INDIAN AGRICULTURAL RESEARCH INSTITUTE, NEW DELHI.


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PRINCIPLES
OF
FARM MACHINERY
THE FERGUSON FOUNDATION
AGRICULTURAL ENGINEERING SERIES

PRINCIPLES OF FARM MACHINERY
Roy Bainer, R. A. Kepner,
and E. L. Barger

SOIL AND WATER CONSERVATION
ENGINEERING
R. K. Frevert, G. O. Schwab,
T. W. Edminster, and K. K. Barnes

AGRICULTURAL PROCESS ENGINEERING
S. M. Henderson and R. L. Perry

TRACTORS AND THEIR POWER UNITS
E. L. Barger, W. M. Carleton,
E. G. McKibben, and Roy Bainer

FARM STRUCTURES
H. J. Barre and L. L. Sammet
In preparing this textbook, the authors have attempted to present the subject of farm machinery from the engineering viewpoint, emphasizing functional requirements and principles of operation for the basic types of field machines. Where feasible, machines for a particular cultural practice (such as planting) have been treated on the basis of the unit operations performed by the functional elements of the machine. Methods for testing or evaluating the performance of certain types of field machinery are included in the appropriate chapters.

*Principles of Farm Machinery* is designed primarily as a textbook for an upper-division course in farm machinery that might be required of all professional agricultural engineering students, regardless of their expected field of specialization. Prerequisites should include a course in static mechanics. Knowledge pertaining to strength of materials and to dynamics would be helpful but is not essential.

In discussing the various machines, only a minimum amount of descriptive material has been included. We have assumed that the reader will be generally familiar with the common types of farm machinery, either from actual experience or from other course work. A student without this background should, from time to time, consult references of a more descriptive nature (trade literature, non-technical textbooks, etc.). The laboratory provides additional opportunity for the student to become familiar with the details of specific machines.

This book represents a summarization and integration of a vast amount of engineering information not heretofore available in one volume. Reference lists at the ends of the chapters indicate the sources for much of the material and provide a handy guide for more detailed study of any particular subject. Such information should be helpful to the practicing agricultural engineer as well as to others in the farm machinery industry.

The subject matter deals primarily with the more common types of field machines but also includes general discussions of materials,
power transmission, economics, and hydraulic controls, as applied to farm machinery. The chapter on seed cleaning is included only because of its relation to the separating and cleaning functions in seed-harvesting equipment. There are many examples of specialty equipment and localized special problems that require engineering attention and offer a real challenge to the farm machinery development engineer, but space does not permit their consideration in this book.

It is recognized that there is considerable variation in the type of approach and technical level of treatment for the various subjects presented. Unfortunately, this inconsistency is an indication of the present status of farm machinery engineering. For some types of equipment considerable engineering information and analytical material are available in the literature, but for other types there is little or nothing other than descriptive material. This situation, however, is changing rapidly and will continue to improve in the future.

Lack of standardization of nomenclature is one of the difficulties confronting a person who writes about farm machinery. The authors have given considerable thought to this problem in an attempt to select the most descriptive and logical terms and perhaps contribute in some degree to future standardization of nomenclature. For example, what we describe as a vertical-disk plow is known in the industry by any one of half a dozen names, including the one we selected.

In a group of closely related books such as The Ferguson Foundation Agricultural Engineering Series, some overlapping of subject matter is unavoidable and perhaps even desirable. For example, the subject of hydraulic controls is discussed in our book as well as in the tractor book of this series. This duplication is felt by the authors to be justified because of the increasing importance of the subject and because of the direct relation of hydraulic controls to farm machinery and its design.

The authors wish to express their appreciation to the many individuals and organizations whose material we have freely used in preparing this manuscript. Farm machinery manufacturers and others have been most cooperative in supplying illustrative material. We are particularly indebted to The Ferguson Foundation, Detroit, Michigan, for sponsoring this work, and to Mr. Harold Pinches of the Foundation for his encouragement and assistance.
Credit is given to the University of California for providing the services and facilities so necessary in such an undertaking.

The preliminary, offset-printed edition was reviewed by many of the leading agricultural engineers, both in industry and in state colleges and universities. Specific suggestions received from approximately fifty of these reviewers, over half of whom were men in industry, represent an important contribution to the accuracy and completeness of the book. We are sincerely grateful for the help received from these men. Our special thanks go to Messrs. N. B. Akesson, A. W. Clyde, and F. W. Duffee for assistance with the subject matter in their respective fields of specialization. Appreciation is expressed to Mrs. Hazel Porter for her cooperation and patience in typing the manuscript, to Messrs. Maurice Johnson and Gerald Lambert who prepared most of the line drawings for illustrations, and to all others who assisted in any way.

Roy Bainer
R. A. Kepner
E. L. Barger

University of California
Davis, California
October, 1955
## Abbreviations

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<td>xi</td>
<td>Various</td>
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### Abbreviations

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<tr>
<td>AISI</td>
<td>American Iron and Steel Institute</td>
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<td>ASAE</td>
<td>American Society of Agricultural Engineers</td>
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<td>ASME</td>
<td>American Society of Mechanical Engineers</td>
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<tr>
<td>bhp</td>
<td>belt horsepower</td>
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<td>Btu</td>
<td>British thermal unit(s)</td>
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<tr>
<td>bu</td>
<td>bushel(s)</td>
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<td>cfm</td>
<td>cubic feet per minute</td>
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<td>cos</td>
<td>cosine</td>
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<td>cosh</td>
<td>hyperbolic cosine</td>
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<tr>
<td>cu ft</td>
<td>cubic foot (feet)</td>
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<td>cu in.</td>
<td>cubic inch(es)</td>
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<td>cu yd</td>
<td>cubic yard(s)</td>
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<td>dbhp</td>
<td>drawbar horsepower</td>
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<td>deg</td>
<td>degree(s)</td>
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<td>diam</td>
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<tr>
<td>fpm</td>
<td>feet per minute</td>
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<td>ft</td>
<td>foot (feet)</td>
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<tr>
<td>ft-lb</td>
<td>foot-pound(s)</td>
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<td>gallon(s)</td>
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<td>gram(s)</td>
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<tr>
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<td>lb</td>
<td>pound(s)</td>
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<td>lb-in.</td>
<td>pound-inch(es)</td>
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<tr>
<td>LP</td>
<td>liquefied petroleum</td>
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<tr>
<td>max</td>
<td>maximum</td>
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<tr>
<td>min</td>
<td>minute; minimum</td>
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<tr>
<td>mph</td>
<td>miles per hour</td>
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<tr>
<td>NIAE</td>
<td>National Institute of Agricultural Engineering</td>
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<tr>
<td>OC</td>
<td>center-to-center distance</td>
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<tr>
<td>OD</td>
<td>outside diameter</td>
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<tr>
<td>oz</td>
<td>ounce(s)</td>
</tr>
<tr>
<td>psi</td>
<td>pounds per square inch</td>
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<tr>
<td>pto</td>
<td>power take-off</td>
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<tr>
<td>Rc</td>
<td>hardness number on Rockwell C scale</td>
</tr>
<tr>
<td>rpm</td>
<td>revolutions per minute</td>
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<tr>
<td>SAE</td>
<td>Society of Automotive Engineers</td>
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<tr>
<td>sec</td>
<td>second(s)</td>
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<td>sin</td>
<td>sine</td>
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<td>square foot (feet)</td>
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<td>sq in.</td>
<td>square inch(es)</td>
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<tr>
<td>tan</td>
<td>tangent</td>
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<tr>
<td>TW</td>
<td>top width of sheave groove</td>
</tr>
<tr>
<td>USDA</td>
<td>United States Department of Agriculture</td>
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<td>yr</td>
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CHAPTER 1

Research and Development in Farm Machinery

1.1. Introduction. The application of machines to agricultural production has been one of the outstanding developments in American agriculture during the past century. The results are to be seen in many aspects of American life. The burden and drudgery of farm work has been reduced, and the output per worker has been greatly increased. Farm mechanization has released millions of agricultural workers to other industries, thus contributing to America's remarkable industrial expansion and to the high standard of living that now prevails in this country.

Our constantly expanding population has required and will continue to demand an ever-increasing agricultural production of foods and fibers. Some of the increased production that has been realized during the past century must be credited to advances in nonengineering phases of agricultural technology such as better crop varieties, the more effective use of fertilizers, and improved cultural practices. A major factor, however, has been the increased utilization of nonhuman energy and of more effective machines and implements. 10

1.2. Objectives of Mechanization. Labor saving has been the principal motivating force in agricultural mechanization. 24 Whereas a hundred years ago about two-thirds of the entire labor force in the United States was on farms, 5 less than one laborer out of six is now engaged in agriculture. 7, 16 Wheat, for example, has been completely mechanized and can now be produced with about 3 man-hr of labor per acre 24 as compared with 60 to 65 man-hr * in 1830 (20 bu per acre). In a more recent example,

Fig. 1.1 illustrates the progressive reduction of labor requirements for cotton as a result of mechanization.

Even today the production of rice in Japan, where hand methods prevail, requires about 900 man-hr of labor per acre * as compared with an estimated 7 1/2 man-hr per acre 30 under the highly mechanized system used in California (yields are comparable).

The average farm unit today represents a sizable capital investment, which calls for increased emphasis on management. Farm mechanization tends to encourage better management by providing more free time for planning and study. Other important contributions and objectives of farm mechanization are greater timeliness for critical operations, the performance of jobs that would otherwise be difficult or impossible by hand methods, and the general improvement of working conditions. Many field operations must be performed within rather short periods of time if optimum results are to be obtained. With high-capacity

* Based upon a study made in Japan in 1948 by Roy Bainer.
mechanized units, operated 24 hr a day if necessary during these critical periods, such operations can be completed in a minimum of time.

1.3. Progress in Mechanization. Although the comparison between present-day methods and those of a century ago is certainly striking, it is of special interest to note the tremendous amount of progress that has been made within the past generation. Among the farm machinery developments that have made their commercial appearance or have become generally accepted since about 1940 are the following: hydraulic controls on trailed implements, automatic draft control for mounted implements, precision planters, processing of seeds for improved planter performance, automatic-tying field balers, mechanical cotton pickers, sugar beet harvesters, improved side-delivery rakes, self-propelled corn pickers, low-volume, low-pressure field sprayers, equipment for applying fertilizer (anhydrous ammonia) in the vapor form, the use of flame for selective weeding of growing crops, the widespread adoption of the airplane as an agricultural tool, and a general increase in the development and application of mounted and self-propelled machines.

One may logically ask why the evolution of farm machinery attained its maximum rate in the United States during the past century. McKibben suggests that it is "the result of a combination of favorable circumstances, a combination unique in the world's history and one which probably will not appear again." He lists twenty-six elements of this combination, including such factors as a stable and equitable government, our system of free enterprise, a rapidly increasing population occupying new lands, a surplus of clear, level land well suited to mechanization, a shortage or infrequent surplus of agricultural labor, a rapidly expanding and effective industrial development, and a remarkable development of transportation facilities. The development of efficient tractors to replace horses and mules as farm power units has greatly broadened the horizon for mechanical agriculture. Without the tractor, much of the progress in fitting the machine to the land and to the crop could not have taken place.

Historically, advances in farm mechanization have been made where a strong demand for labor in other industries has withdrawn workers from the land and forced wage rates up. Severe
labor shortages and high wage rates during World War I and World War II, together with the simultaneous demands for increased agricultural production, have had a marked influence on the mechanization of certain operations in the United States. Before World War II, for example, the mechanization of sugar beet harvesting was held back by the reluctance of processors to accept the product recovered with mechanical harvesters. But the labor shortage resulting from the war forced the industry to modify its standards and accept mechanically harvested beets in order to prevent a drastic reduction in acreage. Since then, mechanization of this operation has progressed to the point where a large percentage of the United States sugar beet acreage is now harvested mechanically.

Outstanding advances in the biological sciences of agriculture have aided mechanization. Plant breeders have produced varieties better suited to mechanical harvesting. Dwarf varieties of grain sorghums having uniform growth, shatterproof small grains, stormproof cotton, and hybrid corn varieties with less tendency to lodge are examples. Planting practices for some crops have been changed in order to modify the growth habits and obtain plants better suited to mechanical harvesting.

1.4. The Future of Mechanization. History indicates that the process of mechanization is dynamic, with no ultimate goal in sight. Under our system of competitive free enterprise, each manufacturer must continually improve his products and develop new ones in order to grow and maintain a profitable position. Many of our present harvesting machines are inefficient or reduce quality. Mechanically picked cotton, for example, has averaged about one grade lower than hand-picked cotton (see Chapter 19). Functional improvements to reduce the tendency of some types of harvesting equipment to clog under difficult conditions would be welcome. Mower cutter bars should be adapted to higher speeds and made more durable and more effective.

Further simplification and standardization of the attachment methods for tractor-mounted equipment would be helpful to the farmer. Corrosion-resistant tillage tools that will still scour satisfactorily, less frequent lubrication for important farm machines, tillage tools designed for higher speeds and increased tractor power, greater simplification of many machines, more
versatility in tools, increased safety and comfort for the operator, and possible reductions in size, weight, and power requirements are other items that need and undoubtedly will receive engineering attention in the future.

A USDA report published in 1947 indicates that, in spite of the tremendous advancements made during the last century, approximately 47 per cent of the 10.4 billion man-hr used in the production of crops in 1944 represented labor performed with the hands or with small hand tools. Considering the combined requirements for crops, livestock, and farm maintenance, the total in 1944 was 21.2 billion man-hr, of which 61 per cent represented hand labor. Although present-day figures are undoubtedly considerably lower, there is still much to be done in further mechanization.

Fruits and nuts, vegetables, and tobacco are among the worst offenders in regard to the amount of hand labor required. Crops such as these are difficult to mechanize because of the judgment and care generally required in operations such as picking, pruning, and thinning. According to Walker, for example, the production of a 10-ton-per-acre crop of cling peaches requires about
300 man-hr of hand labor, with more than two-thirds of this total being used in harvesting operations. Mechanical aids to increase the effectiveness of hand labor, such as mechanical pruning shears and mechanical tree shakers for nuts, offer a means of obtaining partial relief but are not the complete answer.

The economic adaptation of certain types of machines to small farms is a problem that needs more attention before our agriculture can become completely mechanized. On many farms the acreage planted to a particular crop is too small to justify the ownership of expensive equipment. Joint ownership and contract or custom operations are alternatives to individual farmer ownership, but they have their limitations and disadvantages from the farmer's standpoint.

It has been estimated that a 40 per cent increase in agricultural production will be needed by 1975 (as compared with 1950) to supply the needs of our expanding population. Engineers must develop new and improved machines to help the American farmer meet these production goals in the face of a steady decline in the supply of farm labor.

1.5. Characteristics of Farm Machinery Engineering. Agricultural engineering implies the application of engineering knowledge and techniques to agriculture. The successful agricultural engineer must recognize that there are basic differences between agriculture and other industries. The biological factor is an important consideration in engineering applications, and the engineer needs to be familiar with the basic principles and practices of agriculture. Changes in cultural practices are often needed to make a machine adaptable or to increase its effectiveness. Processing equipment and standards may need to be revised to accommodate mechanically harvested crops. The quality or yield of a crop is sometimes reduced by the use of a machine, the resulting monetary losses being chargeable against the machine.

The agricultural industry is basically decentralized or dispersed, with more than 5 million operating units in the United States,* and is greatly influenced by climatic conditions. Power must be taken to the work, rather than bringing the work to a

*USDA Agricultural Statistics (1952) indicates a total of 5,382,162 farms in 1950.
centralized power plant. Most field operations are seasonal in nature, often with only a short period of time available in which to perform the task. Consequently, field machinery in many cases has a low annual duty (i.e., very few hours of operation per year).

It has been said that the field of farm machinery design presents a greater challenge to an engineer's ability than any other field of engineering endeavor. Farm machines must perform satisfactorily over wide ranges in a considerable number of variables. They may be operated where the temperature is well above 100°F or where it is below freezing and are subjected to rain, snow, and sleet. Instead of resting on a solid factory floor or moving over a smooth road, they must operate over uneven terrain through dust, sand, mud, and stones. They must be designed to handle wide variations in crop and soil conditions. Operators often are relatively unskilled, partly because of the limited usage of the machines.

In addition to the difficult environmental conditions, farm machines are subjected to rather stringent economic limitations. Manufacturing costs must be kept to an absolute minimum so that the limited amount of operation will not put the cost per hour into a prohibitive range. Therefore, farm machinery designs must be as simple as possible, must utilize the lowest-cost materials that are available and satisfactory for the job, and must permit the widest manufacturing tolerances that are consistent with good performance.

1.6. Types of Problems Encountered. The student approaching the subject of farm machinery engineering might well consider the types of problems most frequently encountered in this field, together with the general methods ordinarily employed in pursuing these problems. Although the variety of problems that might be encountered is wide, most of them can be grouped into the following general classification:

2. Improvement of a machine, development of a new model similar to existing machines, or design changes to reduce the manufacturing cost of a machine.
3. Comparative testing of several machines or evaluation of the performance of a particular machine.
4. Studies relating to the more efficient utilization of existing machines. An example is the determination of proper adjustments and operating conditions for a combine in order to minimize seed damage and losses.

5. Research studies of fundamental problems not specifically related to a particular machine, such as the study of soil dynamics in relation to tillage and traction.

Commercial manufacturing organizations are, of course, concerned primarily with the development and improvement of farm machinery, the ultimate goal being to obtain a product that is useful and acceptable to the farmer and that can be manufactured and sold at a profit. To this end the farm machinery industry is currently undertaking more and more work of a research nature, the results of which can be applied to some particular class or group of machines. Fundamental work of this type is often done cooperatively by several manufacturing concerns, the organization of the project in many cases being under the auspices of the American Society of Agricultural Engineers (ASAE).

Public research agencies such as state experiment stations and the United States Department of Agriculture (USDA) may undertake any of the types of problems listed above. Studies relating to the more efficient use of machines, comparative testing of machines, and research studies of a fundamental nature are often undertaken by public research agencies rather than by commercial organizations.

The development or improvement of a machine, when undertaken by a public agency, often involves either a special problem of a localized nature (geographically) or one that is likely to require a great deal of development work and close coordination with research workers in other agricultural fields (such as plant breeders, botanists, entomologists, etc.). Emphasis in such a program is generally placed upon fundamental principles and the functional elements required, with the final design of a production model being left to the manufacturer or worked out cooperatively.

Whether the engineer is conducting research, developing a machine, or dealing with any of the other types of problems that may be encountered, he should adopt a positive approach and have faith and determination that the problem will be solved. The need for clear thinking and keen observation throughout the
job cannot be overemphasized. Open-mindedness and tolerance are also important.

**RESEARCH**

Ellison has defined research as "study that is planned to gain basic information not available in the form (in which) it is desired." Research results are fundamental and are comparable with results secured by the same methods at other times. Development work, testing, surveys, investigations, and experimentation may all be parts of a research program but do not in themselves constitute a complete research cycle.

1.7. Research Procedure. A research procedure is primarily an orderly process for analyzing, planning, and conducting an inquiry. The first and probably the most important single consideration is the recognition of and selection of a problem needing solution; otherwise the research may be meaningless. The second step is the accumulation of existing information related to the problem, together with a careful and objective analysis of these data and of the factors (variable or constant) involved in the problem. The variables must then be associated in such a way as to establish their functional interdependence. A good basic analysis should uncover every fundamental fact and relationship without prejudice or favor.

The formation of a hypothesis and the organization of experiments based upon this hypothesis is next in order. Often a considerable amount of time must be spent on experimental methods before a direct attack can be made on the main objective. Methods developed in other fields of activity, such as the systems of plot replication developed by years of agronomic experience, should not be overlooked. If the preceding steps have been performed carefully and objectively, the execution of the selected procedure and subsequent analysis of the results should lead to the ultimate research objectives.

Adequate records (both written and photographic) of experimental conditions and results are important. The findings should be compared with and checked against those of similar research performed by other agencies, and the data should be analyzed so as to give the maximum amount of useful information, particularly in regard to general principles. Prompt and effective
publishing of the results constitutes an important part of the research procedure.

1.8. Replications and Statistical Analysis. Properly conducted research should yield data that are valid and are consist-

![Graphs showing effect of number of replications on draft results for plow disks at three speeds](image)

**Fig. 1.3.** Effect of number of replications upon the comparative draft results for three plow disks at three speeds. (Eugene G. McKibben and Milton O. Berry. *Agr. Eng.*, December, 1952.)

ent with the results from other similar research. If the alleged findings are not real, the research is worthless. When an agricultural engineering research problem involves the soil or some biological phenomenon, as it frequently does, it is extremely difficult to control the variables so that only one of them varies during a particular group of tests. Under such conditions, statistical methods of analysis are important and several replications are needed to permit a satisfactory determination of the reliability of the results.
It is a common failing in planning a research project to include too many treatments and too few replications. McKibben and Berry illustrate the value of replications by means of draft data obtained for three disk-plow blades at three speeds. In the upper left-hand graph of Fig. 1.3, note the variability of the results indicated by each of the four series of runs when considered individually. The second graph shows very little improvement when the data are plotted as means of four combinations of duplicate tests.

However, when the same data are plotted as four combinations of three replications (lower left-hand graph), the merit of the increased number of replications becomes evident, there being considerable similarity between the four sets of curves. Combining the data into a single series of curves representing the means of four replications gives results that appear to be reasonable (in the light of other similar tests of tillage tools); and statistical analysis indicates a "significant" difference between the upper and lower curves. Additional replications might have improved the results still further.

DEVELOPMENT PROCEDURES

The word "development" implies the gradual advancement of a plan toward a specific goal, and describes the over-all process by which most farm machines evolve. Hard work, objective thinking and planning, and numerous disappointments are characteristics of a typical machinery development program.

Early farm machinery developments were often crude and haphazard, with the cut-and-try system predominating. Present-day farm machinery design, however, is rapidly becoming more scientific, with the development of a machine being based increasingly upon fundamental principles and information obtained by research methods. Present-day development programs, whether in industry or in public agencies, tend to follow to a considerable extent the research pattern described in the preceding section.

Throughout the history of agricultural mechanization the farmer has played an important role in the development of equipment to meet his needs. The ideas for many of our present-day farm machines originated on the farm, and in many cases the
first models were built by farmers or under their supervision. A surprising number of ingenious, special-purpose machines are designed and built by farmers, either because commercially available machines are not suitable under their particular operating conditions or because no machines are available. During early field tests of a new commercial model, the assistance and cooperation of farmers and operators are invaluable assets to the manufacturer, sometimes spelling the difference between success and failure for the machine.

1.9. Introduction and Evaluation of the Problem. In public research agencies, farm machinery development projects (as well as other types of problems) are frequently initiated at the request of some influential outside group or organization whose members will be directly benefited by the results, or at the suggestion of personnel within the research agency who recognize the importance of the problem. Cooperative projects between agricultural engineers and various agricultural science groups are common in experiment-station work. In any case a proposed project should first be studied and evaluated in regard to the potential application of the results and its value to the farmer

Fig. 14. A farmer-built machine that simultaneously forms four planting beds, applies fertilizer, prepares the seedbed with rotary tillers followed by rollers, and plants eight rows of sugar beets, in an over-all precision operation. (Photo by Austin Armer.)
in terms of labor saving, increased crop returns, improved quality, increased income, etc.

Development projects in commercial organizations commonly originate as the result of favorable market research or because the judgment of experienced engineers or other administrators in responsible positions indicates their desirability. Some projects originate because of economic considerations, even though the existing device is entirely satisfactory from both the functional and mechanical standpoints. Problems involving the reduction of costs or the substitution of more readily available materials are examples of the latter type.

Evaluation of a proposed project in industry includes consideration of such factors as the estimated potential use or market for a new machine and the feasibility of such a device from both the engineering and economic standpoints. If the proposed machine is of a radically new type, it is difficult to get a reliable reaction from the public as to its probable acceptability or potential application, since the farmer has nothing concrete upon which to base his opinion. New models of existing machines are usually acceptable if they reduce over-all costs or improve performance.

1.10. Determination of Functional Requirements and Fundamental Relations. In either industry or a public research agency, the first step in the development of a new machine is to establish a set of functional requirements or specifications. In other words, what must the machine do? Under what conditions is it expected to operate satisfactorily? The answer to the first question includes consideration of such factors as the optimum distribution and placing of seeds by a planter, the desired effect of a tillage implement upon the soil, or the required action and permissible tolerances of a harvesting machine in regard to recovery and quality of product. General experience or field investigations and surveys may be involved in answering the second question.

The assistance and recommendations of other technical groups (such as plant scientists, entomologists, botanists, and crop processors) are frequently needed in establishing the functional requirements. Oftentimes a compromise must be made between conflicting requirements or between ideal requirements and those attainable with a practical machine.
Closely following this first step, and often directly associated with it, is a critical review and evaluation of existing information (or machines) and past experiences related to the problem. Fundamental relationships that might be of value in solving the problem should be determined, either by field investigations or by laboratory research. What characteristics of the plant, for example, can be utilized to advantage in accomplishing the desired end result? What limitations are imposed by plant or soil characteristics?

1.11. Design and Development of Experimental Machine. If the device under consideration is to be a competitive model or an improved version similar to machines already on the market, the designer can study and test existing units, determine their shortcomings, and then set about to design an improved model. When a radically new machine is being designed, however, the problem is more difficult and requires greater imagination and ingenuity in addition to good basic engineering ability.

The first experimental designs are primarily functional and generally deal with machine elements rather than a complete machine, the chief objective being to test and develop (or discard) certain ideas or principles of operation. Although durability and the refinement of mechanical details are not important in these early models (except to the extent necessary to permit adequate functional testing), the mechanical and economical practicability of the ideas should be given increasing consideration as the development progresses. The ultimate objective is, of course, to be able to perform the specified functions satisfactorily with as simple and efficient a unit as possible.

1.12. Design of a Good Mechanical Unit. If the results from the development of the experimental unit are favorable and indicate that the machine has economic possibilities, a mechanically sound unit, suitable for commercial production, is designed. Although this operation is generally performed by the engineering department of a manufacturing organization, it may be based upon the experimental results obtained by a public research agency as well as upon preceding development work done within the organization.

The design of a production model includes consideration of forces involved, time and motion factors, inertia of moving parts and the severity of accelerating or decelerating forces, weight,
balance, vibration and fatigue problems, and other similar fac-
tors. Close coordination is maintained between representatives
of the engineering, production, and sales departments regarding
materials, fabrication procedures, and other factors contributing
to the most economical manufacture of a machine that will per-
form properly and give the required life with a minimum of
mechanical failures.

Analytical design procedures are being employed to an increas-
ing extent, particularly for new types of machines. Many struc-
tural members, however, are designed by proportion (based upon
the experience of the designer) or by comparison with other simi-
lar applications, because of the saving in time and the difficulty
in predicting the extreme loads likely to be encountered.

A good design from the sales standpoint takes into considera-
tion customer appeal and anticipated consumer prejudices. Sim-
plicity, ease of operation, comfort and safety for the operator,
attractive styling, and a general appearance that suggests capac-
ility and ruggedness are factors that contribute to customer and
operator appeal.4

1.13. Construction of Pilot Run of Machines. If the ma-
chine represents a new development, it is customary to build a
limited number of preproduction or pilot units. They may un-
dergo tests and modifications for several years before the design
is placed in production. The pilot machines are generally built
by experimental-shop methods but follow the proposed produc-
tion design rather closely. These machines are operated by farm-
cers in various areas, preferably under a wide range of conditions,
and their performance is checked periodically by field engineers
and design engineers. Careful consideration should be given to
the suggestions and opinions of the cooperating farmers and
operators.

1.14. Manufacture of Production Model. Although this
might be considered as the final step in the evolution of a ma-
chine, engineering problems still continue to manifest themselves
in such forms as desirable improvements, obtaining satisfactory
life, the use of new and improved materials and fabrication
methods to reduce costs, and expanded applications of the ma-
chine.
1.15. Types of Tests. Farm machinery testing includes (a) functional tests, (b) mechanical tests for structural strength and durability, (c) determination of power requirements, and (d) determination of the external forces acting upon a machine (such as a tillage implement) and accumulation of other similar information that might serve as a basis for design.

The nature of functional tests varies widely, depending upon the type of machine or implement under consideration. The functional requirements and some of the test methods are discussed in subsequent chapters that deal with specific types of machines.

Although the main emphasis in the development of experimental units is placed upon functional testing, determinations of power requirements and external forces may be included as a basis for subsequent design steps. Product design engineers in industry are concerned primarily with functional and mechanical tests but have a secondary interest in power requirements and information that might be utilized in future design work.

1.16. Planning of Tests. As with research, proper planning and careful execution of tests are of extreme importance. In planning tests for a proposed production model, the test engineers should study the structural, mechanical, and functional requirements and then plan suitable tests to determine whether the device will meet the design requirements. The unit should be tested either under simulated operating conditions (if adequately known) or under actual field conditions. The tests should involve the type of use and treatment to be expected after the unit is commercialized and should be conducted in such a manner as to provide a margin of safety for normal variations in materials and processing in the manufacture of the machine.12

1.17. Laboratory or Shop Tests. The laboratory is playing an ever-increasing part in the development and improvement of farm machinery. A great deal of valuable information regarding the mechanical strength and durability of component parts can be obtained by ingenuity in applying appropriate test methods and techniques. Such tests often require the development and construction of specialized testing equipment (Fig. 1.5).
Stresses in structural members at critical sections can be determined with strain gages or other appropriate devices. Structural weaknesses can be found by means of exaggerated loadings and accelerated life tests of component parts or of the entire machine (as on a rough test track). Repetitive tests under simulated field conditions can yield definite results in far less time than would be required for actual field tests. Experience and careful observation are invaluable in relating the results of such tests to the results expected in actual field operation.

Since field tests of most types of farm equipment are seasonal and the available time per season is often short, it is important that laboratory tests be utilized to the fullest extent before the beginning of the field-testing period. With some types of machines (either experimental or production models), preliminary functional tests can be performed in the laboratory. Where this is possible, conditions can generally be controlled more carefully and results can be measured more readily than in the field. For example, material can be fed into a combine to obtain an indication of its capacity and the relative balance and performance characteristics of the various components. Determination of the uniformity of seed distribution from a planter is an example of a functional test that can be performed more readily and more reliably in the laboratory than in the field. High-speed photography is particularly advantageous in the laboratory for studies...
of either the functional or the mechanical behavior of certain ma-
chine elements.

1.18. Field Tests. The final proof of performance and dura-
bility can be obtained only through field tests. In the early
stages of development, tests of an experimental machine may be
confined to a limited number of field conditions in order to reduce
the number of variables. But as the development progresses, the
scope of the tests should be expanded to include a wide range
of expected operating conditions. Pilot machines of a produc-
tion design should be operated in as many areas as possible, under
varying crop, soil, and weather conditions and with typical or
below-average operators.

1.19. Use of Strain Gages for Stress Determination. Elec-
trical-resistance strain gages are being utilized increasingly in
connection with the development and testing of farm machinery.
Although the indicating or recording equipment may be rather
complex and expensive, particularly for high-frequency measure-
ments, the gage elements themselves have a negligible mass in
most applications, are easily applied, and are rather versatile
tools.

Basically, a resistance strain gage consists of a grid or series
of continuous loops of very fine wire bonded to a paper base or
embedded in a thin bakelite sheet (Fig. 1.6a). Gage lengths
(length of loops) range from $\frac{3}{16}$ in. up to several inches. The
gage is cemented firmly to the specimen or member to be tested
and for all practical purposes becomes an integral part of it.
When the outer fibers of the member are strained (deformed) in
tension, the wire is elongated and its resistance increases in direct
proportion to the amount of strain or elongation. Similarly,
compressive strains shorten the gage and decrease its resistance.

An electric current is passed through the wire, and the changes
in resistance caused by strain in the test member are measured
by appropriate instruments. Because the electrical resistance of
a strain gage is affected by temperature as well as by strain, two
or sometimes four gages are generally connected in a Wheatstone
bridge circuit in such a manner that temperature compensation
is obtained. Inactive gages used only for temperature compensa-
tion are mounted on blocks of material having the same properties
and the same environment as the test member.

Many structural members on implement frames are subjected
to complex and unknown loadings or are statically indeterminate to such an extent that mathematical or graphical solutions for load distribution are either impossible or involve broad assumptions. With strain gages, structural weaknesses in such members can be discovered without the necessity of waiting for time-consuming and costly field experience to expose them. Overdesigned parts are also revealed, thus permitting a more economical design in many cases.

Fig. 1.6. (a) Strain-gage element, (b) gage mounted for determination of stress in the outer fibers of a structural member, (c) ring-type dynamometer, (d) orientation of gages on a shaft or tube for measuring torque.

In determining maximum stresses, proper location and orientation of the gages on the test member are important. Several gages in different positions may be needed on a single member if the location of the maximum stress is not obvious. Special strain-sensitive lacquers can be applied to the part in a preliminary step to study stress patterns and determine the proper location for the strain gages. The brittle lacquer coating on the test member, when strained beyond a predetermined value, forms cracks at right angles to the direction of the principal tensile strains.

1.20. Use of Strain Gages on Load-Measuring Devices. The principle of load determination with strain gages is to introduce the load into some member whose resulting deformation can be measured with one or more strain gages. Except for the simplest arrangements, calibration of the load-measuring device is usually required. Strain gages can be utilized for measuring
drawbar pull, either by the simple expedient of applying gages directly to the drawbar or other tensile member or by means of a special device with greater sensitivity such as the ring-type dynamometer described by Hawkins and Rogers and illustrated in Fig. 1.6c. The four gages in the ring are connected in a bridge circuit, gages 1 and 2 being in tension and the other two being in compression (for a tensile load).

![Fig. 1.7. Determining power requirements for component parts of a combine during field tests. The truck contains portable recording equipment for the strain gages mounted on the various shafts to indicate torque. (J. I. Case Co.)](image)

Dynamic torque can readily be measured by applying strain gages to a section of shaft or tubing. Basically, a strain-gage torque meter consists of a short piece of shaft or tubing with four strain gages mounted on it as indicated in Fig. 1.6d and connected in a Wheatstone bridge circuit in such a manner that maximum response is obtained from the torsional strain while the effects of bending, end thrust, and temperature are cancelled. The lines from the four corners of the bridge circuit are brought out from the rotating shaft through brushes and slip rings or by means of some other suitable arrangement. The shaft cross-section under the strain gages is often reduced by boring out (or else tubing is used), to increase the torsional deflection and sensitivity of the device.
Burrough found that by using a specially designed mercury-bath collector instead of brushes and slip rings, variations in contact resistance were minimized to the extent that torques of the magnitude normally encountered in the drive shafts of component parts of a farm machine could be measured with reasonable accuracy by applying strain gages directly to the shaft. The mercury-bath collector is mounted at the end of the shaft to be studied and is driven from it by means of a flexible connection that permits some misalignment. If there is a shaft bearing between the gages and the collector, the shaft must be center-drilled to provide a passageway for the wires. Otherwise the wires are merely taped to the outside of the shaft. According to Burrough, a clearance of about $\frac{1}{8}$ in. radially and 1 in. along the shaft is sufficient for mounting the gages. With an arrangement of this type it is a relatively simple matter to measure the power requirements for various components of a machine such as a combine or a baler.

Because strain-gage torque-measuring devices have very little inertia, they are especially valuable for measuring high-frequency fluctuations and peak torques due either to normal loading or to transient conditions such as those resulting from the sudden engagement of a clutch or from starting under a heavy load (see Section 5.21). Such measurements, however, require rather expensive amplifying and recording equipment.

1.21. High-Speed Photography. High-speed motion-picture photography as an engineering tool is relatively new in the field of agricultural machinery. Although the equipment is expensive and there are some serious engineering problems involved in its use, the results obtained in some types of studies far outweigh the problems and costs. The two principal applications are in studying the mechanical behavior of rapidly moving parts and in studying the behavior of crop materials passing through a machine at high speed. The action of a forage-chopper cutterhead and impeller is a good example of the latter type of application.

High-speed motion-picture cameras are available with top speeds of more than 5000 pictures per second. When film that has been exposed at 5000 frames per second is viewed at the normal projection speed of 16 frames per second, the resulting time magnification is more than 300 to 1. At least one camera manufacturer recommends that for best picture quality the image motion during the exposure time for a single frame should not
exceed about 0.002 in. With this particular camera (16 mm) the exposure time is one-fifth of the total time per frame; the recommended minimum speed for a forage blower operating at a peripheral speed of 6000 fpm would be 3000 frames per second if the field of view is 16 in. wide and 2000 frames per second if it is 24 in. wide (minimum speed is inversely proportional to the linear size of the field of view).

The principal problems encountered in high-speed photography have to do with lighting and with timing. Because of the extremely short exposure times obtained in high-speed filming, a high level of illumination is required. Special lamps are available for high-speed work. Power requirements for illumination alone may be in the order of 10 to 20 kw, depending upon the size of the field of view.

At a speed of 5000 frames per second, a full 100-ft roll of film will pass through the camera in 1 to 1 1/2 sec. About one-third of the film and perhaps one-half of this total time have passed by the time the camera gets up to its normal operating speed. If the event to be filmed occurs for only a short time (say 1/2 sec), the operation of the camera must be synchronized to the event so that the event is photographed on the last two-thirds of the 100-ft roll of film. If the action is continuous there is no problem except turning off the camera at the end of the run to prevent the film from whipping to pieces.

REFERENCES

CHAPTER 2

Field Capabilities
and Cost Analysis

2.1. Introduction. Although the first requirement of a machine is that it be able to satisfactorily perform its intended function, the management and economic aspects of the machine application are also of great importance. In fact, the engineer soon finds that his approach to a farm machinery design problem is largely controlled by economic considerations. To work most effectively, he should have a thorough understanding of the factors affecting field capacities and of the economic principles governing the costs of owning and operating field machines.

2.2. Types of Implements. In this chapter and throughout the entire text, reference is made to four general types of field implements, based upon their relation to the power unit. Some of these types are commonly described by any one of several terms. The following names were selected as being most descriptive and most consistent within the classification.

A **trailed** implement is one that is pulled and guided from a single hitch point and is never completely supported by the power unit.

A **semimounted** implement is one that is attached to the tractor along a hinge axis (rather than at a single hitch point) so that it responds directly to tractor steering but is never completely supported by the tractor.

A **mounted** or **tractor-mounted** implement is one that is attached to the tractor in such a manner that it is steered directly by the tractor and, at least for the raised position of the implement, is completely supported by the tractor.

A **self-propelled** machine is one in which the propelling power unit is an integral part of the implement.
FACTORS AFFECTING FIELD CAPACITY

2.3. Terms Related to Field Performance of Machines.
The rate at which a machine can cover a field while performing its intended function is one of the considerations in determining the cost per acre for the operation.

The theoretical field capacity of an implement is the rate of field coverage that would be obtained if the machine were performing its function 100 per cent of the time at the rated forward speed and always covered 100 per cent of its rated width.

The effective field capacity is the actual average rate of coverage by the machine, based upon the total field time as defined in Section 2.4. Effective field capacity is usually expressed as acres per hour.

Field efficiency is the ratio of effective field capacity to theoretical field capacity, expressed as per cent. It includes the effects of time lost in the field and of failure to utilize the full width of the machine.

Performance efficiency is a measure of the functional effectiveness of a machine, as for example, the per cent recovery of usable product by a harvesting machine.

2.4. Effective Field Capacity. The effective field capacity of a machine is a function of the rated width of the machine, the percentage of rated width actually utilized, the speed of travel, and the amount of field time lost during the operation. With implements such as harrows, field cultivators, mowers, and combines, it would be practically impossible to utilize the full width of the machine without occasional skips. The required amount of overlap is largely a function of speed, ground condition, and the skill of the operator. In some cases the yield of a crop may be so great that a harvesting machine cannot handle the full width of cut, even at the minimum forward speed obtainable.

The rated width of implements with spaced functional units, such as row-crop planters or cultivators, grain drills, and field cultivators is the product of the number of units (rows, grain-drill furrow openers, or cultivator standards, for example) times the spacing of the units. In other words, the rated width is assumed to include one-half space beyond each outside unit. Row-crop machines utilize 100 per cent of their rated width, whereas open-
field implements with spaced functional units are subject to losses from overlapping.

The maximum permissible forward speed is related to such factors as the nature of the operation, the condition of the field, and the amount of power available. With harvesting equipment the limiting factor may be the maximum rate at which the machine can effectively handle the crop.

Lost time is the most difficult variable to evaluate in relation to field capacity. Field time may be lost as a result of adjusting or lubricating the machine, breakdowns, clogging, turning at the ends, adding seed or fertilizer, unloading harvested products, etc. In relation to effective field capacity and field efficiency, as defined and discussed in this chapter, lost time does not include time for setting up or daily servicing of the equipment or time lost due to major breakdowns. It does include time for minor repairs in the field and for any lubrication required in addition to the daily servicing, as well as the other items mentioned above. The total field time is considered to be the sum of the effective operating time (time during which the machine is actually performing its function) plus the lost time. Referring to equation 2.3, effective operating time = \( T_0 \times \frac{100}{K} \).

Time spent in traveling to and from the field is usually included in figuring the over-all cost of an operation but is not considered in determining effective field capacity or field efficiency because the machine should not be penalized for the geographical location of the enterprise.

The effective field capacity of a machine may be expressed as follows:

\[
C = \frac{5280 \times S \times W \times E_f}{43,560 \times 100} = \frac{SWE_f}{825}
\]  

where

- \( C \) = effective field capacity, in acres per hour.
- \( S \) = speed of travel, in miles per hour.
- \( W \) = rated width of implement, in feet.
- \( E_f \) = field efficiency, in per cent.

2.5. Effect of Field Dimensions. Turns at the ends or corners of a field represent a loss of time that is often of considerable importance, especially for short fields. Regardless of whether a field is worked back and forth, laid out in lands, or worked by
travelling around the perimeter, the total number of turns per unit of area with a given width of implement is inversely proportional to the length of the field. For a given rectangular field, worked either in the long direction or around the field, the required total number of round trips (exclusive of opening or finishing lands as in plowing) would be the same for any of the three methods. Working back and forth requires two 180° turns per round, whereas either of the other methods involves four 90° turns per round trip.

Idle travel across the ends of a field represents another loss that is often unavoidable and is particularly important where wide lands are laid out in short fields. If \( w \) is the total width of each land (i.e., width of each area worked out as a unit), the average theoretical travel distance across each end is \( \frac{1}{2}w \). Then, if the length of the field is \( L \), the average total travel per round is \( 2L + w \) and the percentage of idle travel distance is

\[
I = \frac{w}{2L + w} \times 100
\]

Dividing numerator and denominator by \( w \) gives

\[
I = \frac{100}{(2L/w) + 1}
\]  

(2.2)

In actual practice the maximum travel across the end of a land would be a little greater than \( w \), and the minimum travel as the land is narrowed down would be limited by the turning radius of the machine or tractor. Thus, in computing \( I \), a value of \( w \) somewhat greater than the width of the land should be assumed.

Although the above considerations indicate that having fields long in relation to their width tends to improve field efficiency, other factors may be of more importance in determining the proportions of the field. For example, the length may be limited by irrigation requirements. Also, a square field requires less fence than a rectangular field of the same area, and under certain conditions there is some advantage in being able to work a field both ways.

2.6. Interruptions That Are Proportional to Area. Time lost in turning, idle travel across the ends, and adjustment of the machine usually tends to be proportional to the effective operat-
ing time, within reasonable limits of width or speed of the implement. Other time losses, such as those caused by field obstructions, clogging, adding seed or fertilizer, and filling spray tanks or dust hoppers often tend to be proportional to area rather than to operating time.\(^8\) Time losses due to the unloading of harvested crops tend to be proportional to the yield as well as to area. Interruptions of this type become increasingly important as the width or speed of an implement is increased, because they then account for a greater percentage of the decreased total time per acre.

The relative importance of interruptions proportional to area may be determined from the following equation, which is based on the definition of field efficiency:

\[
E_f = K \frac{T_0}{T_0 + T_h + T_a}
\]

where \(K\) = percentage of implement width actually utilized.

\(T_0\) = theoretical time required for one acre (at theoretical field capacity).

\(T_h\) = time lost per acre due to interruptions that are not proportional to area (\(T_h\) usually tends to be proportional to \(T_0\)).

\(T_a\) = time lost per acre due to interruptions that tend to be proportional to area.

For example, if \(K = 100\) per cent, \(T_h = 0.15T_0\), and \(T_a = 0.2\) hr per acre, the field efficiency would be 74 per cent for a machine with a theoretical capacity of 1 acre per hour (\(T_0 = 1\)), but would be only 51 per cent for a similar machine with a theoretical capacity of 4 acres per hour (\(T_0 = 0.25\)).

2.7. Performance of Series Combinations of Machines. As field operations become more completely mechanized, with an increasing number of operations involving a combination of several machine units, the reliability of individual machines becomes increasingly important. For an individual implement, a 5 or 10 per cent time loss from breakdowns, adjustments, clogging, or other stops is generally not considered serious. But if five such units, each with a reliability of 90 per cent, are used in series so that the operation of all units is stopped whenever one unit stops, the expected over-all reliability of the combination
The reliability, as discussed in this section, is basically the same as field efficiency except that only the effective operating time and the time loss due to stops are considered.

The reliability relation is expressed by the following equation, which is based on well-established laws of probability: 10

\[ y = 100 \frac{(x_1)(x_2)(x_3) \cdots (x_n)}{100^n} \]  

(2.4)

where \( x_1, x_2, x_3 \cdots x_n \) are the expected reliabilities of the individual machines, in per cent, and \( y \) is the expected per cent reliability for the combination of \( n \) machines. If each individual machine has the same reliability, equation 2.4 becomes

\[ y = x^n / 100^{n-1} \]  

(2.5)

It should be noted that the reliabilities indicated by equations 2.4 and 2.5 are only the statistically expected values. For short periods of time, the reliabilities of individual machines or combinations may vary widely from the expected values. If the reliability factor and its variation are determined for each machine by observations during a number of periods, it is then possible to determine statistically both the expected value and its variability for a series combination of these machines. 10

Because of the reduced reliability with a combination of machines, preventive maintenance becomes relatively more important than when only a single machine is used. All machines in a combination should be able to operate the same length of time between servicings, and the capacities of the various units should be reasonably well matched.

2.8. Time Studies Related to Field Efficiencies. In reporting the results of three seasons' studies of the loading of a farm tractor (with a Servis-Recorder mounted on it), Bateman 3 includes data for the "labor efficiency" obtained with various implements. These results do not represent field efficiency as it is defined in Section 2.3. They include time for servicing and traveling to and from the field (which field efficiency, as defined in Section 2.3, does not include) as well as field time lost due to minor repairs, adjustments, and unloading of the harvested crop. Time lost in turning and in idle travel across the ends of the field is not included in Bateman's labor efficiency values, and the
reduction of field capacity due to overlapping is not considered.

Bateman's results do, however, provide a basis for determining the percentage of total field time lost due to minor repairs, adjustments, and crop unloading. In general, losses from these causes ranged from 1 to 10 per cent for diskng, plowing, harrowing, or cultivating corn (two-row) and from 10 to 16 per cent when operating an 8-ft grain drill or a 7-ft mower. With a four-row checkrow corn planter, 35 per cent of the total field time was lost in making minor repairs and adjustments, adding seed, and moving the checkwire.

Euler reports the results of time studies for 6-ft trailed combines on eight Indiana farms and for two-row trailed or mounted corn pickers on twenty farms. The average results for the combines were as follows:

<table>
<thead>
<tr>
<th>Activity</th>
<th>Per Cent of Total Field Time</th>
</tr>
</thead>
<tbody>
<tr>
<td>Unloading of harvested grain</td>
<td>17</td>
</tr>
<tr>
<td>Turning</td>
<td>9</td>
</tr>
<tr>
<td>Stopped or waiting (other than unloading)</td>
<td>6</td>
</tr>
<tr>
<td>Actually harvesting</td>
<td>64</td>
</tr>
</tbody>
</table>

Bateman's results with a 12-ft trailed combine indicate that the time lost as a result of minor repairs, adjustments, and unloading was 23 per cent in wheat and 32 per cent in soybeans.

Euler's results for two-row corn pickers did not reveal any striking differences between trailed and mounted units. The averages for all pickers were as follows:

<table>
<thead>
<tr>
<th>Activity</th>
<th>Per Cent of Total Field Time</th>
</tr>
</thead>
<tbody>
<tr>
<td>Changing wagons</td>
<td>4</td>
</tr>
<tr>
<td>Turning and idle travel across ends</td>
<td>10</td>
</tr>
<tr>
<td>Stopped (other than changing wagons)</td>
<td>22</td>
</tr>
<tr>
<td>Actually harvesting</td>
<td>64</td>
</tr>
</tbody>
</table>

Bateman's results for a two-row trailed corn picker indicate that 28 per cent of the total field time was spent in making minor repairs and adjustments and in stopping to change wagons, as compared with 26 per cent in Euler's studies.
2.9. Studies of Plowing Efficiencies as Related to Field Size and Shape. Time studies involving two methods of plowing, conducted by the British National Institute of Agricultural Engineering (NIAE), provide information regarding plowing times and field efficiencies for different sizes of fields. The tests were made with a three-bottom, 12-in., conventional trailed plow and with a mounted two-way plow of the same size. Two adja-

![Diagram](image)

Fig. 2.1. Relation of field efficiency to size of square field, for two methods of plowing. Time for servicing the equipment is not included. (Based on data reported by A. Phillipson and D. I. McLaren.)

cent 2.28-acre square fields were plowed with the two implements, and detailed time measurements were taken for each phase of each operation. With these results, the British engineers were able to calculate the probable times that would be required to plow larger fields of various proportions.

Figure 2.1 indicates the relation of field efficiency to size of a square field and shows the comparative efficiencies for the two methods of plowing. Field efficiencies were computed on the basis of the travel speed, width of cut, and total elapsed times given in the NIAE report. These field efficiencies include allowance for the time required to lay out, open, and finish lands (with the one-way plow), time to plow out headlands, turning and idle travel time, and time for plow adjustments; they do not include
time for servicing. Note that the field efficiency drops rapidly as the area is reduced below about 15 acres.

The NIAE results indicate that the average turning times were about the same (18 to 20 sec per turn at 2.9 mph) for the 180° turns required with the mounted, two-way plow and for the turns across the ends of 90-ft lands with the trailed plow. This suggests that, with the type of equipment used in the NIAE tests, there is little to be gained in turning time by laying out lands with initial widths much less than 90 ft. Practically all the difference between the two methods of plowing in regard to field efficiency was due to the time required to lay out, open, and finish the lands required with the one-way plow.

The effect of the field proportions upon field efficiency may be illustrated by calculated results for a 9.2-acre field, which indicate relative field efficiencies of 77, 85, and 90 per cent for $L/w$ ratios of $\frac{3}{4}$, 1, and 4, respectively, when using the two-way plow.

2.10. Estimating Travel Speeds and Field Capacities. Approximate methods for estimating speeds and field capacities are often handy, both in the field and in the laboratory. The approximate speed of an implement may be determined by walking at the speed of the implement (grasping the frame helps) and counting the number of steps taken in 20 sec. The number of steps divided by 10 gives the speed in miles per hour if the average step length is 2.94 ft. For any other average step length, the number of seconds should be changed proportionately. If the speed of the machine is considerably faster or slower than a normal walking rate, it may be difficult to maintain a normal length of step. However, if the machine positions at the beginning and end of the 20-sec period are marked on the ground, the distance can later be measured at a normal walking rate or with a tape measure.

The time studies discussed in Sections 2.8 and 2.9, together with the results of a field survey made in Nebraska in 1947, provide a reasonable basis for estimating typical field efficiencies. In the Nebraska studies, actual field capacities, together with speeds and implement widths, were obtained in a survey including nearly five hundred farmers. From these results, field efficiencies can be calculated.

Typical ranges for field efficiencies under normal operating conditions are:
FIELD CAPACITIES AND COST ANALYSIS

Most tillage operations (plowing, disking, harrowing, cultivating, etc.) 75–90%
Drill planting of row crops or grain 70–85%
Checkrow planting of corn 50–65%
Combining grain or picking corn 60–75%
Mowing, raking, or baling hay 75–90%

McKibben 8 has pointed out that, if a field efficiency of 82.5 per cent is assumed (which is a reasonable value for many operations), it is evident from equation 2.1 that the capacity of a field machine, in acres per 10-hr day, is equal to the product of the speed in miles per hour times the width of cut in feet.

COST ANALYSIS

2.11. Cost Factors. The total cost of performing a field operation includes charges for the implement or machine, for the power utilized, and for labor. Machine costs include depreciation, interest on investment, taxes, insurance, shelter, repairs and maintenance, lubrication, and fuel (if the machine has an engine). The first five of these items are related to machine ownership. They occur regardless of whether or not the machine is operated and are known as fixed or overhead costs. Expenses for items such as repairs, maintenance, lubrication, fuel and oil, and labor are incurred as a result of actual machine operation. They are known as operating costs.

As mentioned in Chapter 1, farm machinery is characterized by low annual usage. Surveys have indicated that with most field machines (exclusive of tractors, trucks, and wagons), the annual use seldom exceeds 15 days.9,10,11 Machines such as balers, combines, and perhaps corn pickers, are exceptions. They may be operated for as much as 20 or 30 days per year. Planting equipment, on the other hand, is generally not needed for more than 4 or 5 days per year.

As a result of the limited use, overhead costs often represent a large part of the total machine costs. Thus the total cost per unit of work (acre, hour, etc.) can be decreased considerably by increasing the amount of use to distribute the overhead costs more, as will be discussed later. The initial investment and the length of service life are important factors in relation to overhead costs.
2.12. Depreciation. Depreciation is the reduction in value of a machine caused by obsolescence, wear, weathering, accidental damage, etc. For machines operated less than 10 to 15 days per year, the service life is not greatly influenced by the amount of use, and obsolescence is an important factor. A machine may become obsolete because of the development of improved models, changes in farming practices, etc. Where a machine has a relatively high annual use, as with custom-operated equipment on large farms, wear becomes an important factor in determining the service life.

Among the various systems employed for calculating depreciation are the straight-line, constant-percentage, estimated-value, and compound-interest or sinking-fund methods. For calculating the cost of operation of a machine, the straight-line method is the most practical and the most common. It is the simplest method, and it gives a constant annual charge for depreciation throughout the life of the machine. In the straight-line method, the amount of the annual depreciation is equal to the first cost minus the resale or salvage value, divided by the estimated life in years. Some investigators assume a resale value equal to 10 per cent of the first cost, whereas others consider that the machine will have no salvage value at the end of its service life.

The constant-percentage and estimated-value methods deprecate the machine more rapidly than the straight-line method in the early years of the machine's life, and less rapidly in later years. These and other similar methods are more suitable than the straight-line method for determining the market value of a partially depreciated machine and for tax valuation. For income tax purposes, depreciation is often based on a much shorter life than would be justified on the basis of wear.

2.13. Service Life. A common method for determining the probable service life of farm machines is to conduct a survey among a large number of farmers, obtaining from each owner the present age of the equipment and his estimate of the expected future life. The results of such surveys indicate rather wide variations between individual machines of a particular type and a considerable range of average values for different types of machines.

For purposes of cost estimating, it is suggested that machines such as field balers, combines, corn pickers, and field forage...
choppers be depreciated over a 10-yr period, when the annual use does not exceed about 250 hr. A period of 15 yr is reasonable for most tractor-operated tillage and planting equipment, as well as for the less complex types of harvesting equipment. When the annual use is considerably greater than normal, the depreciation period should be shortened accordingly.

2.14. Interest on Investment. Strictly speaking, the charge for interest should decrease over the life of the implement as the value decreases. But for cost estimating, a constant annual charge is desirable. Thus, the interest charge is usually calculated on the basis of the average investment during the life of the machine. With straight-line depreciation, the average investment is merely half the sum of the first cost plus the resale or salvage value (if any). The rate of interest should reflect prevailing rates. A value of 5 per cent is often assumed.

2.15. Taxes, Insurance, and Shelter. These are minor items in the total overhead charges, but should, nevertheless, be considered. Farm machinery is taxed at the same rate as other farm property. The established rate, which varies widely with locality, is applied to a depreciated value for the machine. An accelerated rate of depreciation (not straight-line) is often used in determining the tax base. For purposes of cost estimating, a constant, average yearly charge of 0.5 to 1 per cent of the first cost is commonly assumed.

Farm machinery is sometimes insured, although the owner frequently elects to carry the risk himself. In either case, a charge for insurance is considered by many economists to be justifiable. An annual charge of 0.25 per cent of the first cost is reasonable.

Although it is difficult to demonstrate conclusively that protection from the weather results in monetary savings, shelter is considered desirable for many types of farm machinery. The charge for housing is related to the physical size of the machine but usually ranges from 0.5 to 1 per cent of the first cost of the machine.

As a total annual charge for taxes, insurance, and housing, a value equal to 1.5 per cent of the first cost is suggested.

2.16. Repairs, Maintenance, and Lubrication. One would logically expect the cost of these items to be closely related to the amount of time the machine is operated. But a study of
cost data obtained in various farmer surveys indicates that, for normal amounts of operation, such a relation is not well defined.\(^4\) This can perhaps be explained by the limited annual use and by the fact that many repair operations are seasonal in nature.

Table 2.1 indicates suggested annual charges for repairs, maintenance, and lubrication of various implements, expressed as a percentage of the first cost of the machine. In the case of tillage implements, the costs of sharpening and replacing shares and other soil-working units are included. Fenton and Fairbanks\(^6\) state that these recommended charges are somewhat higher than the costs indicated by farmer surveys. The suggested rates are intended to represent an amount large enough to keep the machine in good repair, pay for transportation and a man's time to obtain the parts, and pay a wage to the mechanic or farmer doing the work. A farmer often tends to overlook or underestimate the last two items in reporting his expenses.

### Table 2.1 SUGGESTED VALUES FOR ESTIMATING ANNUAL REPAIR, MAINTENANCE, AND LUBRICATION CHARGES *

<table>
<thead>
<tr>
<th>Type of Machine</th>
<th>Annual Charge, per cent of first cost of machine</th>
</tr>
</thead>
<tbody>
<tr>
<td>Baler, pickup, with engine</td>
<td>3.8 †</td>
</tr>
<tr>
<td>Combine, trailed, pto (power-take-off)</td>
<td>4.0</td>
</tr>
<tr>
<td>Combine, trailed, with engine</td>
<td>3.5 †</td>
</tr>
<tr>
<td>Combine, self-propelled</td>
<td>3.4 †</td>
</tr>
<tr>
<td>Corn picker</td>
<td>4.0</td>
</tr>
<tr>
<td>Cultivator, field (duckfoot)</td>
<td>3.8</td>
</tr>
<tr>
<td>Cultivator, corn, mounted</td>
<td>3.8</td>
</tr>
<tr>
<td>Disk harrow</td>
<td>3.5</td>
</tr>
<tr>
<td>Field forage chopper</td>
<td>4.5 †</td>
</tr>
<tr>
<td>Grain drill</td>
<td>2.2</td>
</tr>
<tr>
<td>Harrow, spike-tooth</td>
<td>1.1</td>
</tr>
<tr>
<td>Lister</td>
<td>5.5</td>
</tr>
<tr>
<td>Manure spreader</td>
<td>2.0</td>
</tr>
<tr>
<td>Mower</td>
<td>4.2</td>
</tr>
<tr>
<td>Planter, corn</td>
<td>2.5</td>
</tr>
<tr>
<td>Plow, moldboard</td>
<td>7.4</td>
</tr>
<tr>
<td>Plow, vertical-disk</td>
<td>5.5</td>
</tr>
<tr>
<td>Rake, side-delivery</td>
<td>2.5</td>
</tr>
</tbody>
</table>

\(†\) Engine oil not included.

* F. C. Fenton and G. E. Fairbanks.\(^6\)
It should be kept in mind that the charges indicated in Table 2.1 are for normal amounts of annual use, ranging from 50 to 150 hr per yr for most machines, but often being as high as 200 to 250 hr for balers. It seems reasonable to suggest that if a baler is to be operated for more than 250 hr per yr, or any of the other machines more than 150 hr, then the percentage charge might be increased in proportion to the total number of hours. The annual charge for a baler used 350 hr per yr would then be $3.8 \times \frac{350}{250} = 5.3$ per cent of the first cost, whereas the charge for the same number of hours on a moldboard plow would be $7.4 \times \frac{350}{150} = 17.3$ per cent.

2.17. Fuel, Oil, and Miscellaneous Supplies. With machines such as combines, balers, and field choppers, that may be equipped with auxiliary engines, the cost of fuel and oil for the engine must be included in the total machine charge. If the actual average power requirement is known, the fuel consumption (gasoline) can be estimated with reasonable accuracy on the basis of 8.5 hp-hr per gal of fuel.²

In the case of combines working in grain, seasonal averages obtained in farmer surveys suggest a value of about ¾ gal of fuel per acre for trailed machines (regardless of size) and 1 gal per acre for self-propelled machines.

Engine oil consumption, including refills after draining, can be taken as about 3 per cent of the fuel consumption.¹²

Twine or wire required for hay balers represents another operating expense. Average requirements, as given in Section 14.28, are:

<table>
<thead>
<tr>
<th>Type of Baler</th>
<th>Twine Requirement</th>
</tr>
</thead>
<tbody>
<tr>
<td>Automatic wire-tying baler</td>
<td>7.7 lb wire per ton</td>
</tr>
<tr>
<td>Rectangular, twine-tied bales</td>
<td>3.1 lb baler twine per ton</td>
</tr>
<tr>
<td>Round bales, twine-wrapped</td>
<td>2.6 lb binder twine per ton</td>
</tr>
</tbody>
</table>

2.18. Total Cost per Unit of Work. The total cost of performing a field operation is generally desired on either a per-acre or a production-unit basis. Determination of this cost involves the following factors:

1. Total annual overhead charges for the machine (i.e., cost of ownership).
2. Annual use of machine in hours or acres.
3. Total operating costs per hour or per acre (for machine and labor).
4. Cost per hour or per acre for tractor power required by
    trailed or mounted machines.
5. Effective field capacity of the machine.

For quick reference, total overhead charges are indicated in
the accompanying table for four rates of straight-line depreci­
ation. The interest rate is taken as 5 per cent of the average in­
vestment, and the total for taxes, insurance, and shelter is 1.5
per cent of the first cost. Machine operating costs can be esti­
mated by referring to Sections 2.16 and 2.17.

<table>
<thead>
<tr>
<th>Assumed</th>
<th>Assumed Resale Value,</th>
<th>Total Overhead Charges,</th>
</tr>
</thead>
<tbody>
<tr>
<td>Life yr</td>
<td>per cent of first cost</td>
<td>per cent of first cost</td>
</tr>
<tr>
<td>10</td>
<td>0</td>
<td>14.0</td>
</tr>
<tr>
<td>10</td>
<td>10</td>
<td>13.3</td>
</tr>
<tr>
<td>15</td>
<td>0</td>
<td>10.7</td>
</tr>
<tr>
<td>15</td>
<td>10</td>
<td>10.3</td>
</tr>
</tbody>
</table>

Determination of tractor power costs involves the same factors
that have been discussed for implements. The hourly cost is
based on the total annual operating time for the tractor rather
than the annual use with the particular implement involved.
Further discussion and information on estimating tractor power
costs may be found in reference 2. In the absence of specific
data for the tractor, a fair approximation, based on 1954 costs,
is to assume 4¢ per hr per available drawbar horsepower * for
wheel tractors in the 20 to 45 hp range and 5¢ for track-type
tractors in the 30 to 80 hp range. The actual average drawbar
loading for continuous operation should not exceed 75 per cent of
the maximum available horsepower.

Labor charges are based upon prevailing wage rates. The
labor cost per acre is inversely proportional to the field capacity
of the machine. Thus, although increased field capacities often
result in lower field efficiencies because of the effect of interrup­
tions proportional to area (Section 2.6), the over-all cost per
acre is ordinarily reduced.

* Maximum drawbar horsepower from Nebraska tractor tests, adjusted
to standard atmospheric conditions. This is the value ordinarily included
in specifications published by the tractor manufacturers. In the Nebraska
tests, "rated drawbar load" is taken as 75 per cent of the maximum adjusted
drawbar horsepower.
2.19. First Cost of Field Machines. Determination of the overhead charges requires the establishment of a value for the first cost of the machine. If specific price information for a particular situation is not available, a reasonable estimate can be made on the basis of the data presented in Table 2.2. The prices in this table, which represent 1954 conditions, include an allowance for freight to a delivery point about 1000 miles from the factory. Although there is considerable variation between the prices of different manufacturers, the values in Table 2.2 do indicate the relative magnitudes of the costs of various types of equipment. As economic conditions change, these data can all be adjusted up or down accordingly.

Table 2.2 also includes calculated values for cost per pound, based upon factory list prices (i.e., freight not included). Note that the cost per pound is about the same for different machines of any one general type or degree of complexity (when manufactured under quantity-production methods). For example, average costs for various kinds of trailed tillage tools ranged from about 25 to 30¢ per lb in 1954, whereas balers, combines, corn pickers, field choppers, and other similar equipment cost 50 to 65¢ per lb. The mechanical cotton picker is the most complex and costly type of field machinery listed, with prices well over $1.00 per lb. By way of comparison, average tractor costs in 1954 ranged from 55 to 70¢ per lb for the various types and sizes.

2.20. Examples of Cost Determination. For the first example, consider a 12-ft self-propelled combine with an annual use of 150 hr, operating in grain at an average speed of 3 mph. Assume an initial cost of $5300 and a 10-yr service life with a 10 per cent trade-in value. Other pertinent factors are: interest rate = 5 per cent, fuel cost = 20¢ per gal, engine oil cost = 90¢ per gal, operator's wage = $1.25 per hr.

Section 2.10 indicates field efficiencies from 60 to 75 per cent for combines. Assume a value of 70 per cent. Then, from equation 2.1, the effective field capacity is

\[ C' = \frac{3 \times 12 \times 70}{825} = 3.05 \text{ acres per hr} \]

The total area covered per year is \(150 \times 3.05 = 457.5\) acres. From Table 2.1, the annual charge for repairs, maintenance, and
Table 2.2  TYPICAL RETAIL PRICES FOR NEW FIELD MACHINES

(1954 prices)

<table>
<thead>
<tr>
<th>Type of Machine</th>
<th>Size</th>
<th>Painl Cost per Pound</th>
<th>Average Retail Price</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Tillage Equipment</strong></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Cultivator, corn, mounted (with sweeps)</td>
<td>2-row</td>
<td>$0.30</td>
<td>$240</td>
</tr>
<tr>
<td>Cultivator, corn, mounted (with sweeps)</td>
<td>4-row</td>
<td>$0.28</td>
<td>$200</td>
</tr>
<tr>
<td>Cultivator, field (duckfoot)</td>
<td>10-ft</td>
<td>—</td>
<td>$325</td>
</tr>
<tr>
<td>Disk harrow, single-acting, trailed</td>
<td>15-ft</td>
<td>$0.26</td>
<td>$325</td>
</tr>
<tr>
<td>Disk harrow, light-duty tandem, trailed</td>
<td>8-ft</td>
<td>$0.25</td>
<td>$290</td>
</tr>
<tr>
<td>Disk harrow, heavy-duty offset, trailed</td>
<td>8-ft</td>
<td>$0.26</td>
<td>$295</td>
</tr>
<tr>
<td>Harrow, spike-tooth</td>
<td>20-ft</td>
<td>$0.29</td>
<td>$140</td>
</tr>
<tr>
<td>Harrow, spring-tooth</td>
<td>12-ft</td>
<td>$0.23</td>
<td>$160</td>
</tr>
<tr>
<td>Flow, moldboard, mounted</td>
<td>2 14-in.</td>
<td>$0.40</td>
<td>$200</td>
</tr>
<tr>
<td>Flow, moldboard, trailed</td>
<td>3 16-in.</td>
<td>$0.20</td>
<td>$100</td>
</tr>
<tr>
<td>Flow, standard disk, trailed (3-disk)</td>
<td>4-ft</td>
<td>$0.26</td>
<td>$120</td>
</tr>
<tr>
<td>Flow, vertical-disk, trailed</td>
<td>8-ft</td>
<td>—</td>
<td>$575</td>
</tr>
<tr>
<td><strong>Planting and Fertilizing Equipment</strong></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Fertilizer side-dressing attachment</td>
<td>4-row</td>
<td>$0.42</td>
<td>$250</td>
</tr>
<tr>
<td>Grain drill, single-disk opener</td>
<td>10 X 7</td>
<td>$0.31</td>
<td>$190</td>
</tr>
<tr>
<td>Manure spreader, two-wheel</td>
<td>70-hp</td>
<td>$0.29</td>
<td>$180</td>
</tr>
<tr>
<td>Planter, beet and bean, trailed</td>
<td>4-row</td>
<td>$0.38</td>
<td>$200</td>
</tr>
<tr>
<td>Planter, corn, mounted (for drilling only)</td>
<td>2-row</td>
<td>$0.42</td>
<td>$280</td>
</tr>
<tr>
<td>Planter, corn, trailed (for drilling only)</td>
<td>2-row</td>
<td>$0.40</td>
<td>$220</td>
</tr>
<tr>
<td>Planter, corn, trailed checkrow</td>
<td>2-row</td>
<td>$0.38</td>
<td>$280</td>
</tr>
<tr>
<td><strong>Harvesting Equipment</strong></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Baler, automatic twine-tying, pto-driven (light-duty)</td>
<td>14 X 18 in.</td>
<td>$0.83</td>
<td>$1,525</td>
</tr>
<tr>
<td>Baler, automatic twine-tying, with engine</td>
<td>14 X 18 or 16 X 18 in.</td>
<td>$0.61</td>
<td>$2,500</td>
</tr>
<tr>
<td>Baler, automatic wire-tying, with engine</td>
<td>14 X 18 or 16 X 18 in.</td>
<td>$0.61</td>
<td>$2,750</td>
</tr>
<tr>
<td>Baler, automatic, round-bale, pto-driven</td>
<td>—</td>
<td>$0.50</td>
<td>$1,200</td>
</tr>
<tr>
<td>Combine, trailed, pto-driven</td>
<td>6-ft</td>
<td>$0.42</td>
<td>$1,500</td>
</tr>
<tr>
<td>Combine, trailed, with engine</td>
<td>6-ft</td>
<td>$0.48</td>
<td>$1,900</td>
</tr>
<tr>
<td>Combine, trailed, with engine</td>
<td>12-ft</td>
<td>$0.52</td>
<td>$3,400</td>
</tr>
<tr>
<td>Combine, self-propelled</td>
<td>12-ft</td>
<td>$0.63</td>
<td>$5,300</td>
</tr>
<tr>
<td>Corn picker-harvester, mounted</td>
<td>2-row</td>
<td>$0.52</td>
<td>$1,500</td>
</tr>
<tr>
<td>Cotton picker, self-propelled (high-drum type)</td>
<td>2-row</td>
<td>$1.30</td>
<td>$3,500</td>
</tr>
<tr>
<td>Cotton picker, mounted, heavy-duty (high-drum type)</td>
<td>1-row</td>
<td>$1.18</td>
<td>$6,900</td>
</tr>
<tr>
<td>Cotton picker, mounted (fan-drum type)</td>
<td>1-row</td>
<td>$1.10</td>
<td>$5,000</td>
</tr>
<tr>
<td>Cotton stripper, mounted</td>
<td>2-row</td>
<td>$0.70</td>
<td>$1,225</td>
</tr>
<tr>
<td>Field chopper, row-crop attachment, pto-driven</td>
<td>1-row</td>
<td>$0.50</td>
<td>$1,500</td>
</tr>
<tr>
<td>Field chopper, row-crop attachment, engine</td>
<td>1-row</td>
<td>$0.55</td>
<td>$2,200</td>
</tr>
<tr>
<td>Window pickup attachment</td>
<td>—</td>
<td>—</td>
<td>$275</td>
</tr>
<tr>
<td>Mower, mounted</td>
<td>7-ft</td>
<td>$0.50</td>
<td>$300</td>
</tr>
<tr>
<td>Reel, reel-type side-delivery, trailed</td>
<td>4-bar</td>
<td>$0.34</td>
<td>$400</td>
</tr>
<tr>
<td>Sugar beet harvester, mounted, with trailer cart</td>
<td>1-row</td>
<td>$0.90</td>
<td>$2,900</td>
</tr>
</tbody>
</table>

* Cost per pound is based on factory list price (freight not included).
† Retail prices include freight to a point about 1000 miles from the factory.
‡ Includes tire and tube for rear Farrow wheel but not for other two wheels. Add total of $49 to $50 to retail price if new tires are purchased.
§ Tires and tubes not included. Add $40 to $50 to retail price if new tires are purchased. Used automobile tires are often employed.
lubrication is 3.4 per cent of the first cost. Calculations for the various charges and the total cost per acre are as follows:

### Annual Overhead Charges

<table>
<thead>
<tr>
<th>Item</th>
<th>Calculation</th>
<th>Cost per Acre</th>
</tr>
</thead>
<tbody>
<tr>
<td>Depreciation</td>
<td>( \frac{5300 - 530}{10} )</td>
<td>$477.00</td>
</tr>
<tr>
<td>Interest</td>
<td>( 0.05 \left( \frac{5300 + 530}{2} \right) )</td>
<td>145.75</td>
</tr>
<tr>
<td>Taxes, insurance, shelter (Section 2.15)</td>
<td>( 0.015 \times 5300 )</td>
<td>79.50</td>
</tr>
<tr>
<td><strong>Total annual overhead charge</strong></td>
<td></td>
<td>$702.25</td>
</tr>
</tbody>
</table>

### Cost per Acre

<table>
<thead>
<tr>
<th>Item</th>
<th>Calculation</th>
<th>Cost per Acre</th>
</tr>
</thead>
<tbody>
<tr>
<td>Overhead</td>
<td>( \frac{702.25}{457.5} )</td>
<td>1.53</td>
</tr>
<tr>
<td>Repairs, maintenance, lubrication</td>
<td>( \frac{0.034 \times 5300}{457.5} )</td>
<td>0.39</td>
</tr>
<tr>
<td>Fuel, 1 gal per acre (Section 2.17)</td>
<td>( 1 \times 0.20 )</td>
<td>0.20</td>
</tr>
<tr>
<td>Engine oil, 0.03 gal per acre</td>
<td>( 0.03 \times 0.90 )</td>
<td>0.03</td>
</tr>
<tr>
<td>Labor</td>
<td>( \frac{1.25}{3.05} )</td>
<td>0.41</td>
</tr>
<tr>
<td><strong>Total cost per acre</strong></td>
<td></td>
<td>$2.56</td>
</tr>
</tbody>
</table>

As a second example, consider a three-bottom, 16-in. trailed moldboard plow. The first cost is $375, and the annual use is 125 hr. Assume a service life of 15 yr with no salvage value. The plowing speed is 3\( \frac{1}{2} \) mph, and the average actual power requirement is 22 hp. As in the first example, the interest rate is 5 per cent and the operator's wage is $1.25 per hr.

Section 2.10 indicates field efficiencies from 75 to 90 per cent. Assuming a value of 82.5 per cent and applying the short-cut method (Section 2.10), the effective field capacity is \( 3.5 \times 4 = 14 \) acres per 10-hr day or 1.40 acres per hr. The required tractor size, in terms of available drawbar horsepower, is \( \frac{22}{0.75} = 29.3 \) hp (loaded to 75 per cent of maximum). Assume a wheel tractor, with a power cost of 4¢ per hr per available horsepower (Section 2.18). From Table 2.1, the annual charge for repairs, maintenance, and lubrication is 7.4 per cent (including sharpening and replacement of shares).
The details of the various charges are as follows:

**Annual Overhead Charges**

Total (Section 2.18) \( 0.107 \times 375 = 40.12 \)

**Cost per Hour**

<p>| | | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Overhead</td>
<td>( \frac{40.12}{125} )</td>
<td>$0.32</td>
</tr>
<tr>
<td>Repairs, maintenance, lubrication</td>
<td>( \frac{0.074 \times 375}{125} )</td>
<td>0.22</td>
</tr>
<tr>
<td>Tractor power</td>
<td>( 0.04 \times 29.3 )</td>
<td>1.17</td>
</tr>
<tr>
<td>Labor</td>
<td></td>
<td>1.25</td>
</tr>
<tr>
<td><strong>Total cost per hour</strong></td>
<td></td>
<td><strong>$2.96</strong></td>
</tr>
</tbody>
</table>

**Cost per Acre**

Total \( \frac{2.96}{1.40} = 2.11 \)

Note that in this example the total overhead charge is obtained by applying the appropriate percentage value from Section 2.18. This procedure could have been followed in the first example also. Insofar as the mechanics of the calculations are concerned, it makes little difference whether the individual items of cost are reduced to a per-acre, per-hour, or annual basis as a common denominator.

**2.21. Effect of Annual Use upon Costs.** Figure 2.2 shows the relation of total cost per acre to the total hours of annual use, for the machines and conditions of the two preceding examples. For the self-propelled combine, the initial investment is high, and overhead charges represent a large part of the total cost, even for an annual use of 150 hr. The total cost per acre becomes prohibitive as the annual use is reduced to 50 hr (which represents 150 acres per yr in this case).

The moldboard plow is an example of an implement for which power and labor costs are relatively high in relation to fixed charges. But even so, the total cost per acre increases considerably as the annual use is reduced below about 100 hr.

**2.22. Justification for Machine Ownership.** It is evident from Fig. 2.2 that expensive machines such as the large, self-propelled combine are not economically practical for individual ownership on small farms. As pointed out in Chapter 1, this is
one of the serious problems of mechanization. Custom operation and joint ownership by several farmers are two methods of increasing the annual use and thus reducing unit costs. But, with either arrangement the individual farmer sacrifices timeliness and independence in scheduling his operations.

Fig. 2.2. Relation of calculated cost per acre to hours of annual use, for two types of field machines.

Timeliness is an important factor in determining the amount of annual use that will justify the purchase of a particular machine. Figure 2.2, for example, indicates that a self-propelled combine should be operated for at least 200 hr per yr to obtain reasonably low unit costs. But a farmer who can harvest his rice crop in 20 days with one combine may decide to use two machines and complete the operation in 10 days, even though the cost per acre is considerably greater. The additional cost of harvesting with two machines instead of one is insurance against crop losses due to adverse weather conditions that might occur during the last half of the 20-day period. Similarly, the additional cost per acre for harvesting a given total acreage of corn
with a two-row picker rather than a one-row machine may be justified by reduced field losses resulting from earlier completion of the job (see Fig. 18.10).

Timeliness may also result in economic advantages due to greater yields, better market conditions, etc. In some cases, particularly on small farms, the choice may lie between hand labor and a more expensive mechanical method. If the cost difference is not too great, the machine operation may be preferred because of the elimination of problems associated with obtaining and managing hand laborers. Convenience, mobility, and other similar factors are difficult to evaluate but have a definite bearing on economic considerations.

2.23. Mounted and Self-Propelled Machines. At the present time there is a strong trend toward tractor-mounted and semi-mounted equipment and certain types of single-purpose, self-propelled machines. Interchangeable harvesting units for a common, self-propelled chassis or carrier have also entered the picture. Economic adaptability is of prime importance with each type of equipment.

In comparison with trailed machines, self-propelled units tend to provide more flexibility and better maneuverability, better visibility and control by the operator, and improved mobility. With harvesting equipment, field opening losses are minimized because the cutting or gathering unit is across the entire front of the machine rather than projecting to one side.

Because of the greater initial investment, a single-purpose, self-propelled machine must have a relatively large annual use to be economically comparable with a trailed unit. Figure 2.3 shows calculated harvesting costs for 7-ft trailed and self-propelled combines. When the trailed machine is charged with an assumed field opening loss, the costs per acre for the two machines become equal at about 300 acres per yr (150 hr under the conditions assumed). If there is no field opening loss, or if this charge is merely omitted, the trailed machine is more economical even at 400 acres per yr. Cost calculations for two-row mounted and self-propelled corn pickers indicate a similar relation, with the mounted unit being more economical up to more than 400 acres per yr.

One means of reducing the cost of self-propelled machines is to provide a skeleton carrier frame or chassis equipped with a
power unit, upon which various types of machines can be mounted. The interchangeable units might include a combine, a corn picker, a baler, a field chopper, a cotton picker, and perhaps others. Considerable engineering ingenuity is required to provide a carrier that will accommodate such a variety of machines and properly satisfy their basic requirements.

![Graph showing the relation of cost per acre to annual use, for 7-ft trailed and self-propelled combines.](image)

Fig. 23. Relation of cost per acre to annual use, for 7-ft trailed and self-propelled combines. (E. L. Barger. *Agr. Eng.*, March, 1948.)

The use of interchangeable power units, both for self-propelled machines and as auxiliary engines on trailed implements, has been given engineering consideration as a means of reducing total investments. Again, ease of handling and interchanging is of prime importance.

Whereas self-propelled machines are more costly than trailed units, tractor-mounted or semimounted equipment is inherently less expensive. The support wheels and accompanying frame structure required on trailed implements are eliminated, and a single control system serves for all or most of the mounted tools on any one tractor. Maneuverability, visibility of the work,
and ease of transport are features that tend to make mounted implements popular. Since the farmer may need to change implements rather frequently, easy and rapid attaching and detaching of the tools are important features.

The physical size and weight of equipment that can be mounted on a tractor and fully supported by it are limited by the carrying capacity of the tractor chassis and by the transport stability of the combination. Semimounting, in which the tractor carries only part of the implement weight, overcomes these limitations and still retains the advantages of maneuverability, direct steering, and lower investment cost. In the cast of semimounted harvesting equipment, however, visibility from the tractor seat may not be as good as from the operator's position on a self-propelled unit.

REFERENCES


PROBLEMS

2.1. A farmer using a three-bottom, 16-in. moldboard plow at 3½ mph covers 15 acres in 11 hr. What is his field efficiency?

2.2. The over-all time required for the tank-filling operation on a field spraying job amounts to 10 min per acre. The time required for turning at the ends is 14 per cent of the effective operating time. The boom has 20 nozzles spaced 18 in. apart. The average amount of overlap between successive trips through the field is 9 in. and the speed is 4 mph. Neglecting other time losses, compute:
(a) The field efficiency.
(b) The effective field capacity.

2.3. Plot a curve of field efficiency versus length of field, using lengths up to 1 mile, for cultivating corn at 3 mph with a two-row, tractor-mounted cultivator. Assume that time lost due to minor repairs, adjustments, and miscellaneous interruptions is equal to 12 per cent of the effective operating time and that turning at each end requires 20 sec.

2.4. If a person has a step length of 3.3 ft, for how many seconds should he count his steps in determining the forward speed of a machine, in order to have mph equal number of steps divided by 10? Indicate the basic relations used in solving the problem.

2.5. A 10-ft grain drill is pulled at 4½ mph with a 20-hp wheel-type tractor. The initial cost of the drill is $600, and the labor cost is $1.00 per hr.
(a) If 80 acres per year are planted with the drill, what is the total cost per acre for the planting operation?
(b) What is the cost if the annual use is 250 acres? State and justify any assumptions made.

2.6. A farmer has a choice of buying a two-bottom, 14-in. mounted plow for $190 (new) or a three-bottom, 14-in. trailed plow for $450. Assume that the cost per acre for tractor energy would be the same in either case. With either plow, operating speed = 3½ mph and field efficiency = 82.5 per cent.
(a) If the labor cost is $1.10 per hr, what is the minimum number of acres plowed per year that would justify the purchase of the larger plow (i.e., the "break-even" point)?
(b) How would reduced labor costs affect the break-even acreage?

2.7. An automatic twine-tying baler equipped with an engine is pulled with a two-plow-size tractor. The average baling rate (including lost time) is 5 tons per hr, and the corresponding average power requirement from the baler engine is 9 hp. The first cost of the baler is $2400. Assume a 10-yr service life with a $300 trade-in value. For the tractor, first cost = $2000,
total annual charges (overhead, repairs, maintenance, and lubrication) = 18 per cent of the first cost, fuel consumption = 1 1/2 gal per hr, and total annual use = 800 hr. Fuel cost = 19¢ per gal, oil cost = 85¢ per gal, twine cost = 25¢ per lb, interest rate = 5 per cent, labor cost = $1.15 per hr. Calculate the total baling cost per ton for:
(a) 200 tons per yr.
(b) 500 tons per yr.

2.8. Under the conditions of Problem 2.7, how many tons per year must a farmer bale in order to justify ownership of a baler (i.e., the break-even point on cost per ton), if the custom baling rate is $4.00 per ton?
CHAPTER 3

Materials of Construction

3.1. Introduction. Selection of the proper material and treatment for a particular application on a farm implement is important from the standpoints of cost, durability, availability, and machine performance. Implement parts and components should generally be designed to utilize the lowest-cost materials that will perform satisfactorily and give adequate life. Unfortunately, it sometimes becomes necessary to substitute high-cost materials and expensive treatments for low-cost materials to make up for deficiencies in the original design.

Steel and cast iron are the major materials employed in the manufacture of farm equipment. Brass, bronze, and babbit are used to a limited extent, principally for bearings. Wood is suitable for some slow-speed bearings (such as straw walkers on a combine), for reel slats, mower pitmans, and manure-spreader boxes, and for other special applications. Rubber and fabrics are, of course, found in such items as belts, hoses, conveyors, and tires. Synthetic rubbers are being used for torsional bushings to support oscillating hangars as on a combine, and to a limited extent in certain applications where resilient mountings are advantageous. Anderson suggests that aluminum will be used to some extent in the future under conditions where resistance to corrosion and saving in weight are particularly advantageous and that plastics will find some application where weight saving and resistance to wear are important. At the present time both aluminum and plastics are relatively expensive for any but special farm machinery applications. Stainless steel, in spite of its high cost, may prove practical for certain parts in fertilizer
distributors and other machines where corrosion is a serious problem.

3.2. Classification of Steels. Steel is a malleable alloy of iron and carbon or of iron, carbon, and additional alloying elements. The carbon content of steels may be as low as about 0.05 per cent or it may be well above 1 per cent. If the only alloying element present is carbon, the material is known as carbon steel. If the steel owes its distinctive properties chiefly to some element or elements other than carbon, or jointly to such other elements and carbon, it is known as an alloy steel. The physical properties of either carbon steels or alloy steels can be altered considerably by various heat-treating operations.

The chemical composition of a steel is commonly specified according to the SAE (Society of Automotive Engineers) numbering system, which is an accepted standard throughout industry. With a few exceptions, the numbers in this system are composed of four digits. The first figure indicates the class or type of steel. Thus 1xxx indicates a carbon steel, 2xxx a nickel steel, 3xxx a nickel-chromium steel, etc. The last two (or sometimes three) figures indicate the carbon content in "points," or hundredths of 1 per cent. For simple alloy steels the second figure generally indicates the approximate percentage of the predominant alloying element.

Thus, an SAE 1045 steel is a carbon steel containing 0.45 per cent carbon, whereas 2340 indicates a nickel steel with approximately 3 per cent nickel (actually 3.25 to 3.75 per cent) and 0.40 per cent carbon. SAE standards specify the allowable limits of chemical composition for each SAE steel.

The AISI (American Iron and Steel Institute) number for a steel is similar to the SAE number except that a capitalized prefix letter indicates the steel-making process. Steels having the same AISI and SAE numbers (other than the prefix) are identical in regard to chemical composition.

3.3. Heat Treating of Steels. Heat treating is an operation or combination of operations that involves the heating and cooling of a metal or alloy in the solid state for the purpose of obtaining certain desirable conditions or properties. Hardening is a heat treatment for certain steels that generally consists of heating the part to a temperature within or above the trans-
formation range* and then cooling it rapidly (quenching) by
submerging it in an appropriate quenching medium such as water
or oil. Generally speaking, increasing the hardness of a steel
by heat treatment increases the strength and wear resistance
and reduces the ductility. The hardenability of a steel is directly
related to its carbon content. Steels having less than about 0.25
to 0.30 per cent carbon are not hardened to any great degree by
heat treatment.

After quenching, the steel will be very strong and hard, but it
will not be suitable for use because of lack of ductility and the
presence of residual stresses. A subsequent operation known
as tempering (or drawing) relieves the quenching stresses and
recovers a limited degree of toughness and ductility, but the
tensile strength is also reduced. This operation consists of re­
heating the part to a temperature below the transformation range
and then either quenching it or cooling it in air. The amount
of increase in ductility and the accompanying reduction in tensile
strength depend upon the temperature employed in tempering
and upon the rate of cooling.

Bornstein illustrates the effect of heat treatment by citing
the results given in the accompanying table for a carbon-steel

<table>
<thead>
<tr>
<th></th>
<th>As Received</th>
<th>Quenched from 1000°F</th>
<th>Drawn at 1000°F</th>
</tr>
</thead>
<tbody>
<tr>
<td>Yield strength, psi</td>
<td>48,200</td>
<td>99,500</td>
<td>93,500</td>
</tr>
<tr>
<td>Ultimate strength, psi</td>
<td>98,600</td>
<td>152,600</td>
<td>139,500</td>
</tr>
<tr>
<td>Elongation in 2 in., per cent</td>
<td>15.0</td>
<td>14.0</td>
<td>15.5</td>
</tr>
<tr>
<td>Brinell hardness</td>
<td>192</td>
<td>302</td>
<td>275</td>
</tr>
</tbody>
</table>

plow beam containing 0.68 per cent carbon. Note that the heat
treatment increased the ultimate strength by about 40 per cent
and nearly doubled the yield strength.

When maximum softness and ductility are desired, the steel
is full annealed. This process involves heating slowly to a tem­
perature above the transformation range (usually about 100°F
above the upper limit), holding it at this temperature for a con­

*Transformation temperatures are the temperatures at which phase
changes occur as steel is heated or cooled. The transformation range as
steel is heated is the range of temperatures within which austenite is
formed.10
siderable period of time, and then cooling very slowly in the furnace or in an insulated container.

*Normalizing* consists of heating the part to a temperature about 100°F above the transformation range and then cooling it in still air at ordinary temperatures. It is a treatment frequently applied to steel from the mill before machining, as well as to forgings and castings, to insure a uniform grain structure. 6

3.4. Carbon Steels. Carbon is a powerful alloying agent for steels; wide changes in the steel properties can be secured by varying the amount of this element, as indicated in Table 3.1.

Table 3.1 TYPICAL PROPERTIES OF HOT-WORKED STEEL AND OF CAST IRONS *

<table>
<thead>
<tr>
<th>Kind of Material</th>
<th>Approximate Per Cent Carbon</th>
<th>Ultimate Strength, psi</th>
<th>Yield Strength, psi</th>
<th>Elongation in 2 in., per cent</th>
<th>Modulus of Elasticity, psi</th>
<th>Brittleness Modulus of Fracture, psi</th>
</tr>
</thead>
<tbody>
<tr>
<td>Iron (0.07% Fe)</td>
<td>0.01</td>
<td>40,000</td>
<td>20,000</td>
<td>40</td>
<td>29,700,000</td>
<td>50-90</td>
</tr>
<tr>
<td>Soft steel</td>
<td>0.10</td>
<td>60,000</td>
<td>30,000</td>
<td>35</td>
<td>29,100,000</td>
<td>120</td>
</tr>
<tr>
<td>Structural steel</td>
<td>0.25</td>
<td>80,000</td>
<td>40,000</td>
<td>30</td>
<td>28,000,000</td>
<td>150</td>
</tr>
<tr>
<td>Machinery steel</td>
<td>0.40</td>
<td>100,000</td>
<td>50,000</td>
<td>25</td>
<td>28,000,000</td>
<td>180</td>
</tr>
<tr>
<td>Spring steel</td>
<td>0.75</td>
<td>120,000</td>
<td>60,000</td>
<td>12</td>
<td>28,300,000</td>
<td>240</td>
</tr>
<tr>
<td>Tool steel</td>
<td>0.90</td>
<td>140,000</td>
<td>70,000</td>
<td>8</td>
<td>28,000,000</td>
<td>290</td>
</tr>
<tr>
<td>Gray cast iron</td>
<td>3.50</td>
<td>22,000</td>
<td>17,000</td>
<td>6</td>
<td>13,000,000</td>
<td>120-140</td>
</tr>
<tr>
<td>Strong gray cast iron</td>
<td>3.50</td>
<td>30,000</td>
<td>25,000</td>
<td>&lt;1</td>
<td>14,000,000</td>
<td>--</td>
</tr>
<tr>
<td>High-strength cast iron</td>
<td>3.00</td>
<td>50,000</td>
<td>40,000</td>
<td>&lt;1</td>
<td>15,000,000</td>
<td>--</td>
</tr>
<tr>
<td>Alloy cast iron</td>
<td>2.75</td>
<td>70,000</td>
<td>--</td>
<td>&lt;1</td>
<td>18,000,000</td>
<td>--</td>
</tr>
<tr>
<td>Duplex cast iron, annealed</td>
<td>3.50</td>
<td>75,000</td>
<td>--</td>
<td>15</td>
<td>23,000,000,000</td>
<td>185</td>
</tr>
<tr>
<td>Malleable cast iron</td>
<td>2.50</td>
<td>54,000</td>
<td>30,000</td>
<td>18</td>
<td>25,000,000</td>
<td>110-145</td>
</tr>
</tbody>
</table>


† From *Mechanical Engineers' Handbook.* 8

Note: The compressive strength of cast iron is 80,000 to 150,000 psi. 8 For steel the compressive yield strength is the same as the tensile yield strength. *

Strength, hardness, and resistance to abrasive wear (as on tillage tools) increase as the carbon content is increased. The available range of properties is further expanded by the various possibilities for heat treating. The cost also increases as the carbon content is increased, and higher carbon steels become increasingly difficult to machine, weld, and forge. 11

Because of its great versatility and its relatively low cost as compared with alloy steels, carbon steel is used very extensively in the manufacture of farm implements. Some of the typical applications for carbon steels (including some in fields other than the farm machinery industry) are as follows: 8, 9, 11
Per Cent Carbon

Applications

0.05-0.10 Stampings, sheets, wire, rivets, welding stock, cold-drawn parts.

0.10-0.30 Rolled structural shapes, formed structural parts, control levers and rods, shafts, machine and carriage bolts, screws, welded tubing, parts to be carburized, such as gears, sprockets, and small forgings.

0.30-0.60 Shafting and parts to be induction hardened, axles, gears, seamless tubing, large forgings, tool standards, mower cutter bars, structural members.

0.60-0.80 Plow beams, cultivator tool bars, springs (coil or flat), drop-forging dies, hammers, wrenches, band saws, set screws, lock washers.

0.80-1.00 Plowshares, moldboards, harrow and plow disks, cultivator sweeps and shovels, hay-rake teeth, cutting and blanking punches and dies, springs.

1.00-1.20 Drills, taps, milling cutters, lathe tools, knives.

1.20-1.40 Files, reamers, razors, saws, metal-cutting tools.

The first two groups in the above list are often known as mild or low-carbon steels, the 0.30 to 0.60 range as medium-carbon steels, the 0.60 to 0.80 range as medium high-carbon steels, and the higher groups as high-carbon or tool steels. As previously indicated, the medium- and high-carbon steels can be hardened directly by heat treating. In some applications (plow beams, for example) the primary reason for heat treating is to increase the strength, whereas in other applications it may be to increase wear resistance or both strength and wear resistance.

3.5. Structural Shapes in Steel. Most of the steel for structural members in farm machinery is mild or low-carbon steel. Both rolled structural shapes (angle irons, channels, tee bars, tubes, round or square bars, etc.) and structural members formed from sheet steel are used extensively, the choice being influenced by such factors as the nature of the member, the type of loading, and the volume of production. The engineer should recognize that with mill-run, rolled structural shapes rather wide tolerances or variations in cross-sectional dimensions will be encountered, and he must make his design flexible enough to accommodate these variations.

Designs utilizing members formed from sheet steel are virtually unlimited. Channels, for example, can be designed so that the end of one member just fits between the flanges of a
second member for convenience in welding. Material can sometimes be saved by specifying a formed beam that is tapered from end to end, the justification for this design being influenced by the size of the member. Expensive forming equipment is needed in order to make structural members from heavy sheet stock (such as 7-gage), and tooling costs for a particular item may be high. But production stockpile problems are simplified if sheet stock is purchased rather than a large number of different rolled shapes. Designs of formed members may be changed without obsoleting the material already in stock.

3.6. Soft-Center Steel. For many years soft-center steel has been used to obtain plow bottoms that have a smooth, hard surface and yet are tough enough to withstand considerable shock. This material is a three-ply steel, the outer layers being high-carbon steel (usually about 1.00 per cent carbon \(^{11}\)) and the center layer being low-carbon steel (such as SAE 1010). After heat treatment the outer layers are extremely hard whereas the center layer is soft and gives toughness to the product.

3.7. Alloy Steels. Most alloy steel is medium- or high-carbon steel to which various elements have been added to modify its properties, but it still owes its distinctive characteristics to the carbon it contains. Although alloy steels are used to a considerable extent in the manufacture of tractors, their greater cost as compared with carbon steels has generally limited their application in farm implements to such items as gears, shafting, and bearings, and to special conditions where the service is severe or the heat-treating conditions are difficult.

Alloy steels, in comparison with carbon steels, generally have greater strength for a given ductility or greater ductility for a given strength. Alloys do not, however, increase the stiffness of a particular part, since the modulus of elasticity is practically the same for all steels. In many cases stiffness is the governing criterion in design, so that an alloy steel would have no advantage over a carbon steel.

One of the principal reasons for the use of alloys in steel is to increase the hardenability or depth of hardening. This characteristic permits hardening at a lower rate of heat extraction than is required for plain carbon steel. Consequently, most alloy steels are quenched in oil rather than water, which results in less temperature difference between the center and the surface.
during quenching and, hence, less tendency for the steel to warp or crack during the hardening process. In some applications alloys are employed to increase wear resistance by permitting a greater degree of hardness for a given strength level.

3.8. Surface Hardening of Steels. Some machine parts (such as gears) require a hard surface to resist wear, abrasion, or local deformation from impact loads and at the same time must have a tough, shock-resistant core. A structure of this type can be obtained by any one of several surface-hardening processes, such as carburizing, nitriding, cyaniding, carbonitriding, flame hardening, and induction hardening.

In the carburizing or case-hardening process, the parts are subjected to one of the following treatments to increase the carbon content of the outer layers: (a) packed in a carbonaceous material and held at a temperature of 1600 to 1700°F for a number of hours, (b) heated in the presence of a carbon-rich gas, or (c) immersed in a molten carburizing salt bath at 1500 to 1650°F. The outer layers absorb enough carbon so that subsequent heating and quenching, followed by tempering, produces hard surface layers. Carburizing steels are low in carbon (usually 0.15 to 0.25 per cent), so that the core will remain soft and tough after the heat-treating process.

In carbonitriding, ammonia is supplied in addition to a carbon-rich gas. An exceptionally hard surface is obtained, and oil quenching is satisfactory, with less distortion of parts than from the more drastic quenches.

Nitriding produces a very hard surface layer by the absorption of nitrogen during a long heating period (usually 50 to 90 hr) at 900 to 1000°F in the presence of ammonia. No quenching operation is required.

In the cyaniding process, a very hard surface 0.002 to 0.010 in. thick is formed by immersing the part for a short time (15 min to 1 hr) in a bath of molten sodium cyanide and then quenching in water or oil.

Induction hardening with high-frequency electric currents is being employed increasingly in many factories. It permits localized hardening where desired and often allows the use of carbon steel instead of an alloy steel. Both induction hardening and flame hardening depend upon heating and quenching the outer layers so rapidly that the interior of the body is not heated
enough to be affected by the treatment. The chemical analysis of the steel must be such as to respond to heat treatment.

3.9. Hard Facing. The addition of a layer or coating of a special abrasion-resistant alloy on a portion of the surface of a steel part is known as hard facing or hard surfacing. The principal agricultural application for hard-facing alloys is on the cutting edges of tillage tools to reduce the rate of wear (see Section 7.10). These materials, sold under various trade names, are generally nonferrous, chromium-cobalt-tungsten alloys, or high-carbon iron-base alloys containing such elements as chromium, tungsten, manganese, silicon, and molybdenum. They are applied to the cutting edges of plowshares or other tillage tools by means of the electric arc or an acetylene torch. Skill and care are required for their proper application.

3.10. Cast Iron. Cast iron is an alloy of iron containing so much carbon (2 to 4 per cent) that, as cast, it is not appreciably malleable. Gray cast iron contains 0.30 to 0.75 per cent carbon in the combined form as carbides of iron. The balance of the carbon occurs as graphite flakes that are responsible for the relatively low tensile strength of gray iron and give a characteristic gray color to the fracture. Gray iron is also characterized by a considerable amount of silicon (usually between 1.4 and 2.2 per cent), which reduces the ability of the iron to retain carbon in chemical combination and thus has a powerful softening effect.

Gray iron is relatively cheap and is the most common construction material for castings. It is strong in compression but relatively weak in tension (Table 3.1), the strength varying widely with the chemical composition. It is general practice to classify cast irons with respect to tensile strength, suitable compositions to meet the strength specifications being chosen by the producing foundry unless the customer specifies otherwise. The strength and hardness increase as the carbon content is reduced, but the cost also increases. Gray iron is notably lacking in ductility and in the ability to withstand impact or shock, but it has excellent vibration-damping characteristics as well as good resistance to wear.

Cast iron is often employed where rigidity of a part is important and extra weight is desirable, the rigidity being obtained by providing considerably heavier sections than would be economical with steel. But, for parts of identical dimensions, steel would be
more rigid than cast iron because its modulus of elasticity is about twice as great. With steel the modulus of elasticity is practically constant regardless of the chemical composition (Table 3.1), but with cast iron the modulus of elasticity, and hence the stiffness or rigidity, increases as the tensile strength is increased. For bodies of intricate shapes, castings have inherent advantages over built-up or fabricated parts. The final choice, however, depends upon the individual part concerned, the production quantity involved, and the structural requirements of the part. Although more material is required for a casting, much of the welding of steel assemblies is still hand work and is costly for complex shapes. Castings are most advantageous for parts involving a large production tonnage.

Alloying elements such as nickel and chromium can be added to cast iron to increase its tensile strength, hardness, wear resistance, and response to heat treatment. Alloy cast irons have not been employed to any great extent in the manufacture of farm implements.

3.11. White or Chilled Cast Iron. When special low-silicon cast irons are cooled rapidly after pouring, all the carbon remains in the combined form and white cast iron is formed. The absence of graphite flakes gives the fracture a characteristic white color. This material is very hard and brittle and cannot readily be machined. Its principal uses are for parts that are intended to resist abrasion or wear (such as bearings for disk-harrow gangs or wheel bearings for implements) and as an intermediate product in the formation of malleable iron.

Chilled-iron castings are made by cooling all or portions of the outer surface of the part so rapidly that white cast iron is formed in these surface layers while the remainder of the casting is gray iron and remains soft. This is accomplished by making a portion of the mold out of metal rather than sand. The metal portion, known as a chill, removes heat from the molten metal very rapidly. Thus a hard, abrasion-resistant surface is obtained in combination with a soft body that gives some degree of toughness to the product. The principal applications of chilled iron in farm implements are in shares and moldboards of plow bottoms (where only one side is hardened) and in certain types of bearings to which the chilling process is adaptable.
3.12. Malleable Cast Iron. In making malleable iron, a white cast iron of suitable composition is first produced (carbon, all in the combined form). The metal is then subjected to a lengthy annealing process in which it is held at a temperature of about 1600°F for several days and then cooled very slowly. This treatment changes practically all the combined carbon into fine nodules of graphite known as temper carbon. The white castings may be packed in iron oxide for the annealing process, which reduces the total carbon content of the product.

Malleable iron is tough, easily machined, and considerably stronger than gray cast iron (Table 3.1). It can be bent or twisted to some extent, and is preferable to gray cast iron when the part is likely to be subjected to shock or occasional overloads. Malleable iron is used rather extensively in farm implements. Malleable castings are ordinarily made with sections that do not exceed a thickness of $\frac{3}{4}$ in.\textsuperscript{8}

3.13. Ductile or Nodular Cast Iron. A relatively new product, formally announced to the foundry industry in 1940 and known as ductile cast iron, shows promise for certain farm implement applications.\textsuperscript{6} In producing this material, magnesium is added to the molten cast iron in such a way as to convert the usual graphite flakes to a spheroidal form similar to that found in malleable cast iron. The magnesium also acts as a desulfurizer.

According to Geiger and Northrup,\textsuperscript{6} the resulting product may have a yield point of 55,000 to 70,000 psi and an ultimate tensile strength of 85,000 to 95,000 psi, as compared with an ultimate strength of not over 30,000 psi for ordinary gray iron. The material exhibits a definite ductility (3 to 8 per cent elongation as cast\textsuperscript{6}) and has a much greater impact resistance than gray iron.

Chilled ductile cast iron can be made in the same manner as ordinary chilled iron, giving an abrasion-resistant surface with a moderately ductile backing. The wear resistance of this material in plowshares is said to be better than that of ordinary white chilled iron.\textsuperscript{6} Induction or flame hardening is suitable for parts such as sprockets, where chilling is impractical.
REFERENCES


CHAPTER 4

Elements of Rotary Power-Transmission Systems

4.1. Introduction. This chapter is intended to acquaint the student with some of the characteristics of the rotary power transmission systems most common in agricultural machines. Emphasis is placed upon the differences between agricultural applications and industrial drive systems, with only a minimum of space devoted to subjects that are adequately treated in machine design textbooks, engineering handbooks, or other similar references. As the student studies various types of machines during the course he should observe and evaluate specific applications of the various drive systems and their components.

BELT DRIVES

Belts are employed rather extensively in agricultural machinery applications in which it is not necessary to maintain an exact speed ratio. In a properly designed system, belt slippage (including creep) does not exceed 1 to 2 per cent, and power transmission efficiencies (not considering shaft bearing losses) range from 97 to 99 per cent. Belts tend to cushion shock loads, do not require lubrication, are quiet, and can be operated at linear speeds up to 5000 fpm and sometimes higher. They are not suitable for heavy loads at low belt speeds.

4.2. General Considerations. Any type of friction belting is subjected to stresses arising from the following sources:

1. Effective pull required for power load (difference between tight-side and slack-side belt tensions).
2. Tension required to keep the belt tight, as indicated by the
maximum permissible ratio between the tight-side and slack-side tensions.

3. Tension due to centrifugal force.
4. Stresses due to bending the belt around the pulleys.

The horsepower transmitted by a belt is

\[
\text{hp} = \frac{S(T_1 - T_2)}{33,000}
\]

(4.1)

where \( T_1 \) = tight-side tension, in pounds (not including tension due to centrifugal force).

\( T_2 \) = slack-side tension, in pounds (not including tension due to centrifugal force).

\( S \) = belt speed, in feet per minute.

The allowable effective pull \((T_1 - T_2)\) for a given belt is related to the last three items listed above and to the allowable stress for the belt. The relation between the allowable stress and the ultimate belt strength is established primarily by the type of load (smooth, pulsating, high-shock, etc.) and the desired belt life in terms of stress cycles.

**4.3. Tension Ratio.** The maximum permissible ratio between the tight-side and slack-side tensions is

\[
R = \frac{T_1}{T_2} = e^{k\theta}
\]

(4.2)

where \( e \) = base of Naperian logarithms = 2.7183.

\( \theta \) = arc of contact, in radians.

\( k \) = coefficient of friction for a flat belt (for V-belts \( k \) also includes the wedging effect on the belt and the accompanying rather complex force reactions).\(^5\)

The coefficient of friction increases as the amount of slip increases, but slip should not be allowed to exceed \(1\frac{1}{2}\) to 2 per cent.\(^5\)

For rubber belts on steel or cast-iron pulleys, it is common practice\(^7\) to use values of \( k \) ranging from 0.25 to 0.30, which give corresponding values of \( R\) (tension ratio for 180° arc of contact) from 2.2 to 2.6. In designing a V-belt drive,\(^4\) a tension ratio of \( R = 5 \) is commonly assumed (\( k = 0.51 \)). Although considerably larger tension ratios may be satisfactory in regard to V-belt slippage, the drive is likely to become unstable because of low slack-
side tension if \( R \) is much greater than 8. If the tension ratio is smaller than the maximum permissible value, the capacity of the belt for a given allowable stress is decreased.

4.4. **Arc Correction Factor.** It is customary for manufacturers to rate belt drives on the basis of 180° arc of contact, with correction factors being applied to reduce the allowable load for smaller arcs of contact. This compensates for the reduced value of \( T_1/T_2 \) (equation 4.2) and the consequent reduction of effective pull for a given value of \( T_1 \). Arc correction factors are determined from the relation:

\[
C = \frac{(e^{k\theta} - 1)e^{k\pi}}{(e^{k\pi} - 1)e^{k\theta}}
\]  

(4.3)

Tables or curves showing arc correction factors are included in handbooks or catalogs of the various belt manufacturers, as well as in other references such as engineering handbooks and machine design textbooks.

4.5. **Centrifugal Tension.** The tension due to centrifugal force may be expressed as:

\[
T_c = 8.63 \times 10^{-8} w S^2
\]

(4.4)

where \( w \) = weight of belt, in pounds per foot of length. Centrifugal tension is the principal factor that limits the maximum speed at which a belt can be operated, but it may be neglected at low speeds. Most V-belt drives on agricultural machines operate at relatively low speeds (below 2000 fpm) so that centrifugal tension is seldom a limiting factor.*

4.6. **Bending Stresses.** For a given belt the extreme fiber stress due to bending around a pulley is inversely proportional to the pulley diameter. This stress is often neglected for large pulleys, particularly in the case of flat belts, but with small pulleys it reduces the allowable load considerably. The practice in rating industrial drive systems for either flat or V-belts has been to consider only the effect of the smallest pulley, but the design procedure for agricultural V-belt drives that is discussed in Section 4.11 considers the bending stresses at all pulleys.

* Personal correspondence from W. S. Worley of the Gates Rubber Company.
4.7. Service Factor. If a belt drive is likely to be subjected to pulsating loads or be occasionally overloaded, the normal required horsepower is multiplied by a service factor to obtain an appropriate increased design horsepower. Tables of service factors for various applications are included in manufacturers' catalogs.

4.8. Flat Belts. In general, flat belts are most advantageous on drives with long center distances. Their most common application on field machines is in drives involving relatively large power requirements, where a movable idler provides a clutch action by loosening or tightening the belt. In a clutching arrangement, flat belts tend to release more completely than V-belts in grooved sheaves. Rubber flat belts are generally preferred to leather belts, since they are more resistant to weathering.

Although power ratings of flat belts are subject to all the considerations described in Sections 4.2 through 4.7, the following general formula for rubber belts (based on 1 per cent slip) is sometimes used:

\[
\text{hp} = \frac{WPS}{2400} \tag{4.5}
\]

where \(W\) = belt width, in inches.

\(P\) = number of plies in belt.

This rating must be corrected for arcs of contact less than 180° and should also be reduced when pulley diameters are small (less than about 8 and 12 in. for 3-ply and 4-ply belts, respectively).

4.9. V-Belt Drives. Because of the wedging action between V-belts and sheaves, this type of drive can be operated with small arcs of contact, as in close-center shaft arrangements and serpentinite drives, and with rather large shaft-speed ratios. There is also more latitude in possible orientation and arrangement of component parts than with flat belts. V-belts are used either singly or in sets. Multiple belts must be carefully matched as to length in order to assure a uniform distribution of load between them. V-belts are adaptable to clutching arrangements, although a close-fitting guard may be needed to confine the belt and prevent bowing between the sheaves as the tension is released.

Under certain conditions it is convenient or economically de-
sirable to drive a relatively large flat pulley with V-belts from a smaller, grooved sheave (known as a V-flat drive). A sheave and a flat pulley are balanced in regard to power-transmitting capacity when the arc of contact is about $130^\circ$ for the sheave and $230^\circ$ for the flat pulley (equal values for $k\theta$ in equation 4.2, and hence equal values for the maximum permissible tension ratio $R$). In many cases a flat pulley is convenient or more economical, even though it may not be balanced with the small sheave in regard to power-transmitting ability and will therefore require more belt capacity or perhaps greater tension.

**4.10. ASAE V-Belt Standards.** Dimensional specifications for agricultural V-belts and sheaves have been standardized by the ASAE. These standards are included in Appendix A. Agricultural V-belts are distinguished from the corresponding cross-sectional sizes of industrial V-belts by the prefix H (sizes HA to HE). Agricultural *double* V-belts are employed when power must be transmitted to grooved sheaves from both the inside and outside of the belt (serpentine drives) and are designated as sizes HAA, HBB, HCC, and HDD.

The ASAE standards classify belts as to their effective length, based upon an effective outside sheave diameter at a specified groove width. A standard procedure for measuring effective length is prescribed. Thus the effective length is independent of the depth at which a particular manufacturer's belt rides in the groove of an ASAE standard sheave.

Included also are methods of determining required belt lengths, recommended allowances for installation and take-up (including the effects of belt stretch and wear), and information regarding idlers and twisted-belt drives.

**4.11. Capacities of Agricultural V-Belt Drives.** Manufacturers' ratings for industrial applications of V-belts are based upon a life expectancy of several years of continuous operation. Since agricultural belts are often used for only a few hundred hours per year, they can economically be subjected to considerably higher loads than industrial belts.

A system of designing V-belt drives for agricultural implements on the basis of peak stresses and fatigue life expressed in terms of stress cycles has been developed by the Gates Rubber Company. In this method, selection of belts and sheaves or flat pulleys is based upon the geometry and actual service requirements of the
The following factors are considered:

1. The number of pulleys (grooved or flat) on the drive.
2. The effect of each pulley's diameter on the stress.
3. Belt speed.
4. Belt length.
5. The load to be carried by each pulley and the sequence of loaded pulleys on the drive.
6. The arc of contact at each loaded pulley.
7. The location of idlers.

Each pulley on the drive causes a peak stress due to bending of the belt, as illustrated in Fig. 4.1. In designing the system,

\[ T_1 = \text{Tight-side stress} \]
\[ T_2 = \text{Slack-side stress} \]
\[ T_{br} = \text{Stress due to bending over driver wheel} \]
\[ T_{bn} = \text{Stress due to bending over driven wheel} \]
\[ T_c = \text{Centrifugal stress} \]

![Diagram of belt stress](image)

**Fig. 4.1. Belt stresses in relation to position on drive. (W. S. Worley, *Agr. Eng.*, July, 1950.)**

the mean peak stress is used, rather than considering only the stress at the smallest sheave. Tests have indicated that for a given size of belt the relation between mean peak stress and fatigue life may be represented by a curve of the general shape...
For a given number of stress cycles the life of a belt, in terms of operating time, is inversely proportional to the belt speed and the number of pulleys, and directly proportional to the length of the belt. These factors are considered in the Gates design system.

By means of appropriate charts, the Gates system can be used to design a drive for any desired arrangement of components. If the loads to which the drive on the machine will be subjected are accurately known, the drive can be designed for a particular service life, or the life expectancy of the belt on a particular arrangement can be predicted with a reasonable degree of accuracy. If, as is often true in drives on agricultural machinery, the requirements are not known accurately, the probable effect on the service life of any difference between the assumed design load and actual loads experienced can be determined by means of the Gates system.

For example, consider a 5-in. P.D. (pitch diameter) sheave at 2200 rpm driving a 10-in. sheave through an HB-section belt with 20-in. shaft centers and a service factor of 1.00. If the actual load were 5 1/2 hp, the expected service life, according to calculations based on the Gates system, would be approximately 1200 hr. If the requirement turned out to be 6 1/2 hp, the life would be reduced to 400 hr, whereas at 3 1/2 hp the belt would be expected to last approximately 20,000 hr.
VARIABLE-SPEED DRIVES

Both hydraulic and V-belt systems have been employed in farm machinery to vary the speed ratio between two shafts. The term, "variable-speed" implies the ability to change the speed ratio by small increments over the entire range of control while the drive is under load, as contrasted to the selection of specific speed ratios by means of gears, step pulleys, and the like. The need for independent control of ground speed on self-propelled harvesting machines has provided the incentive for considerable development work on variable-speed drive systems capable of transmitting relatively large amounts of power.

4.12. Variable-Speed V-Belt Drives. In comparison with hydraulic systems, the V-belt system has the advantage of relative simplicity, but speed-range ratios are more limited. With variable-speed sheaves and V-belts that conform to the ASAE standards, maximum speed-range ratios of about 3\(\frac{1}{2}\) to 1 are obtainable when two sheaves of the minimum allowable diameter are used.\(^2\) The range for a given belt size varies inversely as the sheave diameter, since the maximum change in pitch diameter is fixed by the groove angle (26° is the ASAE specification) and the belt top width.

Belts designed specifically for variable-speed drives are wider than standard V-belts in relation to their thickness. The extra width is necessary to obtain reasonable ranges of speed ratio as well as increased load capacities; relatively thin belts are needed because minimum operating diameters are necessarily small in this type of drive. Also, the thickness subtracts from the total in-and-out movement of the belt for a given depth of sheave face.

If a single, variable-pitch sheave is used in combination with a fixed-diameter sheave, speed adjustment may be accomplished by (a) providing a thrust spring on the variable-pitch sheave and varying the shaft center distance, (b) providing a thrust spring and moving an idler, or (c) using a spring-loaded idler and moving the adjustable sheave face with a mechanical linkage. With a single variable-pitch sheave the belt will be in alignment for only one speed ratio, a condition that becomes increasingly serious for short center distances.

The speed-range ratio for a combination of two variable-pitch
sheaves is the product of the two individual ranges. Two possible arrangements are indicated in Fig. 4.3. If the faces $A_1$ and $B_2$ in the left-hand arrangement are fixed axially while $A_2$ and $B_1$ are moved simultaneously, proper belt alignment can be maintained at all speed ratios. The arrangement shown at the right is subject to belt misalignment as discussed in the preceding paragraph. With this system, the adjustable sheave must be moved along a path that will keep the required lengths of the two belts constant as the floating center changes its lateral position.

Worley\(^5\) demonstrates that for equal axial movements of the two adjustable sheave faces on a fixed-center system the required belt lengths will not be the same for different sheave adjustments. This condition is aggravated by short center distances and unequal sheave diameters. Thus, if $A_2$ and $B_1$ in Fig. 4.3a are moved by an interconnecting mechanical linkage, a spring should be provided in the linkage system between the two sheaves to maintain proper belt tension.

It is sometimes desirable to incorporate a power-disengaging device in the variable V-belt drive, thereby eliminating the need for a separate clutch. This can be accomplished by mounting a

---

**Fig. 4.3.** (a) Arrangement with two variable-pitch sheaves on fixed centers. (b) Variable-pitch double sheave with floating center. (E. G. Kimmich and W. Q. Roesler. *Agr. Eng.*, July, 1950.)
flat idler inside of the sheave, between the adjustable faces, so that the belt will ride on the idler when the sheave is opened to its maximum width. A similar clutching arrangement can be built into a nonadjustable V-belt sheave by having one face movable to release the belt.

4.13. Variable-Speed Hydraulic Drives. Wolf lists five types of hydraulic drives that could be used as variable-speed units for the propulsion of combines, the most widely accepted one being the Thomas Vari-draulic unit. This drive unit is basically a rotating fluid coupling with a multiple-gear pump built into it to provide the variable-speed feature. Within the sealed, rotating housing is a planetary gear arrangement, with three planet gears and the housing comprising the input unit. The sun gear is keyed to the output shaft. Any difference between the speeds of the input and output shafts causes relative rotation of the sun and planet gears, which then act as a multiple-gear pump within the housing.

An axial-moving control valve can be manipulated by the operator to feed a controlled variable mixture of air and oil (ranging from all air to all oil) through the gear pump. With all air, the planet gears merely coast around the sun gear and the output shaft does not rotate. As the proportion of oil in the pumped mixture is increased, the drag of the planet gears upon the sun gear increases, thus increasing the output torque in the lower range. When the all-oil position of valve travel has been reached, further movement of the control begins to restrict the discharge flow of the oil, thus further restraining the rotation of the pump gears and producing higher torques. At the extreme high-speed position of the control, the pump discharge is completely closed, permitting practically no relative rotation of the gears and producing a substantially positive drive.

This type of drive system does not multiply torque. It is basically a slipping device, with the efficiency varying from virtually 100 per cent at no slip, down to zero at full slip. It does have the advantage of providing a wide range of speeds, and it introduces a cushioning effect into the drive. Although this cushioning effect tends to protect the engine and drive system from shock loads, it has the disadvantage of permitting the output speed to vary considerably if the torque requirements change (for example,
when a combine is propelled over a ridge or changing grades or over varying soil conditions).

Another type of hydraulic drive that can undoubtedly be used to advantage in some farm machinery applications is a combination consisting essentially of a positive-displacement pump and a hydraulic motor. The pump and motor are joined through oil lines, with suitable valves and controls provided to permit changing the oil flow rate and pressure to the motor in order to vary the speed and torque. This means of providing output-shaft speed variations or flexibility in the power transmission system is widely used in other fields, but its relatively high initial cost has limited its adaptability to farm machinery. The efficiency of this type of drive would be somewhat lower than for a well-designed mechanical power transmission system.

GEARS AND CHAIN DRIVES

Both of these types of drive give constant speed ratios, gears being adapted to close-center, intersecting, or crossing shafts, and chain drives being used for parallel shafts with moderate center distances. Gears are generally enclosed so that they can be protected and properly lubricated, whereas chain drives on agricultural machinery frequently operate in the open with periodic oiling or sometimes with no lubrication at all on slow-speed chains in a dusty or dirty environment. Since gear applications on farm machinery are in general very similar to industrial applications, and since detailed design and selection information is available from many standard sources, no further discussion on this subject will be included here.

4.14. General Considerations. Chain size is specified by pitch, which is the effective length of one link. Standard-pitch roller chain, double-pitch roller chain, and detachable-link chain (Fig. 4.4) are all common on agricultural machines. Standardized dimensions for each of these types have been adopted by the American Standards Association. Pintle chain has been used to some extent, principally in applications requiring the transmission of large amounts of power at extremely low speeds. For a given pitch, it is several times as strong as detachable-link chain, but is also considerably heavier and more expensive.
Since a sprocket is essentially a polygon with as many sides as there are teeth or pitches, either the chain speed or the angular velocity of the sprocket must vary as the chain engages or leaves the sprocket. The fewer teeth there are on the sprocket, the greater is the speed variation. Theoretically, a 10-tooth sprocket would give a variation of about 5 per cent. Practically, however, small speed variations as well as sudden load shocks tend to be absorbed or cushioned by the natural elasticity of the chain and the catenary effect of the driving side. Although sprockets with as few as six teeth are available, sizes with less than 17 or 18 teeth are not recommended for high-speed drives.

In general, the load capacity of a chain is based upon the rate of wear rather than the ultimate strength and is a function of the number of teeth on the small sprocket and its rotative speed. For extremely slow drives, however, the chain selection may be based upon the ultimate strength, using safety factors ranging from 5
at 25 fpm to 10 at 250 fpm.\(^7\) As a chain wears, the pitch length increases and the chain rides farther out on the sprocket teeth. The greater the number of teeth on a sprocket, the sooner the chain will ride out too far on the teeth. For this reason, speed ratios greater than about 8 to 1 are not generally recommended.\(^7\) Since chain ratings published in the various catalogs are for the relatively long life expected in industrial applications, somewhat greater loading can often be assumed in designing agricultural drives. However, unfavorable environmental conditions may tend to shorten the life.

4.15. **Standard-Pitch Roller Chain.** Drives of this type are satisfactory at linear speeds from less than 100 fpm up to 4000 or 4500 fpm\(^{12}\) and are well suited for heavy loads requiring a compact drive. The maximum permissible speed decreases as the pitch is increased. For extremely compact drives at high speeds, multiple-width chains of short pitch can be used. Roller chains are precision built and under favorable conditions may have efficiencies as high as 98 to 99 per cent.\(^7\) Chain drives do not require initial tension, and sprockets may be driven from either the inside or the outside of a roller chain. Although oil-bath lubrication is recommended for high-speed drives, this system is often not practical on agricultural machines.

4.16. **Double-Pitch Roller Chain.** This type of chain employs the same pins, bushings, and rollers as standard-pitch roller chain, but the side plates have twice the pitch. Thus, double-pitch chains are lighter in weight and less expensive, although they have the same precision and strength as corresponding standard-pitch chains. They are suitable for drives where the small-sprocket speed does not exceed about 600 rpm.\(^{12}\) Because the roller diameter is only five-sixteenths of the pitch, there is ample space for sprocket teeth, and cast-tooth sprockets are satisfactory (with resultant savings in initial cost as compared with machine-cut teeth).

4.17. **Detachable-Link Chain.** Malleable-cast-iron or pressed-steel detachable-link chains are used for moderate loads at speeds not exceeding 400 to 500 fpm, where low initial cost is important. They are especially well suited for conveyor and elevator work. Under dirty conditions they are subject to greater wear than roller chain, because of the loose-fitting, open hooks. Pressed-steel chain has a lower initial cost than malleable-cast-
iron chain, but malleable cast-iron chain is more resistant to corrosion, and pattern equipment for special attachment links is less expensive.  

SHAFTING AND UNIVERSAL JOINTS

4.18. Shafts. Shafts may be subjected to torsion, bending, axial compression or tension, or to various combinations of these loads. A detailed design procedure would include consideration of the effects of each of these loadings, the nature of the service conditions, the shaft material, and the effect of keyways or other shaft irregularities. However, approximate horsepower ratings for solid circular shafts can be obtained from the equation

\[
hp = \frac{D^3N}{K}
\]

where \( D \) = shaft diameter, in inches.  
\( N \) = shaft speed, in revolutions per minute.  
\( K \) = a constant whose value depends upon the nature of the loading, the shaft material, etc.

For cold-rolled steel shafting in typical installations, \( K \) may be considered as ranging from 75 for shafts with small bending moments (pulleys and sprockets close to bearings) up to about 125 for shafts with relatively large bending moments.

4.19. Universal Joints. The type of universal joint most common on farm machines is the Hooke joint, which is illustrated in Fig. 4.5. The bearings and trunnions can be much smaller when roller bearings are used rather than plain bearings. Roller-bearing joints operating at angles less than 15° have an efficiency of over 99 per cent, run cool, and require less frequent lubrication than plain-bearing joints. Plain-bearing joints may have efficiencies as low as 60 per cent. Roller-bearing joints are also preferable for steering-gear applications because of their small backlash.

A single Hooke universal joint operating at an angle will not deliver uniform angular velocity. Figure 4.6 shows the lead or lag of the driven shaft for one-half revolution, with the universal operating at various joint angles \( \alpha \). At a joint angle of 10° the
rotational fluctuation is small, but at 40° the total fluctuation for a single joint is about 15°. Belts or a long slender shaft in the drive will help absorb these angular fluctuations.

Fig. 4.5. The Hooke universal joint. Roller bearings are commonly used. The ends of the trunnions of the center cross and the bottoms of the bearing caps are precision ground, thereby forming thrust bearings. (Fred M. Potgiester. Agr. Eng., January, 1952.)

The curves in Fig. 4.6 are based on the well-known equation

\[
\frac{\tan Y_2}{\tan Y_1} = \cos \alpha
\]

where \( Y_1 \) = angle of rotation of the driving shaft from the initial position shown at the upper left in Fig. 4.6.

\( Y_2 \) = corresponding angle of rotation of the driven shaft \( (Y_2 - Y_1 = \text{lead}) \).

\( \alpha \) = joint angle (Fig. 4.6).

The peak lag or lead of the driven shaft occurs when the average angle of rotation of the two shafts is 45° or 135° from the indicated reference position. Thus the peak lag occurs when the driving shaft is past 45° by an amount equal to half of the peak lag, as indicated by the broken line in the lower half of Fig. 4.6.

4.20. Universal-Joint Combinations. In many applications, universal joints are used in pairs. If (a) the three involved shafts are all in one plane, (b) the two joints have equal angles, and (c) the yokes on the connecting shaft are in line, then the lead
or lag curves for the two joints have the same magnitudes but are 90° out of phase. Thus, the angular fluctuations of the two joints cancel and the final shaft has uniform rotation. If the shafts connected to one of a pair of joints are in a different plane than the shafts connected to the other, the yokes on the intermediate shaft should be placed out of line by an amount equal to the angle between the two planes (i.e., so that these two yokes will lie in the two planes simultaneously).

When three or more universal joints are used in series, the problem of orienting the shafts and joints to produce uniform angular velocity in the final driven shaft is rather complex. However, Berry and Callum have devised a method for determining the proper setup of multijoint systems by analytical and graphical means.

The maximum permissible angle between shafts depends upon the type of drive, the flexibility of the rotating parts, the inertia of the rotating parts, the speed, and the expected life. Potgieter suggests a maximum of 45° per joint when operated in pairs and

![Diagram showing lead or lag of shaft driven by Hooke universal joint, in relation to the position of the driving shaft.](image-url)
15 to 20° for single joints. For continuous operation in a drive system, these maximum values should be reduced somewhat.

4.21. Universal-Joint Shafts. Provision for axial movement of the shafts in a drive containing universal joints is important. This is usually provided by telescoping shafts and tubes or by splined connections. If the sliding connection is between two joints, the telescoping parts should be keyed in some manner to insure assembly with the correct orientation of the yokes.

Because a universal joint operating at an angle and transmitting torque tends to eliminate its angle and bring the two shafts in line, bending moments are set up in the shafts.11 These bending moments vary with the angle of rotation of the shaft, two stress reversals and two peaks occurring in each revolution. The peak bending moment in each of the shafts connected to a single joint is equal to the product of the torque being transmitted multiplied by the tangent of the joint angle.11 To avoid excessive vibration, the weight of the shaft connecting two universal joints should be kept at a minimum. Therefore the connecting shaft is usually a tubular member.

BEARINGS, SEALS, AND LUBRICATION

Bearings on agricultural implements are often required to operate under extremely dirty conditions. One of the major problems with many types of field machines has been the exclusion of dirt and the application of suitable lubrication procedures to reduce friction and prolong the life of bearings. Although rapid strides have been made in regard to bearing applications and seals, the increasing complexity of farm machines is such that the farmer must still "waste" a great deal of time in daily (or even more frequent) lubrication of many rotating parts.

Self-alignment of bearings is an important feature in many agricultural applications, to compensate for manufacturing tolerances and variable deflection of the supporting members. Ball bearings are available with this feature built into their housings. Other types of bearings may be made self-aligning by proper support of the housing.

4.22. Sliding-Contact (Plain) Bearings. Proper design calls for adequate contact area, dirt exclusion, and proper lubrication. The usual lubricating method for plain bearings in farm machin-
ery is by intermittent application of grease or oil. This results in so-called "thin-film" lubrication, in which there is some contact between the metal surfaces. A conservative range for the coefficient of friction is from 0.12 to 0.15. In some instances, as in certain types of disk harrows, plain bearings operate in an oil bath (see Fig. 8.9c), which gives "thick-film" lubrication with no metal-to-metal contact and considerably lower coefficients of friction.

The most common materials for plain bearings include oil-soaked wood (such as maple), chilled cast iron, bronze, and babbit. Other materials suitable for specialized applications in farm machinery include gray iron, hardened steel, sintered metals (oil-impregnated), nylon, etc. Plain bearings may be designed to take end thrust by means of a thrust collar on the shaft, pressing against the end of the sleeve or fitting into an annular groove in the sleeve.

4.23. Rolling-Contact (Antifriction) Bearings. Although the coefficient of friction for rolling-contact bearings is low (less than 0.01) as compared with that for sliding-contact bearings, probably the most important advantage in farm machinery applications is the improved performance of seals and the consequent reduction of lubrication problems because of the close fits and the absence of bearing wear. In spite of the more complicated shaft and housing design and the higher first cost, antifriction bearings are rapidly replacing sleeve bearings. Lubrication is either by means of an oil bath in an enclosed housing or by grease injected into a scaled bearing.

Antifriction bearings are available for almost any type of application. Although radial-type ball bearings (see Fig. 8.8) are designed primarily for radial loads, they can take a moderate amount of thrust. Some ball bearings are designed specifically for thrust loads or for combination loading (see Fig. 8.2b).

Plain roller bearings have a high radial load capacity but no end-thrust capacity. They are most suitable for low-speed applications, such as on the axle of a grain drill. They are more sensitive than ball bearings in regard to misalignment, dirt, and grit. A needle bearing is a form of roller bearing employing a maximum number of small-diameter rollers without a cage or spacers. Because of their comparatively small radial dimensions and their exceptionally high load capacity, particularly at low peripheral
speeds, needle bearings are well suited to applications involving oscillating motion. They may be installed without a race, with only an outer race, or with both inner and outer races (Fig. 4.7). When the shaft (or the housing) is to serve as a race, it should be hardened and precision-ground to close tolerances. As with other types of straight roller bearings, needle bearings are very sensitive to dirt and misalignment.

Fig. 4.7. Common types of needle bearings. (a) and (b) bearings with outer races only, (c) bearing with inner and outer races, (d) spherical-end roller used in transmissions, etc. (The Torrington Co.)

Tapered roller bearings are designed to take both radial and thrust loads, the relative capacities depending upon the amount of taper. These bearings are used in pairs, as illustrated in Fig. 8.2a (Chapter 8), with axial adjustment usually provided for the inner race. In addition to being adjustable, tapered roller bearings are more tolerant of dirt than are other types of antifriction bearings. They have many applications in farm machinery design.

4.24. Load Capacities of Antifriction Bearings. Manufacturers publish tables that give the load ratings for different bearings at various speeds, based upon a specified average life. These data can be adjusted to give ratings for other life expectancies by means of factors included in the catalogs. In general, the life of an antifriction bearing, in terms of stress cycles, appears to vary about as the inverse cube of the load. Since these bearings fail primarily because of fatigue, the life in hours is inversely proportional to the rotational speed. In many slow-speed and heavy-duty farm machinery applications, the size of the bearing
is determined by the size of shaft needed to carry the load rather than by the load limitations of the bearing.

4.25. Seals. In the early development of the farm machinery industry, when antifriction bearings were seldom used, seals were either inadequate or were not present at all, an excess of lubricant being used to “flood out” the dirt from the bearings. The current trend toward antifriction bearings in farm machinery calls for the inclusion of adequate seals, which depend upon accurate fits and require relatively light contact pressures. Leather and synthetic rubber are probably the most common sealing materials on farm machinery, although felt, cork and other types are used in certain applications.

Applications of sealed, factory-lubricated, ball bearings have been rather limited in the past but are now increasing rapidly, particularly on machines having a large number of bearings (such as a grain combine). These bearings may have no provision whatever for relubrication, or they may have a fitting to permit occasional replenishment of the lubricant if needed (perhaps annually). Their additional cost on complex machines tends to be offset by the reduction in maintenance and lubrication time for the farmer.

OVERLOAD SAFETY DEVICES

In many types of farm machinery a single power source drives various components that have widely differing power requirements and are subject to varying degrees of possible overload. In such a system some overload protection is almost mandatory, especially for the lower-powered components. Three general types of safety devices are commonly used in rotary drives: (a) those that depend upon shearing of a replaceable connecting link in the drive, (b) units in which spring force holds two corrugated members together, utilizing the principle of the inclined plane, and (c) devices depending entirely upon friction. A fourth and less common type drives through an adjustable, spring-loaded trip on one rotating member. The trip must be reset after each overload.

4.26. Shear Devices. This type of device is simple and relatively inexpensive, but the sheared element must be replaced after each overload. Thus, it is most suitable where overloads are
Shear devices can be designed for almost any desired load rating, although pin or key sizes become rather small for low torque ratings unless a material with low shear strength is selected. Typical arrangements for shear devices are:

1. Shear key between shaft and hub (usually brass key, with tapered shaft and bore).
2. Diametral shear pin through hub and shaft (gives double shear).
3. Flange-mounted shear pin parallel to shaft, as illustrated in Fig. 4.8.

Regardless of the arrangement, the driving and driven members must rotate freely with respect to each other after the shear element has failed. With either of the first two types, the shaft or bore is likely to be scored by the sheared element. Removal of the hub from the shaft is necessary in the first type to replace a sheared key.

The flange-mounted shear pin is most easily replaced, but the unit is more costly than a diametral pin and not as well adapted to low torques because of the greater radius to the shear section. For experimental testing, interchangeable pins necked down to
different diameters can be used to vary the load at which failure occurs. In production units a full-sized pin of an ordinary material (such as hot-rolled steel) is desirable for convenience of replacement by the operator.

The torque at which a flange-mounted shear pin will fail is

\[ T' = r_s \left( \frac{1}{4} \pi d_1^2 S_s \right) \]  

(4.8)

and the horsepower is

\[ \text{hp} = \frac{2 \pi N T'}{12 \times 33,000} = \frac{N r_s d_1^2 S_s}{80,250} \]  

(4.9)

where \( T \) = torque, in pound-inches.

\( N \) = shaft speed, in revolutions per minute.

\( r_s \) = distance between shaft center and shear-pin center (Fig. 4.8), in inches.

\( d_1 \) = diameter of shear pin at shear section, in inches.

\( S_s \) = ultimate shear strength of shear pin, in pounds per square inch.

Similarly, a diametral shear pin (double shear) will fail when

\[ \text{hp} = \frac{N D d_1^2 S_s}{80,250} \]  

(4.10)

where \( D \) = shaft diameter (diameter at which shear occurs), in inches.

**4.27. Safety Jump Clutch.** The names “safety clutch” and “slip clutch,” by which this device is commonly known, do not distinguish it from a friction-type safety clutch. Thus the term “jump clutch,” which is descriptive of its action, will be used in this textbook.

The principle of the jump clutch is illustrated in Figs. 4.9 and 4.10. In this example, part \( A \) is keyed to the shaft, and part \( B \) is held against part \( A \) by means of the adjustable spring. The driving sprocket is attached to part \( B \). Consider the condition of impending slip, and assume that the forces acting on part \( B \) (Fig. 4.10) are all concentrated on the inclined face of one corrugation at the mean radius of the contact area. Identification of the symbols is as follows:

\( r \) = mean radius of contact area between parts \( A \) and \( B \), in inches.
\[ F_s = \text{axial spring force plus the axial friction force required to slide part } B \text{ along the shaft, in pounds.} \]
\[ F_t = \text{force on part } B \text{ at radius } r, \text{ due to torque being transmitted, in pounds.} \]
\[ F'_b = \text{reaction of part } A \text{ on part } B, \text{ in pounds.} \]
\[ \phi = \text{angle of static friction (} \tan \phi = \text{coefficient of static friction), in degrees.} \]
\[ \beta = \text{slope of inclined surfaces with respect to the plane of rotation, in degrees.} \]

From the vector diagram it is evident that at impending slip

\[ F_t = F_s \tan (\beta + \phi) \tag{4.11} \]

The torque transmitted is \( r \times F_t \), and the horsepower at impending slip is

\[ \text{hp} = \frac{2\pi NrF_s \tan (\beta + \phi)}{12 \times 33,000} = \frac{NrF_s \tan (\beta + \phi)}{63,020} \tag{4.12} \]

As slip occurs along the inclined surfaces, causing part \( B \) to move axially with respect to \( A \), \( F_s \) will increase due to greater compression of the spring, and \( \phi \) will tend to become sliding rather than static friction, with a slightly lower value. The increase in \( F_s \) generally predominates over the change in \( \phi \), causing the horsepower at impending jump (i.e., maximum torque) to be greater than the horsepower at impending slip. The spring must have sufficient deflection available so that it is not com-
pressed solid before the relative axial displacement of the two corrugated faces is sufficient to permit jumping.

Note from equation 4.12 that the action of the jump clutch is influenced by both the slope of the inclined surfaces and the angle or coefficient of friction, but that the unit would function (at a lower horsepower) even if the coefficient of friction were zero. From trigonometric tables it can be shown that the percentage change of the tangent per degree change of angle is a minimum at an angle of 45°. Thus, it follows that a jump clutch will be least affected by variations in the coefficient of friction when \( \beta + \phi = 45° \). However, the percentage change does not increase very rapidly until the angle exceeds about 60°. Therefore, since the required spring force at 60° is only 58 per cent of the force at 45° for a given horsepower (from equation 4.12), the 60° value for \( \beta + \phi \) gives the better design in this respect. Ideally, the materials and finish of the contact faces should be such as to give a low coefficient of friction or at least one that remains fairly constant.

Because of its automatic resetting feature, the jump clutch is especially suitable where overloads may occur rather frequently. There is no slippage until the load exceeds the setting of the unit, and then the operator may be warned audibly that an overload has occurred. Jump clutches are more expensive than shear devices and are not well suited to large loads because of the excessive physical size required. When they are jumping, they impose high shock loads upon the drive system.

An important factor often overlooked in the design of a jump clutch is the magnitude and effect of the friction force required to slide the movable member (part B in Fig. 4.9) along the shaft. Assume, for example, that part B is driven through a splined section or telescoping shaft whose effective radius is half of the mean clutch contact radius \( r \). Since the torque to be transmitted is \( rF_t \), the tangential contact force at the splines would then be 2 \( \times F_t \). The resulting axial friction force, particularly if the sliding joint is not lubricated, could represent a large part of the total required axial force \( F_a \). Under these conditions, variations in the coefficient of friction would introduce a considerable degree of variability in the torque required for impending jump. For most reliable operation of a jump clutch, the driving torque should be applied to the movable member B on a relatively large
radius and lubrication should be provided for the axial movement, as illustrated in Fig. 4.9.

4.28. Friction Devices. A properly designed belt drive may serve as a friction safety device, although its performance is affected by variations in belt tension and by the increase in coefficient of friction as the percentage of belt slip increases. The performance is more consistent with a spring-loaded idler than with a fixed adjustment.

Single-plate clutches with two friction surfaces, similar to tractor or automotive clutches, are sometimes used as safety devices. The horsepower transmitted by a single-plate clutch (two contact surfaces) is

\[
\text{hp} = \frac{2(2\pi Nr)F_{sh} \mu}{12 \times 33,000} = \frac{Nr F_{sh} \mu}{31,510}
\]

(4.13)

where \( \mu \) = coefficient of friction.

Friction clutches, like belts, tend to be somewhat unreliable as safety devices because of variations in the coefficient of friction. It is possible to have a friction clutch slipping sufficiently to become overheated without the operator being aware of the overload. Friction clutches, however, do not introduce shock loads and are very effective in protecting a drive from high-frequency peak torques, as discussed in Chapter 5 (Section 5.21).

REFERENCES


13. V-Belt Drives for Farm Machines. ASAE Standard. ASAE, Saint Joseph, Michigan, 1950. (See Appendix A.)


PROBLEMS

4.1. (a) Derive equations for belt tensions \( T_1 \) and \( T_2 \) in terms of \( R \), hp, and \( S \), using equations 4.1 and 4.2.

(b) Substitute \( R = 5 \) in the derived equations to obtain practical working equations for V-belts having 180° arc of contact.

(c) For flat-belt equations, substitute \( R = 2 \) (180° arc of contact).

4.2. A V-belt is to transmit 6 hp at a belt speed of 3200 fpm.

(a) Calculate \( T_1 \) and \( T_2 \), assuming \( R = 5 \).

(b) Calculate centrifugal tension \( T_c \), assuming an HB belt with 38° included angle and a density of 0.045 lb per cu in. (Refer to ASAE standard in Appendix A for belt dimensions.)

4.3. The driver and driven pulleys of a flat-belt drive have diameters of 6 in. and 37 in., respectively, and are on 34-in. centers. An 8-in. idler has its center 11 in. from the driver axis and 26 in. from the driven axis. The drive is to transmit 22 hp at a driver speed of 2300 rpm.

(a) Determine belt length and arcs of contact by graphical means.

(b) Select a whole-inch width of 3-ply rubber belt, using equation 4.5 and applying a calculated arc correction factor (take \( k = 0.25 \)) and a factor of 0.9 to compensate for the small driver diameter.

(c) Calculate required tensions \( T_1 \) and \( T_2 \).

(d) Select appropriate sizes for the two shafts, assuming \( K = 90 \) in equation 4.6.

4.4. A V-belt drive has two sheaves with effective outside diameters of 4.9 and 13.7 in. One shaft is to be movable for take-up, with the desired center distance being about 18 in.
4.5. Lay out the drive system shown in Fig. 4.3b., using an enlarged scale, and determine graphically the path of the adjustable-sheave shaft center that will keep the required belt lengths constant.

4.6. (a) Compute the theoretical percentage variation in speed of a chain as it leaves an 8-tooth sprocket rotating at a uniform speed.
(b) Repeat for an 18-tooth sprocket (which is about the minimum size normally recommended for high speeds).

4.7. The two universal joints of a pair are operating at joint angles of 35° and 22°, respectively. The driving shaft, connecting shaft, and driven shaft are all in the same plane, and the yokes on the two ends of the connecting shaft are in line.
(a) Calculate the lead or lag in each joint for each 15° increment of rotation of the driving shaft, from 0 to 90°.
(b) Plot lead or lag versus degrees rotation of the driving shaft, showing a curve for each joint and one for the system. On each curve, indicate where the peak occurs.
(c) What change might be made in this drive system to provide uniform rotation of the driven shaft?

4.8. (a) Specify the location of a 3/8-in. flange-mounted steel shear pin ($S_s = 45,000$ psi) which will fail at 6 hp and 650 rpm.
(b) What size of diametral shear pin would be required if the shaft diameter is 1 in.?

4.9. The jump clutch in a power-mower drive operates at 535 rpm. $\beta = 38^\circ$, and $r = 1\frac{1}{2}$ in. The coefficients of friction are 0.40, static, and 0.32, sliding. Impending jump (maximum torque) is when 1/8 in. of axial spreading has occurred. Spring constant = 850 lb per in.
(a) Determine required axial force $F_a$ at impending jump if the clutch is to jump at 9 hp.
(b) Determine the corresponding hp and $F_a$ for impending slip (no axial displacement). Neglect the effect of reduced axial friction force.
(c) If the movable member of the jump clutch is attached to one part of a telescoping shaft and the effective contact radius between the two telescoping parts is 1 1/8 in., what would be the magnitude of the axial friction force at impending jump? What percentage is this of the total required $F_a$? Assume the telescoping joint is well lubricated, with a coefficient of friction of 0.15.

4.10. If a single-plate clutch with $r = 3$ in. and $\mu = 0.45$ is to be used in place of the jump clutch in Problem 4.9, what total axial force $F_a$ will be required?
5.1. Hydraulic Controls. Mechanical power lifts on tractors were introduced commercially in the late 1920's, following the advent of mounted cultivators and the general-purpose tractor. Hydraulic lifts for mounted implements appeared a few years later, and in the early 1940's hydraulic controls for trailed implements were adopted. Since that time, the development and improvement of hydraulic control systems has been rapid, and this method of control is now used extensively on all types of farm machinery (trailed, mounted, and self-propelled). Electric and pneumatic power lifts have been employed only to a limited extent.

Hydraulic control systems, in addition to providing conveniently located and easily operated control levers, are extremely flexible in regard to application possibilities. Cylinders may be built into the tractor, or external cylinders may be mounted in various positions on the tractor or on trailed implements. Depth can easily and safely be changed while the tractor is moving, and the speed of raising or lowering is adjustable. Cylinders may be single-acting or double-acting, and multiple-control systems with two or more cylinders may be operated from a single pump.

The essential elements of a hydraulic control system include a reservoir, a pump, a control valve or valves, one or more power cylinders, and a relief valve. The entire system, including remote cylinders used on trailed implements, is considered to be part of the tractor. Nevertheless, it is important that the implement designer be familiar with the various control systems in order to utilize them to best advantage, and he must work with the tractor engineer to obtain a balanced system of implements.
and controls. In the case of self-propelled machines, the implement designer has the full responsibility for any hydraulic controls employed.

5.2. Types of Hydraulic Control Systems. Hydraulic controls for agricultural implements are of two general types that may be described as limit-control systems and proportioning-control systems. In a limit-control system, the implement is raised or lowered by moving the control lever to either side of its neutral position. The piston in the power cylinder continues to move until the lever is returned to neutral or until the travel is limited by a stop or by the piston reaching the end of its full stroke. Depth is controlled by adjusting the position of a stop on the cylinder or on the connected linkage or by quickly returning the lever to neutral when visual observation indicates that the desired depth has been reached.

Proportioning-control systems may be subdivided into automatic position control and automatic draft control systems. In an automatic position control system (also known as the follow-up system) there is a definite relation between the position or depth of the implement and the position of the control lever in its quadrant. Thus the operator can preselect and identify an implement position by the location of the hand lever, and the power cylinder will automatically move the implement to the corresponding position or depth. Automatic draft control is similar to position control except that the position of the hand lever represents a given implement draft rather than a particular depth. Variations in draft cause the implement to be automatically raised or lowered as required to restore the draft to the preselected value.

5.3. Pumps and Basic Oil Circuits. Rated oil pressures in early hydraulic systems were sometimes as low as 300 psi. Pressures as high as 3000 psi are now used to a limited extent, but the present trend is toward rated pressures in the order of 800 to 1200 psi. Relief valves (necessary because pumps have positive displacement) are normally set about 25 per cent higher than the rated pressure. Multiple-cylinder, piston pumps are generally employed when rated pressures are greater than about 1500 psi, whereas gear pumps are most common for pressures below this value. Vane-type pumps are also used to some extent.

To allow the pump to operate at low pressure and low load
when no movement of the piston is required, one of the three following arrangements is generally employed:

1. **Open-center valve.** When the control valve is in the neutral position, an open bypass is provided from the pump back to the reservoir.

2. **Pilot-controlled bypass valve.** When all control valves of a multiple-control system (or the one valve of a single-control system) are in neutral, a remote regulator valve opens up a free passage from the pump to the reservoir.

3. **Blocked pump inlet.** When no movement of the piston is called for, the control valve blocks off the suction line between the reservoir and the pump, thus causing the pump to operate at low pressure with no oil flow.

**5.4. Cylinder Types and Applications.** Mounted implements may be controlled through either single-acting or double-acting cylinders, single-acting cylinders being most common for built-in systems. With single-acting cylinders the rate of downward movement of the implement is a function of the implement weight and inertia, the magnitude of vertical components of soil forces, and the resistance to flow of the oil being expelled from the cylinder. A double-acting cylinder, with its positive action in either direction, gives less variation in dropping time than does the single-acting cylinder. With a single-acting cylinder, an implement may be allowed to "float" (i.e., seek its own depth or position) by merely locking the return valve open. The same effect can be obtained with either type of cylinder without requiring the piston to float, by proper design of the linkage.

Some trailed implements, such as disk harrows, grain drills, land levelers, and others, require that the controlling device be able to exert force in either direction. Consequently, most cylinders intended for the control of trailed implements are of the double-acting type. Provision is often made to mount the same cylinder on trailed implements and on the tractor for the control of mounted implements.

**5.5. Single-Cylinder Systems.** Two types of control systems for single-acting cylinders are illustrated in Fig. 5.1. The control valve and other parts of system (a) are submerged in the oil reservoir. When the control valve is in neutral (as shown), the oil in the power cylinder cannot escape and the inlet to the pump
is blocked. The pump then merely cavitates without pumping any oil. When the control-valve plunger is moved to the right, the pump inlet is opened and oil is pumped through the check valve into the power cylinder. Moving the control valve to the left of neutral permits oil to be forced out of the cylinder by the weight of the implement (acting through the lift linkage and piston).

Fig. 5.1. Two types of control systems for single-acting cylinders.

System (b) employs an open-center valve arrangement. With the control lever in the neutral position (as shown), oil from the pump is bypassed through valve A at low pressure and returned to the reservoir while valves B and C prevent oil from leaving the cylinder. Moving the control lever to the “extend” position closes valve A, forcing oil from the pump through check valve B into the cylinder. Opening check valve C by moving the control lever to the right permits oil to flow out of the cylinder and return to the reservoir.

A simple, open-center hydraulic system with a double-acting cylinder is illustrated in Fig. 5.2. Oil flow paths are indicated for each of the three positions of the control valve. In the neutral or “hold” position, only the close fit of the sliding spool in the control valve prevents oil from leaving the cylinder. Any leakage past the spool will cause the piston to creep. This difficulty is not encountered in the system shown in Fig. 5.1b, be-
5.6. Parallel-Cylinder Systems. Selective hydraulic control systems, with two or more individually controlled cylinders operated from a single pump, are becoming increasingly common. Figure 5.3 illustrates a parallel-cylinder system having the control valves in parallel and employing a pilot-controlled bypass valve. This same system could be used for more cylinders than shown or for only one cylinder. Control valve A is shown in the "hold" or neutral position and valve B is in the "extend" position.

As long as one or more of the control valves in such a system is held in an operating position (as shown for valve B), there is a direct passage from pilot-valve line C to return line D and back to the reservoir. Because the small-diameter pilot-valve orifice greatly restricts the flow to line C, the pressure on the bypass-valve piston is low and the spring holds the bypass valve closed.
When all control valves are returned to neutral, there is no longer any passage from line $C$ to line $D$. Then, since there is no flow through the pilot-valve orifice, the pressure on the bypass-valve piston becomes equal to that in the pump discharge line. Since the piston area is several times that of the ball seat, the valve opens and remains open under low pressure until one or more of the control valves is moved from neutral and again opens line $C$ to line $D$.

Cylinders in parallel can also be controlled independently by a stacked, series-parallel arrangement of open-center valves. Such an arrangement, with two identical valve units clamped together between header plates, is illustrated in Fig. 5.4. One valve may be removed or more may be added. Valve $A$ is shown in the neutral position, whereas valve $B$ has been moved to the right to operate cylinder $B$. When all valves are in neutral, the pump output flows through the open-center passages in series and thence back to the reservoir. When any valve is moved from neutral, the bypass path is closed (as shown for valve $B$) and oil under pressure becomes available to this cylinder and to
any others that might be operated simultaneously. Thus, while
the valves are in series when bypassing the oil from the pump
to the reservoir (i.e., all valves in neutral), the simultaneous
operation of two or more cylinders is in parallel.

If only one control valve of a parallel-cylinder system is oper­
ated at any particular time, the behavior is identical with that
of a single-cylinder system. If two or more control valves are
operated at the same time, with unrestricted flow to all cylinders,
the piston with the lowest pressure requirement will be moved
first (as when parallel cylinders are operated by a single control
valve). The other cylinder (or cylinders) cannot operate until
the system pressure rises after the first cylinder reaches the end
of its stroke. The total oil displacement or over-all piston travel
time is equal to the sum of the values for the individual cylinders,
regardless of whether or not the pistons move simultaneously;
but the full rated pressure is available for each cylinder if needed.

5.7. Cylinders in Series. Selective control of individual,
double-acting cylinders in a multiple-cylinder system can also
be obtained by a series arrangement of open-center control

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Fig. 5.4. Stacked arrangement of open-center valves for operating cylinders in parallel.
valves, in which the pump is connected to the input side of only the first valve. The return line from each valve is connected to the input side of the next valve in the series, this being the only interconnection between valves. The return line from the last valve goes to the oil reservoir. With this arrangement the pump output will be bypassed through all valves in series as long as they are all in the neutral position. If any one valve is moved from neutral to extend or retract its cylinder, the system behaves like a single-cylinder system. If two or more control valves are operated at the same time, a series arrangement of the cylinders is obtained, with positive simultaneous movement of the pistons. The oil from the low-pressure side of one cylinder is forced through the control valves into the high-pressure side of the next cylinder being operated. The oil from the low-pressure side of the last cylinder in the series is returned to the reservoir.

When two or more cylinders are operated in series, the rate of movement of the first piston is the same as it would be for a comparable single-cylinder system. The movement of the next piston will be equal to that of the first one if the effective cross-sectional areas of the low-pressure end of the first cylinder and the high-pressure end of the second cylinder (i.e., the series-connected ends) are equal. The input pressure required for a series system is equal to the sum of the pressure differentials represented by the thrust loads of the individual cylinders. Thus if two cylinders require pressure differentials of 300 and 400 psi, respectively, the input pressure must be 700 psi if the cylinders are operated simultaneously.

5.8. Cylinder Check Valves. A system with hydraulically opened check valves in the oil lines between the main control valve and the power cylinder is illustrated in Fig. 53. While cylinder B is extending, check valve E operates in the usual manner and permits oil flow to the high-pressure end of the cylinder. At the same time, check valve F is held open by oil pressure on the head end of the valve piston, thus permitting oil to leave the low-pressure end of the power cylinder. Check valves of this type prevent cylinder creep due to leakage past the valve spools. They also provide a safety feature for personnel, since it is impossible to lower an implement by merely moving the control lever when the oil pump is not operating.

A hydraulic check valve is adaptable to a single-acting cyl-
INDER, provided the main control valve is double-acting so that it can apply pressure to the check-valve piston when the control lever is moved to the "retract" position.

Mechanically opened check valves (similar to valve C in Fig. 5.1b) can be used on double-acting cylinders in conjunction with the main control valve. They protect against cylinder creep but not against accidental lowering of the implement when the pump is not operating.

5.9. Limit-Control Hydraulic Systems. In this type of system there is no indexed relation between the position of the control lever and the implement depth. However, stops of various types (Fig. 5.5) may be provided to limit the piston travel on the retracting stroke and thus predetermine a working position or depth for the tool. The common arrangements include (a) mechanical stop on the cylinder, (b) hydraulic shut-off actuated by piston movement, either directly or through the lift linkage, and (c) electric remote control of hydraulic valves, with electrical limit switches on the cylinder or linkage. All these devices are adjustable, but with many systems a change of the stop setting involves halting the implement and perhaps raising it slightly to free the stop. Another arrangement for mounted implements involves adjustable stops or hand levers that are part of the implement, this system being suitable only where the tool lowers by gravity rather than being forced down by the piston.

Limit-control systems frequently have a detent arrangement.
to hold the control valve in either of its extreme positions while the implement is being raised or lowered, thus freeing the operator's hand for other purposes. When the piston reaches the limit of its travel, the detent is automatically released by hydraulic or mechanical means, and spring action returns the control valve to neutral. It can also be returned to neutral at any time by hand. With some tractor-mounted cylinders, the piston or linkage merely pushes the lever back to neutral instead of a detent being released.

Control levers sometimes have an intermediate, indexed position between neutral and each extreme, which provides slow-speed movement of the piston for making small changes in depth adjustment.

5.10. Automatic Position Control. In this type of system, any given position of the control lever in its quadrant represents a specific position or depth of the implement. The basic linkage for such a system is indicated in Fig. 5.6a. The solid outlines represent a stabilized condition, the control valve being in neu-
tral. If the control lever is then moved from $A$ to $A'$, joint $C$ is moved to the left while joint $E$ momentarily remains in the same position, thus moving $D$ and the control-valve spool to the left. This actuates the power cylinder, which raises the implement until $D$ is returned to the neutral position. With the control valve again in neutral, the new position of the lift arms and connecting linkage is as shown by the broken-line outlines. Similarly, moving the control lever in the opposite direction results in lowering of the implement. In effect, $D$ acts as a pivot point in determining the relative equilibrium positions of the control lever and the implement.

Theoretically, automatic position control can be applied to any type of cylinder and control-valve arrangement. But, since a mechanical interconnection between the control lever and the piston or implement linkage is required, automatic position control is most readily adapted to mounted implements. This system could, however, be applied to trailed implements by transmitting the piston movement to the valve linkage through an arrangement such as a push-pull cable.

For accurate position control, the valve design must be such that only a small amount of valve movement from neutral is required to actuate the cylinder.

5.11. Automatic Draft Control. As indicated in Fig. 5.6b, the linkage for automatic draft control is similar to that for automatic position control except that the top end of equalizer link $CE$ is moved in response to deflection of the load spring rather than by movement of the implement lift linkage. With the control lever in a given position, any change in draft changes the amount of deflection of the load spring, thus moving the control valve from its neutral position, $D$. The hydraulic system then acts to change the implement depth as required to restore the draft and load-spring deflection to the preselected values and thus return the control valve to neutral. In the arrangement illustrated, the draft-responsive hitch member is always in compression.

It is evident from Fig. 5.6b that, for any given position of the control lever, there is only one compressed length of the load spring that will bring the control valve to neutral. But, if the control lever is moved to the left in Fig. 5.6b, less compression
of the load spring, and hence less draft, is required for equilibrium.

Automatic draft control can be applied to trailed implements by mounting the tractor drawbar so that the pull is transmitted to the load spring of the control system. The control valve then actuates the remote cylinder on the trailed implement, just as it would a tractor-mounted cylinder.

Automatic position control and automatic draft control can be used interchangeably in some hydraulic systems for mounted implements, the operator making the selection by merely moving an auxiliary lever. At least one system employs a nonselective combination of automatic draft control and automatic position control. In this system the load spring and lift linkage are interconnected through an equalizer link so that they both affect the control valve. The result is a compromise between a constant implement depth and constant draft.

Sensitivity of the hydraulic control valve and speed of response are important considerations in an automatic draft control system. Fast response is important in maintaining constant draft under varying operating conditions, but too rapid a response may cause "hunting" and instability of the control system.

5.12. Delayed-Action Systems. One of the applications of multiple-control systems is with mounted cultivators. If the front and rear gangs are lifted by independently controlled hydraulic cylinders, the operator can, by raising and lowering them in the proper sequence, cultivate out to the end of the row with all gangs. A similar effect can be achieved with two cylinders operated from a single control valve, through a device that automatically delays the action of the rear cylinder as illustrated in Fig. 5.7.

The pressure required to open spring-loaded valve A and admit oil to cylinder 2 is somewhat greater than the maximum pressure needed to extend cylinder 1. For example, valve A might be set to open at 500 psi in a 1000-psi system. When cylinder 1 reaches the end of its extending stroke, the pump pressure immediately increases sufficiently to open valve A and start to extend cylinder 2. The amount of time delay can be varied somewhat by throttling the oil flow.

When the control valve is moved to lower the implements, both cylinders start to retract simultaneously, but the full-open
clearance of check valve $B$ is restricted by the adjustable stop $C$ so that cylinder 2 retracts more slowly than cylinder 1.

Manual selection and control of the delayed lowering can be provided. Basically, this involves the elimination of valve $B$ and the incorporation of spring-loaded valve $A$ into the main control valve assembly so that a mechanical means (such as a push rod) can be provided for opening this valve with the main control lever. With this arrangement the control lever is first moved to release oil from cylinder 1 and then, after the desired delay, is moved further to open valve $A$ and permit cylinder 2 to retract.

5.13. Standardization of Control Systems for Trailed Implements. To provide complete interchangeability of hydraulic cylinders used on trailed implements, the ASAE and SAE have jointly adopted standards for the application of hydraulic controls to this type of implement. These standards, which are included in Appendix B, specify cylinder mounting and operating dimensions, maximum cylinder outline dimensions, clearance allowances required on the trailed implement, full-stroke operating times, and requirements in regard to hose lengths.

Cylinders with an 8-in. working stroke are specified for tractors (and corresponding trailed implements) having up to 6000 lb maximum drawbar pull, whereas 16-in.-stroke cylinders are required for tractors with drawbar capacities between 11,000 and 20,000 lb. In the range from 6000 to 11,000 lb maximum drawbar pull, both sizes are to be available. Implements are to be raised (or disk harrows de-angled) on the extending stroke of
the piston, because the effective piston area is greater than for the retracting stroke. Stops needed for variable-stroke control are to be incorporated into the cylinder or hydraulic system rather than being part of the implement.

The British have adopted similar standards for tractors having up to 25 drawbar horsepower. Their standards are for an 8-in.-stroke, double-acting cylinder, with dimensional specifications based upon the ASAE-SAE standards. Hose lengths are not specified, but self-sealing, quick-release couplings are recommended. Whereas the ASAE-SAE standards have no specifications regarding pressures, cylinder diameters, or power capacity, the British standards require that the system be able to perform from 50,000 to 55,000 in.-lb of work in 2 sec.


The tractor engineer has the responsibility of providing a hydraulic control system with adequate capacity to lift and control any implement within the power range of the tractor. The implement designer, on the other hand, should be concerned with minimizing peak lifting loads, keeping these loads within the available capacity of the power lift, and generally making the most effective use of the control system.

In designing a lift linkage, the engineer should remember that the total work capacity of a hydraulic cylinder is primarily a function of the effective piston area, the available pressure, the length of stroke, and friction losses in the hydraulic system and cylinder. The lifting time, on the other hand, is related to the piston displacement and the pump capacity at the required lifting pressure. The ASAE standards specify that the full-stroke operating time at rated engine speed and rated pressure shall be from 1\(\frac{1}{2}\) to 2 sec for cylinders with an 8-in. working stroke. This operating time represents a compromise between high power requirements of rapid lifting and greater travel distances for slower lifting.

The total work (or the average force for a cylinder with a given stroke) required to lift an implement is influenced by the lifted weight of the implement, the soil reaction, friction in the linkage, inertia forces and surges, and the height of lift. The ideal linkage, from the standpoint of the hydraulic system, would be such as to require a constant cylinder thrust throughout the stroke, at least under extreme load conditions. This would give maxi-
mum work output with minimum pressures. Frequently, however, the initial design of a linkage results in high peak forces during some portion of the cycle, as illustrated by the work diagram shown in Fig. 5.8.

Assuming it is desired to take corrective measures to obtain a more uniform thrust requirement throughout the stroke, the first steps are to determine (either graphically or experimentally) the relation of implement lift to piston travel and to plot the results on the work diagram. Then divide the piston travel into equal increments on the graph and measure the area under the cylinder thrust curve (i.e., the work) for each increment and for the full piston travel.

The average height of the work curve (total area divided by total piston stroke) represents the desired uniform thrust. Dividing each increment of work area by this average thrust gives the

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Fig. 5.8. Work diagram for lifting a three-bottom 16-in. moldboard plow stalled in hard ground. The first peak is attributed to the load required to break ground, and the high peak near the end is due to increased lifting rates as the slack in the rear furrow-wheel linkage is taken up. (R. D. Barrett. Agr. Eng., October, 1949.)
theoretical portion of the piston stroke that should be used for the amount of implement lift represented by the original increment of piston travel.

As an example, take 1-in. increments of piston travel for the work diagram in Fig. 5.8. Assume that with the existing linkage the implement is lifted 0.9 in. during the first 1 in. of piston travel and an additional 1.2 in. during the next 1-in. increment. Dividing the measured work areas for these first two increments by the average thrust (4050 lb) indicates that only 0.73 in. of piston travel should be used in lifting the implement through this first step of 0.9 in. and 0.81 in. of piston travel for the second lift step of 1.2 in. Thus a piston travel of 0.73 in. for a lift of 0.9 in. represents one point on the ideal lift curve, and a total piston travel of 1.54 in. for a total lift of 2.1 in. represents the next point. Similar treatment of successive increments will yield a complete ideal curve, against which the results of proposed modifications in the linkage can be compared.

If the uniform thrust obtained by redesigning the linkage still exceeds the available capacity, additional corrective measures that may be taken include (a) reduction of total friction in linkage, (b) reduction of total lift height, (c) reduction of implement weight, (d) elimination of unnecessary horizontal movement of the implement in the soil, resulting from the lifting action, and (e) use of auxiliary springs.

Where power-control loads are light so that uniformity of thrust is not important, it is possible to incorporate other desirable features into the system. For example, greater fineness of control for a tillage tool can be obtained by arranging the linkage to give a low lift rate in the range of usual working depths. If, on the other hand, a quick lift is desired, as for a mower cutter bar, the design can be such as to utilize only a portion of the piston stroke for the full lift.

5.15. Cylinder Thrust Requirements for Trailed Implements. Work diagrams such as that shown in Fig. 5.8 have been obtained by means of a recording pressure indicator mounted so that the rotation of the chart drum is proportional to the piston travel and also by taking motion pictures of a pressure gage together with a linear scale indicating piston travel. If the pressure is taken from the hydraulic cylinder or oil lines, rather than using a separate load cell, allowance for losses due to fluid
friction and piston friction must be made in order to determine the implement thrust requirements.

Thrust requirements for various implements, as determined from a large number of field tests by various manufacturers, have been summarized by Worthington and Seiple.\textsuperscript{16} The maximum requirements for various types and sizes of implements when traveling in the field under normally severe operating conditions are indicated by the individual points plotted in Fig. 5.9.

These results indicate the extreme variability of maximum lifting requirements to be expected for different models of a particular type of implement under the wide range of conditions likely to be encountered. In addition, the solid curve shows the over-all thrust requirements needed to satisfy all the conditions represented by the various tests. The two broken-line curves show the range of computed thrusts available from various wheel-type tractors equipped with 8-in.-stroke hydraulic cylinders. The computed thrusts are based upon published dimensional and pressure data and include allowances for average fluid-friction

Fig. 5.9. Maximum thrusts (hydraulic-cylinder pin loads) required to operate implements while traveling in the field. All data are for cylinders with an 8-in. stroke. (Wayne H. Worthington and J. Waldo Seiple. \textit{Agr. Eng.}, May, 1952.)
losses and cylinder efficiencies. It is evident that tractors whose available thrusts lie in the lower part of the area between the two broken lines would not be able to lift all implements in their horsepower range, under the conditions represented by the plotted points.

Most tillage implements require more thrust to lift them when stopped than when traveling, the increase being as much as 50 to 100 per cent for some implements. It is generally assumed that the rated pressure of the hydraulic system should not normally be exceeded while lifting a traveling implement, but that the full relief pressure can be considered as available for the occasional instances when a stalled implement must be raised. This gives an additional available thrust of perhaps 25 per cent for such emergencies.

5.16. Miscellaneous Applications of Hydraulic Controls. Although the preceding discussion has dealt primarily with tractor hydraulic control systems as applied to trailed or mounted implements, the same principles can be applied to similar applications on other farm machinery. For example, self-propelled combines have hydraulic controls for the header and may use them for controlling variable-speed drives, for power steering, etc. In combines where two parallel cylinders support the header (operated from a single control valve), the torsional rigidity of the header frame and the approximate equality of cylinder loads is relied upon to keep the piston movements equalized. With tractor-mounted loaders, hydraulic pumps and control systems are often part of the loader, although powered by the tractor. The implement designer is responsible for control applications of this type.

POWER-TAKE-OFF DRIVES

5.17. Applications. Since power-take-off (pto) drives were introduced, in the 1920's, their use has steadily increased. The availability of this type of power system has been an important factor contributing to the development of implements such as the corn picker, small combines, rotary cutters and shredders, etc. Many implements such as field balers, forage choppers, and combines, that have rotary power requirements within the capacities of the propelling tractors, use pto drives and auxiliary engines
HYDRAULIC CONTROLS AND PTO DRIVES

interchangeably. Although some degree of flexibility is often sacrificed by the use of a pto drive rather than a separate engine, the saving in initial investment is an important factor, especially when annual use of the implement is low.

The introduction of the “live” or independently controlled pto, in the late 1940's,\(^2\) has overcome one of the serious problems with this type of drive on some machines. The live pto allows the forward travel to be stopped without interrupting the operation of the machine, which is particularly advantageous for implements such as combines, that could frequently become clogged.

Power-take-off drives are becoming increasingly popular as a substitute for belt drives on stationary machines such as feed grinders and corn shellers. This application can, however, be rather dangerous because the pto shields often are not installed and because the operators usually have occasion to work near the shafts.\(^2\)

5.18. Standardization of Power-Take-Off Drives. The joint adoption by the ASAE and SAE\(^13\) of a standard pto shaft speed (536 ± 10 rpm) as well as dimensional specifications for the tractor pto shaft and its location with respect to the hitch point (Fig. 5.10) have provided a basis for easy attachment and interchangeable use of pto implements with various makes and models.

![Fig. 5.10. Standardized locations for pto shaft and drawbar hitch on agricultural tractors. The lateral location of the pto shaft is to be within ±3 in. of the tractor centerline, and the hitch point is to be directly beneath the extended centerline of the pto shaft. (ASAE-SAE Standard, revised, 1955.)](image-url)
of tractors. These standards also make it possible for an implement manufacturer to provide adequate shielding for his pto drive with the assurance that it can be attached to the master pto shield on any standardized tractor, thus contributing greatly to the safety of the operator.


The effect of universal-joint angularity on shaft speeds and stresses and the importance of proper orientation of universals in multiple-joint systems are discussed in Chapter 4. Figure 5.11 shows a typical pto arrangement with two universals. Ideally, the hitch point should be midway between the two joints so that the joint angles would be equal for any position of the implement with respect to the tractor. In practice, however, it has been found \(^\text{12}\) that, if the ASAE standard dimension of 14 in. from the hitch point to the end of the pto shaft is used, it is necessary to have the rear universal a little farther from the hitch point to obtain sufficient telescoping action for sharp turns. Then the two joint angles are not equal when the implement is on a turn (Table 5.1) and there will be an increasingly serious fluctuation of angular velocity as the implement is turned more sharply (see Fig. 4.6).

5.20. Triple-Joint Power-Take-Off Drive. Figure 5.12 shows the general arrangement for a pto drive having three universal
Table 5.1  COMPARISON OF UNIVERSAL-JOINT ANGLES FOR POWER-TAKE-OFF DRIVES ILLUSTRATED IN FIGS. 5.11 AND 5.13

<table>
<thead>
<tr>
<th>Angle Between Implement and Tractor, degrees</th>
<th>Two-Joint Drive</th>
<th>Three-Joint Drive</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Front Joint</td>
<td>Second Joint</td>
</tr>
<tr>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>30</td>
<td>16</td>
<td>14</td>
</tr>
<tr>
<td>50</td>
<td>27</td>
<td>23</td>
</tr>
<tr>
<td>70</td>
<td>38</td>
<td>32</td>
</tr>
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<td>40</td>
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<tr>
<td></td>
<td>21</td>
<td>29</td>
</tr>
<tr>
<td></td>
<td>33</td>
<td>37</td>
</tr>
<tr>
<td></td>
<td>51</td>
<td>39</td>
</tr>
</tbody>
</table>

Angles in Universal Joints, degrees

joints. Note that the telescoping action is at the rear joint, with a fixed-length shaft between the two front joints. This type of drive requires more longitudinal space than the two-joint arrangement, but a longer telescoping distance can be provided. With a fixed distance between the two front joints, it is geometrically impossible to have these joints equidistant from the hitch point for all turning angles of the implement. Thus, a compromise is made, as indicated in Fig. 5.13, with the second joint being closer to the hitch point when the implement is not turning and farther than the front joint when the implement is turned 90° from the tractor.

Fig. 5.12. A triple-joint pto drive assembly for trailed implements. Note the dashed outlines of the shields. (Fred M. Potgieter. *Agr. Eng.*, January, 1952.)
The plan view of Fig. 5.13 shows the implement shaft normally in line with the tractor pto shaft. With this arrangement the rear universal joint operates practically straight at all times. The joint angles for the two front joints are indicated in Table 5.1 for various implement turning angles. Note that the second joint has the greater angle for a 70° implement turn but a smaller angle than the front joint for a 90° turn. The drive will be quite rough for an implement turn of 90°.

If the driven shaft of the implement is offset from the center-line of the tractor pto shaft so that the normal operating angle of one or both of the intermediate shafts is greater than about 15 to 20° from the line of the pto shaft, serious velocity fluctuations may occur on sharp turns in the field. Figure 5.14 shows two possible arrangements for obtaining the offset. With the joints phased as indicated, either system theoretically provides

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**Fig. 5.13.** Plan view of typical three-joint pto drive having the implement shaft in line with the tractor pto shaft. (Fred M. Potgieter. *Agr. Eng.*, January, 1952.)
uniform velocity of the driven shaft when the implement and tractor are in line. On turns, the lower arrangement is considerably better than the upper one, having smaller maximum joint angles and much less fluctuation in angular velocity. For a given amount of offset, it has the disadvantage of requiring greater joint angularities in the normal operating position.

5.21. Loads Imposed on Power-Take-Off Shafts. Extensive tests with fast-responding hydraulic and electronic torque meters have indicated that average power or torque requirements have little value as a basis for designing pto drive parts. Maximum instantaneous (high-frequency) peak torques are of far more importance. These peak loads vary tremendously, and are influenced by the following factors:

1. The amount of kinetic energy stored in the rotating parts of the tractor.
2. The moment of inertia of the rotating parts.
3. The amount of resilience in the drive between the heavy rotating parts of the tractor and the rotating parts of the driven implement.
4. The pto horsepower available from the tractor.
5. The horsepower required to operate the implement.

The first three of the above items have a great deal more influence on the magnitude of peak torques than do the last two.
<table>
<thead>
<tr>
<th>Approximate Maximum Tractor bhp</th>
<th>Description of Implement and Reference to Type of Work Performed †</th>
<th>Type of Coupling in pto Drive</th>
<th>Maximum Starting Torque, lb-in.</th>
<th>Maximum Operational Torque Peaks, lb-in.</th>
<th>Average Torque under Normal Operating Conditions, lb-in.</th>
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<tr>
<td>35</td>
<td>Ensilage harvester A (1) †</td>
<td>Standard</td>
<td>4,900-6,400</td>
<td>10,800-15,370</td>
<td>4,680-6,300</td>
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<tr>
<td>35</td>
<td>Ensilage harvester A (1) †</td>
<td>Special slip</td>
<td>8,800 †</td>
<td>5,115-5,723</td>
<td>4,450-7,110</td>
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<tr>
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<td>Ensilage harvester A (1) †</td>
<td>Standard</td>
<td>11,600</td>
<td>4,700-4,925</td>
<td>6,200-8,025</td>
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<tr>
<td>35</td>
<td>Ensilage harvester B (1) †</td>
<td>Standard</td>
<td>2,600-4,000 †</td>
<td>3,200</td>
<td>3,960-7,680</td>
</tr>
<tr>
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<td>Forage harvester C (2)</td>
<td>Standard</td>
<td>14,600</td>
<td>3,730-7,200</td>
<td>6,370-7,200</td>
</tr>
<tr>
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<td>Forage harvester C (2)</td>
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<td>9,800-7,530 †</td>
<td>5,230-6,700</td>
<td>6,100-8,700</td>
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<td>Forage harvester C (2)</td>
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<td>6,060-7,460</td>
<td>9,500</td>
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<td>Corn picker D</td>
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<td>222-1,051</td>
<td>727</td>
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<td>Baler E (3)</td>
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<td>18,300-20,600</td>
<td>5,860-7,470</td>
<td>12,100</td>
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<td>Baler E (3)</td>
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<td>13,100</td>
<td>6,550-8,140</td>
<td>11,600-15,600</td>
</tr>
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<td>35</td>
<td>Baler E (3)</td>
<td>Special slip</td>
<td>10,700-12,100 †</td>
<td>7,250-8,920</td>
<td>11,600-13,300</td>
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<td>8,900-11,100</td>
<td>10,850-12,690</td>
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<td>Baler E (3)</td>
<td>Special slip</td>
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<td>—</td>
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<td>Combine G (5)</td>
<td>Standard</td>
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<td>3,760</td>
<td>9,380</td>
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<td>7,350-8,650 †</td>
<td>7,760-9,130</td>
<td>1,700</td>
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<td>Special slip</td>
<td>4,160-4,200</td>
<td>7,470</td>
<td>1,000</td>
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<td>9,030</td>
<td>11,500</td>
<td>14,800</td>
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<td>35</td>
<td>Hammer mill H (7)</td>
<td>Special slip</td>
<td>21,210</td>
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<td>35</td>
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<td>8,220</td>
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<td>Hammer mill J (7)</td>
<td>Standard</td>
<td>18,150</td>
<td>7,800</td>
<td>13,000</td>
</tr>
</tbody>
</table>

† Description of work being performed: (1) chopping heavy drilled corn, (2) chopping green alfalfa, (3) baling alfalfa, (4) baling straw, (5) combining from windrow, (6) direct combining, (7) grinding ear corn.
‡ Safety clutch in pto drive slipped, limiting torsional load to this value.
§ Universal joints aligned before this run with baler E.
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items. This is particularly true when pto drives are used on implements such as balers, hammer mills, ensilage or forage choppers, and combines, which are capable of causing very rapid changes in pto rotating speeds. Table 5.2 includes maximum and average torque values obtained in tests with these implements operating under various conditions that might be encountered in regular service. Figure 5.15 shows typical torque-meter charts. Note that even under normal operating conditions, peak torques were several times as great as the average values.

Although mechanical failures of pto drives have generally been in the tractor components, the implement designer has the responsibility for investigating the pto load characteristics of his implements and making any practical corrections that will reduce excessive peak torques. Tests with a tractor driving an electric dynamometer through a typical pto power-shaft arrangement have demonstrated that peak torques are increased considerably by using universal-joint angles greater than 15 to 20° for a pair of joints at equal angles or greater than 8 to 10° for a single joint (Fig. 5.16). Thus minimum joint angles during normal operation (i.e., implement not turning) are important.

Peak torques can be reduced to any desired value by the use...
of a friction slip clutch in the pto drive, as indicated in Table 5.2 and by the two lower charts in Fig. 5.15. However, many implements that are subject to occasional plugging or infrequent overloads now take advantage of the fact that pto drives can absorb occasional peak loads about twice as great as the maxi-

![Graph showing the effect of universal-joint angularity on maximum and minimum instantaneous peak torque values in a pto drive when transmitting 30 hp.](Merlin Hansen. Agr. Eng., February, 1952.)

maximum allowable values for infinite fatigue life. A protective device cannot be expected to limit loads to values within the infinite-fatigue-life specifications and yet permit occasional torsional loads of twice that magnitude. In these cases it is particularly important that the normal high-frequency torsional peaks of the implement be within the infinite-fatigue-life limit of the drive so that the protective device can be adjusted to allow the occasional higher loads.

5.22. Recommendations for Power-Take-Off Load Limits.
As a result of the investigations discussed in the preceding section, the ASAE has adopted an official recommendation regard-
HYDRAULIC CONTROLS AND PTO DRIVES

ing the operating requirements for pto drives. This recommendation (see Appendix G) discusses the importance of proper hitching, adequate telescoping action of the power shaft, and provision to prevent locking of universal joints when the implement is on a sharp turn. It specifies the maximum bending load to be imposed upon the tractor pto shaft and recommends torque limit settings for power-line protective devices under various types of loading.

REFERENCES


PROBLEMS

5.1. Assume that, for the conditions represented in Fig. 5.8, the actual increments of plow lift for successive 1-in. increments of piston travel were 0.9, 1.2, 1.4, 1.6, 1.9, 2.7, 3.6, and 3.9 in. (17.2 in. total lift). Plot this assumed curve of plow lift versus piston travel (cumulative values). Then, using the curve in Fig. 5.8, determine and plot the ideal lift curve for constant thrust requirement.

5.2. The hydraulic control valves of a two-cylinder system are manipulated to connect the cylinders in series as indicated in the accompanying diagram. Each cylinder has a bore of 3 in. and a piston-rod diameter of 1 in. The thrust loads are indicated on the diagram. The pump output is 8½ gpm. Calculate:

(a) Required pump pressure, neglecting line losses.

(b) Rate of piston movement for each cylinder, in inches per second.

(c) Horsepower input to the pump, assuming an over-all system efficiency of 40 per cent.

5.3. The two hydraulic cylinders in a parallel system have the same dimensions and thrust loads as in Problem 5.2. The pump output is 8½ gpm.

(a) Describe the behavior of the system when both control valves are moved simultaneously to extend the two cylinders.

(b) Calculate the required pump pressure, neglecting line losses.

(c) Determine the rate of piston movement for each cylinder, in inches per second.

(d) What is the total time required for both pistons to move through a full 8-in. stroke (extending) when both valves are held in the operating position?
5.4. (a) For each of the pto drive arrangements shown in Figs. 5.11 and 5.13, calculate the amount of telescoping action required in turning the implement through a 90° angle.

(b) For each arrangement, determine the additional amount of telescoping required if the inside rear wheel of the tractor drops into a 12-in. ditch while the implement is turned at 90°. The tractor drawbar and pto shaft are at the ASAE-SAE recommended heights and the tractor rear wheel tread is 60 in. Assume that all shafts remain horizontal.

5.5. Referring to Chapter 4, determine and compare the maximum lead or lag angles of the implement driven shaft with respect to the tractor pto shaft, for the two pto drive arrangements represented in Table 5.1, using 30°, 50°, and 70° implement turning angles. For the conditions of this problem, it will be sufficiently accurate to consider that the peaks for the two joints occur simultaneously.

5.6. Calculate the average horsepower requirements for each of the implements represented in Table 5.2, assuming the standard pto shaft speed.
CHAPTER 6

Tillage Force Analysis and Hitching

6.1. Introduction. Since it is important that the student be thoroughly familiar with the fundamental definitions and relations of mechanics, the pertinent terms will be reviewed here along with additional concepts that are used specifically in connection with farm machinery.

*Force* is any action that changes or tends to change the state of rest or motion of a body. A force is completely specified by its magnitude, direction, and point of application. The common unit in the English system is the pound.

*Pull* on an implement is the total force exerted upon the implement by a power unit. With tillage tools it is generally at some angle above the horizontal, and it may or may not be parallel to the vertical plane containing the line of motion.

*Draft* or *directional pull* is the horizontal component of pull, parallel to the line of motion.

*Side draft* is the horizontal component of pull, perpendicular to the line of motion.

*Torque* is the moment of a force tending to produce rotation about a point. It is the product of the force times its radius of rotation and is commonly expressed as pound-feet or pound-inches to distinguish it from work. A *couple* consists of two equal and opposite forces that are parallel but not concurrent. The magnitude or moment of a couple is equal to the product of one of the forces times the perpendicular distance between the two forces. A couple may tend to produce rotation about any point in the plane of the two forces. Thus, torque is a special case of a couple with the center of rotation on the line of one of the forces.

*Work* is the product of force (in the direction of motion) times
the distance through which the force acts. The foot-pound is the common unit in the English system.

Power is the rate of doing work. Common units are foot-pounds per second, foot-pounds per minute, and horsepower.

\[ \text{Horsepower (hp)} = \frac{33,000 \text{ ft-lb of work}}{\text{minute}} \]

Drawbar horsepower (dbhp), in relation to either a trailed or a mounted implement, is the power actually required to pull or move the implement at a uniform speed. It is sometimes convenient to recall that at 3.75 mph the drawbar horsepower is \( \frac{1}{100} \) of the draft in pounds. The horsepower at other speeds would, of course, be directly proportional to the speed.

\[ \text{Horsepower-hour (hp-hr)} = \text{quantity of work performed when one horsepower is used for one hour} \]

It is equal to 1,980,000 ft-lb of work.

6.2. Forces Acting upon a Tillage Implement. The engineer is concerned with the forces acting upon tillage implements from the standpoints of total power requirements, proper hitching or application of the pulling force, designing for adequate strength and rigidity, and determination of the best shapes and adjustments of tools. A tillage tool moving at a uniform velocity is subjected to three main forces: (a) gravity (weight of implement), (b) pull applied to maintain motion, and (c) the total soil reaction. A fourth force, due to torque from a pto drive (as in a pto-driven potato digger), may be present in a few cases. These forces must all be in equilibrium for the condition of uniform motion.

Clyde subdivides the total soil reaction into useful and parasitic forces. He defines useful soil forces as those which the tool must overcome in cutting, breaking, and moving the soil. Parasitic forces are those (including friction or rolling resistance) that act upon stabilizing surfaces such as the landside and sole of a plow or upon supporting runners or wheels. Under a given set of operating conditions with a specific tool, the operator has little control over the useful soil-resistance forces. However, both the designer and the operator have some control over parasitic forces.

6.3. Measurement of the Useful Soil Resistance by the Pulling Method. In this method, the tool is pulled by means of a free link of some length (such as a chain or cable) to indicate
the line of pull. The position and angle of the pulling force are adjusted until the tool operates at the desired depth, width, and angle. The pulling force is then measured, and the location of the line of pull is established with respect to the implement. Then, after weighing the tool and finding its center of gravity, the weight and pull forces can be combined to determine the total soil reaction required for equilibrium. For determining the useful soil force alone, the pulling method is applicable only to implements such as a group of cultivator shovels or a gang of disks, which will operate rather steadily with practically no parasitic force. It is not suitable for determining the useful force on tools such as the moldboard plow because it is almost impossible to get stable operation without some side or vertical support. The results are often not very accurate because of soil variations and the difficulty of making measurements “on the go.”

6.4. Multiple-Support Force-Measuring Devices. If a tool is attached to a frame that is entirely supported by force-measuring devices, the soil reaction, including rotational effects, can be determined completely. Devices of this type have been in use by the Pennsylvania Agricultural Experiment Station since 1935 and by the USDA Tillage Machinery Laboratory at Auburn, Alabama, since 1936. In each of these two units, the subframe to which the tool is attached is held in place by six hydraulic load-measuring cells, each connected to a pen on a strip-chart recorder which also records time and distance. Three of the cells indicate horizontal forces (either one or two of them being at right angles to the direction of travel), and the other three support the subframe vertically.

With this arrangement, tools having any combination of translational and rotational soil reactions can be tested. Parasitic forces may be included in the measured soil reaction, or they can be eliminated or minimized by adjustment of the tool or by removing any stabilizing or supporting surfaces. The implement weight is eliminated from the force considerations by taking initial load readings with the tool suspended and then dealing only with load changes caused by the soil reaction. The changes in load-cell reactions are combined graphically or analytically into one of the two following forms 7 to express the entire effect of soil resistance upon the tool (Fig. 6.1):
TILLAGE FORCE ANALYSIS AND HITCHING

1. Two nonintersecting forces, one vertical and one horizontal.
2. A single force plus a couple (the couple usually being placed in a vertical plane perpendicular to the direction of travel).

If there is no rotational effect on the tool, the vertical and horizontal resultants in Fig. 6.1a will intersect and the value of the couple in Fig. 6.1b will be zero.

The USDA Tillage Machinery Laboratory\textsuperscript{11,12} has nine soil bins, each 20 ft wide, 250 ft long, and 6 ft deep. Two of them are divided across the middle, thus providing places for a total of eleven selected soils. Rails on the walls between bins support the wheels of all soil-fitting and test units. Equipment is available for preparing the soil, sprinkling it, and protecting it from the weather. The test equipment (Fig. 6.2) consists of a power car and an attached car for carrying the tool to be tested. Thus, tests at the Tillage Laboratory can be performed under carefully controlled and uniform soil and operating conditions. This arrangement is particularly suitable for repetitive tests involving the comparison of different designs or tool adjustments under various controlled soil conditions. Quantitatively, the laboratory results do not necessarily represent field conditions because, until recently, soil in the bins has not been subjected to the effects of root growth, incorporation of plant residues, and other natural processes.

The Pennsylvania test unit,\textsuperscript{7} which is known as the tillage meter, can be moved from one field to another. This permits
some choice of soils but introduces the inherent variability of field conditions. Emphasis has been placed on the determination of soil resistances under actual field conditions with easily tilled, typical, and rather difficult soils, to give a range of conditions as a basis for design and operation. Lateral control of the tillage meter during a test run is obtained by metal wheels running in movable steel-channel tracks, while depth is controlled by two rubber-tired wheels running on undisturbed ground. A telescoping frame permits 20 ft of width to be covered before resetting the track.

6.5. Draft Measurement. As defined in Section 6.1, draft is the component of pull in the direction of travel. Barger and his associates describe three types of dynamometers used for measuring the drawbar pull of a tractor or the pull on an implement. A hydraulic type, transmitting pressure to a Bourdon gage, is easier to read than the simpler spring type, because force fluctuations can be dampened considerably. Some hydraulic dynamometers record the pull on a strip chart driven by a ground wheel so that a continuous record can be obtained. A timer may be used in conjunction with the chart, so that all components required for power determination are recorded.
If the force-measuring element is connected directly in the line of pull of the implement, the total pull will be measured, rather than the draft component. Thus, the inclination of the line of pull from the horizontal and the angle from the direction of travel must be observed so that the draft or directional pull can be computed from the total dynamometer pull. In determining power requirements, the draft must be considered rather than the total angled pull.

The draft of plows is commonly expressed as pounds per square inch of furrow cross-section. For some implements, such as planters, pounds per row may be specified; for most other farm machines, draft is usually given in pounds per foot of width. Draft requirements for a particular tillage implement vary greatly for different soil types and conditions. A table indicating ranges of draft and power requirements for various farm implements is included in Appendix C.

6.6. Symbols Used in Tillage Force Analysis. The most common symbols occurring in the several chapters dealing with tillage tools are identified in the following list. Others are defined as they appear in the various sections.

- \( R \) = resultant of all useful soil forces acting upon the implement (Fig. 6.1b). Where useful and parasitic forces cannot be determined separately, \( R \) includes both.
- \( L \) = longitudinal or directional component of \( R \) (Fig. 6.1).
- \( S \) = lateral component of \( R \) (Fig. 6.1).
- \( V \) = vertical component of \( R \) (Fig. 6.1).
- \( R_h \) = resultant of \( L \) and \( S \) (Fig. 6.1a).
- \( R_v \) = resultant of \( L \) and \( V \) (i.e., component of \( R \) in a vertical, longitudinal plane).
- \( a \) = lateral distance between \( V \) and \( R_h \), for tools having a rotational effect (Fig. 6.1a).
- \( V_a \) = couple tending to rotate the tool about the longitudinal axis (Fig. 6.1b).
- \( Q \) = resultant of all parasitic forces acting upon the implement.
- \( Q_h \) = component of \( Q \) in a horizontal plane. It includes stabilizing side forces and the accompanying longitudinal friction forces.
- \( Q_v \) = component of \( Q \) in a vertical, longitudinal plane. It includes vertical support forces and the accompanying longitudinal friction forces or rolling resistance.
\( P \) = resultant pull exerted on the tool by the power unit.
\( P_h \) = component of \( P \) in a horizontal plane.
\( P_v \) = component of \( P \) in a vertical, longitudinal plane.
\( W \) = weight of tool, acting through the center of gravity.
\( H \) = horizontal center of resistance of the tool, which is the point of concurrence of \( R_h \) and \( Q_h \), or of two \( R_h \) components as in a disk harrow.
\( G \) = point of concurrence of \( Q_v \) and the resultant of \( W \) and \( R_v \).

It might be called the vertical center of resistance.

The subscripts \( x, y, \) and \( z \), as applied to \( P \) and \( Q \), refer to force components in the longitudinal, lateral, and vertical directions, respectively.

### Hitching of Trailed Implements *

Directions for hitching trailed implements often refer to the center of resistance of a tool as a fixed point and state that the hitch should be adjusted so that the line of pull passes through this center of resistance. Actually, in determining the location of the so-called center of resistance, the weight of the implement must be considered in addition to all soil forces, both useful and parasitic. The line of pull must necessarily pass through this center of resistance, regardless of how the implement is hitched; otherwise the system could not be in equilibrium.

As the hitch is changed the soil forces automatically readjust themselves (within the limits for stabilized operation of the tool) to move the center of resistance to the new line of pull, and thus maintain equilibrium. The function of proper hitching is to establish the center of resistance in the position that is most desirable from the standpoints of the effect of the pulling force upon the tractor and the magnitude and distribution of parasitic forces acting upon the implement.

6.7. Vertical Hitching. Figure 6.3 shows the force relations, in a vertical plane (or planes) parallel to the direction of travel, for a trailed implement that receives no support from the tractor. For uniform motion, the forces \( W, R_v, P_v, \) and \( Q_v \) must be in equilibrium. Knowing the magnitudes and locations of the weight

*For a discussion of safety devices in hitches for trailed implements, see Barger, et al.,¹ pp. 325-331.
TILLAGE FORCE ANALYSIS AND HITCHING

$W$ and the useful soil resistance $R_v$ under the particular operating conditions involved, the first step in analyzing the hitch is to combine these forces graphically into the resultant $AB$ (Fig. 6.3).

The line of pull is established next. It must pass through the hitch point $F$ on the tractor and through the hitch hinge axis selected at $E$. The line of pull and the resultant $AB$ intersect at $G$. The line of action of the support force $Q_v$ is now drawn through $G$, although its magnitude is not yet known. In Fig. 6.3, $Q_v$ is shown with some backward slant to include the rolling resistance of the wheels furnishing the vertical support. If the support were mostly on sliding surfaces, more slant would be needed to include the friction force.

Since $P_v$ must be in equilibrium with $AB$ and $Q_v$, the magnitudes of $Q_v$ and $P_v$ can be determined by moving $AB$ along its line of action to $DG$ and then completing the force parallelogram as indicated.

The top example in Fig. 6.3 represents a desirable hitch adjustment for a moldboard plow, with $Q_v$ located well behind the front wheels so that there is enough weight on the rear wheel for stable operation. The lower example illustrates an extreme condition in
which \( E \) is so high on the plow that \( Q_v \) is about under the front wheels, with practically no weight being carried on the rear wheel. The rear of the tool will be very unstable, particularly when momentary variations in the direction and magnitude of \( R_v \) are considered.

Hitching at too low a point on the tool has the opposite effect. The resultant support force \( Q_v \) is moved toward the rear, thus reducing the weight on the front wheels. Increasing or decreasing the slope of \( P_v \) without changing the location of \( G \) decreases or increases \( Q_v \) but does not change its location.

Although the moldboard plow has been taken as an example, the same principles can be applied to any other trailed tool not receiving support from the tractor. Additional discussion of the soil forces and recommended hitching for a trailed moldboard plow is included in Chapter 7.

In the case of a single-axle implement such as a chisel cultivator, which obtains vertical support only through its wheels, the location of \( Q_v \) is fixed. The line of \( Q_v \) must pass slightly behind the axle centerline (Fig. 6.4) in order to supply torque to overcome wheel-bearing friction and cause rotation of the wheels. Point \( G \) is fixed by the intersection of \( AB \) and \( Q_v \), and the line of pull is through \( G \) and the vertical hitch point \( F \) at the tractor.

**6.8. Horizontal Hitching.** It is not always possible to have the horizontal center of resistance of an implement directly behind the center of pull of the power unit, particularly for narrow implements and wide-tread tractors. In such cases, the alterna-
tives are a central angled pull, an offset straight pull, or an offset angled pull, when referred to the center of pull of the power unit. These conditions are sometimes all referred to loosely as "side draft," but technically, side draft should be considered only as the side component of an angled pulling force.

The central angled pull does not affect tractor steering, whereas the offset pulls do. An angled pull (either central or offset) introduces a side force on the tractor rear wheels, which is sometimes of sufficient magnitude to be objectionable. Some implements are not able to resist the side forces of an angled pull, and for others the effects of an angled pull are undesirable. Usually, some compromise in hitching is best, with part of the adverse effect being absorbed by the tractor and part by the implement. For tools such as the moldboard plow and the offset disk harrow, the lateral location of the center of resistance can be controlled somewhat by design relations. These considerations will be covered in subsequent chapters that deal with specific implements.

6.9. Offset-Trailed, Wheel-Type Implements. Although the discussion in this chapter deals primarily with tillage implements, some consideration may well be given to the horizontal force relations for trailed, wheel-type implements such as field balers, forage harvesters, combines, and corn pickers, which must have cutting or pickup units located to one side of the tractor path.

The force relations for an implement of this type are indicated in Fig. 6.5. Wotja and his associates include in their analysis the effects of angular and translational accelerations, to show the effects of crossing ditches or other surface irregularities and of velocity changes during highway travel. The following discussion, however, applies only for uniform velocity under stabilized conditions.

Summation of the forces in the $x$ and $y$ directions yields the following relations:

$$ P_x = R_4 \sin \theta + (R_1 + R_2) \cos \theta + R_3 \quad (6.1) $$
$$ P_y = R_4 \cos \theta - (R_1 + R_2) \sin \theta \quad (6.2) $$

Taking moments about the hitch point $P$ gives

$$ bR_2 - aR_1 - (f - e)R_3 \sin \theta + dR_3 \cos \theta - fR_4 = 0 $$

from which
\[ R_4 = \frac{bR_2 - aR_1 - (f - e)R_3 \sin \theta + dR_3 \cos \theta}{f} \] (6.3)

where \( P_x \) = horizontal component of pull in the direction of travel.
\( P_y \) = lateral component of pull (side draft).
\( R_1 \) = rolling resistance of inner wheel.
\( R_2 \) = rolling resistance of outer wheel.
\( R_3 \) = resistance of gathering unit (pickup, header, etc.) in the direction of travel.
\( R_4 \) = horizontal skid reaction of soil on both wheels.
\( \theta \) = tire slip angle, with respect to direction of travel.

Fig. 6.5. Horizontal force relations for an offset-trailed implement under conditions of uniform motion.

\[ a, b, d, e, \text{ and } f \] are distances indicated in Fig. 6.5, having positive values as drawn.

Note that the required skid reaction \( R_4 \) becomes less as \( f \) is increased, with resultant decrease in \( P_y \) (and in \( P_x \) if \( \theta \) is of appreciable magnitude). The value of \( \theta \) is affected by the required magnitude of the skid reaction, the nature of the supporting surface, and the design and operating conditions of the tires.
HITCHES FOR MOUNTED IMPLEMENTS

6.10. Design Considerations. Among the factors to be considered in designing or analyzing a system for mounting tillage implements on a tractor are the following: 6,10

1. Ease of attachment and adjustment, and versatility.
2. Uniformity of tillage depth as the tractor passes over ground-surface irregularities.
3. Ability to obtain penetration of the implement under adverse conditions, particularly with tools such as disk harrows and disk plows.
4. Rapidity with which tools such as plows and listers enter the ground.
5. Trailing characteristics of the implement around contours and on side hills.
6. Effect of the implement upon the tractive ability of the tractor (weight transfer).
7. Effect of the raised implement upon the transport stability of the tractor.

6.11. Types of Linkages. Most linkages for attaching mounted implements to tractors may be classified, in regard to their effects in a vertical plane, as (a) single-axis or single-point hitches, (b) two-axis or three-point hitches with vertically converging links, and (c) two-axis hitches with parallel links. Either of the first two types may be laterally rigid, or they may swing laterally (within limits) about a real or a virtual horizontal hitch point. The virtual horizontal hitch point for the second type is obtained by horizontal convergence and pivoting of the two lower links. Parallel-link hitches are found primarily on cultivators and are generally designed to be laterally rigid (as are single-axis hitches intended only for cultivators).

Any of these types of hitches may be operated with the hitch members acting as free links or with the tool supported vertically through the lift mechanism on the tractor (restrained links).

6.12. Single-Axis or Single-Point Hitch. This is the simplest method of attaching a mounted implement. As indicated in Fig. 6.6, it consists of a real hitch point \( F_v \) beneath the body of the tractor, to which the implement is attached. When operated
as a free-link system, the tool is free to rotate about $F_v$ in a vertical plane, any connections farther back being provided merely to lift the implement or to stabilize it laterally.

The force relations for free-link operation are indicated in Fig. 6.6. The analysis is basically the same as for a single-axle trailed implement (Fig. 6.4). The line of pull must pass through $F_v$, with all support ($Q_v$) for the tool being applied through the rear furrow wheel and landside in the example shown. Support is sometimes supplied by a gage wheel running on the surface of the ground, rather than by the furrow wheel. This will move $Q_v$ forward, since the usual position for a gage wheel is considerably ahead of the location of the rear furrow wheel. As in analyzing a trailed implement, $W$ and $R_v$ are first combined into the resultant $AB$. Since the line of $Q_v$ is fixed by the location of the supporting surface, point $G$ is determined by the intersection of the lines of $AB$ and $Q_v$. The line of pull must then pass through $G$ and $F_v$. As before, the magnitudes of $P_v$ and $Q_v$ are determined by shifting $AB$ to $DG$ and completing the force parallelogram. Note that changing the fore-and-aft location of $Q_v$, by moving the gaging device, changes the slope of $P_v$.

For the conditions represented in Fig. 6.6, the hitch point $F_v$ could be a little higher, giving $P_v$ more slope and adding load to the tractor drive wheels. The limit would be with $F_v$ so high that $P_v$ coincided with $AB$, in which case $Q_v$ would become zero. Operation at this limiting condition would be impractical, however, because of momentary fluctuations in the soil forces.
6.13. **Two-Axis Hitch with Vertically Converging Links.**

The three-point or three-link hitch, in which the two lower links resist rotation about the longitudinal axis, is the most common application of this principle. When operated as a free-link system, the convergence of the links in a vertical plane provides a virtual hitch point or instantaneous center of rotation as shown at $F_v$ in Fig. 6.7. An important advantage of this arrangement over the single-axis hitch is that there is no physical interference between the tractor frame and a virtual hitch point. In some cases, however, the power take-off or other parts of the tractor may project so far behind the rear axle that it is difficult to mount the implement up close to the tractor (for stability) without using excessively short links.

With the converging-link system, the location of $F_v$ can readily be changed by modifying the arrangement of the links, and it shifts automatically as the implement is raised or lowered. The broken-line position of the links in Fig. 6.7 illustrates how $F_v'$ is lower than $F_v$ and farther to the rear when the tool is entering the ground. This shift promotes more rapid entry of tools that have appreciable bottom support surfaces (such as a moldboard plow).

As with the single-axis system, the three-point linkage can be in equilibrium as a free-link system only when $P_v$ passes through the hitch point $F_v$. The location of $G$ in Fig. 6.7 is the same as that shown in Fig. 6.6 and is determined in the same manner.

6.14. **Two-Axis Hitch with Parallel Links.**

The two systems already discussed cause the implement to enter the ground with
a steeper-than-normal pitch, which is a desirable and necessary feature for tools such as moldboard plows and listers. Parallel-link systems are not suitable for tools of this type because there is no change in pitch as the height varies. But because the pitch does remain constant, parallel-link hitches are used extensively on mounted row-crop cultivators.

Although the depth of cultivation is most often controlled

![Diagram of parallel-link hitch](image)

Fig. 6.8. Vertical force relations for a parallel-link hitch operated as a free-link system.

through the lifting mechanism (restrained links), gage wheels are sometimes employed in conjunction with free-link operation. With a free-link system (Fig. 6.8), the virtual hitch point is at the intersection of the parallel links or, in other words, at infinity. Thus, $P_v$ with such a system must be parallel with the links and the magnitude of $Q_v$ can be changed (for given operating conditions) by changing the slope of the links.

**6.15. Depth Control with Free-Link Systems.** For stable operation with a free-link (floating) hitch, most tillage implements must have an appreciable area of supporting surface or else be equipped with gage wheels. This is necessary to compensate for fluctuations in the soil force $R_v$. With a moldboard plow the support may be obtained from the bottom of the furrow, through the rear furrow wheel and the heel of the landside, or through gage wheels operating on the surface of the ground.
Listers, cultivators, and other chisel-type tools pulled through free links are generally supported by gage wheels.

When a moldboard plow is pulled by free links and supported on the bottom of the furrow, as in Figs. 6.6 and 6.7, the depth is adjusted by changing the tilt or pitch of the tool. The plow then rotates about the real or virtual hitch point until the supporting surface becomes level at a new depth. The pitch on a single-axis hitch (Fig. 6.6) may be changed by raising or lowering the hitch point $F_v$ or by an adjustment built into the linkage. With a three-point hitch the length of the upper link is generally changed.

**6.16. Operation with Restrained Links.** With restrained-link operation, the tool gets all its vertical support from the tractor, the hitch links being free only when the tool is entering the ground. As soon as a moldboard plow, for example, reaches its working depth, it is held by the hydraulic system. Its landside and rear furrow wheel must have clearance above the furrow bottom so that the plow can go deeper when the controls call for greater depth.

![Diagram](image)

Fig. 6.9. Vertical force relations when a mounted implement is supported by restrained links. $W$ and $R_v$ are the same as in Figs. 6.6 and 6.7.

Since the tool obtains no support from the soil, $P_v$ is merely the resultant of $W$ and $R_v$, as indicated in Fig. 6.9, and the effect of the implement upon the tractor is independent of the type of hitch linkage. It is merely a coincidence that the line of pull passes close to the front joint of the lower links in Fig. 6.9.

When operating with restrained links, the only significance of the real or virtual center of rotation ($F_v$ in Figs. 6.6 through 6.9) is that the line of pull cannot go below this point, since most lifting mechanisms for mounted implements are single acting and
cannot force or hold the tool down (except for hold-down springs sometimes used on cultivators).

Operating with restrained links rather than free links increases the weight on the tractor rear wheels and thus provides greater tractive ability. This is because the support forces that would act upon the gaging units in a free-link system are transferred to the tractor when the links are restrained. But when the implement is suspended in a fixed position with respect to the tractor (as with automatic position control), the depth fluctuations caused by ground-surface irregularities are greater than with a gaged, free-link system.

6.17. Automatic Draft Control. This is a type of restrained-link system, in which the depth of the implement is automatically adjusted to maintain a preselected, constant draft, as explained in Chapter 5 (Section 5.11). Under conditions of uniform soil resistance, depth fluctuations caused by irregular ground surfaces are less with automatic draft control than when the implement is suspended in a fixed position. In a field with nonuniform soil conditions, however, the depth will vary as a result of variations in soil resistance. Thus, in smooth fields the principal advantage of draft control is in maintaining the draft within the available power or tractive ability of the tractor.

Figure 6.10 shows the response of an automatic draft control system to changes of draft in a smooth field when operating a single-bottom lister. The broken-line curve indicates the variability of draft under these particular soil conditions during a run in which the depth was held constant. These results were obtained by taking motion pictures of a depth scale and of dial indicator gages mounted so as to measure deflections and twisting of a specially mounted 1½-in. square tool bar. The curves for the first portion of the automatic-draft test run show the response to a sudden overload of considerable magnitude (which apparently occurred just before the camera was started). During about the first 8 in. of forward travel in this run, the rate of lifting of the implement was limited by the capacity of the hydraulic pump. During the next 6 or 8 in. the implement dropped rapidly to increase the draft and counteract the initial overshoot in control. During the remainder of the run the automatic draft control gradually increased the depth of the implement until a stabilized
condition was reached, presumably at about the preselected control value of draft.

As with any single-acting hydraulic lift mechanism, the rate at which the implement drops is a function of the implement weight and inertia, the soil reactions, and the oil-flow resistance (oil viscosity, amount the valve is open, etc.), and the maximum depth is limited to the condition under which the line of pull passes through $F_v$.

**6.18. Vertical Effects upon Tractor.** When the magnitude and line of action of $P_v$ are known, the effect of the implement on the drive-wheel loading of the tractor can readily be determined. The force relations in a vertical plane are shown in Fig. 6.11. $R_r$ and $R_f$ are the vertical supporting soil reactions upon the wheels, and $R_t$ is the tractive effort. $R_r$ is slightly in front of the rear axle (perhaps 1 to 2 in.) because of the rolling resistance of the tractor. $P_v'$ is the pull of the implement upon the tractor. It is equal and opposite to $P_v$, and may be from either a trailed or a mounted implement. $P_v'$ and $P_a'$ are the vertical and horizontal components of $P_v'$. The distance $y$ is the height above the ground at which $P_v'$ intersects the vertical line.

Fig. 6.10. Response of an automatic draft control system to changes of draft when a single-bottom lister was operated in a smooth field of Yolo fine sandy loam. Forward speed was approximately 4 ft per sec. (Philip R. Bunnelle. Unpublished thesis. University of California, 1950.)
of action of $R_v$. $W_t$ is the weight of the tractor, acting through the center of gravity.

![Diagram](image)

**Fig. 6.11.** Vertical force relations for a tractor when pulling a mounted or trailed implement at a uniform velocity on level ground.

Taking moments about $C_2$ yields the following relation:

$$R_rx_1 - Wtx_2 - P_z'x_1 - P_x'y = 0$$

from which

$$R_r = W_t\left(\frac{x_2}{x_1}\right) + P_{z}' + P_{x}'\left(\frac{y}{x_1}\right) \quad (6.4)$$

Taking moments about $C_1$ gives

$$W_t(x_1 - x_2) - P_{x}'y - R_fx_1 = 0$$

from which

$$R_f = W_t\left(\frac{x_1 - x_2}{x_1}\right) - P_{x}'\left(\frac{y}{x_1}\right) \quad (6.5)$$

These equations indicate that the effect of the implement pull is to add the vertical force $P_z'$ to the rear wheels and to transfer from the front wheels to the rear wheels a weight equal to $P_{x}'y/x_1$, thus increasing the tractive ability.

Instability of the tractor when the tool is in a raised or transport position is sometimes a limiting factor for rear-mounted equipment. This is particularly true for tools that extend a con-
siderable distance to the rear and for tools that require considerable weight to obtain penetration.

6.19. Horizontal Effects of Hitching. In regard to horizontal effects, three common linkage or hitch arrangements for mounted implements are:

1. Implement attached to the tractor through a real hitch point beneath the tractor and free to swing laterally (within limits) about this hitch point.
2. Implement not permitted any lateral movement with respect to the tractor.
3. Three-point hitch with the two lower links converging toward the front and free to swing laterally within limits.

In addition, some implements designed for laterally rigid hitch linkages have pivot points or short converging links incorporated in the tool frame itself.

The first arrangement is sometimes used in conjunction with the single-point hitch shown in Fig. 6.6, whereas the third type is often found in three-point hitches with vertically converging links (Fig. 6.7). Any of the several types of hitches may be designed so that there will be no appreciable lateral movement of the tool with respect to the tractor. It is desirable to be able to lock a laterally swinging hitch so that side movement can be eliminated with tools such as bedders, planters, and some cultivators.

The trailing characteristics of a mounted tillage implement should be such that a reasonably uniform width of cut is maintained when operating around a curve (such as on a contour) without adversely affecting the tractor steering. If a rear-mounted implement is not permitted any lateral movement with respect to the tractor, the tool will cut to the outside when operating on a curve, and steering response may be poor because of the side forces introduced by the tool. These effects are particularly objectionable with a tool such as a plow. A laterally swinging hitch gives easier steering, but the tool ordinarily cuts to the inside on a corner, as explained in the following paragraphs.

Figure 6.12 shows the plan view of a common arrangement for a three-point hitch in which the two lower links converge toward the front and are free to swing laterally within limits. With the tractor operating on a slight curve as indicated, $M$ is the instan-
taneous center of rotation of the two lower links. At first glance one might think that \( M \) would be a virtual hitch point, with a nondirectional tool (one that has little resistance to side movement) tending to move toward it. This, however, is not correct in the arrangement illustrated (front end of top link farther to rear than front ends of lower links) because of the effect of the thrust in the top link, whose line of action does not pass through \( M \).

![Diagram](image)

**Fig. 6.12.** Trailing characteristics with horizontally converging, pivotal links, when operating around a curve.

Considering the horizontal components of the forces acting upon the frame of the tool in Fig. 6.12, \( F_{1h}, F_{2h}, \) and \( F_{3h} \) must be in equilibrium with \( P_h' \). The force \( P_h' \) is equal and opposite to the horizontal resultant of the pull upon the implement, and its line of action passes through \( H \), the horizontal center of resistance of the tool. Knowing the geometry and force relations in a vertical plane, plus the geometry in the horizontal plane as indicated in Fig. 6.12, the resultant direction of pull, \( BHI \), can be determined.

A nondirectional tool tends to move along the line of pull when the tractor is on a curve, the linkage adjusting itself so that \( BHI \) is perpendicular to a radius drawn through \( H \) and the turning center for the tractor. The effect is the same as that of a trailed implement pulled from a point within the small area \( A_1 \) on the tractor centerline. The tool cuts the corner more than it would if \( M \) were the virtual hitch point as is sometimes assumed. The total amount of corner cutting is the distance from \( H \) to arc \( CD \), as indicated in Fig. 6.12.
Some tools, such as a moldboard plow or a cultivator equipped with a fin or guiding coulter, are directional. They tend to go in the direction in which they are pointed, rather than in the direction of pull. In this case, the tool is pulled toward the virtual hitch zone $A_2$, representing the intersection of the tool centerline and the tractor centerline (Fig. 6.12). Since $A_2$ is farther forward than $A_1$, a directional tool cuts the corner even more than a tool that is free to move in the direction of pull.

For ideal trailing of a tool around a curve (no corner cutting), the horizontal hitch point (real or virtual) should be on the tractor centerline, equidistant from the center of resistance of the tool ($H$) and the center of pull of the tractor ($D$). On a side hill, however, where the rear of the tractor tends to slide downhill, the lateral position of the implement will be affected least when the hitch point is well forward on the tractor. A hitch point somewhat forward of the rear axle is also best in regard to ease of steering. Thus a compromise must be made to determine the best over-all location for the horizontal hitch point.

REFERENCES


PROBLEMS

6.1. The line of pull on an implement is 15° above the horizontal and is in a vertical plane which is at an angle of 10° with the direction of travel.

(a) Calculate the draft and side-draft forces for a pull of 2500 lb.

(b) What drawbar horsepower would be required at 3½ mph?

6.2. If the wheels in Fig. 6.4 are mounted on plain, grease-packed bearings with 2½-in. diameter bearing surfaces and the coefficient of friction of the bearings is 0.15, what must be the distance from the wheel centerline to the line of $Q_0$ to cause rotation of the wheel? Draw a free-body diagram of the wheel. Neglect the weight of the wheel, but indicate all other forces and the point of contact between the axle and the hub.

6.3. Take accurate measurements on a tractor with a moldboard plow attached through a three-point, converging-link hitch.

(a) By graphical methods, determine the location of the virtual hitch point when the plow is operating at a depth of 6 in., with a free-link system.

(b) Locate the virtual hitch point when the front plow point is just touching the surface of the ground (as the plow starts to enter the ground).

6.4. Assume that $P_v = 1470$ lb in Fig. 6.7. Determine the force in the top link and the total force in the two bottom links, indicating whether tension or compression. Scale the dimensions and angles from the text, and solve by graphical methods.

6.5. Compare free-link operation with restrained-link operation, in regard to the effect on front and rear wheel loading of a tractor, for the conditions represented by Figs. 6.7 and 6.9. $W$ and $R_0$ for the tool are the same in both cases. Take $P_v = 1470$ lb in each case, since the values are nearly the same under the conditions shown. For free-link operation, the slope of $P_v$ is $11\frac{1}{2}°$ and $y$ (Fig. 6.11) is $6\frac{1}{2}$ in. With restrained links the corresponding values are $20°$ and 19 in. The effective wheelbase of the tractor ($x_1$ in Fig. 6.11) is 76 in.
7.1. Introduction. Although tillage is one of the oldest practices of agriculture and rapid advances have been made during the last century, it is still far from being an exact science. We say that one of the major objectives of tillage is to improve soil tilth. Yet, we cannot even define tilth except by vague statements which recognize that tilth involves both the biological and physical conditions of the soil in relation to plant growth. There is no adequate method available for evaluating the tilth produced by an implement, and the specific tilth requirements for the different crop plants are not definitely established. It is small wonder that conflicting results are obtained from the multitude of tests that have been and will continue to be conducted for comparison of tillage methods in relation to crop yields.

Tillage machinery is sometimes classified as primary and secondary, but the distinction is not always clear-cut. Whereas the plow is basically a primary tillage tool, other implements such as the disk harrow may be used either for primary tillage or for subsequent seedbed preparation. For this reason, no such distinction is made in discussing the various tillage tools in this textbook.

7.2. Functions of Tillage. Even though we cannot establish a definite set of performance specifications that a tillage implement must meet, it might be well to set forth in a general way the following purposes of tillage:

1. To control weeds. In general, this is the most important reason for tillage.
2. To develop a desirable soil structure for a seedbed or a rootbed. A granular or crumby structure is desirable to
allow rapid infiltration and good retention of rainfall, to provide adequate air capacity and exchange within the soil, and to minimize resistance to root penetration. A good seedbed, on the other hand, is generally considered to imply finer particles and greater firmness in the vicinity of the seeds.

3. To facilitate the placement of surface residues. Thorough mixing of trash is desirable from the tilth and decomposition standpoints, whereas retention of trash in the top layers reduces erosion. On the other hand, complete coverage is sometimes necessary to control overwintering insects or to prevent interference with precision operations such as planting and cultivating certain crops.

4. To minimize soil erosion by following such practices as contour tillage, listing, and proper placement of trash.

5. To prepare land for irrigation.

6. To smooth or otherwise prepare the soil surface for harvesting operations.

7. To incorporate and mix fertilizers or soil amendments into the soil.

It should be evident that some of the above requirements are conflicting, with compromises often necessary for a particular situation.

7.3. Mechanics of Tillage. Nichols lists the properties that affect the reactions of soils to applied force as (a) cohesion, (b) adhesion, (c) frictional resistance, (d) resistance to compression, and (e) resistance to shear. The summation of the effects of these items is manifested by the force required to pull an implement through the soil. Nichols has shown that the reactive forces of all classes of soils are dominated by the film moisture on the colloidal particles and thus directly related to the soil moisture content and colloidal content.

Soils may be classified as nonplastic and plastic, the term plastic implying that the soil is moldable within a certain range of moisture contents and that it will retain its molded shape after drying. If a plastic soil is saturated with water and then allowed to dry, it passes through the following stages, in order: sticky, plastic, friable (crumbly), and hard (cemented). The friable stage represents optimum conditions for tillage. Soil compaction by tillage implements and power units, which is a serious prob-
lem in some areas, is promoted by working the soil when too wet. Excessive tillage of a dry soil may result in too much pulverization.

7.4. Plasticity Constants. The plasticity of a soil is commonly expressed in terms of the Atterberg consistency constants, which may be defined as follows: the upper plastic limit is the percentage moisture content (dry basis *) at which the soil will barely flow when jarred by a blow; the lower plastic limit is the moisture content at which the soil can barely be rolled out into a thin, wire-like formation; and the plasticity number is the numerical difference between the upper and lower plastic limits. Nichols 10 expresses the plasticity number as an exponential function of the colloidal content of the soil, whereas Baver 4 relates it linearly to the total clay content. The lower plastic limit shows a moderate increase as the clay or colloidal content is increased; it is affected by the chemical composition of the colloid and is increased by the presence of organic matter. Sands or other soils containing less than 15 to 20 per cent colloids or clay are generally considered to be nonplastic.

7.5. Soil and Metal Friction. Both the life and the draft of plows and other tillage implements are materially affected by soil friction. Nichols 17 determined coefficients of friction by pulling flat pieces of metal across smooth surfaces of uncemented (i.e., not hard) soils uniformly moistened to various moisture contents.

Nichols 17 classifies the friction between a soil and a metal surface into the following distinct phases, each involving a different set of controlling factors:

A. Compression phase. When the water from the soil does not adhere to the metal and when the bearing power of the soil is less than the applied pressure (i.e., the metal sinks in), the coefficient of friction varies with the speed, the unit pressure, and the smoothness and materials of the surface. The occurrence of this phase is generally limited to dry soils not containing any appreciable amounts of colloids.18

* Soil moisture contents are customarily expressed as a percentage of the dry weight, whereas moisture contents of crop materials are ordinarily based upon the wet weight (i.e., the weight of the dry matter plus the weight of the water).
B. Friction phase. When the bearing power of the soil is greater than the applied pressure and soil water does not adhere to the metal, true friction is obtained and it is independent of the moisture content.

C. Adhesion phase. When there is enough moisture present to cause soil adherence to the metal but not enough to have moisture brought to the soil surface, the coefficient of friction increases rapidly with moisture content.

D. Lubrication phase. When there is enough moisture to give a lubricating effect, the coefficient of friction decreases with further increase in moisture content.

The relation of the $B$, $C$, and $D$ phases to moisture content, and the effects of other variables upon the coefficient of friction, are indicated in Fig. 7.1. Identification of symbols is as follows:

\[ M_p = 0.7 P_l \]
\[ M_{np} = 0.13 C + 4.77 \]
(M and $M_{np}$ are also related to the wettability of the metal surface)

\[ \mu_p = 0.40 - 0.0001 H \]
\[ \mu_{np} = 0.24 + 0.005 C - 0.0001 H \]

For chilled iron in $B$ phase, $\mu_p = 0.50$
\[ \mu_{np} = 0.0076 C + 0.28 \]

The $A$ phase is not shown because it depends upon pressure.
SOIL TILLAGE: MOLDBOARD-TYPE TOOLS

\[ C = \text{colloidal content of soil, in per cent.} \]
\[ P_n = \text{plasticity number.} \]
\[ P_l, P_u = \text{lower and upper plastic limits, respectively.} \]
\[ M_p, M_{np} = \text{moisture content for plastic and nonplastic soils, respectively, in per cent (dry basis).} \]
\[ \mu_p, \mu_{np} = \text{coefficient of sliding friction for plastic and non-plastic soils, respectively.} \]
\[ H = \text{metal surface hardness as determined by the Brinell test.} \]

Note that \( \mu \) is independent of moisture content in the friction phase and that for plastic soils \( \mu \) is about independent of the colloidal content (according to Nichols, the division is in the vicinity of \( C = 32 \)). The equations indicated for steels and chilled iron are for surfaces with an ordinary polish. Highly polished steel samples showed no reduction of \( \mu \) for sand, but 5 to 20 per cent reduction for soils having colloidal contents of 16 to 80 per cent.

7.6. Shear Values of Uncemented Soils. Practically all tillage machines consist of devices for applying pressure to the soil, often by means of inclined planes or wedges.18 As the implement advances, the soil in its path is subjected to compressive stresses which, in an uncemented soil, result in a shearing action. The influence of the soil shear value upon the draft of an implement is important, although the exact relation is not known. The shearing of soils is considerably different from the shearing of most solids, in that the reaction may extend for a considerable distance on either side of the shear plane because of the cohesive action of moisture films and the interlocking of particles (internal friction).18

With a laboratory setup whereby shear could be measured at controlled soil pressures, Nichols18 determined shear values over a wide range of moisture contents for various types of soils, using pressures up to 30 psi. He found that for a plastic soil the shear force at a given pressure increased with moisture content up to about the lower plastic limit and then decreased uniformly to a value of zero at the upper plastic limit. The following equation shows the relation of shear force to the plasticity constants,
moisture content, and pressure, for moisture contents below the lower plastic limit:

\[ F_s = 0.06 \frac{M}{P_t} (P_n + 20) + P_c + 0.6 \quad (7.1) \]

where \( F_s \) = shear force, in pounds per square inch.
\( M = \) soil moisture content, in per cent (dry basis).
\( P_c = \) compressive pressure on soil at right angles to shear plane, in pounds per square inch.

Note that for any plastic soil with zero moisture content, \( F_s = P_c + 0.6 \).

The shear force for sand was essentially constant at all moisture contents \( (F_s = 0.7P_c) \), but nonplastic soils with appreciable colloid content showed an increase of shear force with increased moisture, similar to the behavior of plastic soils. The maximum shear force for the nonplastic soils containing colloids was found to be

\[ F_{ms} = 0.2C + 0.7P_c \quad (7.2) \]

where \( F_{ms} = \) maximum shear force, in pounds per square inch.
This maximum occurred at a moisture content of

\[ M = 0.3C + 5 \quad (7.3) \]

It should be emphasized that equations 7.2 and 7.3 apply only for nonplastic soils (up to a maximum \( C \) of about 15 per cent) whereas equation 7.1 is for plastic soils.

7.7. Moldboard Plows. The plow is one of the oldest of all agricultural implements and is generally considered to be the most important tillage tool. Plowing accounts for more traction energy than any other field operation, requiring an estimated 21/2 to 3 billion hp-hr annually. Although yield studies have indicated that under certain conditions with some crops there is no apparent advantage in plowing, the moldboard plow is still by far the most-used implement for primary tillage in seedbed preparation.

Through the years there has been a vast amount of development and research work pertaining to moldboard plows; yet plow-bottom design (as with other tillage tools) is still largely dependent upon cut-and-try methods. Many excellent plow bottoms have been developed, but there are still important soil
types and conditions for which present equipment is not suited. One of the outstanding plowing problems is to obtain a moldboard plow that will scour in the heavy, waxy soils of Texas and the Black Belt of Alabama and Mississippi. Another problem concerns the scouring or at least shedding of the sticky "push" soils common to many sections of the southeastern part of the United States.

7.8. Types of Moldboard Plows. Of the 400,000 plows comprising the total United States domestic shipments in 1952 and 1953, approximately 56 per cent were tractor mounted, 37 per cent tractor drawn, and 7 per cent horse drawn. The popularity of mounted plows has increased tremendously since the advent of hydraulic controls on tractors. Mounted plows generally have from one to three bottoms, whereas common sizes of trailed plows have from two to five bottoms. Thus the ratio between the composite capacities of all trailed and all mounted plows is considerably greater than is indicated by the relative numbers of plows.

Most moldboard plows are designed to turn the furrow slices only to the right. Some plows, however, have two sets of opposed bottoms that can be used selectively. With this arrangement, known as a two-way plow, all the furrows can be turned toward the same side of the field by using the right-hand bottoms for one direction of travel and the left-hand bottoms on the return trip. The two sets of bottoms may be mounted on separate frames so that they can be raised and lowered independently, or they may be on opposite sides of a common frame that is rotated about either a longitudinal or a lateral axis when the plow is raised at the end of the field.

One-way plowing (with a two-way plow) is especially desirable for plowing irrigated lands, since it eliminates back furrows and dead furrows and leaves the field more nearly level. It is also advantageous for terraced fields or contour plowing and for small fields of irregular shape. Tests reported in Chapter 2 (Fig. 2.1) indicate that a higher field efficiency can be obtained with a three-bottom, 12-in., mounted two-way plow than with a trailed one-way plow of the same size (fields laid out in lands when using the trailed plow).

Because two sets of bottoms are needed, two-way plows are more expensive than comparable one-way plows, and two-way
plows with more than two or three bottoms tend to be heavy and cumbersome. However, trailed units with as many as four 18-in. bottoms (per set) are commercially available. About 5 per cent of the total number of moldboard plows mentioned above were of the two-way type (2 per cent mounted and 3 per cent trailed).

7.9. The Plow Bottom. The basic unit of a moldboard plow is the plow bottom. Essentially it is a three-sided wedge with the landside and the under side of the cutting edges of the share as flat sides and the moldboard as a curved side. The primary functions of the plow bottom are to cut the furrow slice, loosen or pulverize the soil, and invert the furrow slice to cover trash.

As with any cutting tool, the moldboard plow bottom requires clearance behind the cutting edge. This is provided by shaping the share so the point is directed downward and toward the unplowed land. The vertical clearance is known as down suction and the lateral clearance as side suction (Fig. 7.2). When there is insufficient down suction or side suction it is difficult to maintain the desired depth or width of cut with the plow bottom. Down suction and side suction should generally be from $\frac{1}{8}$ to $\frac{3}{40}$ in. when there is no rear furrow wheel. If a rear furrow wheel is used, it should be adjusted to give about $\frac{1}{4}$ to $\frac{1}{2}$ in. vertical clearance between the heel and the furrow bottom and about $\frac{1}{4}$ in. lateral clearance between the rear of the landside and the furrow wall.

The size of a moldboard plow bottom is the width of furrow that it is designed to cut. With the standard length of share, the size is the perpendicular distance from the landside to the share.
wing tip. The most common sizes are 12-, 14-, and 16-in., although both larger and smaller widths are available.

In the past, shares have been predominantly of the general type shown in Figs. 7.2 and 7.3a, having a vertical portion, known as the gunnel, that acts as a forward extension of the landside. These gunnel-type shares are removed and resharpened when they become dull. The present trend, however, is toward disposable shares of the types illustrated in Figs. 7.3b and 7.3c. One contributing factor is the gradual disappearance of the local blacksmith and the resulting difficulty in getting gunnel-type shares properly sharpened.

Fig. 7.3. Three types of shares for moldboard plows: (a) gunnel-type, (b) single-piece disposable type, and (c) two-piece disposable type.

Disposable shares are made by merely cutting specially rolled strip stock into the proper length and shape and pressing the attachment bolts into countersunk holes. Since their initial cost is perhaps one-third that of the gunnel-type shares, they can economically be replaced when dull or worn, rather than being resharpened. Because the point wears much faster than the rest of the share, some disposable shares are two-piece units (Fig. 7.3c). Moldboard shins are also replaceable on many plows, since this is another part of the cutting edge and is subject to considerable wear.

7.10. Materials for Moldboards and Shares. Soft-center steel is widely used because it gives a smooth, hard surface that scours well in sticky soils such as clay and clay loams. Because of the soft inner layer of steel, plow bottoms made of this material are tough and can withstand considerable shock without breaking. Some manufacturers obtain similar characteristics by
carburizing a low-carbon steel on both sides. Chilled cast iron is the usual choice for sandy soils because of its resistance to abrasion and wear. Chilled cast-iron plows are lowest in first cost but are heavier and more brittle than steel and do not scour as well in sticky soils. Gunnel-type shares of solid steel (about SAE 1080 steel) have a lower first cost than soft-center steel shares and are often preferred for nonsandy soils that scour readily. Disposable shares are generally made from a carbon steel in the range of SAE 1090 or 1095. Specifications for marking plow-shares and other soil-working shapes to indicate the type of material have been set up by the ASAE.\textsuperscript{51}

Steel shares (solid or soft-center) are sharpened by forging, but chilled shares must be either ground or replaced. Hard-facing alloy materials are often added to the cutting edges of steel shares to reduce the rate of wear. Field tests on Kansas farms\textsuperscript{6} indicated that a gunnel-type solid-steel share could be used for 8 to 10 acres per sharpening and could be sharpened from four to seven times before being worn out, whereas a similar share could be hard-faced three or four times and would plow 40 acres per treatment. Resharpened soft-center shares, unless properly heat treated after each sharpening, were found to wear very little better than solid-steel shares. The heat-treating problem is not encountered with disposable shares, since they are not resharpened.

7.11. Types of Moldboards. Because soil types and plowing conditions vary widely, many different shapes of moldboards have been developed. The most common general types are the sod or breaker bottom, the stubble bottom, the general-purpose bottom, and the blackland bottom. There are many variations of shape within each of these general classifications.

The sod bottom has a long, low moldboard with a gradual twist that completely inverts the furrow slice with a minimum of breakup, thus covering vegetative matter thoroughly. It is used for plowing virgin sod or land that has been idle for a long time. A stubble bottom has a relatively short and broad moldboard that is curved rather abruptly near the top, resulting in a greater degree of pulverization than with other types (Section 7.16). The general-purpose bottom lies in between these two extremes and is suitable for a wide range of conditions. Special shapes of general-purpose bottoms have been developed for plowing
efficiently at relatively high speeds. The blackland bottom has a relatively small moldboard area, and its shape tends to promote scouring in soils such as the heavy blacklands of Texas. A less common type is the slat moldboard, which has portions of the moldboard cut out and is sometimes used in extremely sticky soils.

7.12. Expression of Moldboard Shapes. In comparing and analyzing the performance of different plow bottoms, it is desirable to have some significant system for identifying a particular moldboard shape. Several methods of expressing the shape mathematically or identifying it by empirical measurements have been devised. White reported in 1918 that many plow bottoms (particularly forged types) have surfaces that can be fitted by the general relation

\[ \frac{x^2}{a^2} + \frac{y^2}{b^2} - \frac{z^2}{c^2} = 1 \]  

(7.4)

which is the equation for a hyperboloid of one sheet. Surfaces of this shape contain two sets of intersecting straight lines, none of which are parallel. White found that this equation could be fitted equally well to extreme shapes such as sod bottoms and stubble bottoms, the only difference being in the values of the constants \( a \), \( b \), and \( c \).

![Fig. 7.4. Three-dimensional view showing method of measuring moldboard surface by moving point \( A \) of the arc of a circle along the \( x \) axis and rotating the arc (as indicated by angles \( \varphi \) and \( \theta \)) to fit the moldboard surface, or by moving some point \( B \) of the chord along a line \( CD \) directly above the \( x \) axis and rotating the arc as required. The arc is always in a vertical plane.](M. L. Nichols and T. H. Kummer. *Agr. Eng.*, November, 1932.)
Ashby, in 1931, proposed a set of nine standard measurements or indices of plow-bottom shape. Nichols and Kummer, in a study involving twenty-two typical plows of various types, found that the entire surfaces of all plows studied could be covered by arcs of circles moved along and rotated about the line of travel of the share wing tip or about a parallel line directly above the wing tip (Fig. 7.4).

Reed has identified plow-bottom shapes by measuring horizontal contours at 1-in. vertical intervals and plotting them on a plan view of the bottom, as illustrated in Fig. 7.5.

Fig. 7.5. Plan views of two 14-in. general-purpose plow bottoms, showing contours for 1-in. vertical intervals. (a) Conventional moldboard with warped surface. (b) Moldboard with cylindrical surface and disposable share. (I. F. Reed. Agr. Eng., March, 1941.)

7.13. Attachments for Moldboard Plows. Rolling coulters are employed to help cut the furrow slice and to cut through trash that might otherwise collect on the shin or beam and cause clogging. A coulter is usually mounted with its axis directly above or a little behind the point of the share and with the disk blade ½ to ¾ in. to the land (i.e., beyond the landside, toward the unplowed ground). The coulter depth under usual conditions should be about half the depth of plowing but may need to be greater than this in sod and less in hard ground. A large-diameter coulter goes through heavy trash better than a smaller one but does not penetrate hard ground as readily. Diameters of 15 to 18 in. are typical for 12 to 16 in. bottoms. A notched coulter cuts trash better than a smooth one of the same diameter but wears more rapidly.

A stationary jointer is a miniature plow bottom that cuts a narrow, shallow furrow ahead of the shin. Its function is to
move the trash and roots from this strip over toward the main
furrow in such a manner as to insure complete coverage by the
main plow bottom. The point of a jointer should be above the
share point and \( \frac{3}{8} \) to \( \frac{5}{8} \) in. to the land. The usual operating
depth is \( 1\frac{1}{2} \) to 2 in. The most common application of a jointer
is in conjunction with a rolling coulter, as shown in Fig. 7.6,

![Fig. 7.6. A mounted plow equipped with a coulter-jointer combination. Rotating the coulter standard as indicated changes the lateral position of the combination. (Massey-Harris-Ferguson, Inc.)](image)

where complete coverage of heavy trash is desired. In this
combination, the clearance between the jointer and the coulter
should be not more than \( \frac{1}{4} \) in. at the point and from \( \frac{3}{8} \) to \( \frac{1}{2} \) in.
at the top. A stationary jointer without the coulter is sometimes
used in stony or hard ground but is unsatisfactory in heavy trash.

Disk jointers, which are concave disks set at an angle to the
direction of travel, are sometimes used in place of the coulter-
stationary-jointer combination. Weed hooks or covering wires
are occasionally employed to fold tall plant growth over just in
front of the moldboard to improve coverage.

Conditions. The wide range of soil conditions encountered in
tillage materially affects the reactions of the soil on moldboard
surfaces. Nichols and Reed classify the various soil conditions
and describe the reactions as follows:
1. **Hard cemented soils.** These soils break into large irregular blocks ahead of the plow, with no definite pattern to the soil reactions.

2. **Heavy sod.** Because of the surface reinforcement from the mass of roots, normal shear planes are not generally observed. However, normal soil reactions occur below the sod.

3. **Packed or cemented surface.** This is a rather unusual situation, with relatively loose soil immediately beneath the packed surface. Blocks of the surface layer are broken loose irregularly and lifted like boards.

4. **Freshly plowed soil.** In this condition, the soil has insufficient rigidity and compressive resistance to allow the plow to function properly.

5. **Push soils.** These soils, when settled, act somewhat similarly to freshly plowed soil. Adhesion of the soil to the moldboard builds up a pressure ahead of the plow bottom, which, because of insufficient rigidity in the furrow slice, causes the soil to be pushed to one side rather than being elevated and turned.

6. **Normal soil condition.** The soil has settled and reached a firm condition, primarily as a result of natural agencies, and is within its proper moisture range for good plowing.

### 7.15. Reactions under Normal Soil Conditions

At the ordinary speeds of plowing, the movement of soil on the moldboard is due to the resistance to compression of the soil ahead of the plow, and the average speed of movement of the soil across the moldboard would be expected to approximate that of the plow. With the exception of a change in apparent volume due to pulverization, there is no evidence of any appreciable dimensional change of the furrow slice in passing over a properly functioning plow, although there are, of course, changes in direction and acceleration of the soil particles.

In general, the moldboard surface (including the share) can be divided functionally into three sections: 10 (1) the lower, or share portion, which forms a wedge for breaking the soil loose, (2) a central area where most of the pulverization takes place, and (3) a turning and inversion area on the upper part of the moldboard. Pulverization may be largely confined to one portion of the plow surface, or it may extend over most of the mold-
board area. Lifting and turning extend throughout the length of the plow, but the rate of turning generally increases on the upper and rear parts of the moldboard.

7.16. Pulverizing Action. As a plow moves forward, its double wedging action exerts pressure both upward and toward the open furrow. Nichols and Reed \(^{20}\) found that the stresses set up by this action cause blocks of soil to be sheared loose at regular intervals on parallel, inclined shear planes. These primary shear planes extend forward from the plow point at an angle of about \(45^\circ\) in both the horizontal and vertical planes, retaining their relative positions as they move across the moldboard surface (Figs. 7.7 and 7.8). Note in Fig. 7.8 that, toward the rear of the moldboard, rotation of the furrow slice has caused the primary shear planes to become about parallel with the undisturbed furrow wall. The soil blocks formed by the primary shearing action break down as they move up the moldboard, forming secondary shear planes at right angles to the primary shear planes. Further pulverization is caused by the sliding of these soil blocks over each other.

Nichols and Kummer \(^{19}\) show that pulverization is promoted if the moldboard curvature is such as to produce simultaneous movement on all primary shear planes (Fig. 7.7) as the plow moves forward. They demonstrate that theoretically this involves uniformly accelerated motion of the soil in the direction of the shear planes and is accomplished by a surface whose vertical profiles in planes parallel to the landside have the mathematical form

\[
 z = ae^{bx} \tag{7.5}
\]

where \(z\) and \(x\) are the vertical and longitudinal coordinates, \(a\) and \(b\) are constants, and \(e\) is the base of Naperian logarithms. This equation indicates an increasing rate of curvature from front to rear of the moldboard. Measurements on a number of typical,
successful plows showed that vertical profiles in the central or pulverization areas of all moldboards studied could actually be fitted by equations of this type. The lower, or front, portion was steeper than indicated by this equation, in order to obtain strength and suction. Because of inversion requirements, the upper portion was also steeper than the equation would indicate.

Fig. 7.8. Moldboard plow in moist, sandy soil, showing the location of primary shear planes in the furrow slice. The effect of secondary shear planes can be seen in the turned furrow slice. (M. L. Nichols and I. F. Reed. *Agr. Eng.*, June, 1934.)

7.17. Turning and Inversion. Since the cutting edge of the share is normally at an angle of about 40 to 50° from the direction of travel, elevation of the shin-side of the furrow slice begins before elevation of the wing-side. Thus, turning and inversion start immediately. Most of the turning, however, is accomplished by the upper part of the moldboard. The final action is to push or throw the soil from the upper moldboard into the preceding furrow, the amount of throw depending largely upon the forward speed and the direction of release of the soil. The inversion of the soil and the accompanying forward movement during plowing are indicated in Fig. 7.9 for typical conditions, as determined by Ashby and reported by Nichols and Reed. Nichols and Kummer found that in a group of typical plows
the turning and inversion of the furrow slice on the upper moldboard was accomplished by uniform-pressure curves similar in principle to those used in highway and railway engineering. As mentioned in Section 7.12, they were able to describe moldboard surfaces by moving and rotating arcs of circles (Fig. 7.4). They

![Fig. 7.9. Movement of soil during plowing, as determined by placing small blocks in the unplowed furrow slice and noting their final positions in the turned soil. (M. L. Nichols and I. F. Reed. Agr. Eng., June, 1934.)](image)

found that with a plow designed for proper inversion of tough or sticky soils, one end of the arc moved along the $x$ axis (line through share wing-tip, parallel to landside) as the arc was rotated. For general-purpose plows it was common to have a point on the chord move along a line directly above the $x$ axis (as point $B$ in Fig. 7.4). With stubble plows the upper part of the moldboard is swept out by a second arc having a smaller radius than the arc used for the pulverization area.

7.18. Scouring of Moldboard Surfaces. Scouring of moldboard plows in sticky soils is one of the most difficult problems
that has confronted plow designers. Many different moldboard shapes have been tried, as well as a number of different moldboard materials. Bacon reported in 1918 that moldboards coated with plaster of Paris and those covered with hog hides scoured better in Texas soils than any other type of moldboard tried (including steel, iron, glass, brass, and aluminum). He also cites instances in which heat apparently improved the scouring of plow bottoms.

Kummer found that when wood slats (maple or beech) impregnated with paraffin or linseed oil were used in place of steel slats on a slatted moldboard, scouring in clay soils was improved considerably, especially in the higher moisture ranges where adhesion-phase friction (Section 7.5) was extremely evident. Under these soil conditions, sliders made of the impregnated wood gave coefficients of friction only one-half to two-thirds as great as soft-center-steel sliders. Kummer states that the improved scouring of the treated wood was probably due to less adherence of water to the wood surface than to steel.

Kummer's studies also included tests in which the solid moldboard was replaced by endless belts and by a series of wooden rollers. More recently, Skromme reports the development of a belt-type moldboard to obtain scouring in clays of volcanic origin on which pineapples are grown in Hawaii. This plow cuts a furrow 32 in. wide and 13 in. deep and is operated at $2\frac{1}{2}$ mph. Ordinary plows do not scour in this soil because of its low shear strength and very high adhesive properties.

7.19. Effect of Moldboard Shape on Scouring. It has been stated by various investigators that uniform scouring depends upon uniform soil pressures over the entire moldboard surface. On the basis of a mathematical study of the forces acting upon the surface of a moldboard, Doner and Nichols show that high curvature in the path of soil travel near the share increases the tendency for the soil to stick to the moldboard, whereas an increasing curvature from front to rear tends to equalize soil forces and thus improve scouring. Their analysis, together with general field observations, indicates that failure to scour would be expected to originate at any point or area having an excessively high coefficient of friction or excessive curvature, i.e., at a rough, unpolished spot or at a concavity or bend
in the surface. Cast moldboards are more likely to have surface irregularities than are forged steel bottoms.

7.20. Forces Acting upon a Plow Bottom. As defined in Chapter 6 (Section 6.2), the useful soil forces acting upon a moldboard plow bottom are those resulting from the operations of cutting, pulverizing, lifting, and inverting the furrow slice. These soil forces practically always introduce a rotational effect on the plow bottom. Parasitic forces include those that act upon the side and bottom of the landside (including friction), as well as the rolling resistance of support wheels.

In the following discussion, the force $R$ and its components $L$, $S$, $V$, $R_h$, and $R_v$ refer to the resultant of all useful soil forces. The term $Q$ indicates a parasitic force, whereas $P$, $P_v$, and $P_h$ (or draft) include the effects of both useful and parasitic soil forces and the weight of the implement. These terms are defined more completely in Chapter 6 (Section 6.6).

7.21. Horizontal Forces. Figure 7.10a shows the typical position of $R_h$ for general-purpose plows with shares in good condition, based on field tests with the Pennsylvania State College tillage meter over a wide range of soil conditions. $S$ is shown as 24 per cent of $L$ in this typical situation. In tests performed at the USDA Tillage Machinery Laboratory with general-purpose bottoms, the $S/L$ ratio was about twice as great in a fine sandy loam as in a clay loam.*

In the USDA tests a rolling coulter was mounted ahead of the plow, but, unfortunately, the forces acting upon the coulter were not included in the measurements. Thus the actual magnitudes of the $S/L$ ratio cannot be compared directly with the Pennsylvania State College field results. It should also be kept in mind that the USDA laboratory tests are not intended to necessarily represent field conditions.† The results do, however, indicate that all three of the soil-force components for the plow bottom ($L$, $S$, and $V$) increase as the speed is increased.

If the soil were perfectly uniform, the plow could theoretically

*The descriptions of soil texture for the Tillage Laboratory tests discussed in Chapters 7 and 8 were suggested by Mr. I. F. Reed, Senior Agricultural Engineer, USDA Tillage Machinery Laboratory, in personal correspondence.

†The USDA Tillage Machinery Laboratory and the Pennsylvania State College Tillage Meter are described in Chapter 6.
be pulled (with no side force on the landside) by a force whose horizontal component was equal and opposite to $R_h$ (Fig. 7.10). When the horizontal pulling force is in the direction of travel (as $P_x$ in Fig. 7.10a), a parasitic side force is introduced on the landside. $Q_h$ is the resultant of this side force and the accompanying friction force on the landside. Taking a typical value of 0.3 for the coefficient of friction, the horizontal component of pull $P_x$ is, in this example, greater than $L$ by 0.3S. A pull angled to the left, as in Fig. 7.10b, increases the landside force and would theoretically increase $P_x$.

Point $H$, the intersection of $R_h$ and $Q_h$, represents the horizontal location of the center of resistance of the plow bottom. Tests have shown that a long landside (Fig. 7.10c) moves the resultant landside force $Q_h$ to the rear, thus relocating $H$ farther back and closer to the landside (since the line of $R_h$ does not change). Taking most of the side force on a rear furrow wheel has a similar effect. Soil conditions and other factors also cause $R_h$ (and $H$) to vary somewhat from the position shown.
7.22. Vertical Forces. A moldboard plow bottom by itself generally has a downward-acting vertical component of the useful soil force (suction), but rolling coulters must always be forced into the ground (Fig. 7.11a). The $V$ component for the combination may be either up or down, as indicated in Fig. 7.11b, depending primarily upon the resistance of the soil to coulter penetration. According to Clyde,7 a downward $V$ of 100 lb (for plow plus coulter) with $L = 700$ to 800 lb (for a 14-in. plow) is common under good plowing conditions (10 to 12 per cent moisture). In the USDA tests24 the $V/L$ ratio for the plow bottom alone (a coulter was used but its force was not measured) was considerably greater (i.e., more suction) in the fine sandy loam than in the clay loam.

With worn shares in sod having 24 per cent moisture, Clyde9 found that $V$ for the plow alone was practically zero, as compared with 130 lb downward for good shares in the same soil.
Thus, considering the upward component of pull exerted by the power source, badly worn shares plus rolling coulters may prevent penetration of a normal-weight plow in hard ground. Under hard conditions it would be helpful to set the coulters very shallow or remove them entirely. Jointers may be substituted without incurring adverse effects in regard to penetration.

**7.23. Couples.** A right-hand plow bottom by itself is generally subjected to a counterclockwise couple as viewed from the rear. The highest value reported by Clyde is about 2000 lb-in. A rolling coulter decreases this couple, and in hard ground changes it to clockwise (up to 1100 lb-in. has been observed). In a complete plow, this rotational effect is counteracted primarily by parasitic support forces on the wheels or the bottom surfaces of the plow.

**7.24. Horizontal Hitching of Trailed Moldboard Plows.** General considerations in regard to the hitching of tillage imple-

![Fig. 7.12. Recommended horizontal hitching for a wide tractor. (R. J. McCall and A. W. Clyde. *Pennsylvania Agr. Ext. Circ. 259.*)](image)
the rear of the share. Other writers suggest a distance from the landside of one-fourth the width of cut.

The ideal hitch is obtained when the tractor tread can be adjusted so that the center of pull (a point approximately midway between the rear wheels and slightly ahead of the rear axle) is directly ahead of the center of resistance. In some cases, however, such a narrow wheel setting cannot be obtained or is not practical, especially for tricycle-type tractors on rough or rolling ground. Fortunately, a moldboard plow will operate satisfactorily even when the line of pull is at a considerable horizontal angle from the line of travel (Fig. 7.10b), although slightly better results are obtained if the pull is straight ahead. With a wide-tread tractor it is common practice to divide the effects of the offset, as indicated in Fig. 7.12, so that the line of pull (established by points H and F) passes just a little to the right of the center of pull of the tractor but not far enough over to cause steering troubles.

7.25. Vertical Hitching. As pointed out in Section 6.7 and illustrated in the upper view of Fig. 6.3, proper vertical hitching of a trailed plow places the resultant support force \( Q_u \) well behind the front wheels so that the rear wheel will carry a reasonable amount of weight. This condition is necessary for stable operation and uniform depth of the plow.

Clyde found that, over a wide range of soil conditions, \( R_v \) for a plow bottom with a rolling coulter passes through a common locality almost directly above the plow point and a little below the ground surface (Fig. 7.11b). He points out that the center of gravity of trailed tractor plows is usually somewhat ahead of the average location of all share points on the plow but suggests that the location over the plow point (or average position of all points) and just below the ground surface is a good trial point for preliminary adjustment of the vertical hitch. This is not the same point used for horizontal hitching. The point commonly used for the center of resistance in vertical hitching of a walking plow is not correct for most trailed tractor plows because walking plows are seldom equipped with rolling coulters and their center of gravity is farther to the rear.

7.26. Draft of Plows. Because of the tremendous amount of energy consumed each year for plowing, a great amount of work has been done in evaluating the various factors that affect the
draft of a plow and investigating possible means for reducing the energy requirements. It is common practice to refer to the draft of plows in terms of pounds per square inch of furrow cross-section. In the following discussion this quantity will be designated by the term unit draft, to distinguish it from the total draft of the plow. The unit draft of plows varies widely under different conditions, being affected by such factors as the soil type and condition, plowing speed, shape of moldboard, sharpness of share, depth of plowing, width of furrow slice, type of attachments, and adjustment of the plow and attachments.

Tests in a sandy loam soil in good plowing condition showed that, with a trailed plow at $2\frac{1}{2}$ mph, approximately 50 per cent of the total draft was required to cut the furrow slice, 30 per cent for elevating, pulverizing, and inverting the soil, and 20 per cent for overcoming the rolling resistance of the plow wheels. In these tests, increasing the speed to 4 mph increased the draft for elevating, pulverizing, and inverting the soil so that it became 50 per cent of the total, whereas the magnitudes of the other components did not change. The draft requirements for cutting the furrow slice were determined by noting the reduction in total draft when the slice was precut with a special L-shaped knife. It was noted that the draft of the slice cutter did not increase with speed.

7.27. Effect of Soil Type and Condition. Soil type and condition are by far the most important factors contributing to variations in unit draft. Values of unit draft range from 2 to 3 psi for sandy soils up to 15 to 20 psi for heavy gumbo soils. Sandy or silt loams may have unit drafts from 3 to 7 psi, whereas 6 to 12 psi would be typical for clay loams and heavy clay soils.

Soil moisture content is an important factor in regard to both draft and quality of work. A dry soil requires excessive power, in addition to accelerating wear of the cutting edges. For example, Ashby and his associates found that a rainfall of 1.3 in. on a fairly dry clay loam reduced the unit draft by about 40 per cent. In tests at the USDA Tillage Machinery Laboratory, an increase of moisture content from 9.1 to 11.7 per cent reduced the unit draft in a fine sandy loam by 15 to 35 per cent.

Other pertinent soil factors include the degree of compaction, the previous tillage treatment, and the type or absence of cover
crup. Ashby found that compaction of a moist clay loam by the tractor and by equipment employed in preparatory tillage operations definitely increased the draft. Laboratory tests in a fine sandy loam showed a 15 to 35 per cent increase of draft when the apparent specific gravity of the soil changed from 1.68 to 1.83.

The effect of soil variations upon draft and the difficulty in maintaining a uniform width and depth of cut under field conditions make it rather difficult to obtain consistent results from field tests without a large number of replications. The arrangement of the USDA Tillage Machinery Laboratory permits more accurate control of many of the variables encountered in field work.

7.28. Effect of Depth of Plowing. Since the force for cutting the bottom of the furrow slice is independent of depth, one would expect the unit draft to become less as the depth is increased. Ashby and his associates found an average reduction of 14 per cent in unit draft when the plowing depth with 12-in. to 18-in. plow bottoms was increased from 6 in. to 8 in. in a clay loam. Ocock found a decrease of about 17 per cent between 4-in. and 8-in. depths in a brown silt loam. In tests at the USDA Tillage Machinery Laboratory (landside removed and coulter used but its draft not included) the minimum $L$ for a 14-in. general-purpose bottom was with a plowing depth of 5 to 7 in. The increase of $L$ at greater depths was explained as probably being due in part to choking of the thick furrow slice in the curvature of the mold.

7.29. Effect of Width of Cut. Results reported for a limited number of laboratory tests in a sand indicate that, for this one soil condition, widths of cut from 8 to 16 in. for a 12-in. bottom and from 8 to 20 in. for a 16-in. bottom had little effect on the magnitude of the unit draft of the plow bottom alone (landside removed). However, since the total components of draft due to the rolling coulter, the landside friction, and the rolling resistance of plow wheels (which were not included in the measured forces) would change very little for different widths of cut, these factors would cause a marked increase in unit draft at the narrower cuts.

7.30. Effect of Moldboard Shape. The shape of the moldboard has a definite influence on the draft, although the relative
effects are apparently influenced by other factors. For example, Ocock 22 found that in silt loam a sod bottom had a higher unit draft than a stubble bottom, whereas Reed 27 found the opposite effect in laboratory tests with sand and fine sandy loam soils but a similar effect in a stiff clay loam. Reed explains the greater draft of the sod bottom as probably due to the large area of moldboard in contact with the soil and to the under edge of the wing being lower. Stubble bottoms tend to have a higher draft than general-purpose bottoms, because of the more abrupt curvature and the resulting greater pulverization. In general, shapes that give the best trash coverage or the greatest degree of pulverization tend to have the highest drafts, although the converse is not necessarily true. Reed found distinct differences of unit draft between various plow bottoms within the general-purpose classification.

7.31. Effect of Attachments. Ashby and his associates 2 found that with 16-in. and 18-in. bottoms in sandy loam and clay loam, removing the jointer from a jointer-coulter combination reduced the draft by about 7 per cent. Two 10-ft covering wires absorbed about 2 per cent of the total power. Clyde’s force diagram for typical conditions (Fig. 7.11a) indicates that about one-sixth of the total draft is absorbed by the rolling coulter. Tests showed that the total draft of a 16-in. plow equipped with an experimental disk jointer was about 12 per cent less than when using a rolling coulter plus a conventional stationary-type jointer.8 Little information is available to show the relative drafts of a plow-coulter combination as compared with a plow without a coulter. Although various investigators 14,25 have stated that a rolling coulter frequently decreases draft, limited tests by Clyde * in a firm sod indicated a slight increase in total draft when the coulter was used.

7.32. Effect of Rear Furrow Wheel. Comparative tests in loam soils 5,6 indicate some reduction in draft (perhaps 5 to 7 per cent) by taking most of the side thrust on the rear furrow wheel rather than all on the landside. In tests with a 14-in. general-purpose bottom under controlled soil conditions at the USDA Tillage Machinery Laboratory,24 removing the landside entirely and putting all of S on the test equipment reduced the

* Reported in personal correspondence.
draft of the plow bottom alone by 30 to 40 per cent in a sand
and by about 20 per cent in a fine sandy loam. In the USDA
tests the rolling coulter was used but its draft was not included
in the measurements. The percentage reductions would be less
if the draft of the coulter were included.

7.33. Effect of Speed upon Draft and Performance. The
draft of a moldboard plow includes a range of components, from
those that are relatively independent of speed (such as the force
to cut the furrow slice) to soil-accelerating forces that would
theoretically vary as the square of the speed. McKibben and
Reed\textsuperscript{15} have consolidated the many test results reported by
various investigators during the period from 1919 to 1949
and have plotted the per cent increase in draft as a function of speed,
taking the draft at 3 mph as 100 per cent in each case. These
data include several hundred runs distributed throughout the
speed range from 1 to 6 mph, as well as a considerable number
of runs at speeds as high as 7 to 8 mph. For most of the plotted
points, the calculated ratio $D_s/D_3$ is within ±20 per cent of the
value indicated by the empirical relation

$$D_s/D_3 = 0.83 + 0.0189s^2$$

(7.6)

where $D_s =$ draft at speed $s$.

$D_3 =$ draft at 3 mph.

$s =$ speed, in miles per hour.

The form of equation 7.6 was arbitrarily selected to include two
draft components, one being independent of the speed and the
other representing the effect of soil-acceleration forces. This
equation indicates that the average increase of draft between 2
and 4 mph is only 25 per cent, whereas the increase between 3
and 6 mph (doubling the speed in each case) is 51 per cent.
Thus, future increases in normal plowing speeds will involve con-
siderably more sacrifice in power economy than have past in-
creases.

It should be recognized that operation at higher speeds with a
given plow undoubtedly results in greater pulverization of the
soil, so that the increased power input is not entirely wasted.
Although a plow bottom can be designed to give the desired
pulverization and placement of the furrow slice at any selected
speed, the performance will be different and perhaps unsatisfac-
tory if the operating speed is greatly changed from the design speed (for example, operation at 2 mph when the plow is designed for 10 mph).

7.34. Listers. A lister or middlebreaker is basically the same as two opposed moldboard plow bottoms placed back to back, with the landside eliminated and soil being thrown to both sides. Various sizes (measured between outside wing tips of the lister share) are available, 14-in. being a common size. In a lister the side components of the useful soil forces balance each other so that the resultant useful force lies in a vertical plane in the direction of travel. Since there is no landside to stabilize the lateral forces, the pull must be approximately in the direction of travel. Normally no rotational effect would be expected. Lister bottoms are sometimes mounted on trailed tool carriers but more often are attached to tractor tool bars. Depth-gage wheels are frequently employed in the tractor-mounted arrangement.

In some areas crops such as corn, soybeans, and cotton are planted with listers having planter attachments that place the seed in the soil at the bottom of the furrow. Where lister planting is practical, the labor and power requirements for this operation are considerably lower than for surface planting of corn on plowed land. Weeds are more easily controlled (by covering them as the corn grows) and erosion is reduced when lister planting is on the contour. However, because seed planted in the bottom of the furrow may be in cool, wet soil without previous seedbed preparation, poor stands are sometimes obtained.

Contour listing is often done for erosion control, and beds for planting vegetable crops may be preformed with listers. In semiarid areas grain land is sometimes blank listed instead of being plowed. The ridges are later split out or “busted” open when the grain is planted. This method of double listing is considerably faster than plowing and requires less energy per acre.

7.35. Moldboard-Type Ditchers. Tools similar to listers but with much larger moldboards are available for making ditches for the distribution of irrigation water or for drainage. The bottom width of such ditchers is usually in the order of 12 to 16 in., but corresponding top widths (between outer extremities of moldboard wings) may be as great as 36 to 60 in. Ditchers of this size generally are mounted on trailed tool carriers.
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REFERENCES


PROBLEMS

7.1. Analysis of a medium clay soil showed 49.6 per cent colloids and plasticity limits of 21.0 and 36.4 per cent. A fine sandy loam (nonplastic) had 15.5 per cent colloids.

(a) With each soil, compare coefficients of sliding friction in the \( B \) phase for chilled iron and two soft-center steels having Brinell hardnesses of 250 and 600.

(b) For each soil, compute the maximum coefficient of friction (\( C \) phase) and the moisture content at which it occurs.

7.2. Plot complete curves of soil shearing force versus moisture content for pressures of 5, 10, and 15 psi, using the medium clay soil of Problem 7.1.

7.3. Taking Collins' division of power between the various operations of plowing at \( 2 \frac{1}{2} \) mph, determine the probable per cent decrease in unit draft if the plowing depth with a 14-in. bottom is increased from 6 in. to 8 in. Assume that the cutting force is proportional to the length of perimeter cut and that the total energy required for elevating and turning is proportional to the amount of dirt moved (i.e., furrow cross-section). Compare your results with Ashby's findings as reported in the text.
7.4. (a) By what percentage is the draft \( P_h \) of a plow bottom increased if the horizontal component of pull is at an angle of 10° from the direction of travel rather than being straight ahead (Fig. 7.10, b versus a)? Assume \( R_h \) is at an angle of 12° from the direction of travel and the coefficient of friction on the landside is 0.33.

(b) By what percentage is the perpendicular force on the landside increased? Graphical solution is suggested, with the scale at least four times that of Fig. 7.10.

7.5. The total draft of a four-bottom 16-in. moldboard plow when plowing 7 in. deep at 3.5 mph was 3400 lb.

(a) Calculate the unit draft in psi.

(b) What is the actual horsepower requirement?

(c) If the field efficiency is 75 per cent, what is the rate of work in acres per hour?

7.6. Determine the proper wheel tread for a tractor to pull a four-bottom 14-in. plow without side draft. Assume 13-in. rear tires and a 1-in. clearance between the tire and the furrow wall. Show pertinent dimensions on a diagram.

7.7. (a) Assuming the average relation between draft and speed given in Section 7.33, determine the most economical speed (minimum cost per acre) for plowing under the following conditions: four-bottom 14-in. plow operating at a depth of 6 in. in a soil that has a unit draft of 7 psi at 3 mph; tractor loaded to 75 per cent of its rated horsepower; field efficiency = 80 per cent; labor charge for operator = $1.00 per hour; energy charge = 5¢ per hour per rated horsepower; implement charge = 28¢ per hour + 24¢ per acre (the 24¢ is for sharpening and replacing shares). Suggested procedure: Set up an algebraic relation between speed and cost per acre, and then differentiate to obtain minimum cost.

(b) How would this speed be changed by increased labor costs? How would it be changed by increased energy costs?
8.1. Standard Disk Plows. The standard disk plow consists of a series of individually mounted, inclined disk blades on a frame supported by wheels (Fig. 8.6). It is most suitable for conditions under which the moldboard plow does not work satisfactorily. A disk plow can be operated in hard, dry soils where a moldboard plow will not penetrate, in sticky soils where a moldboard plow will not scour, in stony or stumpy fields, in soil containing heavy roots, in loose, push-type soils such as peat lands, and in abrasive soils. It is also suitable for deep plowing. According to Reed, the moldboard plow, in soils where it works properly, has a lower unit draft than a disk plow. The disk plow does not cover trash as thoroughly as a moldboard plow, and under usual plowing conditions it leaves the field rougher and more cloddy, thus requiring a greater number of subsequent operations to obtain a good seedbed. These characteristics, however, may be advantageous where erosion is a problem.

Standard disk plows generally have from one to seven concave disk blades, spaced to cut from 7 to 12 in. per disk. On some models one or more disks can be removed, and in some cases the disk spacing along the frame can be changed. The disks are tilted backward at an angle of 15 to 25° from the vertical (tilt angle, Fig. 8.1.), and are generally operated with the plane of the disk face at a horizontal angle of 42 to 45° from the direction.
of travel \(^9\) (disk angle, Fig. 8.1). The disks on standard disk plows generally have diameters between 24 and 32 in. Disk concavities also vary, the spherical radius of curvature usually being between 18 and 25 in.\(^9\) The trend is toward the larger-diameter disks because they take a wider cut, permit deeper plowing where it is desired, and cut through trash better.\(^9\)

Scrapers are furnished as regular equipment with most standard disk plows, the three most common types being the moldboard (or universal), the hoe (flat), and the rotating (flat disk). The hoe and rotating types are best in sticky soils, whereas the moldboard type assists more in covering trash.

Under most conditions, and particularly in hard, dry soils, a disk plow must be forced into the ground by its weight, rather than depending upon suction as does a moldboard plow. Consequently, standard disk plows are built with heavy frames and wheels (total weights of 400 to 1200 lb per disk blade), and even then additional weight must sometimes be added. Whereas the moldboard plow absorbs side forces mainly through the landside, a disk plow must depend upon its wheels for this purpose. Because of the large, off-center thrust forces encountered, the disks are supported through antifriction bearings (Fig. 8.2), tapered
roller bearings being the most common type. Tapered roller bearings are often used in the wheels also.

8.2. Vertical-Disk Plows. The vertical-disk plow (also known by various other names such as wheatland plow, one-way disk plow, disk tiller, and harrow plow) was first sold on a large scale in about 1927. It is similar to the standard disk plow in regard to the frame, wheels, and depth control, but the disks are uniformly spaced along a common axle or gang bolt and clamped together through spacer spools so that the entire gang rotates as a unit (as in a disk harrow). This tool is used rather extensively in the Great Plains area and in other grain-growing regions for shallow plowing and mixing the stubble with the soil. Since operating depths are shallow (often only 3 to 4 in.), energy requirements per acre are much less than for ordinary plowing.

The disks of a vertical-disk plow are somewhat smaller than those of a standard plow, the most common diameters being between 20 and 24 in. They are generally spaced 8 to 10 in. apart along the gang bolt. The width of cut per disk depends upon the spacing and upon the angle (adjustable) between the gang axis and the direction of travel. Disk angles range from 35 to 55°, with 40 to 45° being most common.

Widths of cut obtainable with various sizes of vertical-disk plows range from about 2½ to 20 ft. Some of the larger sizes have several gangs of disks in line, joined by flexible couplings. With some of these larger units, one or more of the gangs can be removed to reduce the width of cut. Since vertical-disk plows are primarily for relatively shallow plowing, they are built much lighter than standard disk plows (usually 100 to 400 lb per disk).

One of the important construction features on a vertical-disk plow is the method of absorbing the large end thrust of the disk gang. This may be done at the rear end of the gang by a single antifriction thrust bearing or a well-constructed plain thrust bearing. Or the radial supports can absorb the end thrust through combination radial-thrust bearings (such as tapered roller bearings).

8.3. Soil Reactions on Plow Disks. The net effect of all soil forces acting upon a disk blade as a result of the operations of cutting, pulverizing, elevating, and inverting the furrow slice, plus any parasitic forces acting on the disk, can be expressed in any one of several ways. In Fig. 8.3a, the resultant effect is ex-
pressed by two nonintersecting forces, one being a thrust force $T$, parallel to the disk axis, and the other being a radial force $U$. This method is particularly advantageous in calculating loads on the disk support bearings. (See reference 2 for a description of the procedure.) The thrust force is well below the disk centerline because the soil acts against the lower part of the disk face. The radial force, which includes the vertical support force on the disk blade, must pass slightly to the rear of the disk centerline to provide the torque necessary to overcome bearing friction and cause rotation of the disk.

The resultant effect can also be expressed by either of the methods illustrated in Fig. 6.1, which are based on the longitudinal, lateral, and vertical components $L$, $S$, and $V$ and resultants of these forces. This type of force representation, illustrated in Fig. 8.3b, is more useful than the other when considering the effects of soil forces upon an implement as a complete unit. In Fig. 8.3b, the $L$ and $S$ components are combined into the horizontal resultant $R_h$ so that the entire effect is represented by the two nonintersect-
ing forces $V$ and $R_a$ (as in Fig. 6.1a). Because these two forces do not intersect, they introduce a couple $Va$ that tends to rotate the implement about the axis of forward travel (the distance $a$ is identified in Fig. 8.3b). This couple is always clockwise for a disk plow as viewed from the rear, which is opposite to the effect on a moldboard plow without a coulter (see Section 7.23).

The forces indicated in Fig. 8.3a can be obtained directly from those in Fig. 8.3b (or vice versa) by proper application of the methods of statics. The same two methods of representation can also be applied to a tilted disk, as on a standard disk plow.

The influence of different variables upon soil reactions has been investigated in a series of tests made under carefully controlled soil conditions at the USDA Tillage Machinery Laboratory. Two soils were used, one a fairly heavy clay loam and the other a fine sandy loam. Most of the tests were with a 26-in. plow disk having a 22.4-in. radius of curvature. The results reported for these tests include values of $L$, $S$, $V$, and the calculated thrust $T$ but do not indicate the values of $a$ or the magnitudes of the couples involved. Clyde reports that in field tests with the Pennsylvania State College tillage meter, the magnitude of the $Va$ couple for a plow disk has usually ranged from 1100 to 1900 lb-in. (always clockwise).

The effect of speed upon these force components is indicated in Fig. 8.4. In plotting this graph from Gordon's data, forces were converted to unit values (pounds per square inch of furrow slice) because two different widths of cut were involved. The percentage rate of increase in draft ($L$) in the clay loam was about the same as the average rate for moldboard plows indicated in Chapter 7 (Section 7.33), but the rate of increase in the fine sandy loam was considerably greater. Note that the side force $S$ increased with speed whereas the vertical upward force $V$ decreased and in some runs was negative (i.e., the disk had suction). Thus with the blade tilted, increasing the speed would improve penetration under these soil conditions. Other tests indicate that if the blade is vertical the effect is reversed and penetration is decreased at the higher speeds.

The effect of disk angle is indicated for two soils and two tilt angles in graphs (a) and (b) of Fig. 8.5. Note that the draft was a minimum in each case at about a 45° disk angle. The increased draft at greater angles is probably due in part to greater throw
of the soil. At smaller disk angles the draft tends to increase because of greater contact area between the furrow wall and the convex (rear) side of the disk. This increasing contact is also indicated by the reduction in measured side force at smaller angles, particularly in the fine sandy loam. Penetration is im-

Fig. 8.4. Effect of speed on soil reactions, for a 26-in. disk (22.4-in. radius of curvature) with a tilt angle of 18 to 20°, a disk angle of 45°, and an operating depth of 6 in. Width of cut was 9 in. for the clay loam and 7 in. for the fine sandy loam. (E. D. Gordon. *Agr. Eng.*, June, 1941.)

proved by increasing the disk angle, since the upward \( V \) decreases considerably.

Increasing the angle of tilt, within the 15 to 25° range normally encountered in disk plows, increases the draft and the vertical upward force but decreases the measured side force (Fig. 8.5c). Thus, penetration is better at the smaller tilt angles.
Gordon also found that increasing the disk concavity (i.e., smaller radius of curvature) increased the vertical upward force somewhat, especially in the heavier soils, and tended to increase the draft. Comparative tests with 20-in. and 26-in. disks at a

45° disk angle slightly favored the larger disk in regard to draft and penetration when the disks were vertical but favored the smaller disk at a tilt of 19°.

Gordon concluded that soil type and soil condition have the most pronounced effect on soil reactions, as evidenced by the comparative results for the two soils in Figs. 8.4 and 8.5. It should be kept in mind that these results were obtained in carefully prepared soils that had not been subjected to the effects of plant growth, etc. Under field conditions the presence of roots, crop residues, stones, etc., would substantially affect disk-blade operation, particularly in regard to the magnitude of soil reactions.
8.4. Draft of Disk Plows. Reed⁹ lists the following ranges for the unit draft of a 26-in. disk at 3½ mph, presumably based upon tests at the USDA Tillage Machinery Laboratory:

<table>
<thead>
<tr>
<th>Soil Type</th>
<th>Draft Range</th>
</tr>
</thead>
<tbody>
<tr>
<td>Sand (5.7–5.9 per cent moisture content)</td>
<td>1.9– 3.0 psi</td>
</tr>
<tr>
<td>Sandy loam (9.3–10.7 per cent moisture content)</td>
<td>6.3– 8.5 psi</td>
</tr>
<tr>
<td>Clay (14.0–16.0 per cent moisture content)</td>
<td>9.0–11.3 psi</td>
</tr>
</tbody>
</table>

The draft values for dry, hard soil would be considerably higher than those given above.

8.5. Adjustments for Disk Plows. With either a standard or a vertical-disk plow, the width of cut for a particular size of unit may be changed by adjusting the angle between the frame and the land-wheel axle, to rotate the entire frame in a horizontal plane. This also changes the disk angle. The adjustments of the front and rear furrow wheels and the hitch are then modified as required so that the land wheel will run straight ahead. For most conditions the furrow wheels should be given a slight lead away from the furrow wall (i.e., toward the plowed ground), to compensate for the effects of side thrust.

As indicated in Fig. 8.5a, a disk angle of 40 to 45° gives the minimum draft for a given width of cut, although larger disk angles for a vertical-disk plow reduce the total draft because the width of cut is less. If the disk angle is made too small (with either type of disk plow), the convex sides of the disks may have enough contact with the furrow wall to cause difficulty in holding the cut at the desired width under some soil conditions. Note in Figs. 8.5a and 8.5b that the measured side force in the fine sandy loam was very small at a disk angle of 35°. In field tests with a 24-in. vertical disk in a silt loam, Clyde⁹ found that at a disk angle of 30° the side force decreased from 60 lb at a 3-in. depth of cut to zero at a 6.3-in. depth.

In addition to reducing the width of cut, increasing the disk angle by the above method improves penetration in hard ground, as discussed in Section 8.3. Some standard disk plows have wedges in the disk mountings so that each disk angle can be changed to improve penetration without changing the width of cut. Penetration is also improved by decreasing the tilt angle on a standard disk plow. The tilt angle should be increased for sticky soils. If penetration is not difficult, the use of a larger tilt angle will result in better turning of the furrow slice.⁹
8.6. Hitching of Trailed Disk Plows. The general considerations in regard to vertical hitching of disk plows (either type) are the same as for moldboard plows (see Section 7.25). With the usual position of the center of gravity, the suggested trial point for the vertical center of resistance is at the ground surface midway between the centers of the front and rear disks.\textsuperscript{7} If the rear furrow wheel has the proper lead and still tends to climb out of the furrow, the hitch point on the plow frame may be lowered.

Fig. 8.6. Horizontal force relations and hitching for a trailed disk plow. Although a standard disk plow is shown, the relations are basically the same for a vertical-disk plow.

The horizontal force relations (Fig. 8.6) are somewhat different for a disk plow than for a moldboard plow, because all the side thrust must be taken through the wheels and because the pulling member on a disk plow ($DF$ in Fig. 8.6) is essentially a free link in regard to horizontal forces. Whereas the horizontal line of pull on a moldboard plow must pass through the hitch point on the tractor and through a center of resistance established primarily by the plow and soil characteristics (see Fig. 7.12), the horizontal line of pull for a disk plow is determined by the location of hitch points $D$ and $F$. The position of the horizontal center of resistance $H$ and the location of the resultant side force $Q_h$ are then established by the point of intersection of $P_h$ and $R_h$ (Fig. 8.6).

For the side force to be divided equally between the front and rear furrow wheels, the line of $Q_h$ must pass midway between them.
With most trailed disk plows this condition will be approximated if the hitch is adjusted so the line of pull passes through a point slightly to the left of the average position of all disk centers (thus establishing $H$ in the desired location). If hitch point $D$ in Fig. 8.6 is moved to the left on the plow frame, $H$ and $Q_h$ will be moved toward the rear of the plow, and the rear furrow wheel will carry a greater proportion of the side thrust. Moving $D$ to the right (or $F$ to the left) puts more of the side thrust on the front wheel.

8.7. Disk Harrows. The disk harrow first attained wide popularity during the latter part of the nineteenth century. It is now second only to the moldboard plow in its importance as a tillage implement in the United States. Heavy-duty disk harrows are used for controlling weeds, cutting up and mixing stubble or heavy cover crops with the soil, and for primary tillage in orchards and vineyards as well as in open fields. Lighter units are often used in seedbed preparation subsequent to plowing.

A single-acting disk harrow has two opposed gangs of disk blades, both throwing dirt outward from the center of the tilled strip (Fig. 8.7). A tandem disk harrow has two additional gangs that throw the dirt back toward the center as a second operation, thus tilling the soil twice and leaving the field more nearly level. The offset disk harrow, which made its commercial appearance in about 1925, has one right-hand gang (i.e., a gang that moves the soil to the right) and one left-hand gang, operating in tandem. As will be pointed out in a later section, the forces acting upon an offset disk harrow are such that, when it is operating with no
side draft, the center of the tilled strip is considerably to one side of the line of pull. This feature makes the offset disk harrow especially well suited for working under low-hanging branches in an orchard. This type of disk harrow is generally designed for normal right-hand offset, as illustrated in Fig. 8.7.

Both the single-acting and the tandem disk harrows leave an untilled strip of soil between the center blades of the front gangs and leave the field in an uneven condition. A properly adjusted offset disk harrow overcomes both of these difficulties. In open-field work with a wide-tread tractor and a relatively narrow offset-type disk harrow, however, the implement must be operated with an angled or offset pull in order to trail directly behind the tractor and cover its tracks.

Trailer, tandem disk harrows are by far the most common type, with widths usually ranging from about 5 to 12 ft. Single-acting disk harrows are available in widths up to about 20 ft. Offset disk harrows with rigid gangs generally range in size from 4½ to 12 ft, but units with hinged gangs may be somewhat wider. For extremely wide units that have the flexibility needed for uneven fields, two or three offset disk harrows with rigid gangs are coupled together by means of flexible hitchs to give "squadron" units up to 36 ft wide. The rear gangs of squadron arrangements are usually in line (but have hinged joints). Most trailed disk harrows have power angling devices, operated either hydraulically or by moving the tractor forward or backward while a mechanical lock is held open.

Some of the small and medium-size trailed disk harrows have wheels between the front and rear gangs that are used for leveling and depth control, for raising the harrow at the end of the field, and for transport on roads or highways. Hydraulic control is generally employed.

Since the advent of tractor-mounted tools in the late 1930's, mounted disk harrows have been developed and are popular for certain applications. Their weight is necessarily limited because they must be lifted by the tractor. Their principal advantages are in maneuverability and ease of transport.

8.3. Bearings. As with other disk tools, bearings are subjected to both radial and thrust loads of considerable magnitude. The most common bearing arrangements on disk-harrow gangs are
(a) bearings mounted on the spools (spacers) between disk blades, 
(b) the enclosed tubular or oil-bath type, and (c) end-mounted 
bearings. In end-mounted bearings, sealed antifriction bearings 
are placed in self-aligning housings (Fig. 8.8). Spool-mounted 
bearings may be of the self-aligning, antifriction type (Fig. 8.9b), 
or they may be plain bearings made of chilled cast iron or other 
wear-resistant materials. In the past, plain bearings made of oil­
impregnated wood were common. End thrust in plain bearings is 
taken through matching collars and grooves, as indicated in Fig. 
8.9a. Since these bearings run in contact with dirt much of the 
time, lubrication is a problem and wear of plain bearings is 
usually rapid. Antifriction bearings are more expensive than 
plain bearings, but for either the spool-mounted or the end­
mounted arrangements they have the advantage of better perform­
ance of seals and the consequent reduction of lubrication prob­
lems.

The tubular type with bear­
ings running in an oil bath (Fig. 
8.9c) may have either plain or 
antifriction bearings. However, since plain bearings in an oil 
bath wear well and since the friction load of the bearings under 
these conditions is a very small part of the total draft, there ap­
ppears to be little justification for the added cost of antifriction 
bearings in the tubular arrangement (see Problem 8.3).

8.9. Disk-Harrow Blades. Most disk harrows have blades 
from 16 to 24 in. in diameter, spaced 6 to 10 in. apart. The 
larger sizes and wider spacings are desirable for cutting up 
heavy cover crops; they are found primarily on offset disk har­
rows and the heavy-duty tandem units. Single-acting disk har­
rows and light-duty tandem units generally have the smaller 
sizes and closer spacings, thus obtaining more thorough tilling be­
tween disk blades. Scrapers are available for operation in sticky or 
wet soils. Disk-harrow blades generally have less concavity 
than plow disks, although there is a tendency to use plow disks
Increasing the concavity turns the soil more but tends to reduce the penetration.

Some disk harrows are equipped with cut-out or notched blades and are used for such operations as pasture renovation and cutting heavy trash. Cut-out blades penetrate a little better than plain disks and cut heavy trash more readily because they tend to pull it under instead of pushing it ahead. But they are also more expensive and may have a greater tendency to break, as discussed in Section 8.19.

8.10. Soil Reactions on Disk-Harrow Blades. The studies pertaining to soil reactions on plow disks, discussed in Section 8.3, were mostly with disk angles of about 45°, whereas disk harrows are seldom operated with disk angles greater than about 25°.
However, most of the trends discussed can probably be applied to disk harrows.

With the Pennsylvania State College tillage meter, Clyde made a series of tests in a silt loam soil to determine soil reactions on groups of four or five 18-in. and 22-in. disk-harrow blades. In these tests the value of $V$ was varied by applying different weights to the disk assemblies and then allowing the disks to seek their own depth during the run, thus simulating the usual operation of a trailed disk harrow. The results for the 22-in. disks, covering the range of $V$ usually encountered in heavy-duty disk harrows without extra weight added, are shown in Fig. 8.10. Average values of $a$ are indicated, which, when multiplied by $V$, give the magnitudes of the soil couple $Va$ about the axis of forward motion. Note that at disk angles of 19° and 15°, measured values of $S$ varied widely for a given $V$. This difference was probably due to varying amounts of side support furnished by the furrow walls at different depths and under different soil conditions. In the 23°
runs this side support was not so evident. The ratio between 
$L$ and $V$ was reasonably consistent for each of the three disk 
angles.

The tests with 18-in. disks (1½-in. concavity, 6⅔-in. spacing) 
were at considerably lower values of $V$ (20 to 70 lb per blade), 
because light-duty disk harrows with this size of blades usually 
weigh only 30 to 50 lb per disk. The wide spread of $S$ was not 
evident in these tests. The ranges of $L/V$ and $S/V$ indicated 
by Clyde’s curves for 18-in. blades are as tabulated below. $R_h$

<table>
<thead>
<tr>
<th>Disk Angle</th>
<th>$L/V$ Ratio</th>
<th>$S/V$ Ratio</th>
</tr>
</thead>
<tbody>
<tr>
<td>15°</td>
<td>0.5-0.75</td>
<td>0.6-0.9</td>
</tr>
<tr>
<td>19°</td>
<td>0.7-1.0</td>
<td>1.0-1.3</td>
</tr>
<tr>
<td>23°</td>
<td>0.9-1.2</td>
<td>1.25-1.55</td>
</tr>
</tbody>
</table>

usually met the plane of the disk face 1 to 2 in. back of the vertical 
centerline, and $R_v$ met it about 2 in. above the bottom.4

8.11. Forces Acting upon a Disk Harrow. McKibben is 
credited with making the first complete force analysis on a disk 
harrow. His studies,5 which included qualitative field observa-
tions, were initiated soon after the commercial appearance of the 
offset disk harrow, primarily to determine the factors responsible 
for its ability to operate in an offset position with no side draft.

Soil reactions on an individual blade have already been dis-
cussed and are illustrated diagrammatically in Fig. 8.3. The
combined soil reactions for a group or gang of disks can be con-
sidered as acting upon a single disk blade in the average position 
of all blades (i.e., at the center of the gang). Then the forces act-
ing upon a complete disk harrow are (a) the resultant soil reac-
tion on each gang, (b) the weight of the implement plus any extra 
weight added, (c) any supporting soil forces provided by wheels 
or as a result of being mounted on a tractor, and (d) the pull of 
the power source. For uniform motion these forces must be in 
equilibrium. If there is to be no side draft, the sum of the side 
components of all soil reactions must equal zero.

8.12. Horizontal Forces. Consider first a single-acting disk 
harrow, as indicated in Fig. 8.11. The position of the horizontal 
center of resistance $H$ is determined by the intersection of $R_{ha}$ 
and $R_{hb}$ (the resultant soil reactions on the two gangs). Since 
this implement is symmetrical, with the two gangs normally
operating at the same disk angle and under similar soil conditions, the side forces $S_a$ and $S_b$ are equal and opposite and the pull $P_h$ is in the direction of travel (i.e., no side draft) and at the center of cut.

A similar situation exists in the two rear gangs of a tandem disk harrow except that the $S$ components are directed away from, rather than toward, each other. Thus in a tandem unit the horizontal component of pull $P_h$ would still be at the center of the tilled strip and would be equal to the sum of the $L$ components of the four gangs.

Figure 8.12b shows the horizontal forces acting upon an offset disk harrow when it is operating with no side draft. Again, the location of the horizontal center of resistance $H$ is determined by the intersection of $R_{h_1}$ and $R_{h_2}$. For the condition of no side draft, the hitch linkage of the disk harrow must be adjusted so that the hitch point $F_0$ is directly in front of $H$.

If the hitch linkage is changed to move the implement either to the right or to the left from the no-side-draft position, side draft is introduced and the operating conditions of the harrow are changed. If, for example, the hitch point in Fig. 8.12b is moved from $F_0$ to $F_2$, the force equilibrium is momentarily destroyed and the side component of the new pull, acting at point $H$, rotates the implement counterclockwise about $F_2$. Rotation continues until the disk angles of the two gangs have readjusted themselves (the front increasing and the rear decreasing, since the total included angle remains constant) so that the difference between their lateral force components $S_f$ and $S_r$ becomes equal to the side

![Diagram](image-url)
Fig. 8.12. Horizontal force relations for a trailed, right-hand offset disk harrow without wheels. The disk angles and pulls shown in these examples were measured in a silt loam soil. Each gang had ten 22-in. disks with a 2 1/2-in. concavity and a 9-in. spacing. The probable distribution of $L$ and $S$ for these conditions was determined on the basis of tests discussed in Section 8.10. (A. W. Clyde. Agr. Eng., June, 1939.)
draft $P_y$ (as in Fig. 8.12a). Note that the magnitudes of $L_f$ and $L_r$ and the position of $H$ also change during this readjustment. The change in disk angles shown between positions (b) and (c) seems small, but Clyde states that these results were consistent with those from other tests with this tool.

In changing from condition (b) to condition (a), the amount of soil moved by the rear gang is decreased. Conversely, in the extreme right-offset position (Fig. 8.12c), the rear gang is doing most of the work. Even with no side draft (Fig. 8.12b), the rear gang operates at a greater angle and moves more dirt than the front gang, because it is in softer soil. In orchards this condition results in a gradual movement of soil away from the trees.

8.13. Amount of Offset Obtainable. Let $e$ be the amount of offset (lateral distance from hitch point $F_1$ to center of cut), $d$ the longitudinal distance between the centers of the two gangs, and $b$ the longitudinal distance from the center of the front gang to the hitch point (Fig. 8.12c). Taking moments about $F_1$ yields the following relation (assuming that $R_{hf}$ and $R_{hr}$ pass through the centers of the gangs):

$$eL_f + eL_r + bS_f - (b + d)S_r = 0$$

From which

$$e = \frac{b(S_r - S_f) + dS_r}{L_f + L_r}$$  \hspace{1cm} (8.1)

For the condition of no side draft, $S_r = S_f = S$. Hence, from equation 8.1, the offset obtainable with no side draft is

$$e_0 = \frac{dS}{L_f + L_r}$$  \hspace{1cm} (8.2)

Equation 8.2 states that the amount of offset obtainable without side draft is a function only of the distance between gangs and of the relative magnitudes of the lateral and longitudinal soil reactions. The force relations, however, are affected by a number of variables, such as soil condition, disk angles, speed of travel, disk-blade size and concavity, etc. According to Clyde, a soft field and the consequent deep penetration give a smaller $e_0$ than a very firm field, because in the soft field $L$ tends to be large in relation
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to S. The distance between gangs is limited by practical considerations such as frame construction and required turning room.

8.14. Vertical Forces. Figure 8.13 shows the general vertical force relations for a double-acting disk harrow (tandem or offset) having a rigid frame (no horizontal hinge axis between the front and rear gangs). The numerical values given are for the example of the offset disk harrow with no side draft, as shown in Fig. 8.12b. The position of the vertical center of resistance G is established by the intersection of W (the weight of the implement) and the line of pull. \( R_{uv} \) and \( R_{vr} \) automatically adjust themselves so that their resultant \( R_v \) passes through point G and is in equilibrium with \( W \) and \( P_v \). Note that even with the special low implement hitch point at \( E \), \( V_f \) was considerably larger than \( V_r \) (although \( L_r \) was nearly as great as \( L_f \) because the rear gang operates in soft ground). Raising the hitch on the implement to one of the regular holes (above \( E \)) would increase the “weight transfer” from rear to front, thus increasing the penetration and soil-moving action of the front gang at the expense of the rear gang.

With many of the tandem disk harrows, the rear gangs are pulled from a joint on the front gangs that acts as a horizontal hinge axis. Vertical force relations for this type of arrangement are shown in Fig. 8.14. Since the joint \( E_r \) is actually the vertical hitch point for the rear gang, \( P_{vr} \) must pass through it as well as

Fig. 8.13. Vertical force relations for a double-acting disk harrow with no hinge axis between the front and rear gangs. Numerical values given are for the conditions indicated in Fig. 8.12b. (A. W. Clyde. Agr. Eng., June, 1939.)
being in equilibrium with $W_r$ (the weight of the rear gangs) and $R_{wr}$. At the front gang, then, $W_f$ is combined with $-P_{vr}$ to give a resultant force that must be in equilibrium with $P_v$ and $R_{vf}$. Inspection of the relative magnitudes of the force vectors in Fig. 8.14 (which represent typical values, based on Clyde's tests) indicates that if more weight is added to increase penetration it is most effective when placed toward the rear of the implement.

Fig. 8.14. Vertical force relations for a tandem disk harrow with hinged joint. (A. W. Clyde. Agr. Eng., June, 1939.)

8.15. Couples Acting on Disk-Harrow Gangs. It is a well-known fact that the concave end of a disk-harrow gang tends to penetrate more deeply than the convex end. This condition exists because the soil force component $T$, perpendicular to the disk blades, is applied well below the axle (Fig. 8.15) while the balancing force $T'$ is applied at axle height (through the bearings or through bumpers on the inside ends of the gang bolts in the case of two opposed front gangs), thus forming a couple $T_f$. The forces $V$, $M$, and $U$ are in the plane of the disk face.

For uniform penetration, $V$ (Fig. 8.15) will act approximately at the center of the gang. To obtain uniform penetration with a single gang, the resultant downward force $W'$ (weight of gang, plus any extra weight added, minus any upward component of pull) must act at a distance $h$ from the center of the gang (toward the convex end) such that

$$W'h = fT$$  \hspace{1cm} (8.3)

If the weight of the tool and the upward component of pull act at
the center of the gang, they do not contribute to the resisting couple. Then $W'$ and $h$ in equation 8.3 can refer directly to any weight added.

In single-acting and tandem disk harrows it is a relatively simple matter to obtain uniform penetration by having the couples of the laterally opposed gangs counteract each other through the frame. The design problem is more complex in the case of an offset disk harrow because the opposing couples subject the frame between the gangs to torsion. Careful design of the frame and adequate provision for rotational adjustment of one gang with respect to the other are important. Another complicating factor in an offset disk harrow is the fact that the operating conditions for the two gangs are different, so that the couple of the front gang is generally larger than that of the rear gang. For the example shown in Fig. 8.13, the front couple was 6400 lb-in. whereas the rear couple was only 4090 lb-in. Because of the larger front couple, there is a tendency for the right side of the entire harrow, and particularly the right end of the front gang, to run deeper than the rest of the harrow (considering a right-hand offset disk harrow).

If weight must be added to a disk harrow to obtain adequate penetration, proper placement will tend to counteract the rotational effects of the gangs, thus promoting uniform penetration and in some cases relieving the frame of some of its load. But if the hinge axis of the hitch on a right-hand offset disk harrow (point $E$ in Fig. 8.13) is fairly high so that the front gang tends to run deeper than the rear one, weight should first be added to the left (concave) end of the rear gang, even though it adds to the natural turning moment of this gang. This placement is best
DISK TOOLS

for the disk harrow as a whole, under these conditions, because of the tendency for the larger couple of the front gang to make the entire right side of the harrow run deeper.

8.16. Penetration of Disk Harrows. Among the factors that tend to increase depth of penetration under given soil conditions are the following:

1. Increasing the disk angle.
2. Adding weight.
3. Lowering the hitch point on the tractor.
4. Reducing the forward speed.
5. Use of sharp, relatively thin disk blades.
6. Use of blades with less concavity.
7. Use of smaller-diameter blades.

Although it is generally conceded that smaller blades improve the penetration of a disk harrow, it is interesting to note that Gordon's tests with 20-in. and 26-in. plow disks at a 45° disk angle tended to favor the larger disk in this respect.

With wheel-type or mounted disk harrows, depth control is no particular problem. With trailed disk harrows that do not have wheels, depth is controlled primarily by changing the angle between the gangs (disk angles) or by the addition or removal of weight when the gangs are fully angled. The depth data included in Fig. 8.10 illustrate these relations. In soft or sandy soils it is sometimes desirable to add depth-limiting devices to permit operation at the full disk angle. Depth gages similar to split pulleys, clamped around the spacer spools between blades, are most common. Two gages are generally attached to each gang.

8.17. Wear Resistance of Disk Blades. The relation of wear resistance to steel hardness, carbon content, and method of manufacture has been determined in a series of tests performed at the USDA Tillage Machinery Laboratory. Field tests were conducted by installing groups of different kinds and hardnesses of disks on tandem disk harrows that were owned and operated by farmers. Disks of one particular type and hardness were included on all the disk harrows, to provide a common basis for comparing the relative wear rates of the various groups on any one implement. The weight losses were taken as the index for relative wear.
In addition, an apparatus was developed that permitted rapid
determination of relative wear rates in the laboratory and yielded
results that correlated closely with those of field tests in regard
to relative wear ratings. In this wear-test machine (Fig. 8.16) a test sample about 2 in. square is held against an abrading
disk with a constant force while carefully screened sand is fed
down between the sample and the disk. The \( \frac{3}{8} \times 4 \) in. disk is
made from some type of relatively uniform soft iron plate and is
driven at 45 rpm.

Standardized conditions with a 12-lb contact force and a total
peripheral travel of 300 ft for the disk were established. Under
these conditions, the weight losses for many of the steels used in
tillage tools were found to be in the order of 0.10 gm (measured
to nearest 0.0001 gm), and replica tests on a metal sample could
be made in a total time of about 30 min. This type of machine,
however, is not suitable for measuring the relative wear of hard
surface layers less than about 0.025 in. thick.

The USDA tests indicated that with carbon steels having be-
 tween 0.80 and 0.96 per cent carbon, the greatest resistance to
wear was obtained when the hardness was in the range from 44 to 48 on the Rockwell C scale (Rc). The weight loss was definitely greater when the hardness was reduced to Rc40 or Rc36 or increased to Rc52. At any given hardness over this range, the wear resistance increased as the carbon content was increased. In comparative laboratory wear tests with SAE 1080 and SAE 6160 steels at various hardnesses, the weight losses were a little less for the 1080 steel (0.80 per cent carbon) than for the 6160 steel (an alloy steel containing 0.60 per cent carbon, 0.70 to 0.90 per cent manganese, and 0.80 to 1.10 per cent chromium). Thus, although the addition of the chromium and manganese tends to increase the toughness of the steel (discussed in Section 8.19), it does not compensate for the reduction in carbon content, in regard to wear resistance.

8.18. Disk Sharpening. In manufacturing disks, the edges are either sharpened by grinding or rolled out to a thickness of about \( \frac{3}{4} \) in. and not ground. Similar methods are employed in some areas to resharpen worn disks. In the USDA field tests with new disks, the ground edges, until they became rounded, lost weight much more rapidly than the rolled edges. After about 250 hr of use the rates of loss were about the same for the two types. When worn disks are resharpened, the rolling method has the additional advantage of increasing the diameter of the disk somewhat during the rolling operation and thus saving the metal that would be removed if sharpened by grinding.

8.19. Impact and Fatigue Resistance of Disks. In conjunction with the wear tests described above, the USDA also conducted laboratory tests to determine the relative resistance of disk blades to impact shocks and fatigue. Relative impact resistance was determined by dropping a guided weight (usually 100 or 200 lb) onto the edge of the disk from preselected heights. In a given test the weight was dropped from successively greater heights until the disk failed or the limit of the test equipment was reached. Tests were made with the disk axis horizontal and with it vertical. Accelerated fatigue tests were made with a setup of the type illustrated in Fig. 8.17, using disk edge loads of 1800 to 2175 lb. The disk was inspected after each 100 revolutions, the test being continued until the disk failed or was considered no longer serviceable.

In the laboratory tests, both impact and fatigue resistance decreased rather rapidly as the hardness of the disks (SAE 1080
steel) was increased from $R_c36$ to $R_c52$. Observations during the field tests for wear resistance indicated that disks hardened to $R_c44$ or slightly above should withstand severe service with relatively low breakage. Considering the resistance to wear, impact, and fatigue, Reed and McCreery recommend making disk-harrow blades from SAE 1085 or 1090 steel, hardened to about $R_c42$ or $R_c44$.

Disks made from cross-rolled steel were definitely superior to disks from straight-rolled steel, in regard to impact and fatigue resistance. The difference tended to become less as the hardness was increased to $R_c52$. Disks made from SAE 6160 steel were somewhat superior to disks of 1080 steel in regard to impact and fatigue, when both types were hardened to $R_c39$. At $R_c42$ and $R_c45$ there was no difference, and at $R_c36$ the alloy steel was better only in regard to fatigue. In general, field experience indicates that alloy steel will stand up better than carbon steel in large disks subjected to severe operating conditions.

The notches in cut-out disks are currently made by one of two methods. In the first method the notches are all punched out simultaneously by a die moving parallel to the disk axis and are then sharpened by grinding. In the second method the notches are cut individually with a die moving at an angle from the disk axis. The resulting notches are left sharp without the necessity for grinding. The USDA results indicate that disks with angle-cut notches are superior to those with ground notches in regard to impact and fatigue resistance and will stand up better in field service. In laboratory tests on 22-in. carbon-steel disks, those
with angle-cut notches tended to stand up about as well as the unnotched disks.11

REFERENCES


PROBLEMS

8.1. A 7½-ft offset disk harrow with ten 22-in. disks per gang is operated under the soil conditions represented by Fig. 8.10. For the front gang the disk angle is 15° and \( V = 1000 \) lb. For the rear gang the disk angle is 23° and \( V = 600 \) lb. Determine the draft in pounds per foot of width. For the front gang use the average value of \( L \) indicated in Fig. 8.10. Since the rear gang is in soft ground, use the lower limit of the band in determining \( L \) for it.

8.2. The resultant effect of all soil reactions upon each 22-in. diameter blade of a disk-harrow gang, as measured by Clyde, under heavy loading, can be represented approximately by the values indicated for the radial force \( U \) and the axial thrust force \( T \) in the diagram on page 196. \( U \) actually passes slightly to the rear of the disk center to provide torque for
rotating the gang but is shown through the center to simplify the problem. Consider a gang that has seven disks and is supported by two combination radial-thrust bearings 45 in. apart and equidistant from the plane of the resultant radial force for the gang. The total weight of the rotating assembly (disks, spools, gang bolt, etc.) is 250 lb.

(a) Determine the radial load on each bearing. Use graphical methods where possible. Suggestion: see reference 2.

(b) Determine the total thrust load. Discuss its probable distribution between the two bearings, and factors that might affect the distribution.

8.3. Under the conditions of Problem 8.2, the draft per disk is 175 lb. Assume that the disks make one revolution per 5 ft of forward travel. Determine what percentage of the total draft is required to overcome bearing friction for each of the following arrangements:

(a) Chilled-iron bearings (Fig. 8.9a), with the diameter of the cylindrical bearing surface = 2 1/2 in., the average diameter of the thrust face = 3 in., and the coefficient of friction = 0.15.

(b) Enclosed, oil-bath-type plain bearings (Fig. 8.9e), with the diameter of the cylindrical bearing surface = 1 3/8 in. and the average diameter of the thrust face = 1 1/4 in. Assume the coefficient of friction = 0.02 for the oil-bath bearings, which is a conservatively high value for thick-film lubrication under these conditions.

8.4. What percentage of the total cross-sectional area between the ground surface and the plane of the bottoms of the disk blades is cut out by:

(a) A vertical-disk plow with 24-in. disks spaced 9 in. apart, when operating with a disk angle of 43° and at a depth of 4 in. (to bottom of disk blade)?

(b) A disk harrow with 22-in. blades spaced 9 in. apart, when operating with a disk angle of 20° and at a depth of 4 in.?

8.5. A disk-harrow gang (22-in. blades) is operating at a disk angle of 19°, with a V of 80 lb per disk.

(a) Taking the average values indicated in Fig. 8.10, construct a force diagram similar to Fig. 8.3b.

(b) By application of static mechanics, determine the location and magnitude of the corresponding radial and thrust forces as in Fig. 8.3a.
CHAPTER 9

Miscellaneous Tillage Equipment

9.1. Rotary Tillers. Although there has been a marked increase of interest in rotary tillers in the United States during the last few years, the principle of rotary tillage is not new. Patents on devices of this type were issued as early as 1850-1860, but it was not until about 1910 that a successful rotary tiller, using the elastic-claw principle with a staggered arrangement, was developed by Von Meyenberg of Switzerland. Swiss-made rotary plows or tillers were introduced into the United States in about 1930, and soon afterwards several American manufacturers started making this type of equipment.

Rotary tillers are available as (a) garden-type, self-propelled units (two-wheel, with 8 to 30 in. widths of cut), (b) trailed or tractor-mounted units with pto drives (usually 3 to 4 ft cut), (c) trailed units with auxiliary engines (4 to 7 ft cut, with engines as large as 90 to 110 hp), and (d) field-type, self-propelled machines (6 to 8 ft cut, with engines as large as 180 hp).

The garden-type rotary tillers have proved themselves practical and effective for preparing seedbeds from unplowed ground for nurseries, greenhouses, vegetable crops, lawn planting, and similar jobs. These narrow machines are also suitable for shallow cultivation and weed control between rows of certain crops. In addition, special, tractor-mounted units (see Chapter 12) have been developed for rotary cultivation of row crops on a field scale. The practice of preparing farm-scale seedbeds with rotary tillers is still rather limited.

9.2. Mechanical Arrangements. In the usual arrangement, the rotor consists of a power-driven, transverse shaft on which knives or tines are mounted to cut the trash and soil. Rotor speeds are generally in the order of 200 to 300 rpm. Many types
and shapes of tines or knives are available, none being ideal under all conditions. The hook or pointed type of tine (as shown in Fig. 9.1) is most suitable for deep tillage in relatively clean ground, but clogging and wrapping of trash on the rotor are likely to occur when heavy cover crops are encountered. The various types of knife tines, such as the one illustrated in Fig. 9.2 and the heavier L-shaped knives used on some machines (see Fig. 12.6a) are better for trashy conditions where deep penetration is not required and for weed control.

![Diagram](a) Typical cross-section of garden-type rotary tiller in operation. The divider shown is to split out the strip beneath the drive housing, which would otherwise not be tilled. (b) Path of tip of tine in relation to forward travel. (C. W. Kelsey. Agr. Eng., April, 1946.)

One of the serious problems with rotary tillers is that of tine breakage or bending in hard or stony ground. To combat this problem, various systems are employed to reduce the impact shock and protect against overloads. These include resilient tines, spring-mounting of tines, rubber-mounting of tines, resilient couplings in the drives, and friction slip clutches in the drives. The machines with the heavier, L-shaped knives or hoes generally have rigid knife mountings and rely upon the slip clutch for protection.

**9.3. Principle of Operation.** The tines or knives are spaced along the axis of the rotor and are staggered around the periphery to take incremental bites of soil and distribute the load on the machine more uniformly. As the rotor revolves, each tine or knife cuts a slice from the untilled soil as illustrated in Fig. 9.1, the cross-sectional area of the slice depending upon the depth of tillage.
and the amount of forward travel $F$ per cut. Note that the thickness of the slice varies with the distance above or below the rotor axis. Frevert shows by a mathematical analysis that the ratio of the distance of tine travel in the soil to the depth of tilling is a minimum when the diameter of the tine circle is about 1.25 times the depth. Thus, the power efficiency of the machine could theoretically be improved by reducing the rotor diameter to approach this optimum ratio.  

### 9.4. Performance

One of the criticisms of the rotary tiller is that it stirs the soil more than most other machines and can

![Fig. 9.2. Soil particle-size distribution resulting from various tillage treatments applied to clover stubble on a silt loam soil having a moisture content of 28 per cent. (Richard K. Frevert. Unpublished thesis. Iowa State College, 1940.)](image-url)
iliary engine, the operator has some control over the type of seedbed by proper selection of forward speed. It has been a general observation that the rotary tiller produces a satisfactory seedbed under most conditions \(^8,13,16\) although in some cases the soil is so loose that it is desirable or necessary to pull a roller behind the tiller to firm the seedbed.

The rotary tiller is an excellent mixing device for distributing organic matter or other materials throughout the tilled soil,\(^8,10,11,16\) but coverage is not as complete as with a moldboard plow. The surface trash is usually well cut up, with varying amounts left on the surface. However, materials such as wet, tough corn stalks and green weeds may pass through in rather large pieces and be left on the surface to interfere with planting and cultivating.\(^9\) Cutting knives must overlap sufficiently to cut off all the vegetation; otherwise the plants will continue to grow. The fast rotating speed of the cutting knives will destroy many injurious soil insects, but it may also destroy beneficial ones.\(^10,16\)

9.5. Power Requirements and Operating Conditions. Frevert\(^8\) found that the energy requirements per unit volume of tilled soil were more than four times as great with a forward travel of 1.58 in. per slice as with \(F = 8.75\) in. (Fig. 9.3). This is to be expected, since the first condition resulted in greater pulverization of the soil (Fig. 9.2). Even the lowest unit energy requirement with the rotary tiller (at \(F = 8.75\) in.) was about three times as great as for a moldboard plow in the same soil. However, the additional energy and cost for disk ing and harrowing after using the moldboard plow must be included to give a fair comparison, and the rotary tiller should be credited with the increase in fuel efficiency obtained when a tractor transmits power through the pto or belt rather than through the drawbar. Frevert found that increasing the rotor speed from 100 to 400 rpm, with a corresponding increase in forward speed (i.e., a constant \(F\)), increased the unit energy requirements by an average of about 23 per cent. He found that pointed tines (Fig. 9.1) required less energy than the “S” tines (Fig. 9.2).

Rotary tillers can prepare a seedbed to a depth as great as 6 to 9 in. if sufficient power is available. The small, garden-type units are ordinarily operated at speeds between \(\frac{1}{2}\) and 2 mph, whereas the usual speed for the larger units (with relatively more
power available) is from $1\frac{1}{2}$ to $2\frac{1}{2}$ mph. Shawl lists the engine power requirements for the larger machines at these speeds as being from 10 to 15 hp per ft of width. If less power is available, slower forward speeds must be used. This may necessitate the installation of special low-speed drive gears in the smaller tractors if they are to handle pto-driven units satisfactorily.

![Graph](image)

**Fig. 9.3.** Effect of forward travel per slice upon rotor energy requirements for the same rotary tiller and the same soil as in Fig. 9.2, but at a 21 per cent soil moisture content. The indicated units for energy input correspond numerically to the psi unit draft of a conventional plow. (Richard K. Prevert. Unpublished thesis. Iowa State College, 1940.)

### 9.6. Subsurface or Mulch Tillage

Increasing emphasis on subsurface tillage, particularly in arid and semiarid regions, has resulted in the development of special implements for this purpose as well as the modification of conventional types of equipment. The chief objectives of subsurface tillage are to reduce erosion and to conserve water by reducing run-off. This practice involves cutting the roots of weeds and other plants and leaving the crop residue on the surface or mixed into the top few inches of soil, the proper disposition of the residue depending
upon the amount present and the subsequent operations and equipment involved. The large amounts of trash at or near the surface introduce problems in planting (since the planter must penetrate the mulch) and in cultivation of row crops.

In addition to killing weeds and leaving the residue at the surface, a subsurface tiller is required to perform the initial tillage at normal plowing depths and to prepare a satisfactory seedbed with a minimum of surface pulverization. The most common types of subsurface tillers are implements with large V-shaped sweeps, implements with straight blades at right angles to the direction of travel, and modified rod weeder. In extremely heavy mulches where some of the residue must be cut up and mixed into the top few inches of soil to facilitate subsequent operations, disk tools, moldboard plows with part or all of the moldboard surfaces removed, and "double-cut" moldboard plows may be employed. The double-cut moldboard plow has an additional set of smaller shares, without moldboards, mounted behind and below the standard bottoms. The standard bottoms invert only the top 3 or 4 in. of soil and the auxiliary bottoms merely loosen the next 3 or 4 in.

9.7. Blade-Type Subsurface Tillers. V-shaped sweeps designed for subsurface tillage (Fig. 9.4) have cutting widths ranging from 15 in. to 8 ft or more per sweep. For ordinary amounts of residue a sweep width of at least 24 in. is recommended, to provide adequate trash clearance between standards. There should be at least 18 in. vertical clearance between the sweep and the beam, as well as adequate fore-and-aft distance between the staggered rows of sweeps. The V-angles range from 60 to 90° for the commonly used 24-in. to 30-in. sweeps and from 85 to 100° for sweeps with widths of 5 ft or more. The smaller angles shed trash better but require excessively long wings for sweeps with wide cuts.

The blades are generally made from 4-in. curved or flat steel drawn down to a thin cutting edge. These blades must perform the same operation as a plowshare and need to be kept
sharp. The heel or rear side of each blade is usually about $1\frac{1}{8}$ to $1\frac{3}{8}$ in. above the cutting edge (for a 4-in. blade), to provide suction and to give a pulverizing action as the soil is raised and allowed to fall. Blades that are convex upward have more suction and give greater pulverization than flat blades. High-lift sweep blades give better pulverization than flatter sweeps but tend to furrow and ridge the soil when it is loose and may cause clogging under some conditions. The shanks that support the V-blades must be strong and laterally rigid and yet be designed to leave only a narrow kerf or trench and to gather a minimum amount of trash and roots. A rolling coulter should be mounted ahead of each standard whenever there is any appreciable quantity of residue. The paths of the sweeps should overlap by 3 or 4 in. to obtain a good weed kill.

Subsurface tillers with V-sweeps give good performance under favorable conditions and are suitable for relatively shallow depths. However, they generally produce insufficient pulverization and must be supplemented by secondary tillage operations. Shallow-rooted plants are often retarded but not killed in the initial tillage. Soil moisture contents should be about the same as for good plowing, or a little drier.

Straight blades operated at right angles to the direction of travel are usually 6 to 7 in. wide (flat or curved) and 5 to 12 ft long. They are mounted on heavy frames and built to stand rough treatment. Straight blades are generally suitable for initial tillage but do not shed trash as well as V-sweeps and tend to clog when operated at shallower depths during subsequent operations.

9.8. Chisels and Subsoilers. These tools are used primarily to break through and shatter compacted or otherwise impermeable soil layers and other barriers to the movement of water and roots. The most effective results are obtained when the soil is dry, thus contributing to the shattering action. With some soils the beneficial effect tends to be lost rather rapidly as subsequent rains or irrigation cause the soil to run together again. Subsoilers have one or more heavy standards that can be operated at depths of 24 to 30 in. or more. Chisels or chisel plows have a series of heavy, rigidly mounted standards, spaced 12 to 18 in. apart and equipped with replaceable teeth or shovels, similar
to the arrangement of a field cultivator. They can be operated at depths well below the normal plowing zone if the impervious layer is relatively thin and if adequate power is available.

Under adverse soil conditions or when it is desirable that the soil not be inverted, chisels are sometimes used for deep tillage in place of plowing. Since chisels do not pulverize the soil as well as plows, a greater number of subsequent operations may be needed to obtain a good seedbed after chiseling.

9.9. Field Cultivators. Many different combinations of standards and shovels are found on field cultivators. The standards, which are arranged in two or three ranks, are much closer together (usually 6 to 12 in.) than on a subsurface tiller with sweeps. Rigid standards (with or without spring trips), light or heavy spring teeth, and heavy coil-spring standards are among the common types. Tool bars may be on trailed tool carriers, or they may be tractor mounted.

The field cultivator, with either tooth equipment or sweeps, is used primarily for weed control, seedbed preparation, and other secondary tillage operations. When equipped with sweeps, the field cultivator, or duckfoot cultivator as it is often called, sometimes replaces the rod weeder for shallow subsurface tillage operations in stony soils. However, because of the relatively close spacing of standards, the duckfoot cultivator is not satisfactory where there are large amounts of trash. Clogging may be reduced by increasing the vertical clearance and lateral spacing of standards and by employing spring teeth to obtain maximum tooth chatter.

9.10. Rod Weeders. This implement has been used for many years in the Pacific Northwest and to some extent in other areas, primarily to control weeds during fallow periods after plowing. A rotating round or square rod about 1 in. in diameter is driven by a ground wheel and turns slowly in the opposite direction at a depth of 1 to 2 in. below the soil surface. This action lifts or severs the plant roots and leaves them near the surface. It also has the desirable effect of firming the soil below the rod level. Rod weeders are usually about 10 to 12 ft wide, with four or five standards to support the rod. They are best suited to loose soil that is free from stones and other obstructions.

To convert a rod weeder for subsurface tilling, a bar carrying narrow teeth or small shovels is attached below the revolving
rod. This modified tool is particularly well adapted to weed control and soil pulverization after primary subsurface tillage has been performed with some heavier tool. The rod tends to clear trash, although clogging may occur if the initial tillage has placed considerable residue at a depth of 3 to 4 in.\textsuperscript{15}

9.11. Spike-Tooth Harrows. The spike-tooth harrow is a universally used implement and is as old historically as the plow. Its principal function is to finish the seedbed by smoothing it and breaking surface clods, particularly in a mellow, friable soil. A spike-tooth harrow is often pulled directly behind the plow. It is not very effective in breaking clods after they have become hard. This tool is effective in killing small weeds that are just starting and is also useful in covering broadcasted seeds and in breaking crusts that have formed over newly planted crops.

The spike-tooth harrow is made in sections that are ordinarily 4 to 6 ft wide. The teeth are either clamped or welded to cross-bars (usually five bars) and staggered so that each tooth works a strip 1\% to 2 in. wide. Straight, pointed teeth are used, the most common cross-section being a \( \frac{3}{8} \times \frac{3}{8} \) in. diamond shape. Tooth bars can usually be rotated to change the tooth angle from vertical to nearly horizontal, to control the depth of penetration. Weight is sometimes added also. The teeth shed trash more readily when inclined considerably from the vertical. The frames of most harrow sections are rigid, but flexible harrows having the tooth bars connected by individual links are available, primarily for rough or stony land.

The weight of spike-tooth harrows averages about 20 to 25 lb per ft of width. Since the draft is light (Appendix C lists 30 to 60 lb per ft of width), multiple-section combinations can cover large areas rapidly with a small amount of power. Both spike-tooth and spring-tooth harrows are available for attachment to the lift linkage on a tractor in such a manner that they retain their flexibility when on the ground but can be raised clear for turning or transport.

9.12. Spring-Tooth Harrows. The spring-tooth harrow is somewhat similar to the spike-tooth, but has long, curved teeth (Fig. 9.5) made from spring steel \( \frac{1}{4} \) to \( \frac{3}{8} \) in. thick and usually about 1\% in. wide. Teeth with various types and widths of points are available for different conditions. The spring-tooth harrow is not used as extensively as the spike-tooth but is
adapted to the cultivation of firm, previously tilled soils to depths as great as 5 or 6 in. Because of the spring action and “give” of the teeth, this tool is suitable for use in rough or stony ground. It is useful for loosening and bringing roots of certain obnoxious grasses and weeds to the surface, for bringing clods to the surface where they can be pulverized, and for renovating or cultivating alfalfa fields.

Spring-tooth harrows generally have three tooth bars (some heavy-duty units have four) with teeth staggered and arranged to till strips 3 to 6 in. wide. Depth is gaged by runners (Fig. 9.5) and adjusted by rotating the tooth bars. Most units are made up of sections 2½ to 6 ft wide, although some have a single, rigid frame for widths as great as 10 ft. Weights usually range from 40 to 60 lb per ft of width, although some heavy-duty harrows have weights as great as 100 lb per ft.

Clyde found that \( R_v \), the resultant soil reaction on the teeth, was substantially horizontal in soft to moderately-firm ground, ranging from 30 to 60 lb per tooth for a depth of 3 to 3½ in. (Appendix C indicates 75 to 150 lb per ft of width.) Figure 9.5 shows the forces and vertical hitching for a spring-tooth harrow. The relations are similar to those shown in Fig. 6.3, \( Q_v \) being the parasitic soil force on the runners and \( DG \) being the resultant of \( W \) and \( R_v \). According to Clyde, the center of gravity of a spring-tooth harrow is a little ahead of the average tooth position. The greatest angle of pull that will allow the teeth to remain in the ground depends upon the relative magnitudes of \( W \) and \( R_v \), as indicated in Fig. 9.5. Hitches that give less slope to the line of pull put less weight on the tractor wheels and more on the harrow runners, both of which are undesirable.
9.13. **Rotary Hoes and Treaders.** The rotary hoe consists of a series of pronged wheels about 18 to 20 in. in diameter (Fig. 9.6), placed approximately 6 in. apart on an axle and rotated by contact with the ground. Two gangs, with the hoe wheels staggered and overlapping, are usually used in tandem, thus giving a 3-in. spacing. The longer gangs are generally arranged in sections connected by hinged joints, to provide flexibility for following ground irregularities.

The rotary hoe, with the wheels rotating in the normal direction indicated in Fig. 9.6, is used for shallow cultivation, crust breaking, and uprooting of small weeds from among established crop plants such as corn, cotton, soybeans, and small grains. The rotary hoe works over the tops of rows as well as between them, without serious injury to well-established plants. The best performance is obtained at relatively high ground speeds (see Section 12.8). Some single-section units have support wheels for depth control and transport, whereas others depend upon weight adjustments. A typical rotary hoe weighs about 50 lb per ft of width per gang.

With the pronged wheels of a rotary hoe reversed (Fig. 9.6), this tool becomes a subsurface packer or treader and is a good seedbed-finishing implement in mulch tillage. It smooths the field, loosens and spreads bunches of residue, breaks up clods, and performs some tillage. The weeding action is improved considerably if the treader is pulled at an angle. Two single gangs operating at an angle can be hitched to oppose each other laterally in an arrangement similar to that of an offset disk harrow. Most rotary hoes, however, are not designed for this "skew" pulling, and bearings may wear rapidly. Only the heavier units are suitable for this application.
9.14. Rollers and Packers. The most common types of surface packers and pulverizers are those consisting of a series of flanged or notched rollers, placed together on an axle (Fig. 9.7a). Because of the nature of the soil surface left by them, implements with this type of flanged roller are known as corrugated rollers. Loose-running sprocket rings are sometimes used between the rollers (Fig. 9.7b) to give some surface mulching. Packers without the sprockets often have two gangs in tandem, oriented so that the rear rollers split out the ridges left by the front gang. Other units have the center holes in the rollers several inches larger than the supporting shaft so that individual rollers can move up or down to follow the contour of planting beds.

9.15. Stalk Cutters and Shredders. With crops such as cotton and corn, that produce large and tough stalks, adequate disposal or size reduction of the crop residue is an important step in preparing the seedbed for the succeeding crop. Large sections of undecayed roots or stalks may interfere with planting, cultivation, and the operation of mechanical harvesters, as well as harboring undesirable insects and pests during the winter. According to Smith,\textsuperscript{17} it is best from the standpoints of decaying of the residue and control of insects to process and plow under cotton stalks and other crop residues while still green or as soon as possible after harvesting.

Rolling stalk cutters have been employed for many years to break up cotton and corn stalks. A rolling cutter merely has a reel-type arrangement with blades parallel to the axis, the reel being rotated by contact with the ground as it is pulled.
along the rows. According to Smith and Miller, a heavy, two-row trailed cutter does a fairly good job of breaking up dry and brittle cotton stalks but often merely mashes down large or green plants.

Power-driven rotary stalk cutters or shredders have been developed within the last few years and are now being used increasingly on a variety of crops. Some units are designed specifically for row crops (usually two-row machines), whereas others cover a solid strip 5 to 7 ft wide and are suitable for either solid plantings or row crops. Some have rigidly mounted radial knives on high-speed cutterheads, and others have free-swinging hammers or flails. At least one commercial unit has two high-speed reel-type cutters (perpendicular to the rows) with a heavy, stationary shear bar suspended from the axle inside of each reel.

The various power-driven cutters or shredders may be classified as (a) those with knives rotating in a horizontal plane, (b) those with knives or beaters rotating in vertical planes perpendicular to the row, and (c) those with knives or beaters rotating in vertical planes parallel to the row (axis perpendicular to the row). Most of these machines depend primarily upon impact to shatter or cut the residue (see Section 14.4). Peripheral speeds of the knives or beaters of all three types are in the order of 8000 to 12,000 fpm.

Cutters with knives rotating in a horizontal plane ordinarily cover a solid strip at least 5 ft wide, often with only a single cutterhead having two radial blades. They may be used for clipping weeds or pasture grasses at any desired height as well as for cutting stalks. The blade support arms may be shaped to create an updraft and thus draw cut and loose material up into the blades for further size reduction. Machines with knives or beaters cutting across the row (axis parallel to the row) have a gathering tunnel or chute on each row just ahead of the cutter to fold the stalks down along the row and hold them for cutting.

Shredders or beaters with the axis of rotation perpendicular to the direction of travel generally have a series of free-swinging hammers or flails attached to the rotor at intervals of about 2 in. along the axis. When equipped with hardened steel hammers supported by either steel arms or short pieces of link chain, they are good for shredding stalks, small brush, heavy cover crops, and the like. When equipped with flexible rubberized flails and
operated at reduced peripheral speeds, these machines are useful for removing the top growth of such crops as sugar beets and potatoes in a preharvest operation.

The results of tests with five models of power-driven stalk cutters in green cotton plants indicate that all units performed satisfactorily at forward speeds of about 3 1/2 to 5 mph. In the various tests, from 64 to 92 per cent of the cut pieces of stalk were less than 3 in. long and 3 to 23 per cent were longer than 6 in.

One of the incentives for the development of power-driven stalk cutters and shredders was to obtain a method for thoroughly shredding corn stalks as an aid in the control of the corn borer in the corn belt. Lilly and his associates report tests with fourteen models to determine their effectiveness in this respect. The degree of control obtained was extremely variable, even in different tests with the same machine. The kill ranged from about 20 to 90 per cent in all their tests. They point out that stalk shredding may be quite effective as an aid to clean plowing and in soil and water conservation where the stalks are cut and the field left unplowed over the winter.

REFERENCES


PROBLEMS

9.1. Calculate the ratio of the tine peripheral speed to the forward speed for each of the rotary-tiller curves in Fig. 9.2 (one revolution of rotor per slice).

9.2. Compare the energy requirements for rotary tilling to a depth of 6 in. with the total energy required for preparing a seedbed by plowing to the same depth with a moldboard plow, disking twice with a light-duty tandem disk harrow, and then finishing with a spike-tooth harrow. For the rotary tiller take \( F = 8.75 \) in., with an energy requirement of 18 ft-lb per 12 cu in. of tilled soil (from Fig. 9.3). The unit draft of the moldboard plow in the same soil was about 7 psi at 2 1/2 to 3 mph. Assume the draft per foot of width is 150 lb for the disk harrow and 40 lb for the spike-tooth harrow.

9.3. (a) At what forward speed could a 54-in. rotary tiller be operated under the conditions of Problem 9.2 if a tractor with a belt or pto rating of 33 hp is to be loaded to 80 per cent of its rating?

(b) What would be the maximum speed if \( F \) is reduced to 4 in.? (See Fig. 9.3.)

(c) What actual drawbar horsepower would be required to pull a four-bottom 14-in. moldboard plow at 3 mph under the conditions of Problem 9.2?

(d) What would be the power requirement for a 10-ft tandem disk harrow at 4 mph?

(e) What would be the power requirement for a 40-ft spike-tooth harrow at 5 mph?
CHAPTER 10

Earth-Moving Equipment

10.1 Introduction. Strictly speaking, any tillage operation involves earth moving. In this chapter the term will be applied only to those operations whose principal function is to modify the contour of a field surface by moving earth from one place to another. Specialty equipment such as trenchers and post-hole diggers will not be discussed.

The farmer may be confronted with any of a number of earth-moving operations, such as terrace building, filling of gullies or depressions, land smoothing for irrigation or surface drainage, construction of drainage ditches, building of earth dams or embankments, construction of ponds, and other similar jobs. Heavy earth-moving equipment designed for construction work is sometimes needed on the farm for certain jobs, but it is usually obtained on a contract or rental basis. Ordinarily the agricultural engineer is not directly concerned with the development of this type of equipment. He does, however, have the responsibility of providing lighter equipment adapted to the power available on the farm.

10.2. Principles and Types of Earth-Moving Equipment. In general, earth-moving involves controlled cutting or loading, moving, and spreading or dumping. Methods for moving the earth after cutting or loading may be described as (a) lifting and rolling, (b) throwing, (c) pushing, and (d) carrying. The first class includes moldboard and disk plows, which are sometimes employed in terrace building. The whirlwind terracer (described in Section 10.12) is an example of the throwing type. Blade scrapers and graders of various kinds, bulldozers, V-drags, and land smoothers are common types of equipment that utilize the pushing or crowding principle (the pushing, however, is often
accompanied by some rolling). Rotary scrapers, the small mounted or trailed scoops, and carryall scrapers carry the soil, either on wheels or by dragging the loaded bowl. Elevating graders, used for open-ditch work and for terrace building, also employ the carrying (conveying) principle.

10.3. Angle-Blade Scrapers and Graders. The most common application for equipment of this type is in crowding soil to one side by operating with the blade at an acute angle from the direction of travel. However, the blade may also be set in the right-angle position for smoothing or moving moderate amounts of dirt by direct pushing. Blades should be adjustable in three planes (vertical pitch, horizontal angle, and side tilt). For general farm use, rear-mounted utility blades in the 5 to 7 ft size range are popular and have many applications, particularly for farmstead jobs and occasional earth-moving or smoothing work in the field. As will be pointed out in Section 10.9, a rear-mounted blade with proper attachment linkage and gaging device can be rather effective as a field smoother.

Trailer, two-wheel blade graders have been developed especially for terrace building. In comparison with four-wheel machines, the two-wheel terracers have a lower first cost, better maneuverability, and better penetration. Other desirable features that any blade terracer should have include an easily reversible blade with means for rapid depth adjustment, ample provision to take side thrust when on the side of a terrace ridge, and sufficient strength and weight to cut well under unfavorable soil conditions.¹

10.4. Push-Type Scrapers. The bulldozer is one of the most common tools that moves earth by pushing. Bulldozer blades are normally mounted on the front of the tractor and are in general much heavier and more rugged than the rear-mounted utility blades. They should be designed to withstand the full power output of the tractor in a direct push or when turning. Some bulldozer blades can be angled, whereas others can be operated only in the right-angle position. The bulldozer is a versatile tool, adaptable to a wide variety of jobs, particularly in construction work.² On the farm it is suitable for such applications as land-clearing, backfilling of ditches, and moving earth over relatively short distances. The bulldozer is generally considered economical for loaded hauls up to a maximum of
about 150 to 200 ft.\(^\text{a}\) It is not well adapted to smoothing operations because of the difficulty in accurately gaging a front-mounted blade.\(^{\text{a}}\)

Trailed scrapers of the push type (commonly known as land levelers) have bottomless bowls, similar to a bulldozer blade with skirts added at the ends. They are carried on wheels that control the depth of cutting or spreading, usually through a remote hydraulic cylinder. Some scrapers of this type have wheels that are interchangeable between positions several feet behind or at the ends of the bowl. The wheels are placed in the rear position for land smoothing and in the side positions for making borders or levees.\(^{\text{a}}\) Widths of cut ranging from 6 to 14 ft are common, with bowl capacities (i.e., amount of dirt being pushed) in the order of 1 to 4 cu yd. The larger units have power requirements up to 50 or 60 hp.

### 10.5. Drag-Type Carrying Scrapers

These tools are distinguished from blade-type scrapers in that the load is supported by the bottom of the scraper bowl behind the cutting edge, rather than being pushed. The rotary or roll-over scraper is a common example of this type. According to Harper,\(^{\text{4}}\) when a roll-over scraper is properly adjusted, filling should take place in not less than 20 ft of travel and can be done at transport speed. When loading is completed, the bowl is tipped backward enough for the blade to clear, and the loaded scraper is then dragged on its bottom to the desired location. Unloading is accomplished by releasing a catch that permits the bowl to rotate forward against adjustable stops whose position determines the depth of spreading or dumping. Subsequent release from these stops permits the bowl to complete the revolution and start loading again.

Roll-over scrapers are relatively inexpensive and are reasonably efficient for making small fills and performing other jobs that involve moderate hauling distances.\(^{\text{4}}\) Bowl capacities are usually in the order of \(\frac{1}{2}\) to 2 cu yd, and power requirements are within the range of the average farm tractor.

### 10.6. Wheel-Type Carrying Scrapers

Self-loading, wheel-type scrapers are available in sizes ranging from two-wheel units with a capacity of 1 cu yd up to the large carryall units with bowl sizes of 30 cu yd or more. This type of scraper is capable of digging its own load, hauling it on wheels, and then spreading
it in controlled layers. Loading is accomplished by means of a blade across the front of the bowl that is pulled through the soil at a controlled depth. As the blade is lifted for transport after loading, a forward lip or apron closes the bowl to keep the dirt from spilling. Since the pull required for loading a wheel scraper is much greater than that needed for transport, it is common practice in construction work with the larger carryall units to provide a pusher tractor to assist during loading.

Wheel scrapers are the most economical type of earth-moving equipment under most conditions involving hauls of any appreciable distance. Figure 10.1a gives a general idea of the relative production capacity of a farm-size wheel scraper (3 cu yd) as compared with a roll-over scraper and with a bulldozer, all operated with a 35-hp tractor. Note that the bulldozer is indicated as being economical only for distances of less than 200 ft, whereas a wheel scraper of the size indicated is suitable for distances up to about 800 ft. The roll-over scraper has somewhat less production capacity than a wheel scraper requiring the same power but is economical for considerably longer hauls than is the bulldozer.

Where conditions permit and the size of the job justifies operation of the larger carryall units, savings as compared with the use of other types of equipment are rather substantial, particularly for the longer hauls. Figure 10.1b shows typical capacities for two sizes of trailed carryall scrapers, the larger one employing a second tractor to assist in loading. Rubber-tired, self-propelled
carryall scrapers are most economical for hauls longer than those indicated in Fig. 10.1b. Although this type of equipment was developed primarily for the large earth-moving jobs encountered in road building and other construction work, it can be used to advantage in some farm earth-moving jobs such as the roughing operation in leveling land for irrigation, the building of earth dams, etc.

**10.7. Land Smoothing.** Land smoothing is an essential operation for irrigated fields and is now being considered in many areas as a means of surface drainage for relatively flat lands. Smooth fields also facilitate the operation of field equipment and are important for certain types of harvesting operations (such as picking up nuts from the ground with a mechanical harvester). In connection with irrigation or drainage, the purpose of land smoothing (often erroneously referred to as land leveling) is to produce a plane surface with a continuous and relatively uniform slope. Land smoothing is a finishing operation performed after any major earth-moving or rough-leveling work has been done with other types of equipment previously discussed. Best results with a land smoother are obtained if the field is covered several times in different directions.

**10.8. Blade-Type, Long-Span Smoothers.** The effectiveness of a land-finishing implement in producing a plane surface is determined largely by its span or bridging effect. Special, long-wheelbase smoothers, often referred to as land planes, have been developed primarily for finishing irrigated fields but are also used to some extent in nonirrigated areas. The land plane consists essentially of a long frame supported at each end by wheels or skids and carrying an adjustable leveling blade at some intermediate point. Effective lengths or spans range from about 30 to 80 ft, with widths of cut from 5 to 15 ft.

When the blade is properly adjusted, a device of this type automatically cuts off the high spots and fills shallow depressions, carrying just enough dirt so that it never completely unloads. It is important that the frame units be rigid so that the height of the cutting edge with respect to the wheels will remain constant for any particular blade adjustment. Likewise, the rear wheels should have sufficient bearing area to support the implement without sinking in appreciably as they roll over a soft fill left by the blade.
Several distinct arrangements are employed by the various manufacturers of long-span smoothers. The conventional arrangement has a stiff frame (Fig. 10.2a), with the distance between the blade and the rear wheels ranging from one-half to one-fourth of the wheelbase. When the blade is mounted behind the midpoint of the span (broken-line outline in Fig. 10.2a), the vertical fluctuations of the blade as the front wheels pass over irregularities are less than if the blade is at the midpoint. How-

![Diagram](attachment:image.png)

**Fig. 10.2. Typical arrangements for long-span land smoothers.**

ever, any fluctuations caused by the rear wheels sinking into soft spots or going over irregularities left by the blade will be greater than if the blade is at the midpoint. The frame arrangement shown in Fig. 10.2b is jointed or articulated, with a linkage arrangement that minimizes vertical fluctuations of the blade due to irregularities encountered by the front wheels.

Another type of smoother, somewhat shorter than the larger sizes of land planes, has a stiff frame supported by a skid in front and by either a skid or wheels at the rear. The blade is at the midposition, with wheels located at each end of the blade and approximately in line with it. The wheels do not support the frame (except in transport) but are connected to the blade through a crank and linkage so that if the wheels go up or down with respect to the main frame the blade moves a lesser distance in the opposite direction. Thus the blade tends to overcut the high spots and overfill with loose dirt in the low spots, which are desirable features.
10.9. Tractor-Mounted Land Smoothers. If a blade scraper with a properly located gaging device is attached to a tractor through a vertically converging-link hitch as indicated in Fig. 10.3, a smoothing device is obtained whose effective span extends from the rear gaging device (wheel or skid) up to the position of the virtual hitch point \( F_v \). The length of span is affected somewhat by the relative heights of the scraper and the tractor wheels, since \( F_v \) shifts as the hitch links are moved.

The linkage arrangement shown in Fig. 10.3 is laid out to place the average virtual hitch point about at or above the front wheels.

![Fig. 10.3. Gaged scraper blade mounted on vertically converging-link hitch.](image)

The broken-line position of the links and the new location of the virtual hitch point at \( F_v' \) show the effect when the rear wheels pass over a ridge. Since \( F_v \) and \( F_v' \) are near the vertical line through the front axle, there is very little effect on the blade height. In a similar manner it can be shown that, as the front wheels pass over a ridge or through a depression, the vertical blade movement is about in proportion to the ratio between the effective span and the distance from the blade to the rear gaging device, as it is with a conventional trailed smoother. If the virtual center is placed ahead of the front wheels, the effect of vertical movement of the front wheels is amplified and an effect from rear-wheel movement is introduced.

10.10. Terracing Equipment. Terracing, as practiced in the United States, is a method of erosion control involving the construction of broad channels across the slope of rolling land in such a manner that they do not interfere with ordinary farming operations. A terracing machine must move the soil a comparatively short distance laterally and lift it from 1 to 2 ft. It should be able to construct terraces at a high rate of speed and to build them effectively and economically on all slopes up to 15 or 20
Other desirable features in a terracing machine include a hitch arrangement to permit sharp turning, the ability to follow the tractor well on sharp curves, and depth control that is independent of the tractor drawbar height. Ideally, the topsoil should be placed at or near the surface of the ridge and a minimum amount of subsoil should be exposed.

Most of the common types of earth-moving equipment have been used for terrace building, including blade scrapers, V-drags, motorized road graders, bulldozers, rotary scrapers, and carryalls.

In addition, such machines as trailed, two-wheel blade graders or terracers (described in Section 10.3), farm-size elevating graders, and the whirlwind terracer have been developed primarily for terrace building. Moldboard plows and disk plows, because of their availability, are practical where the farmer wants to do his own terracing. Disk plows are generally more satisfactory than moldboard plows if the soil is dry or where there is considerable trash.

10.11. Elevating Graders. Machines similar to the elevating graders employed in construction work have been developed for use with ordinary farm tractors in building terraces. Basically, an elevating grader consists of an inclined cross-conveyor and a device such as a plow disk to loosen the soil and deposit it on the conveyor belt. According to Fletcher, the elevating grader is suitable for terrace building where large quantities of soil are to be moved through a distance of 10 to 14 ft into a ridge, but it is more expensive and more complex than other types of terracing machines. Supplementary operations are required to smooth and finish the terrace.

10.12. Whirlwind Terracer. This device is a combination plow and soil thrower developed at Iowa State College in about 1932. It was originally designed as a high-speed pulverator to kill white grubs but was later adapted to terrace building. It consists of a moldboard plow bottom with part of the moldboard removed and a rapidly revolving vertical auger placed in the cut-out section. As the soil leaves the moldboard the auger catches it and throws it to one side. Most of the early tests were made with a 14-in. diameter auger having two 14-in.-pitch blades and operating at about 1060 rpm.
REFERENCES


PROBLEMS

10.1. Analyze and compare the smoothing actions of the three arrangements indicated in Fig. 10.2. Consider fluctuations in blade height caused by the front wheels passing over irregularities, fluctuations in blade height caused when the rear wheels pass over irregularities left by the blade or when they sink into soft spots, and the relative lengths of effective spans in regard to the removal of long irregularities.
11.1. Introduction. Crop planting operations may involve placing of seeds or tubers (such as potatoes) in the soil at a predetermined depth, random scattering or dropping of seeds on the field surface (broadcasting), or setting of plants in the soil. Machines that place the seed in the soil and cover it in the same operation create definite rows. If the rows or planting beds are far enough apart to permit the operation of machinery between them for intertilling or other cultural operations, the result is known as a row-crop planting; otherwise it is considered to be a solid planting. Thus, grain drilled in rows 6 to 14 in. apart is a solid planting, whereas sugar beets, with rows commonly 20 in. apart, are grown as a row crop.

With appropriate planting equipment, seeds may be distributed according to any of the following methods or patterns:

1. Broadcasting (random scattering of seeds over the surface of the field).
2. Drill seeding (random dropping and covering of seeds in furrows to give definite rows).
3. Precision drilling (accurate placing of single seeds at about equal intervals in rows).
4. Hill dropping (random dropping of groups of seeds at about equal intervals in rows).
5. Checkrow planting (accurate and indexed placement of hills or groups of seeds to give rows in two perpendicular directions).

Solid planting is generally done by one of the first two methods, whereas row-crop planting may involve any of the methods except broadcasting.
11.2. Plant Population Requirements. The primary function of any planting operation is to establish an optimum plant population and plant spacing, the ultimate goal being to obtain the maximum net return per acre. Population and spacing requirements are influenced by such factors as the kind of crop, the type of soil, the fertility level of the soil, the amount of moisture available, and the effect of plant and row spacing upon the cost and convenience of operations such as thinning, weed control, cultivation, and harvesting.

With many crops (such as corn, Fig. 11.1) there is a fairly narrow range of plant populations that will give maximum yields under a particular combination of soil and fertility conditions, the optimum number of plants per acre increasing as the productivity of the soil is increased. For other crops such as cotton (Fig. 11.2) and the small grains there appears to be a rather wide range of plant populations over which yields do not vary appreciably, the principal requirement from the yield standpoint being to keep the number of plants per acre above some minimum value.

Most crops can tolerate reasonable variations in uniformity of plant spacing in the row without seriously affecting yields, provided the average population density (or land area per plant) is within the optimum range. Uniform spacing is important with
certain crops, however, for other reasons. Thus, from the standpoint of mechanical harvesting of corn or cotton, drill planting is favored over hill dropping or checkrow planting because it results in more even flow of material into the harvester.

Factors other than the yield are sometimes of considerable importance in establishing the best population for a particular crop and set of conditions. In upright crops such as corn, cotton, and the small grains, increased populations are accompanied by a greater tendency for the stalks to lodge or break, which is undesirable from the harvesting standpoint. Tests with cotton, on the other hand, have shown that increasing the plant population tends to raise the height of the lowest fruiting nodes, which is an aid to mechanical harvesting (see Fig. 10.6).

11.3. Factors Affecting Final Stand. Seedling emergence rates in the field are affected by a number of factors such as the viability of the seed (per cent germination under controlled laboratory conditions), the physical condition of the seedbed, soil moisture conditions, intimacy of contact between the seed and the soil, depth of planting, soil temperature, planter performance, formation of surface crusts after planting, and losses due to diseases, insects, and adverse environmental conditions.

Field emergence rates of 80 to 90 per cent are typical for corn and other crops that tolerate a fairly wide range of planting conditions. In such cases, planting the proper amount of seed to obtain the desired final stand is not a serious problem. With sugar beets and many of the smaller-seed vegetable crops, however, field emergence is so low and unpredictable that it is customary to plant a considerable excess of seed and then thin as required to obtain the desired stand. Cotton may have average emergence rates of only 50 to 60 per cent in some areas, and up to 75 or 80 per cent in others. But, since it can tolerate rather wide variations in plant population without seriously affecting the yield, it can in many cases be planted to final stand by using delinted seed and precision planting equipment.

11.4. Planting-Surface Profiles. Planting may be done on the flat surface of a field, in furrows, or on beds, as illustrated in Fig. 11.3. Furrow planting (or lister planting) is widely practiced under semiarid conditions for row crops such as sorghum, corn, and cotton. In many cases, lister bottoms mounted directly ahead of row-crop planters perform the only tillage in the
seedbed preparation. Grain drills may be equipped with special openers to permit furrow planting in semiarid regions where protection from winter killing is needed. Furrow planting places the seed deep into moist soil, and young plants in the furrows tend to be protected from the weather. With row crops, small weeds in the row can be covered and effectively controlled by moving dirt into the furrow from time to time as the crop plants grow.

Flat planting generally predominates where natural moisture conditions are favorable. In the corn belt much of the corn is flat-planted and checkrowed to permit cross-cultivation for weed control.

Bed planting is common for many row crops in irrigated areas and is often practiced in high-rainfall areas to improve surface drainage. With close-spaced row crops such as sugar beets, lettuce, and certain other vegetable crops, two or more rows are sometimes planted close together on a single bed (Fig. 11.3), thus leaving more width in the spaces between beds for the operation of machinery. For example, beds may be on 40 to 42 in. centers, with two rows 12 to 16 in. apart on each bed. A similar arrange-
ment is sometimes employed in flat planting, with two to five closely spaced rows between wider spaces left for the tractor wheels.

11.5. Mechanical Functions of a Seeding Machine. With the exception of broadcasters, a seed planter is required to perform all the following mechanical functions:

1. Open the seed furrow to the proper depth.
2. Meter the seed.
3. Deposit the seed in the furrow in an acceptable pattern.
4. Cover the seed and compact the soil around the seed to the proper degree for the type of crop involved.

As indicated in Section 11.3, good planter performance in itself does not insure good emergence. Precise performance of the above functions is, however, one of the requirements for obtaining a good stand with crops whose emergence is critical. Since timeliness is of extreme importance in the majority of planting operations, it is desirable that a planter be able to perform these functions accurately at fairly high rates of speed (perhaps 5 mph or more).

The primary function of a broadcaster is to meter the seed and distribute it with reasonable uniformity over a given width of land. Covering is a separate operation or is omitted entirely under some conditions.

11.6. Seed-Metering Devices. Most seed-metering devices may be classified as: (a) those having cells on a moving member, the cells being sized to accommodate single seeds or groups of a few seeds each, (b) the so-called "force-feed" devices, each having a moving member to remove seed from the hopper and discharge it in a more or less continuous stream, and (c) stationary-opening units, usually with an agitator above the opening.

The horizontal-plate planter is the most common example of the cell type. Figure 11.4 shows three types of plates for corn planters. Plates with round or oval holes (Fig. 11.16) instead of the edge cells are also used interchangeably for drilling or hill-dropping of various row crops. A large selection of plates is necessary to meet the requirements for the many types of seeds and spacings. Edge-cell, edge-drop plates are well suited to the planting of relatively large, flat seeds like corn, but the stationary ring surrounding the plate may become worn and thus enlarge
Horizontal-plate planters have spring-loaded cutoff devices that ride on the top of the plate and "wipe" off excess seeds as the cells move beneath them (Fig. 11.4). Spring-loaded knockout devices push the seeds from the cells when they are over the seed tube or boot.

Some planters have vertical rotors with cells around the periphery (Fig. 11.17), and others have inclined-plate metering devices (Fig. 11.5). In a few cases, as in some bean planters, vertical-rotor units merely have seed cups that move up through a shallow layer of seeds, picking up one or two seeds per cup and carrying them over the top of the circle. Another cell-type arrangement has the cells in a flat belt, as illustrated in Fig. 11.6.

With any of the cell-type devices (except on checkrow planters), the average spacing of seeds or hills in the row is determined by the ratio of the linear or peripheral speed of the cells to the forward speed of the planter and by the distance between the cells in the metering unit. Both factors can be changed on most row-crop planters.

The two common types of force-feed metering devices are the fluted wheel (Fig. 11.7) and the internal double run (Fig. 11.8). Although they have been used to some extent on row-crop planters, these devices meter seed less uniformly than do most plate-type units and cannot be used for single-seed drilling. Their principal
application is in drills for solid planting (grain and grass drills). The fluted-wheel feed tends to be more positive in its metering action than is the double run and is generally favored where only relatively small seeds (such as the small grains) are to be handled. Small-diameter fluted wheels are used for metering the small grass seeds. The double-run feed is suitable for larger seeds such as soybeans and peas, as well as being satisfactory for the small grains.

The rate of seeding with the fluted wheel is customarily controlled by moving the wheel axially to change the length of flutes exposed to the grain in the feed cup. Thus an infinite number of selections is available between zero and full exposure of the wheel. (Changing the speed of rotation would also change the rate.) An adjustable lip at the outlet (on the bottom of the feed cup,
Fig. 11.7) can be locked in any one of several positions to accommodate different ranges of seed size or to change the range of seeding rates, and can be opened wide for cleaning.

With the double-run feed the rate of seeding is controlled primarily by changing the speed of the wheel, usually by means of a relatively simple, gear-type speed-selector transmission. One common type of speed-change device has a pinion that is movable along its shaft and can be engaged with any one of several rows of teeth at various diameters on the face of a disk or a cone. A typical drive for a grain drill might include ten gear combinations with two sprocket changes to give a total of twenty speeds, each speed change giving a 10 to 15 per cent change in seeding rate from the preceding value. Only one side of the double-run wheel (Fig. 11.8) is used at once, the choice depending upon the size of seed. Gates in the hopper bottom cover whichever side of the feed cup is not in use. On

some units an adjustable gate in the lower part of the housing inside of the wheel rim gives further control of the range of seeding rates.
In the arrangement of stationary-opening metering devices usually found on broadcasters, the seeding rate is controlled by adjusting the size of the opening, and agitators are provided above the openings to prevent bridging and reduce the effect of seed head on the flow rate. The stationary-opening principle is applied in a little different manner on the unit illustrated in Fig. 11.9, which is used extensively for planting vegetables and to some extent for other row crops. In this device a rotatable plate has a
series of various-size metering holes around the outer part (Fig. 11.9). The plate is turned by hand until the desired size of hole is within the hopper; it is then locked in position and remains stationary during the planting operation. The corrugated-disk agitator, which is usually ground driven (speed ratio not adjustable), merely moves the seed back and forth across the opening. Several interchangeable plates with different hole-size ranges are available.

11.7. **Furrow Openers.** Examples of both rotating and fixed types of furrow openers are shown in Fig. 11.10. The choice among these types or others similar to them is influenced by a number of factors. The optimum depth of planting varies widely with different crops and is influenced by soil moisture conditions, soil temperature, time of year, etc. Some seeds are rather sensitive to environmental conditions and require careful control of the planting depth, whereas others can tolerate a considerable range of conditions. Cotton has occasionally been planted with variable-depth furrow openers in an attempt to have some seeds at the proper depth regardless of changes in conditions.

The full runner is a simple device that works well at medium depths in mellow soil free of trash and weeds. It is suitable for the average conditions encountered by corn and cotton planters. Horizontal, plate-type depth gages may be attached to the runner for soft soils. Various types of furrow-planting attachments are available for use in prepared soils. Angled disk blades, stationary crowder blades, moldboard-type blades, or similar devices are mounted at either side of the runner in such a manner that the runner extends below the furrowing device and deposits the seed at the desired planting depth below the bottom of the main furrow.

The stub runner is sometimes used on corn planters in rough or trashy ground. Hoe-type openers, when equipped with spring trips as shown in Fig. 11.10, are suitable for stony or root-infested soils. They, or similar shovel-type openers, may also be used for deep placement of seeds if the soil is relatively free of trash. Some vegetable planters have shoes or runners considerably smaller than those shown.

Disk-type openers are suitable for hard or trashy ground, and in wet, sticky soils they are more satisfactory than fixed openers because they can be kept reasonably clean with scrapers. The single-disk opener is more effective than the double disk in regard
to penetration and cutting of trash. It is suitable for a wide variety of conditions and is the usual type of opener found on grain drills. With a moldboard attachment as shown at the lower right in Fig. 11.10, it is used for furrow planting of grain, the drilled surface being left as a series of furrows and ridges. Double-disk openers are particularly well adapted to medium or shallow seeding of row crops that are critical in regard to planting depth, because the depth can be controlled rather accurately with removable depth bands (Fig. 11.11).

The furrow-forming wheel shown beneath the hopper in Fig. 11.12 is a recent development resulting from an extensive study of possibilities for improving the emergence of sugar beet seedlings. The wheel presses a V-shaped groove in the soil instead of opening a loose furrow as does a disk or runner. A small shoe immediately behind the wheel holds the furrow open until the seed has been deposited at the bottom of the groove. Depth is controlled by the height of the flange beyond the rim. The wheel

Fig. 11.10. Some of the common types of furrow openers. The single-disk openers are slightly concave; the double disks are flat and contact each other at the front.
Fig. 11.11. An independently mounted row unit for a row-crop planter, showing a double-disk opener with depth bands and a double concave presswheel assembly. Spring pressure is divided between the opener and the presswheel in proportions determined by the point of application on the presswheel pull bars. (International Harvester Co.)

Fig. 11.12. Combination of V-shaped furrow-forming wheel and V-shaped presswheel that has improved emergence of sugar beets. (R. D. Barmington and S. W. McBurney. *Colorado Agr. Expt. Sta. Bull. 420A.*)
could undoubtedly be built with interchangeable depth bands. With this type of opener, even when followed by standard presswheels, the average sugar beet seedling emergence in a considerable number of tests was about 20 per cent greater than with a standard double-disk opener.6

Somewhat the same principle as that illustrated in Fig. 11.12 is utilized to a lesser degree in a grass seeder that deposits seed in corrugations formed by the front gang of a special, corrugated roller.22 The rear rollers then split the front ridges in the usual fashion (see Section 9.14), thus covering the seeds uniformly at a shallow depth and firming the soil around them. The rollers on this seeding unit are about half as wide as on the ordinary corrugated roller, being only 2 in. wide and leaving corrugations about 1 to 1½ in. deep.

11.8. Covering Devices. Among the many types of covering devices employed on seeders are drag chains, drag bars, scraper blades, steel presswheels, rubber-covered or pneumatic presswheels, disk hillers, and various combinations of these units. Ideally, a covering device should place moist soil in contact with the seeds, press the soil firmly around the seeds, cover them to the proper depth, and yet leave the soil directly above the row loose enough to minimize crusting and promote easy emergence.

Some kinds of seed are more critical than others in regard to these factors. Thus, simple drag chains, which merely cover the seeds with loose soil, are satisfactory for grain drills under most conditions where there is ample moisture. In loose, sandy soils or for furrow drilling of grain in heavy residue, narrow presswheels with steel or rubber rims may be used behind the openers; they tend to give increased stands and yields in areas where moisture is a limiting factor. For corn and many other row crops, concave steel presswheels (open-center, closed-center, or double-wheel as in Fig. 11.11) have predominated through the years and have given satisfactory results.

Recent tests have indicated that emergence of certain row crops can be increased by using presswheels of improved design. Smith and Brown22 found in tests with cotton that certain types of rubber-covered wheels gave much faster emergence than was obtained with steel wheels or no pressing, and the resulting yields were greater even though there was not much difference in the final stands. Tests with sugar beets, which normally have a rather
low emergence rate (perhaps 35 per cent), indicate that emergence is increased by using presswheels that exert a high unit pressure on the soil in the vicinity of the seeds. One type of presswheel that has given good results on beets is the V-shaped wheel shown at the right rear in Fig. 11.12. The flange runs directly over the seed, pressing it firmly into the soil, and then a drag chain behind the presswheel (rear center in Fig. 11.12) fills the remaining groove. The flange and rim on this particular wheel are of molded rubber and may be fitted over a standard, concave steel presswheel. Narrow, rubber-tired presswheels (1 in. wide) have been developed for grain drills.

SOLID PLANTING

11.9. Broadcast Seeding. Seeds may be broadcast with centrifugal-type spreaders, with full-width-feed broadcasters that have spaced openings along the full length of a hopper (similar to a grain drill without furrow openers), or by distribution from helicopters or fixed-wing aircraft. If broadcasted seed is to be covered, this is done as a separate operation, usually with a spike-tooth harrow.

The centrifugal-type broadcaster, commonly known as an endgate seeder, provides a rapid and inexpensive method of seeding crops such as the small grains and some grasses. It is particularly useful for fields that are small, wet, irregularly shaped, or have surface or subsurface obstructions. Seed is metered from the hopper by either a fluted-wheel feed or an adjustable opening with an agitator above it. The metered seed is dropped onto one or two horizontal, radially ribbed disks that rotate at a fairly high speed and spread the seed by centrifugal force. The width of strip covered ranges from about 20 to 50 ft, depending upon the physical characteristics of the seed (size, shape, specific gravity, etc.), the speed of the rotating disks, and the height of the disks above the ground. Distribution is not as uniform as with a grain drill and is affected by wind.

11.10. Airplane Seeding. The first reported use of an airplane for seeding rice was in 1929, when it became necessary to replant some California fields that had already been flooded. At the present time most of the rice in California, as well as some in other states, is presoaked and broadcasted onto the flooded
Other crops such as wheat, barley, and pasture grasses have been seeded by airplane to a limited extent. Aircraft are particularly valuable for reseeding hilly range lands or burned-over areas. Pellets containing groups of seeds are often used for range reseeding.

The equipment for aircraft seeding is about the same as that used in aircraft dusting. For a fixed-wing airplane it includes a hopper in the fuselage, an adjustable-gate seed-metering opening with an air-driven agitator in the hopper, and a venturi-type spreader beneath the fuselage (see Fig. 21.13). The usual flying height over relatively smooth terrain is 20 to 30 ft, and the swath covered is about 20 to 25 ft wide (which allows for considerable overlap). Metering devices must be calibrated for a given air speed and type of seed. This is usually done by distributing seed while flying over a known course.

11.11. Grain Drills. In comparison with broadcast seeders, grain drills tend to give higher yields because of the greater uniformity of seed distribution and the more uniform seeding depth. The maximum depth of the furrow openers is controlled in gangs, usually through suitable adjustments in the lift arrangement. Each opener, however, is held down by spring pressure and can raise independently to pass over irregularities.

A fertilizer-grain drill has a divided hopper, the front section being for seed and the rear section for fertilizer (Fig. 11.13). The fertilizer may be deposited through the same tubes with the seed or through separate passages behind the seed tubes. Fertilizing units are also available as removable attachments for plain grain drills. Fertilizing equipment is discussed in more detail in Chapter 13. Attachments for drilling small grass seeds are available for either plain or fertilizer drills. An auxiliary hopper equipped with small-diameter fluted wheels is provided (Fig. 11.13), the grass seed either being deposited through the grain seed tubes or allowed to fall onto the soil behind the furrow openers. The operations of drilling grain, distributing fertilizer, or seeding grasses can be performed independently or in any desired combination.

The size of a grain drill is usually given by the number of furrow openers and their spacing (20 by 6, for example). Spacings generally range from 6 to 14 in., with 6, 7, and 8 in. being common for standard drills. The wider spacings are primarily for furrow
drilling (as with the single-disk, deep-furrow opener in Fig. 11.10). In comparative tests with 6-in. and 7-in. disk spacings and equal seeding rates per acre, Promersberger and Swaller found no difference between the yields of grain or of straw from the two spacings (barley, wheat, and flax were planted). However, the stubble from the 7-in. spacing did not hold up a swath as well as that from the 6-in. rows. An advantage of the wider spacing is the lower initial cost of the drill and the reduced power requirements for pulling the drill.

**ROW-CROP PLANTING**

11.12. Planter Arrangements. Most row-crop planters are adaptable to a variety of crops by merely changing seed plates or hopper bottoms. Two-row and four-row corn planters are probably the most widely used for various crops. With two-row
planters the presswheels can be the main support wheels, but with four-row units flexibility is needed to allow for field surface irregularities. Independent row units can be hinged to a main frame that is supported by separate wheels, the depth of each unit being gaged by its presswheel, or a pair of two-row units can be linked together laterally.

Beet-and-bean and vegetable planters are generally made up of one to six independent row units hinged to the main frame of a trailed planter, mounted on a tractor tool bar, or mounted on a sled that does the final shaping of beds during the planting operation and orients the rows accurately with respect to the beds. The depth of each unit is controlled by bands on double-disk openers, by shoes on runner openers, by the presswheels, or by individual gage wheels.

Seed hoppers may be mounted directly on the individual row units just above the furrow openers, or they may be on the main frame and have telescoping or flexible seed tubes. In the latter arrangement, seed-metering devices are driven from the main support wheels of a trailed planter or, in the case of a mounted planter, from a tractor rear axle or a separate traction wheel provided with the planter. When the hoppers are on the individual row units, seed-metering devices are driven from the presswheels or individual gage wheels, from double-disk openers, or from the main frame through flexible drives.

Fertilizing attachments, with which bands of commercial fertilizer can be placed in the soil at various depths on either or both sides of the seed row (see Chapter 13), are available for most row-crop planters.

11.13. Grading and Processing Seed. One of the requirements for accurate seeding of row crops is that the seeds or seed units be of uniform size and shape and of such a nature that they can readily be singled and handled by cell-type devices. For best results this involves accurate grading of the seed within acceptable size limits, as well as selection of the proper seed plate. Seeds that are too large for a given size of cell will remain in the hopper or will protrude from the cell and be damaged as the plate passes the cutoff. Small seeds will either give multiples in the cells, or the top seed will protrude and may be damaged by the cutoff. Smooth seeds approaching a spherical shape are best adapted to precision planting.
Some kinds of seeds are fairly uniform, but the introduction of hybrid corn varieties has resulted in wide variations in seed size and shape. This poses a difficult grading problem, particularly since each of the three major dimensions must be considered. Accurate classification requires a large number of grades, and

planter seed plates are needed to fit each grade. Improved planter performance, however, may justify the added expense of better graded seed. Grading also tends to improve the quality of the seed by separating out the fractions most likely to have poor germination.

In some cases the individual seed units are actually modified as to size, shape, or surface condition, to make them more suitable for precision drilling. One of the most striking examples from the standpoint of economic savings is the processing of sugar beet seed. Natural sugar beet seed (Fig. 11.14a) consists of seed balls, each of which contains from one to five or more germs or true
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seeds and may produce up to that many seedlings. Before the introduction of processed seed, the laborious task of hand thinning required 15 to 30 man-hr per acre. Two processes have been developed to reduce the number of germs per seed unit. The oldest is a shearing or segmenting procedure, first tried experimentally in 1941. A more recent development is a grinding and decorticating sequence that breaks up the larger seed balls in a preliminary operation and then removes most of the outer corky material by passing the seeds between a neoprene pressure pad and the flat face of a rotating abrasive stone (axis vertical).

The products from both processes have a large percentage of single-germ seeds and have few seeds with more than two germs. Properly graded seed from either process is suitable for precision, single-seed drilling. Decorticated seeds (Fig. 11.14) are smoother and more nearly spherical than segmented seeds and give a somewhat higher field emergence. When processed sugar beet seed is planted with precision-type planters, the labor required for hand thinning is much less than when whole seed is planted, and complete mechanical thinning is feasible. Processed seed is now planted in most of the United States sugar beet acreage, usually at seeding rates of 4 to 6 lb per acre (as compared with 18 to 20 lb per acre formerly used for whole, ungraded seed). Processed sugar beet seed is generally graded (with round-hole screens) to size limits within the over-all range from $\frac{3}{64}$ to $1\frac{1}{64}$ in., with either $\frac{3}{64}$ or $\frac{1}{64}$ in. difference between the upper and lower limits (7 to $\frac{3}{64}$, 7 to $\frac{1}{64}$, etc.).

An outgrowth of the introduction of segmented sugar beet seed was the development of a practical process whereby individual seeds could be coated with an inert material to make the segmented seed units larger and more nearly spherical, primarily to improve planter performance. However, the subsequent development of the decorticating process and of suitable precision planters largely eliminated the need for coating sugar beet seed. Since then, coating or pelleting has been applied to various kinds of small or irregularly shaped seeds such as lettuce (Fig. 11.14b), carrots, other vegetable crops, and some annual flowers. Coating of small seeds that cannot otherwise be drilled singly can result in a considerable reduction in thinning costs for crops such as head lettuce, as well as a reduction in seed requirements.

In one coating process the seeds are placed in a rotating drum
somewhat like a concrete mixer and sprayed lightly with a fine mist of water. Then a small amount of the coating material is added as a fine dust. Additional amounts of spray and dust are added from time to time until the pellets, each containing a single seed, have reached the desired size. The pellets, which are nearly spherical, are then screened to a uniform size and dried slightly. The coating must be substantial enough to withstand handling and shipping and porous enough to permit respiration of the en-

Fig. 11.15. One common type of special hopper-bottom assembly for planting fuzzy or gin-run cotton seed. The inclined fingers on the horizontal plate guide the seed to the opposite-moving vertical picker wheel. Note the agitator rod attached to the top of the horizontal plate. The seeding rate is adjusted by means of a sliding gate that changes the width of the slot through which the picker wheel carries the seeds. (Deere & Co.)

closed seed. It should not absorb moisture from the atmosphere in storage, but should soften rapidly when in contact with moist soil and hold the moisture to the seed to promote germination and growth.¹⁰

Cotton is another type of seed that is much easier to plant uniformly after it has been processed. Because gin-run cotton seed is covered with fuzzy lint (Fig. 11.14c), the seeds stick together in a fluffy mass and will not flow freely. Special planting units (Fig. 11.15) are required, and accurate seed distribution is difficult if not impossible to attain. Seed that has had the lint removed by either mechanical or chemical means can readily be handled by the same types of planting units employed for other row crops, and more uniform planting rates can be obtained than with fuzzy seed. Many thousands of acres of cotton are now being planted to stand, with delinted seed.
11.14. **Row-Crop Drilling.** Planters having horizontal-plate seed-metering devices (Fig. 11.4) are by far the most common for drilling row crops, although planters with stationary-opening metering devices (Fig. 11.9), inclined plates (Fig. 11.5), or vertical rotors (Fig. 11.17) are used to a considerable extent for certain crops. Drill planters are less complicated and less expensive than checkrow planters, are more easily adapted to high-speed operation, and cause less lost time in the field. Drilled stands are favored for mechanical harvesting, and drill seeding is practiced in contour planting for soil erosion control.

11.15. **Precision Drilling.** Since the introduction and widespread adoption of processed sugar beet seed, a tremendous amount of development work has been done to improve planter performance in order to take full advantage of the uniformity and single-germ characteristics of the processed seed. The primary objective in precision drilling of sugar beets and other crops requiring thinning is to obtain single plants with a reasonably uniform spacing in the row so that thinning can be done mechanically or with a minimum amount of hand labor. Planters are available that will do an acceptable job of precision drilling with processed sugar beet seed at speeds up to about 3 mph. The same planters, with appropriate seed plates, are satisfactory for precision drilling of coated vegetable seeds.

The principal requirements for precision drilling, established as a result of extensive development and test work, are as follows:

1. Seeds must be of uniform size and shape, preferably about spherical.
2. The planter cells must be of the proper size for the seed. Plates and other parts of the metering device must be accurately made. With decorticated sugar beet seed, best results are generally obtained when cell diameters are about \( \frac{3}{16} \) in. larger than the upper limit of the seed size range and the plate thickness is slightly greater than the lower limit of the seed size range. A slight taper in the cell walls of horizontal plates (bottom larger than top) promotes easy unloading of the cells.
3. The seeds must have sufficient opportunity to enter the cells. Exposure distance of the cells in the hopper must be ade-
quate, and the linear speed of the cells is important (Section 11.16).

4. A good cutoff device is needed to prevent multiple cell fill and breakage of seed.

5. Unloading of the seeds from the cells must be positive. (See Figs. 11.16 and 11.17.)

6. The seed passageway from the metering device to the furrow should be a small-diameter ($\frac{1}{2}$ to $\frac{3}{4}$ in.), smooth, straight tube, preferably with the discharge end near the bottom of the furrow. When drilling at 3 mph with seeds spaced $1\frac{1}{2}$ in. apart in the row, a seed that is delayed only $\frac{1}{2} \text{sec}$ will be overtaken by the next seed. The flexible steel-ribbon seed tubes used on grain drills (Fig. 11.10) and on some row-crop planters are not suitable for precision drilling because of seed delays due to bouncing within the tube. If the seed hoppers are mounted on the main frame of the planter, rather than on the individual row units, small-diameter telescoping tubes are satisfactory.

7. The seeds should be placed at the proper depth in the soil
and should not be displaced appreciably by bouncing or rolling in the furrow.

11.16. Factors Affecting Cell Fill. Tests with processed sugar beet seed indicate that the per cent cell fill (total number of seeds divided by total number of cells passing the discharge point) for a given planter and seed plate is influenced by such factors as the range of seed sizes in relation to the cell size, the distribution of seed sizes within the range, the shape of the seed, and the linear speed of the cells.\(^5\) In these tests, combinations of seed size, cell size, and plate speed that gave about 95 to 100 per cent cell fill also gave the most uniform seed distribution patterns,\(^3\) and there was a tendency toward minimum seed damage under these conditions.\(^5\) It should be pointed out that 100 per cent cell fill, as defined above, does not necessarily mean that every cell has a single seed but merely implies that any empty cells are offset by extra seeds in multiple fills.

In all the tests, the per cent cell fill for a given combination of seed size and cell size decreased as the linear speed of the plate cells was increased (Fig. 11.18). Thus, best performance for a given combination can be obtained only within a small range of plate speeds. For example, the horizontal-plate planter represented in Fig. 11.18 gave 100 per cent cell fill at a plate speed of 50 fpm and would probably give best results between about 35 and
60 fpm. With 72 cells spaced around a 6\(\frac{3}{16}\)-in. diameter circle, 50 fpm would represent a planting rate of 35 seeds per second. The corresponding forward speed would depend upon the desired spacing of the seeds in the row. A similar plate of the same diameter but with fewer cells would deliver correspondingly fewer seeds per second at the optimum plate speed.

The effect of seed size upon cell fill is indicated by a series of tests with a horizontal-plate planter having \(1\frac{1}{64}\)-in. diameter cells, in which sugar beet seed closely graded to size limits of 7 to \(\frac{5}{16}\) in., 8 to \(\frac{5}{16}\) in., and 9 to \(\frac{3}{16}\) in. gave cell fills of about 140, 100, and 90 per cent, respectively. A cell fill of 100 per cent was obtained with the broader size range of 7 to \(\frac{3}{16}\) in.

11.17. Checkrow Planting. Checkrow planting is a precision operation, the primary function of which is to simplify the control of weeds by permitting cross-cultivation. The checking operation should be accurate to give straight cross-rows, and the desired number of seeds should be dropped in each hill without appreciable scattering. The principal application of checkrow planting is for corn in the corn belt. Improved methods for controlling weeds in the row may, however, permit drilling in many areas where checkrowing has been used in the past.

As a checkrow planter moves across the field, buttons on a wire passing through one of the two checkheads on the planter engage the checkfork and move it backwards with respect to the planter frame, thus opening two valves simultaneously in each seed boot (Fig. 11.19). Seeds that had been resting on the lower valve are ejected a relatively short distance into the furrow, while seeds accumulated on the upper valve are released to fall down onto the lower valve. Spring action closes the valves before the seeds from the upper valve have time to reach the lower valve.

The movement of the checkfork also engages a clutch in the seed-plate drive, which subsequently rotates the feed shaft through one revolution and then automatically disengages. During the one revolution of the feed shaft, the seed plate (with single-seed cells) is moved sufficiently to meter out and accumulate on the upper valve the number of seeds required for a hill. Gear-change arrangements are provided in the drive between the clutch and the seed plate to permit selection of the desired number of cells or seeds per hill (usually two, three, or four). In one type of checkrow planter the seed plates rotate continuously rather than
having a clutch, and a considerable number of speed ratios is available between the usual limits of two and four seeds per hill. This permits closer selection of plant populations (the average number of seeds per hill with this arrangement need not be a whole number).

Fig. 11.19. Valves in the boot of a checkrow planter designed for planting at speeds up to 5 mph. Left: Valves closed, each holding the proper number of seeds for one hill. Right: Both valves are open. Kernels from the lower valve have been ejected into the furrow, and kernels from the upper valve are falling, to be caught by the lower valve when it closes. (J. I. Case Co.)

11.18. Checkrow Valves for High-Speed Operation. Since timeliness of planting is often of extreme importance, present-day checkrow planters are often operated at speeds up to 5 mph and are expected to function satisfactorily at these speeds. For checkrow planters to do an accurate job under these conditions, careful design of the valves is required. The valve pockets should be narrow and V-shaped (as in Fig. 11.19) to minimize bouncing of the seeds as they fall into the pocket and to permit dropping the entire hill in a compact bunch. The seeds at rest on the lower valve should be in contact with the ejecting surface so that they will be pushed out with a minimum of scatter rather than being struck and batted out. The rearward component of the ejection velocity should offset the forward motion of the planter so that the seeds will drop more or less vertically into the furrow.

11.19. The Checkwire and Its Use. Figure 11.20 shows how the checkwire is handled at the ends of the field. The anchor
stake should be placed at least six or eight buttons beyond the ends of the rows, so that the angularity of the wire will not become too great. When the planter reaches the end of the rows the wire is automatically released from the checkhead, the release being actuated either by the wire due to its angularity or by the lifting mechanism of the planter. The operator then turns the tractor and heads into the next rows, after which he moves the anchor stake and places the wire in the checkhead. The stake should be set along a fence line or on some other straight reference line.

One of the problems in the use of the checkwire, particularly with four-row planters, is button travel (longitudinal movement parallel to the rows) due to changing angularity of the wire between the planter and the stakes as the planter moves across the field. In tests with the anchor stake set directly behind the center of a four-row corn planter in a field $\frac{3}{4}$ mile long, Shedd found that the button travel varied from $7\frac{1}{2}$ in. in the center portion of the field to practically zero at the ends. A perfect check in all parts of the field can be obtained only if the button travel is zero or is uniform throughout the length of the row, in which case the checkhead can, if necessary, be moved forward or backward on the planter to compensate for button travel. Shedd

Fig. 11.20. Method of turning two-row and four-row checkrow planters, showing recommended location for tension-meter stake.
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found that setting the stake somewhat to one side of the planter centerline (Fig. 11.20) increased the button travel at the ends without appreciably changing the travel in the center portion of the field, so that a better job of checking was obtained.

A theoretically perfect job of checking can be obtained if the stake is set directly behind the checkhead, but the required length of wire ahead of the planter increases considerably as the planter approaches the end of the row, because of the greater angularity. One of the two common types of stakes for tractor planters (known as the payout stake) has provision to compensate for this increase and is normally set directly behind the checkhead. As the planter approaches the stake, the increasing angularity of the wire releases a ratchet on the stake and permits a drum to pay out rope at a preset tension. When the stake is reset for the next round, the rope is rewound and the ratchet locked. The other common type of stake for tractor planters is known as the tension-meter stake. It is set as in Fig. 11.20 and merely has a gaging device that aids in obtaining the same tension on the wire each time.

One of the serious disadvantages of checkrow planting is the extra time required to handle the wire and move the stakes. In time studies with experienced operators on a number of farms, Hansen found that an average of 1 min was required to dismount from the tractor, set a tension-meter stake, and get under way again. The average time for the payout type was 1 1/2 min. Since this loss of time is proportional to area and inversely proportional to the length of rows, it becomes increasingly serious as planting speeds are increased (see Chapter 2) and as row lengths are reduced (see Problem 11.3). Simple calculations show that the time required for setting a payout stake is as great as the travel time for a 40-rod (1/2-mile) row at 5 mph.

11.20. Hill Dropping. Checkrow planters can be used for hill dropping without the checkwire, but the hills will not be in cross alignment. In this application the clutch on the feed shaft remains engaged and an arm on the feed shaft trips the valves once each revolution. Hill dropping can also be accomplished with a rotary valve of the type shown in Fig. 11.21, which is generally driven from the planter feed shaft. Properly designed rotary valves are more suitable for high-speed planting than are the lip or plunger valves on checkrow planters.
Seed plates with cells large enough to hold a full hill of seeds can be used for hill dropping, but there is likely to be considerable scattering of the seeds, particularly at high forward speeds. Seed dispersion may result from time delays as the seeds leave the plate cells (cell dispersion), time delays as they fall through the seed tube (seed-tube dispersion), or scattering when they strike the furrow.

In tests with acid-delinted cotton seed, Autry and Schroeder investigated the factors affecting seed dispersion from hill-drop plates in a conventional horizontal-plate planter (Fig. 11.4) without boot valves. Seed dispersion was determined with a greased-board test setup of the type described in Section 11.21. In these tests, cell dispersion was minimized by using plates with short, broad cells rather than long, narrow ones (less unloading time) and by having a star-wheel type of knockout device rather than the roller type. A smooth seed tube, designed to fit the trajectories of the seeds leaving the plate and give small impact angles, was much superior to the conventional straight tube with funnel top. The trajectory-type tube designed by Autry and Schroeder had a straight taper from the bottom end to the larger-diameter top.

Slow plate speeds (25 to 40 fpm) gave the least dispersion. With an 8 to 1 ratio between forward speed and plate speed, dispersion on the greased board (which does not include the ground-scattering effect) was moderately greater at 4½ mph than at lower speeds and was a minimum at about 3 mph. The number of cells in the plate (4 to 16) had little effect on dispersion. Therefore, when changing hill spacing, it is best to change the number of cells rather than plate speed.

With the trajectory-type seed tube, cell and seed-tube dispersions were not affected by the height of fall. Ground scattering,
however, was decidedly affected by the height of fall, since it is related to the kinetic energy of the falling seed. The effect can be reduced by keeping the hopper as low as possible and by restricting the furrow just ahead of the seed tube to eliminate bouncing in that direction. The effect of forward speed on ground scatter was not investigated.

11.21. Planter Testing. Although the ultimate criterion for evaluating a complete planting operation is the stand obtained in the field, these results are influenced by seed viability and environmental factors beyond the control of the planting machine. The effects of different types of furrow openers or presswheels can only be determined by field emergence trials, but the performance of a seed-metering device can more readily be checked in the laboratory by any of several methods.

The regularity of seed spacing can be observed rather simply by mounting the hopper and metering device on a suitable stand and passing a grease-coated board beneath the seed tube at a rate representing the ground speed of the planter. The resulting seed pattern is representative of the performance of the metering device with its seed tube but does not show the effect of bouncing in the furrow. One test unit has been devised that actually simulates a furrow, so that the results include some effect of bouncing. Photosensitive devices and electronic units have also been used to record the paths or frequencies of falling seeds.

One of the problems in determining regularity of seed spacing is to arrive at a single criterion of performance for direct comparison of different metering units or modifications of the same unit. Brooks and Baker have developed a statistical method whereby a dispersion coefficient for drill planters is obtained from a mathematical analysis of the grease-board distribution pattern. As the value of the dispersion coefficient approaches zero, the seed distribution pattern approaches absolute uniformity. Field tests have substantiated the validity of this method of analysis; i.e., planters giving low dispersion coefficients in the laboratory have given the best performance in the field.

The most common method of determining cell fill is to weigh the seed collected while a certain number of cells pass the discharge point, count the number of seeds in a known weight, and then calculate the total number of seeds collected. Damage is determined on a weight basis by screening out the broken particles of
Another method is to count the number of seeds collected or recorded by one of the arrangements for determining regularity of spacing. The counting method gives a direct indication of the number of seeds discharged by each individual cell and the probability of a cell's selecting zero, one, two, or any other number of seeds, whereas the weighing method merely indicates the over-all average.

With any cumulative-type dropping mechanism (such as check-row and hill-drop planters using single-seed plates) the accuracy of selection for a complete hill is somewhat lower than the accuracy of each individual selection and is influenced by the number of individual selections involved per hill. If the accuracy of selection by individual cells is known, the expected accuracy of accumulation for any size hill can be predicted from mathematical probability expressions developed by Collins and Morrison.

11.22. Potato Planting. Potatoes are generally grown from seed pieces cut from the whole tubers, although small potatoes are sometimes planted without cutting. Since planting rates are in the order of 800 to 1500 lb per acre, large seed hoppers are necessary. Fertilizing units with hoppers holding several hundred pounds per row are available for most potato planters and are integral units on some of them. The fertilizer is deposited in bands on either or both sides of the row by means of disk openers. The seed furrows are generally made with runner or shoe-type openers. Concave covering disks bury the seed pieces to a depth of perhaps 4 in. and leave a ridge over each row.

Automatic potato planters have vertical, rotating picker wheels with devices to either pierce or grip individual seed pieces and then drop them into the furrow. Semiautomatic or assisted-feed planters require the assistance of an operator to see that each pocket on a horizontal, rotating feed ring has a seed piece in it. Planters of the semiautomatic type are not adaptable to high-speed operation. Trailed one-, two-, and four-row automatic planters are available that can be operated at speeds up to 5 mph.

High-speed automatic planters generally have feed mechanisms of the picker-pin type (Fig. 11.22). Each arm or head of the picker wheel has two sharp picking pins that pierce a seed piece in the picking chamber (at point D in Fig. 11.22), carry it over to the front, and then release it above the furrow (at point C in Fig. 11.22). To avoid excessive picker-head speeds in high-speed
planting, two wheels per row, each with six to eight picker heads, are mounted side by side with the arms staggered. The position of the picker pins on each head is adjustable to accommodate various sizes of seed pieces. The spacing of seed pieces in the row is controlled by the speed ratio between the ground wheels and the picker wheels.

**TRANSPLANTING**

Crops such as cabbage, tomatoes, tobacco, sweet potatoes, and a number of others are generally propagated in special beds and then transplanted in the field. If any appreciable acreage is involved, mechanical transplanting aids are frequently employed. Mechanical plant setters are also suitable for planting small trees in reforestation work and for other kinds of nursery stock. Plant-setting rates generally range from 1500 to 2000 or more plants per hour for each man on the transplanter, provided the forward speed is such that the men are working at full capacity.

**11.23. Basic Components of a Transplanter.** The essential components of the simpler transplanting machines are a furrow opener, provision for carrying a supply of plants, low seats for
operators who place the plants directly in the furrow, and press-wheels or other suitable devices to cover the roots and firm the soil around them. **Spacing of the plants in the row is by judgement, sometimes assisted by a mechanical clicking device.** A water supply tank is usually provided, with appropriate valving for either intermittent application of water around each plant or continuous application along the row.

Machines designed for setting crop plants in prepared seedbeds are generally equipped with runner-type furrow openers. The principal requirements are that the furrow be of uniform depth and that it have sufficient width to accommodate the plant roots without crowding. But, for planting trees on sod or burned areas without previous tillage, Bruhn and Trenk found that better results could be obtained with a heavy trencher consisting of a lister bottom to remove sod and brush from the planting strip, a runner-type opener directly beneath the lister, and an inclined snout across the front of the opener to raise loose dirt and leave it in position for covering the roots behind the opener.

**11.24. Semiautomatic Plant Setters.** A number of transplanting machines have mechanical transfer devices that are hand-fed but which automatically place the plants in the furrow. This arrangement permits the operators to work in more comfortable positions and tends to give more uniform placement in the furrow. The transfer device must be carefully designed to insure that the plants will not be damaged and to preclude any

![Fig. 11.23. A transplanter with a mechanical transfer device that automatically places plants in the furrow. (Deere & Co.)](image-url)
CROP PLANTING

possibility of injury to the operator. Proper timing of the release is required so that the plants will remain upright as the roots are covered.

Fig. 11.23 illustrates one type of mechanical transfer device. The large rubber disks are normally held together by radial, spring-steel fingers; they are spread apart for loading, and again for unloading, by ball-bearing rollers mounted between them at appropriate positions. The tops of the plants are placed between the disks, with the roots protruding. Plant spacing on this particular unit is in response to a mechanical clicker or bell, driven from a presswheel.

Another type of mechanical transfer device has rubber-lined gripping pockets mounted on a conveyor chain. Opening and closing of the grippers is by cam action. With this arrangement the spacing of plants in the row is uniform and is controlled by the speed ratio between the ground wheels and the chain.

REFERENCES

PRINCIPLES OF FARM MACHINERY


PROBLEMS

11.1. The fertility of a particular field is such that maximum corn yields are obtained with a population of 15,000 plants per acre. The rows are to be 40 in. apart, and an average emergence of 85 per cent is expected.

(a) How many seeds per hill should be planted, if the hills are 40 in. apart?

(b) What would be the seed spacing if the crop is drilled?

11.2. Seeds spaced 1½ in. apart at 3 mph (such as sugar beets) are metered at the rate of about 35 seeds per second.
(a) Compare this with the rate when planting corn at 5 mph with three-kernel hills spaced 40 in. apart.

(b) If the corn seed plate has a diameter of 7 in., rotates continuously, and has 16 single-seed edge cells, what is the linear cell speed in fpm?

11.3. Plot curves of field efficiency versus length of field (up to \( \frac{1}{2} \) mile) for drill planting and for checkrow planting at speeds of 3 and 5 mph with a four-row planter and a 40-in. row spacing. Consider the average turning time at each end to be 20 sec for either method and either down-the-row speed (assuming the operator throttles down from 5 mph to turn). The time consumed in filling seed hoppers, in making adjustments, and by miscellaneous interruptions amounts to 1.0 min per acre. Assume that the time required to set the tension-meter stakes in checkrow planting is the same as the average value given in Section 11.19.

11.4. A vertical-rotor planter with a metering device similar to the one in Fig. 11.17 is operated without a seed tube, the seeds being released at the lowest point of travel and falling freely by gravity to the furrow bottom 3\( \frac{1}{2} \) in. below. The peripheral speed of the rotor is 70 fpm, and the ground speed is 3 mph. How far does a seed move horizontally (indicate whether forward or backward) between the point of discharge and the point of impact in the furrow,

(a) If the rotor turns in the same direction as the ground wheels?
(b) If the rotation is opposite?
(c) At what angle from the vertical does the seed strike the bottom of the furrow in each case? Neglect air resistance.

11.5. A horizontal-plate planter has plates with 72 cells on a 6\( \frac{1}{4} \)-in. diameter circle. The diameter of the ground wheels is 26 in. A 15-tooth sprocket on the main axle drives an 8-tooth sprocket on the feed shaft. The final drive is through a 9-tooth bevel gear on the feed shaft and a 20-tooth gear on the plate drive shaft.

(a) Calculate the seed spacing in the row.

(b) If sugar beets are to be planted and the per cent cell fill is represented by the curve in Fig. 11.18, what forward speed (mph) would give 100 per cent cell fill?

(c) What forward speed could be used for 100 per cent fill if the sprockets were changed to give only 6 seeds per foot?

11.6. In calibrating a 16 X 7 grain drill, one wheel is jacked up and the seed from each delivery tube is collected while the wheel is turned (by hand) through a given number of revolutions at about the speed to be used in the field. The loaded radius of the pneumatic tires is 14 in., and the estimated amount of wheel slippage in the field is 15 per cent (see Section 22.4). For the seed-meter adjustment selected, the chart furnished by the manufacturer indicates a seeding rate of 1.2 bu of wheat per acre.

(a) At about how many revolutions per minute should the wheel be turned to be equivalent to a field speed of 5 mph?

(b) If only half of the drill is calibrated at one time, how many revolu-
tions of the wheel are required, to be equivalent to covering \( \frac{1}{4} \) acre in the field?

(c) The total amount of seed collected from half the drill during the number of wheel revolutions indicated in (b) is 20.5 lb, and the measured bushel weight of the wheat is 61 lb. What is the actual seeding rate in bushels per acre?

(d) By what correction factor should the chart values be multiplied to obtain the actual seeding rate for wheat with a particular seed-meter adjustment?
Chapter 12

Row-Crop Cultivation, Flaming, and Thinning

12.1. Introduction. Since a large part of this chapter is devoted to equipment and practices whose primary function is to control weeds, a brief discussion of the general problem and the methods of combating weeds seems appropriate. The control of weeds and grasses has always been one of the greatest time- and labor-consuming operations in the production of crops. The average annual loss due to weeds in the United States has been estimated by the USDA as $5 billion.\textsuperscript{1}

In addition to requiring extensive control measures, weeds rob nutrients and water from the crop plants, often serve as hosts to insects and other pests that prey on crop plants, and create equipment problems, especially in harvesting and processing of certain crops.

12.2. Methods of Controlling Weeds. Mechanical cultivation or tillage is still the most important method for controlling weeds and is generally the most economical method where it can be used.\textsuperscript{10} The weeds may be uprooted, covered, or cut off. The principal problem in the tillage method is the killing of weeds and grasses in the crop row.

In the early stages of growth for some crops, implements such as the rotary hoe (see Section 9.13) and the spring-tine or pencil-point weeder can be operated directly over the rows to uproot small weed seedlings from among established crop plants. The action of these tools depends upon differential resistance between the weeds and the crop plants, and there is generally some crop mortality. Indiscriminate or “over-all” coverage of both the middles and the rows with tools of this type is fast and economical, and power requirements are low. Individual-row, rotary-hoe at-
Attachments are often mounted between the sweeps or shovels of a row cultivator for early cultivations (Section 12.8).

Selective burning or flaming shows some promise for control of in-the-row weeds in certain crops, such as cotton, sugar cane, and corn, whose stems are not injured by a short exposure to an intense heat. This method, however, cannot be applied during the early stages of crop growth.

The use of chemicals for weed control has increased tremendously in the past decade, primarily because of the impetus given by the introduction of selective weed killers such as 2,4-D. Selective chemicals, if properly chosen and applied, will kill weeds in certain crops without injury to the crop.

In pre-emergence spraying of row crops, general-contact herbicides (which kill most kinds of vegetation and are not selective) are applied directly over the rows (or as a continuous blanket spray) after the seed has been planted but before emergence of the crop plants. The weeds are allowed and encouraged to emerge on the beds before planting, and then the soil in the rows is disturbed as little as possible just before and during planting, so that additional weed seeds will not be brought to the surface layers for germination.

Directed post-emergence spraying with general-contact weed killers is practical with row crops that are resistant to injury from such sprays applied at the plant base. The nozzles must be carefully placed and directed so that the spray does not strike susceptible portions of the crop plants. Shields are sometimes provided as additional protection. Spraying and the equipment for chemical weed control are discussed more fully in Chapter 21.

Sometimes the crop plant itself is effective in combating weeds by shading them, but other methods of control are required until shade takes over. It might be said that any weed-control problem can be solved with the hand hoe or by finger weeding, but these are time-consuming, tedious, and costly operations. However, a moderate amount of hand hoeing is sometimes needed to clean up the rows after applying other control methods.

12.3. Selection of Weed-Control Method. In general, the selection of a method or methods for controlling weeds is influenced by the type and age of the crop, the type and size of the weeds or grasses, timeliness, the equipment available, etc. Good weed control usually involves a combination of the available
methods, plus timeliness and good farming. Weeds and grasses are most effectively controlled when small (not more than 1 or 2 in. tall), especially when the control methods involve uprooting, covering, flaming, or the application of directed post-emergence general-contact sprays. Larger weeds in checkrowed or cross-blocked crops can be removed reasonably well by cross-cultivation, but if the weeds in drilled rows get out of control, hand hoeing is about the only method available for their removal.

ROW-CROP CULTIVATORS

Cultivation of row crops refers primarily to tillage operations performed after the seed has been planted. The general purpose of cultivation is to promote plant growth, the most important reason for cultivating being to kill weeds. In irrigated sections additional functions are to prepare the land for the application of irrigation water and to improve water penetration. In certain crops, preparation of the field for harvesting operations is an important consideration in the final cultivations. Incorporation of chemical fertilizers into the soil is another function of cultivation.

Row-crop cultivators range in size from small, hand-pushed units up to four-row, tractor-mounted cultivators with effective widths of 13 to 14 ft or more. Horse-drawn cultivators of both the walking and the riding types have, of course, been common for many years, but in most areas of the United States they have now been largely replaced by tractor-mounted units. Since the development and production of trailed tractor cultivators for row crops are now rather limited, the present-day farm machinery designer is interested primarily in tractor-mounted cultivators.

12.4. Tractors for Mounted Row-Crop Cultivators. The tricycle-type, all-purpose tractor with adjustable rear-wheel tread was developed primarily for mounted cultivators and other row-crop equipment. Since the general adoption of hydraulic controls for mounted equipment, the trend to mounted cultivators, as well as to other types of mounted equipment, has been rapid. Four-wheel tractors with adjustable front-wheel tread are available in addition to the tricycle type with single or dual front wheel. The tricycle-type tractor is not suitable for straddling a single row or bed (or any odd number). A recent development for the cultivation of tall row crops is the high-clearance tractor with
minimum vertical clearances of more than 30 in. beneath the axles and frame. Small tractors with rear-mounted engines have been developed to provide improved visibility for close cultivation of vegetable crops.

12.5. Types of Mounted Cultivators. Two common types of mounted row-crop cultivators are illustrated in Fig. 12.1. The separated-gang cultivator (Fig. 12.1b) is designed for a specific number of rows and has either one or two gangs per row. The tool bars of the individual gangs drop down between the rows, an arrangement that provides maximum vertical clearance for the plants. A cultivator of this type is often known as a cotton-and-corn cultivator, since they are the crops for which it is most used. The size of a separated-gang cultivator is designated as the number of rows covered (usually one, two, or four rows). The gang spacing is adjustable for the row spacings ordinarily encountered in cotton, corn, and similar crops.

The continuous-tool-bar cultivator shown in Fig. 12.1a (one version is known as a beet-and-bean or vegetable cultivator) has tool bars that extend across the tops of the rows rather than dropping down between them. The front unit may have divided tool bars with right- and left-hand gangs, or it may consist of a single gang with tool bars continuous across the full width. The rear-mounted unit of a continuous-tool-bar cultivator is usually a single gang. Front gangs commonly have double tool bars (one ahead of the other, as in Fig. 12.1a) to permit staggering of tools. Rear gangs often have this arrangement. The continuous-tool-bar cultivator is adaptable to a wide range of row spacings, the maximum number of rows depending upon the length of tool bar and the row spacing. Good lateral stability between tools is characteristic of this type of cultivator and is an essential requirement for cultivating extremely close to the rows. Vertical clearance is limited by the maximum practical length of the tool standards.

Cultivating units of either type may be front-mounted (as shown) or rear-mounted or may include both front and rear gangs. Front-mounted cultivators afford good visibility of the work and direct response to steering. Rear-mounted gangs permit loosening of the soil behind the tractor wheels and are adaptable to simple and easy attachment.

If a rear-mounted cultivator is on a laterally rigid hitch, the initial response to steering is in the opposite direction from that
Fig. 12.1. Two general types of mounted row-crop cultivators. (a) Continuous-tool-bar. (b) Separated-gang.
of the front wheels. Thus, such an arrangement is not satisfactory for cultivating close to the plants. This problem of rear-mounted cultivators can be largely overcome by a hitch that provides a real or virtual horizontal hitch point in front of the rear axle, as explained in Chapter 6 (Section 6.19). A guiding fin or coulter (Fig. 12.2) provides lateral stability for the tool and

Fig. 12.2. A rear-mounted cultivator for a three-point hitch with horizontally converging, pivotal links. Note the rigidly mounted guiding coulter and the plant shields. (Ford Motor Co.)

makes it directional. A visual steering guide is attached to the front axle of the tractor, directly over a row.

12.6. Attachment of Cultivators to Tractors. Attachment and dismounting of tractor cultivators should involve a minimum of time and physical exertion and should not require the services of more than one person. With rear-mounted units these conditions can be met without a great deal of difficulty, since the tractor can be merely backed up to a cultivator unit, and appropriate hydraulically operated lift links attached. If the lift linkage is considered to be part of the tractor (as with most three-point hitches), such an arrangement lends itself readily to standardization and interchangeability.
CULTIVATION, FLAMING, AND THINNING

Front-mounted cultivators normally have parts of one or more gangs directly behind the front wheels (Fig. 12.1), so that some hand maneuvering and lifting of the cultivator parts are necessary in attaching or detaching most present-day units. In addition, most of the frame and lift linkage, particularly with the larger units, must be attached or removed along with the gangs each time. This further increases the time and manpower requirements, as well as limiting the possibilities for utilizing the tractor lift system to help. Rigidity of the frame after attachment should not be sacrificed in attempting to simplify the mounting.

Although there has been considerable improvement in the mounting arrangements of some front-mounted cultivators, others still require heavy manual lifting and an excessive amount of time for attaching or detaching. Interchangeability of front-mounted cultivators undoubtedly warrants further attention, in addition to the problem of ease of attachment.

12.7. Lift Arrangements and Depth Control. Most front-mounted cultivators are attached through parallel or nearly parallel links so that, when a gang is raised or lowered, the tool pitch is not affected and the depths of all tools on a gang are changed by equal amounts. Rear-mounted cultivators with a single tool bar often have a simple, single-axis hitch, since all tools on the bar will be in about the same fore-and-aft position. Other rear-mounted units on which the linkage is part of the cultivator rather than part of the tractor have parallel-link systems. Where the same hitch serves for the cultivator and for other rear-mounted implements, two-axis hitches with vertically converging links are the rule.

Hydraulic controls have largely replaced hand and mechanical lifts for row-crop cultivators, with the exception of some of the cultivators for small tractors. Depth may be controlled by any of the methods described in Chapter 5 or by gage wheels on individually floating gangs. Gage wheels are especially desirable for wide cultivating units and under conditions where the field is uneven, soft, or sandy. They permit each gang to follow ground-surface irregularities with a minimum of variation in depth.

The lift linkage often has springs to push or hold down on the gang and thus aid in obtaining penetration; otherwise the
only downward forces are the weight of the gang and the downward component of the soil forces (suction) on the tools. In a few cases, heavy-duty cultivators are held down rigidly by a double-acting hydraulic cylinder.

As discussed in Chapter 6, the maximum depth obtainable with a parallel-link hitch, when operated with free links (and gage wheels) or when depth is controlled by a single-acting lift, is that which causes the resultant of all useful soil forces \( R_e \) and the weight of the gang \( W \) to become parallel with the hitch links. If hold-down springs are used, their effect upon the gang should be combined with the weight. Thus, the penetration with a parallel-lift linkage can be improved by changing the cultivator setup to raise the rear ends of the links or lower the front ends. With the single-axis arrangement sometimes employed on rear gangs, the maximum depth is reached when the resultant of \( W \) (including spring effects) and \( R_e \) passes through the pivot axis.

With a cultivator having both front and rear gangs, some method of delaying the raising or lowering of the rear gang is desirable so that all operations can be started and stopped at about the same position with respect to the ends of the rows. Two hydraulic cylinders operated from a single control valve can be arranged to provide an automatic delay for the rear gang as described in Chapter 5 (Section 5.12), or the front and rear gangs can be lifted (or lowered) selectively through two cylinders with independent controls. A third arrangement gives automatic delay in lifting and independent manual control for lowering.

12.8. Cultivating Tools and Attachments. Many types and combinations of cultivating tools are used in row crops, the selection being influenced by such factors as type and size of crop plants, soil type and field condition, and the purposes for which cultivation is being performed. A few of the common types of shovels and sweeps are illustrated in Fig. 12.3. Various sizes and shapes of these tools are available. Among the many other types of equipment are tools such as disk weeder and disk hiller for moving dirt to or from the row, rotary hoe units, and special rotary cultivating equipment for row crops.

In the close cultivation of small plants, row shields (stationary or rotating) are often needed to prevent covering of the plants with dirt or clods (Fig. 12.2). When working in large plants,
special fenders or shields are sometimes needed to prevent the plants from being damaged by the tractor wheels or the cultivator frame.

Sweeps are used extensively for weed control, since shallow cultivation is generally desired. Special sweeps have been developed that can be operated at high speeds (often 5 mph or more in crops such as corn and cotton) without throwing excessive amounts of dirt. Where subsurface or mulch tillage is practiced, the crop residue tends to clog sweeps and prevent penetration. In these areas, disk hillers have given good results.

Rotary-hoe attachments for row-crop cultivators (Fig. 12.4) are very effective at 5 to 6 mph for pre-emergence and early post-emergence cultivation of corn, cotton, and other crops. These attachments have groups of three or four wheels similar to those on the field rotary-hoe units described in Chapter 9 (Section 9.13). They are run directly over the crop rows, between the front sweeps or shovels. In addition to breaking any crust that might be present and uprooting small, tender weeds, the hoe wheels serve as excellent rotary shields for the sweeps or other shovels in the middles. They allow sweeps to work at high speeds without cover-
ing, even when the plants are small. In cotton, rotary hoe units can be used until the plants are 8 to 10 in. tall.  

**12.9. Cultivator Adjustments.** The three general adjustments for cultivator tools are the relative horizontal positions (lateral and fore-and-aft), the depth, and the pitch. In addition, some tools, such as disk hillers and other tools for moving dirt to or from the row, require a directional adjustment. This adjustment may be obtained by using standards whose upper portions have a circular cross-section so they can be rotated in the clamps. Vertical adjustment of the standards in the clamps is provided so that the relative depths of tools on a gang can be adjusted. Clamps should be easy to remove or shift.

Standards for separated-gang cultivators ordinarily have an adjustable joint for changing the pitch of individual tools, whereas
continuous-tool-bar cultivators generally have one-piece standards and provision for rotation of the entire tool bar or gang to adjust the pitch. Some tools, such as weed knives (Fig. 12.3), have slotted holes for pitch adjustment at the point of attachment to the standards.

12.10. Cultivator Setup. In general, a cultivator is set up for the same number of rows that were planted at one time, since the guess rows (row spaces between adjacent planter trips) will always have some variation in width. For close cultivation it is important that the cultivator tool spacings correspond accurately with the spacing of the planter units. The tools that are to be operated close to the rows are mounted on the front gangs. Rear gangs are ordinarily fitted with tools to till or loosen the centers of the row spaces.

A floor diagram affords a convenient and accurate means of setting up and adjusting a cultivator before going to the field. In this system a line diagram is painted on a concrete floor, paving, or other permanent, flat surface. Lines are accurately laid out to represent the tractor centerline, crop rows, and the centers of the middles.

After the tractor has been driven onto the floor and properly aligned on the diagram, the gangs are lowered to a working height and the tools are attached in the desired relation to the rows and middles. If all tools (sweeps, for example) are to run at the same depth, they will be placed so that they all touch the floor and have the proper pitch (usually flat for sweeps and weed knives). If various tools are to run at different depths, their heights above floor level can be set accordingly. Some types of tools, particularly those that may be rotated about the vertical axis to control the amount of dirt moved to or from the row, may need final adjustment in the field. In general, however, the floor diagram will give a more accurate setup than could be obtained in the field. The same diagram can be used for spacing the units on other row-crop implements such as planters.

12.11. Power Requirements. Mounted row-crop cultivators are in general considerably overpowered, particularly for the shallow cultivation in which the primary objective is weed control. As an example, tests with a two-row mounted cultivator (18 dbhp tractor) working in cotton on a light sandy-loam soil in North Carolina gave the following results:
Standard 11-sweep cultivator at 3 mph 3.8 hp
Standard 11-sweep cultivator at 5 mph 5.2 hp
Seven sweeps and 2 rotary-hoe attachments at 5.1 mph 7.5 hp

These figures include the power required to overcome the rolling resistance of the tractor, which accounted for 30 to 40 per cent of the totals shown.

12.12. Protective Devices for Cultivator Standards. Continuous-tool-bar cultivators seldom have any protection for individual tools or standards, but spring-trip standards are common on cotton-and-corn cultivators. Their purpose is to provide overload protection in case the tool encounters a stone, root, or other solid object. Other types of protective devices include wooden break pins and friction releases.

A typical arrangement for a spring trip is shown in Fig. 12.5. Clyde lists the following as desirable characteristics for a spring trip:

Fig. 12.5. Spring trip on a cultivator standard.

1. The force on the shovel required to operate the spring trip should be large enough at the start of tripping to hold the shovel rigid until an obstruction is encountered and should then decrease as the shovel moves back, particularly near the end of the travel.
2. The spring trip should provide enough force to automatically return the shovel to its working position under normal conditions.

Clyde also points out that an inherent defect in most spring trips is that the pivot point is located behind the point of the tool rather than above or slightly ahead of it, which causes the shovel to try to go deeper when the device trips (Fig. 12.5). The results are high stresses and a tendency for the shovel to fail to return to its working position without help.

12.13. Rotary Cultivation. Within the last few years considerable interest has developed in rotary tillage for cultivation of certain row crops, particularly those with close-spaced rows
such as sugar beets and various vegetable crops. One of the early applications for rotary cultivators was for controlling weeds in peat soils, where sweeps and other cultivator shovels have a tendency to merely push or displace the weeds in the loose soil without killing them.

Special multirow units, driven from the pto, have been developed for mounting on row-crop tractors. When equipped with suitable shields, such units can be operated extremely close to young plants with small root systems. It is not uncommon to leave an untilled row strip only 2½ or 3 in. wide in young sugar beets and vegetable crops. Such close cultivation, with either rotary cultivators or other cultivating tools, must be done at low forward speeds to permit accurate steering and reduce operator fatigue.

Two types of rotary cultivating units are illustrated in Fig. 12.6, one with L-shaped knives mounted more or less radially on central hubs (as on the rotary tillers discussed in Chapter 9) and one having straight blades mounted parallel to the axis of rotation. With either type, the widths of the individual units can be changed to accommodate various row spacings. The vertical crop clearance for rotary cultivators of the types shown is limited by the radius of the tillage units, since the support shaft is continuous...
across the rows. Another type (not shown) has individual row units with separate drives from a common, elevated shaft, thus providing greater clearance over the rows.

Rotary cultivators equipped with longitudinal blades or overlapping knives as shown are very effective weders and mulchers. As with other rotary tillers (see Chapter 9), the degree of pulverization can be controlled to some extent by the relation of forward travel to rotor speed, and the unit energy requirements are reduced by increasing this ratio (i.e., by taking a longer “bite” per knife or blade).

FLAME WEEDING

Control of unwanted vegetation by flaming has been practiced to a limited extent for many years in such places as railroad right-of-ways, drainage ditches, etc. However, the practical application of selective flaming to control weeds and grasses within a specific crop is relatively new, dating back to the early 1940's.

12.14. Principles of Selective Flame Weeding. The differentiation in burning depends upon the weeds being small and tender and upon the crop plants having stems that are resistant to the intense heat and being tall enough so that flame directed at the ground in the row will not strike the leaves or other tender parts. Since the flame is likely to be deflected up into the row by ridges of dirt or by large clods, it is essential that the plant beds be as flat and smooth as possible. To be most effective, flame weeding must be done when the weeds and grasses are not over 1 or 2 in. tall. The theory is that the intensity of heat (fuel rate) and the time of exposure (forward speed) are adjusted so that enough heat is applied to the weeds and grasses to cause expansion of the liquid in the plant cells and consequent rupture of the cell walls but not enough heat to cause actual combustion. Thus, the effect of flaming may not become fully apparent until several hours after the operation has been performed.

12.15. Applications. Most of the early development work on flame weoders was in cotton, sugar cane, and pineapples, but flaming has been successfully tried in a number of other crops. Wright reports that flame weeding was successful in tests with field corn over 6 in. tall, sweet corn, set onions, seed onions over 12 in. tall, potatoes, lima beans, and some types of nursery stock. He reports tests on other crops, including soybeans and a number
of vegetable crops, in which the results showed only fair promise or were discouraging. Barr reports the initiation of tests on sugar cane in Louisiana in 1942 and states that about 70 flame weeder were being used in this crop in 1944.

By far the greatest amount of development work and the most extensive application of flame weeder has been in cotton, where the control of weeds and grasses in the row is a serious problem, particularly in relation to mechanical harvesting. Cotton is well suited to flame weeding, although it does not become resistant to the action of the flame until the plant reaches a height of 6 to 10 in. and has a stem diameter of about \( \frac{3}{16} \) in. at the base. Thus, in the early stages of cotton growth, weeds in the row must be controlled by other means such as directed oil sprays or rotary-hoe attachments.

Flaming and shallow cultivation of cotton are usually carried on as a combined operation during midseason growth of the crop, sweeps being used to clear the middles of weeds and the flames being so directed as to destroy the small weeds and grass in the rows. The operation is repeated at intervals of 5 to 10 days, since flaming is most effective when the weeds are not over 1 or 2 in. tall. Periods of extended rainfall may result in weed infestations so thick and vigorous that flame weeder will not be effective in cleaning the rows. Flaming alone is used if weed control is necessary during the latter part of the season after the cotton has been "laid by" (i.e., cultivation discontinued) and until the first bolls open. According to Meek, flaming of cotton is not the complete answer in weed control, but when properly done under favorable conditions it is an effective method that reduces weed-control costs to some extent and hand-labor requirements tremendously and has no significant effect on cotton yields. Fuel requirements are in the order of 4 gal per acre per flaming, and the fuel cost is perhaps 60¢ per acre.

12.16. Fuels and Distribution System. The early models of flame weeder (before 1946) burned diesel fuel or kerosene in air-atomizing burners and required an air compressor. Present units operate on LP (liquefied petroleum) gas exclusively. LP gas for agricultural applications is usually propane or a mixture of butane and propane. These fuels exist in the gaseous form at ordinary atmospheric pressure and temperatures but liquefy when subjected to moderate pressures. Thus LP gas can be stored and
transported in pressurized tanks as a liquid, and at normal temperatures the tank supplies the pressure needed by the burners.

The components of a typical LP-gas fuel system for a flame weeder are shown in Fig. 12.7b. In the system shown, liquid is withdrawn from the bottom of the tank and the heat for vaporization is obtained from the cooling water of the tractor engine.

Heat could also be obtained from the engine exhaust, but usually not so conveniently. It is possible to obtain vapor directly from the top of the tank; but, with the high rates needed on four-row units, the tank temperature under some ambient conditions is soon reduced to the point where insufficient pressure is available.

The quick cut-off valve (Fig. 12.7b), which should be where the operator can readily reach it, is used to shut off the burners during turns or for midfield stops. A small internal hole or orifice through the valve provides enough fuel to keep pilot flames burning during these temporary shut-offs. The tank valve (or some other valve in the line) is used for complete shut-off.

The tank is generally mounted on the rear of the tractor. It

![Diagram of LP-gas fuel system for a flame weeder.](image)
should be large enough to hold at least half a day’s fuel supply, which means about a 100-gal tank for a four-row unit.

12.17. Burners and Their Placement. The original oil burners and the first LP-gas burners had round cross-sections. A recent development is the flat burner shown in Fig. 12.7a, which gives a broad, short, and comparatively thin flame.14 Whereas the round burners were set at an angle of only 15 to 20° above horizontal, the flat burners can be set at a much steeper slope (Fig. 12.7a) and are less sensitive to deflection by ground-surface irregularities in the row. Exposure times at a given forward speed are greater than with the round burner, which tends to increase the effectiveness.

The flat burner, which is built to about the proportions illustrated, has a standard fan-type weed-spraying nozzle as the fuel jet. The usual size is rated at 0.4 gpm of water at 40 psi and has a spray angle of 40° with water at this pressure. Jones 14 measured the flow rates for LP gas through a particular nozzle of this size and obtained values of 1.52, 2.02, and 2.25 gph at pressures of 20, 30, and 40 psi, respectively. With nozzles of this size and two burners per row, operating speeds commonly range from 2½ to 4½ mph, nozzle pressures being between 30 and 55 psi. In a series of tests in cotton, with pressures of 20 to 50 psi at a speed of 3 mph, Tavernetti and Miller 10 found no significant effect on the yield, although there was some temporary visible damage at 40 and 50 psi.

The proper placement of flat burners with respect to the row, as recommended for cotton, is indicated in Fig. 12.7a. The burners are inclined at an angle of about 45° and should be set so the flame strikes the ground 2 or 3 in. from the row center, on the near side. In plan view the burners are staggered to prevent any flame interference that might cause deflection upward into the crop plants.

Because of the tremendous heat output from the burners (1½ to 2 million Btu per hr from a four-row unit with 8 burners), it is highly desirable from the standpoint of the operator and the tractor to mount the burners at the rear. There has, however, been some objection by safety engineers to the rear mounting because of the proximity of the fuel tank. For this reason some installations have the burners at the front. To give accurate con-
trol of height, the original round burners were supported by skids hinged to the tool bar or a similar support. But, because the present flat burners can be set at steeper slopes and are therefore less sensitive to variations in height, they are sometimes attached rigidly to the tool bar.

PLANT THINNING

12.18. Reasons for Thinning. With some row crops, such as sugar beets, cotton, and many vegetables, emergence rates are generally low and rather unpredictable because of the inability to control all the pertinent factors. With these crops it is common practice to plant a considerable excess of seed and then to thin the plants to the desired stand after emergence. Ideally, it would be desirable to eliminate the thinning operation by planting to the desired final stand. Evidence indicates that, with cotton, it is possible in many cases to do this without affecting yields adversely 16 (see Fig. 11.2).

In addition to reducing the plant population, thinning removes or thins the weeds in the row. Consequently, if thinning is eliminated, the early weeds must be adequately controlled by other means.

12.19. Methods of Thinning Plants. Blocking or thinning of row crops may be accomplished by hand methods, with mechanical devices, with flame, or with chemicals. Hand thinning, in addition to being tedious work, is costly, involves high peak labor requirements, and may necessitate staggered plantings to permit timely thinning.

Hand thinning is basically a selective operation, but present mechanical thinners (including flame and chemical units) are random stand reducers. They take out weak or strong plants indiscriminately and may leave multiple plants or no plant at all in the block or space skipped. They do, however, give a reasonably consistent average percentage reduction in stand. Crops such as cotton and sugar beets may be thinned mechanically with no serious reduction in yields and with a considerable saving in cost.5,19 Although good stands are highly desirable for random mechanical thinning, uniformity of distribution (in terms of
plants per foot of row) is more important. The theoretical proportion of the total plants removed is equal to the ratio between the length of block cut out and the center distance between blocks.

12.20. Mechanical Blockers and Thinners. Mechanical thinning may be done either with machines that operate across the rows at right angles to them or with down-the-row thinners. Cross-blocking is accomplished by cross-cultivation with sweeps, knives, or other cutting tools of the proper width mounted on a cultivator tool bar so as to give the desired widths of cuts and skips. Since this operation is performed by traveling across the rows, it is suitable only for flat-planted fields. Depth control is not as good as with down-the-row blockers.

Down-the-row mechanical thinners may be of the power-driven rotary type (driven from the tractor pto or from a ground wheel on the thinner), power-driven oscillating type, or direct ground-driven rotary type. The axis of rotation of the power-driven cutting head may be parallel to the direction of travel or at an angle from it. Various types of cutting units are used, as indicated in Fig. 12.8. Different lengths of blades are available for types a and c, whereas the knives of type b are adjustable (by rotation in their sockets) to leave various lengths of skips or blocks. The number of blades on type c may be varied by means of interchangeable heads. Type d is a spring-tine unit for random thinning of very small weeds from among larger crop plants. The power-driven rotary units are often arranged so that individual row units are "floating," with each unit having its own depth-gage wheel.

The direct ground-driven thinner rotates because of contact with the ground and therefore can only be operated with its axis at some angle from the direction of travel (in a position similar to that shown for type a in Fig. 12.8). It commonly consists of a disk-like arrangement with cut-out sections and may have circumferential knives attached to the front side of the remaining sections. The oscillating type has cutting elements on arms suspended from a cross-bar, with a pto-driven mechanism to swing the cutters back and forth across the rows at regular intervals.

12.21. Action of Rotary, Down-the-Row Thinners. The vector diagram in Fig. 12.9a indicates the general velocity relations for a rotary thinner and Fig. 12.9b shows the paths of suc-
Fig. 12.8. Various types of cutting heads for power-driven, down-the-row, rotary thinners.

Fig. 12.9. (a) Velocity relations for a down-the-row, rotary thinner. (b) Paths of successive blades with respect to the row. (Peripheral distances and velocities are shown in the horizontal plane.)
cessive blades with respect to the row. The various symbols are defined as follows:

\[ V_f = \text{forward velocity of machine.} \]
\[ V_b = \text{peripheral velocity of blade with respect to the machine.} \]
\[ V_r = \text{resultant velocity of blade with respect to the row.} \]
\[ \theta = \text{angle between plane of rotation and direction of forward travel, in degrees.} \]
\[ \alpha = \text{angle between } V_r \text{ and direction of forward travel, in degrees.} \]
\[ \beta = \text{angle between the direction of forward travel and the axis drawn through the blade extremities that define the width of sweep } W, \text{ in degrees.} \]
\[ D = \text{diameter of cutting head.} \]
\[ N = \text{number of blades on cutting head.} \]
\[ w = \text{effective width of sweep of blade, perpendicular to } V_r. \]
\[ S = \text{blade spacing around periphery} = \pi D/N. \]
\[ b = \text{effective blade length.} \]
\[ L = \text{center distance between blocks in the row.} \]
\[ L_c = \text{length of row cut out.} \]
\[ L_s = \text{length of row skipped (length of block).} \]

Since a blade rotates through the peripheral distance \( S \) in the same time that the machine moves forward a distance \( L \), it follows that

\[ L = S \frac{V_f}{V_b} = \frac{\pi D V_f}{NV_b} \quad (12.1) \]

From the geometry of Fig. 12.9b it can be shown that

\[ L_c = \frac{b \sin(\beta + \alpha)}{\sin \alpha} \quad (12.2) \]

\[ \frac{L_c}{L} = \frac{b \sin(\beta + \alpha)}{S \sin(\theta + \alpha)} \quad (12.3) \]

and

\[ \frac{L_s}{L} = 1 - \frac{L_c}{L} \quad (12.4) \]

If the axis of rotation is parallel to the line of travel (\( \theta = 90^\circ \)), \( \alpha \) will always be an acute angle. When \( \theta \) is less than 90° (as in
When a rotary thinner is driven by direct contact with the ground, the direction of skidding (along \( V_r \)) must be approximately along the axis of rotation, assuming there is no appreciable resistance to rotation in the bearing and axle assembly. Since \( V_r \) is then approximately at right angles to \( V_b \), \( \theta + \alpha = 90^\circ \) (Fig. 12.9a) in this case.

12.22. Determination of Required Setup. To determine the required stand reduction for random mechanical thinning of sugar beets, for example, stand counts are first made to determine the average number of hills* per 100 in. of row (or number of plants in some crops). Enough counts should be taken over the field to give a good average. The ratio of the desired final stand to the average initial stand count gives the percentage of plants to be retained. A suitable setup for the thinning unit is then selected to obtain the desired result, as discussed below. The stand reduction may be accomplished in a once-over operation if the initial stand is not too heavy. But if more than half the plants are to be removed, a twice-over sequence with a down-the-row thinner is generally used for sugar beets. The second operation is done with narrower knives than used for the first time over and with more cuts per foot (usually twice as many).

Each time a random mechanical thinner is run over the field, the theoretical percentage of plants left, in terms of the stand just before the operation in question, is \( 100 \times \frac{L_2}{L_1} \). For a twice-over operation, the ratio of the final stand to the initial stand, in per cent, is

\[
R = 100\left(\frac{L_{s1}}{L_1}\right)\left(\frac{L_{s2}}{L_2}\right)
\]  

(12.5)

Assume, for example, that an initial sugar beet stand of 30 beet hills per 100 in. of row is thinned first with a setup that leaves 55 per cent of the stand \((L_1/L = 0.55)\) and then with a second combination that leaves 65 per cent of the hills remaining after the first operation. The theoretical average stand after the first operation will be \(30 \times 0.55 = 16.5\) hills per 100 in. of row. After the second time over, the final average stand should be \(30 \times 0.55 \times 0.65 = 10.7\) hills per 100 in.

* Present practice for sugar beets is to consider that each 1-in. increment of row containing one or more plants is a hill.
12.23. Flame and Chemical Thinners. A flame thinner has burners of the same type used for flame weeding but, in addition, has wheels perhaps 2 ft in diameter that run directly on the rows. Each wheel has metal boxes or hoods at intervals around its periphery to cover and protect the plants in the blocks to be saved. If two burners are directed at each row from opposite sides, they should be placed in the same fore-and-aft position (not staggered as for flame weeding). This orientation is desirable to give maximum flame interference and promote spreading on the row, since the function of the burners is to kill all plants in the unprotected areas. In some units, burners are suspended from the axles inside of the wheels and are aimed directly downward onto the rows.

Chemical thinners, which have been tried experimentally to a limited extent, are similar to flame thinners except that a general-contact herbicide is sprayed onto the unprotected parts of the row instead of using flame. Either flame or chemical thinning must be done when the crop plants are small, and both methods have the advantage of not disturbing the soil in the row.

12.24. Selective Mechanical Thinning. Although all the previously described mechanical blockers and thinners are of the random type, at least one machine that is selective has been developed. It has been used to a limited extent in sugar beets and some vegetable crops. This machine employs phototubes or "electric eyes" to select the plants that are to be saved. An enclosed box, containing a phototube and having a window in the bottom, passes over the row, close to the ground. By reflection of light from the foliage, it locates the first crop plant occurring beyond the preselected distance from the preceding plant that was retained. Through a "memory" device, the impulse resulting from the phototube signal then causes a pair of intermittent-rotating knives to make a cut across the row on either side of the selected plant after the phototube unit has passed over it. The knives then stop until the phototube locates the next plant to be saved. After each cycle the phototube is automatically rendered inoperative until the preselected plant spacing has been covered along the row.

This device can be adjusted to select only the larger crop plants and, until the plants grow into a solid row, is suitable for weeding in operations subsequent to thinning. Although it is a costly and
complex mechanism, the electronic thinner does illustrate the broad applicability of the various branches of engineering to the field of farm machinery.

REFERENCES

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PROBLEMS

12.1. The total lifting time for each gang of a row-crop cultivator with hydraulic controls and delayed-action lift is $1\frac{1}{2}$ sec at rated engine speed. The distance between the front and rear gangs is 12 ft. Assume that two-thirds of the total lifting time is with the tools above ground.

(a) Plot forward travel while tools are lifting in the soil versus miles per hour (up to 5 mph).

(b) On the same graph plot required time delay (seconds) between start of lifting of front and rear gangs versus miles per hour.

(c) With a delayed-action valve of the type shown in Fig. 5.7 (Chapter 5), what would be the actual delay in lifting time for this tractor at rated engine speed?

(d) For what forward speed (at rated engine speed) would this be the proper amount of delay on this cultivator?

12.2. Determine the rate of cultivating, in acres per hour, when operating a four-row tractor cultivator at 5 mph in corn with 40-in. rows. Assume a field efficiency of 75 per cent.

12.3. Consider the special case of a ground-driven disk thinner with no extra blades attached (i.e., merely a cut-out disk).

(a) Draw diagrams similar to those of Fig. 12.9, indicating the various symbols involved, and develop expressions for $L$, $L_\omega$, and $L_\omega/L$, in terms of $b$, $S$, and $\theta$. Neglect friction in the axle bearings of the disk.

(b) Check your results by substituting appropriate angle relations in equations 12.2 and 12.3.

12.4. With the aid of the equations developed in Problem 12.3, design a ground-driven disk-type thinner to leave 3-in. blocks on 10-in. centers. Select a disk whose diameter is an even number of whole inches. The cut-out notches should be at least 2 in. wide (measured circumferentially) to permit easy passage of clods and trash, and $\theta$ should not exceed 40°.

12.5. The average initial-stand count in a field of sugar beets is 28 hills per 100 in. of row. The desired final stand is 10 hills per 100 in. The follow-
ing cutting units (type c, Fig. 12.8) are available for a down-the-row thinner: 10-knife head with \( L_c = 2 \) in., 10-knife head with \( L_o = 1 \frac{3}{2} \) in., 20-knife head with \( L_o = 1 \) in., and 20-knife head with \( L_o = \frac{3}{2} \) in. The cutting head makes one revolution per 40 in. of forward travel. Recommend the head to be used in each operation of a twice-over sequence, and calculate the probable average final stand for this combination.
13.1. Introduction. Fertilizers are applied to the soil to increase the available supply of plant nutrients (principally nitrogen, phosphorus, and potassium) and thus promote greater yields or better crop quality. The use of commercial fertilizers has increased steadily during recent years, with over 18 million tons being applied to United States crops in 1950. As the total crop production requirements for our expanding population continue to increase, fertilizers and their proper placement in the soil will become more and more important.

13.2. Application of Manure. It has been estimated that the production of manure on American farms amounts to about 1 billion tons annually, but that only one-fourth to one-third of its potential value is actually utilized on crop and pasture land. The remainder is lost by misplacement, drainage, leaching, or fermentation. In addition to the immediate effect of increasing the available nutrients, manure has the long-time effect of increasing the supply of humus. Thus, it acts as both a fertilizer and a soil conditioner.

Manure is applied to the surface of the soil with special spreading machines and is generally incorporated into the soil by plowing or diskng (except for light top dressings added to growing crops such as grain or pasture). The manure must be placed below the surface if losses from drying out are to be prevented, and it should be incorporated so as to admit sufficient air for decomposition. Application rates are generally in the order of 6 to 12 tons per acre, although heavier rates are sometimes used, as well as light top dressings of 2 to 4 tons per acre. A manure rate of 8 tons per acre is equivalent in nutrient value to about 1000 lb of a 20-unit mixed commercial fertilizer.

* Mixed commercial fertilizers are sold on the basis of chemical composi-
A manure spreader is basically a special-purpose farm wagon or trailer with a mechanical unloading and spreading arrangement (Fig. 13.1). A chain-and-slat conveyor moves the load to the rear, where two beaters and a widespread distributor with spiral blades (augers or paddles) shred it and distribute it over a strip of land usually 6 to 8 ft wide. In the most common arrangement, the beaters, distributor, and conveyor are all ground driven, the conveyor having a ratchet drive with an adjustable-stroke pawl (Fig. 13.1). Some of the larger manure spreaders, however, emission in terms of the three important elements, often designated as NPK. For example, a 4-12-4 fertilizer contains 4 per cent nitrogen, 12 per cent phosphoric acid (P₂O₅), and 4 per cent potash (K₂O), for a total of 20 units of plant-food constituents.⁹

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Fig. 13.1. **Top:** Rear view of a manure spreader. The beaters and the widespread distributor all rotate counterclockwise as viewed from the right. **Bottom:** Ratchet-type drive for conveyor. The feed control arrangement limits the return stroke of the pawl. (J. I. Case Co.)
ploy pto drives for all the components or for all except the conveyor. This type of drive permits operation of the spreader under more adverse field conditions.

Tractor-drawn manure spreaders are ordinarily of the twowheel type, since maneuverability in the barnyard is important. It is frequently desirable to unhitch from the spreader, either to release the tractor for some other purpose while the manure is being loaded or to use the same tractor with a front-mounted manure bucket for the loading operation. Adjustable or automatic tongue stands are provided on most spreaders to facilitate hitching and unhitching. Capacities of manure spreaders range from about 40 to 120 bu, with 60 to 90 bu sizes being most common. Bottoms are ordinarily of treated wood and the sides may be of either wood or steel. The rear width of the box is usually an inch or two greater than the front width, to facilitate movement of the load by the conveyor.

In some cases, general-purpose wagons with pto-driven, conveyor-type unloaders, designed primarily for hauling or feeding chopped forages and the like (see Chapter 22), have attachments to adapt them for manure spreading.

13.3. Commercial Fertilizers. Although commercial fertilizers are predominantly dry materials, plant nutrients may also be supplied as liquids, dissolved materials, or gases. Dry fertilizers are applied in connection with practically every kind of field operation performed in the production of crops. Some of the application methods are as follows:

1. Broadcasted before plowing or placed at the plowing depth by a distributor on the plow that drops fertilizer in each furrow.
2. Deep placement with chisel-type cultivators.
3. Broadcasted and mixed into the soil (or drilled into the soil) after plowing and before planting.
4. Applied during the planting operation.
5. Side-dressing applications on growing row crops (generally during a cultivating operation) or top dressings on solid-planted crops.
6. Drilled into established pastures and other sods with special equipment.
Application rates of 100 to 500 lb per acre are common for many crops, although rates may exceed 1000 lb per acre.\textsuperscript{9,13} The use of aqua ammonia as a source of nitrogen is well established throughout the western part of the United States, and anhydrous ammonia is becoming increasingly popular in certain areas of the country.\textsuperscript{2,11} Aqua ammonia is a solution of ammonia in water and is a liquid under ordinary conditions. Anhydrous ammonia (NH\textsubscript{3}) is a water-soluble gas at normal temperatures and atmospheric pressure but is stored and handled as a liquid under pressure in special cylinders or tanks similar to the LP-gas tanks required for flame weeding. The vapor pressure of liquefied ammonia is approximately 125 psi (gage) at 75°F and 250 psi at 115°F. Anhydrous ammonia contains approximately 82 per cent nitrogen, as compared with 25 to 40 per cent for aqua ammonia and lesser amounts for dry fertilizers. It is a cheaper source of nitrogen than are other fertilizing materials but introduces problems in transportation and storage under pressure.

The application of fertilizer materials through irrigation water has long been a common practice in certain areas of the United States, particularly in California.\textsuperscript{2} Dry materials, dissolved materials, or liquids are introduced into the irrigation supply stream by means of special metering devices. Since its introduction in 1931, the practice of injecting anhydrous ammonia into irrigation water has spread rapidly in many areas of the West. The liquid anhydrous ammonia is supplied in special cylinders (usually 150-lb capacity) from which it is metered by a simple orifice and discharged through a pipe into the irrigation stream below the water surface. The application of fertilizers through irrigation water is a relatively simple practice and is particularly advantageous when quick stimulation is desired at certain stages of plant development. Careful control of the irrigation water is required to obtain uniform distribution of the fertilizer.

Anhydrous ammonia, aqua ammonia, and other liquids or solutions are also applied to the soil by means of field implements, although it is only during the last few years that this practice has been followed to any appreciable extent.\textsuperscript{12} Equipment for the injection of liquids and gases into the soil is discussed in the latter part of this chapter.

Inorganic (chemical) fertilizers dissolved in water may be applied directly to plants as a spray, with the assurance that at
least part of the material will be absorbed. The main purpose of such an application is to quickly overcome some particular mineral deficiency that would otherwise seriously impair the growth or yield of the plant. Excessive concentrations will injure the leaves or fruit, and spray applications need to be repeated at frequent intervals.

13.4. Soil Amendments. Materials such as lime, gypsum, and the newly developed synthetic soil conditioners are not fertilizers, but are used to improve the chemical or physical condition of a soil. Lime is useful for correcting soil acidity and is the most common commercial soil amendment. It is ordinarily applied before planting and then worked into the soil, although it can be applied to a crop at any stage without injury. Either centrifugal broadcasters or full-width-feed broadcasters are used. Application rates are much higher than for commercial fertilizers, ranging from 1000 lb per acre up to several tons per acre.

13.5. Placement of Commercial Fertilizers. Because movement of fertilizers in the soil is very limited, proper placement in relation to the seeds or plant roots is important for maximum response and the most efficient utilization of the nutrients. A comprehensive report on this subject has been prepared by the National Joint Committee on Fertilizer Application. This report, published in 1948, summarizes the important principles involved in proper placement and gives specific recommendations for various crops, based upon the results of many fertilizer experiments.

Localized placement of fertilizers near the seeds at the time of planting (rather than uniform distribution over the entire area) favors early stimulation of the seedlings and results in more effective utilization of the plant nutrients. However, excessive concentrations of soluble nutrients in contact with the seeds or small roots may seriously injure the initial roots or even impair germination. Best results with most row crops have been obtained when the bands were a little below the level of the seed (as much as 3 in.) and spaced 1 to 4 in. laterally from the row on one or both sides. This placement is generally accomplished with applicators that are independent from the seed furrow opener (Fig. 13.2) and are adjustable both vertically and laterally. Limiting the application to a single band per row re-
Fig. 13.2. Independent fertilizer applicators (shown on a runner-type planter) for placing bands to one side of the seed row and at any desired depth with respect to the seed level. From left to right: runner, disk, and deep-placement types. (International Harvester Co.)

results in simplified mechanical construction of the distributor and less disturbance of the seedbed.

When a split-band fertilizer boot (Fig. 13.3) is used for hill-planting of corn, the fertilizer is deposited in short bands on each side of the hill and is separated from the seed by \( \frac{1}{2} \) to 1 in. of fertilizer-free soil, usually in a depth zone at or a little above the seed level.\(^{12}\) Research indicates, however, that placement a little below seed level is preferable.\(^{12}\) In Fig. 13.3 the seed is covered with soil beneath the V-shaped divider that splits the fertilizer stream. A valve above the divider operates simultaneously with the seed valves when hill-dropping or check-planting and remains open when drilling. Because of the closeness of the fertilizer to the seed, only relatively light applications (not over 300 lb per acre)
are recommended for check-planting or hill-dropping. Larger amounts are safe if the fertilizer is drilled in continuous bands.

With small grains it is recommended that the seed and the fertilizer be drilled simultaneously, which places the fertilizer close to and in partial contact with the seed. This method has given better results than separate application by either drilling or broadcasting. Side-band application has not been investigated. Most grain drills have provision to permit dropping the fertilizer either through the same tube with the seed or through a separate passage directly behind the seed tube (see Fig. 11.13).

Fertilizers applied as side dressings are of most immediate benefit when placed in moist soil within the root zone, but excessive mechanical destruction of the root system must be avoided. Side dressings are ordinarily applied in conjunction with a cultivating operation. The fertilizer is generally dropped in furrows opened by regular cultivator shovels, but it can be placed at other locations or depths with separate openers such as the one shown at the right in Fig. 13.2.

APPLICATION OF DRY COMMERCIAL FERTILIZERS

13.6. Types of Equipment. Because of the wide variation in types of fertilizers and operating conditions encountered, fertilizer equipment is one of the most diversified classes of farm machinery on the market. In general, however, the various devices may be classified as those that broadcast the material onto the surface of the ground and those designed for placing the fertilizer in rows or bands beneath the surface.

Centrifugal fertilizer broadcasters are similar to centrifugal seed broadcasters (see Section 11.9), in that the material is metered from a hopper and distributed by horizontally rotating, ribbed disks. Uniformity of distribution across the broadcasted strip is likely to be poor and is affected by wind. The principal application for centrifugal broadcasters in the United States is in spreading lime. Some manure spreaders have centrifugal attachments available to adapt them for lime spreading.

Drop-type or full-width-feed broadcasters have adjustable openings or other metering units spaced at regular intervals along the full length of a hopper (Fig. 13.10). Machines of this type
are suitable for spreading either lime or fertilizers. Some full-width-feed broadcasters have furrow openers available for placing bands of fertilizer at various depths and spacings, either in tilled soil or in pasture and other forage-crop sods. Row crops can be side-dressed with these machines by plugging part of the outlets.

In certain areas fertilizers are broadcast by aircraft, particularly on crops such as rice and the small grains and on hilly pasture lands. In 1952, airplanes in the United States were flown more than 56,000 hr for fertilizing agricultural crops, which represents a coverage of several million acres. Lathrope reported in 1951 that airplane fertilizing of wheat in Indiana had given good results for several years. The additional cost of airplane application as compared with the use of ground equipment was ordinarily offset by the greater yields resulting from earlier application. The equipment and operating procedures for airplane fertilizing are in general the same as for airplane seeding (see Section 11.10).

Equipment for row or band placement of fertilizers includes (a) special fertilizer drills for deep or shallow placement, (b) attachments for planters, cultivators, and various tillage implements, and (c) combination units such as the fertilizer-grain drill. In some instances, particularly where deep placement is involved, the fertilizing equipment is the permanent or basic machine, with interchangeable planting units and cultivators available as attachments.

13.7. Design Considerations. To meet the requirements of (a) uniform distribution at low as well as high rates of application, (b) positive and accurate control of delivery rates, (c) mechanical durability, and (d) convenience of operation, Cummings lists the following characteristics as being important in a fertilizer distributor:

1. Positive dispensing action with fertilizers covering a wide range of drillabilities. (Drillability is defined as the ease with which a fertilizer flows.)
2. Low amplitude of cycles or impulses of delivery.
3. Delivery rate independent of the fertilizer head (depth) and of reasonable inclinations of the distributor.
4. Accuracy and refinement of dispensing parts.
5. Rate adjustable in small increments, with a suitable reference scale provided.
6. Provision for ready calibration or determination of actual delivery rates.
7. Unit should be easy to empty and disassemble for thorough cleaning. Many fertilizing materials are corrosive and will tend to "freeze" rotating parts if even small amounts are left in the distributor and become moist or wet.
8. Use of corrosion-resistant metals where possible.

Guelle\(^{a}\) points out that the corrosive effect of fertilizing materials is one of the major problems now facing manufacturers. In addition to shortening the life of the mechanism, corrosion reduces the accuracy of distribution. Satisfactory resistance to corrosion can probably be obtained by making the working parts from stainless steel, but this is a costly solution to the problem. When metal coatings are employed, the effect of abrasion, followed by corrosive action after the coating has been worn away, is a problem.

13.8. Metering Devices for Dry Fertilizers. Many different types of dispensing devices have been developed in an attempt to

![Fig. 13.4. Star-wheel feed arrangement for a grain-drill fertilizing unit or a full-width-feed broadcaster. Shown at the right are three sizes of interchangeable wheels. (Deere & Co.)](image)

obtain a positive and uniform metering action under the wide variety of conditions encountered in distributing commercial fertilizers. Some of the principles employed in present-day distributors are described in the following paragraphs. The rotating
members of these metering devices are generally driven from a ground wheel. Row or band placement units usually have an automatic feed clutch operated in conjunction with the raising or lowering of cultivator shovels or planting units.

The star-wheel feed is used extensively on grain-drill fertilizing units (Fig. 13.4), as well as on some row-crop attachments (Fig. 13.5) and some full-width-feed broadcasters. Each wheel carries fertilizer through a gate opening into the delivery compartment. Fertilizer carried between the teeth of the feed wheel falls into the delivery tube by gravity, while material carried on top of the wheel is scraped off into the delivery opening. The feed rate is controlled by raising or lowering a gate above the wheel. Two or more speed ratios are often provided. Each wheel should be driven through a safety device such as a shear pin, to give protection if the feed becomes blocked with caked fertilizer or other solid objects.

Many row-crop fertilizing attachments utilize the revolving-bottom principle. In one arrangement (Fig. 13.6) a stationary plow scoops material out of a rotating pan immediately beneath the hopper, elevates it over the side of the pan, and diverts it into the delivery tube. The depth or thickness of the layer carried to the plow is determined by the vertical clearance between the rotating pan and the bottom edge of the hopper base (see right-hand view of Fig. 13.6). The feed rate is adjusted by raising or lowering the hopper and its base, to change this vertical clearance.

Other revolving-bottom distributors have a rotating, horizontal flat plate that is held up against the bottom of the hopper base. In some cases the material is scraped off of the revolving plate into the delivery tube by an adjustable deflector located beneath
a stationary cover plate or housing that projects inward from the periphery of the hopper base. Other units have an adjustable gate at the side, as in Fig. 13.7. In this particular unit, note the two dispensing units that permit application of two fertilizer bands from a single hopper.

Two other types of metering devices for row crops are illustrated in Figs. 13.8 and 13.9. With the auger-type unit the feed rate is controlled entirely by the speed of the screw, generally through a speed-change arrangement similar to that on a double-
run grain-drill feed (see Section 11.6). The feed rate with a belt-type dispenser is controlled with a gate above the belt (Fig. 13.9), and, in addition, several speed ratios are usually available.

Full-width-feed broadcasters often have adjustable openings spaced along the bottom of the hopper, with an agitator or rotor of some sort directly above each opening. The design of the rotor should be such as to break up lumps, promote uniform feeding, and minimize the effects of the changing fertilizer head as the hopper empties. Two types of feed rotors above adjustable
openings are illustrated in Figs. 13.10 and 13.11. Similar feed units are occasionally employed on row-crop distributors.

All the devices described above are of the bottom-delivery type. Top-delivery distributors, which are found only to a limited extent in the United States, usually have an arrangement such as a revolving hopper with ascending bottom that lifts the mass of material at a preselected rate to a stationary top deflector where it is shaved off and directed into the delivery tube. In other arrangements, used to some extent in Europe, the hopper bottom is stationary while the sides and dispensing mechanism move downward. The hopper or dispenser of a top-delivery distributor must be repositioned each time fertilizer is added.

13.9. Factors Affecting Delivery Rates. Although many of the bottom-delivery distributors have a positive action in dispensing a portion of the material, most of them depend at least partially upon gravity to supply the increased flow obtained when the delivery opening is enlarged by means of a gate. Consequently, the delivery rates from most types of distributors are materially affected by changing conditions, the extent of the effects being related to the degree of dependence upon gravity.
flow. Top-delivery distributors operate on the positive-displacement principle so that the discharge rate is independent of most of the variables affecting the bottom-delivery units. Since all types of distributors operate on a volumetric basis, the apparent specific gravity of the material affects the weight rate with any of them.

One of the important factors affecting delivery rates is the drillability of the fertilizer. The drillability or ease of flowing is affected by such factors as the hygroscopicity of the fertilizer, the relative humidity at which it is stored, the state of subdivision (size and shape of particles, presence of lumps, etc.), the apparent specific gravity, and the compaction characteristics of the material.¹⁰

In a large number of tests with a star-wheel feed unit from a grain drill, all made with the control gate raised about one-third way, Mehring and Cumings¹⁰ found a definite relation between the kinetic angle of repose of the fertilizer and the delivery rate, as indicated in Fig. 13.12. They concluded that the drillability of a fertilizer is inversely proportional to the angle of repose and that fertilizers with angles of repose greater than about 55° are usually undrillable with present equipment (with the exception

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Fig. 13.12. Relation of delivery rate to angle of repose of the fertilizer, for two types of distributors. (Armon L. Mehring and Glenn A. Cumings. USDA Tech. Bull. 182.)
of top-delivery distributors). They suggest that fertilizers having about a 40° angle of repose and composed of 20-mesh-size rounded grains with rough surfaces are best adapted to present types of equipment.

Comparison of the two curves in Fig. 13.12 shows the advantage of having a metering unit with positive volumetric displacement (such as the top-delivery feed) rather than one depending partially upon gravity flow. Closing the gate of the star-wheel feed would tend to reduce the slope of the curve because the positive-action portion would become relatively more important. With materials that do not flow by gravity, opening the gate does not increase the rate much above that due to positive action.\textsuperscript{10}

In tests with metering devices similar to those illustrated in Figs. 13.4 through 13.9, Mehring and Cumings found that the depth of fertilizer in the hopper (using a free-flowing material with a 35° angle of repose) had little effect on the delivery rate of the belt-type and auger-type distributors (depths from 2 to 14 in.). With the star-wheel and revolving-bottom types, there was little variation at depths greater than 3 or 4 in. but the rates were reduced by 8 to 14 per cent at a depth of 2 in. and even more at a depth of 1 in. The top-delivery dispenser had a virtually constant weight rate except for a slight increase due to the effect of compaction of the material in the bottom portion of the hopper.

With a free-flowing fertilizer (35° angle of repose), tilting the star-wheel or revolving-bottom hoppers 10° toward the discharge opening increased the delivery rate by 11 to 21 per cent (because of the increased effect of gravity). A 10° tilt in the opposite direction decreased the rate by 11 to 15 per cent as compared with the level rate. Tilting had less effect on the belt-type and auger-type units and had no effect on the positive-acting, top-delivery unit.

13.10. Factors Affecting Uniformity of Distribution. Uniform distribution along the row is essential for fertilizers to be most effective but is difficult to obtain, especially for low rates of application.\textsuperscript{10} Cycles and impulses of delivery due to mechanical imperfections or to the design of the dispenser are the principal causes of irregular distribution of free-flowing fertilizers.\textsuperscript{10} A revolving bottom that is not quite perpendicular to the axis of
rotation is an example of mechanical imperfection. The fingers of a star-wheel feed give definite impulses of delivery, as indicated in Fig. 13.13. Likewise, the auger feed tested by Mehring and Cumings had a rather pronounced cycle corresponding to one revolution of the screw, when discharging a free-flowing material. With either of these metering devices, some 1-ft increments received about four times as much fertilizer as did other increments within the cycle.

![Diagram](image)

Fig. 13.13. Uniformity of distribution in relation to distance traveled, for two types of distributors, when metering a fertilizer that had a 45° angle of repose. Each point represents the total delivery in a 1-ft increment of travel. (Arnon L. Mehring and Glenn A. Cumings. USDA Tech. Bull. 182.)

Figure 13.14 shows the average per cent deviation from the mean delivery rate, based on 1-ft increments of travel, for several types of distributors and with fertilizers having various degrees of drillability. It should be remembered that the actual extreme deviations are considerably greater than the indicated average deviations. Note that, with the star-wheel and auger-type feeds, the deviations due to the inherent cyclic impulses predominated regardless of the condition of the fertilizer and were relatively large. The other types of units had considerably smaller deviations with free-flowing materials.

Fertilizers of low drillability (large angle of repose) are delivered unevenly regardless of the type of dispenser, as indicated by the 54° points in Fig. 13.14. Minimum cohesion between the fertilizer particles, or some mechanical means of overcoming the
effect of cohesion, is necessary for uniform distribution. If some mechanical means is provided for breaking down the fertilizer into a finely divided state at the point of delivery, greater uniformity can be obtained and the range of drillability at which fertilizers can be uniformly distributed is widened.

Fig. 13.14. Relation between the angle of repose of fertilizers and the average deviation from the mean delivery rate, for several types of distributors. The 35° fertilizer flowed like coarse, dry sand. The 54° material was a damp-appearing powder that partially retained its form when squeezed in the hand and did not flow much by gravity unless continually agitated. (Data from Arnon L. Mehring and Glenn A. Cumings. *USDA Tech. Bull.* 182.)

13.11. Agitators. Most fertilizer distributors are equipped with rotating agitators in the hoppers (Figs. 13.4 to 13.10) or at least have agitators available. Mehring and Cumings found that an agitator influences the delivery rate only with fertilizers that tend to cake or bridge. The increase of delivery rate obtained as a result of the action of an agitator is a measure of its effectiveness in breaking up lumps and preventing caking or bridging. Although an agitator makes possible the dispensing of fertilizers that could not otherwise be handled, it does not completely com-
pensate for the decrease in rate caused by low drillability. If the drillability is extremely low, the agitator may merely revolve within the mass without any beneficial effect.

**INJECTION OF LIQUIDS AND GASES INTO THE SOIL**

Some of the fertilizer materials that are liquids or solutions at atmospheric pressure and normal temperatures can be applied directly to the soil surface, as on grass and other sod crops. But for row crops proper placement in the soil, either behind cultivator shovels or with separate openers (Section 13.5), is more effective. With aqua ammonia or anhydrous ammonia, it is essential that the material be released in narrow furrows at a depth of at least 4 to 6 in. and covered immediately to prevent the escape of ammonia.

Equipment for placing liquids and gases in the soil consists essentially of a supply tank, a metering device, a narrow furrow opener with a tube attached to the rear to release the liquid or gas at the bottom of the furrow, and sometimes a separate covering device to seal the furrow. Both mounted and trailed types of equipment are available. Tank capacities generally range from 60 to 100 gal on the smaller units, but some of the large, custom-operated ammonia injectors in the West have tanks that hold over 1000 gal (on four-wheel trailed units).

**13.12. Metering Devices.** For liquids and solutions at atmospheric pressure, gravity flow through fixed openings or adjustable valves is the simplest arrangement, but the head changes and small orifices are likely to become clogged. With anhydrous ammonia, the vapor pressure in the tank (125 to 250 psi, depending upon the liquid temperature) can be used to discharge the liquid through a pressure-regulating valve and fixed orifices. Either of these arrangements requires a constant forward speed for uniform distribution, since the discharge rate is independent of the forward speed.

If metering is done by a positive-displacement pump driven from a ground wheel, the discharge rate is proportional to the forward speed and is independent of head or pressure. One such unit, designed for anhydrous ammonia but also suitable for other liquids, employs a piston pump that is driven from a ground
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wheel. The discharge rate is controlled by a variable-stroke arrangement. Anhydrous ammonia flows from the tank through a small heat exchanger, in which it is cooled to condense any gas bubbles, through the pump, where it is further cooled by the partial vaporization accompanying a reduction in pressure, back through the heat exchanger, into a dividing manifold, and finally into the various delivery tubes.

![Diagram of simple metering pump](image)

**Fig. 13.15.** A simple metering pump for liquids at relatively low pressures.

A rather simple metering pump for materials that are liquid at atmospheric pressure was developed at the Tennessee Agricultural Experiment Station. The basic arrangement is shown in Fig. 13.15. At low pressures (i.e., gravity head) it tends to give positive displacement, and, since it is ground driven, the volume from each tube is proportional to the forward travel. The application rate is a function of the speed ratio, the reel diameter, and the inside diameter of the rubber tube.

**13.13. Applicators.** The applicators for injecting liquids and gases into the soil consist of narrow furrow openers, each with a delivery tube attached to the rear side and extending nearly to the bottom of the furrow (Fig. 13.16). The ammonia is discharged from holes in the sides of the tube near the lower end. When the narrow furrow openers are operated at a depth of at least 5 to 6 in. in loose, friable soils, effective sealing in of am-
monia may be obtained without the use of covering devices. But, under some conditions, devices such as small press wheels, covering shoes, scrapers or disk hillers should follow immediately behind the applicators.

Since the release and subsequent vaporization of anhydrous ammonia is actually a refrigeration process, the applicators must be carefully designed so that the furrow opener and standard will not be chilled sufficiently to cause a build-up of ice and soil on them. To promote sealing of the furrows, the applicator shanks should shed trash freely and the soil should flow smoothly around them. Spring-trip or coil-spring standards are often employed where there is danger of encountering obstructions in the soil.

REFERENCES

APPLICATION OF FERTILIZERS

presented at Third Annual Short Course on Use of Aerial Equipment in Agriculture, Purdue University, 1951.


PROBLEMS

13.1. A side-dressing fertilizer unit is to place two bands per row on a crop with a 40-in. row spacing. It is desired to apply 500 lb per acre of a fertilizer having an apparent specific gravity of 0.85. If the distributor is calibrated by driving the machine forward a distance of 100 ft, what weight of material should be collected from each delivery tube when the distributor is properly adjusted?

13.2. A distributor for liquid fertilizer has gravity feed through fixed orifices. The tank is 30 in. deep. The bottom of the tank is 2 ft above the ground, and the orifices are 6 in. below ground level.

(a) What is the ratio between the flow rates with the tank full and with it practically empty?

(b) What changes in the tank or mounting would you suggest to reduce the variation in rate?
CHAPTER 14

Hay Harvesting: Mowing, Raking, and Baling

14.1. Introduction. Hay is grown on more than half of all the farms in the United States, with the acreage averaging about 20 per cent of the total harvested crop land. Hay production in the United States amounts to about 100 million tons annually. In addition, forage crops are harvested extensively for silage. Minor methods of using forage crops include direct feeding of freshly harvested green hay and the production of meal (primarily alfalfa) from either artificially dehydrated or sun-cured hay.

Forage handling is complicated by the nature of the product. Hay is a crop of great bulk and weight, containing 70 to 80 per cent water (wet basis) when first harvested. For storage it must be dried, either naturally or artificially, to a safe moisture content. Limits of 20 to 25 per cent are considered safe, although long loose hay or extremely loose bales can tolerate slightly higher moisture contents without serious damage. The relatively low cash value per acre for hay crops limits the economic feasibility of mechanization for small acreages. In addition, hay is frequently grown on rolling land and steep slopes or under other conditions unfavorable to mechanization.

14.2. Quality of Product in Relation to Harvesting Methods. In general, all methods of harvesting and storing forage crops involve some reduction in quality. When hay is dry, the leaves, which are the most nutritious part of the plants, are easily lost in handling. Prolonged exposure to sun, dew, and rain results in the loss of valuable nutrients, particularly protein and carotene (a source of vitamin A), and also causes bleaching or loss of color. Thus, any harvesting procedure that reduces the time between
cutting and storing tends to minimize losses in quality, as well as reducing the weather hazard. Partial curing in the field, followed by artificial drying in the barn or stack with forced circulation of either heated or unheated air, is one method of reducing the time the hay is in the field.

In comparative tests of harvesting methods with alfalfa, Hodgson and his associates found that protein losses during harvesting and storage, in terms of the protein content of the original crop, amounted to 32 percent for field-cured hay, 26 percent for barn-cured hay, and 14 percent for silage. Many attempts have been made to measure the relative effects of various types of machines on the quality of hay produced under a given harvesting method, but most of these tests have shown nothing conclusive. Weather is still the major factor involved.

14.3 Hay Harvesting Methods. Many different combinations of operations are employed in harvesting hay, some of them requiring considerable amounts of hard, manual labor. All methods, with the exception of field chopping of standing crops for direct feeding or dehydration, include mowing and raking. Subsequent operations involve handling the hay in the long loose form, baling it, or chopping it. The usual practice is to windrow the hay within a few hours after cutting, because the leaves dry considerably faster in the swath than do the stems and would shatter badly by the time the stems were down to a safe moisture content.

Hauling and storing hay in the long loose form involves the least investment in equipment and is the most economical method for small tonnages, but it may involve a considerable amount of hand labor. The hay is commonly hauled with wagons or trailers, but it is often moved short distances with sweep rakes. Loading and stacking may be done entirely by hand or with mechanical aids. In some areas the hay is stacked in the field and fed directly from the stack.

Most baling is now done in the field with hand-tie or automatic-tying machines of the windrow-pickup type. Although field baling in itself has been developed into a highly efficient operation in regard to labor requirements, the loading and handling of bales still requires considerable hand labor. The bales are usually allowed to drop from the baler onto the ground and are picked up later with mechanical bale loaders or by hand. Mechanical
bale loaders eliminate most of the heavy lifting in loading, but the bales must still be positioned by hand on the truck or trailer.

Another method of loading bales is with a direct bale chute, by means of which the baler forces the bales up onto a trailer (see Fig. 1.2). This method of loading bales requires the least labor, but it involves heavy work. In addition, tests have shown that towing a trailer reduces the capacity of a baler by an average of about 15 per cent.*

Unloading and stacking of bales may be done entirely by hand or with mechanical aids such as inclined elevators and forks or slings operated from overhead barn tracks. Pallet handling is practiced to a limited extent.

The field-chopping method of harvesting hay requires less labor than any other method but involves a large investment in equipment. In this method the hay is picked up from the windrow with a field chopper. The chopped hay is blown directly into vans or special trailers, from which it can be unloaded mechanically and put into storage without hand lifting. Stationary chopping requires more labor than field chopping but less investment in equipment. In this method, long loose hay is hauled to the chopper with sweep rakes or wagons (loaded mechanically), and the chopper blows the product directly into the barn. Forage choppers and blowers are discussed in Chapter 15.

MOWING

14.4. Principles of Cutting. In order to have cutting take place, a system of forces must act upon the material in such a manner as to cause it to fail in shear. This shear failure is almost invariably accompanied by some deformation in bending and compression, which increases the amount of work required for the cutting operation. A common way to apply the cutting forces is by means of two opposed shearing elements which meet and pass each other with little or no clearance between them. Either one or both of the elements may be moving, and the motion may be linear with uniform velocity, reciprocating, or rotary. A single cutting element, either moving or stationary (with respect to the machine), is sufficient if the nature of the operation permits a fixed surface, such as the ground, to act as one of the shearing elements (as with weed knives on cultivators). If the material
being cut is adequately supported and is relatively strong in bending (sugar beets, for example), the material itself may transmit the force required to oppose a single cutting element.

An impact cutter has a single, high-speed cutting element and relies primarily upon the inertia of the material being cut to furnish the opposing force required for shear. An excellent example is the rotary cutter that has blades in a horizontal plane and is used for clipping pastures, weeds, or other debris, above the surface of the ground. These devices often have blade tip diameters in the order of 5 ft, with tip speeds of 10,000 to 12,000 fpm. This particular type of machine is most useful where it is permissible or desirable to have the material cut into relatively short lengths, as with stalk cutters (see Section 9.15).

A considerable amount of experimental work with rotary or impact-type lawn mowers has been done by engineers in the Lawn Mower Division of Reo Motors, Inc. With blade diameters ranging from 10 to 20 in., they found* that a peripheral speed of about 10,000 fpm was adequate for clean cutting of fine fescue grasses at a blade height that left stubble lengths ranging from 1 to 1\(\frac{1}{2}\) in. Under open-field conditions, where various types of grasses were encountered, a speed of about 8000 fpm was found to be adequate for cutting at a height of approximately 3 in. Operating at speeds greater than needed for clean cutting resulted in increased power requirements with no definite advantages observed in regard to cutting ability.

14.5. Conventional Mower Cutter Bar. The construction and mounting of a typical mower cutter bar are illustrated in Fig. 14.1. Not shown are the outer divider and shoe assembly and the grass board. The function of the grass board, which is attached to the rear of the outer divider, is to clear the cut material from a narrow strip next to the standing crop and thus provide a place for the inner shoe to operate on the next trip. As indicated in Fig. 14.1, the knife of the conventional mower is driven through a pitman attached to the knife head by means of a ball-and-socket joint. The knife stroke and guard spacing are ordinarily 3 in., and crank speeds on tractor mowers without a reciprocating counterbalance are in the order of 800 to 1000 rpm (1600 to 2000 cutting strokes per minute).

*Personal correspondence from Paul Rosenberg, Chief Engineer, Lawn Mower Division, Reo Motors, Inc., dated January 6, 1953.
The height of cut is ordinarily gaged by adjustable shoes at the ends of the cutter bar. The adjustable support spring shown in Fig. 14.1 acts through the lift linkage to carry most of the weight of the cutter bar so that it "floats" along the ground. The lift linkage (Fig. 14.1) can be adjusted to change the relative amounts of weight on the inner and outer shoes. These weights should be just enough to prevent bouncing of the cutter bar, the optimum amounts being influenced by the roughness of the field and the forward speed. Typical values are 80 to 100 lb on the inner shoe and 20 to 30 lb on the outer shoe.

Figure 14.2 shows a cross-section of a cutting unit. The ledger plates are ordinarily serrated on the under side and are occasionally replaced but never sharpened. The cutting edges of the knife sections may be either smooth or underserrated, both types being resharpened at frequent intervals by grinding the beveled top surfaces. Underserrated knives are good for coarse-stemmed crops but give trouble in grasses because the fine stems tend to wedge under the blade. Knife clips and wearing plates are mounted together and are generally spaced three or four guards apart (Fig. 14.1). The wearing plates provide vertical support.
for the rear of the knife sections and also absorb the rearward thrust of the knife. They have a fore-and-aft adjustment but must be replaced when the top surface wears down sufficiently to tip the knife and cause an appreciable vertical clearance between the ledger plates and the points of the knife sections.

14.6. Knife Clearances and Cutting Velocities. Because many of the materials commonly cut with a mower are weak in bending, it is important to hold the knife sections down close to the ledger plates. This is done by bending the knife clips (Fig. 14.2) down until there is only enough clearance to prevent binding. If the clearance is allowed to increase and approach the thickness of the material being cut, deformation in bending produces a wedging effect that increases power requirements and may result in failure to cut some stalks.

This situation is less critical if the cutting unit is designed so that all cutting occurs during portions of the stroke where knife velocities are relatively high. A combination of impact and shear can then be utilized, rather than depending upon shear alone. Since the peak knife velocity with a 3-in. stroke at 1000 rpm is less than 800 fpm (as compared with 10,000 fpm for a rotary cutter), there is probably very little cutting by impact of the knife sections alone.

14.7. Register and Alignment. Two adjustments that are important to proper functioning of a mower are known as knife register and cutter-bar alignment. A knife is in proper register when the knife sections are centered in the guards at each end of the stroke. Adjustment is commonly made by moving the entire cutter bar in or out with respect to the pitman crankshaft,
although some mowers have provision for changing the pitman length instead.

When a cutter bar is in proper alignment, the pitman and the knife will be in line (in plan view) when the mower is operating in the field. To allow for the rearward deflection of the outer end of the cutter bar during operation, it is customary to adjust the mower so that, when not operating, the outer end of the cutter bar has a lead of about \( \frac{3}{4} \) in. per ft of bar. Lead is measured by stretching a string parallel to the vertical plane of the pitman and determining the difference in horizontal fore-and-aft distances from the string to the knife back at the inner and outer ends. One method of adjusting lead is by rotation of an eccentric bushing on one of the hinge pins (Fig. 14.1).

If the crankshaft is mounted directly on the inner end of the cutter bar (as in Figs. 14.6 and 14.7), the problems of maintaining alignment and register are virtually eliminated.

### 14.8. Tractor-Mounted Mowers

Common arrangements for mowers on tractors include side mounting between the front and rear tractor wheels, rear mounting, and semimounting at the rear of the tractor. Side mounting gives better visibility of the work and direct response to steering. The rear-mounted (or semimounted) mowers are generally easier to attach or remove and are a little better for making square corners when cutting around a field.

Cutter bars on tractor mowers are usually 6 or 7 ft long, whereas most horse-drawn mowers had 5-ft bars. Elles points out that, for high-speed operation of tractor mowers, the present cutter bar is too flexible and is subjected to excessive bending stresses that may cause fatigue failures. He suggests the adoption of a new standard cross-section that would be thicker than the present main bar (Fig. 14.2) and have a section modulus about 50 per cent greater.

Since the inertia of all but the smallest tractors is relatively high, it is important that the cutter bar be provided with a safety break-away coupling that will allow the bar to swing back if it strikes an obstruction. This safety device becomes increasingly important for the larger tractors and as forward speeds are increased. A safety device in the pto drive is also important, because of the large excess of power generally available.
14.9. Cutting Pattern for a Conventional Mower. Figure 14.3 shows the cutting pattern, based upon a theoretical analysis,\textsuperscript{14} for a conventional mower when operated with a feed rate (forward travel per cut) of 3 in. Sinusoidal motion of the knife is assumed, and the dimensions shown are typical. The shaded portion represents the land area from which stalks are cut during the stroke under consideration. All stalks originating in this shaded area are cut along line $EF$. As the mower moves forward and the knife section moves from left to right, cutting begins when the front center of the section is at point 1, such that the rear corners
of the cutting portions of the knife and the ledger plate just come together (at $E$). Cutting finishes at $F$ when the knife center is at point 2.

According to this analysis, all stalks originating along a typical path $ZV$ are deflected by the guards and ledger plates to point $V$ as the mower moves forward. The knife section deflects all stalks originating along $JLUV$ to point $V$, where they are cut by point $U$ of the knife. The slope of $ZV$ is the sum of the angle of the ledger-plate edge with respect to the direction of travel and the angle whose tangent is the coefficient of friction between the stalks and the ledger plate. Similarly, the slope of $JL$ represents the condition under which stalk slippage along the knife edge is impending or occurring. Between $L$ and $V$, the slope of the path of point $U$ on the knife is less than the slope of $JL$, and no slipping occurs. The coefficients of friction assumed in Fig. 14.3 are (a) between stalks and serrated ledger plates, $\tan 20^\circ = 0.364$, and (b) between stalks and smooth knife edge, $\tan 17^\circ = 0.306$. These values are based upon friction-angle measurements reported by Bosoi (reproduced in reference 14).

If the included angle between the two cutting edges is too great, stalks will tend to slip forward rather than being cut. The maximum permissible angle depends upon the coefficients of friction between the stalk and the cutting edges. The primary function of serrations on the cutting edges (discussed in Section 14.5) is to increase the coefficient of friction.

It will be noted in Fig. 14.3 that for a feed rate of 3 in. there is a sizable shaded area behind stalk paths $HE$ and $EK$, from which stalks must be crowded forward in order to be cut by the rear portion of the knife. In this example, which would represent a forward speed of 5 mph if the crank speed were 880 rpm, the area behind $HEK$ is equal to 25 per cent of the total area cut per stroke. Such a condition is undesirable because of the bunching effect at the rear of the knife and the resulting large cutting force required at the start of the cutting and because of the excessive stalk deflection involved. If the feed rate for the mower represented in Fig. 14.3 is reduced to 1.55 in., points $Q$ and $E$ coincide and the area behind $HEK$ becomes zero. The analysis indicates that with a 2-in. feed rate this rear area would be only 9 per cent of the total.
14.10. Stalk Deflection. In Fig. 14.3, the theoretical maximum deflection of the rear stalks is from $M$ (point of stalk origin) to $E$ (point of cutting), a distance of 2.4 in. in this example. The maximum deflection of side stalks is from $B$ to $F$, a distance of 3.2 in. Corresponding deflections for a feed rate of 1½ in. instead of 3 in. were found to be 0.8 and 2.8 in. Thus it is seen that deflection of side stalks is rather large with the conventional mower, even at low feed rates, whereas the deflection of rear stalks becomes large as the feed rate is increased.

Excessive stalk deflection, particularly when cutting close to the ground, results in stubble of uneven length. It also increases the tendency for the stalks to slip forward out of the cutting unit and thus be cut higher or missed entirely. For best performance when cutting within 2 or 3 in. of the ground, the maximum deflection of rear stalks (from $M$ to $E$) should not exceed the amount indicated in Fig. 14.3 and should preferably be less. This represents a maximum feed rate of 2½ to 3 in. for the conventional cutter bar. Increasing the effective length of the cutting elements would move point $E$ to the rear (Fig. 14.3) and thus permit greater feed rates. Changing the widths of the knife sections or ledger plates or changing the length of stroke also affects the maximum deflection of rear stalks.

Side deflection can be minimized by having a short stroke or by an arrangement to obtain two cuts per stroke. Thus, doubling the number of guards on a conventional mower (as is occasionally done for certain crops) reduces the side deflection. This change also increases the permissible forward speed for a given feed rate and crank rpm, since the number of cuts per stroke is doubled, but the reduced width of opening between guards may cause feeding troubles. If the cutter bar is equipped with two reciprocating knives, each having a stroke equal to half the knife-section spacing (1½-in. stroke, for example), the side deflection is reduced and yet there is ample width of feed opening when the knives are at the ends of their strokes. Two reciprocating knives that have longer strokes (such as 3 in.) and cross over to give two cuts per stroke would accomplish the same results and require only half as great a crank rpm for a given forward speed.

14.11. Reciprocating Unbalance in a Mower. In designing a mower with a reciprocating cutting unit that will do a good job of cutting at high forward speeds, one of the serious problems is
vibration caused by unbalanced forces of the reciprocating parts. Figure 14.4 shows the force relations between a reciprocating slider, the pitman, and the crankpin. In analyzing the unbalanced forces, it is convenient and sufficiently accurate in this case to consider the pitman weight as being divided into components $W_1$ and $W_2$ concentrated at the two ends, $W_1$ being sub-

![Diagram](image)

Fig. 14.4. Forces due to the inertia of the reciprocating parts in a crank and offset-slider arrangement.

jected to rotary motion and $W_2$ having reciprocating motion. $W_s$ is the weight of the slider (knife), and $W_c$ is the weight of the crankpin. Uniform angular velocity of the crank is assumed. If the ratio $\frac{R}{\sqrt{L^2 - S^2}}$ is less than 0.25, the inertia force of the reciprocating parts can be represented by the following approximate relation, using any consistent set of units:

$$F_h = \frac{W_s + W_2}{g} R \omega^2 \left( \cos \phi_c + \frac{KS}{R} \sin \phi_c - K \cos 2\phi_c \right) \ (14.1)$$

where $F_h =$ inertia force of the slider and the $W_2$ portion of the pitman weight.

- $g =$ acceleration of gravity.
- $\omega =$ angular velocity of the crank, in radians per second.
- $\phi_c =$ angle of crank rotation beyond reference position indicated in Fig. 14.4.
- $L =$ length of pitman.
- $R =$ radius of rotation of crankpin.
- $S =$ height of crankshaft centerline above the plane of the connection between the slider and the pitman.
Because of the angle between the pitman and the slide, a periodic, alternating, vertical reaction $F_v$ is introduced at the knife head, which results in vibration and also causes the pitman force $F_p$ to be greater than $F_h$. At the crankpin, $F_p$ is divided into radial and tangential components (Fig. 14.4). The tangential component is opposed by the flywheel and power source, but $F_r$ is opposed only by the centrifugal force of the counterweight commonly provided on the flywheel. From Fig. 14.4, it is seen that the reciprocating unbalance at the crankpin is

$$F_r = F_p \cos (\phi_c - \phi_p) = F_h \frac{\cos (\phi_c - \phi_p)}{\cos \phi_p} \quad (14.2)$$

where $\phi_p$ = angle of the pitman above horizontal.

In addition to $F_r$, there is at the crankpin a rotating weight (crankpin plus remaining component of pitman weight) which gives an unbalanced or centrifugal force of

$$F_c = \frac{W_c + W_1}{g} R \omega^2 \quad (14.3)$$

Since $F_c$ is independent of the crank angle $\phi_c$, it can be balanced for all crank positions by adding a balance weight opposite the crankpin as may be required to produce the same centrifugal force.

The reciprocating unbalance, however, as determined from equations 14.1 and 14.2, is a function of $\phi_c$, varying as indicated in Fig. 14.5. It cannot be balanced by a rotating mass because the centrifugal force of the mass would be constant at all crank angles. The usual compromise is to provide a rotating counterweight opposite the crankpin, to introduce a centrifugal force equal to $F_c + \frac{1}{2} (F_r \ \text{max})$. The result of this half-balancing, which only minimizes the unbalance, is indicated by the lower solid curve in Fig. 14.5.

Improved balancing of reciprocating parts can be obtained by the addition of a second reciprocating mass that moves in direct opposition to the first mass. However, unless the opposed forces
are collinear, the resulting couple will set up a new vibratory effect.

One way to improve balancing is by having two opposed, reciprocating knives. If only one cut per stroke is obtained, this arrangement requires only half the stroke of a comparable single-knife cutter bar, thus permitting higher crank speeds for the same inertia forces. Reciprocating balance in itself allows higher crank speeds, thus further increasing the maximum forward speed at which good cutting can be obtained.

14.12. Recent Developments in Mowers. The conventional, pitman-type cutter bar discussed in the preceding sections is basically the same, both in principle of operation and in construction, as its horse-drawn predecessor. It requires a great deal of maintenance and repair and is not well adapted to high-speed tractor operation. In the last few years, however, farmers' demands for improved, high-speed mowers have resulted in a considerable amount of development work and the commercial appearance of several improved models.

One of the new models has the conventional type of cutter bar
(except that the stroke is only $2\frac{1}{2}$ in. for a guard spacing of 3 in.) but also has a reciprocating counterbalance weight to obtain dynamic balancing. The entire drive unit and the counterweight are mounted on the inner shoe (Fig. 14.6), and the cutter-bar hinge axis is in line with the crankshaft so that operation is not affected by raising or lowering the bar. Specially designed, sealed antifriction bearings are employed throughout. The ball-bearing knife-head joint is in line with the knife, to eliminate the bending moments characteristic of the conventional ball-and-socket type of connection that has its center about $1\frac{3}{4}$ in. above the knife. The knife spring shown in Fig. 14.6 is attached to the knife about 1 ft from the inner end. Its purpose is to absorb the slight variations in height of the knife head with respect to the knife.

Another type of dynamically balanced mower is illustrated in Fig. 14.7. This machine has two reciprocating knives, each with a stroke of 1 in. and with knife sections 2 in. apart. The knives are driven at 1800 to 2000 rpm by a hydraulic motor mounted on the inner shoe. Scotch yokes, attached directly to the knives, are driven by eccentrics on the drive shaft, the entire assembly operating in an oil bath. The absence of guards greatly reduces the tendency to plug in tangled or matted crops but may present problems under stony conditions.

![Fig. 14.6. Drive unit for dynamically balanced mower with reciprocating counterweight. (L. E. Elies. *Agr. Eng.*, March, 1954.)](image)

Draft requirements for horse-drawn mowers are given in Appendix C as 60 to 100 lb per ft of cutter bar. Power-take-off power requirements for tractor mowers are commonly considered to be \( \frac{1}{4} \) to \( \frac{3}{4} \) hp per ft of bar, although Elles measured values considerably lower than this. With a 7-ft mounted mower having a conventional type of cutter bar and a newly developed drive system (Fig. 14.6), he made limited tests in moderately heavy, mixed hay (1\( \frac{1}{4} \) to 1\( \frac{3}{4} \) tons per acre) and obtained the results given in the following table.

<table>
<thead>
<tr>
<th>Feed Rate, in. per stroke</th>
<th>Average pto Horsepower</th>
<th>Average Peak pto Horsepower</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inertia and friction load (no cutting)</td>
<td>1.70</td>
<td>6.18</td>
</tr>
<tr>
<td>Mowing at 4.9 mph</td>
<td>2.68</td>
<td>2.55</td>
</tr>
<tr>
<td>Increase due to cutting</td>
<td>0.85</td>
<td>0.82</td>
</tr>
</tbody>
</table>

These results, obtained with a 7-ft mower at a crank speed of 942 rpm, indicate an average input of only a little over \( \frac{1}{4} \) hp per ft of cutter bar but peak values (one peak per stroke) of about 1 \( \frac{1}{4} \) hp per ft of bar. It is not known how well these power requirements would apply to other types of mowers or to other cutting conditions. Elles found that for satisfactory life the drive system (V-belt) had to be designed on the basis of the cyclic peak requirements.

HAY CRUSHERS

One method of reducing field curing time for hay is to crush or crack the stems of the freshly cut hay so that they will dry at a rate more nearly equal to that of the leaves. Although the general use of field hay crushers is a relatively recent development, tests...
on experimental mower-crushers were reported in 1931 and 1933. The present-day crusher consists of a pair of steel rollers held together under pressure by adjustable springs and driven by the tractor pto. A pickup unit lifts the swath and feeds it between the rolls. The crushed hay is dropped back onto the stubble, still in the swath. The crushers of some mower-crusher combinations follow directly behind the tractor and crush the swath that was mowed on the previous round, whereas other combinations have the crusher directly behind the cutter bar. Separate crushers are also available, but they require an extra time over the field.

Comparative tests indicate that crushed hay generally dries to a safe storage moisture content in one-third to two-thirds of the time required for uncrushed hay. Although this reduction in time is seldom sufficient to allow cutting and storing in the same day, it does reduce the weather hazard considerably and the faster curing should result in less loss of nutrients. Crushing may be done in conjunction with any of the three general methods of handling hay (baling, chopping, or leaving the hay in the long loose form).

Raking

Most methods of harvesting hay, as well as some seed-harvesting and green-crop harvesting operations, involve windrowing. Two general types of rakes are in common use for windrowing hay. Dump rakes produce windrows that are not continuous but are characteristically larger than those formed by side-delivery rakes. The larger windrows are desirable for either hand loading of long loose hay or moving it with sweep rakes. Side-delivery rakes were introduced just before the turn of the century, for use with mechanical hay loaders. Interest in this type of rake was greatly increased by the more recent advent of pickup field balers, field choppers, and pickup combines. Side-delivery rakes produce uniform windrows that are continuous and thus well suited to pickup machines. They handle the hay more gently than dump rakes, and the more recent models are better suited to high-speed operation.

14.14. Types of Side-Delivery Rakes. Since about 1945, side-delivery rakes have undergone considerable development, and a number of new types have appeared. The rakes now avail-
able may be classified as (a) cylindrical-reel, (b) oblique reel-head, (c) finger-wheel, and (d) drag-type. The general arrangements for the first three types are indicated to some extent in Fig. 14.13. In the cylindrical-reel type, the teeth rotate in planes perpendicular to the axis of the reel, maintaining parallel positions throughout the circle. One method of obtaining the parallel relationship is illustrated in Fig. 14.8. Rotating the eccentric about the reel axis changes the pitch of the teeth.

The oblique reel-head rake, which originated in England, was further developed and first manufactured in the United States in 1948. Its unique feature is that the reel heads are set at an acute angle from the reel axis but in parallel planes, as indicated in Fig. 14.13. The tooth bars are shaped so that the axes of the tooth-bar bearings and the main bearings on the reel heads are perpendicular to the planes of the reel heads (Fig. 14.9). This arrangement automatically maintains the teeth in parallel positions (usually about vertical) as the reel rotates, and the path of rotation of any tooth is in a plane parallel to the reel-head planes. Thus, the horizontal movement of the teeth with respect to the rake can be at an angle of as much as 90 to 100° from the direction of forward motion without requiring an unreasonably long reel to cover the required width of swath. Tooth pitch can be changed only by rotating the reel frame about the reel axis.

Most cylindrical-reel rakes have four tooth bars, whereas oblique reel-head rakes are available with four, five, or six bars. The reels of both types of rakes are power driven, either from ground wheels or from the tractor pto. Trailed rakes are usually driven by the ground wheels; this arrangement provides the desirable feature of maintaining a constant ratio between reel po-
ripheral speed and forward speed. A mounted, pto-driven rake can be built with less weight and at a lower cost than a trailed rake, but the speed ratio is changed when the tractor is operated in different transmission gears.

The finger-wheel rake was first manufactured in 1946. It consists of a series of individually floating wheels set at an angle to the direction of forward motion and overlapping each other as indicated in Fig. 14.13. The floating feature allows the rake to adjust itself to the contour of surface irregularities such as terrace channels or irrigation borders. Each wheel has raking teeth on its periphery and either rotates because of contact between the teeth and the ground or is power driven at a comparable speed. Since each wheel is set at an angle to the direction of travel, there is a velocity component perpendicular to the plane of the wheel in addition to the component causing rotation. The perpendicular component results in a dragging action of the teeth, approximately parallel to the axle. Although contact with the ground results in a clean job of raking, it tends to create a dust cloud in some fields and may cause some trash and debris.
to be put into the windrow. Under windy conditions, disk shields on the wheels are desirable to keep hay out of the spokes.

Drag-type side-delivery rakes have a series of dragging bars or teeth with the rear portions curved upward and arranged to provide a raking surface at an acute angle with the direction of travel, somewhat similar to an angled scraper blade. The upper ends of the bars are usually curved towards the discharge side of the rake to introduce somewhat of a rolling action. The windrower attachment for a mower cutter bar is an example, and at least one implement that operates on the drag or crowder principle has been developed for raking from the swath.

14.15. Desirable Raking Characteristics. Among the factors to be considered in evaluating or comparing rake performances are the following:

1. Amount of leaf loss due to shattering.
2. Amount of hay missed.
3. Amount of trash, dirt, stones, and other debris put into the windrows.
4. Uniformity and continuity of windrow.

Leaf loss is one of the most important considerations, particularly if the hay is a little too dry when raked. Shattering of seed crops such as beans or alfalfa would represent a similar type of loss. The amount of leaf loss caused by a rake is influenced by such factors as the distance the hay is moved from the swath into the windrow, the average hay velocity, and the accelerating and decelerating action or impact of the rake teeth upon the hay as it is being moved. Although the various types of rakes can be compared analytically in regard to these factors, the relative importance of the factors in relation to leaf loss can best be determined experimentally. This is not a simple test, however, because of the problems involved in controlling the variables. At the present time there are very few reliable data available on the subject.

14.16. Analysis of Raking Action for Reel-Type Rakes. In analyzing and discussing the various types of rakes, the following general symbols and terms will be used (others will be identified as needed):

\[ V_p = \text{peripheral velocity of teeth.} \]
\[ V_{tr} = \text{reel component, which is the average horizontal component} \]
of tooth velocity with respect to the rake, during the angle of rotation in which the teeth are in contact with the hay.

\( V_f \) = forward velocity of rake.

\( V_t \) = resultant tooth velocity with respect to the ground, which is the vector sum of \( V_f \) and \( V_{tr} \).

\( V_h \) = average hay velocity with respect to the ground, as the hay is moved from the swath into the windrow along the hay path.

\( V_{hr} \) = average hay velocity with respect to the rake as the hay is moved along the raking front.

\( \gamma \) = angle between the raking front and the line of forward motion. (With a reel-type rake, the raking front is parallel to the centerline of the reel.)

\( \theta_{tr} \) = reel-component angle, which is the angle between \( V_{tr} \) and the line of forward motion.

\( \theta_{t} \) = resultant tooth-path angle, which is the angle between \( V_t \) and the line of forward motion.

\( \beta \) = angle between tooth bars on reel-type rakes.

\( R_{pf} \) = ratio of tooth peripheral velocity to forward velocity = \( V_p/V_f \).

Consider the action of the teeth that rotate in plane CD, as indicated in Fig. 14.10. As the reel rotates, the teeth on bar A (see vertical section) start to rake at some angle \( \alpha_1 \) from the centerline and continue to move the hay until the vertical clearance is sufficient to allow the teeth to pass over the roll of hay (at angle \( \alpha_2 \) in the vertical section and line EF on the plan view). The total vertical clearance would then be \( y_2 \) plus the height of the teeth above the ground at their lowest point. The forward motion of the rake during the angle \( \alpha_1 + \alpha_2 \) is indicated by \( x_1 \). After bar A drops the hay at EF, the reel will continue to rotate through the angle \( \beta - \alpha_1 - \alpha_2 \) before a tooth on bar B contacts the previously moved hay at F. The forward motion during this angle is \( x_2 \). If \( r \) is the radius of the tooth circle, then, by definition (expressing \( \alpha_1, \alpha_2, \) and \( \beta \) in degrees),

\[
R_{pf} = \frac{2\pi r (\beta - \alpha_1 - \alpha_2)}{360x_2}
\]

from which

\[
x_2 = \frac{2\pi r (\beta - \alpha_1 - \alpha_2)}{360R_{pf}} \tag{14.4}
\]
Fig. 14.10. Diagram showing the raking action and the effective raking stroke \((L_1 + L_2)\) for reel-type rakes.

The geometry of Fig. 14.10 indicates that

\[
x_2 = L_2 \cos \theta_{tr} + \frac{(L_1 + L_2) \sin \theta_{tr}}{\tan \gamma} + L_1 \cos \theta_{tr}
\]

\[
= (L_1 + L_2) \left( \cos \theta_{tr} \frac{\sin \theta_{tr}}{\tan \gamma} \right)
\]

(14.5)

Since \(L_1 = r \sin \alpha_1\) and \(L_2 = r \sin \alpha_2\), it follows that

\[
x_2 = r(\sin \alpha_1 + \sin \alpha_2) \left( \cos \theta_{tr} \frac{\sin \theta_{tr}}{\tan \gamma} \right)
\]

(14.6)

Combining equations 14.4 and 14.6 and solving for \(\beta - \alpha_1 - \alpha_2\),

\[
\frac{2\pi r(\beta - \alpha_1 - \alpha_2)}{360 R_{pf}} = r(\sin \alpha_1 + \sin \alpha_2) \left( \cos \theta_{tr} \frac{\sin \theta_{tr}}{\tan \gamma} \right)
\]

\[
\beta - \alpha_1 - \alpha_2 = \frac{360}{2\pi} R_{pf}(\sin \alpha_1 + \sin \alpha_2) \left( \cos \theta_{tr} \frac{\sin \theta_{tr}}{\tan \gamma} \right)
\]

(14.7)

The angle \(\theta_t\) between the resultant tooth path and the line of forward motion is determined from Fig. 14.10 as follows:
but, as in equation 14.4,

\[ x_1 = \frac{2\pi r(\alpha_1 + \alpha_2)}{360R_{pf}} \]  

Hence,

\[ \tan \theta_t = \frac{(L_1 + L_2) \sin \theta_{tr}}{(L_1 + L_2) \cos \theta_{tr} + \frac{2\pi r}{360R_{pf}} (\alpha_1 + \alpha_2)} \]  

In order to solve equation 14.7, a value for either \( \alpha_1 \) or \( \alpha_2 \) must be assumed. Since, in practice, \( y_2 \) will vary, depending upon the position along the reel, the size of the hay crop, and the clearance between the reel and the ground, it is convenient to assume reasonable values for \( y_2 \) (or determine them experimentally) and then calculate the corresponding values for \( \alpha_2 \) and \( L_2 \) from the relations,

\[ \cos \alpha_2 = \frac{r - y_2}{r} \]  

and

\[ L_2 = r \sin \alpha_2 \]  

Then equation 14.7 can be solved for \( \alpha_1 \), and the corresponding values of \( y_1 \) and \( L_1 \) can be computed from

\[ y_1 = r(1 - \cos \alpha_1) \]  

and

\[ L_1 = r \sin \alpha_1 \]  

Equations 14.11 through 14.14 are derived directly from trigonometric relations in Fig. 14.10. In applying them, careful attention must be given to the algebraic signs of \( \alpha_1 \) and \( L_1 \). Values of \( \alpha_1, \alpha_2, L_1, \) and \( L_2 \) are considered positive as shown in Fig. 14.10. Negative values for \( \alpha_1 \) and \( L_1 \) indicate that the start of the raking stroke is beyond the lowest point of tooth travel.

The next step is to determine the resultant tooth-path angle \( \theta_t \) from equation 14.10. Velocity vector diagrams can then be constructed (as in Figs. 14.12 and 14.13) to obtain values for the reel component \( V_{tr} \), the resultant tooth velocity \( V_t \), and the average hay velocity \( V_h \), all in relation to the forward velocity \( V_f \). The
reel component $V_{tr}$ will always be less than the tooth peripheral velocity, the ratio for a given rake depending upon the value taken for $y_2$ in Fig. 14.10.

**14.17. Analysis of Raking Action for Finger-Wheel Rake.** The action of one wheel is indicated in Fig. 14.11. The velocity of the axle in the plane of the wheel is the component $V_a$ of the forward velocity $V_f$. Since the tips of the teeth are in contact with the ground, causing the wheel to rotate, the peripheral velocity $V_p$ is numerically equal to $V_a$. The wheel will act upon the hay at some average height $y$ above the ground, with a reel (or wheel) component of

$$V_{tr} = \frac{r - y}{r} V_a = \frac{r - y}{r} V_f \cos (180 - \theta_{tr}) \quad (14.15)$$

where $\theta_{tr}$ is expressed in degrees. After assuming a value for $y$ and calculating $V_{tr}$, the resultant tooth velocity $V_t$ and the angle $\theta_t$ can be determined from a vector diagram as indicated in the plan view of Fig. 14.11.

**14.18. Action of a Drag-Type Side-Delivery Rake.** In general, the crowding action would be similar to that of a scraper
blade set at an acute angle with the line of forward motion. The raking unit exerts upon the hay a force perpendicular to the raking axis and a friction force parallel to the raking axis. The hay would be expected to move in the direction of the resultant force, or a little forward of the perpendicular force. However, it is possible to warp the rear or upper ends of the bars toward the discharge side of the rake and obtain a definite rolling action toward the windrow. This could cause the hay path to be a little behind the perpendicular to the raking axis.

The drag-type windrower depends upon contact with the stubble to provide a force at a low elevation that opposes the resultant of the rake forces on the hay and creates a couple, causing the hay to be rolled and moved to the windrow. For this reason, drag-type windrowers work best in moderate to heavy crops. Inherently they tend to move the hay more smoothly than reel-type rakes, because of the intermittent action and successive impacts of the reel tooth bars.

14.19. Determination of Length of Hay Path and Average Hay Velocity. Ideally, the average hay velocity should be the same as the resultant tooth velocity, a condition that theoretically exists in a drag-type windrower. But with rakes that move the hay by incremental steps, such as the reel types and the finger-wheel rake, the average hay velocity is always less than the resultant tooth velocity. After determining the resultant tooth-path angle $\theta_i$, the theoretical average hay velocity $V_h$ can be determined in relation to the forward velocity $V_f$ (for any type of side-delivery rake) by means of a vector diagram as indicated in Fig. 14.12, assuming that the hay follows the resultant tooth path.

The theoretical maximum length of hay path $S_h$, for moving hay across the full width $W$ of the swath along the resultant tooth path $AB_1$ (as the rake moves forward from $A$ to $A_1$), is

![Fig. 14.12. Vector diagram showing relation of average hay velocity to forward velocity.](image-url)
If field observations indicate that the hay-path angle is not the same as the resultant tooth-path angle \( \theta_t \), then the observed hay-path angle should be used in calculating the actual length of hay path and the actual average hay velocity.

**14.20. Comparison of Typical Rakes.** Calculations were made for typical rakes of three types, using the dimensions of rakes tested and reported by Giles and Routh.\(^9\) (It was necessary to assume a value of \( r \) for the finger-wheel rake.) The results are indicated graphically in Fig. 14.13 and summarized in Table 14.1.

### Table 14.1 THEORETICAL COMPARISON OF THREE TYPES OF RAKES

<table>
<thead>
<tr>
<th>Rake Type</th>
<th>Effective Stroke</th>
<th>Tooth-Path Angle</th>
<th>Average Hay Path Velocity</th>
<th>Theoretical Maximum Hay Path Ratio, ( V_r/V_h )</th>
<th>Swath, ft</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cylindrical-reel rake (four-bar, ( R_{RF} = 1.16 ), ( \theta_r = 50^\circ ), ( \gamma = 40^\circ ), ( r = 11\frac{1}{4} ) in.)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Assumed ( y_2 = 1 ) in.</td>
<td>8</td>
<td>24</td>
<td>26.6</td>
<td>70</td>
<td>2.75</td>
</tr>
<tr>
<td>Assumed ( y_2 = 0 ) in.</td>
<td>-23 *</td>
<td>62</td>
<td>22.5</td>
<td>72</td>
<td>2.20</td>
</tr>
<tr>
<td>Oblique reel-head rake (six-bar, ( R_{RF} = 1.20 ), ( \theta_r = 100^\circ ), ( \gamma = 50^\circ ), ( r = 12 ) in.)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Assumed ( y_2 = 1 ) in.</td>
<td>9</td>
<td>24</td>
<td>58.0</td>
<td>81</td>
<td>1.83</td>
</tr>
<tr>
<td>Assumed ( y_2 = 6 ) in.</td>
<td>-23 *</td>
<td>60</td>
<td>18.3</td>
<td>78</td>
<td>1.02</td>
</tr>
<tr>
<td>Finger-wheel rake (( \theta_p = 135^\circ ), ( \gamma = 25^\circ ), ( r = 27 ) in.)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Assumed ( y = 0 )</td>
<td>—</td>
<td>—</td>
<td>45.0</td>
<td>45</td>
<td>1.53</td>
</tr>
<tr>
<td>Assumed ( y = 3 ) in.</td>
<td>—</td>
<td>—</td>
<td>38.5</td>
<td>47</td>
<td>1.51</td>
</tr>
</tbody>
</table>

* Negative sign indicates that start of stroke is beyond lowest point of tooth travel.

Note that the table includes calculated results for two assumed values of \( y_2 \) or \( y \) for each rake. With the reel-type rakes it is probable that the hay is initially picked from the swath under conditions approximating those with \( y_2 = 1 \) in. However, in moving the hay ahead of the reel, \( y_2 \) must be great enough to clear the accumulated hay. Thus the conditions with \( y_2 = 6 \) in. might well represent the effective transporting stroke for a medium-size windrow; these conditions (shown in Fig. 14.13) should be assumed in calculating velocities and hay-path lengths.

It is of interest to note that the effective strokes shown for the four-bar cylindrical-reel rake and the six-bar oblique reel-head rake considered in this comparison are almost identical (same
values for $\alpha_1$ and $\alpha_2$ in Table 14.1). If this same oblique reel-head rake had only four bars, $\alpha_1$ for the condition with $y_2 = 1$ in. would be $26^\circ$ instead of $9^\circ$. In other words, the raking stroke would start earlier, at a time when the teeth were farther above their lowest position. There would then be a greater tendency to miss hay at the beginning of the stroke, unless the reel were set closer to the ground.

The results in Table 14.1 indicate that, for the particular implements selected, the cylindrical-reel rake has a much longer theoretical hay path and a higher ratio of resultant tooth velocity to average hay velocity than have the other two types. Both conditions are undesirable, the latter because of the indicated greater
impacts upon the hay by the teeth. The oblique reel-head rake has the shortest hay path but the highest average hay velocity. The finger-wheel rake shows the lowest average hay velocity and lowest velocity ratio, and has a relatively short hay path. In tests with these three rakes in fields of soybeans, lespedeza and alfalfa at average speeds of 4.8 to 5.9 mph, the leaf loss with the finger-wheel rake was considerably less than with the other two types, indicating a general correlation with the theoretical comparisons. However, the relative importance of the various velocity and distance factors has not been established.

**BALING**

Hay baling is essentially a packaging operation. Although it is one of the most expensive methods of handling hay, it is also the most popular. In 1951, 62 per cent of the entire hay crop was baled, in addition to 11 million tons of straw. Many of the factors that make baling attractive have to do with the handling of packaged material. Baling facilitates handling, selling, and storing. Storage space requirements are reduced, since the normal density of baled hay is 10 to 12 lb per cu ft as compared with a commonly assumed value of 4 for field-cured long loose hay.

**14.21. Classification of Balers.** Although stationary balers appeared in the United States as early as about 1850 and for many years were the only type available, very few stationary machines have been manufactured since 1949. Most balers are now of the field pickup type. These machines, however, are suitable for stationary baling as well as for field operation. Both hand-tie (wire) and automatic-tying balers are common; semiautomatic balers, with devices to automatically place the wires around the bale for hand tying, have been used to a limited extent. Automatic balers are available as wire-tying machines or twine-tying machines, both making rectangular-cross-section bales, or as round-bale machines that wrap the bale with twine. Round bales introduce some inherent difficulties in stacking and transporting, but the initial cost of the baler is considerably less than for other types.

Hand-tie field balers were introduced in 1930 but were used for less than 3 per cent of the total hay crop in 1939. The automatic twine-tying baler appeared commercially in about 1940,
followed within a few years by automatic wire-tying machines. The relative amounts of hay baled with the different types of balers in the United States in 1951 are indicated by the following figures, representing percentages of the total hay crop harvested:

<table>
<thead>
<tr>
<th>Type of Baler</th>
<th>Percentage</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hand-tie, wire (field and stationary)</td>
<td>12%</td>
</tr>
<tr>
<td>Automatic wire-tying</td>
<td>12%</td>
</tr>
<tr>
<td>Automatic twine-tying, rectangular bales</td>
<td>31%</td>
</tr>
<tr>
<td>Round bales (automatic twine-wrapping)</td>
<td>7%</td>
</tr>
</tbody>
</table>

14.22. Baler Sizes. In regard to size, balers making rectangular bales may be classified in terms of the cross-sectional dimensions of the bale chamber, by the number of wires or strings around each bale, or by the size of the crew (including the tractor driver) required to operate the baler. Thus, automatic-tying balers are considered to be one-man machines, whereas hand-tie balers require three or four men.

Some stationary balers and a few hand-tie field balers have 17 × 22 in. bale chambers (cross-sectional dimensions), producing three-wire bales 40 to 46 in. long that weigh 125 to 150 lb. There are also several models of automatic-tying balers (three-wire or three-twine) that produce large bales, a 16 × 24 × 48 in. size being typical. The use of these large-bale machines is confined mainly to the Pacific Coast, where a considerable quantity of hay is sold and transported commercially. Most balers now have 14 × 18 in. or 16 × 18 in. bale chambers and make bales about 36 in. long, with two wires or strings per bale.

The trend is toward smaller bales because they are easier to handle and because speed of tying is not a limiting factor in number of bales per minute when automatic balers are used. The following average bale weights were obtained in a USDA nationwide survey covering the 1951 season:

- Rectangular, hand-tied bales: 73 lb
- Rectangular bales, automatic wire-tying machines: 71 lb
- Rectangular bales, automatic twine-tying machines: 60 lb
- Round bales (36 in. long, usually 16 to 18 in. diameter): 57 lb

Any of the several types of balers are capable of making considerably heavier bales than indicated by these average weights.

14.23. Round-Bale Machines. Balers of this type pick up the hay and feed it between two belts (Fig. 14.14) that roll up
the mat of incoming hay under adjustable pressure. As the bale grows, rolls $D$ and $F$ are automatically moved to the rear. When the bale reaches the desired diameter (adjustable between 14 and 22 in.), the conveyor automatically stops feeding material and the bale continues to revolve while it is being spirally wrapped with ordinary binder twine (about 35 ft per bale). No knot is tied, adhesion between the hay and twine being relied upon to prevent unwrapping. After wrapping has been completed, rolls $D$ and $F$

Fig. 14.14. Bale being formed between the belts of a round-bale machine (Allis-Chalmers Mfg. Co.)

separate to discharge the bale. Then all rolls return to the broken-line positions shown in Fig. 14.14 and the conveyor drive is automatically engaged again. Since the pickup and conveyor are automatically stopped during wrapping, the forward motion of the machine must also be stopped each time a bale is being wrapped. Bale density is controlled by adjusting a friction brake that regulates the tension in the bale-forming belts.

14.24. Plunger-Type Field Balers. Basically, a field baler that produces rectangular bales includes the following components (Fig. 14.15):

1. A unit to pick up hay from the windrow and elevate it.
2. A conveyor to move the hay to the bale-chamber entry.
3. Packers to place the hay in the chamber while the plunger is on its retracted stroke.
4. A reciprocating plunger to compress the hay and move it through the bale chamber.

5. A means for applying forces to resist the movement of hay through the bale chamber and thus control the degree of hay compression and the resultant bale density.

![Diagram of field baler with pto drive](image)

Fig. 14.15. An automatic twine-tying field baler with pto drive. The flywheel on this model is inside of the frame, at the front. (International Harvester Co.)

6. A means of separating consecutive bales and placing the wires or strings around each bale.

7. Either automatic tying devices that operate when the bale reaches the preselected length or facilities for hand tying.

Although mounted engines (usually 15 to 20 hp) are the most common source of power for field balers, pto drives are satisfactory if the tractor has adequate power and sufficient range of forward speed to accommodate heavy and light windrows. Moderate variations in baler rpm do not seriously affect the performance. A live or constant-running pto is advantageous. The lower first cost and the reduced weight with the pto drive are both desirable.

14.25. Windrow-Pickup Devices. Field machines that may have windrow-pickup devices, either as an attachment or as an integral component, include field balers, field choppers, hay loaders, and combines. Most pickup units can be classified as (a)
cylinder-type with fixed spring teeth and stationary stripper plates, (b) cylinder-type with spring teeth mounted on cam-controlled tooth bars, (c) cylinder- or drum-type with retracting fingers, (d) chain-and-bar type with fixed teeth, (e) chain-and-bar type with spring fingers mounted on cam-controlled bars, or (f) flat-belt type with spring teeth attached to the face of the belt (principal application is on combines). The first two types are probably the most common on balers.

Pickup units may be driven either from the power source (engine or pto) or from a ground wheel. Ground drive maintains a constant ratio between the peripheral speed of the pickup teeth and the forward speed. It is the most desirable type of drive where shatter is a serious problem (as in seed harvesting). For best performance and minimum shattering, the peripheral speed should be 10 to 20 per cent greater than the forward speed.

An engine-driven pickup must be operated fast enough to have a peripheral speed greater than the maximum forward speed normally encountered. Consequently, the action on the windrow at any but the maximum forward speed is more vigorous than with a ground drive. The engine drive is most common on field balers, probably because reducing or interrupting the forward speed of the baler allows the pickup to tear apart heavy windrows or bunches, thus promoting more even feeding from non-uniform windrows. On some balers the operator can select either power drive or ground drive, at will.

14.26. Conveying and Feeding. Many different arrangements and combinations are employed to move the hay from the pickup device into the bale chamber. An auger is used for cross-conveying in most cases, sometimes with a belt-type conveyor beneath it. Top-feed balers (hay enters bale chamber from the top) generally have an elevating conveyor between the pickup unit and the cross-conveyor. Most of them utilize an overhead "wad board" or feeder head to force the material down into the bale chamber while the plunger is retracted. Side-feed balers may have packer fingers (Fig. 14.15), a ribbed drum held down against the top of the entering hay, or other suitable arrangements to feed the material from the cross-conveyor into the chamber.

14.27. Compressing Hay and Control of Bale Density. The speed of the reciprocating plunger that compresses the hay in the bale chamber varies with different baler models but is generally
between 40 and 65 rpm. As the plunger moves on its compression stroke, the hay charge is compressed until the plunger force becomes large enough to move the completed and partly formed bales along the chamber. On the return stroke of the plunger, the compressed hay is held by fixed wedges and by spring-loaded dogs that project into the bale chamber.

The density of the bales is primarily a function of the type of material, its moisture content, and the total resistance that the plunger must overcome in moving the material through the bale chamber (and up the bale chute if a trailer is used). Fixed wedges may be added or removed to obtain large changes in the resistance, but the principal method of controlling bale density is by squeezing together two sides or all four sides of the bale chamber at the discharge end. This convergence of the bale-chamber sides causes the hay to be compressed laterally as it moves through the chamber.

The increased resistance due to lateral compression is a function of (a) the required compressive force for the hay in relation to the amount of deflection (corresponding to the modulus of elasticity for structural materials), (b) the coefficient of friction between the hay and the bale-chamber surfaces, (c) the amount of deflection of the bale-chamber sides, and (d) the length of the converging or deflected section of bale chamber.¹⁸

One of the problems encountered in baling hay is the change of bale density as the moisture content varies from one part of the field to another, or as it changes with time. According to Raney,¹⁸ this is because dry materials have a considerably lower modulus of elasticity and a lower coefficient of friction than do materials with a higher moisture content and thus require more deflection of the sides to produce a given resistance force. Burrough and Graham⁷ demonstrated with alfalfa that, although part of the increase in density with increased moisture content at a given adjustment is due to the additional water, the amount of dry matter per bale is also considerably greater. They also found that for a given adjustment there was an appreciable increase in density as the baling rate was increased.

With the conventional type of so-called tension control (deflection adjustment), coil springs permit some change in deflection if the lateral force changes (Fig. 14.15), but compensation for changing conditions is primarily by manual adjustment of the
tension bolts. A system that gives substantially uniform lateral pressures at all deflections, such as a hydraulic cylinder maintained at constant oil pressure, overcomes the effects of changes in the modulus of elasticity but does not compensate for changes in the coefficient of friction. Theoretically, the effects of changes in both the modulus of elasticity and the coefficient of friction could be overcome with a deflection system having a control device that automatically increased the lateral pressure as the deflection increased.

Many types of devices have been developed in an attempt to automatically maintain more uniform bale densities. One of the more recent arrangements is illustrated in Fig. 14.16. This device employs a hydraulic cylinder and a linkage such that all four sides of the bale chamber are squeezed together. The hydraulic pressure is controlled by a spring-loaded star wheel that runs in contact with the bale, the points penetrating to a greater or lesser depth depending upon the bale density. If the density increases, for example, the wheel is moved outward and operates a control valve to automatically reduce the oil pressure in the hydraulic cylinder.

14.28. Bale Separation and Tying Systems. With stationary balers and the earlier hand-tie field balers, consecutive bales in the machine were separated by means of wooden blocks. These
blocks, which were grooved on each side to permit insertion of the
tie wires through the bale chamber, were dropped vertically into
the feed opening while the hay feed was interrupted for one or
two strokes of the plunger. All present-day rectangular-bale field
balers have a knife on the plunger that acts in conjunction with
a stationary ledger plate to completely separate successive hay
charges by slicing. This slicing facilitates the breaking up of
bales for feeding livestock and eliminates the need for blocks
to separate bales. Most hand-tie machines now have straight
needles, grooved on each side, that are inserted between hay
charges to provide passageways for manually putting the wires
through the bale chamber, or else have automatic wire-threading
devices. The needles of automatic-tying devices pass through
slots in the plunger face while the plunger holds the hay in the
compressed position.

The usual method of controlling bale length with an automatic
baler is by means of a wheel with pointed projections around its
periphery that is rotated by contact with the bale in the chamber.
When the material in the chamber has moved through the pre-
determined bale length, the metering device engages the clutch for
the tying unit. The cycle does not begin, however, until the
plunger reaches the proper position for correct timing between the
needles and the plunger.

Automatic-tying devices overcome the capacity limitations im-
posed by the time required to make a tie by hand. Some auto-
matic balers unlatch the plunger from the crank for one revolution
while each bale is being tied, leaving the plunger in the extended
position (hay compressed). This allows 1 to 2 sec for the com-
plete tying cycle. If no stroke is missed, the complete cycle,
including needle travel, takes place in somewhat less than one
crank revolution, in a total time of less than 1 sec.

Wire-tied bales (hand or automatic) are tighter and more
durable than twine-tied bales and, for this reason, are favored
where hay is sold and must be handled several times. Twine bales
are looser and more tolerant to high moisture contents. Twine is
not injurious to livestock and is easier to dispose of than is wire.

A survey made in 1951 indicated that automatic wire-tying
balers required an average of 7.7 lb of wire per ton. Rectangu-
lar, twine-tied bales used 3.1 lb of baler twine per ton and round
bales required 2.6 lb of binder twine per ton. Based on the prices
that prevailed in Central California in 1954, the calculated costs of twine or wire per ton of hay are $0.80 to $0.90 for any of the three types of bales.

14.29. Automatic Twine-Tying Devices. The tying units on twine balers are basically the same as those used for many years on grain binders, but they have appropriate refinements and im-

![Diagram of a typical knotter head from an automatic baler.](International Harvester Co.)

provements. Baler twine is considerably heavier than binder twine, having an average tensile strength of at least 270 lb.21 Figure 14.17 shows the basic components of a typical knotter. In the left-hand view the needle has just brought the twine around the bale and placed it in the twine holder. The two outside disks of the holder now rotate through the angle between adjacent notches while the center disk remains stationary, thus pinching the twine between the spring-loaded disks and holding it as the needle withdraws. In the right-hand view the knotter-bill assembly has rotated almost one turn to form a loop in the string, around the knotter bills. As the knotter-bill assembly rotates further, the bills close over the strings held by the twine holder. Then the stripper moves forward to pull the loop off of the bills.
over the ends held between them, and the knife attached to the stripper arm cuts the twine.

An automatic-tying system, whether twine or wire, is a complex mechanism requiring careful adjustment and proper timing of the parts, together with adequate maintenance and replacement of worn parts. The operating or service manuals for most automatic balers include rather complete descriptions and instructions pertaining to their particular tying mechanisms.

14.30. Automatic Wire-Tying Devices. Some automatic balers require only one coil of wire for each strand around the bale and make a single tie in each strand. With this system, as the bale moves through the chamber the wires must be pulled around the front end of the bale. Other machines use two coils per strand, one on each side of the bale chamber (four coils on a two-wire baler), so that wire need not be pulled around the front end of the bale. In this system, a single, twisted tie is made in each strand and then cut in the center to leave two ties per strand, one on each end of the bale.

In order that coils of wire from different manufacturers can be interchangeable in various balers, the ASAE has adopted standards for baling wire. These standards, which are included in Appendix E, specify 14½-gage annealed wire having a tensile strength of 50,000 to 70,000 psi. Dimensions and winding con-

![Diagram of a wire-tying mechanism for a four-coil baler](image-url)

Fig. 14.18. Two views of an automatic wire-tying mechanism for a four-coil baler. (Deere & Co.)
ditions are specified for two standard sizes of coils (for two-coil and four-coil balers) and for one interim size.

Figure 14.18 shows two views of the wire-tying mechanism on a baler using the four-coil system (two coils per strand). In the left-hand view the needle is bringing the wire from one coil around the end of the completed bale to be placed in the twister-pinion slot against the wire from the other coil. In the right-hand view the twister pinion has revolved several times, twisting the two wires together. The kinker shafts are rotating through 180° to double-kink the splice on each side of the twister (for increased strength). A shearing edge on the right-hand kinker, operating against the face of the pinion, is about to cut the twist in two. The needle then retracts, leaving the two wires joined on the completed bale and leaving the two coils joined at the right to form a continuous strand across the front of the next bale to be formed.

14.31. Baler Capacities. Some of the machine characteristics that affect the tonnage capacity of a baler are (a) the size of the bales, (b) the number of plunger strokes per minute, (c) capacity limitations of the pickup and feed mechanisms, (d) the amount of power available, (e) type of tying system, i.e., hand or automatic, and (f) the durability and reliability of the machine. Important operating factors include (a) the size and uniformity of the windrow, (b) the condition of the field surface, as it limits forward speeds, (c) the condition of the hay, (d) the density of the bales, and (e) the skill of the operators, particularly with hand-tie balers.

It should be apparent from a consideration of the above items that baling capacities can vary widely. According to a USDA survey, the average baling rate with three-man hand-tie field balers in 1945 was 2.4 tons per hr. For each of two automatic balers (one twine and one wire), Barger reports seasonal averages of a little more than 5 tons per hr. Rates as high as 8 to 10 tons per hr are not uncommon for a single day's operation of an automatic baler.

14.32. Power Requirements. By means of strain gages mounted on the plunger connecting rod and on appropriate shafts, Burrough and Graham determined the power inputs to the various components of an automatic twine-tying baler over a rather wide range of conditions. Figure 14.19 shows the relation
between average plunger power requirements and feed rate* for four crop conditions, as well as the total power required by all other baler components. The data obtained by Hansen and presented in Table 5.2 (Chapter 5) indicate total average power requirements ranging from 10 to 19 hp, but crop conditions are not given and baling rates may have been considerably above the range of Fig. 14.19.

Comparison of curves A and B indicates that with the rather dry wheat straw the plunger power requirements were about doubled when the bale density was increased by only 25 per cent, from 8.4 to 10.5 lb per cu ft. Curve D shows the plunger power requirements for alfalfa at 23 per cent moisture content, which is a good baling condition for this crop. At this moisture content the effect of bale density upon power requirements was much less than with the dry straw.

In another series of tests, using alfalfa and adjusting the density to maintain a constant amount of dry matter per bale, the plunger power requirements increased as the moisture content was reduced. The effect was most pronounced at low moisture contents,

*The feed rate is the rate at which material enters the baler during the actual operating time (i.e., at 100 per cent field efficiency).
with very little change above 20 to 25 per cent moisture. Thus, curve C in Fig. 14.19, in spite of the lower bale density, indicates greater power requirements than curve D because the moisture content of the hay was considerably lower (13 per cent versus 23 per cent).

Figure 14.20 shows the relation of plunger-face or press-plate

![Plunger work diagrams for two feed rates in alfalfa. Plunger friction and inertia forces are not included. (D. E. Burrough and J. A. Graham. Agr. Eng., April, 1954.)](image)

force to plunger displacement, for two feed rates. These curves are actually work diagrams for compressing the hay and moving the bales through the chamber, the area under each curve representing the total work involved. The two curves are similar except that compression starts later with the smaller hay charge and the peak force is lower. Peak forces as high as 17,000 lb were encountered when baling wheat straw at 5 per cent moisture. The small peaks in the vicinity of 12 to 16 in. displacement (Fig. 14.20) are due to shearing of the hay charge by the knife on the plunger. Their magnitude and location will vary for different baler models, usually occurring later than on the particular baler tested. Otherwise, these work diagrams can be applied to other
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balers by making proper allowances for bale-chamber size and crank rpm.7

From the work diagram, plus plunger friction and inertia forces, one can determine the relation of crank torque to crank angle. Consideration of crank speed then yields plunger horsepower as plotted in Fig. 14.19. In their tests, Burrough and Graham found that the peak plunger power requirements during the compression part of the stroke were about 8 to 12 times the average plunger horsepower when baling wheat straw at 5 per cent moisture (curves A and B), and 7 to 9 times the average for alfalfa at 23 per cent moisture (curve D). These peak requirements must be supplied by slow-down of the baler flywheel.7

14.33. Overload Protection. The transmission system and compression chamber are often subjected to high overloads as a result of variations in moisture content of the hay, picking up of foreign objects, or careless operation. Thus a safety clutch or shear device in the drive between the flywheel and the plunger is important. As explained in Chapter 5 (Section 5.21), a friction slip clutch in the drive ahead of the baler flywheel is important on pto-driven balers to protect the pto drive from peak torques caused by the compression stroke of the plunger. Additional desirable protection includes (a) a device that releases if the needles strike an obstruction, (b) a shear pin in the drive to the tying mechanism, (c) a safety clutch in the pickup and conveyor drive, and (d) a device to prevent overloading of the feed mechanism.

REFERENCES


**PROBLEMS**

14.1. (a) How much water must be removed to obtain 1 ton of cured hay at 20 per cent moisture content (wet basis), starting with green hay at 75 per cent moisture content?

(b) What percentage of this amount is removed in the swath if the hay is raked at 55 per cent moisture content?
14.2. Consider a mower with dimensions indicated in Fig. 14.3, operating at a crank speed of 1000 rpm. Assume sinusoidal motion of the knife. Compute:

(a) Crank angles from beginning of stroke, for start and finish of cutting.
(b) Per cent of stroke during which cutting occurs.
(c) Per cent of forward travel during which cutting occurs.
(d) Maximum velocity of knife, in feet per minute (with respect to ledger plate).
(e) Knife velocities at start and finish of cutting.

14.3. A mower operating at a crank speed of 1000 rpm has $S = 9\frac{1}{4}$ in.; $L = 41\frac{3}{4}$ in.; $R = 13\frac{1}{2}$ in. (see Fig. 14.4); knife weight = 9\frac{3}{4} lb; pitman weight = 7\frac{3}{4} lb, with center of gravity 19 in. from crank end; crankpin weight = \frac{3}{4} lb. Calculate:

(a) Inertia force $F_h$ at each end of stroke.
(b) Mean value of reciprocating unbalance $F_r$ for the two ends of the stroke.
(c) The rotating unbalance $F_c$.
(d) The required centrifugal force for a counterweight on the flywheel.
(e) The resulting maximum unbalance.

14.4. Measurements of an existing four-bar oblique reel-head rake gave $R_p = 1.34$, $\theta_r = 72^\circ$, $\gamma = 63^\circ$, and $r = 11\frac{3}{4}$ in.

(a) Assuming $y_0 = 6$ in., calculate $a_1$, $a_2$, $L_1$, $L_2$, and $\theta_r$.
(b) Draw a vector diagram (similar to those in Fig. 14.13) and determine graphically the values of $V_t$, $V_h$, and $V_{tr}$ in relation to $V_f$.
(c) Indicate on a cross-sectional view (to scale) the beginning and end of the effective pushing or transporting stroke (as in the top part of Fig. 14.13).

14.5. An automatic baler averages 5 tons per hr, with a field efficiency of 70 per cent. The plunger makes 55 strokes per minute, and the average bale weight is 65 lb. The yield is 1\frac{3}{4} tons per acre.

(a) Compute the actual number of compression strokes required per bale.
(b) Recommend the number of 7-ft rake swaths that should be put into one windrow and the corresponding forward speed required.

14.6. Compute the range of baling-wire strengths corresponding to the ASAE standards (Appendix E) and compare with the average strength of baler twine mentioned in the text.

14.7. The average power supplied to the crank arm of a baler is 9 hp. During the compression portion of the stroke, the amount of energy required above the average is 3100 ft-lb. This energy must be supplied by the flywheel, whose mass moment of inertia is 14.8 lb-ft-sec$^2$. A 109-tooth gear on the crankshaft is driven by a 14-tooth gear on the flywheel shaft.

(a) Assuming that the crank speed is 57 rpm at the start of compression, calculate the crank rpm at the end of compression and the percentage decrease in rpm during compression.
(b) If the slow-down occurs during 55° of crank rotation, what is the average horsepower released by the flywheel during this period?
Forage Chopping and Handling

15.1. Introduction. Stationary choppers for corn silage date back to the latter part of the nineteenth century, whereas field choppers, commonly known as forage harvesters, appeared in the late 1930's. The annual production of corn silage in the United States averaged about 35 million tons during the five years from 1940 through 1944. Corn for stationary chopping is usually cut with a binder and fed into the ensilage cutter by hand, which involves considerable hard labor. Stationary choppers are also used to some extent for hay. Field choppers, however, are rapidly replacing stationary cutters, both for silage and for hay chopping, primarily because of the inherent reduction in labor requirements.

With a field chopper and its complementary equipment, the hay-making or silage-making operation can be completely mechanized. The chopped material is blown directly into trailers that are towed behind the chopper or pulled beside it with a separate power source, or into trucks driven behind or beside the chopper. These vehicles are usually covered, especially for cured or partially cured material, to prevent excessive losses during the loading operation and in transit. Chopped forage may readily be unloaded by mechanical means, the most common method being with devices that move the load to the rear of the vehicle at a controlled rate and drop it onto the feed table or hopper of an elevating device. Transport and mechanical-unloading equipment is discussed more fully in Chapter 22. Elevation of the chopped material into storage is most commonly done with an impeller-blower, although mechanical elevators are employed to a limited extent for moderate heights.

Although field chopping represents the most highly mechanized
method of harvesting hay or making silage, this system involves a large investment in equipment. Thus it is economically practical only where a considerable tonnage of material is to be chopped each year. If several different products, such as grass silage, hay, and corn silage, are all chopped with the same machine, the annual use may be increased and the cost per ton reduced. Chopped hay requires 50 to 75 per cent as much storage space as does long loose hay.

15.2. Basic Components of Choppers. In general, forage choppers, whether field or stationary, include the following components:

1. A conveyor or feed table for the material to be chopped.
2. Spring-loaded feed rolls or aprons to compress and hold the material for chopping.
3. A cutterhead or chopping unit.
4. An impeller-blower (or other conveying device) to elevate the chopped material into storage or deposit it in the transporting vehicle.

The field chopper must, in addition, have a gathering unit to cut standing plants or to pick up windrowed material.

Specially designed stalk shredders, similar to the rotary beaters described in Chapter 9 (Section 9.15), are used to some extent for chopping green forage. These machines have a series of free-swinging chains, flails, or hammers attached to a horizontal rotor whose axis of rotation is perpendicular to the direction of travel. Knife sections may be attached to the ends of the flails. The power-driven rotor rotates in the opposite direction from that of the ground wheels, shredding the standing material by impact and carrying it up over the top of the rotor where a curved hood or deflector directs it upward and rearward into the trailed wagon. Although there is some question about the effect of the severe beating upon the quality of the product, the first cost of these units is far less than for the conventional cutterhead-type of field chopper.

Another type of low-cost machine for chopping green forage employs an impact cutter with the knives rotating in a horizontal plane at a peripheral speed of about 10,000 fpm (see Section 14.4). The height of the knives above the ground is adjustable. Impeller blades, similar to those on a flywheel-type cutterhead
(Section 15.5), are attached to the outer ends of the knife arms. These impellers collect the chopped material within a horizontal, concentric housing in much the same manner as in the flywheel-type cutterhead, and discharge it through a pipe into a trailed wagon.

15.3. Gathering Units. Field choppers have one or more of the following interchangeable gathering attachments, which

makes them adaptable for a variety of jobs: (a) windrow-pickup unit, for green or wilted hay intended for silage or direct feeding, for partially cured hay that is to be barn cured, or for field-cured hay; (b) cutter-bar unit for direct mowing and chopping of hay or grass for silage or direct feeding; and (c) row-crop attachment for direct cutting of such crops as corn and sorghum for silage.

Windrow-pickup units are similar to those on field balers and are discussed in Chapter 14 (Section 14.25). A typical row-crop attachment is shown in Fig. 15.1. One-row units are most common, but two-row units are available. The cutting unit of the cutter-bar attachment is similar to that of a conventional mower and is located across the front of the conveyor assembly as on the header of a combine. A reel of some sort is generally
mounted above the cutter bar. Widths of cut ranging from 4 to 5 ft are common.

15.4. Feed Mechanisms. Whether the chopper is of the stationary or the field type, the material is deposited on a conveyor that moves it to the feed rolls. The functions of the feed rolls are to compress the material to be cut, feed it positively to the chopping unit, and hold it while cutting takes place. An auxiliary clutch, to permit stopping or reversing the feed mechanism if an overload occurs, is a desirable feature. A typical feeding arrangement is shown in Fig. 15.2. The lower feed roll is usually smooth, whereas the upper feed roll has longitudinal ribs, to provide maximum ability to hold the material. A chain-and-slat apron with notched angle-iron cross-bars sometimes takes the place of one or both of the upper rolls.

Note that both of the upper feed rolls in Fig. 15.2 are spring loaded, with provision for considerable vertical movement to accommodate varying amounts of material. The cross-sectional area defined by the minimum width of the feed opening at the feed rolls and the maximum operating clearance between the upper and lower feed rolls is known as the throat of the chopper.
Throat area is one of the factors affecting the capacity of a chopper, as discussed in Section 15.16.

For the most positive feed, the peripheral speeds of all feed rolls should be the same as the linear speed of the conveyor, taking into account the effective diameter of corrugated or ribbed rolls rather than the outside diameter. The upper and lower feed rolls should be as close as possible to the vertical cutting plane so that long pieces of material will not be pulled through by the knife. The top of the shear bar and the top of the lower feed roll should be at about the same height.

15.5. The Cutterhead. Two general types of chopping devices, known as the flywheel or radial-knife type and the cylinder or helical-knife type, are found on both field and stationary choppers. The knives of the flywheel type (Fig. 15.3) are mounted in radial or nearly radial positions on one side of a steel-plate flywheel or on heavy steel spokes. The fan or impeller blades for elevating the chopped material are attached to the outer edge of the same wheel or spokes. The orientation of the knives on the flywheel and the relation between the heights of the stationary shear bar and the flywheel shaft are such that cutting occurs progressively along each knife, starting from either the inner or the outer end (Fig. 15.3).

The cylinder-type cutterhead has helically shaped knives mounted around the outside of a cylindrical reel. Some machines elevate the chopped material with a separate fan wheel, similar to the impeller-blower of a flywheel-type cutterhead. On others, the knives are designed to act as the impellers (Fig. 15.4). If the cylinder is long (3 ft, for example), cutting will take place in a relatively thin layer. This requires less convergence in feeding and provides a better shearing action than with narrow, thick layers. In addition, the short lift required for the upper feed rolls facilitates protection against large stones. On the other
hand, a thick layer of material, as required with a flywheel-type cutterhead or a short cylinder (14 to 18 in. throat width), is considered better for chopping corn silage because the ears are held more firmly in the thick mat.

Either flywheel- or cylinder-type cutterheads ordinarily have four or six knives, with provision to remove some of them for changing the length of cut. Knife removal and replacement is somewhat easier on the flywheel type, but the cylinder type is adaptable to built-in sharpeners, thus reducing the frequency of removals.

15.6. Chopping Lengths. The chief objective in chopping hay is to reduce the material to lengths that can be handled by an impeller-blower and moved in a pipe along with an air stream.1 With silage, additional functions of chopping are to facilitate packing for exclusion of air and to make feeding easier. Chopping into lengths shorter than necessary increases the power requirements and may reduce the capacity of the chopper. A theoretical cut of \( \frac{3}{2} \) to \( \frac{3}{4} \) in. is usually fine enough for silage, although shorter settings are sometimes used. Feeding experience indicates that the theoretical cutting length for cured hay should be at least 3 in. if the chopped hay is to be consumed as readily as long hay.2 Shorter cutting of cured hay also makes it more dusty and less tolerant to moisture in storage.11

The theoretical length of cut is the amount of advance of the feed mechanism between the cuts of successive knives. The theoretical length is adjusted by changing the speed ratio be-
between the feed mechanism and the cutterhead and/or by changing the number of knives, as discussed in Section 15.17. The actual length of cut will be the same as the theoretical only when the stalks feed in straight, as with row crops such as corn. With windrowed crops many of the stems do not feed in straight, and the actual length may average twice the theoretical setting, with some pieces being three or four times this length.

15.7. Feed Interference. The problem of obtaining relatively long cuts of cured hay to improve its palatability is com-

![Diagram showing clearance relations in regard to feed interference in a flywheel-type cutterhead.](image)

Fig. 15.5. Clearance relations in regard to feed interference in a flywheel-type cutterhead. (Orrin I. Berge. *Agr. Eng.*, February, 1951.)

plicated by feed interference and obstructions to flow within the cutterhead. Adjusting for a long theoretical length of cut does not assure long lengths of material if there is further breaking up or pulverization due to feed interference or to an excessive beating and pounding action upon the chopped material within the housing.

If the theoretical length of cut is too great in relation to the knife clearance angle $\beta$ (Fig. 15.5), there will be interference between the uncut hay feeding in and the back of the knife or its support. This will cause undesirable breaking up of the hay. For a flywheel-type cutterhead, the maximum theoretical length of cut without feed interference, or the minimum cutting radius for a given length of cut, can be obtained from the following geometric relation (Fig. 15.5):

$$2\pi r = Ny = N \frac{L}{\tan \beta}$$ (15.1)
where \( r \) = distance between the flywheel shaft centerline and the inner edge of the throat.

\[ N = \text{number of knives on the cutterhead.} \]

\[ y = \text{circumferential distance between knives at radius } r. \]

\[ \beta = \text{clearance angle between the knife and support assembly and the plane of rotation.} \]

\[ L = \text{maximum theoretical length of cut without feed interference.} \]

A similar relation can be derived for a cylinder-type cutterhead.

As an example of the application of equation 15.1, consider a flywheel-type cutterhead having four knives mounted with the beveled sides facing the shear bar (as in Fig. 15.5) and with \( \beta = 20^\circ \). The minimum radius for obtaining a 3-in. theoretical cut with no feed interference is

\[ r = \frac{4 \times 3}{2\pi \tan 20^\circ} = 5.25 \text{ in.} \]

But, if the knives are mounted with the flat side toward the shear bar and with \( \beta \) equal to perhaps 5° or less (Fig. 15.2), then even with only two knives the minimum radius for a 3-in. cut becomes 11 in. or more.

Proper streamlining and location of knife supports on the cutterhead are important in minimizing the further breakup and pulverization of cut material. Stroboscopic observations of material being cut and passing through a flywheel-type cutterhead indicated that the material, after being cut, was almost instantaneously accelerated to the speed of the knife and thereafter moved along with it. Thus, most of the material flow after cutoff was radial with respect to the knife, and long-cut material tended to collect on the outer knife mountings of the typical arrangement illustrated in Figs. 15.2 and 15.3. The result was unnecessarily severe action on the hay, and there was also a tendency for the collected bunches to cause clogging when they broke away.

To overcome these difficulties, a streamlined knife mounting, properly located to avoid interference with the natural flow paths of the cut material, has been developed. On this cutterhead (Fig. 15.6) the knives are mounted with the bevel facing
the shear bar, to reduce feed interference. This cutterhead has given good performance on long-cut hay.³

Cylinder-type cutterheads with cupped knives (Fig. 15.4) handle the cut material without excessive pounding, as long as the theoretical length of cut does not exceed the radial depth of the cupped sections. When flat knives and a separate impeller-blower are employed, long-cut hay tends to collect inside of the cylinder. The material is also subjected to more impacts and sudden changes of direction, which probably results in more breakup and pulverization of long-cut hay.

15.8. Theory of the Impeller-Blower. The impeller-blower, which is the most common type of device for elevating chopped forages (as well as some other materials), depends primarily upon the “throwing” action of the blades rather than upon air velocity. According to Segler,⁵ the air in the pipe of such a device normally has a lower velocity than the solid material leaving the blades and, therefore, has a retarding effect during the first part of the conveying. Since the “thrown” material slows down because of the effect of gravity while the air velocity remains constant in a given size of the pipe, the air may give some conveying energy to the material in the last part of the pipe, particularly when moving dry materials such as chopped hay.

The impeller-blower has a relatively small number of blades (usually two to four on field choppers and four to six on forage blowers), and the housing is concentric, with close clearances maintained between the blades and the housing. Duffee⁶ recommends a radial clearance of not over \( \frac{1}{8} \) in., especially from the bottom of the housing around to the point of discharge, and side clearances not greater than \( \frac{3}{4} \) to \( \frac{1}{2} \) in.

15.9. Elevating Efficiency. In general, it is reasonable to assume that the cut material leaves the impeller blades at about the peripheral speed of the impeller. Theoretically, material leaving the impeller blades at a vertical velocity \( V \) would be
elevated to a height of \( h = \frac{V^2}{2g} \). The actual height attained with a given peripheral speed of the impeller is considerably less than the theoretical value because of friction losses in the pipe and because the material must have sufficient energy at the top to carry it around the elbow and deflect it into the receptacle. If the peripheral speed for a given height is too low, blockages will occur in the pipe. Operation at speeds greater than needed for elevation results in excessive power requirements. If the pipe is too small, increased friction losses reduce the elevating efficiency so that greater speeds and more power are required. There are no data available on the effect of having too large a pipe. Diameters from 6 to 9 in. are common on forage choppers and forage blowers.

In a series of tests with stationary corn ensilage cutters, Duffee 10 found that for heights of 35 to 75 ft (from ground level to elbow at top of pipe), the actual heights were from 53 to 42 per cent of the theoretical values represented by the minimum peripheral speeds. For a height of 100 ft, the elevating efficiency dropped to 38 per cent. With corn silage and elevations up to 75 ft, he recommends speeds representing an elevating efficiency of 40 per cent. There is little information available regarding elevating efficiencies for other materials, but field experience indicates that minimum peripheral speeds for grass silage should be about 15 per cent greater than for corn silage. Segler 15 presents curves and a formula to show that the elevating efficiency varies inversely with the cube root of the weight rate of material. Although his formula (based on tests with 10-in. pipe and heights up to 66 ft) indicates elevating efficiencies of only 30 per cent at 10 tons of green maize per hour and 26 per cent at 15 tons per hr (which appear to be rather low), it is reasonable to expect that the elevating efficiency would decrease as the material feed rate is increased.

Field choppers are required to elevate the material only a few feet, but the discharge pipes are sometimes inclined rather than vertical (thus increasing friction losses), and the material leaving the deflector must have considerable velocity to carry it to the rear of the wagon. As a result of field tests in chopping

cured hay, Berge\textsuperscript{3} found that a minimum peripheral speed of about 4400 fpm was satisfactory for full loading of the wagon, provided the delivery pipe had a properly designed elbow or curve. At this low speed there was very little separation of leaves and stems and very little leaf pulverization. A minimum peripheral speed of about 4000 fpm is recommended for corn silage.\textsuperscript{2}

\textbf{Fig. 15.7.} Distribution of power in a flywheel-type field chopper at one peripheral speed of the impeller-blower and one theoretical length of cut. (Fredrick Z. Blevins. Unpublished thesis. Purdue University, 1954.)

\textbf{15.10. Distribution of Power Requirements.} The power required to operate a forage chopper is utilized in the following ways: (a) to gather, convey, and compress the material to be cut, (b) to shear the material, (c) to move the air pumped by the cutterhead and impeller-blower, (d) to accelerate the cut material to approximately the peripheral speed of the impeller, (e) to overcome losses due to friction of the cut material within the housing, and (f) to overcome mechanical losses in the machine.

Figure 15.7 shows the distribution of power as a function of feed rate, for one peripheral speed of a flywheel-type cutterhead,
as obtained by Blevins \(^4\) in tests with green alfalfa. Strain gages were installed on appropriate drive shafts to obtain power inputs to the flywheel and to the pickup and feed mechanism. Curve \(A\), representing the power required for cutting, was obtained from tests made after removing the impeller blades and the peripheral band of the fan housing. The observed results were corrected for kinetic energy imparted by the knives and for air horsepower as determined with no material passing through the cutterhead. The difference between curves \(B\) and \(A\) represents the calculated kinetic energy of the discharged material when operating with the fan shroud and impeller blades in place.

Curve \(D\) represents the total power input to the flywheel shaft. Curve \(C\) was obtained by measuring the air horsepower with no material being chopped, and then subtracting this amount from curve \(D\), assuming that the air horsepower was not affected by the feed rate. The friction horsepower, then, is the remaining difference between curves \(C\) and \(B\).

The difference between curves \(E\) and \(D\) represents the actual power determinations for the pickup and feed mechanism. Note that it increases more rapidly at high feed rates than at the lower rates. This power component should be relatively independent of cutterhead speed, unless serious feed interference occurs.

15.11. Air Horsepower. According to well-established fan laws, the power required to move the air should vary about as the cube of the peripheral speed. If this power component is relatively independent of the feed rate, as has been assumed, the energy input to the air per ton of chopped material varies inversely with the feed rate. Although the air power requirements are somewhat different for different machines, the requirements reported by several investigators, involving four field choppers and two forage blowers, can all be represented reasonably well (within \(\pm\) 30 per cent) by a curve that follows the cube law and passes through the point representing \(1\frac{1}{2}\) hp at 6000 fpm.

15.12. Cutting Energy. The cutting energy per ton of material varies with the type of material, the moisture content, and the length of cut. In fifteen different groups of tests with alfalfa, Blevins \(^4\) found that for a given moisture content and theoretical length of cut, the cutting horsepower was directly proportional to the feed rate (as indicated in Fig. 15.7). In other words, the cutting energy per ton is constant, regardless of the feed rate.
When the cutting unit is in good condition, the cutting energy should be relatively independent of knife speed. With dull knives, there is probably some advantage in the higher knife speeds.

The right-hand graph in Fig. 15.8 shows the effect of cutting length upon cutting energy requirements for green alfalfa. The left-hand graph indicates the effect of moisture content upon the cutting energy per ton of dry matter. In the tests with a 1-in. theoretical cut (and also for a ½-in. cut with another machine), the wilted alfalfa required more cutting energy per ton of dry matter than did the green or the cured alfalfa. On the wet-weight basis, the difference between green and wilted alfalfa would be even greater.

15.13. Friction Energy. As indicated in Fig. 15.7, a considerable amount of power may be absorbed (and wasted) by friction between the chopped material and the periphery of the housing. It can be shown analytically that the friction energy per ton of material, lost as a result of the effect of centrifugal force, is

\[ E_f = 1.522 \times 10^{-10} \mu \theta V^2 \]  

(15.2)
where $E_f = \text{friction energy, in horsepower-hours per ton of material.}$

$\mu = \text{coefficient of friction between the chopped material and the housing.}$

$\theta = \text{angle subtended by the average arc of housing periphery rubbed by the chopped material, in degrees.}$

$V = \text{peripheral speed of impeller, in feet per minute.}$

The angle $\theta$ must represent the average arc of contact for all material passing through the unit, since all the material from a flywheel cutterhead does not strike the housing at the same place.

Note that the friction energy per ton is independent of the feed rate but increases as the square of the peripheral speed. With large particles, such as chopped corn, there is probably very little wedging action between the impeller blades and the housing. But it is believed that with grass silage the smaller particles result in considerable wedging, which increases the energy loss due to friction. Field observations of the housing becoming hot when chopping grass silage provide evidence in support of this effect.²

Coefficients of friction for chopped materials on polished galvanized steel were determined by Richter¹⁺ in a series of tests at various normal pressures and sliding velocities. On the basis of his results, Richter recommends the following coefficients for design purposes:

<table>
<thead>
<tr>
<th></th>
<th>Static</th>
<th>Sliding</th>
</tr>
</thead>
<tbody>
<tr>
<td>Chopped hay and straw</td>
<td>0.35</td>
<td>0.30</td>
</tr>
<tr>
<td>Silages</td>
<td>0.80</td>
<td>0.70</td>
</tr>
</tbody>
</table>

Blevins ⁴ got about the same values as Richter for the sliding coefficients of green alfalfa (silage) and chopped hay (23 per cent moisture) on stainless steel but a considerably lower value for corn silage. His experimental determination of the friction horsepower for two field choppers (as described in Section 15.10) indicated a value of 41.8 for the product $\mu \times \theta$ in equation 15.2.

15.14. **Kinetic Energy.** Assuming that the chopped material leaves the impeller blades at about the peripheral speed of the impeller, its kinetic energy is represented by the familiar relation, $\text{K.E.} = Wv^2/2g$. Converting this to horsepower-hours per ton gives
where $E_{ke} = \text{kinetic energy, in horsepower-hours per ton.}$

$V = \text{peripheral speed of impeller, in feet per minute.}$

Note that this component also increases as the square of the peripheral speed.

15.15. Total Energy Requirements. In summarizing data from a large number of tests with ensilage cutters, Duffee has plotted the total energy requirements per ton for cutting and elevating corn silage as a function of the peripheral speed of the impeller blades. Most of his runs were made with feed rates between 10 and 25 tons per hr and a theoretical cut of about $\frac{1}{2}$ in. Results reported by Stewart and Berge agree remarkably well with Duffee's curve, as indicated in Fig. 15.9.

Energy requirements for chopping green alfalfa (74 per cent moisture content) with a $\frac{1}{2}$-in. theoretical cut were obtained by Blevins with three field choppers and peripheral speeds ranging from 4000 to 9000 fpm. His results indicate total energy requirements about 25 to 35 per cent greater than those represented by the curve for corn silage in Fig. 15.9.
In field chopping of cured hay (8 to 30 per cent moisture content and 3-in. theoretical cut), Berge obtained energy requirements of about 1 hp-hr per ton at a peripheral speed of 3600 fpm and $1\frac{1}{2}$ hp-hr per ton at 5000 to 6000 fpm. As indicated in Fig. 15.8, these power requirements would be increased considerably by making shorter cuts.

Since several of the power or energy components are proportional to the cube or the square of the peripheral speed, as discussed in the preceding sections, power requirements will be unnecessarily large if speeds are greater than needed to obtain the required height of elevation. Duffee found that, at a given peripheral speed, the total energy requirements per ton (for corn silage) were practically the same for any actual elevating height below about 60 ft and that there was only a moderate increase for greater heights (for example, 8 per cent increase between 10 and 75 ft). Thus, if the required elevating height for corn silage is 35 ft and the operating speed is 8000 fpm (sufficient for 100 ft elevation) instead of the 4400 fpm suggested by the dotted curve in Fig. 15.9, the total power requirement for a stationary chopper would be just twice as great as necessary. Although Duffee's findings with regard to speed and power requirements were published more than 25 years ago, the use of unnecessarily high speeds on both field choppers and forage blowers is still common.

**15.16. Capacity of Forage Choppers.** The capacity of a field chopper may be limited by the capacity of the feed mechanism, by the amount of power available, by the ability of the cutterhead to chop and handle the material, or by other factors. The maximum theoretical capacity of the feed mechanism is determined by:

1. The cross-sectional area of the throat opening (defined in Section 15.4).
2. The rate of advance of the material through the throat (generally considered to be the same as the linear speed of the feed mechanism).
3. The density of the forage as it passes between the feed rolls.

The actual maximum working capacity of the feed mechanism will ordinarily be somewhat less than the theoretical maximum because of the difficulty in maintaining a uniform feed rate. It has been suggested that rated capacities (at 100 per cent field
efficiency) be taken as 70 per cent of the theoretical maximum in the case of corn silage, and 60 per cent of the theoretical maximum when chopping hay.\(^2\)

The speed of the feed mechanism is related to the length of cut, as will be discussed in Section 15.17. The density factor is primarily a function of the type of material and its moisture content, but it is also influenced by the way the material is placed on the feed table and by the pressure of the feed rolls. Thus, the effective feeding density of a given material may not be the same as the storage density and will vary somewhat for different machines. It can best be determined by actual capacity tests with the machine or machines involved.

Duffee's results for stationary ensilage cutters indicate an average effective density of 20.7 lb per cu ft for corn (about 75 per cent moisture), but limited tests indicate that the value is somewhat higher for field choppers.\(^2\) When chopping hay at 26 per cent moisture, Berge\(^3\) found the effective density to be 3.22 lb per cu ft, but he suggests that a value of about 3.45 be assumed because maximum feed rates were not attained in the tests.

When chopping cured hay, the capacity of the feed mechanism is generally the limiting factor. But when chopping silage with present-day field choppers, the availability of power is likely to be the controlling factor, especially when cutterhead speeds are greater than actually needed for satisfactory loading. Theoretical maximum capacities for the feeding mechanisms on many of the current machines, when chopping corn with a theoretical cut of \(\frac{3}{8}\) in., are in the range from 50 to 75 tons per hr.\(^2\) Even at a peripheral speed of only 4500 fpm, the corresponding power requirements would be 35 to 50 hp. At 6000 fpm they would be 50 to 70 hp. On most field choppers, pto drives and auxiliary engines are optional, with typical ratings for the auxiliary engines being in the order of 30 to 40 hp.

15.17. Adjusting the Length of Cut. As previously mentioned, the theoretical length of cut is a function of the rotational speed of the cutterhead, the number of knives on the cutterhead, and the linear speed of the feed mechanism. The length of cut can be increased or decreased by changing any one (or more) of these three factors.

Decreasing the length of cut by increasing the speed of the
cutterhead results in unnecessarily large power requirements if the elevating requirements remain unchanged (unless the impeller-blower is separate from the cutterhead and its speed is not changed). Reducing the feed-mechanism speed to obtain a shorter cut reduces the capacity of the feed mechanism, as explained in the preceding section. Changing the number of knives does not affect either the feeding capacity or the kinetic-energy power requirements. But changing knives is a time-consuming and somewhat hazardous operation, and the selection of lengths obtainable by this method alone is limited.

Thus, it is evident that some compromise is often necessary or advisable to obtain the desired length of cut with a minimum of physical effort and without sacrificing capacity. Field choppers making short cuts are often operated at cutterhead speeds considerably greater than required for elevation, in order to increase their feeding capacity. The penalty is excessive power requirements, thus emphasizing again the importance of using the longest acceptable length of cut. For a given length of cut and a given feeding capacity, a six-knife cutterhead will require only two-thirds as great a peripheral speed as a four-knife unit. A cylinder-type cutterhead with the knives acting as impellers has the advantage of an inherently smaller diameter and correspondingly lower peripheral speeds.

FORAGE BLOWERS

15.18. Mechanical Features. Forage blowers are popular for handling chopped material because of their simplicity, dependability, and high capacities. Essentially, a forage blower consists of a feed hopper and conveyor that feeds the material into an impeller-blower similar to the units on forage choppers. If a forage blower is to be operated at full capacity (perhaps half a ton of green material per minute), mechanical unloading of the vehicle is a necessity. Many forage blowers can be equipped with an adjustable-speed attachment to drive the mechanical-unloading devices on wagons, thus eliminating the need for a separate power source.

The hopper is generally hinged and counterbalanced by springs so that it can be lifted to a vertical position while the wagon or truck is being moved forward into the unloading position. The
hopper must be long enough to extend across the full width of the rear of the vehicle, and it should be broad and low to minimize spilling during the unloading operation. Both augers and drag-type conveyors, usually with an adjustable-speed drive, are employed to feed the material into the blower. A counter-rotating leveling roll (or sometimes an adjustable gate) at the blower entry smooths off the upper surface of the advancing material and gives a more uniform feed rate. Callum\(^5\) recommends that the leveling roll be operated at a peripheral speed about five times that of the feed conveyor.

### 15.19. Power Requirements and Elevating Efficiencies

The preceding discussion pertaining to impeller-blowers on forage

![Graph](image_url)

**Fig. 15.10.** Total energy requirements for elevating corn silage with a forage blower. (Data from Sherwood S. DeForest. Unpublished thesis, Iowa State College, 1917.)

choppers can, in general, be applied directly to forage blowers. Total energy requirements for elevating corn silage with a forage blower having eight impeller blades and a 34-in. diameter wheel are indicated in Fig. 15.10. The bottom curve is for the same unit after removing four blades. The reduction in energy requirements per ton resulted partly from the reduced air flow with the smaller number of blades and partly because of a higher feed rate which distributed the air-moving power component over
a greater number of tons. Since most present-day forage blowers have either six or four blades, rather than eight, the lower curve is probably the more typical. The broken-line curve in Fig. 15.10 is included to give some comparison between the energy requirements for chopping and elevating and for elevating alone.

When a forage blower is putting material into a barn mow, the terminal velocity from the discharge pipe must be great enough to carry the hay across to the opposite side of the mow. It is generally considered that if a blower is capable of elevating to an actual height of 100 ft, the speed will be adequate for most conditions. According to Fig. 15.9, this maximum requirement would be met for corn silage by a peripheral speed of about 8000 fpm.

Operating speeds recommended by manufacturers ordinarily range from 7000 to 10,000 fpm or higher. Since all the important power components in a forage blower (air, friction, and kinetic energy) vary according to exponential functions of the peripheral speed, the waste of power resulting from unnecessarily high speeds is even greater than with forage choppers.

**15.20. Performance of Forage Blowers.** The performance of present-day forage blowers is generally satisfactory for short-cut material, but the recent trend toward long cuts for chopped hay has introduced serious problems in the operation of a forage blower. One of the problems is that blockage occurs in the impeller housing. Stroboscopic studies indicate that this blockage is caused by "hairpinning" of the long pieces over the sloping inner edge of the blades. The accumulated material tends to slide outward along the sloping edge. If it reaches the small-clearance section at the outer end of the sloping portion, it sometimes wedges so tightly between the housing and rotor that the rotor is stopped within a few revolutions. In other instances, the sudden release of a large mass of material collected on the blade will cause a stoppage in the discharge pipe.

**15.21. Material Flow Paths.** As mentioned in Section 15.7, the chopped material in a forage-harvester cutterhead is very rapidly accelerated to the angular velocity of the knife. The material entering a forage blower may not be accelerated quite as fast, since it is not acted upon so positively. In either case, centrifugal force immediately causes a radially outward acceleration of the particles, in addition to the rotational effect. If one
assumes that the entering particles are instantaneously accelerated to the angular velocity of the impeller and considers only the effect of centrifugal force on radial acceleration, the following expression for the flow path of a particle can be derived analytically: \(^\text{5,13}\)

\[
r = r_0 \cosh \theta
\]

(15.4)

where \(r_0\) = radial distance at which the particle enters the impeller-blower (point at which cutting occurs, in the case of a chopper with a flywheel-type cutterhead). The radial velocity at this point is zero, but the angular velocity is assumed to immediately become equal to that of the flywheel.

\(\theta\) = angle of rotation of the impeller beyond the point of particle entry at \(r_0\).

\[
\cosh \theta = \text{hyperbolic cosine} = \frac{1}{2}(e^\theta + e^{-\theta}), \text{ where } \theta \text{ is expressed in radians.}
\]

\(r = \) radial distance to particle at rotational angle \(\theta\) beyond the point of entry.

Note that the ratio \(r/r_0\) is independent of the rotational speed of the impeller-blower, the feed rate, and the characteristics of the material. If \(r\) is taken as the radius of the blower housing, \(\theta\) becomes the theoretical angle of rotation required for the particle to reach the periphery. This relation neglects the effects of air resistance, radial friction, and obstructions within the cutterhead, but does indicate a minimum required angle for a given amount of radial travel.

Theoretical, minimum-angle paths for various points of entry on a forage blower are plotted in Fig. 15.11. The diagram indicates that a considerable amount of material is carried past the outlet opening before it reaches the periphery of the housing, a situation substantiated by experimental observations.\(^6\) This material must be carried around again before it can be discharged, thus increasing friction losses and damage to the material. Particles from the upper right-hand side of the inlet strike the housing before the outlet is reached. Stroboscopic observations\(^7\) with long-cut hay indicate that this material is carried in front of the impellers in a relatively long and thin layer against the housing (usually not over \(\frac{3}{2}\) in. radial thickness).
For the best performance of a forage blower, the feed inlet must be positioned to minimize carry-over past the outlet and keep the lengths of flow paths at a minimum. Present designs can undoubtedly be improved in this respect.

In the case of a flywheel-type cutterhead on a forage chopper, the material is introduced about on the horizontal axis of the wheel, to the right of the center in Fig. 15.11. Most of the material then strikes the periphery before reaching the outlet opening. Stroboscopic observations indicate that material is carried against the housing through an angle of about 90° before being discharged. In tests with the lower half of the peripheral band removed, Blevins found that about 16 per cent of the material went up the discharge pipe and that practically none was carried past the outlet opening. In other words, this amount of material was carried through an angle of 180° or more before reaching the periphery. According to equation 15.4, the minimum angle required for all the material to reach the periphery, under the conditions of Blevins' tests, would be 135°.

Fig. 15.11. Theoretical flow paths for particles in a forage blower. (Charles L. Callum. Unpublished thesis. University of Wisconsin, 1951.)
REFERENCES


15.1. Duffee reports one test with a three-knife, flywheel-type cutterhead in which the angle between the plane of rotation and the knife faces on the shear-bar side was 2° 41' (knife bevel on the side away from shear bar), and the theoretical length of cut was 0.52 in. If the minimum distance from the wheel centerline to the throat was 6 in., would there theoretically have been any feed interference?

15.2. Compute the weight and volume (cubic inches) of corn silage carried on each impeller blade if a four-blade wheel is operated at 750 rpm and the feed rate is 17 tons per hr. Assume a density of 21 lb per cu ft.

15.3. Derive equation 15.2.

15.4. Derive equation 15.4.

15.5. At a feed rate of 600 lb per min for green alfalfa, and a peripheral speed of 6820 fpm, horsepower values read from the curves in Fig. 15.7 are as follows: curve $A = 9.5$ hp, $B = 13.5$, $C = 19.0$, $D = 22.3$, and $E = 25.2$. The power required for the pickup and feed mechanism at 300 lb per min is 45 cent of the requirement at 600 lb per min. For each of the following conditions, calculate the total power requirement, the total energy requirement in horsepower-hours per ton, and the distribution of component power requirements in percent of the total, tabulating your results:

(a) A feed rate of 600 lb per min and a peripheral speed of 6820 fpm.
(b) A feed rate of 300 lb per min and a peripheral speed of 6820 fpm.
(c) A feed rate of 600 lb per min and a peripheral speed of 4500 fpm.

15.6. (a) Neglecting air effects, what horizontal discharge velocity from the pipe of a field chopper is required to carry the chopped material 12 ft to the rear while it drops 2 ft?

(b) If the material leaves the impeller blades at 4400 fpm, what percentage of the kinetic energy at this speed has been lost by the time the material leaves the pipe at the velocity calculated in (a)?

15.7. A field chopper with a throat 14 in. wide and 5 in. high (flywheel-type cutterhead) is to be designed for a rated capacity of 25 tons of corn silage per hour ($\frac{3}{4}$ in. theoretical cut) at a peripheral speed of 5000 fpm. Consider the rated feeding capacity (at 100 percent field efficiency) to be 70 percent of the theoretical maximum, and assume an effective density of 21 lb per cu ft for the corn.

(a) Determine an appropriate wheel diameter and number of knives.
(b) Estimate the total power requirement at rated capacity.
(c) What would be the power requirement if the cutterhead were operated at a peripheral speed of 8000 fpm (which is common practice)?
CHAPTER 16

Seed Cleaning
and Sorting *

16.1. Introduction. The chief reasons for cleaning and sorting seeds are:

1. To remove foreign material such as chaff, broken straw, clods, stones, and seeds of other plants.
2. To obtain restricted size groups as an aid to precision planting.
3. To separate the seed into different density groups for controlling the germination of the final product.

Some physical difference must exist between the seeds and the foreign material or between different grades of seed before a mechanical separation can be made. The principal characteristics utilized in making separations are size, shape, specific gravity, surface characteristics, and color. Separations on the basis of color are highly specialized, usually involving electronic equipment, and will not be discussed in this book.

The operation of a mechanical separator often depends upon more than one type of characteristic difference. Separations made on the basis of size, shape, and specific gravity predominate. Machines utilizing size and shape characteristics for sorting seeds or separating them from foreign material may have screens or sieves, moving pockets, spiral cones, angle sieves, or inclined belts. Specific-gravity separations ordinarily involve an air blast, often in conjunction with some other mechanism.

16.2. Screens. Screens (sometimes referred to as sieves) are used more widely than any other type of device for sorting

* For a more thorough treatment of this subject, see Henderson and Perry. 4
granular products. An air blast in combination with screens adds a cleaning feature based upon specific gravity. This combination is known as a fanning mill (Fig. 16.1). A similar combination is found in the cleaning shoe of a threshing machine or combine (see Chapter 17). The screens in a fanning mill consist of

![Diagram of a fanning mill](image)

**Fig. 16.1.** Sectional view of a fanning mill. Mixtures to be separated are fed to the top screen through an aspirating air column. The bulk of the coarse material is collected at 1, additional material larger than the seed at 3, the desirable seed at 7, and the fine material at 4 and 8. Light seeds may be collected at 2, 5, and 6. (A. T. Ferrell & Co.)

framed, perforated metal sheets or wire cloth and have nonadjustable openings that are generally round, square, oblong, or triangular in shape. Sieves with adjustable openings are common in combines.

The screens are supported by hangers and oscillated by means of an eccentric, crank, or reaction drive, generally in the direction of material flow, but sometimes crosswise (known as side shake). The hanger mounting is such that the screens have a horizontal oscillating motion and a smaller vertical motion. This combination of motions thoroughly agitates the material and gradually moves it toward the point of discharge. The slope of the screen
is usually adjustable for controlling the rate of flow. Screens are sometimes fitted with brushes that move slowly from side to side beneath the screen and remove lodged material from the openings.

The function of the top screen (scalper) in an arrangement such as that shown in Fig. 16.1 is to remove material that is larger than the seed. Therefore, a size and shape of screen openings is selected to permit easy passage of the seeds. Round-hole screens are suitable for spherical-shaped seeds such as grain sorghums, vetch, and onions, whereas elongated holes may be preferable for seeds such as wheat, rye, and oats. The lower screen of each pair removes material that is smaller than the seed. When two sets of screens are in series, as in Fig. 16.1, the screens in the top set are selected to give a rough separation and remove the bulk of the large and small undesirable material, thus permitting a finer and more accurate separation to be made with the lower set.

Air flow may aid separation in a fanning mill in either of two ways. The seed may fall from a feeding device through an aspirating column (Fig. 16.1) to the top screen. The lighter material is drawn off, thus reducing the load on the screen and increasing its effectiveness. The final cleaning in a fanning mill is done by dropping the material through an air blast as it leaves the lower screen (Fig. 16.1). In the cleaning shoe of a combine, air is directed up through the sieves, removing much of the lighter material as the seeds fall through the openings and the remainder of the material is moved toward the rear of the sieve.

In difficult separations such as screening vetch (round) from oats (elongated), a sheet of smooth material such as oil cloth, floating on top of the mass of material on the scalper screen, will prevent the oats from up-ending and passing through the round holes selected for the vetch. A side-shake screen will give about the same effect. More effective methods for making this type of separation will be discussed in the next three sections.

16.3. Pocket-Type Separators. This type of equipment includes indented vertical disks, indented cylinders, and inclined belts with indents.

A disk separator (Fig. 16.2a) consists of a series of rotating disks having pockets on each side (Fig. 16.2b). Selection of seeds
is primarily on the basis of differences in length. The seeds picked up by the disks are carried over the top and discharged into trough-like receivers as shown at the left in Fig. 16.2a. The remaining material is propelled axially from one disk to the next by means of the blade-shaped spokes.

Disks with various sizes of pockets are available for different separations. When the separation involves only two kinds of seeds, all disks have the same size of pockets. Taking vetch and oats as an example, a size is selected that will permit the vetch seeds to enter but will reject the longer oat kernels. When several kinds of seeds of varying lengths are to be separated, the machine can be made up of groups of disks having different sizes of indentations. The first group contains disks with pockets that will remove the shortest seeds, and the last group is made up of disks with pockets that reject the longest seeds in the mixture. The grouping and selection of the intermediate disks depends upon the type and number of separations to be made.

The revolving-cylinder sorter (Fig. 16.2c) utilizes differences in seed widths, carrying the narrower seeds up out of the mixture in internal indentations. The position of the separator...
plate determines the degree of separation. The material caught by the separator plate is collected in a central receiver and conveyed along the axis of the cylinder. The rate of flow through the machine is controlled by the tilt of the cylinder.

16.4. Inclined Belts. Inclined belts with indentations represent another form of pocket-type separator. The indentations can be made to reject on the basis of either length or width. The rejected material is crowded off the belt by a diagonally mounted brush as the seeds to be recovered are carried up the slope and over the top in the indentations.

Smooth, inclined belts are often used for making separations based on shape, as, for example, the removal of leaves and twigs from fruit and nuts. Another common application is the separation of stemmy material from sugar beet seed. As the mixture is fed onto the belt midway up its length, the more-or-less spherical-shaped sugar beet seeds roll down the inclined surface while the stems remain on it to be carried up to the point of discharge.

16.5. Spiral Cone. This device is also known as a spiral gravity separator. Basically it consists of two concentric, helical-shaped spirals. Greater capacity is gained when the inside section contains a group of spirals in parallel, as shown in Fig. 16.3. This type of equipment is most effective in separating spherical-shaped seeds from elongated seeds (as in separating vetch from
oats). The mixture is fed to the inside spirals through a divider at the top. As the mixture flows down the spiral paths, the spherical-shaped seeds accelerate until enough centrifugal force is developed to cause them to roll over the edges of the inner spirals into the outer spiral. The elongated seeds never acquire enough velocity to cause them to leave the inner spirals. At the bottom the inside spirals all discharge into one container and the outside spiral discharges into another.

The spiral separator is quite effective in making separations of the type described above. It requires no power for operation, is inexpensive, and is simple, but has a relatively low capacity.

16.6. Angle Sieves. The angle sieve is another example of a device that separates on the basis of differences in length. It is commonly called a wild-oat kicker. The angle sieve (Fig. 16.4) permits short seeds to pass through the angle in the throat, while the oats are held back because of their length. The sieve is oscillated longitudinally with sufficient amplitude to move the oats from one section to the next, thereby giving the appearance of walking the oats out of the machine. Additional screens and an air blast are provided for recleaning the seeds that pass through the angle sieve.

16.7. Separation on the Basis of Specific Gravity. Representative specific gravities of a number of grains, as determined by Zink, are indicated in Table 16.1. Differences between the

Table 16.1 REPRESENTATIVE SPECIFIC GRAVITIES FOR SEVERAL KINDS OF GRAIN *

<table>
<thead>
<tr>
<th>Kind of Grain</th>
<th>Number of Varieties Included</th>
<th>Specific Gravity</th>
</tr>
</thead>
<tbody>
<tr>
<td>Barley</td>
<td>5</td>
<td>1.13–1.33</td>
</tr>
<tr>
<td>Buckwheat</td>
<td>1</td>
<td>1.10</td>
</tr>
<tr>
<td>Corn</td>
<td>1</td>
<td>1.19</td>
</tr>
<tr>
<td>Flax</td>
<td>1</td>
<td>1.10</td>
</tr>
<tr>
<td>Grain sorghum</td>
<td>2</td>
<td>1.22–1.26</td>
</tr>
<tr>
<td>Millet</td>
<td>1</td>
<td>1.11</td>
</tr>
<tr>
<td>Oats</td>
<td>4</td>
<td>0.95–1.06</td>
</tr>
<tr>
<td>Rice</td>
<td>2</td>
<td>1.12</td>
</tr>
<tr>
<td>Rye</td>
<td>1</td>
<td>1.23</td>
</tr>
<tr>
<td>Soybeans</td>
<td>2</td>
<td>1.13–1.18</td>
</tr>
<tr>
<td>Wheat</td>
<td>3</td>
<td>1.29–1.32</td>
</tr>
</tbody>
</table>

specific gravities of various kinds of seeds, as well as between those of seeds and other foreign material, can serve as a basis for separation. Often there is sufficient variation within the same lot of seed to permit separations within the sample.

16.8. The Specific Gravity Table. The separating unit on a specific gravity table consists of an oscillating, perforated deck (Fig. 16.5) covered with a material such as fabric, woven wire, or perforated metal. Provisions are made for elevating the deck at the rear and at the right side (Fig. 16.5b). Air from a centrifugal blower is directed up through the deck surface by a system of baffles. The air volume is controllable over a wide range by means of an iris-type gate in the main duct.

The deck is oscillated in such a manner that anything in direct contact with it will tend to be moved in the direction of conveyance (Fig. 16.5a). During normal operation the deck is elevated on the right side (Fig. 16.5b) and is tilted slightly downward from back to front. Proper tilt of the deck and proper adjustment of the air volume comes only with experience in operating the machine. Both will vary for different lots of seed.

Material fed onto the deck on the low side moves in the general direction of the broken lines (Fig. 16.5a), depending upon the setting of the machine and the relative densities of the different elements of the mixture. Air moving through the deck agitates the mixture and gives it the appearance of a boiling liquid. This action produces stratification of the material according to the specific gravity of the individual particles. The

Fig. 16.5. A specific gravity table. (a) Plan view of deck. (b) Front view, showing discharge collectors.
heavier seeds sink through the mass until they are in direct contact with the deck surface and are then moved in the direction of conveyance. The lighter materials lose contact (or have less contact) with the deck surface, with the result that they gravitate more rapidly toward the discharge edge.

The material leaving the deck is collected in a series of converging spouts along the discharge edge (Fig. 16.5b), the specific gravity of the various lots increasing from left to right.

The specific gravity of certain types of seeds may vary as much as 30 per cent within a given sample.† There is some evidence that germination within a given lot is related to the specific gravity of the individual seeds. Certain vegetable seeds, such as those of lettuce and carrots, are often graded into density groups that are subsequently blended as a control of the average germination of the final product. Table 16.2 gives the results of grading a sample of segmented sugar beet seed on a gravity table, in terms of the subsequent germination. The original germination of this lot was 79 per cent. By discarding the seed collected in spouts 4 and 5 (9 per cent of the sample) the germination of the remaining seed was raised to 83 per cent. Further upgrading could be accomplished by rerunning the increment caught in spout 3 and discarding the lighter material separated in the second run.

Table 16.2 RELATION OF GERMINATION OF SEGMENTED SUGAR BEET SEED TO SEPARATION OBTAINED WITH A SPECIFIC GRAVITY TABLE *

<table>
<thead>
<tr>
<th>Spout Number</th>
<th>Per Cent of Sample</th>
<th>Per Cent Germination</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>31.0</td>
<td>92.0</td>
</tr>
<tr>
<td>2</td>
<td>34.6</td>
<td>83.0</td>
</tr>
<tr>
<td>3</td>
<td>25.4</td>
<td>72.0</td>
</tr>
<tr>
<td>4</td>
<td>6.1</td>
<td>36.5</td>
</tr>
<tr>
<td>5</td>
<td>2.9</td>
<td>36.0</td>
</tr>
</tbody>
</table>


16.9. Aspirating Column. A well-designed aspirating column is capable of producing rather accurate separations based

† Unpublished data collected by Roy Bainer.
upon a combination of differences in specific gravity, size, and shape of the particles. The principle of operation is to introduce the material into a column of air that is moving vertically upward at a uniform, controlled velocity. Seeds (and other particles) that are too heavy to be picked up by the air stream fall into a container directly below the column inlet. Lighter seeds and trash are lifted by the air stream and conveyed to a separating cyclone.

The vertical column should be round and have a properly shaped entry nozzle to give a uniform velocity distribution across the column. In one model (Fig. 16.6) the seed is introduced from a concentric hopper, through an adjustable, annular opening in the column wall. The air is drawn from the column to the inlet of a blower. The separating cyclone should be between the aspirating column and the blower, so that material carried over in the air stream does not have to pass through the blower. The air velocity in the column is controlled by a suitable valve between the aspirator and the cyclone.

Comparative tests were made with a gravity table and the aspirating column illustrated in Fig. 16.6, using decorticated sugar beet seed.* When the machines were set to give the same percentage recovery of seed, there was no significant difference in the germination from the two separations.

In separating or moving different materials by air, the size and shape of the particles must be considered in addition to the specific gravity. Referring to Table 16.1, it will be noted that typical specific gravities for wheat, corn, and oats are about 1.3, 1.2, and 1.0, respectively. On the basis of specific gravity alone, it would be expected that the air velocity to lift corn would be somewhere between those required to lift wheat and oats. How-

* Unpublished data collected by Roy Bainer.
ever, Brown and Reed found that the velocity required to lift corn was greater than that required to lift wheat (Fig. 16.7), presumably because of the greater mass of each seed and the flat nature of the seeds.

16.10. Sorting on the Basis of Surface Characteristics. An example of the utilization of this principle is in the separation of beans from small adobe clods that are similar in size, shape, and specific gravity. A unique machine developed at the California Agricultural Experiment Station (available commercially) makes this separation on the basis of differences in surface characteristics, the beans being smooth and the clods being rough. Basically, the machine consists of a rough, saw-tooth type of surface and a soft bristle brush (Fig. 16.8). A mixture of beans and clods is deposited at one end. As the brush moves against the points of the rough surface (Fig. 16.8a), the beans are swept over the points ahead of the brush while the rough clods are held by the points so that the bristles of the brush pass around them. When the brush reaches the end of the surface the beans fall into a receiver. On the return trip (Fig. 16.8b) the clods are swept ahead of the brush because the surface offers less resistance in this direction.

In the commercial model, the brush is stationary and the rough surface (similar to that on a fluted reamer) is on a horizontal cylinder. A full-length slot through the cylinder provides a
passageway (after each one-half revolution) for dropping the beans held back by the brush. The rough surface of the cylinder moves the clods through the brush, after which they fall away from the cylinder by gravity.

The difference in surface characteristics between hulled and unhulled rice is used in rice mills for separating the two. The hulling process in a mill is never completed in one operation, it being necessary to separate and return unhulled kernels to the huller. Separation is accomplished on a vibrating, inclined, rough surface that moves the rough, unhulled kernels laterally out of the normal flow of hulled rice.

REFERENCES

CHAPTER 17

Grain and Seed Harvesting

17.1. Introduction. Most grain and seed crops are now harvested with combined harvester-threshers, commonly known as grain combines. Consequently, the following discussion pertains primarily to this type of machine. Except for differences in the feeding arrangement and the addition of a straw stacker, stationary threshers employ the same principles and include the same basic components as combines.

Although the greatest application of combines is in harvesting the small grains and soybeans, these machines are also used for a wide variety of small-acreage or speciality crops. Thus, although most emphasis in the following discussion will be placed upon grain harvesting, special considerations relating to other seed crops will be mentioned from time to time. The relatively new practice of harvesting corn with modified grain combines is covered in Chapter 18, along with the discussion of specialized corn-harvesting equipment.

17.2. Harvesting and Threshing Methods. For many centuries the procedure for harvesting grain involved hand cutting with a sickle, scythe, or cradle, threshing by the treading of animals or by flailing, and separating the threshed grain from the straw and chaff by hand raking, sieving, and winnowing.

The systems followed in mechanically harvesting grain (and other seed crops) include (a) direct combining, (b) windrowing and combining, (c) binding or heading and stacking, followed by threshing in a stationary machine, and (d) windrowing, picking up the windrows with a field chopper, and threshing in a stationary machine.

Direct combining and windrow combining require the least
amount of labor and, in the United States, have largely replaced stationary threshing methods. In 1950, for example, 85 to 95 per cent of the barley, wheat, and soybean crops in the United States were harvested with combines. The windrow-combine method involves an extra operation as compared with direct combining but is advantageous under certain conditions. Windrowing permits the curing of green weeds and unevenly ripened crops before threshing. The weather hazard to the standing crop is reduced because windrowing can be started several days earlier than direct combining. In tests conducted in Ohio, windrows on grain stubble 9 to 12 in. tall cured more rapidly than standing or shocked grain, regardless of whether or not rain occurred. Heavy vegetative crops, such as alfalfa grown for seed, are often harvested by the windrow-combine method. In areas having a hot, dry climate, the system of spray curing followed by direct combining is being practiced to an increasing extent for small-seed legumes such as alfalfa and clover. In this method, general-contact herbicides are applied to kill the top growth. The crop is then direct combined after the leaves are dry but before regrowth starts (usually from 1 to 5 days after treatment). Spray curing is most effective when the stands are uniformly mature, open, and erect.

Stationary threshers are still used to some extent where the fields are small and the conditions not well suited to combine operation. Approximately one-fourth of the 1950 oat and rye crops in the United States were threshed with stationary machines. Binding or heading the grain prior to threshing provides the advantages of curing of green material and reduction of weather hazards (as with windrowing) but involves a considerable amount of labor. The stationary thresher accumulates the straw in a stack, which is an advantage if the straw is to be saved for future use.

The windrow-field-chopper method, in conjunction with stationary threshing, is a recent development in the Wisconsin area. After the windrowed material has cured, a field chopper picks up the grain, chops it (using a long cut), and deposits it in a wagon. By means of a mechanical unloading arrangement, the chopped material is then fed directly from the wagon into a stationary thresher. This system is most feasible where all of
the following conditions exist: (a) the grain should be windrowed because of green material, uneven ripening, etc., (b) the terrain is rolling and not too good for combining, (c) a forage harvester and wagons are already available because of other applications, and (d) the farmer wants to save his straw. Labor requirements are lower than for the binder-thresher method.

17.3. Types and Development of Combines. Although combines were rather common in California as early as 1885, the large-scale adoption of these machines in the wheat belt of the Great Plains area did not occur until the 1920's. The early combines were all large, trailed units (mostly 12 to 16 ft cut, with some as wide as 35 ft or more), and were not adaptable to the small farms of the eastern part of the United States. Small, trailed combines (mostly 5 to 7 ft cut but some as small as 3½ ft) were introduced commercially in 1935. Since that time, the combine has become nationally accepted. More than 95 per cent of the trailed combines built in 1952 and 1953 had widths of cut less than 12 ft.

Self-propelled, flat-land combines appeared commercially in about 1939. These machines ordinarily have widths of cut ranging from 9 to 16 ft and have the header mounted across the front of the combine (Figs. 17.1 and 17.3). Their popularity has increased rapidly in recent years. In 1952 and 1953, over 60 per cent of the domestic shipments of combines larger than 6-ft cut were self propelled (although self-propelled machines represented only 21 per cent of all combines of all sizes).

As compared with trailed machines, self-propelled combines have the advantages of (a) saving of grain in opening up fields surrounded by fences, other crops, levees, etc., (b) greater flexibility and better maneuverability, and (c) better visibility and control by the operator. Power steering is available on many of the current models. The first cost of self-propelled machines is considerably greater, however, and accessibility for making repairs and adjustments is often sacrificed in comparison with trailed machines.

The first trailed, hillside combine was built in the late 1880's. Combines of this type have provision for shifting one or both of the main wheels vertically, to keep the threshing and separating assembly level when harvesting along the contour on steep slopes. The header, hinged to the body of the combine and supported at
the outer end by an outrigger wheel, remains parallel with the surface of the field. Leveling of present-day trailed machines is done through manually controlled, power-driven mechanisms. Self-propelled hillside combines were introduced in the early 1950's. The header platform is attached to the feeder housing through a swivel joint, as indicated in Fig. 17.1. A linkage between the platform and the main axle assembly automatically keeps the platform parallel with the plane of the ground beneath the main wheels. The leveling action is accomplished by means of hydraulic cylinders and can compensate for maximum slopes of about 35 to 40 per cent toward either side.

Although some self-propelled machines have manual control, the trend is toward fully-automatic leveling. One automatic system has a heavy, damped pendulum that actuates spool-type hydraulic control valves of the general type illustrated in Fig. 5.3 (Chapter 5). The valve design must be such that only a small amount of spool movement is required to cause oil flow to or from the hydraulic cylinders. On another machine, a fluid reservoir exerts pressure on a sensitive pressure switch located to one side of the reservoir. If the body of the combine starts

Fig. 17.1. An automatic-leveling, self-propelled hillside combine. Note the oversize tires. (Deere & Co.)
to tilt, the reservoir is raised or lowered with respect to the diaphragm. This change of pressure actuates the pressure switch, which in turn operates solenoid valves in the hydraulic system and levels the machine.

Most hillside combines can be leveled only for side slopes. At least one self-propelled unit, however, is automatically leveled in the fore-and-aft direction as well. One trailed model has a cleaning unit that is pivotally mounted at the front and is automatically kept level in the fore-and-aft direction by means of direct linkage to a heavy pendulum. The principal use of hillside combines is in the Pacific Northwest.

17.4. Functional Elements of a Combine. The ultimate purpose of any harvesting and threshing operation is to recover the seed, free from plant residue, with (a) a minimum of seed loss, (b) a minimum of external damage, and (c) a minimum of internal injury, if the seed is to be used for planting. The five basic operations performed by a combine in recovering the seed are:

1. Cutting (or picking up from the windrow).
2. Conveying and feeding the cut material to the threshing mechanism.
3. Threshing or removal of the seed from the head or pod.
4. Separating the seed and chaff from the straw.
5. Cleaning the chaff and other debris from the seed.

The functional elements of a combine are indicated in Fig. 17.2, which shows a cut-away view of a small, trailed combine having a 6-ft cut. The machine illustrated (widths of cut for current models of this type range from 5 to 7 ft) is known as a straight-through combine because the cut material is fed directly into the cylinder without any cross-conveying. This simple arrangement is possible because the length of the threshing cylinder (and usually the width of the separating unit) are nearly as great as the width of cut. The cleaning unit is generally somewhat narrower. Some of the small combines have a full-width, inclined feeder canvas as in Fig. 17.2 but are not strictly of the straight-through type in that a short cylinder is employed, with a cross-conveying auger at the top of the feeder canvas.

Figures 17.1 and 17.3 illustrate platform-type headers on self-propelled machines. A header platform with a cross-conveyor is
needed on a combine (trailed or self-propelled) when the width of cut is greater than about 6 or 7 ft, because the length of cylinder required to give a straight-through arrangement would be impractical. Platform-type combines have shorter cylinders and narrower separators than do straight-through machines, because when a cross-conveyor is used it is difficult to get uniform feed distribution across a long cylinder.

![Diagram of a combine](image)

**Fig. 17.2.** Cut-away view of a trailed, straight-through type of combine. (International Harvester Co.)

**17.5. Flow Path of Material.** Although each of the basic functions is discussed in detail in later sections, it should be helpful to the reader to first obtain a general, over-all picture of the entire sequence of operations. In direct combining, the reel pushes the standing stalks against the cutter bar and then delivers the cut material back onto the header platform (Fig. 17.1) or directly onto the inclined feeder canvas (Fig. 17.2). In Fig. 17.1 the platform cross-conveyor (auger, in this case) delivers the material to the feeder conveyor (usually chain-and-slat type). The feeder canvas or feeder conveyor elevates the material and feeds it into the cylinder-and-concave assembly where the threshing and much of the separation take place. The
seed and chaff separated by the concave grate (and concave-extension grate, Fig. 17.4, if one is provided) fall directly onto the oscillating grain pan or onto a conveyor.

The cylinder beater (Fig. 17.2) tends to strip the threshed material from the cylinder, aids in further separation at this point, and directs the straw and remaining seed onto the straw carrier (oscillating rack or rotary walkers). The check curtain (or curtains) prevents threshed seed from being thrown out of the rear of the machine by the beater or the cylinder. The straw carrier agitates the material to separate out any remaining seed and unthreshed heads as the straw is moved rearward to be discharged from the machine. The material separated from the straw is collected by the grain return pans (or a conveyor) and delivered to the grain pan at the front of the chaffer sieve.

The mixture of threshed seed, unthreshed heads or pods, chaff, and other small debris is moved from the grain pan onto the front of the oscillating chaffer sieve. As the mixture is moved rearward over the chaffer sieve, an air blast directed upward through the sieves aids in separating out the free (threshed) seed and unthreshed heads and blows the light chaff out the rear of the machine. Most of the unthreshed heads ride over the chaffer sieve and drop through the larger openings of the chaffer extension into the tailings auger. The tailings are then returned to the cylinder for rethreshing or sometimes, if the seed is easily damaged, to a point just behind the cylinder.

The free seed falls through the chaffer sieve and is further cleaned by passing through the shoe sieve, which has smaller openings. The cleaned seed is then delivered to the grain tank by means of the clean-grain auger and elevator. Sometimes, when bulk handling of the seed is not convenient, the grain tank is replaced by a sacking platform. Two seed delivery spouts are provided in an inverted Y arrangement that has a movable gate hinged at the junction of the two legs. When the sack on one spout is full, the gate is rotated by hand to close off this spout and direct the seed into the empty sack on the other spout. The filled sacks, after being sewed, are deposited on an inclined chute or trough and then discharged onto the ground in groups of several sacks each.

17.6. Types and Sources of Seed Loss. Seed losses from a combine can occur in connection with any of the five basic opera-
tions listed in Section 17.4. These losses can be described as header, cylinder, rack, and shoe losses. Header losses include heads (or pods) and free seed lost during the cutting and conveying operations or during the pickup and conveying operations in windrow combining. Cylinder losses are represented by unthreshed seed discharged from the rear of the machine, either in the straw or in the material from the cleaning unit. Rack losses consist of free (threshed) seed carried over the rack or walkers in the straw and discharged from the machine. Shoe losses include free seed carried over the shoe (cleaning unit) in a blanket of chaff or blown over by an excessive air blast.

Seed damage does not represent a direct loss of yield (except for seeds broken into pieces too small for recovery), but it may reduce the quality and value of the product, depending upon the intended use for the seed or grain. Thus, the presence of cracked kernels does not reduce the feeding quality of grain, but it is undesirable in malting barley or in rice that is to be milled. Seed damage, as it affects germination, is an important consideration when the harvested product is intended for planting. With some kinds of seed, such as lima beans, germination may be reduced by internal damage even though there is no visible evidence of damage. As will be pointed out later, most seed damage occurs in the threshing unit.

17.7. Cutting and Conveying. The cutting and conveying assembly, known as the header, includes the reel, the cutter bar, a platform or conveyor for receiving the cut material, and conveyors for delivering the material to the cylinder. The header is attached through a hinge axis and is adjustable to obtain heights of cut ranging from 1 1/2 or 2 in. up to 3 ft or more on some machines. The grain is ordinarily cut just low enough to recover all or nearly all the heads. If the straw is to be saved, cutting may be at a lower height even though more material must be handled by the machine.

The cutter bar is similar in construction to that of a mower (see Chapter 14), but the ledger plates are usually smooth and the knife sections serrated on the upper surface. In some instances the guard surfaces are hardened instead of providing replaceable ledger plates. The operating conditions and the nature of the material cut with a combine are such that knife speeds can be considerably lower than for a mower. Speeds in
the range from 400 to 450 rpm (800 to 900 strokes per minute) are typical. The top-serrated knife ordinarily is not resharpened, but sections may be replaced occasionally.

The most common type of reel has four to eight slats (bats) mounted rigidly on radial arms (Figs. 17.1 and 17.2). In some cases the reel is ground driven, but power drives with interchangeable sprockets are most common. The peripheral speed of the reel should ordinarily be 25 to 50 per cent greater than the forward speed of the machine. Faster speeds are more effective in laying the cut material back onto the platform but may result in excessive shatter losses. The position of the reel with respect to the cutter bar is adjustable both vertically and horizontally to accommodate different crop conditions. The height normally should be such that the bottom edges of the slats, at the lowest point of their travel, are a little below the lowest heads of the uncut grain. The axis of the reel is usually from 6 to 12 in. ahead of the cutter bar.

Figure 17.3 shows a special type of unit known as a pickup reel, that is very effective in picking up lodged crops ahead of the cutter bar. An eccentric-and-spider arrangement maintains
the slats in parallel positions. The slat pitch should be adjusted so the spring teeth point slightly rearward, and the peripheral speed should be 10 to 25 per cent greater than the forward speed of the machine. This causes the teeth to hook into the mat of lodged material and lift it as the cutter bar passes beneath it. The center of the reel should be 9 to 12 in. ahead of the cutter bar. In a badly lodged crop the teeth should clear the cutter bar by only 2 or 3 in.

In a straight-through type of combine (Fig. 17.2), an inclined conveyor carries the cut material directly from the cutter bar to the cylinder. The larger, trailed combines have header platforms extending to one side of the threshing units, whereas most self-propelled machines have platforms across the front of the combine. Canvas drapers or large augers are generally employed to convey the cut material to the end or center of the platform. A separate feeder conveyor is ordinarily provided to elevate the material from the platform to the cylinder. In some special applications (as for heavy rice crops) auger-type platforms have canvas cross-conveyors mounted in front of the auger to provide more platform area for the collection of cut material.

Most combines have a beater (Fig. 17.2) or a short canvas draper (Fig. 17.5) mounted above or behind the upper end of the elevating conveyor to assist in feeding the material uniformly into the restricted area between the cylinder and concave.

17.8. Types of Threshing Cylinders. Removal of seeds from the heads or pods is ordinarily accomplished with rotating cylinders whose threshing action depends primarily upon impact. When the relatively slow-moving material comes in contact with the high-speed cylinder (peripheral speed of 5000 to 6000 fpm for wheat), the impact shatters the heads or pods and frees a considerable portion of the seed from the straw. Further threshing is obtained by the rubbing action as the material is accelerated and passes through the restricted clearance space between the cylinder and the concave.

The three common types of cylinders are illustrated in Figs. 17.4 and 17.5. The arrangement of the spike-tooth cylinder and concave (Fig. 17.4a) is such that the cylinder teeth pass midway between staggered teeth on the concave, thus producing a combing action. The concave assembly is pivoted at the rear, whereas the front is adjustable vertically to control the amount
of overlap of the cylinder and concave teeth. The lateral clearance is also changed slightly by this adjustment, since the teeth are tapered (Fig. 17.4b). The concave assembly is adjusted laterally to give equal clearances on both sides of the cylinder teeth. The teeth in the concave are mounted on perforated, re-

Fig. 17.4. Two types of cylinders, showing the concave-and-grate assembly and a finger-grate type of concave extension. (a) Spike-tooth cylinder. Note that the front and rear concave sections (removable) are each shown with two rows of teeth, whereas the middle section is blank (no teeth). Inset (b) shows the clearance relations between the teeth in the spike-tooth cylinder and those in the concave. (c) Rasp-bar cylinder. In this particular unit the concave bars are built into the concave grate. (J. I. Case Co.)

movable sections, usually with two rows of teeth per section. The total number of rows of teeth needed in the concave (usually two, four, or six) depends upon the crop and the threshing conditions. Teeth with corrugated sides are sometimes installed for difficult threshing conditions.

In a rasp-bar cylinder (Fig. 17.4c), threshing is done between corrugated cylinder bars and stationary bars built into or attached to the concave grate. Corrugated bars are occasionally used on the concave as well as on the cylinder. A special type of rasp-bar cylinder, known as a V-bar cylinder, has V-shaped
rasp bars rather than straight ones. The bars are attached to the cylinder so that the points of the V are leading as the cylinder rotates. The two ends of each V-bar are about in line with the point of the following bar.

The angle-bar cylinder (Fig. 17.5) is equipped with helically mounted, rubber-faced, angle-iron bars rather than corrugated bars. The concave is ordinarily fitted with a rubber-faced "shelling plate" and steel-jacketed rubber bars, as indicated in Fig. 17.5.

Fig. 17.5. An angle-bar cylinder and concave arrangement. Note the variable-speed, V-belt drive for the cylinder. (Allis-Chalmers Mfg. Co.)

There is considerable difference of opinion among operators and manufacturers regarding the relative merits of the different types of cylinders, and very few published data are available to provide a basis for comparison. Until about 1930, the spike-tooth cylinder was used almost exclusively. Since then the trend has been toward the other two types, with the rasp-bar construction predominating. The rasp-bar and angle-bar types are readily adaptable to a wide variety of crop and threshing conditions, have less tendency to break up the straw, and are more tolerant of green growth (weeds). The spike-tooth cylinder has a more positive feeding action and is preferred in some areas, particularly for heavy rice crops and for windrowed beans.

The long cylinders in straight-through combines are all of the rasp-bar or angle-bar types, with lengths in current models ranging from 48 to 64 in. and diameters generally being 15 to 16 in.
All three types are found in the narrow-body, short-cylinder machines. Cylinder lengths range from about 24 to 48 in. but are mostly between 28 and 32 in. Diameters generally range from about 19 to 22 in.

17.9. Rubber Rolls for Threshing. With some podded crops, a large part of the threshing can be done with a squeezing action, a rubbing action, or a combination of the two. Rubber-covered steel rolls have been used to a limited extent for threshing beans. An experimental bean combine developed at the California Agricultural Experiment Station had a series of three pairs of rubber-covered rolls, operating at peripheral speeds of 250 to 300 fpm, instead of a conventional cylinder. Threshing in this machine was entirely due to squeezing and rubbing of the bean pods. Satisfactory threshing was obtained without appreciable seed damage.

A combination of one rubber-covered roll and one steel roll (known as flax rolls) is sometimes installed just ahead of the conventional cylinder when harvesting podded crops such as flax and alfalfa. The upper roll is spring loaded and is driven about 10 per cent faster than the lower roll, to give a rubbing action in addition to the squeezing effect. Adjustable stops on the spring-loaded roll are set to provide a minimum clearance sufficient for the passage of the seeds without damage. Peripheral speeds are less than 500 fpm. In addition to their threshing action, flax rolls tend to hold the material against the pulling action of the cylinder, thus promoting more uniform feeding.

17.10. Threshing Effectiveness. Thoroughness of threshing is related to (a) the peripheral speed of the cylinder, (b) the cylinder-concave clearance, (c) the number of rows of concave teeth, in the case of a spike-tooth cylinder, (d) the type of crop, (e) the condition of the crop in terms of moisture content, maturity, etc., and (f) the rate at which material is fed into the machine. Covering the concave openings, so that the heads or pods must pass over all the concave bars, is sometimes helpful. Under certain conditions the type of cylinder may have some effect, although there is no conclusive evidence that any one type is consistently superior to the others.

High cylinder speeds and close clearances favor thoroughness of threshing. The actual requirements vary with different crops and often change during the day as weather conditions change.
Early-morning threshing, following a dew, may call for a higher cylinder speed or smaller clearance than is needed later in the day. The effect of moisture content upon threshability of Ladino clover (a crop rather difficult to thresh) is indicated by the results of laboratory tests under controlled conditions. As the moisture content was reduced from 16 to 9 per cent, the percentage of thresh (one time through the cylinder) increased from about 55 to 80 per cent.

![Graph](image)

**Fig. 17.6.** Effect of feed rate upon loss of unthreshed seed. Results are for a 12-ft self-propelled combine, operated in the Sacramento Valley, California. (Data from California Agricultural Experiment Station, 1951–1954.)

The general effect of feed rate * upon the loss of unthreshed seed from the rear of the machine (cylinder loss) is illustrated in Fig. 17.6. Note that, under these conditions, the per cent loss was nearly proportional to the feed rate. These results are not a direct measure of the thoroughness of threshing, because some of the unthreshed seed from the cylinder is recovered in the tailings and returned for rethreshing. Thus the net loss of unthreshed seed is influenced to some extent by the effectiveness of the separating and cleaning units.

*The term, feed rate, as used in this chapter, includes all plant material except the seed. It is the sum of the discharge rates from the straw rack and from the shoe, less the total free and unthreshed seed losses over the rear.
17.11. Seed Damage. Although seed damage can occur because of the cylinder-concave clearance being too small to permit free passage of the seeds, this is not ordinarily a controlling factor. In most cases the damage occurs primarily because of impact blows received in the threshing process and is directly related to the cylinder peripheral speed. The damage may be visible or it may be internal, the latter type being determinable only by germination tests. The susceptibility of different kinds of seeds to impact damage varies widely. The seeds of some dicotyledonous plants (such as beans) are extremely susceptible to this type of damage, whereas Ladino clover can withstand peripheral speeds of over 7000 fpm without appreciable damage.

Results obtained when threshing lima beans with a spike-tooth cylinder at various peripheral speeds are presented in Fig. 17.7. As the speed was increased the damage was also increased, but the unthreshed-seed loss was reduced. Note that at each speed

Fig. 17.7. Relation of seed damage and threshing losses to cylinder speed (spike-tooth) for Wilbur baby lima beans having a moisture content of 16.4 per cent. (Roy Bainer and H. A. Borthwick. California Agr. Expt. Sta. Bull. 580.)
the per cent internal damage exceeded the per cent visible damage. Maximum recovery of undamaged beans was at 940 fpm. The results of laboratory tests, in which lima beans were dropped from various heights onto a steel block, substantiated the premise that the seed damage was primarily due to impact. These laboratory tests, as well as other field tests, also indicate that

![Graph showing the effect of cylinder speed upon damage to alfalfa seed](image)

**Fig. 17.8.** Effect of cylinder speed upon damage to alfalfa seed, as determined by germination tests. (L. G. Jones, R. A. Kepner, Roy Bainer, and J. P. Fairbank. *California Agriculture*, August, 1950.)

the amount of damage at a given peripheral speed increases rapidly as the moisture content of the beans is reduced. To increase the recovery of undamaged beans, especially for seed crops, special bean threshers have been built with two spike-tooth cylinders in series. The first cylinder is operated at a peripheral speed of perhaps 1000 to 1300 fpm and removes most of the seeds without excessive damage. The second cylinder is operated somewhat faster to remove the beans left by the first cylinder.

Figure 17.8 shows the effect of cylinder speed upon damage to alfalfa seed when harvested under the dry conditions found in the interior valleys of California. Results for 16 different combines are included. With the exception of the two direct-combin-
ing runs noted on the graph, these tests were all made when com-
brining from the windrow, with seed moisture contents ranging
from 5.2 to 6.8 per cent. Cylinder speeds of 4500 to 5000 fpm
were required for satisfactory threshing under the conditions of
these tests. When flax rolls were added, satisfactory threshing
was obtained at a cylinder speed of 4000 fpm, with correspond-
ingly less damage. Other tests indicate that, as with beans, seed
damage is reduced as the seed moisture content is increased.

The results of tests with barley in Minnesota show the rela-
tion of visible damage to peripheral speed (Fig. 17.9). They also
indicate that, at a given peripheral speed, the per cent visible
seed damage was reduced by having rather large cylinder-concave
clearances. This effect is probably due in part, at least, to the
greater cushioning effect of the thicker layer of material. Bun-
nelle and his associates report a similar effect with Kenland
red clover and alfalfa, in that seed damage was reduced as the
feed rate was increased, presumably because of greater cushion-
ing and protection for the seed.

17.12. Cylinder Adjustment. The method of adjusting cyl-
inder-concave clearances varies with different machines. In
some cases the cylinder is raised or lowered, whereas in others
the height of the concave assembly is adjustable. Where the
heights of the front and rear of the concave can be adjusted inde-
pendently, the clearance is generally made a little greater at the
front than at the rear. Clearances should be as great as can be
used and still obtain satisfactory threshing. Close clearances
result in more breakup of the straw, thus decreasing the effective-
ness of the separating and cleaning units.

Cylinder speeds may be changed by means of adjustable-pitch
V-belt sheaves (Fig. 17.5) or by changing sheaves or sprockets.
It should be evident from the preceding discussion that some
compromise is often necessary to obtain the best cylinder speed.
With some crops, such as lima beans (Fig. 17.7) and Kenland
red clover, a cylinder speed high enough to do a good job of
threshing may cause excessive seed damage. High speeds also
break up the straw more.

Typical peripheral speeds and clearances for various crops,
based for the most part upon a summary of recommendations
found in operator's manuals published by the various manufac-
turers, are indicated in Table 17.1. By considering peripheral
Fig. 17.9. Effect of cylinder speed and clearance upon visible damage to barley having a moisture content of 12 to 15 per cent. (H. H. DeLong and A. J. Schwantes. Agr. Eng., March, 1942.)
Table 17.1  TYPICAL CYLINDER PERIPHERAL SPEEDS AND CLEARANCES FOR VARIOUS CROPS

<table>
<thead>
<tr>
<th>Crop</th>
<th>Peripheral Speed, feet per minute</th>
<th>Clearance for Rasp-Bar Cylinder, inches *</th>
</tr>
</thead>
<tbody>
<tr>
<td>Clovers</td>
<td>5500-6500</td>
<td>1/16-1/4</td>
</tr>
<tr>
<td>Flax</td>
<td>5000-6000</td>
<td>1/16-1/4</td>
</tr>
<tr>
<td>Oats, rye, wheat</td>
<td>5000-6000</td>
<td>3/16-5/8</td>
</tr>
<tr>
<td>Alfalfa †</td>
<td>4000-6000</td>
<td>1/16-5/16</td>
</tr>
<tr>
<td>Barley</td>
<td>4500-5500</td>
<td>1/4-3/4</td>
</tr>
<tr>
<td>Grain sorghum</td>
<td>4000-5000</td>
<td>1/4-5/8</td>
</tr>
<tr>
<td>Rice</td>
<td>4000-5000</td>
<td>3/16-3/4</td>
</tr>
<tr>
<td>Soybeans</td>
<td>2500-3500</td>
<td>5/16-7/8</td>
</tr>
<tr>
<td>Peas</td>
<td>2000-3000</td>
<td>5/16-3/4</td>
</tr>
<tr>
<td>Edible beans</td>
<td>1500-2500</td>
<td>5/16-3/4</td>
</tr>
<tr>
<td>Seed beans ‡</td>
<td>1000-1500</td>
<td>1/16-3/4</td>
</tr>
</tbody>
</table>

* Clearances recommended for angle-bar cylinders are generally a little greater than for rasp-bar cylinders.

† Operating instructions from the various manufacturers indicate speeds of 5500 to 6500 fpm and clearances from 1/16 to 5/16 in., presumably for Midwest conditions. In dry climates, clearances should be greater and speeds must be kept below about 5000 fpm to prevent excessive seed damage.6,14

‡ Based on data in reference 2.

speeds rather than rotational speeds (rpm), the effect of cylinder diameter is eliminated.

17.13. Separating. Under normal operating conditions, a large portion of the threshed seed (along with some chaff) is separated from the straw at the threshing unit, falling through openings in the concave-and-grate assembly or through the concave-extension grate (finger grate). Separation of the remaining free seed and unthreshed seed takes place on the straw carrier as the straw is agitated and moved to the rear of the machine.

The most common types of straw carriers are the one-piece oscillating straw rack (Fig. 17.2) and the multiple-section straw walker. In the latter arrangement, three or four narrow sections are placed side by side in the machine. Each section is attached to a multiple-throw crankshaft at the front and another at the rear. The crank throws for the individual sections are equally spaced around the circle of rotation (i.e., 120° or 90° apart).

The action of the straw rack (or straw walkers) accelerates
the straw in a rearward and upward direction during a portion of the cycle. While returning to the forward position, the rack tends to leave the straw momentarily in midair. The material then falls onto a section of the rack nearer to the discharge end and is moved another step toward the rear by the next stroke of the rack. It is this tossing and agitation that is responsible for the separating action. Crank speeds for straw racks or walkers are in the order of 200 to 250 rpm. If the speed is either too high or too low, seed losses may be increased.

In the larger combines (over 7-ft cut) the amount of separating area ranges from about 1 1/2 to 3 1/2 sq ft per ft of cut, whereas for the smaller combines the range is from about 4 to 6 sq ft per ft of cut. The relatively larger area for the small combines contributes to a greater capacity per foot of cut, thus permitting higher forward speeds in many cases.

17.14. Cleaning. After the seed has been separated from the straw, it is still mixed with large quantities of chaff and other plant residue that have passed through the grates or straw rack with the seed. The functions of the cleaning unit (shoe) are to separate and clean the threshed seed, return partially threshed heads to the cylinder, and dispose of the remaining debris.

The essential components of a cleaning shoe (Fig. 17.2) include two (or sometimes three) oscillating sieves, a fan (usually an undershot, paddle type), a chaffer extension at the rear of the upper sieve, and augers for conveying the tailings and the clean seed to elevators. An adjustable wind-direction vane is often placed between the fan and the sieves. Ordinarily no provision is made in the shoe for removing particles smaller than the seed, but auxiliary attachments are available for this purpose.

The air blast from the fan is directed upward and rearward through the sieves, agitating or "floating" the mixture on the chaffer (upper) sieve to aid in separation of the seed as the material is moved rearward. The air blast also blows out most of the light, chaffy material as the seeds fall through the relatively large openings of the chaffer sieve and then through the smaller openings of the shoe sieve.

Small pieces of unthreshed heads and other heavy material that pass through the chaffer sieve but do not fall through the openings of the shoe sieve are discharged into the tailings auger.
Most of the unthreshed heads pass over the chaffer sieve onto the chaffer extension, which has openings large enough for them to fall through into the tailings auger. If the openings of the chaffer sieve or chaffer extension are too large in relation to the amount of air blast, there will be excessive amounts of chaff and other debris in the tailings. If they are too small, seed will be carried out in the blanket of material.

Adjustable-type chaffer sieves are most common, although sieves with fixed openings are sometimes used. Adjustable chaffer sieves (Fig. 17.10) have a series of cross-members mounted on cross-rods and linked together so they can all be rotated simultaneously to increase or decrease the size of the openings. With an adjustable-lip chaffer sieve (Fig. 17.10a), there is a tendency for straws to become oriented parallel with the cross-members and then fall through the slots between them, particularly with small-seed crops requiring relatively low air velocities. The vertical partitions on the type shown in Fig. 17.10b tend to minimize this problem.

Since the shoe sieve is primarily for size separation, the openings should be just large enough to permit free passage of the seed. Adjustable shoe sieves, similar to the chaffer sieve shown in Fig. 17.10a but with smaller lips and openings, are most common. Round-hole sieves are, however, preferable for harvesting
small-seed crops such as alfalfa and clover, and sieves with elongated holes are occasionally used for other crops.

The air volume is controlled by means of adjustable shutters on the fan inlet openings (Fig. 17.2) and by changing the fan speed. If the air blast is excessive or is directed too far toward the rear of the shoe, seed will tend to be lifted and blown out of the rear of the machine. High air velocities also interfere with the passage of seed through the openings in the shoe sieve, thus increasing the amount of free seed in the tailings. Too little air results in insufficient agitation of the mixture on the chaffer sieve, thereby causing seed to be carried out in the blanket of material. Insufficient air also results in a poorer cleaning job (more trash with the clean seed) and increased amounts of tailings (particularly chaff and other debris). The maximum amount of air that can be used is related to the size, specific gravity, and aerodynamic characteristics of the seeds. More air is permissible with large seeds than with small ones.

The effects of too little air and of too much air, in relation to free-seed losses over the shoe, are indicated for two crops in Fig. 17.11. Note that in these tests, all made with the same 12-ft self-propelled machine, the effect of too little air was far more serious for barley than for alfalfa. The relatively high seed rate
indicated for barley, in comparison with the rate for alfalfa (total seed into combine), may partially account for the difference in response to insufficient air.

Adequate cleaning area in the shoe is important. The larger combines (over 7-ft cut) generally have from about 1 to 2 1/4 sq ft of cleaning area (including all sieves and the chaffer extension) per foot of cut, whereas the smaller machines have 2 to 3 sq ft per ft of cut. As in the case of the separating area, the small combines have more cleaning capacity in relation to the width of cut than do the larger machines.

17.15. Performance of Separating and Cleaning Units. Maximum efficiency in combining is generally attained through balancing of losses occurring at different places within the machine, to obtain a minimum over-all loss. The performance of the shoe, for example, is affected by the performance of the straw rack or walkers, since all the material falling through the straw carrier must be handled by the shoe. Reducing the size of the openings in the straw rack to decrease the load on the shoe may reduce the shoe loss but also tends to increase the rack loss.

The relative load that must be handled by the shoe is influenced by the type of crop and the threshing conditions. Tests have shown that, when harvesting small grains in Ohio, 90 to 95 per cent of the total input of straw and chaff was retained by the straw rack. In recent tests under the drier conditions found in the interior valleys of California, the walkers or rack retained only 60 to 70 per cent of the material when harvesting wheat or barley. With windrowed alfalfa (in California), only half of the material was retained by the walkers. Recirculation of large amounts of tailings, in the case of alfalfa and other small-seed legumes, further increases the shoe load.

Reducing the threshing effect of the cylinder may increase the cylinder loss (unthreshed seed) but tends to result in lower rack and shoe losses because of less straw breakage, with improved separation on the rack and less load on the shoe. Figure 17.12 illustrates an example of this effect, wherein increasing the cylinder-concave clearance from 1/4 in. to 3/4 in. more than tripled the cylinder loss but reduced the over-all loss by one-third because the rack and shoe losses were less.

Free-seed losses over the rack and shoe may increase rapidly when the combine is overloaded, as illustrated for two conditions
in Fig. 17.13. These curves represent the same tests for which unthreshed-seed losses were plotted in Fig. 17.6. The curves shown in Fig. 17.13 for barley are probably steeper than normal, but they demonstrate that under some conditions the problem of overloading can be rather serious. Note that, for a given feed rate, the amount of seed passing through the combine was more than five times as great for barley as for alfalfa (76 per cent of feed rate versus 14 per cent).

Fig. 17.12. Effect of cylinder-concave clearance upon over-all grain losses from a combine. (G. W. McCuen and E. A. Silver. *Ohio Agr. Expt. Sta. Bull.* 643.)

When direct-combining standing crops, the total seed loss often can be reduced by cutting higher to reduce the load on the rack and shoe, even though header losses may be increased somewhat. In tests with bearded wheat in Ohio, reducing the length of straw cut from 38 in. to 26 in. reduced the over-all loss from 302 to 73 lb per acre. The seed rate can also be reduced by operating at a lower forward speed or by utilizing only part of the cutting width of the header.

With windrowed crops, the feed rate (once the windrows have been formed) can be controlled only by changing the forward speed of the combine. In heavy, windrowed crops such as alfalfa, forward speeds must sometimes be as low as ½ mph to prevent excessive seed losses, even when the windrows are spaced only 10 ft apart. With most self-propelled machines, special attachments are required to obtain these low speeds.
GRAIN AND SEED HARVESTING

17.16. Testing of Combines. In order to make an accurate and complete study of the losses from various parts of the machine, provision should be made for simultaneous collections of the material discharged from the straw rack, the material from the shoe, and the seed at the grain tank or sack. These collections are made while the machine is operating at a constant speed over a measured and timed distance and are then weighed separately.

Separate collections from the rack and the shoe are made by dragging two canvas sheets behind the machine and holding the front ends, one above and behind the other, in appropriate positions. The material collected on the top canvas consists of straw, threshed seed (rack loss), and some of the unthreshed seed (cylinder loss). The lower canvas collects the chaff and free seed from the shoe, as well as additional unthreshed seed.

After weighing the collections, the material from each canvas is put through a fanning mill to recover the free seed. It is then run through a threshing cylinder or a slow-speed hammer mill to thresh out the remaining seed and, finally, over the fanning mill again to recover the seed that was originally unthreshed. The percentage losses at the various points are then calculated on the basis of the total weight of all seed collected during the run.
Header losses may be determined by placing a frame of known area in a number of locations at random over the field and picking up the shattered heads or seeds within the frame each time. The frame should not be placed where seed that has been lost from the rear of the machine would be included in the count. If there has been some shattering prior to the harvesting, a pre-harvest count is necessary to permit determination of the additional shatter caused by the combine.

If seed damage is a problem, samples taken from the clean-seed spout are checked visually for external damage and are then subjected to laboratory germination tests for internal damage. Hand-threshed samples are also checked. If it is suspected that damage is occurring in parts of the machine other than the cylinder, samples should be taken from pertinent locations along the path of the seed within the machine.

17.17. Power Sources for Combines. Power for operating the smaller combines may be furnished either through the tractor pto or by an auxiliary engine. Combines with pto drives have a lower first cost but lack the flexibility provided by an auxiliary engine. Because the threshing mechanism must be driven at a constant speed for best results, the tractor engine speed must be kept constant when using a pto drive. Consequently, the forward speed can be changed only by selection of transmission gears. A "live" or continuous-running pto is desirable, so that the tractor can be stopped or started without stopping the threshing mechanism.

Practically all of the larger, trailed combines (over 8-ft cut) are equipped with auxiliary engines. On most self-propelled combines, the same engine that operates the threshing mechanism also propels the machine. In order to obtain flexibility in regard to forward speeds (without changing the engine speed), variable-speed drive units are employed in conjunction with gear-type transmissions. The two common types are variable-speed V-belt drives and variable-speed hydraulic drives (see Chapter 4).

17.18. Power Requirements. Figure 17.14 shows the total power requirements and the distribution of power between the various components, for a straight-through type of combine with an 80-in. cutter bar, a 7-ft gathering width, and a 60-in. cylinder. In these tests the feed rate was determined by picking up the material from the test strip after each run. In the interest of
clarity, the experimental points have been omitted from the four lowest curves.

Note that the cylinder curves in Fig. 17.14 have quite different shapes for soybeans and wheat. Burrough explains this as being the result of the way the material was fed into the cylinder. The wheat always entered as a full-width layer so that increased rates required greater compression of the material as it passed between the cylinder and concave, thus causing a rapid increase in power requirements. The soybeans, on the other hand, entered as two narrow bands from the two rows being harvested and did not utilize the full width of the cylinder. As the rate increased, the bands merely widened out without requiring a great deal more compression and the power requirement increased almost linearly with feed rate.

The difference between the summation curve and the measured total-power curve for soybeans is presumed to represent friction losses in the various power-transmission systems on the combine. Total power requirements were not measured for the wheat, but an estimated curve is shown, based on the assumption that the friction losses would be the same as for soybeans. The average total power requirement at the highest feed rates in these tests

![Fig. 17.14. Power distribution in a 7-ft, straight-through type of combine (D. E. Burrough, Agr. Eng., January, 1954.)](image-url)
was about $1\frac{1}{4}$ to $1\frac{1}{2}$ hp per ft of cut. Adequate allowance for overloads, particularly in the cylinder, must be made in selecting a power unit and designing the drive systems. Appendix C lists typical engine power requirements for trailed combines as being 2 to 3 hp per ft of cut.

Traction power requirements for a self-propelled combine are influenced by the weight of the machine plus the seed in the grain tank, the size of tires, the condition of the ground surface, and the slope of the ground. Manufacturers' ratings for the engines on current models of self-propelled combines average about 4 hp per ft of cut.

**17.19. Windrowing.** This operation is ordinarily performed with a machine that resembles the header of a platform-type combine. Such a windrower includes a reel, a cutter bar, and one or more cross-conveyors (usually canvas), to discharge the cut material into the windrow. For cutting tangled crops, a short, vertical cutter bar is sometimes mounted at one end of the main cutter bar. Windrowers may be trailed, self-propelled, or mounted on the rear of a tractor that is operated in reverse. The last two types are better for opening up fields. Widths of cut ordinarily range from 8 to 15 ft. It is preferable that the reel be ground driven. The knife and the conveyors are ordinarily driven from the tractor pto or from the mounted engine on self-propelled units.

Some of the smaller, trailed windrowers discharge the material beyond the end of the cutter bar, on the previously cut strip. It is preferable, however, to discharge the material behind the cutter bar so that the first windrow when opening a field will rest on stubble. This may be accomplished by having the cutter bar extend beyond the discharge end of the cross-conveyor, or by providing two cross-conveyors to move the material from both sides toward the center of cut.

For best curing and good combine performance, the windrows should be continuous, uniform, and not too tight. In windrowing grain, the stubble should be at least 6 to 8 in. tall to permit free air circulation beneath the windrow. The arrangement of the windrower should be such that the material is not discharged onto stubble that has been run over by a wheel.

With some crops, such as the small-seed legumes, it is necessary to cut close to the ground to recover as much of the seed as
possible. Under these conditions, a drag-type windrower attached to the rear of a mower cutter bar (see Section 14.14) is sometimes used instead of the platform-type windrower. In light stands, however, the windrows are not as uniform as those obtained with the platform-type unit.

17.20. Windrow-Pickup Attachments. For windrow combining, a pickup is attached to the front of the combine header. It should have support runners operating on the ground and be hinged to the header so that it can "float" without the necessity for precise control of the header height. For best performance and minimum shattering, the peripheral speed of the pickup unit should be 10 to 20 per cent greater than the forward speed.

The various types of windrow pickup units are described in Chapter 14 (Section 14.25). The types most common on combines are (a) cylinder with retracting fingers, (b) cylinder with fixed fingers and stripper bars, and (c) flat-belt type with slats and spring teeth attached to the face of the belt. Where seed shattering is a problem, the belt type is preferable to the others because much of the shattered seed will be carried up on the belt.

REFERENCES


**PROBLEMS**

17.1. Construct a graph showing the relation of peripheral speed to cylinder rpm for cylinder diameters of 14, 16, 18, 20, 22, and 24 in. and peripheral speeds up to 6000 fpm.

17.2. An actual field test with a combine harvesting barley in central California gave the following data: width of cut = 12 ft; length of test ran = 40 ft; time = 21.3 sec; total material over rack = 18.9 lb; free seed over rack = 70 gm; unthreshed seed over rack = 55 gm; total material over shoe = 8.8 lb; free seed over shoe = 264 gm; unthreshed seed over shoe = 74 gm; total seed collected at grain tank = 35.5 lb. Calculate:

(a) Cylinder, rack, shoe, and total grain losses, in per cent of total yield.

(b) Seed yield and total loss, in pounds per acre.

(c) Rates of straw and chaff over the rack and over the shoe, in pounds per minute.

(d) Total feed rate, in pounds per minute.

(e) Per cent of straw and chaff retained by rack.

17.3. From material presented in the text, or from other sources, develop a list of possible causes for each of the following conditions:

(a) Excessive header loss.

(b) Excessive amount of unthreshed seed.

(c) Excessive free-seed loss over straw rack.

(d) Cracked grain.

(e) Excessive free-seed loss over rear of shoe.

(f) Excessive amount of chaff in tailings.

(g) Excessive amount of free seed in tailings.
17.4. Alfalfa seed is to be harvested with a self-propelled combine under the following conditions: width of swath = 12 ft; field efficiency = 75 per cent; straw and chaff yield = 2½ tons per acre; seed yield = 600 lb per acre. The cost factors are: first cost of combine = $5500; total annual overhead charge = 13 per cent of first cost; fuel cost = $0.50 per acre; cost for engine oil and lubrication = $0.20 per hr; operator's wage = $1.25 per hr; value of seed = $0.30 per lb. Assume that the machine is to be used for 200 hr per yr (the total acreage depending upon the speed of operation) and that the curves in Figs. 17.6 and 17.13 represent the seed losses as a function of feed rate.

(a) Plot a curve of total cost per acre (including a charge for the value of the seed lost) versus feed rate. From this curve determine the most economical feed rate at which to operate.

(b) What forward speed corresponds to the most economical feed rate?

(c) How many acres can be harvested in the 200-hr period when operating at the most economical speed?
CHAPTER 18

Corn Picking
and Shelling

18.1. Introduction. Corn is our greatest-acreage field crop, as well as being our most important source of feed. In recent years almost 90 per cent of the total corn acreage has been harvested for grain. Although some corn is produced in every state in the United States, 90 per cent of the total production is confined to the 12 north-central states.

Although a few one-row, ground-driven corn pickers were built during the decade preceding World War I, there was little interest in mechanical pickers until the tractor pto was adapted to them, in the late 1920's. Two-row mounted pickers appeared about this time, and picker-shellers became available in the middle 1930's. Even then, acceptance of the mechanical picker was slow, with less than 15 per cent of the crop being harvested mechanically in 1938.

The general adoption of the mechanical picker in the corn belt came immediately following World War II. In 1951, corn pickers harvested 90 per cent or more of the total acreage in each of seven corn-belt states, although they were still used for only a small per cent of the crop in the southern states. The average for the United States was 68 per cent of the total acreage.

18.2. Types of Machines. Corn harvesters may be classified as (a) snappers, (b) picker-huskers, commonly known as pickers, and (c) picker-shellers. A fourth type, introduced commercially in 1954, is a combination picker-chopper. This machine first picks the ears from the standing stalks and delivers them to a wagon trailed behind the machine. The stalks are then cut and fed into a conventional forage-chopper cutterhead. The chopped

*United States Crop Census.

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material may be discharged onto the ground for incorporation into the soil, or it may be delivered into a truck or wagon moving along beside the harvester. The picking unit is actually an attachment for a field chopper. At the present time there is also considerable interest in adapting modified grain combines for picking and field shelling of corn.

Snappers and picker-huskers are available in both one-row and two-row sizes, whereas picker-shellers are generally two-row units. Most two-row machines are designed with snapping-unit center distances ranging from 39 to 41 in., but they can accommodate row spacings 2 or 3 in. greater or less than the spacing of the snapping units. Trailed machines have predominated in the past, but the current trend is toward mounted and self-propelled units. In 1951, almost half of the United States production was of the mounted and self-propelled types.

18.3. Functional Elements of Corn Harvesters. Basically, a picker-husker includes the following components:

1. An arrangement to guide the stalks into the machine.
2. Snapping rolls to remove the ears from the stalks.
3. Lugged gathering chains above the snapping rolls to assist in feeding the stalks into the rolls and moving the stalks and snapped ears rearward through the snapping zone.
4. A unit to remove the husks from the ears.
5. A conveying system to elevate and deliver the husked ears into a wagon.

A snapper is essentially the same as a picker-husker except that there is no provision for removing the husks, the snapped ears being conveyed directly to the wagon. Some husks, however, are removed in the snapping operation, most of the removed husks remaining attached to the stalks. In tests with mechanical snappers in Texas, 15 to 40 per cent of the ears were completely husked in the snapping operation.16

A picker-sheller does not have a husking unit as such, but it does include a shelling and cleaning unit. The shelled corn is generally delivered to a bulk tank on the machine, from which it is unloaded into a wagon or truck.

18.4. Gathering Devices and Snapping Units. Typical, streamlined gatherers are illustrated in Fig. 18.1. The front sections are generally hinged to enable them to follow the con-
tour of the ground. For minimum loss of corn ears, the gathering points should be operated on the ground or as low as surface irregularities will permit. The gathering units lift the lodged and leaning stalks and direct the mass of stalks into the machine.

The stalks enter the throat between the tapered front ends of two downwardly rotating snapping rolls (Figs. 18.2 and 18.3). One or more gathering chains equipped with finger links (Figs.

![Diagram of corn picker](image)

*Fig. 18.1. Front view of a two-row, mounted corn picker, showing the streamlined gathering units. (International Harvester Co.)*

18.1, 18.2, and 18.3) assist in moving the material into and through the snapping zone. The gathering chains should be timed so that the fingers on opposite chains are evenly staggered, and their speed in relation to the forward speed of the machine is important.

The snapping rolls are generally operated with peripheral speeds in the order of 500 to 600 fpm. As the stalks are pulled downward between the rolls, the ears, being too large to pass through the available space, are snapped off. In most cases, the snapped ears, with the aid of the gathering chains, are moved onto a conveyor that is adjacent to one side of the snapping rolls. In some machines, however, the rolls are combination units that perform both the snapping and the husking operations. The snapped ears then are moved directly onto the husking sections by means of the gathering chains.
18.5. Types of Snapping Rolls. The most common type of snapping roll is illustrated in Fig. 18.2. Rolls of this type are ordinarily made of cast iron or cast steel, with spiral ribs or lugs on their surfaces. Their aggressiveness can be varied for different operating conditions by the attachment or removal of screws, bolts, or additional lugs. Roll lengths generally range from about 40 to 55 in.

The direction of wrap of the spirals, from front to rear, is opposite to the direction of rotation of the roll. Thus the spiral ribs tend to move the stalks rearward, particularly if there is slippage between the stalks and rolls. Most manufacturers time the two rolls so that the ribs are staggered, but in some models the spirals are directly opposite each other.
Figure 18.3 shows a combination snapping and husking unit in which the snapping rolls are considerably more aggressive than the spiral-ribbed rolls and can therefore be relatively short in length. Another unit of this type has intermeshing, fluted snap-

Fig. 18.3. Combination snapping and husking rolls on a corn picker. (a) Cross-sectional view of husking rolls. (b) Cross-sectional view showing the action of the short, aggressive, snapping rolls and the stripper plates. (c) General arrangement of one picking and husking unit of a two-row machine. (Allis-Chalmers Mfg. Co.)

ping rolls. Stripper plates are provided so that the ears can be snapped without coming in contact with the rolls (Fig. 18.3b). Otherwise, shelling losses would be excessive because of the extremely aggressive action of this type of roll.

18.6. Adjustment of Snapping Rolls. The two principal adjustments for the snapping rolls are the clearance between the rolls (and between stripper plates if used) and the aggressiveness
of the roll surfaces. Safety clutches in the roll drive system should also be properly adjusted. On most machines, roll clearance is adjustable only at the lower end of the rolls. Since this adjustment may need to be changed rather frequently to accommodate various crop conditions, it should be a simple and convenient operation. On some of the more recent tractor-mounted pickers, the roll spacing can be adjusted from the tractor seat without stopping. The aggressiveness of the rolls is changed by adding or removing lugs of various types and by replacing worn rolls or surface attachments.

The snapping rolls should be operated as close together as the condition of the crop will permit (¼ to ½ in. under normal conditions). The general effect of having the rolls too far apart (when stripper plates are not used) is to increase the loss due to shelling at the rolls and to increase the number of small ears and nubbins that pass between the rolls and are lost. When the rolls are too close together, the stalks are broken more and small pieces of stalk may be delivered with the corn. The rolls should be close together (but not touching) when the stalks are damp and the ears hard to snap. Spacings should ordinarily be wider than normal when a large percentage of the stalks are down or when the stalks are standing but are either brittle or tall and heavy. When stripper plates are used they should be just far enough apart to permit ready passage of the stalks without breakage.

If the roll surfaces are too aggressive or the roll speed is too great, excessive shelling will occur. If the roll surfaces are not aggressive enough, the passage of the stalk is retarded because of slippage at the time the ear is being snapped, especially when the stalks are dry. This slippage causes the end of the ear to be held in contact with the rolls for a longer time, which tends to increase the shelled-corn loss. Insufficient aggressiveness under dry conditions may also result in the accumulation of trash on the rolls.

18.7. Trash Removal. Special trash rolls are often provided to remove trash and broken stalks not expelled by the snapping rolls. Fluted sections may be incorporated on the upper ends of the snapping rolls (Fig. 18.4), and transverse, fluted rolls are sometimes mounted at the discharge end of a conveyor (Figs. 18.3c and 18.5). The principal function of the transverse rolls is
to remove relatively long sections of stalks. Small blowers may also be used to assist in removing trash.

**18.8. Husking Units.** Whereas the components discussed in the four preceding sections are characteristic of all types of corn harvesters, husking rolls are found only on picker-huskers. The husking rolls may be in a separate husking bed at the discharge end of a short elevator from the snapping rolls (Fig. 18.6), they may be incorporated in the snapping-roll elevator (Fig. 18.7), or they may be direct extensions of the snapping rolls (Fig. 18.3).

With any of the various arrangements, the husks are engaged by the rolls and pulled down between them, leaving the husked ears on top, as illustrated at the upper left in Fig. 18.3. The rolls of each pair are usually held together by adjustable springs. Various devices are employed to distribute the ears across the husking bed, hold them down against the rolls, and move them along the rolls at a uniform rate. If the husked ears discharge from the lower end of a sloping bed (Fig. 18.6), the device is known as an ear retarder. If the ears move upward along the rolls, it is a forwarder. A properly designed roll with a spirally grooved surface helps to advance the ears along the rolls.4

Most husking units have one roll of each pair made of rubber. The other may be of steel (Fig. 18.6), cast iron, wood, or rubber
(Fig. 18.7) and may have steel or rubber inserts (Fig. 18.3a) to increase the effectiveness of the combination. Lugs, screws, etc., can be added to the surfaces of some rolls to increase their aggressiveness. A sieve arrangement of some sort is generally placed beneath the husking bed to recover shelled corn from the husks.

The practice of using rubber for at least one of the husking rolls is in conformance with results obtained by Collins and his associates in laboratory tests with various roll combinations (reported in 1940). In their tests, a combination of a smooth steel roll and a spirally grooved cast-iron roll removed only about one-third of the husks, whereas nine other combinations, each having at least one rubber roll, removed 88 to 98 per cent of the husks. A prohibitive amount of shelling was obtained
with two combinations in which both rolls were rubber. These tests also indicated that changing the roll speed in the range between 250 and 700 rpm had no significant effect upon husking effectiveness but that reducing it below 200 rpm decreased the effectiveness.

18.9. Adjustment of Husking Unit. The husking unit should be adjusted to obtain maximum husking effectiveness with minimum shelling. USDA tests have indicated that, for storage with a minimum amount of spoilage, the total amount of shelled corn in the crib should not exceed 1 per cent. Husks and trash in the crib are undesirable because they occupy storage space and reduce the natural air flow through the corn.

The principal adjustments are (a) contact pressure between rolls, (b) aggressiveness of the rolls, as modified by adding or removing husking pegs, pins, screws, etc., and (c) the relation between the rolls and the feed mechanism. Increasing the aggressiveness of the rolls improves the husking effectiveness but also tends to increase shelling. But, if the surfaces are not aggressive enough, shelling may be increased because the ears are held against the rolls longer. Excessive shelling may also result from insufficient roll contact pressure (especially when the ears are small), insufficient clearance between the rolls and the feed mechanism, accumulation of trash on the husking bed (which slows the progress of the ears), and excessive roll speeds.

18.10. Shelling and Cleaning Unit. In a picker-sheller, the snapped ears are discharged onto a unit that removes the husks not taken off by the snapping rolls, shells the corn from the cobs, and separates the shelled corn from the cobs and trash.

The shelling unit generally consists of a beater or cylinder, operating over a concave made from round bars (Fig. 18.8) or from a perforated metal sheet. The kernels are removed from the cobs by the action of the beater in agitating and moving the ears over the concave. The kernels fall through the concave as the cobs and husks are moved axially through the shelling unit by the spiral arrangement of the cylinder elements.

By means of an adjustable gate at the discharge end of the cylinder, the length of time that the ears remain in the shelling unit can be varied to accommodate different crop conditions. Final separation and cleaning is accomplished in a cleaning unit similar to that of a grain combine (Fig. 18.8).
18.11. Field Losses with Corn Harvesters. Loss of corn in the field is one of the serious problems associated with mechanical harvesting. It is estimated that, in general, field losses average about 10 per cent of the total yield.\textsuperscript{14,9} Hand-harvesting losses (ears overlooked) are estimated to be about 1 bu per acre,\textsuperscript{15} which would be 1\% per cent of an 80-bu crop.

Field losses can be minimized by proper adjustment of the machine (as described in preceding sections), by careful driving to keep the snapping units centered on the rows, and by avoiding high forward speeds. High speeds and poor centering on the row whip the stalks vigorously, thus increasing ear losses, especially when the stalks are dry and brittle. Under ordinary conditions, speeds should probably not exceed 3 to 3\(\frac{1}{2}\) mph.

Field losses are also influenced by the date of picking, varietal differences, and other factors. As the season advances, stalks become dry and brittle and tend to lodge as a result of adverse weather conditions, thereby making it increasingly difficult to recover the ears. Shelled-corn losses at the snapping rolls also increase during the season, because of the progressive drying of
the kernels. Figure 18.9 indicates that shelled-corn losses increase rapidly when the kernel moisture content drops below about 17 per cent.

Figure 18.10 shows the general relation of field losses to harvesting date, based upon results for picker-huskers as reported by several investigators. Most of the data indicate shelled-corn losses somewhat greater than machine ear losses. These results emphasize the importance of harvesting the crop as soon as it is dry enough for safe storage.

The losses in the picking unit of a picker-sheller are influenced by the same factors as the losses from a picker-husker and should be of about the same magnitude under comparable conditions. Sheller losses (primarily unshelled kernels left on the cobs) are related to the moisture content of the cobs, as indicated in the left-hand graph of Fig. 18.11, to the capacity of the shelling and cleaning unit, and to the load through the unit. Whereas losses from the picking unit increase during the season, sheller losses are highest at the start of the season and decrease as the ears dry out.

The right-hand graph in Fig. 18.11 shows the relation of sheller damage to the moisture content of the kernels. Note that reducing the moisture content decreased the damage, which is opposite.
Fig. 18.10. Relation of field losses to harvest date, for picker-huskers.\(^1,9,15\)

Fig. 18.11. Left: Relation of sheller loss to cob moisture content. Right: Effect of kernel moisture content upon mechanical damage from the sheller. (D. E. Burrough and R. P. Harbage. *Agr. Eng.*, January, 1953.)
to the effect of moisture content upon thresher damage to beans (see Section 17.11).

18.12. Application of Harvesting Methods. Picker-huskers are the predominant type of corn harvester in the corn belt. In many areas of the South, corn is snapped (by hand or with machines), primarily because of the prevailing belief that leaving the husks on the ears retards weevil infestation. Field shelling of corn may be done with a picker-sheller, with a stationary-type shelling unit on a trailer behind a snapper or picker-husker, or with a modified grain combine. Field shelling is more economical than other harvesting methods (Fig. 18.12) and requires less labor. Cobs are left in the field (no disposal problem), and less storage space is required for the harvested product.

In the corn belt and other similar areas, corn shelled at the time of picking usually contains 17 to 24 per cent moisture and must be dried artificially (to about 14 per cent) before it can safely be stored. In dry climates, and especially late in the season, artificial drying of field-shelled corn may not be necessary. Ear corn, on the other hand, can safely be cribbed without artificial drying when the kernel moisture content is about 20 per cent (the moisture content of the cobs is then about 30 per cent and the average for the ears is about 22 per cent).

Kiesselbach found that, under Nebraska conditions, corn

![Fig. 18.12. Comparative costs of picking, hauling, and shelling corn with various harvesting methods. (Illinois Vocational Agriculture Service, unit 8, revised, 1951.)](image-url)
maturity was reached when the moisture contents of the kernels, cobs, and ears had decreased to approximately 34, 53, and 38 per cent, respectively. Theoretically, the crop could be harvested at this stage without any loss of potential yield or quality. But field tests have indicated that satisfactory performance of the shelling unit on a picker-sheller is difficult to obtain at these high moisture contents.\textsuperscript{3}

18.13. Safety in Corn-Picker Operation. Probably more serious accidents occur with corn pickers than with any other type of farm machinery. Records kept in Illinois indicate that the accident rate has increased from 4.3 accidents per 1000 machines in 1945 to 7.5 in 1950. Most of the corn-picker accidents are caused by the operator trying to unclog the snapping rolls while they are running and accidentally touching them with his clothing or hand.

The corn picker is inherently a dangerous machine, since the snapping rolls cannot be completely shielded. From the design standpoint, various types of modifications to improve safety have been studied, and a number of them have been tried in the field.\textsuperscript{13} Among these are (a) adjustable, stationary strippers above the rolls, (b) a manual device to permit opening the rolls from the tractor seat in case of clogging or imminent clogging, (c) spring mounting of the rolls, (d) speeding up the rolls to increase their capacity, and (e) a roll-reversing mechanism to assist in cleaning out the machine. Although some of these arrangements show promise and the first two are available on commercial machines, they all have disadvantages. None of them can completely guarantee safe operation if the operator is careless.

From the operator's standpoint, safety is promoted by such practices as keeping the machine in proper adjustment, harvesting only under favorable conditions, and driving carefully so that the snapping units are always centered on the rows (to minimize clogging). No attempt should ever be made to unclog the rolls until the picker mechanism has been stopped.

18.14. Harvesting Corn with a Grain Combine. Two of the chief objectives in attempting to adapt the grain combine for harvesting corn are (a) to increase the annual use of the combine and thus reduce the cost per acre for harvesting both the grain and the corn crop, and (b) to reduce field losses below the usual values for corn pickers. Although attempts to harvest
corn with a grain combine were made prior to 1930,\textsuperscript{11} it is only since about 1950 that any great interest has been shown in this possible harvesting method.

The Illinois Agricultural Experiment Station made laboratory studies in 1951 and 1952 to determine the feasibility of shelling corn with a combine cylinder,\textsuperscript{7} and in 1953 made field trials with a straight-through type of combine equipped with a modified header.\textsuperscript{12} During the 1954 season, several farm machinery manufacturers had combines with experimental headers in the field. A number of machines have also been converted by farmers and local machine shops. Both the straight-through type of combine, with the long cylinder, and the self-propelled machines, with short cylinders and narrow separators, have been tried.

The principal problem appears to be getting the material into the machine. Two general approaches to this problem are being made. In one system, the stalks are left and the entire plants fed through the machine. This may be accomplished by adding suitable attachments to a standard header,\textsuperscript{12} or by installing an entirely new header unit designed specifically for corn (similar to the row-crop attachment for a field chopper). The chief objectives in this system are to eliminate the high snapping-roll losses common with conventional pickers, and to permit the use of a stalk shredder on the rear of the machine.

The other approach is to replace the combine header with a snapping unit similar to those found on conventional corn pickers and then feed only the ears through the combine. The principal advantage in this system is the smaller amount of material handled by the combine.

For the 1953 field tests with the Illinois combine (feeding stalks through the machine), total field losses of about 6 per cent are reported.\textsuperscript{12} Over half of this amount was from ears lost at the header. The kernel moisture content was 18 per cent and the average yield was 82 bu per acre.

18.15. Combine Cylinder Studies. The recent Illinois studies\textsuperscript{7} were made with a rasp-bar cylinder from a straight-through type of combine. In these laboratory tests it was found that, for satisfactory results with this long cylinder, it was necessary to either double the number of cylinder bars (twelve instead of six) or add filler plates between the standard six bars. A concave screen with \( \frac{3}{4} \)-in. round holes permitted separation
of up to 60 per cent of the shelled corn at this point. When the number of concave bars was increased above the standard number, trouble was experienced in feeding.

The 12-bar cylinder gave somewhat better shelling than the 6-bar cylinder with filler plates, whereas the 6-bar arrangement resulted in less damage to the kernels. With either type of cylinder the percentage of unshelled corn compared favorably with that from a picker-sheller and varied less with changes in moisture content. With the 12-bar cylinder at a peripheral speed of about 3000 fpm, the unshelled-corn losses were less than 1 per cent over the range of moisture contents from 16 to 28 per cent. Both cylinder arrangements resulted in a higher percentage of mechanical damage than with a picker-sheller, although the same characteristic of decreasing damage with decreasing moisture content was evident.

It appears from these laboratory trials, as well as from limited field experience, that the best cylinder speed for effective shelling and minimum damage is between 2500 and 3500 fpm (peripheral speed). The clearance between the cylinder and the concave should be 1 to \( \frac{1}{8} \) in. at the front and \( \frac{1}{2} \) to \( \frac{3}{8} \) in. at the rear. Limited field experience also indicates that under some conditions it is possible to obtain satisfactory results without adding extra bars or fillers to the cylinder.

Measurements by Hopkins and Pickard indicate that maximum forces on the concave are only slightly higher when harvesting corn than when harvesting small grains, but that each group of ears causes momentary forces higher than the steadier loads in threshing grain. As would be expected, the forces on the concave diminish as the clearance is increased. Extensive field experience will be necessary to determine the effect of the corn-harvesting operation on the repair and upkeep costs for the combine.

REFERENCES

**PROBLEM**

18.1. If the field efficiency for a two-row corn picker is 65 per cent and the forward speed is 3 mph, what is the capacity in acres per hour? Assume a 40-in. row spacing.
19.1. Introduction. Picking cotton by hand is tedious, hard work that accounts for 50 to 85 per cent of the total labor required to produce the crop when it is not harvested mechanically. Yet, until recently all the cotton produced in the world was harvested by hand. It is estimated that, on the basis of 150 lb of seed cotton picked per worker per day, 3 1/2 million laborers, each working 40 days, would be required to pick the United States annual production of 15 million bales.

Mechanical cotton harvesters available today are of two basic types, commonly known as pickers and strippers. Mechanical pickers are selective in that the seed cotton is removed from the open bolls, whereas green, unopened bolls are left on the plant to mature for later picking. In high-yielding areas, and in other areas where serious weather hazards make it important to start harvesting as early as possible, it is common practice to go over the field twice (about 4 to 6 weeks between pickings). Under some conditions a second picking is not economically justifiable.

The stripper, on the other hand, is a once-over machine. All bolls, whether open or closed, are removed from the plant in a single pass. Harvesting with a stripper is, therefore, usually delayed until the plants shed their leaves following the first frost. Chemical defoliants are sometimes applied to permit earlier stripping.

Mechanical pickers are best adapted to irrigated areas and regions of high rainfall where yields are ordinarily high, the fibers are long, the bolls are of the open type, and vegetative growth is rank. Strippers are most successful in the High Plains area of Texas and Oklahoma where the plants are small, yields are relatively low, and the fibers are short, hard bodied, and easily
cleaned in the gin. Because the initial cost of a stripper is only a fraction of the first cost of a picker (see Table 2.2) and the maintenance costs are also lower, strippers are economically practical for much smaller acreages than are pickers.

MECHANICAL PICKERS

19.2. Development. The development of a mechanical cotton picker has challenged the minds of men for more than 100 years. The first patent was issued to Rembert and Prescott in 1850. Approximately 900 patents covering various types of harvesting devices have since been granted. A patent issued to Campbell in 1895 covered some of the basic principles found in two of the commercial machines now available. Two other commercial units employ principles developed later by John and Mack Rust.

The first commercial pickers appeared in the early 1940's, but it was not until 1946 that these machines were manufactured in any appreciable volume. By 1952 over 12,000 mechanical pickers had been made available. In a study of 63 one-row mechanical pickers reported by Hedges and Bailey, the average output per machine was equivalent to that of 25 hand pickers.

19.3. Picker Arrangements and Basic Requirements. Essentially, a mechanical picker consists of the following functional units:

1. An arrangement for guiding the cotton plants into the picking zone and providing necessary support while the seed cotton is being removed.
2. Devices to remove the cotton from the open bolls.
3. A conveying system for the picked cotton.
4. A storage basket or container in which the picked cotton can be accumulated.

A typical arrangement is illustrated in Fig. 19.1. Carrying the harvested cotton in a basket on the tractor (rather than in a trailer) improves maneuverability, both in the field and in unloading. Height adjustment for the picking heads is obtained through hydraulic controls.

Heavy-duty or high-drum pickers are available as either one-row or two-row machines. These units are either self-propelled
or mounted on a modified wheel-type tractor (Fig. 19.1). Either arrangement provides good maneuverability, but overhead charges tend to be less for the mounted type because the tractor can be used for other operations before the picking season. However, the changes that must be made in the tractor to adapt it for mounting a high-drum picker are rather extensive and time consuming, partly because the direction of travel is reversed.

Lighter and less expensive, mounted one-row models, usually with smaller picking heads (shorter drums or fewer slats), are also available. These machines can readily be mounted on a conventional tractor. They are suitable for areas where the cotton is not too tall and, because of their lower initial cost, are economically practical for smaller acreages than are the larger units.

A mechanical picker must be capable of gathering mature seed cotton with a minimum of waste and without causing serious damage to the fiber, plant, or unopened bolls. To insure the highest possible grade of ginned cotton, the harvested material should have a minimum of leaves, stems, hulls, weeds, and other foreign matter entwined in the lint. How well this requirement is met depends somewhat upon how gently the plant is handled while
it is passing through the machine. Streamlining of the plant passageway, the use of suitable limb lifters, and synchronization of moving picker parts with the forward travel of the machine all tend to minimize disturbance to the plant.

19.4. Spindles. The basic principle of a revolving spindle penetrating the cotton plant, winding the seed cotton from the open boll, and retreating to a doffing zone is employed by all the commercial pickers now available. The rearward movement of the spindles while in the picking zone is substantially the same as the forward movement of the machine (generally 2 to 3 mph) so that the spindles, while in the picking zone, do not move forward or backward with respect to the cotton plant. Each rotating spindle merely probes straight into the cotton plant from the side of the row, works on an open boll if it encounters one, and then withdraws straight out to the side with a minimum of disturbance and damage to the remainder of the plant. The spacing of the spindles (approximately $1 \frac{1}{2} \times 1 \frac{3}{4}$ in.) is such that they can slip past unopened bolls and leave them on the plant to mature for a later picking.

Two general types of spindles are found in current models of mechanical pickers. Tapered spindles have three or four longitudinal rows of sharp barbs (Fig. 19.2) for engaging the cotton. Their tapered shape facilitates removal of the cotton (doffing) after they leave the picking zone. In current models, spindle speeds range from about 2000 rpm at 2 mph to 2700 rpm at 2.5 mph.

Straight spindles (Fig. 19.3) are longer than the tapered type and are considerably smaller in diameter. Although their picking surfaces are either fluted or slightly roughened, their picking ability depends primarily upon the spindles being wet when they contact the cotton. On present-day machines, spindles of this type are generally rotated at a speed of about 1200 rpm (at 3 mph) while in the picking zone and are not driven during the remainder of the cycle.

19.5. Drum-Type Spindle Arrangements. The spindles are carried either on columns or bars arranged in vertical drums or on vertical slats attached to endless chain belts. Two units may be arranged in tandem to pick from the two sides of the row in succession, or a single unit may be used alone, picking entirely from one side of the row.
The working elements of a typical, tandem-drum picker are shown in Fig. 19.2. A spring-loaded, adjustable pressure plate opposite each drum crowds the plants toward the spindles in the picking zone. The clearance between the pressure plate and the ends of the spindles varies from $\frac{3}{8}$ to 1 in., depending upon the size and density of the plants. In this particular machine the front drum has 16 spindle bars and the rear drum has 12 bars, with 20 spindles per bar (high-drum type). This gives a total of 560 spindles per row of cotton, each spindle requiring a precision-fit sleeve bearing and being driven through bevel gears by a shaft inside the spindle bar. In another make of picker the front and rear drums each have 300 spindles.
The proper orientation of the spindle bars in relation to the row (or to the doffers) is obtained by means of a stationary cam and followers on the bars. All linear motion of the spindles while in the picking zone is at right angles to the row. The spindles enter the cotton plant through a grid section (Fig. 19.2c) which subsequently prevents the plants from being pulled into the doffing area as the loaded spindles withdraw.

19.6. Chain-Belt Spindle Arrangement. The picking process with a chain-belt unit is essentially the same as with the drum-type picker, although the chain-belt principle (Fig. 19.3)
permits the spindles to remain in the picking zone for a longer time. In currently available pickers, the chain-belt units have straight spindles. The particular picking unit illustrated in Fig. 19.3 has 80 vertical slats, each with 16 spindles (total of 1280). Each spindle is rotated by means of a roller in contact with a stationary, rubber drive rail (Fig. 19.3c), but only while on the picking side of the unit.

Guide strips (Fig. 19.3c, top) hold the chains in position between the main sprockets and provide the proper curvature for moving the spindles laterally into and out of the row. Each spindle slat is pivoted between the upper and lower chains. While the spindles are being rotated, the action of the drive rollers on the rails maintains the spindles in a position normal to the curvature of the drive rails. As the spindles approach the stripper (F in Fig. 19.3a), contact with a stationary restraining block or hold-back device rotates the slats to angle the spindles toward the rear and thus orient them for the stripping operation.

Machines having the chain-belt arrangement and straight spindles ordinarily pick from only one side of the row, as illustrated in Fig. 19.3. However, one manufacturer of a two-row machine has provision for mounting the two picking units in tandem to pick a single row from both sides when in high-yielding cotton.

19.7. Spindle Moistening. Spindles of either type are moistened by water for two reasons: (a) as an aid to picking because cotton adheres better to a wet steel surface and (b) to keep the spindles clean. Spindles pick up a gummy substance from the plants, which, if allowed to accumulate, will collect dust and trash and thus interfere with picking. The addition of a wetting agent (detergent) reduces the amount of water required for moistening and at the same time makes it more effective.

A spindle-moistening system is provided for each picking unit, water being metered in equal amounts to each level of spindles. Application is made to each spindle just before it enters the picking zone, by means of a specially designed rubber wiping pad (Fig. 19.2b and part D in Fig. 19.3a).

19.8. Removal of Cotton from Spindles. In machines with tapered spindles, the seed cotton is removed from the spindles by means of rotating doffer plates (Fig. 19.2d). The clearance be-
tween the surface of the spindle and the rubber lugs on the doffer is about $\frac{3}{8}$ in. As the doffer lug moves over the spindle surface toward the tip, the cotton is forced off. With the small-diameter straight spindles, stripping is accomplished by moving the spindles axially through the spaces between closely fitted stripper shoes ($F$ in Fig. 19.3a).

19.9. Conveying and Carrying. Either a pneumatic conveying system (Fig. 19.1) or a mechanical elevator can be used to move the cotton from the doffing area to the storage container, the pneumatic system being the most common. Some machines have had a two-stage system, the first stage being mechanical and the second stage being pneumatic. The pneumatic discharge lends itself more readily to uniform distribution and filling of the storage basket. Some machines with dual picking units have two separate elevating systems (two blowers) to provide more uniform and positive conveying from each unit.

Storage baskets are ordinarily carried on the picker, as in Fig. 19.1. Capacities are generally about 750 lb (seed cotton) on one-row machines and 1200 to 1500 lb on two-row units. Baskets are emptied (normally into a parked transport trailer) either by tipping with a hydraulic cylinder or by means of a mechanical conveyor in the bottom of the basket.

MECHANICAL STRIPPERS

The cotton stripper is an outgrowth of the home-made cotton sleds that came into widespread use in the High Plains region of Texas in 1926. As early as 1914, sleds equipped with a section of picket fence were employed to salvage the cotton crop in the Plains area. The practice in this region, before the stripper was introduced, was to hand-snap the cotton bolls rather than to pick the seed cotton from the burs. During the past 25 years, the Texas Agricultural Experiment Station has made important contributions to the development of cotton-stripping machinery.

It has been estimated that in 1952 there were about 20,000 strippers available in the United States. In the High Plains area of Texas, the use of two-row strippers has reduced the labor requirements per acre from about 17 man-hr on dry-land cotton and 33 man-hr on irrigated cotton to about 1.5 and 2 man-hr, respectively (1947 to 1949 studies).
19.10. Principles and Development. The mechanical stripping of cotton is accomplished by forcing the plants through an area too small for the bolls to pass. The snapped bolls are collected in the machine while the plants remain in place in the row. In removing the bolls, an upward and forward force is applied to the plant. Since this force must be opposed by the root system, the plants must be firmly anchored in the row.

Early strippers consisted of parallel, inclined fingers mounted on the front end of a sled. The plants passed between the fingers while the bolls were retained to be raked into the sled. The action was that of combing the bolls from the plants as the machine moved down the row. This combing action also broke off many limbs, stems, and leaves, resulting in the collection of a large quantity of trash. Oftentimes the amount of trash collected was equal to the weight of the seed cotton harvested. Field losses, too, were excessive.18

In an endeavor to reduce the amount of trash collected and to improve the efficiency of the operation, agricultural engineers at the Texas Agricultural Experiment Station developed a roll-type stripper.27 This machine consisted of two spring-loaded, parallel rolls, 48 in. long, mounted at an angle of approximately 30° above the horizontal. The rolls were smooth and were driven so that their adjacent surfaces moved upward. Note that this is opposite to the direction of rotation of corn-snapping rolls. As the bolls were snapped off, they were moved away from the plant by the surface of the rolls and delivered onto adjacent conveyors.

19.11. Types of Rolls. In recent years further work has been done at both the Texas and the Oklahoma Experiment Stations in regard to the nature and composition of stripper rolls. Among the types that have been tried in addition to steel rolls are steel-wire, tampico, palmetto, calabar, and nylon brushes, and longitudinal rubber paddles. Bristles have been attached in a spiral pattern (Fig. 19.4) as well as longitudinally.

Commercial strippers either have one roll and an adjustable, stationary stripper bar (Fig. 19.5) or have two rolls. Smith15 recommends that the peripheral speed of stripper rolls be 25 to 50 per cent greater than the forward speed of the machine. The rolls may be smooth, toothed, or fluted, or may have brushes or paddles as described above. Recent trends have been toward surface materials other than steel. The roll illustrated in Fig.
19.5 has stripper fingers only on the upper one-third of the length and is intended for stormproof varieties. An interchangeable roll having fingers the full length is available for this machine and is suitable for open-type cotton.

In 1952 the Texas Station ran comparative tests with the Oklahoma brush-type unit (Fig. 19.4), their own (consisting of two rolls, each having eight longitudinal rubber paddles), and a commercial stripper having a steel roll and a stationary stripper bar. When operating in a high population of a stormproof variety, the percentage recovery of the cotton amounted to 97.4, 96.1, and 85.1 for the rubber, brush, and steel-roll strippers, respectively.

19.12. Miscellaneous Components of Strippers. Gathering devices (Fig. 19.4) are necessary to lift the plants and guide
them into the rolls. The height of the stripping unit may be controlled by means of a gage wheel or by adjusting the suspension height from the tractor. Mechanical elevating and conveying devices, with slotted bottoms to aid in removal of dirt and small debris, are most common. Augers, chain-and-slat conveyors, and finger rolls (Fig. 19.5) are employed. The Oklahoma experimental brush-type stripper shown in Fig. 10.4 is equipped with a pneumatic conveying system.

Strippers are generally tractor mounted, with two-row units being most common. The harvested cotton is ordinarily deposited in a trailer pulled behind the tractor.

FACTORS AFFECTING MECHANICAL HARVESTING

19.13. Varietal Characteristics. It is estimated that about 500 different varieties of cotton were produced in the United States in 1930. By 1952 the number of varieties had dropped to 28. More important, 89 per cent of the 1952 acreage was planted with the ten leading varieties. The growing of a minimum number of varieties is important to the successful development and application of mechanical harvesters. Plant breeding can also be directed more effectively toward the development of strains most suitable for mechanical harvesting. A California law designating the one and only variety of cotton that can be produced commercially within the principal cotton-producing area of the state (San Joaquin Valley) has been a factor in promoting mechanical picking in that area.

Smith and Jones found that many commonly grown varieties of cotton are not well suited to mechanical harvesting, particularly with the stripper-type machines. They describe the ideal variety for stripping as one producing a semidwarf plant having relatively short-fruited, short-noded branches, storm-resistant* bolls borne singly but having fairly fluffy locks for good extracting, and a medium-size boll stem that can be pulled from the limbs with a force of 3 to 5 lb. Stripping a variety that produces a wide, spreading plant with numerous vegetative and fruiting branches results in low recovery of cotton and excessive field losses.

*"Storm resistance" in cotton (or stormproof) refers to a characteristic whereby the bolls tend to resist shattering from the burs when subjected to the action of wind and rain.
Harrison points out that with mechanical pickers the conformation of the bolls largely determines how well the spindles perform their task. Tests have shown no correlation between picker performance and the size of the bolls. Cotton bolls exhibiting too much of the storm-resistant characteristic, although well adapted to stripping, are difficult to pick mechanically. Tests with mechanical pickers in Texas, for example, showed a picking efficiency of only 76 per cent in a variety developed primarily for stripping, as compared with 91 per cent in a variety more suitable for mechanical picking.

The size of the plant, the type of growth, and the nature of the boll all have more influence on the efficiency of the mechanical picker than does the yield. Where the plant characteristics are suitable, a machine will pick high-yielding cotton just as efficiently as it will low-yielding cotton.

**19.14. Plant Population and Spacing.** Although several investigators feel that continuous, dense stands of cotton are essential for the best performance of mechanical cotton harvesters, the available evidence is somewhat controversial. Comparative tests at the Shafter Station (California) in hill-dropped and drill-planted plots having equal plant populations (48,700 plants per acre in 1949 tests and 35,700 in 1950 tests) showed no difference in either picking efficiency or the amount of trash collected.

Other tests at the Shafter Station in drill-planted stands ranging from 8000 to 68,000 plants per acre showed very little difference in picking efficiencies for populations above 20,000 plants per acre and only a slight decrease at the lower populations (Fig. 19.6). The Shafter results, however, represent a rather high level of picker efficiency (92 to 97 per cent), and large differences would not be expected. Under less ideal conditions, Smith and Brown observed losses of 18.0 and 8.9 per cent at populations of 11,000 and 71,000 plants per acre, respectively.

Figure 19.6 indicates that increasing the plant population has the desirable effect of raising the height of the lowest fruiting nodes on the plants. Fairbank and Smith found in 1948 that, under the conditions of their observations, 41 per cent of the total picking losses were in the lower 6 in. of the plant zone, 17 per cent in the zone above 6 in., and 42 per cent shattered. Thus, further raising of the fruiting zone would probably improve picker effi-
ciency, particularly under adverse conditions. It would also permit the picking unit to be operated at a higher level, thereby reducing wear which now results from the necessity of operating the mechanism practically in the dirt.

![Graph showing cotton picker efficiency and height of first fruiting node](image)

**Fig. 19.6.** Effect of plant population upon height of first fruiting node and upon picker efficiency. (Data from James R. Tavernetti and H. F. Miller, Jr. *California Agr. Expt. Sta. Bull. 747.)*

**19.15. Weed Control and Cultivation Practices.** The problem of effective weed control is one of the main obstacles to the complete mechanization of cotton production. Weeds and grass not only compete with the cotton plants for water and soil nutrients, but also interfere with harvesting and lower the grade of ginned cotton. Grass is extremely difficult to remove at the gin. Late cultivation, chemical weeding, and flame weeding (see Chapter 12) all offer promise in reducing weed populations to a minimum. Throwing of soil to the cotton plants ("dirt ing") during cultivation has been a common practice in many areas as a weed-control measure, but it tends to interfere with mechanical picking.

The shape of the row profiles left by the final cultivation is important in regard to mechanical harvesting. Final profiles should be uniform in height, width, and shape and should be smooth and free of clods, with the crest at the base of the cotton stalks.

Smooth row profiles (Fig. 19.7) have been made at the
Shafter Station (California) by the simple expedient of welding extension "whiskers" of 10-in. lengths of $\frac{3}{8}$-in. round rod to the wings of 22-in. sweeps used for the last cultivation.

19.16. Defoliation. When practiced in conjunction with mechanical picking or stripping, defoliation serves three valuable purposes, namely: removal of the bulk of the leaves that tend to interfere with machine harvesting, prevention of green leaf stain on the lint, and elimination of a source of dry leaf trash that is difficult to remove and becomes an increased burden on the gin.

Defoliation is described as a natural process for cotton, a crop that belongs to a group of plants that are inherently deciduous. A live cotton plant normally sheds its leaves through the influence of a number of natural causes. Drought, starvation, insect and fungus attacks, cold weather, and even light frosts may induce the plant to develop a layer of abscission cells at the base of the leaf stem, resulting in separation at that point. These causes may be considered as injuries or unfavorable alterations of vital plant processes. Chemical defoliation merely involves the infliction of an applied injury that ultimately induces the plant to cut off (abscise) its leaves.

Too much injury, whether the result of a hard freeze or a heavy application of chemicals, may kill the tissue in the abscission layer and thus prevent the vital plant process needed for defoliation. When this happens, the entire plant is usually killed and the leaves remain attached to dry up and cause excessive leaf trash in the harvested cotton.
Chemical defoliants may be applied in the form of dusts or sprays, with either ground rigs or aircraft (sprayers and dusters are discussed in Chapter 21). In regions where vegetative growth is extensive and defoliation is most needed, difficulty has been experienced in obtaining sufficient penetration and leaf coverage to insure a thorough job. Defoliants in the form of dusts are most effective when applied to plants wet with dew, because moisture is needed for their activation and because the dust sticks to the leaves better.

**EFFECTS AND COSTS OF MECHANICAL HARVESTING**

19.17. Gin Turn-Out. When seed cotton arrives at the gin, it contains dirt, hulls, moisture, leaf fragments, stems, and other trash such as weeds and grass. With stripped or snapped cotton, the burs are also included. The gin must remove this foreign matter in addition to separating the lint from the cotton seed. Thus the gin plays an important role in the over-all mechanization picture. Additional drying, cleaning, and extracting facilities have been required to properly handle the rough cotton coming in from mechanical harvesters and to keep abreast of the field mechanization program.

Gin turn-out is the ratio of the weight of recovered lint to the weight of field-run seed cotton. For example, a gin turn-out of 36 per cent means that approximately 1400 lb of seed cotton from the field are required to obtain a 500-lb bale of lint. The presence of excessive amounts of trash and other foreign matter reduces the gin turn-out, increases the cost of ginning, and lowers the final grade of the lint.

Watson reports the average gin turn-outs for cotton produced in the cotton belt during 1950 and 1951 as follows:

<table>
<thead>
<tr>
<th>Method</th>
<th>Gin Turn-Out</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hand-picked</td>
<td>37 per cent</td>
</tr>
<tr>
<td>Machine-picked</td>
<td>36 per cent</td>
</tr>
<tr>
<td>Hand-snapped</td>
<td>26 per cent</td>
</tr>
<tr>
<td>Machine-stripped</td>
<td>23 per cent</td>
</tr>
</tbody>
</table>

Results reported for other areas are in remarkably close agreement with the above figures. In general, the gin turn-out for machine-harvested cotton is a little lower than for cotton harvested by hand.
19.18. **Cotton Grades.** The various ways in which machines can reduce grades of lint below the grades that might be obtained by hand picking are summarized by Watson as follows:

1. Discoloration of lint by green leaves and by oil and grease from the machine.
2. Introduction of additional amounts of weeds, grass, and trash from the cotton plant.

3. Addition of excessive moisture (in spindle moistening) to the lint and trash, thus making trash removal at the gin more difficult and inducing graying or mildew if ginning is delayed.
4. Twisting or tangling of the lint by the spindles, thus increasing the difficulty of normal gin preparation.

Machine-picked cotton averages lower in grade than hand-picked cotton. Although improvements in harvester operation, cultural practices, chemical defoliation, and ginning facilities have helped to narrow the difference, there is on the average about a one-grade differential, as illustrated by the example in Fig. 19.8. This reduction in grade may represent a loss of 5 to 10 percent of the total value of the crop. Machine-harvested lint carries a greater amount of foreign material than does hand-harvested lint, and in addition, may be discolored. Otherwise it compares favorably with hand-picked cotton in regard to quality.
19.19. Field Loss. Field losses resulting from mechanical harvesting vary a great deal from one area to another. Often they vary from field to field. Probably the greatest single factor affecting field loss is the skill of the operator. Other factors (most of which have already been discussed) include weeds and grass in the row, poor defoliation, varieties not well adapted to the type of harvester used, poor row profile, uneven or insufficient headland for turning, improper speed relations between picking unit and forward travel (which may be due to slippery ground condition), climatic conditions, plant population, and the mechanical condition of the machine.

General experience with mechanical pickers indicates that, with careful attention to the various production and machine-operating factors involved, machine losses will usually range from 5 to 10 per cent of the yield. Under less favorable conditions, picker losses may be as great as 15 to 20 per cent, or even more.

Tests with commercial strippers in Texas during a 7-yr period indicated an average machine loss of 11.0 per cent for 17 varieties at College Station and 3.6 per cent for 14 varieties at Lubbock. In other tests, limited to varieties having storm-resistant characteristics, stripper losses as low as 2 to 4 per cent were not uncommon.

19.20. Costs of Mechanical Harvesting. The costs of mechanical picking are characterized by high overhead charges, resulting from the high first cost of the machine, and by the predominance of repair charges in the operating costs. Since mechanical strippers are less complex and have a much lower first cost than mechanical pickers, total harvesting costs per acre are considerably lower for stripping. Because of the high overhead charges for mechanical pickers, the amount of cotton harvested per season is an important factor in determining the picking cost per bale.

In a detailed cost analysis for 63 one-row mechanical pickers in California (1949), the average seasonal output per machine was 229 bales (Table 19.1). Most of the acreage was picked twice, with about 20 per cent of the total yield being obtained in the second picking. The total picking cost averaged $14.65 per acre, of which 52 per cent represented overhead charges, 30 per cent was machine operating costs, and 18 per cent was labor. When the charge of $13.47 per bale for machine field loss and
grade loss was added to the machine-picking cost, the total of $28.12 was still $16.88 less than the $45.00 cost for hand picking.

Studies in North Carolina (1948) in which one-row mechanical pickers averaged only 100 bales per machine during the season, showed an advantage of only $3.50 per bale over hand picking (after making adjustments for field and grade losses and additional gin fees). In these same studies, the total cost for mechanical stripping (average seasonal output of 46 bales per machine) was $35.75 per bale as compared with $68.84 for hand snapping.

The studies in California, North Carolina, and Mississippi (Table 19.1) all indicate that, under 1947 to 1949 conditions, the minimum economical output per season for a one-row mechanical picker (high-drum type) was about 100 to 150 bales. This rules out individual ownership of pickers on thousands of small cotton farms in the southeastern part of the United States. Joint ownership or custom harvesting seems to be the only way of completely mechanizing these areas. However, both mechanical strippers and the smaller, less expensive, mounted pickers mentioned in Section 19.3 are adaptable to somewhat smaller acreages than are the heavy-duty pickers.

Table 19.1 AVERAGE HARVESTING RATES AND ANNUAL MACHINE USAGE FROM THREE ECONOMIC STUDIES OF MECHANICAL HARVESTING

<table>
<thead>
<tr>
<th>Location</th>
<th>California 8</th>
<th>Mississippi 3</th>
<th>Texas 22</th>
</tr>
</thead>
<tbody>
<tr>
<td>Date of study</td>
<td>1949</td>
<td>1947</td>
<td>1947-1949</td>
</tr>
<tr>
<td>Type of machine</td>
<td>1-row picker</td>
<td>1-row picker</td>
<td>2-row stripper</td>
</tr>
<tr>
<td>Number of machines</td>
<td>63</td>
<td>26</td>
<td>217</td>
</tr>
<tr>
<td>Average seasonal use</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>hours per machine</td>
<td>407</td>
<td>308</td>
<td>—</td>
</tr>
<tr>
<td>days per machine</td>
<td>47</td>
<td>31</td>
<td>—</td>
</tr>
<tr>
<td>bales per machine</td>
<td>229</td>
<td>109</td>
<td>—</td>
</tr>
<tr>
<td>acres per machine</td>
<td>284 *</td>
<td>—</td>
<td>206</td>
</tr>
<tr>
<td>Average bales per day</td>
<td>7.0 (first picking)</td>
<td>3.5</td>
<td>5.5 (dry-land)</td>
</tr>
<tr>
<td></td>
<td>2.3 (second picking)</td>
<td>—</td>
<td>9.5 (irrigated)</td>
</tr>
<tr>
<td>Average acres per day</td>
<td>5.0 (first picking)</td>
<td>—</td>
<td>14.0 (dry-land)</td>
</tr>
<tr>
<td></td>
<td>6.8 (second picking)</td>
<td>—</td>
<td>10.8 (irrigated)</td>
</tr>
<tr>
<td>Average yield, pounds of lint per acre</td>
<td>700</td>
<td>—</td>
<td>211 (dry-land)</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>456 (irrigated)</td>
</tr>
</tbody>
</table>

* Includes 145 acres of first picking and 139 acres of second picking.
REFERENCES


**PROBLEM**

19.1. At a forward speed of 3 mph, how many revolutions does each spindle of a cotton picker make while in the picking zone:

(a) For a chain-belt arrangement in which each spindle has a rotational speed of 1200 rpm and remains in the picking zone during 30 in. of forward travel?

(b) For a drum-type arrangement in which each spindle has a rotational speed of 2700 rpm at 2½ mph and is in the picking zone during 6 in. of forward travel?
CHAPTER 20

Harvesting of Root Crops

20.1. Introduction. According to the United States Crop Census, root crops were grown on approximately 5 million acres in 1951. Peanuts, potatoes, sugar beets, sweet potatoes, and onions, in the order listed, constitute the principal root crops produced.

Hand labor has, in the past, been used almost entirely for the harvesting of root crops, and it continues to play a major role. In recent years, however, new machinery developments have provided partial or complete mechanical harvesting of certain crops.

PEANUT HARVESTING

20.2. Harvesting Methods. Peanut harvesting involves the operations of digging, shaking to remove adhering soil, windrowing or stacking, and picking (removal of peanuts from vines). The prevailing practice in the past has been to cut the roots below the tuber zone with a half-sweep type of blade and then to lift, shake, and windrow the vines by hand in preparation for curing and picking.

In recent years, however, considerable effort has been directed toward the mechanization of peanut harvesting. Approximately 75 per cent of the 1951 Texas crop, for example, was threshed by combines. Improved methods and equipment in other regions offer promise of reducing the harvest labor to about 4 to 5 man-hr per acre as compared with 30 to 50 man-hr for hand harvesting.

20.3. Digging, Shaking, and Windrowing. In the southeastern and southwestern peanut-growing areas of the United States, specially designed digger blades of the type shown on the
tractor in Fig. 20.1 are used to cut the tap roots and lift the peanuts. The rods attached to the blades and to the standards help raise the tubers out of the loosened soil and move the two rows toward each other in the center. In the Virginia-North Carolina area, the peanuts are larger and the stems are more easily broken. Consequently, diggers for this area lift the soil and plants onto an open conveyor arrangement that shakes out the soil, somewhat like a potato digger.

![Fig. 20.1. A peanut digger and shaker-windrower. (J. I. Case Co.)](image)

Side-delivery hay rakes have in the past been used rather extensively for windrowing, but the results have been rather unsatisfactory because of excessive losses of peanuts, tight and tangled windrows, too much soil left on the tubers, and high upkeep on the rake.\(^6,^{17}\) Combination shaker-windrowers of the type illustrated in Fig. 20.1 have been developed in recent years and have largely replaced side-delivery rakes in peanut harvesting. The chain-and-slat conveyor picks up the plants, elevates them to a height of about 6 ft, shaking off most of the soil in the process, and then discharges them onto a cross-conveyor for windrowing.

20.4. **Threshing or Picking.** Peanuts are generally removed from the vines either with cylinder-type machines similar to grain threshers and combines or with pickers that employ the carding or combing principle. Cylinder-type machines may be equipped with a conventional spike-tooth cylinder from which part of the teeth have been removed, or spring teeth may be used on the cylinder and in the concaves. Cylinder peripheral speeds of 1200
to 2000 fpm are typical. The separating and cleaning units are ordinarily similar to those of a grain combine, but revolving stemmer saws are added in the bottom pan of the cleaning shoe to remove the stems from the peanuts.

The Georgia Coastal Plain Experiment Station, in cooperation with the USDA, has recently developed a windrow-pickup type of peanut harvester that picks the peanuts from the vines with two intermeshing spike-tooth cylinders. Separation is then accomplished by means of a series of three spring-tooth cylinders operating over expanded-metal concaves, after which the peanuts fall through an air blast and go to the stemmers.

A peanut combine employing the carding principle is illustrated in Fig. 20.2. The pickup conveyor moves the plants through a series of steel springs that comb the peanuts from the vines. The peanuts then pass through an air blast for cleaning and through the stemmer saws.

The usual procedure is to allow the peanuts to cure or partially cure in stacks or windrows in the field before picking. In some areas, weather conditions at harvest time do not permit complete
windrow curing. The partially cured peanuts are picked (threshed) after 3 to 10 days in the windrow. They can then be dried successfully by forced convection with unheated air or, in some areas, by merely stacking in the field or in an open shed.

Peanuts can be combined from the windrow immediately after digging, but artificial drying is necessary.

**POTATO HARVESTING**

Mechanical potato harvesters may be classified as elevating diggers and combined digger-pickers (potato combines). With either type of machine, final separation of the tubers from clods, vines, stones, and other foreign material is done by hand. Both types are available in one-row and two-row sizes. Power for operating the conveying mechanism is generally furnished through the tractor pto in the case of diggers, but potato combines often have separate engines.

### 20.5. Diggers

Present-day power-operated diggers have shaker chains that elevate the potatoes and sift out most of the soil, dropping the tubers and remaining debris (vines, clods, occasional stones, etc.) from the rear of the conveyor onto the ground. The potatoes are then picked up by hand and placed in bags, boxes, barrels, or other containers. Sometimes a roller is mounted beneath the rear of the conveyor to firm and smooth the soil on which the potatoes are to be dropped. Where rank vine growth is encountered, flail-type rotary beaters (see Section 9.15) are often used to remove the vines ahead of digging.

Various types of blades or shovels are employed for digging or scooping up the soil and potatoes. One-row diggers are ordinarily equipped with pointed blades. A two-row digger may have either a separate, pointed blade for each row (Fig. 20.3) or a continuous, straight blade across both rows. The blade is operated at a depth just below the tuber zone.

The blade delivers the entire mass of potatoes and surrounding soil onto a rod-chain type of elevating conveyor. The links of the elevator chain are generally offset alternately up and down from the plane of the chain (Fig. 20.3), but they may also be straight. Two-row diggers have a separate elevator for each row (Fig. 20.3).

Elongated or oval-shaped idler sprockets support the elevator chain and agitate it with an up-and-down motion to aid in sifting
out the soil. Some diggers have provision for changing the linear speed of the elevator to suit the operating conditions. The higher speeds give a better cleaning action, but the potatoes are more likely to be bruised.\textsuperscript{15}

\begin{figure}[h]
\centering
\includegraphics[width=\textwidth]{fig203.png}
\caption{\textit{Left:} Front view of a two-row potato digger that consists essentially of two single-row machines. \textit{Right:} Rear view of a two-row digger with no partition between the two elevators. (Deere & Co.)}
\end{figure}

\subsection*{20.6. Potato Combines.} Potatoes harvested with a combined digger-picker are lifted with the soil in the same manner as with a standard digger. But the potatoes, instead of being dropped back onto the ground to be picked up by hand, pass from the elevator onto a sorting conveyor. Workers standing on platforms on each side of the conveyor, and in some instances along the lower sections of the elevator, remove vines, stones, clods and other foreign material that has not been separated mechanically. The potatoes are then discharged from the sorting conveyor into sacks or barrels or are elevated into bulk-handling trucks. In some cases the sorting is done on a separate conveyor-sacker unit trailed behind a conventional digger. The over-all length of this combination is a disadvantage in turning at the ends.

From three to nine hand pickers are needed on a two-row machine, the number depending upon the amount of foreign material to be removed and the forward speed. Martin and Humphrey\textsuperscript{9} recommend an operating speed of 1\textsuperscript{1/2} mph in fields yielding 200 to 400 sacks per acre. Removal of the vines with a rotary beater prior to harvesting simplifies the sorting job, as well as permitting the skins to "set" so the potatoes are not so easily bruised.
Although potato combines are not well suited to conditions that result in the formation of a large number of clods in the digging operation, they are rapidly becoming popular in some areas. In Idaho, for example, 72,000 acres (half of the state acreage) were harvested with potato combines in 1949, using either digger-sacker units or bulk handling. Either of these systems eliminates the tedious, hard work of picking potatoes from the ground by hand,

![Image](image)

**Fig. 20.4.** A two-row potato combine for bulk handling. Three men can ride beside each horizontal sorting conveyor, and three more behind the cross-conveyor, to remove rocks, clods, and vines. (J. W. Martin and E. N. Humphrey. *Agr. Eng.,* May, 1951.)

... reduces labor requirements, and reduces over-all harvesting costs. Bulk handling eliminates all hand lifting and results in the lowest labor requirements. A disadvantage of bulk handling is that delay of any one piece of equipment halts the entire operation.

Although a number of commercial models of potato combines are now available, the majority of these machines have in the past been built by farmers or in local shops. An additional conveyor is added at the rear of the regular elevator to increase the mechanical separating area and to provide a place for hand sorters to work. For bulk handling, a cross-conveyor and elevator assembly is also added at the rear (Fig. 20.4).

To minimize damage to the tubers, Martin and Humphrey recommend the use of rubber-covered links in all elevator chains and suggest that the conveyor extension be of the rubber-roller type as illustrated in Fig. 20.4. Each extension consists of a series...
of six to twelve 3-in. diameter rubber rollers (on 1-in. steel shafts) spaced on 4\(\frac{3}{4}\)-in. centers and driven at a peripheral speed of about 100 fpm. Free-fall distances should be as short as possible (preferably not over 6 in.) and rubber padding should be used on all surfaces against which the potatoes may strike.9

When potatoes are handled in bulk, the floor and sides of the truck where loading begins should be padded. Care must be exercised to first build the load to the full height in this padded area and then position the truck so that subsequent potatoes always fall on a previously built mound, thereby reducing the distance they fall.

20.7. Sweet Potato Harvesters. The harvesting of sweet potatoes requires considerable hand labor. Machinery available for digging has not been entirely satisfactory because of numerous problems such as the disposition of rank vine growth, susceptibility of the tubers to bruising, poor underground distribution of tubers, and difficulty of lifting all the tubers to the surface.9

Sweet potatoes produce rank top growth that must be cut and moved, at least out of the immediate row, before harvesting can progress. A simple and satisfactory cutting device developed at the South Carolina Agricultural Experiment Station consists of two mower knife sections mounted vertically, 5 in. apart, on the underside of a spring-loaded slide (downward force on the slide is approximately 100 lb).10 The knives cut the vines vertically on each side of the crown. Shielded coulters represent another arrangement for cutting the vines. After they have been cut, the vines are crowded to one side by rod-type vine turners, disk wheels, or finger-type rake wheels. A V-crowder arrangement with cutting blades was developed at the Louisiana Agricultural Experiment Station and is commonly used in that area for cutting and moving the vines from the row.10

The usual procedure after cutting the vines is to plow the tubers to the surface with either a moldboard plow or a lister (middle buster) and then to pick them up by hand. The number of potatoes exposed on the surface by either method may vary from 50 to 90 per cent, depending upon the root distribution, soil type, and moisture conditions.10 A certain amount of hand “scratching” is necessary to recover the potatoes not visible on the surface.

Park and his associates10 found in 1951 that an experimental rod-type digger (Fig. 20.5), exposed 93 per cent of the tubers.
as compared with 73 and 83 per cent, respectively, for the plow and the lister. There was no great difference in the degree of bruising by the three devices.

The rod-type digger consists of a special, wide-base lister with short moldboards for plowing the potatoes out. Bent rods, extending back from the lister and from side brackets on the frame, receive the soil and potatoes from the lister. As the machine moves down the row, the soil is sifted through the rods, forcing the tubers to the surface. In the above tests the mower-section vine cutter previously described was mounted beneath the tractor differential. The vines were moved out of the way of the digger by the angled disk wheels shown ahead of the side rods in Fig. 20.5.

**SUGAR BEET HARVESTING**

Before the introduction of mechanical harvesters, beets were loosened from the soil and lifted slightly by means of a subsoiler arrangement having horizontal wings attached to the base of the standard, or with special lifter blades. Each beet was then picked up by hand and its top portion (crown) cut off with a heavy
knife. The topped beets were placed in windrows and later loaded by hand into trucks.

Mechanical sugar beet harvesters were first used to an appreciable extent in 1943, when approximately 3 per cent of the United States crop was harvested by machine. Eight years later it was estimated that 75 per cent of the nation's crop was harvested mechanically.

The essential operations performed by a mechanical beet harvester (Fig. 20.6) are (a) cutting off the unwanted top portion of the beet at the desired height, (b) proper disposition of the tops to prevent interference with the other steps in the harvesting operation, (c) loosening the beets from the soil, (d) elevation of the beets and separation from clods and other foreign material, and (e) deposition of clean beets in a truck or trailer.

Fig. 20.6. Schematic arrangement of a sugar beet harvester employing in-place topping. The inset shows the converging wheels that loosen and lift the beets. The elevator discharges the topped beets into a special, two-wheel trailer, from which they are mechanically unloaded into a truck. (Deere & Co.)
20.8. Topping. This operation may be performed while the beets are still in the ground (in-place topping), or it may take place in the machine after the beets have been lifted. The tops are generally moved to one side onto the harvested area, either with a conveyor or by means of flinger wheels equipped with spring teeth (Fig. 20.6) that throw the tops against a curtain mounted to one side of the row. In regions where the tops are not saved for livestock feed, the vegetative top growth is sometimes removed with rotary heaters (see Section 9.15) before harvesting.

The ideal thickness of crown to be removed in the topping operation is related to the size of the beet. Topping standards set up by the sugar beet processors for hand topping specified that beets up to 3¾ in. in diameter be topped at the lowest leaf scar and larger beets ¾ in. above the lowest leaf scar.\(^\text{12}\) As a basis for approaching this topping standard with machine topping, Powers\(^\text{12}\) made measurements on individual beets in fields throughout several western states.

Fig. 20.7. Relation of sugar beet heights and topping planes to beet diameter. (John B. Powers. *Agr. Eng.,* August, 1948.)

Curves A and B in Fig. 20.7 indicate the results of these meas-
urements, showing the height of the top of the beet and the height of the lowest leaf scar in relation to the greatest diameter of the beet. Each plotted point represents the average of measurements of several hundred beets. The difference between curves $A$ and $B$ is the crown thickness. Curve $C$ indicates the height of the topping plane above ground level if the hand-topping standards are to be met. Curve $D$ represents an approximation of curve $C$ that can readily be obtained with a mechanical linkage.

Fig. 20.8. (a) Top view of a two-blade lifter and a kicker-wheel cleaning trough, for an in-place topper. (b) A variable-cut, in-place topping unit with a notched-disk cutter and a shoe-type finder. (International Harvester Co.)

To satisfy curves $A$ and $D$, a gaging device (finder) that rides over the tops of the beets must lift the knife 0.71 in. for each inch the finder is raised. A topper based on this variable-cut principle was built at the University of California, and its performance approached the precision of hand topping. Some commercially available harvesters with in-place topping now use the variable-cut principle (Fig. 20.8b). Machines that top the beets after they have been lifted (Fig. 20.10), as well as some machines with in-place topping (Fig. 20.6), remove approximately the same thickness of crown from all beets, regardless of size. The thickness of crown removed can, however, be adjusted for the best average condition in a particular field.

20.9. Gaging for In-Place Topping. Various types of finders are employed, including sliding shoes (Fig. 20.8b), power-driven wheels (Fig. 20.6), and driven belts. Field measurements
on high beets in wet fields have indicated that the horizontal component of the force exerted on the beet by the finder as it raises the topping mechanism should not exceed 60 lb. Otherwise the beet may be overturned, particularly in moist or sandy soils. Driven wheels or belts reduce the tendency of the finder to overturn high beets.

At normal harvesting speeds a free fall of the topping mechanism, after topping a high beet, is too slow to properly top a closely following low beet. The rate of fall can be accelerated with springs, but the downward force must not be so great that beets will be overturned in their effort to lift the finder. Since the rate of fall is governed by the ratio of the effective spring force to the inertia of the moving parts, a limitation is imposed upon the permissible weight of the topping unit. For a comprehensive treatment of the relations between effective dynamic weight, effective spring thrust on the gaging surface, and minimum skip distance between beets at a given forward speed, see references 11 and 12.

20.10. Cutting. It has been found that drawing a knife through a beet supported only below the cutting plane (as by the soil) will frequently cause breakage of the root. The break usually occurs when the knife is about two-thirds of the way through the beet and takes place along a plane sloping downward and forward from the knife at an angle of about 45°. The failure occurs along this 45° plane, as explained by Powers, because the beet is relatively strong in shear and weak in tension. Laboratory tests have indicated that breakage occurs when the thrust of the cutting unit exceeds 7.7 lb per in. of beet diameter.

To prevent breakage, the cutting thrust may be reduced or an opposing force may be applied to the beet crown by means of a driven finder. The cutting thrust has been found to be approximately proportional to the effective included angle of the knife cutting edge. With a fixed knife perpendicular to the direction of motion, root breakage can be prevented by grinding the knife to an included angle of not more than 5°. The resulting edge, however, is too fragile to be practical (12° was found to be the minimum practical angle).

The effective included angle of the knife edge can also be reduced by imparting a component of velocity parallel to the blade. This can be accomplished by mounting a fixed knife at an acute
angle to the row, oscillating the knife longitudinally (i.e., across the row), or cutting with the edge of a rotating disk. Harvesters that top the beets in the machine ordinarily have overlapping, rotating disks (Fig. 20.10), whereas all three arrangements have been employed for in-place topping (Figs. 20.6 and 20.8b). Mounting the knife at an angle, or the use of a rotating disk, increases the amount of space required between beets in the row for proper topping of successive high and low beets.

20.11. Digging, Elevation, and Cleaning of Beets Topped in the Ground. Beets that have been topped in place are loosened from the ground and lifted onto a conveyor by specially designed blades or lifting points such as those illustrated in Fig. 20.8a, by means of pronged, converging wheels (Fig. 20.6), or with other similar arrangements. The two blades of the first type of unit operate along either side of a beet row at a depth of perhaps 6 to 8 in., loosening the soil and raising the beets by a wedging action. The prongs on the machine illustrated in Fig. 20.6 are forced into the ground and converge as they move rearward, thus grasping the beets and lifting them. A paddle wheel operating between the two lifter wheels at the rear transfers the beets onto a conveyor.

Shown in Fig. 20.8a (and also in Fig. 20.6) is a typical arrangement of a cleaning trough and kicker wheels for separating the beets from the soil. From the cleaning trough, the beets are elevated and further cleaned by a rod-chain type of conveyor and are then discharged into a special trailer. Under cloddy conditions, when separation in the cleaning trough is not complete, the beets may be delivered to a horizontal sorting belt above the trailer. The beets are picked from the belt by hand and dropped into the trailer, while the clods are carried to the rear and discharged onto the ground. Two hand sorters can handle 5 to 6 tons per hr.

20.12. Harvesters That Lift the Beets before Topping. Machines of this type lift the untopped beets either by gripping their top growth or by impaling the beets on a spiked wheel. With either method the beets are lifted free of the soil and the problem of separating the roots from large clods occurring in heavy soils is eliminated or minimized. The beets are topped in the machine after being elevated.
The first complete sugar beet harvester to become available in the United States (improved models are currently available) was a trailed machine that lifted the beets by their tops. In this machine, a pair of inclined chains, held together by spring-loaded idlers, grasps the tops as a single-blade lifter cuts the tap roots. The chains move rearward at about the same speed that the machine moves ahead, thereby giving practically a vertical lift to the beets. Machines of this type can be used only under conditions where the tops have adequate strength to support the weight of the root.

A harvester that lifts the untopped beets by means of a spiked wheel is illustrated in Figs. 20.9 and 20.10. Both trailed machines (mostly two-row) and mounted, one-row units have been built, the mounted type representing the more recent design and having a considerably smaller spiked wheel.

On the mounted model (illustrated), the spiked wheel is 30 in. in diameter and has a rim 8 in. wide with four staggered rows of $\frac{5}{16} \times 3\frac{3}{4}$ in. curved spikes spaced 2 in. apart in the rows. The wheel and frame assembly are mounted on the right-hand side of a wheel-type tractor in such a manner that the spiked wheel is free to rise over the high-crown beets and drop to the smaller ones. The spiked wheel is chain driven from the rear axle of the tractor, at a peripheral speed slightly less than the forward speed of the machine.

A two-blade lifter (Fig. 20.9) loosens the beets as the spikes pierce the crowns. After the spiked wheel has elevated the beets, inclined steel stripper bars mounted between the rows of teeth (Fig. 20.10) raise the beets up off of the spikes far enough to permit topping at the proper level with a pair of overlapping, rotating disks. Both the disks and the strippers are adjustable to control the thickness of top removed.

After topping, the strippers remove the tops completely from
the spiked wheel onto a cross-conveyor for windrowing. A spiral-rod auger transfers the topped beets to a rod-chain type of conveyor, with a rolling and tumbling action that removes most of the dirt. The conveyor deposits the beets in a specially designed two-wheel trailer, from which they are periodically unloaded into a truck by means of another conveyor.

Fig. 20.10. Spiked-wheel harvester, showing how the beets are topped by two rotating disks. (Blackwelder Mfg. Co.)

20.13. Onion Harvesting. Onion growing involves considerable stoop labor, with hand labor for the harvesting operation alone often accounting for as much as 50 per cent of the total production costs. Thus, successful mechanical harvesting would result in substantial monetary savings as well as reducing the labor requirements.

The principal requirements for a mechanical onion harvester are that it perform the functions of digging, lifting, topping, and sacking. A harvester developed at the University of California has shown promise of meeting the above requirements in handling onions (as well as flower bulbs). The machine loosens the
onions and, with two inclined, round, rubber belts held together by spring-loaded idlers, lifts them by their tops. The topping is done by a pair of rotating disks located near the upper end of the elevating section. The topped onions drop onto an elevator to be conveyed to a sacking platform, and the tops are discharged onto the ground.

REFERENCES


CHAPTER 21

Spraying and Dusting

21.1. Introduction. Sprayers and dusters have been used for many years to apply chemicals for the control of agricultural pests. Although the early machines were designed primarily for orchards, the introduction, during recent years, of new pest-control chemicals such as 2,4-D has resulted in the development of new and improved types of equipment and in a much broader field of application, particularly in regard to low-pressure and low-volume spraying of field crops.

Present-day agricultural pest-control equipment includes (a) high-pressure orchard sprayers (400 to 800 psi), (b) field-crop sprayers (usually 20 to 200 psi, but sometimes higher, depending upon the application), (c) blower-sprayers, which use an air blast as a carrier for low-pressure sprays and were developed primarily for orchards, (d) dusters (ground rigs), and (e) aircraft equipped for spraying or dusting. Thermal aerosol generators, which atomize the chemicals by means of steam, hot gases, or direct impingement upon a hot plate, have only limited application for agricultural pest control.

Among the many uses for the various types of spraying and dusting equipment are the following:

1. Application of insecticides to control insects on plants.
2. Application of fungicides to control plant diseases.
3. Application of herbicides to kill weeds, either indiscriminately or selectively (see Section 12.2).
4. Application of preharvest sprays to defoliate or condition crops for mechanical harvesting (see Chapters 17 and 19).
5. Application of hormone (growth-regulating) sprays to increase fruit set or prevent early dropping of the fruit.
6. Application of sprays to thin fruit blossoms.

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7. Application of plant nutrients (sprays) directly to the plant foliage (see Section 13.3).

21.2. Particle Size in Relation to Effectiveness. The size of particles is important in relation to such factors as the penetration and carrying ability obtained with hydraulic sprayers (Section 21.22), the efficiency of "catch" of sprays or dusts by plant surfaces, uniformity and completeness of coverage, effectiveness of individual particles after deposition, and drift. Dusters have no direct influence on particle size, except as they influence agglomeration of particles during application, but sprayers can be made to produce droplets covering a wide range of average sizes. Present atomizing equipment, however, does not produce a uniform size of droplets under a particular set of conditions but gives a wide band or spectrum of sizes (Section 21.9).

Considerable work has been done in regard to the most effective size of aerosol particles for control of insects in flight, but little positive information is available about the best size of individual particles for pesticides applied to plant surfaces. The optimum particle size for many insecticide applications is believed to be related to the size, state of development, and species of insect. In applying fungicides and certain insecticides (as for scale insects), continuous coverage seems to be most important. For 2,4-D and other sprays of the translocation type, large droplets (100 to 200 microns * average diameter) give satisfactory results and minimize drift hazard.

Particle size is important in relation to the dynamic "catch" or impingement of sprays or dusts upon foliage when an air stream carries the particle. As an air stream approaches an obstruction, all particles of a particular size will be removed from a central portion of the approaching stream but all particles of the same size that are outside of this central width will be carried around the obstruction. The efficiency of dynamic catch is defined as the percentage of the total frontage of approaching air stream (having the same cross-sectional area as the obstruction) that is cleaned of droplets of a particular size. Thus, a catch efficiency of 100 per cent for particles of a particular size means that air sweeping through the foliage would be stripped of all particles of that size only in a cross-sectional area equal to that

* One micron = \(1 \times 10^{-6}\) meters or approximately \(\frac{3}{40,000}\) in.
presented by the foliage. Increasing the size of the particles or the velocity of approach increases the percentage of catch because of the greater momentum of the particles (Fig. 21.1). The catch also varies inversely with the size of the obstruction (broken-line curves in Fig. 21.1) and to some extent with the shape of the object.32

![Figure 21.1](image)

Fig. 21.1. Effect of droplet size and velocity of approach upon the dynamic catch of two sizes of cylinders. (F. A. Brooks. *Agr. Eng.*, June, 1947, and additional calculations based upon reference 27.)

In addition to the frontal deposits by impingement, particles are deposited on the back side of obstructions, as a result of turbulence and eddies. Yeomans and his associates33 found that the smaller particles tend to deposit on the backs of obstructions and the larger particles on the fronts.

Uniformity and completeness of coverage are affected by the total number of droplets available per unit area. For a given rate of application, the number of droplets is inversely proportional to the cube of the diameter. Thus an application of 1 gal of liquid per acre would give only 1150 drops per sq in. of land area if they were all 100 micron in diameter, but would give 9200 50-micron drops per sq in. or 144,000 20-micron drops. Evaporation losses are, however, greater with the smaller droplets, because of the increased total surface area.

21.3. Drift of Dusts and Sprays. Drift is essentially a function of the rate of fall of particles in relation to the horizontal
velocity, which in turn is related to particle size and local climatology. Figure 21.2 shows the relation of droplet size to the distance water droplets are carried while settling 10 ft in straight air flow having a uniform horizontal velocity of 3 mph. Note that particles smaller than 10 microns can be expected to drift several miles, whereas 100-micron particles drift only 50 ft under these conditions.

Drift may be a very serious problem, particularly when applying hazardous materials. Since many of the commercially available dusts have average particle sizes of only 2 to 30 microns (and a wide range of sizes within a given sample), these materials are especially susceptible to drift. Tests have indicated that over 70 per cent of the dust applied by an airplane may drift away from the treatment area. In addition to representing inefficient use of material, drift may cause serious damage either by injury to susceptible crop plants in nearby fields or as a result of poisonous deposits left on edible plant parts. Although the problem of drift is most acute for aircraft dusting, it is also very real for ground-rig dusting and for spraying with either aircraft or ground rigs.

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### Fig. 21.2. Relation of drift to particle size, for water droplets. (F. A. Brooks. *Agr. Eng.*, June, 1947.)

<table>
<thead>
<tr>
<th>Diameter of droplets, microns</th>
<th>Distance drifted, feet</th>
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<td>5</td>
<td>10,000</td>
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<tr>
<td>3</td>
<td>20,000</td>
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- **Moderate rain**
- **Light rain**
- **Drizzle**
- **Mist**
- **Cloud**
- **Coarse aerosol**
- **Sea fog**

Vertical settling distance = 10 ft
Average air velocity for the 10-ft depth = 3 mph

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ATOMIZING DEVICES

21.4. Types of Nozzles. In general, devices for the atomization of liquids utilize one or more of the following principles: 21

1. Pressure or hydraulic atomization, which depends upon liquid pressure to supply the atomizing energy. The liquid stream from an orifice or a nozzle is broken up by its inherent instability and its impact upon the atmosphere or by impact upon a plate or another jet.

2. Gas atomization, in which the liquid is broken up by a high-velocity gas stream, the breakup occurring either entirely outside of the nozzle or within a chamber ahead of the exit orifice.

3. Centrifugal atomization in which the liquid is fed under low pressure to the center of a high-speed rotating device, such as a disk or cup, and is broken up by centrifugal force as it leaves the periphery.

The principle of centrifugal atomization is not common in agricultural sprayers but can be applied to produce uniform droplets of controlled sizes for laboratory studies of the effectiveness of different sizes of droplets.30 The gas-atomizing principle is employed in blower-sprayers equipped with shear-plate devices (Section 21.25). Both field sprayers and high-pressure orchard sprayers rely entirely upon the principle of hydraulic atomization. Most aircraft sprayers and blower-sprayers equipped with spray orifices depend mainly upon hydraulic atomization, although gas atomization is undoubtedly involved to a small degree.

Hydraulic-type spray nozzles may be classified as hollow-cone, solid-cone, fan, impact, and solid-stream. The impact and solid-stream types have been used to a limited extent on aircraft sprayers but very little on ground rigs.

21.5. Hollow-Cone Nozzles. In this type of nozzle the liquid is fed into a whirl chamber through a tangential side-entry passage, or through fixed spiral passages, to give it a rotating motion. The orifice is located on the axis of the whirl chamber and the liquid emerges in the form of a hollow conical sheet, which then breaks up into droplets.21 Three common methods for obtaining the rotary motion are illustrated in Fig. 21.3a, b, and c. All of
them are found in agricultural sprayers. With any of these types of nozzles, various size combinations of whirl devices and orifices are generally available. The core-insert device is found mainly in small-size nozzles. (This type of nozzle is common in oil burners.)

Figure 21.4 illustrates adjustable and non-adjustable, disk-type, hollow-cone nozzles for high-pressure orchard spraying.

In the adjustable type (primarily single-nozzle hand guns), a fixed whirl plate may be bypassed (as in Fig. 21.4) or the entire whirl plate may be moved axially by a plunger. Deepening the whirl chamber results in a coarser spray and narrower discharge angle, the extreme condition being a solid stream.

21.6. Solid-Cone Nozzles. The solid-cone nozzle is a modification of the hollow-cone type and provides complete coverage of an area at close range. The construction is essentially the same as that of hollow-cone nozzles except for the addition of an internal axial jet of the proper size (Fig. 21.3b), which strikes the rotating liquid just within the discharge orifice. The breakup is largely due to this impact and the resulting turbulence, with the
liquid apparently leaving the orifice in droplet form rather than as a sheet.\footnote{21}

21.7. **Fan-Type Nozzles.** By means of a milled cut or channel across the outside face of the orifice plate, and sometimes an elongated orifice in addition to the slot (or by means of two inclined, impinging jets), the liquid is caused to emerge as a flat, fan-shaped sheet, which is then broken up into droplets.\footnote{21} Nozzles of this type (Fig. 21.3c) are used extensively for low-pressure weed spraying and in other similar applications. Discharge rates from available sizes of fan-type nozzles, at the usual pressures of 20 to 100 psi, range from about 0.02 to 3 gpm. The smallest sizes, however, are not very satisfactory for field spraying because of clogging problems.

21.8. **Nozzle Flow Rates and Spray Angles.** In general, the flow rate for a particular nozzle is about proportional to the square root of the pressure. For nozzles with geometrically similar passages, the discharge rate is about proportional to the orifice area. Because core-type, hollow-cone nozzles and the smaller sizes of disk-type, hollow-cone nozzles have an appreciable pressure drop in the whirl devices, the flow rate increases more slowly than the orifice area if the whirl openings are not enlarged proportionately. With some nozzles the pressure drop through the whirl device is low enough that it does not have any great effect on the flow rate.
Nozzles on field sprayers generally have spray angles of 60 to 90° (included angle of cone or fan), whereas high-pressure orchard-sprayer nozzles have smaller angles (except for close-range work). Spray angles of most types of nozzles, with the exception of the solid-cone, core type, decrease considerably as the pressure is reduced in the range below 50 to 75 psi (Fig. 21.5).

Operating pressures below about 20 psi are undesirable because of poor performance of nozzles.

It is customary for nozzle manufacturers to provide tables giving volumetric flow rates and spray angles for their nozzles at various pressures, usually for water. Some manufacturers have a nozzle numbering system that indicates the flow rate and spray angle for a particular liquid at a standardized pressure, others assign numbers that indicate the orifice diameter, and still others have no direct correlation between the nozzle number and the nozzle characteristics.

21.9. Droplet Sizes. In general, the degree of atomization depends upon the characteristics and operating conditions of the nozzle and upon the characteristics of the liquid being atomized. The average droplet size from a given nozzle is decreased if either the surface tension or the density of the liquid is reduced,
but it is not greatly affected by viscosity in the range from one to ten times the viscosity of water.\(^{21}\)

Limited tests indicate that, within the range of conditions normally encountered with either field sprayers or high-pressure orchard sprayers, the average droplet size produced by a given nozzle handling a particular fluid varies about as the inverse square root of the pressure.\(^4,^{14}\) For example, the average droplet size for a fan-type nozzle might be increased from 75 to 150 microns by reducing the pressure from 100 to 25 psi. Any hydraulic spray nozzle, however, produces a wide spectrum of droplet sizes. Increasing the average droplet size does not eliminate small droplets, but merely shifts the emphasis, as indicated in Fig. 21.6.

Atomization is also influenced by the size of the nozzle orifice, the type of whirl device and size of openings, and other design features. There is little information available to indicate the exact nature of these relationships, although Akesson's tests\(^4\) with two sizes of fan-type nozzles and two sizes of core-insert hollow-cone nozzles (the hollow-cone nozzle with the smaller orifice also having smaller whirl openings) showed that in each

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**Fig. 21.6.** Range of droplet sizes produced in one region of the spray from a fan-type nozzle, at three different pressures. (Data from Norman B. Akesson.\(^4\))
case the nozzle with the smaller orifice gave the smaller average droplet size. French, on the other hand, found that for a given high-pressure orchard spray gun the droplet size was independent of the orifice size when the same whirl device was used with all orifice disks. In this case the effect of decreasing the orifice size (greater atomization) tended to be counteracted by the reduced whirl velocity accompanying the smaller discharge rates.

21.10. Determination of Droplet Sizes and Uniformity of Coverage. Uniformity of spray distribution, as well as some indication of relative droplet sizes, can be determined in the laboratory or in the field by means of sensitized paper or by spraying colored liquid onto plain paper. Another method for determining uniformity of application involves spraying colored liquid onto distributed metal or glass sheets from which it can be washed and analyzed quantitatively by means of colorimetry or spectrophotometry. Glass slides coated with a material such as Dri-film, soot, or magnesium oxide can be used for checking coverage as well as in determining droplet sizes with a microscope. Correction factors must be applied, however, to determine the original spherical diameters from the observed sizes of the flattened impressions.

For direct measurement of water droplets as spheres, the oil-immersion method is satisfactory. The water droplets are caught in a shallow dish containing an oil with low surface tension. Since the water has a somewhat greater specific gravity than the oil and is not miscible with it, the droplets go to the bottom of the dish and remain as approximate spheres.

Uniformity of coverage on plant surfaces can be checked by adding fluorescent dyes to the spray or dust and then viewing the surfaces with a fluorescent light (ultraviolet, with filter) after dark. A permanent record can be obtained by means of ultraviolet photography.

PUMPS FOR SPRAYERS *

21.11. Piston or Plunger Pumps. Positive-displacement pumps found on sprayers include piston or plunger, rotary, and diaphragm types. These pumps are all self-priming, and they

*For a more thorough discussion of pumps, see Henderson and Perry or other suitable references.
all require automatic (spring-loaded) bypass valves to control the pressure and to protect the equipment against mechanical damage if the flow is shut off. Piston or plunger pumps are well suited for high-pressure applications such as high-pressure orchard sprayers and small multipurpose sprayers that are designed for both high- and low-pressure spraying. Because they are more expensive than other types, occupy more space, and are heavy, piston pumps are not popular for low-pressure sprayers. As with any reciprocating pump, a surge chamber is necessary to smooth out pressure pulsations.

High-pressure sprayer pumps are equipped with either ball-type or disk-type valves, usually made of hardened stainless steel. Pistons may be fitted with plunger cups of specially molded rubber-and-fabric composition, which seal against the cylinder walls, or each plunger may operate through stationary packing that serves as the wall of the displacement chamber. The cylinder walls, especially on pumps with plunger cups, are usually either porcelain lined or made of stainless steel, to retard corrosion and wear from abrasive materials.

The volumetric efficiency of a plunger pump in good condition is generally high (90 per cent or more), and the discharge rate is essentially a direct function of crank speed and volumetric displacement. Crank speeds on sprayer pumps are generally in the order of 100 to 200 rpm, with rated outputs between 10 and 60 gpm being typical. Mechanical efficiencies may range from 50 to 90 per cent, depending upon the size and condition of the pump.

21.12. Rotary Pumps. Rotary pumps are popular for low-pressure sprayers, the most common types being gear pumps (internal or external) and vane or roller pumps (Fig. 21.7). Nylon is a common material for the rollers in roller pumps, although steel and carbon are also used. In operation, the rollers are held against the case by centrifugal force. Rotary pumps of these types are compact and relatively inexpensive, and can be operated at speeds suitable for direct connection to the tractor pto. Their pumping action depends upon the maintenance of close clearances between the housing and the gears or impellers. Although they are classed as positive-displacement pumps, there is a moderate decrease in flow as the pressure is increased, because of leakage past the rotors.
Pressures above 100 psi are not ordinarily recommended for rotary pumps when pumping non-lubricating liquids. Gear pumps are unsatisfactory for pumping suspensions of wetttable powders or any other abrasive materials, because of rapid wear and short life. Roller pumps tend to be somewhat better in this respect but still have rather short lives. In tests with a mixture of kaolin clay and water (80 psi discharge pressure) Garton and Morgan found that gear-type and sliding-vane pumps lasted less than 20 hr. The output from roller pumps (with nylon rollers) decreased by 20 to 30 per cent during 130 hr of operation in these tests, and the seals started to leak during this time.

21.13. Centrifugal Pumps. Because pumps of this type depend upon centrifugal force for their pumping action, they are essentially high-speed, high-volume devices (especially if high pressures are needed) and do not have positive displacement. The pressure or head developed by a given centrifugal pump at a particular speed is a function of the discharge rate, as indicated by the typical performance curves in Fig. 21.8. Note that the peak efficiency, which occurs at a relatively high flow rate, is well above 70 per cent for this particular unit, whereas efficiencies at small flows are low.

* Vane pumps are similar to roller pumps but have radial, sliding vanes in place of the rollers.

Fig. 21.7. Left: External-gear pump. Right: Roller-type rotary pump. Reversing the rotation of either type interchanges the inlet and outlet.
For a given pump and a given point on the efficiency curve, the discharge rate varies directly with the speed, the head varies as the square of the speed, and the power varies as the cube of the speed. If two or more stages are connected in series, the head and horsepower at a given discharge rate are increased in proportion to the number of stages. Thus, multistaging provides increased pressures without increasing the capacity range.

Centrifugal pumps are becoming increasingly popular for certain types and sizes of sprayers because of their simplicity and their ability to handle abrasive materials satisfactorily. They are well suited to equipment such as blower-sprayers and aircraft sprayers, for which high flow rates are needed and the required pressures are relatively low, and are used to some extent on low-pressure weed sprayers. The high capacities are advantageous for hydraulic agitation and for tank-filling arrangements. Speeds in these applications are generally in the range between 1000 and 4000 rpm, depending upon the pressure required and the diameter of the impeller.

Within the last few years, centrifugal pumps have appeared on a number of high-pressure orchard sprayers. With either single-stage or multistage pumps and speeds as great as 10,000 rpm, pressures of 500 to 700 psi are obtained at discharge rates of 100 gpm or more.
Since centrifugal pumps do not have positive displacement, they are not self-priming and do not require pressure relief valves for mechanical protection. Priming is usually accomplished by (a) mounting the pump below the minimum liquid level of the tank, (b) using an aspirator operated by the engine exhaust, or (c) providing a built-in reservoir on the pump that always retains enough liquid for automatic priming.

21.14. Miscellaneous Pumping Systems. Regenerative turbine pumps, which represent a special type of centrifugal pump, are occasionally used on sprayers. At a given speed, they can develop a much higher pressure than a single-stage centrifugal pump with the same diameter of impeller. Their efficiency is relatively low and they are affected to some extent by abrasive materials.

Flexible-impeller pumps have been used to some extent but are generally not suitable for pressures greater than 20 to 50 psi. In this type of pump, the vanes of a flexible impeller (usually neoprene) are flattened down by the eccentric housing during part of the rotation cycle, thus reducing the effective volume between the vanes at that point and creating the pumping action.

Diaphragm pumps are used to a limited extent for flow rates up to 5 or 6 gpm where required pressures do not exceed about 80 psi. Since the valves are the only moving parts in contact with the spray material, these pumps can readily handle abrasive materials.

Some small weed sprayers have an air compressor to pressurize the tank. Thus, the spray material does not pass through the pump, but agitation of the material in the tank presents mechanical problems. The practical size of this type of system is limited by the requirement that the tank be pressure-tight and able to withstand operating pressures up to 100 psi and by the cost of the larger compressors.

**AGITATION OF SPRAY MATERIALS**

Many of the spray materials are suspensions of insoluble powders or are emulsions. Consequently, most sprayers are equipped with agitating systems. Both mechanical and hydraulic systems are used. Either type, if properly designed, will provide satisfactory mixing.
21.15. Mechanical Agitation. Mechanical agitation is commonly obtained by means of either flat blades or propellers on a shaft running lengthwise of the tank near the bottom and rotating at a speed of 100 to 200 rpm. The following relations apply to round-bottom tanks with flat, I-shaped paddles sweeping close to the bottom of the tank; they are based on results originally reported by French and later analyzed by Yates and Akesson.31

\[
S_m = 69.2 \frac{A^{0.422}}{R^{0.531}} F^{0.293} \tag{21.1}
\]

\[
H_s = 1.93 \times 10^{-11} R^{0.582} S^{0.41} L \tag{21.2}
\]

where \(S_m\) = minimum peripheral speed of paddles, in feet per minute.

\(A\) = depth of liquid above agitator shaft centerline, in inches.

\(R\) = total combined width of all paddles, divided by tank length.

\(L\) = length of tank, in inches.

\(H_s\) = shaft input horsepower at any peripheral speed \(S\).

\(F\) = factor indicating relative power requirement for agitating a given mixture (hydraulically or mechanically).

The factor \(F\) is taken as unity for the most difficult oil-water mixture, which was found to be 40 per cent oil and 60 per cent water, by volume.31 For other oil-water mixtures, \(F = 0.5\) for 1 to 2 per cent oil, 0.8 for 10 per cent oil, and 0.7 for 60 per cent oil.31 No data are available to indicate values of \(F\) for suspensions containing heavy solids.

Paddle tip speeds in excess of about 500 fpm may cause serious foaming of some mixtures.31 For mechanical agitation in flat-bottom tanks with rounded corners, the minimum tip speed from equation 21.1 must be multiplied by the factor, 1.22. This increase in minimum speed causes the minimum power requirement to be approximately doubled (equation 21.2).

21.16. Hydraulic Agitation. To obtain agitation hydraulically, a portion of the pump output is returned to the spray tank and discharged into it through a series of solid-stream nozzles or
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orifices located in a boom along the bottom of the tank. The energy and turbulence from the jets provide the mixing action. In tests with various sizes of cylindrical tanks, Yates and Akeson 31 found that best results were obtained when solid-stream nozzles were mounted on a boom 1 to 2 in. above the tank bottom and aimed upward at an angle of about 30° above the horizontal. Changing the nozzle spacing in the range from 3 to 28 in. did not affect the performance, provided the orifice size was changed accordingly to maintain the same total flow rate per foot of boom length.

The minimum total recirculation rate for hydraulic agitation in a cylindrical or round-bottom tank, based on complete mixing of a full tank of material within about 1 min after agitation is started, 31 was found to be

\[ Q_m = 1.25 \frac{V F}{P^{0.56}} \]  \hspace{1cm} (21.3)

where \( Q_m \) = minimum total recirculation rate, in gallons per minute.

\( V \) = tank volume, in gallons.

\( P \) = agitator nozzle pressure, in pounds per square inch.

This will ordinarily be about the same as the spray-nozzle pressure.

From basic hydraulic relations, the hydraulic horsepower (useful output) required for any recirculation rate and pressure is

\[ H_h = \frac{Q P}{1714} \]  \hspace{1cm} (21.4)

where \( Q \) = total recirculation rate (not necessarily the minimum required for agitation), in gallons per minute.

Note that when hydraulic horsepower is expressed in terms of pressure and volumetric rate, it is independent of the fluid density. It can be shown from equations 21.3 and 21.4 that as the nozzle pressure is increased the minimum flow rate required for agitation decreases but the corresponding power requirement increases. Thus, the most efficient agitation is obtained at low pressures.

The principal advantage of the hydraulic system is its simplicity as compared with the mechanism and drive required for
mechanical agitation. With hydraulic agitation, however, the spray pump must have additional capacity and the power requirements will be considerably greater than for mechanical agitation, particularly at high pressures. For high-pressure orchard sprayers, mechanical agitation is definitely the more economical system (see Problem 21.2).

**FIELD SPRAYERS**

The general-purpose field sprayer must be able to handle many types of materials, including solutions of chemicals in water or oil, suspensions of insoluble materials (wettable powders), various oils, and oil-water emulsions. Many of them are corrosive, and others are abrasive and result in rapid wear of pumps and nozzles. Application rates range from as low as 5 to 10 gal per acre for 2,4-D (at 20 to 50 psi), up to 200 or more gallons per acre for some insecticides and general-contact herbicides. Required pressures ordinarily do not exceed 150 to 200 psi. A sprayer must be rather versatile if it is to meet all these conditions. As a result, many sprayers are designed for one specific type of job, such as the low-pressure, low-volume application of 2,4-D. Tractor mounting is most popular for the low-volume sprayers, whereas larger units are generally of the two-wheel, trailed type. Sprayers are sometimes mounted on skids so they can be placed on trailers or trucks when they are to be operated. High-clearance, self-propelled units are available for tall row crops.

**21.17. Basic Components of a Field Sprayer.** Figure 21.9 shows a typical, schematic arrangement for a field sprayer. Tanks range in size from 50 to 500 gal or more, depending on the intended type of application, the pump capacity, and the boom length. The strainers indicated in the tank-fill opening and at the pump inlet should be of 50 to 80 mesh screen, whereas the boom line strainer is perhaps 100 to 150 mesh, depending upon the size of nozzles on the boom. A quick-acting shut-off valve, readily controlled from the operator's position, is included in the line to the boom.

Rotary pumps are used extensively, particularly on units that are designed specifically for low-volume, low-pressure spraying of soluble materials such as 2,4-D. Centrifugal pumps are often employed on sprayers designed for higher application rates or
on machines with long booms. On any sprayer the pump capacity should be at least 10 to 15 per cent greater than the maximum expected spraying requirements. The piping may be arranged to permit filling the tank with the spray pump, as indicated by the broken-line portion of Fig. 21.9.

With a positive-displacement pump, an automatic relief valve or a bypass pressure regulator (Fig. 21.9) is essential for protection of the equipment. These devices generally have a spring-loaded ball or diaphragm and are adjustable.

Fig. 21.9. Diagram showing the basic components of a field sprayer having mechanical agitation.

With a centrifugal pump, a bypass regulator is generally not satisfactory for controlling pressure, because of the head-flow characteristics of the pump. In the usual operating range for sprayers, the pressure or head changes so slowly with discharge rate (Fig. 21.8) that the required capacity for the bypass regulator and lines would be unreasonably large and sensitivity of control would be poor. The pressure from a centrifugal pump can be controlled reasonably well by changing the speed of the pump or by a simple, manually adjusted throttle valve in the line to the boom. More precise control over a range of flow rates can be obtained with a diaphragm-type pressure-reducing valve in the boom feed line.

21.18. Booms and Nozzles. Practically all field sprayers have horizontal booms with nozzles attached directly to the boom or to the lower ends of vertical drop pipes extending downward from the boom. The drop pipes are for between-the-row spray-
ing of tall row crops. To obtain more accurate control of nozzle height, nozzles are occasionally connected to the boom through drop hoses and mounted on individual skids that move along the ground between the rows.\textsuperscript{26} Boom lengths are usually from 10 to 14 ft but may be as much as 80 ft.\textsuperscript{5} Booms are ordinarily sectionalized and hinged to permit reducing the over-all width for transport. All but the shortest booms require both vertical and fore-and-aft support. The longer booms are sometimes equipped with outrigger wheels to maintain more uniform nozzle heights. Outlets for nozzles should be attached to the side or top of the boom (rather than the bottom) to provide a settling place (in the boom) for dirt particles and to prevent boom drainage when the shut-off is closed.

Fan-type nozzles seem to be preferred for weed spraying, whereas hollow-cone nozzles are more popular for insecticides and fungicides. There is, however, little concrete evidence to justify the choice of one type over the other. Brass is the most common material for nozzles. Hardened stainless steel tips or orifice disks are considerably more expensive than brass ones but are preferred for some applications because the rate of wear from abrasive materials is much lower.

Nozzles of the sizes generally found on field sprayers (orifice diameters as small as 0.02 in.) have built-in screens (Fig. 21.3) to prevent or minimize plugging of the small passages and orifices. Screen openings should be only a little smaller than the orifice opening so that fine particles, such as those of a wettable powder in suspension, can pass through both the screen and the orifice. The nozzle screens should supplement those indicated in Fig. 21.9, rather than replacing them.

21.19. Nozzle Spacings and Boom Height. For open-field work (i.e., continuous coverage) the proper height of the boom above the deposition surface is a function of (a) the nozzle spacing on the boom, (b) the nozzle spray angle, and (c) the amount of overlap required for uniform coverage, as determined by the nozzle spray pattern. In testing a considerable number of nozzles, Akesson\textsuperscript{1} found three general types of distribution profiles for individual nozzles, as follows: (a) profiles having a broad, flat section across the center (i.e., uniform flow), with steep sides (rapidly decreasing flow), as idealized by the broken-line curves in Fig. 21.10d, (b) profiles with more gradually sloping sides and
a much narrower flat top, as idealized by the broken-line curves in Fig. 21.10a, and (c) profiles with sloping sides and a very narrow or peaked center section. Profiles for three of the four fan-type nozzles reported by Barger and his associates \(^5\) can be approximated very closely by the pattern of Fig. 21.10a.

Fig. 21.10. Effect of the nozzle distribution pattern and the boom height upon uniformity of coverage. The broken-line curves indicate distribution patterns (at the deposition level) for individual nozzles. The solid curve in each case shows the combined discharge pattern for all nozzles (i.e., the sum of the broken-line curves).

Figure 21.10 shows the amount of overlap required for uniform coverage with two types of idealized nozzle distribution patterns, and the effect of variations of boom height upon the uniformity. Note that the steep-sided profiles in (d) require less overlap but are far more sensitive to variations in boom height than are the narrower-topped and gradually sloping profiles of (a). With either pattern the uniformity of distribution is affected less by having the boom too high (excessive overlap) than by having it too low. With nozzles having profiles of the type shown in Fig. 21.10a, the height of the boom should be such that the over-all width of each nozzle spray pattern at the deposition level is about 50 per cent greater than the nozzle spacing. More overlap
is required for profiles having narrower peaks, and less for steep-sided profiles.

If the boom is placed at twice the minimum height for uniform coverage, so-called double coverage is obtained and the sensitivity to changes in height is reduced. If the application is such that the boom must be kept low to minimize drift, double coverage requires that the nozzles be spaced closer together than for "single" coverage (or else have wide spray angles). Closer spacing means smaller nozzles for a given application rate, with resulting finer atomization (more drift) and increased possibility of clogging. Nozzle spacings for open-field work generally range from 12 to 24 in., the closer spacings being for double coverage and the wider spacings for single coverage.

The spacing of nozzles or drop pipes for row crops is usually either one or two outlets per row. If the crop plants are being sprayed, one nozzle is located directly above each row. Additional nozzles on drop pipes between the rows may or may not be provided, depending upon the crop. When spraying weeds in tall row crops, only the nozzles on drop pipes are used. Laboratory tests by Barger and his associates demonstrated that, for spraying weeds in corn, uniform distribution could be obtained either with two inclined nozzles (fan-type) on a single drop pipe centered between the 40-in. rows or by having two drop pipes per row (20 in. apart) with single, vertical nozzles.

21.20. Control of Application Rate. The amount of spray liquid applied per acre by a field sprayer is a function of (a) the spacing of nozzles on the boom, (b) the nozzle rating or orifice size, (c) the nozzle pressure, and (d) the rate of forward travel. The relation between these variables is expressed by the following equation, in which the combined effects of nozzle size and pressure are represented by the nozzle discharge rate:

\[
\text{(gph per nozzle) } = \frac{0.0101 \text{ (gal per acre)} \times \text{(mph)} \times \text{(nozzle spacing, in.)}}{}
\]  

(21.5)

If the total amount of liquid per acre is not critical, the concentration of active ingredient may be changed, which introduces a fifth variable. Since all these factors are equally important, they must all be controlled carefully in order to obtain the desired application rate. Nozzles must have uniform flow rates and pat-
terns; the pressure gage must be accurate, with proper allowance being made for any pressure losses in the lines; and the forward speed must be checked accurately by timing or with a low-range speedometer mounted on the tractor or sprayer.

HIGH-PRESSURE ORCHARD SPRAYERS

High-pressure sprayers (usually 400 to 800 psi) have been used in orchards for many years. They may be of the two-wheel or four-wheel trailed types or may be mounted on a truck. Tank sizes are usually in the range from 250 to 600 gal, with mechanical agitation provided. Application rates of 600 to 800 gal per acre are common for many types of mature fruit trees, although rates several times this high are sometimes required in spraying citrus and olive trees.

In the usual arrangement, the liquid is pumped either through hose lines to hand guns manipulated by the spraymen or to various types of booms or masts for automatic spraying. When spraying by hand, towers or raised platforms are often provided to permit better accessibility and coverage of tree tops.

21.21. Pumps and Pressure Regulators. High-pressure orchard sprayers are generally equipped with piston or plunger-type pumps (Section 21.11). However, as mentioned in Section 21.13, several of the more recent models are equipped with centrifugal pumps operated at extremely high speeds. The high capacity of these centrifugal pumps (100 gpm or more) makes them well suited for automatic spraying with mechanical booms.

The bypass regulator required with a piston pump is generally of the automatic-unloading type, in that it allows the pump to operate at a greatly reduced pressure when no material is being sprayed. A spring-loaded diaphragm or plunger (Fig. 21.11) moves to open a bypass valve (ball or disk) whenever the liquid pressure is sufficient to overcome the resisting force of the compression spring. To obtain satisfactory regulation, some liquid must bypass through the regulator at all times. If all guns or booms are shut off completely, the sudden surge of pressure opens the bypass valve rather widely. Simultaneously, the unloader check valve closes to maintain high pressure on the regulator (and in the spray lines) so that the bypass valve is held open.
When subsequent opening of a gun or boom valve reduces this pressure below the normal spraying pressure, the spring moves the plunger downward, thus permitting the bypass valve to close.

The compression spring may be adjusted to obtain any pressure within the normal range of high-pressure spraying. If a high-pressure sprayer is to be used for low-pressure work as well, it is generally desirable to install an additional low-pressure bypass regulator in parallel with the high-pressure regulator, since the high-pressure device is often not very sensitive at pressures below 100 psi.

21.22 Spray Guns and Booms. The high-pressure orchard sprayer depends upon liquid pressure to atomize the liquid and furnish the energy for carrying the spray to and into the trees. Conditions that favor long carrying distances are: (a) high discharge velocity from the nozzle, (b) high volume rate of discharge, (c) large droplets, and (d) small spray angle. As the pressure is increased, the carrying distance increases up to the point where the adverse effect of reducing the droplet size equals or overbalances the benefits of the higher discharge velocity and greater volume of material. French found that the average size of droplets produced by a hand gun ranged from about 400 microns at 200 psi to 200 microns at 800 psi. In these tests the maximum carrying distance with the spray gun (Fig. 21.4) adjusted for close-range spraying (wide-angle spray) increased rapidly between 200 and 600 psi but increased very little between 600 and 1000 psi. When the gun was set for long-range spraying (solid spray stream), the carrying distance increased with pressure up to about 800 psi and then decreased between 800 and 1000 psi.

Orchard spray nozzles or guns are generally identified by a
number that represents the orifice diameter in 64ths of an inch. Typical discharge rates for single-nozzle guns are from 5 to 15 gpm at 600 psi. Multiple-nozzle guns or "brooms" employ a group of three to eight non-adjustable nozzles with smaller orifices (1/64 to 1/8 in., for example) to give a similar range of total flow rates. The smaller droplets obtained with these smaller nozzles tend to give better coverage with less material, and the mutual reinforcement of the nozzles creates an air movement sufficient to carry even the smaller droplets a considerable distance.\(^{13}\)

Sprayers having pumps with adequate capacity (at least 40 to 50 gpm) may have several guns or groups of nozzles mounted on masts or on a boom shaped in a vertical arc around the tank. Some booms consist of groups of nozzles oscillated mechanically to sweep the trees as the sprayer moves down the row. The principal advantage of these devices, which move along the row without stopping, is in the saving of labor. Coverage is also more rapid and often tends to be more uniform than that obtained with hand-spray operations, but more material per tree may be needed.\(^{28}\)

**BLOWER-SPRAYERS**

21.23. Applications. The main use for blower-sprayers is in applying spray materials to trees. As the name implies, blower-sprayers utilize an air stream to carry the spray droplets, rather than depending upon the energy from hydraulic pressure. Consequently, they can handle smaller liquid droplets (75 to 100 microns, average size) than the high-pressure orchard sprayers. The effectiveness of a blower-sprayer depends upon its ability to displace the air in the tree with spray-laden air from the machine.

The maximum advantage of blower-sprayers over high-pressure orchard sprayers is realized in applications where satisfactory coverage can be obtained with low-volume rates of semiconcentrate or concentrate materials. In some applications, however, the volume of liquid needed with a blower-sprayer is as great as that required with high-pressure sprayers.

In semiconcentrate spraying, the foliage is wetted only to the point of little or no run-off,\(^{19}\) using perhaps one-third to one-fourth as much total liquid per acre as for bulk or high-volume spraying. Concentrate sprays are put on with perhaps 10 to 15 per cent as much total liquid as for bulk spraying (under some
conditions this may not provide enough liquid for thorough tree coverage). The principal advantage of low-volume spraying is the reduced amount of water that must be supplied and the resultant saving in filling and hauling times. Because of the absence of run-off, there also tends to be some reduction in the amount of active ingredient required per acre as compared with bulk spraying. Most of the large blower-sprayers for orchards are suitable for both semiconcentrate and bulk spraying, although some of them do not have enough capacity for bulk spraying. Other machines are designed primarily for concentrate spraying.

In addition to the reduction in the amount of liquid that must be hauled and the saving in active ingredients, blower-sprayers cover the orchard two or three times as fast as hand-operated high-pressure sprayers. Labor requirements are much less than for hand-operated sprayers and somewhat less than for high-pressure sprayers with automatic booms. The chief disadvantage of the blower-sprayer is its high initial cost, which makes its ownership economically impractical for small orchards. Complete coverage is difficult to obtain in trees with dense foliage, such as citrus and olive trees.

21.24. Blowers and Outlets. Most blower-sprayers can be fitted into the following arbitrary classification:

1. Low-volume, high-velocity (under 5000 cfm, and over 150 mph discharge velocity).
2. Medium-volume, medium-velocity (5000 to 25,000 cfm, and 100 to 150 mph).
3. High-volume, low-velocity (over 25,000 cfm, and under 125 mph).

The low-volume, high-velocity machines are principally shade-tree machines, although some small orchard models are available in this group. They are generally equipped with centrifugal blowers and have a 4 to 6 in. diameter flexible discharge hose that is directed by hand to cover the trees.

Most of the large, orchard-type blower-sprayers are in the third class and are equipped with either centrifugal or axial-flow fans. The discharge is usually through a slot (circumferential opening around the blower housing, "fishtail" outlet, etc.), although some machines have round or nearly-square outlets that are oscillated mechanically to sweep the trees. In any event, the arrangement
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is such that one side of one row, or the adjacent sides of two rows, can be covered as the machine is driven along between the rows. The included angle of delivery (in a vertical plane) on each side should be adjustable to take care of different sizes of trees. Since power requirements are high, especially for the high-volume units (50 hp or more), careful design of the air system and proper selection of the fan or blower best suited to the operating conditions are important.

21.25. Pumps and Atomizing Devices. Most orchard-type blower-sprayers introduce the spray material into the air stream through hydraulic nozzles (usually hollow-cone), the degree of atomization being principally a function of the liquid pressure and the nozzle characteristics. Bulk and semiconcentrate machines generally operate with nozzle pressures below 100 psi and are equipped with centrifugal pumps capable of delivering 50 to 200 gpm. These pumps can be used for rapid filling of the tank, and part of their output is often recirculated for hydraulic agitation. Concentrate machines may operate with pressures as high as 500 psi, to obtain greater atomization, and are ordinarily equipped with piston-type pumps (5 to 20 gpm).

The large-volume orchard machines designed for bulk or semiconcentrate spraying have as many as 100 small nozzles in the air stream. These nozzles may be changed (different sizes), or part of them removed, for various spraying conditions. In order to obtain uniform coverage, it is generally necessary to install more or larger nozzles in that portion of the air stream which is directed to the tree tops. Concentrate machines have only a small number of nozzles, and blower-sprayers of the high-velocity, low-volume type usually have a single nozzle in the outlet duct, directed against the air stream or at right angles to it.

Shear-plate atomizing devices, which are used to a limited extent, introduce the spray material at extremely low pressures (1 to 10 psi) and rely entirely upon the air stream for atomization. Since air velocities of 150 to 250 mph are needed for fine atomization, these devices are suitable only for the low-volume, high-velocity, shade-tree sprayers, or for low-velocity sprayers that have a separate high-velocity tube for atomization.9
GROUND-RIG DUSTERS

All dusters utilize an air blast in which the dust particles are airborne. Various types of high-speed blowers are employed on ground rigs, whereas the blast for aircraft dusting is supplied by the propeller and the forward motion. In general, a duster is a simpler, less troublesome, and less costly machine than a sprayer. Since dust applications are usually only 20 to 50 lb per acre, there is much less weight to haul than when spraying. Dusting is advantageous where the water supply is inadequate or inconvenient for spraying.

The most serious problems in connection with dusting are drift (Section 21.3) and inefficient use of material. The weather must be calm for dusting, whereas spraying can safely be done with winds up to 8 to 12 mph. Tests have indicated that in many cases only 10 to 20 per cent of the material discharged by the duster is actually deposited on plant surfaces. Some of the methods that have been tried in an attempt to reduce drift and increase deposition are (a) canvas tunnels dragged along over the plants and into which the dust is discharged, (b) addition of an oil or water spray at the exit of the dust nozzles, and (c) electrostatic charging of the dust particles in such a manner that they will be attracted to the plant surfaces.

21.26. Hoppers and Feed Mechanisms. The conventional feed system consists of an adjustable opening in the bottom of the hopper, which meters dust into the blower inlet, and one or more agitators above the discharge opening. With this arrangement the feed rate may be extremely uneven and unpredictable because of the influence of such factors as (a) the density of the dust, (b) the degree of compaction when put into the hopper, (c) the change in head of dust as the hopper empties, and (d) changes in the fluidity of the air-dust mixture in the hopper (due to compaction or to the addition of air as a result of excessive agitation). In tests with 47 dust materials, Irons found that densities ranged from 9 to 96 lb per cu ft. He reports feed-rate variations that exceeded a 3 to 1 ratio for a given dust and duster adjustment. Fractionation, or separation of the active ingredients from the diluent, is also a problem.

A recent improvement in dust feed mechanisms is the vertical-
auger feed developed by the USDA. In this device, dust is moved continuously at a rapid rate from the bottom of the hopper to the top by means of a vertical auger in a central tube open at both ends. The auger tube has an adjustable port near the top, through which a portion of the circulated dust is metered into the fan inlet. This arrangement reduces or eliminates the problems of compaction, fractionation, and change of head, and gives remarkably uniform feed rates.

21.27. Blowers and Discharge Arrangements. Various types of blowers, including accurately built, multiblade centrifugal fans, paddle-type centrifugal fans with four to six welded steel blades, and propeller or axial-flow fans, are employed on dusters. The present trend is toward high-volume blowers with discharge velocities somewhat lower than have been customary (for example, 20,000 cfm with a discharge velocity of about 80 mph). Blower-sprayers are sometimes adapted for dusting by mounting a hopper and feed device at the rear of the machine (and not using the tank).

Discharge equipment may be of the single-outlet, multiple-nozzle, or hollow-boom types. Single outlets with one or two vertical "fishtails" attached, or with peripheral openings as on blower-sprayers, are sometimes used for orchard or vineyard dusting. Multiple-nozzle units for field or row crops have a group of flexible hoses connected to a manifold or to peripheral outlets around the fan housing. The nozzles are then spaced along a supporting boom with the outlets located as close to the plants as is practical. Differences in delivery rates from individual tubes of different lengths is sometimes a problem.

Hollow booms 3 to 6 in. in diameter are sometimes used in field crops to convey the material and distribute it through appropriately-spaced openings. Some booms are tapered, whereas others (metal or canvas) have a uniform diameter and employ internal baffles in an attempt to improve uniformity of discharge. Long booms are usually jointed in the middle and supported by outrigger wheels, with the blower discharging into each half separately.
21.28. Applications and Types of Aircraft. In 1952, airplanes in the United States were flown more than 300,000 hr for dusting agricultural crops and approximately 230,000 hr for spraying, which represents a total coverage of perhaps 30 million acres. Less than 2 per cent of the total number of aircraft involved were helicopters. Currently, the Stearman-type biplanes (used by the army and the navy as trainers during World War II) are probably the most widely used fixed-wing aircraft, although some of the light monoplanes are popular in the Midwest. The Texas A. and M. College has built an experimental low-wing monoplane designed especially to meet the requirements for agricultural applications such as spraying and dusting. The trend in the design of new airplanes for agricultural applications is toward monoplanes.

The principal advantages of aircraft over ground equipment are in the speed of coverage and the resultant improvement in timeliness, and in the ability to apply the materials at times when ground equipment could not get onto the field. Aircraft spraying is not as thorough as ground-rig spraying, especially in crops more than 3 to 5 ft tall, but it gives adequate control for many types of applications. Dust applications by aircraft will ordinarily give good penetration, but adequate deposition is sometimes difficult to obtain. Drift of hazardous materials is one of the most serious problems in aircraft application, particularly in connection with dusting. Aircraft operations are hazardous, are more dependent on optimum weather conditions than are ground operations, and are less efficient on small acreages.

In comparison with fixed-wing aircraft, helicopters have greater maneuverability in closely bounded fields. Because less landing area is required, ferrying time is usually much less. Helicopters also seem to give better dust penetration in dense foliage on tall crops when flown at low forward speeds (20 to 25 mph, or less). One of the principal disadvantages of the helicopter is the high initial cost, which is five to ten times that of a suitable fixed-wing plane. Low forward speeds are also a handicap from the cost standpoint when dusting large areas. Helicopter pay loads are small, generally ranging from 300 to 600 lb as com-
pared with 800 to 1200 lb for fixed-wing planes such as the Stearman.

21.29. Aircraft Spraying. Because of the limitations in payload for aircraft, spray materials are always applied in concentrate form, usually at rates of 5 to 10 gal per acre. Agricultural sprays are normally applied from a height of 5 to 10 ft above the tops of the plants. Flagmen on the ground guide the pilot to obtain a controlled swath width that is perhaps 5 ft greater than the wing span. Fixed-wing planes are generally flown at speeds of 80 to 100 mph during application.

Wing-length (or rotor-length) booms equipped with hydraulic nozzles are used extensively on both fixed-wing planes and helicopters. Other types or arrangements of atomizing devices found on aircraft sprayers include (a) short booms (6 to 8 ft long) beneath the fuselage, (b) wing-tip nozzles or booms, (c) rotating brushes or disks driven at 2000 to 3000 rpm by auxiliary propellers (centrifugal atomization), and (d) breaker-bar booms in which liquid streams discharged from many small openings along the boom strike a breaker bar close behind. Short booms, wing-tip nozzles, and whirling devices require greater flying heights than does the wing-length boom, in order to obtain uniform distribution across the swath.

Wing-length booms (which, on fixed-wing planes, are actually a little shorter than the wing because of wing-tip vortices) may be fitted either with close-spaced nozzles or with clusters of nozzles. The nozzles or clusters are equipped with individual, quick-acting shut-off valves controlled simultaneously from the pilot's cockpit or else have individual spring-loaded check valves that close (at about 5 psi) when the boom flow is cut off and the boom pressure is released. The latter system is becoming increasingly popular and is frequently augmented by a venturi "suck-back" arrangement (Fig. 21.12), which, when the pump flow is diverted into the tank, creates a slight suction on the boom, thus reducing the boom pressure more rapidly and preventing leakage from faulty check valves. The suck-back jet must be relatively large so that the pressure drop through it will not be excessive when spraying. The check-valve and suck-back system is also sometimes employed on ground-rig field sprayers.

To minimize drift and reduce evaporation losses from the falling droplets, aircraft sprayers generally apply a coarser spray
than do ground rigs. Average droplet sizes between 100 and 200 microns are typical for aircraft spraying, the smaller drops giving more complete coverage but more drift. In applying 2,4-D, it appears desirable to use the largest obtainable droplet sizes that will still permit uniform distribution across the swath. Various types of nozzles are used in aircraft spraying, with the side-entry hollow-cone type (Fig. 21.3a) being rather popular because of its relative freedom from clogging. Nozzle orifice sizes range from $\frac{1}{4}$ to $\frac{1}{4}$ in., depending upon the desired droplet size, flow rate, and spacing of nozzles. Pressures range from 20 to 100 psi on fixed-wing aircraft and up to 200 psi on helicopters.

Centrifugal pumps are most common, primarily because of their high capacity and ability to handle abrasive materials. They may be driven by an auxiliary propeller (on fixed-wing planes only), from a power take-off, or by an electric or hydraulic motor. Hydraulic agitation is generally employed, the entire pump output being directed back into the tank when not spraying, but with little or no recirculation during the short intervals of actual spraying.

21.30. Aircraft Dusting. Dust hoppers in fixed-wing aircraft are equipped with propeller-driven, low-speed agitators. Some type of slide-gate feed-control device is generally employed, with felt gaskets provided to give dust-tight closure and with a lever mechanism in the cockpit for operating the gate. Variation of feed rate is a problem, just as it is with conventional
ground-rig feed devices. The metered dust enters the throat of a venturi-spreader arrangement located in the propeller blast beneath the fuselage (Fig. 21.13), from which it is discharged into the wing downdraft. The characteristic wing-tip vortices draw the dust out laterally, but there is a tendency toward heavy deposits in the center of the swath.

Fig. 21.13. Venturi-spreader on an aircraft duster. The small propeller beneath the wing drives the dust agitator.

In order to obtain more uniform distribution across the swath, dusting with fixed-wing aircraft is ordinarily done at a somewhat greater height than when spraying and there is considerable overlap. To counteract the reduced efficiency of aircraft application, dusts are often applied at rates a little greater than would be used with ground rigs.

Helicopters have two dust hoppers, one on each side of the plane, with agitators that are usually driven by electric motors. The feed devices are of either the slide-gate or the fluted-roll type. The engine cooling fan provides air for venturi tubes under the two hopper discharge outlets. The dust is fed into these tubes and then discharged down and outward into the rotor downdraft.
REFERENCES


SPRAYING AND DUSTING


PROBLEMS

21.1. A 250-gal cylindrical sprayer tank is 58 in. long and has a diameter of 36 in. Mechanical agitation is to be provided, with four paddles 11 in. long (tip diameter) and 8 in. wide mounted on a shaft 6 in. above the bottom of the tank.

(a) Calculate the minimum rpm for agitating a mixture of 10 per cent oil and 90 per cent water.

(b) If the mechanical efficiency of the power transmission system is 90 per cent, what brake horsepower (bhp) would be needed for agitation?
21.2. (a) Under the conditions of Problem 21.1, what recirculation rates would be required for hydraulic agitation at 60 psi and at 400 psi?
(b) If the pump efficiency is 50 per cent, what brake horsepower would be needed for hydraulic agitation at each pressure?
(c) Prepare a table to compare the results of Problems 21.1 and 21.2.
21.3. A field sprayer with a 30-ft boom (i.e., covers a strip 30 ft wide) has nozzles 18 in. apart and is to be designed for a maximum application rate of 150 gal per acre at 125 psi and 4 mph.
(a) Determine the required pump capacity in gallons per minute, assuming 10 per cent of the flow is bypassed under the above maximum conditions.
(b) If mechanical agitation requires 0.5 bhp and the pump efficiency is 50 per cent, what should be the engine rating if the engine is to be loaded to not more than 80 per cent of its rated horsepower?
(c) What discharge rate per nozzle (gpm) is required under the above conditions?
(d) If the nozzles have 70° spray angles and the pattern is such that 50 per cent overlap is needed for uniform coverage (i.e., spray pattern 50 per cent wider than nozzle spacing), at what height above the tops of the plants should the boom be operated?
21.4. A field sprayer is equipped with nozzles having a rated delivery of 0.11 gpm of water at 40 psi. The nozzle spacing on the boom is 20 in. Each pound of active ingredient (2,4-D) is mixed with 10 gal of water and the desired application rate is 14 oz of chemical per acre. What is the correct forward speed for a nozzle pressure of 30 psi?
21.5. A blower-sprayer is to be operated at 2½ mph and the desired application rate is 15 gal per tree. The tree spacing is 30 × 30 ft and each nozzle delivers 1.05 gpm at the operating pressure of 60 psi.
(a) If one-half row is sprayed from each side of the machine, how many nozzles will be needed?
(b) How many acres can be covered with a 500-gal tank full of spray?
CHAPTER 22

Farm Transport

22.1. Introduction. Transportation, in one form or another, is an indispensable part of most farm operations. It may involve hauling seed or fertilizer to the field, removing harvested crops from the field, distributing feed to livestock, moving farm equipment from one place to another by means of trailers or special implement carriers, or various other applications. Even the moving of an implement on its own wheels and the driving of a tractor to and from the field might well be considered transport operations.

Although motor trucks and other types of automotive equipment are being used to a great extent in many farm operations, the discussion in this chapter will be limited to equipment or components with which the farm machinery designer is directly concerned.

TRANSPORT WHEELS

22.2. Rolling Resistance. As an implement or vehicle moves over a given surface, the draft or resistance to forward travel that results from the wheels rolling along the surface is known as the rolling resistance. If the surface is soft, each wheel sinks in, and in effect is continually trying to climb an incline as it moves along. The rolling resistance includes this effect, as well as other losses due to soil disturbances. The coefficient of rolling resistance is defined as the ratio of the rolling-resistance force to the vertical load on the wheel.

One of the important advantages of pneumatic tires as compared with steel wheels is the reduced rolling resistance. In a series of tests with individual implement wheels of various sizes and with a range of loads and field surface conditions, the coefficients of rolling resistance for pneumatic tires averaged 28
per cent less than those for steel wheels of comparable sizes. On concrete, however, the steel wheels were somewhat superior in regard to rolling resistance.

In these tests the most important factors affecting the coefficient of rolling resistance were found to be (a) the condition of the road or field surface, (b) the outside diameter of the wheel or tire, (c) the inflation pressure for tires, and (d) the rim width on steel wheels. For a given outside diameter and inflation pressure the cross-sectional diameter of the tire had comparatively little effect except under extreme conditions, and the usual differences in tread design of pneumatic implement tires did not appear to be of any consequence.

The relation of the coefficient of rolling resistance to the important variables is indicated for pneumatic tires in Fig. 22.1. The four broken-line curves in the left-hand graph represent the results of a considerable number of runs for each of the soil or road conditions indicated (experimental points omitted). The tests on the fall rye seeding were on a loam soil after fall plowing, disk ing, spike-tooth harrowing, and seeding. The tilled loam and the loose sand had been freshly plowed 10 in. deep and a spike-tooth harrow had been pulled behind the plow. Moisture contents and weights per cubic foot (dry basis) were as follows: (a) bluegrass pasture, 22 per cent, 67 lb per cu ft, (b) fall rye seeding, 19 per cent, 69 lb per cu ft, (c) tilled loam, 18 per cent, 62 lb per cu ft, and (d) loose sand, 8 per cent, 83 lb per cu ft.

The solid curves in the left-hand graph were cross-plotted from groups of runs made with two sizes of tires on soils of various hardnesses. These results indicated a definite correlation between soil penetration readings and the rolling resistance. The Rototiller penetrometer used in these tests is a relatively simple device developed by Stone and Williams.  

The right-hand graph in Fig. 22.1 indicates that inflation pressure is an important factor in regard to the rolling resistance on soft, loose surfaces. Reducing the inflation pressure increases the work of flexing the tire; but, because of the larger bearing surface, it decreases the energy expended in displacing the soil. Thus, decreasing the inflation pressure greatly reduces the rolling resistance on extremely soft surfaces but increases it somewhat
Fig. 22.1. Coefficients of rolling resistance for pneumatic tires in relation to outside diameter, road or field surface conditions, and inflation pressure. For most implement tires, the nominal outside diameter (calculated from the tire size rating) can be used in place of the actual outside diameter in determining rolling resistances from this graph. (Eugene G. McKibben and Dale O. Hull. *Agr. Eng.*, June, 1940.)
on a hard, smooth surface (such as concrete) where there is no surface displacement.

22.3. Effect of Wheel Arrangement. Tests with 6.00-16-in. implement tires under three soil conditions indicated that the coefficient of rolling resistance for a dual arrangement of wheels did not differ greatly from that of a single tire. A tandem arrangement gave about the same rolling resistance on bluegrass pasture, but in the tilled loam and the loose sand the coefficient of rolling resistance of the tandem combination was 25 to 35 per cent less than for a single wheel. When a single tire was run along the same path several times in the loam and in the sand, the coefficient of rolling resistance was only half as great in the second pass as in the opening run. These results emphasize the desirability of arranging wheels on field equipment so they will track, especially when they are likely to be operated in soft ground.

22.4. Effective Tire Radius and Slippage. The right-hand graph in Fig. 22.1 includes a curve to show the relation of wheel slippage to the coefficient of rolling resistance. The per cent slip is based upon the effective radius and the loaded radius, the effective radius being determined from the measured forward travel per revolution of the wheel and the loaded radius being the height of the wheel center when supporting the test load on a concrete surface. It was found that, in general, when a transport wheel is operated on a friable medium (i.e., nonelastic in tension), the effective radius is always greater than the loaded radius. Thus, the bottom of the wheel is not stationary with respect to the soil but is continually slipping forward.

The forward slippage of transport wheels is of particular concern when some mechanism on the implement (such as a planting unit) is driven by ground wheels. In such applications the forward slippage may be increased as a result of the torque load imposed by the driven mechanism, the amount of increase being influenced by the field surface condition, the type of tire tread, the vertical load on the wheel, etc. In many cases the rotational resistance due to the driven mechanism is small in relation to the vertical load on the wheel so that slippage from this cause can be neglected.

Analysis of dimensional data published by four of the leading tire manufacturers indicates that, for agricultural implement
tires at rated loads and normal inflation pressures, the loaded radius averages about 8 per cent less than the nominal outside radius calculated from the tire size rating. Figure 22.1 indicates that when implement tires with outside diameters in the common range from 25 to 35 in. are operated on moderately firm, tilled soil, the natural slippage is about 5 to 10 per cent (i.e., the effective radius is 5 to 10 per cent greater than the loaded radius). Under these conditions an effective diameter equal to the nominal outside diameter \((16 + 2 \times 6.00 = 28\) in.) for a 6.00-16-in. tire can ordinarily be assumed with sufficient accuracy for speed calculations on ground-driven mechanisms. For other conditions, the effects of natural slippage (Fig. 22.1) and slippage due to the torque load can be applied to the known or estimated value for the loaded radius (estimated value = 92 per cent of the nominal outside radius).

22.5. Standardization of Tire Load Ratings and Disk Wheels. Although the exact dimensions of pneumatic tires vary slightly between different manufacturers, allowable inflation pressures and load ratings have been standardized for various sizes of tractor and implement tires. A table of load ratings for agricultural implement tires with either ribbed or traction tread is included in Appendix F. In order to provide interchangeability between certain sizes of agricultural wheels for pneumatic tires, the SAE and ASAE have adopted an official recommendation specifying the pertinent dimensions. This recommendation covers 12-, 15-, 16-, 18-, and 20-in. rim sizes.

WAGONS AND TRAILERS

22.6. Four-Wheel Wagons. With pneumatic tires, antifriction bearings, automotive-type steering, and steel frames, present-day farm wagons represent a vast improvement over their horse-drawn predecessors. Modern wagons or trailers are adapted to relatively high-speed operation and are used extensively for both field work and highway transport. Various types of wagon boxes or beds are available to meet the range of hauling requirements represented by such products as (a) loose or baled hay or straw, (b) chopped forages, (c) corn, in the ear, shelled, or in bundles, (d) grain, in sacks or in bulk, and (e) cotton, loose or baled.

Although the so-called standard tread of about 56 to 62 in.
has the advantage of tracking behind automotive vehicles, there appears to be a trend toward wider treads because of increased lateral stability and a better fit for straddling two rows of corn.

22.7. Standardization of Wagons. One of the important uses for farm wagons is that of trailing behind field harvesters for direct loading of such crops as corn (in the ear or shelled) or chopped forages. In these applications it is important that the position of the wagon box with respect to the elevator and discharge spout of the harvester be such that proper distribution of material in the wagon can be obtained and that there is no physical interference between the elevator and the wagon, either in straight traveling or on turns.

This poses a problem in standardization involving the wagon-tongue length, the wagon-box height, and the location of the hitch point on the rear of the harvester. Based upon a study of existing equipment and the general nature of the problem, an ASAE recommendation for farm grain wagons has been officially adopted. The recommended dimensions are indicated in Fig. 22.2. If these recommendations are followed by the wagon manufacturer, the harvester manufacturer will be able to properly locate the wagon hitch point on his particular machine.

22.8. Telescoping Tongues. A desirable feature, especially for wagons that are trailed behind field harvesters, is a telescoping hitch on the tongue. Such a device allows one man to ex-
tend the tongue as required to make the connection, without having to accurately maneuver and position the towing machine. After hitching, the operator merely backs the machine until the tongue locks in the retracted position. In tests with wagons trailed behind a two-row corn picker, changing wagons equipped with the standard (nontelescoping) tongue took two men and required 12.8 per cent of the net operating time. With telescoping tongues on the wagons, one man alone made the changes in 4.5 per cent of the net operating time.

22.9. Two-Wheel Trailers. Whereas four-wheel wagons are generally preferred for field operations in the United States, two-wheel units are rather popular in England and some of the other European countries, particularly for general-purpose use on small farms. Two-wheel trailers are compact and more easily maneuvered than four-wheel units, and the load can be distributed so as to add considerable weight to the tractor wheels for increased traction needed under muddy, hilly, or other adverse conditions. There has been some European interest in pto drives for the trailer wheels to secure additional traction. The drive is usually designed for the lowest tractor gear and includes an overrunning clutch for protection when operating in other gears. The drive is ordinarily engaged only when needed for traction.

Two-wheel trailers are more difficult than four-wheel units to hitch and unhitch, especially when loaded, and therefore do not lend themselves as well to direct loading from field harvesters or to loading and unloading while detached. However, coupling and uncoupling of two-wheel trailers (even when loaded) can be greatly simplified by the use of automatic tractor hitches that are raised or lowered by means of the tractor's hydraulic system.

MECHANICAL UNLOADING

22.10. Tilting Beds. For free-flowing materials such as grain, shelled corn, and ear corn, the dumping method of unloading is entirely satisfactory. It may be used for chopped forage that is to be dumped into a pile or a bin, but it is not too satisfactory for obtaining controlled rates of unloading with chopped materials. Since forages do not have a fixed flow angle or angle of repose, the bed must be raised to a steep angle before movement begins and then the load tends to slide out as a body
or in bunches (unless it is raked out by hand before sliding occurs). Thus it is difficult to match the rate of unloading to the capacity of the forage blower. Wagon or trailer beds are usually dumped by means of external hoists or special hydraulic jacks, but they are available with built-in hydraulic cylinders operated from a pto-driven pump or from the tractor's hydraulic system. Beds may be arranged to tip to either side or to the rear.

22.11. Drag-Type and Conveyor-Type Unloaders. Wagon-unloading devices of these types have been developed primarily for handling chopped forages. In order to distribute overhead costs, they should be readily adaptable to other hauling jobs as well. Common arrangements for moving the load to the rear at a controlled rate include:

1. A false front endgate that is dragged to the rear as cables are wound onto a roller.
2. A canvas apron or covering that is laid on the floor before loading and then wrapped onto a roller at the rear during unloading.
3. A chain-and-slat conveyor that drags the load along the bottom.
4. A steel-mesh conveyor carried around a roller at one end and driven by sprockets at the other end.

The rate of linear travel of any of the four types can be controlled by adjusting the speed of the driven roller or sprockets, and one man at the rear of the wagon can easily feed the blower at any desired rate. Uniform feeding without manual assistance requires rotating beaters or other equivalent devices at the rear of the wagon.

All the systems listed above have given good performance. The false endgate tends to pack the material somewhat, and there is some delay before the rear of the load starts to move, but its low first cost and simplicity are offsetting features. If the chopped forage (particularly cured hay) is to be merely unloaded in a pile, the false endgate is sometimes pulled out directly by means of a cable and tractor. Either the false endgate or the canvas apron may readily be removed to free the wagon for general hauling. Both types must, after each unloading, be re-
placed in their proper positions in the wagon, either by hand or mechanically.

Canvas aprons (Fig. 22.3) have the advantages of starting to move the load immediately and of tending to stretch the load somewhat rather than compressing it. However, unless a suitable front-end fastening is provided, the canvas may be difficult to hold in place at the start of a load in windy weather. The canvas is subject to misalignment as it is rolled up, particularly if it contains longitudinal seams. Unless the drive roll is stiff enough to withstand the large bending loads with a minimum of deflection, the canvas will stretch out of shape and double up in the center, thus reducing its life.

The chain-and-slat and steel-mesh-conveyor types are more durable but also more expensive than the false-endgate or canvas-apron types. Mechanical problems are involved in obtaining clearance for the returning leg of the conveyor beneath the bed unless the frame is specially designed (Fig. 22.4) or a false bottom is installed. Another alternative is to have a narrow conveyor down the center with the sides sloping inward to the conveyor. Where the conveyor passes beneath the endgates it is difficult to seal the box tightly enough to hold small grain and shelled corn. Although wagons with conveyor-type unloaders are not as readily adapted to general hauling as are the other types, attachments
can be provided to permit spreading manure with them. Beaters and cross-conveyor attachments for either the front or rear end are available on some conveyor-type wagons to adapt them for the distribution of livestock feed along continuous mangers or feed racks.

22.12. Drives for Controlled-Rate Unloaders. In moving the load out of wagons with controlled-rate mechanical unloaders, linear speeds are normally between 1 and 5 fpm. Power requirements are low, seldom exceeding \( \frac{3}{4} \) hp with any of the four arrangements described in the preceding section. In tests to compare false-endgate and canvas-apron types of unloaders, Barger and his associates found that for corn silage the maximum force required to move the load (about 21/2 tons) in rectangular boxes with various types of bottoms was from 80 to 100 per cent of the load weight, whereas the average force during the entire unloading operation was from 45 to 75 per cent of the weight.

In these tests the canvas was somewhat superior on a waxed wood floor and the false endgate was equally superior on sheet metal, but the differences were not large enough to be an im-

Fig. 22.4. A chain-and-slat type of wagon unloader. Note the dual conveyor arrangement to reduce the slat length. Steel-mesh conveyors have a similar arrangement. (Pick Mfg. Co.)
important factor. Making the box 2 ft wider at the rear than at the front reduced the power requirements for the canvas apron by about 16 per cent. Hodges reports little difference in the power requirements of canvas-apron and chain-and-slat unloaders, and his data indicate maximum forces a little greater than the load weight. Loads of 5 tons or more are not uncommon when hauling silage materials.

Even with small-diameter drive rollers (2 to 4 in.) and continuous rotation, the large pulling forces result in maximum driving torques as high as 12,000 to 16,000 lb-in. Although a ratchet drive is entirely satisfactory from the functional standpoint and affords an easy method of adjusting speed, it introduces momentary peak torques considerably higher than the maximum torque with a continuously rotating roller. Variable-speed devices, such as variable-pitch V-belt sheaves, are often incorporated in the drive units.

Common power sources for portable drive units are 1/2-hp or 3/4-hp electric motors (Figs. 22.3 and 22.4) and small gasoline engines. Some of the conveyor-type unloading devices are driven from the tractor pto, particularly when the wagon has attachments for manure spreading or for feed distribution. Variable-speed drive attachments to operate unloaders are available for many of the newer forage blowers, thus utilizing the same power source that drives the blower.

TRACTOR-MOUNTED LOADERS

22.13. Types and Applications. With the general adoption and rapid improvement of hydraulic controls, tractor-mounted loading and transport equipment has become increasingly popular. Front-mounted loaders are used extensively for such operations as manure loading, earth moving and loading, and other miscellaneous jobs. Rear-mounted transport boxes or platforms, hydraulically lifted, are popular in some of the European countries. A recent development in the United States is that of fork-lift attachments to fit on the rear of farm tractors for handling pallets of boxed fruit, baled hay, etc. In some cases the tractor with fork-lift attachment moves pallets of fruit out of the orchard and loads them directly onto highway trucks.
22.14. **Front-Mounted Loaders.** The basic and almost universal attachment for front-mounted loaders is a bucket or scoop for handling dirt and other loose materials. Manure buckets or forks, which are generally available, have prongs or teeth extending forward from the bucket to aid in penetrating and tearing apart matted manure and bedding. Bulldozer blades may be mounted on many of the loader units. Other useful but less common attachments include hoists, sweep rakes, snow scoops, snow plows, etc.

Practically all front-mounted loaders are hydraulically operated. Although a few present-day models are built to be operated from the tractor's hydraulic system, the most common arrangement is a separate pump driven from the tractor pto. These pumps, which are furnished as part of the loader attachment, usually have working capacities of at least 10 to 15 gpm, which is somewhat higher than for many of the built-in pumps on medium-sized tractors. The most common pressure rating for the hydraulic systems of loaders is 1000 psi. Although rated lifting capacities generally range from 1500 to 3000 lb, these capacities usually cannot be fully utilized by the ordinary tractor for reasons explained in the following section.

22.15. **Effect of Mounted Loaders on Tractors.** One of the serious problems introduced by mounted loaders is that of weight distribution on the tractor wheels. Figure 22.5 shows a typical example with a front-mounted loader carrying a 1000-lb load. The broken-line outline at the rear indicates an alternative arrangement in which a 1500-lb load is carried at the rear (as with a fork lift) instead of having a front-mounted loader. Comparative wheel reactions for these arrangements, with and without counterweights, are presented for this example in Table 22.1. Although the weights of the loaders themselves would have an appreciable effect, they have been omitted to simplify this example.

From Table 22.1 it is evident that with 1000 lb on the front-mounted loader as shown in Fig. 22.5, the front wheels would be greatly overloaded and additional weight on the rear would be desirable for increased traction. The entire front end of the tractor would need to be specially designed to properly handle the loads imposed by the front-mounted loader.
Fig. 22.5. Typical arrangement and loading for a front-mounted loader. An alternative arrangement for a rear load, as with a fork lift, is also indicated.

Table 22.1 WHEEL REACTIONS FOR VARIOUS LOAD POSITIONS INDICATED IN FIG. 22.5

<table>
<thead>
<tr>
<th>Arrangement</th>
<th>$R_r$, lb</th>
<th>$R_f$, lb</th>
</tr>
</thead>
<tbody>
<tr>
<td>Maximum rated load for 4-ply tires shown</td>
<td>4200</td>
<td>1510</td>
</tr>
<tr>
<td>Tractor alone</td>
<td>1800</td>
<td>1200</td>
</tr>
<tr>
<td>Front-mounted loader</td>
<td>1300</td>
<td>2700</td>
</tr>
<tr>
<td>Front-mounted loader with 400-lb counterweight 30 in. behind rear axle</td>
<td>1850</td>
<td>2550</td>
</tr>
<tr>
<td>Rear-mounted loader or carrier</td>
<td>4275</td>
<td>225</td>
</tr>
<tr>
<td>Rear-mounted loader with 400-lb counterweight 40 in. ahead of front axle</td>
<td>4075</td>
<td>825</td>
</tr>
</tbody>
</table>

The 1500-lb rear load is about the maximum permissible for the tires in the example. A front counterweight is necessary to give sufficient weight on the front wheels for stability and steering. Lift boxes supported by the usual hydraulic lift linkage of the tractor ordinarily would not be subjected to large loads and probably would not be so far behind the rear axle.

REFERENCES


PRINCIPLES OF FARM MACHINERY


   VI. McKibben, Eugene G., and J. Brownlee Davidson. Effect of Steel Wheel Rim Shape and Pneumatic Tire Tread Design on Rolling Resistance. 21:139-140, April, 1940.


   X. Davidson, J. Brownlee, and Eugene G. McKibben. The Value and Cost of Pneumatic Tires. 21:319-321, August, 1940.
PROBLEMS

22.1. Determine the rolling resistance of the tractor indicated in Fig. 22.5 (without loaders) when operating in a settled loam with Rototiller penetrometer reading of 8 in. The rear tires are inflated to 16 psi and the front tires to 36 psi. Consider that the front and rear wheels do not track.

22.2. The speedometer on a tractor is driven by contact with a 5.00-15-in. front tire (25.6 in. OD, 12.0 in. loaded radius) inflated to 32 psi, and indicates its peripheral speed directly. Plot speedometer correction factor versus Rototiller penetrometer reading.

22.3. A 14-ft (bed length) wagon with a 5500-lb load of chopped corn is to be unloaded into a forage blower at a rate of 1000 lb per min, using a canvas apron and a 3½-in. diameter roller. Calculate:

(a) The required rpm of the roller.
(b) The maximum torque required by the roller, assuming the maximum force equals 100 per cent of the load weight.
(c) The maximum horsepower input to the roller.

22.4. A front-mounted loader is equipped with a 12-gpm hydraulic pump and has a rated lifting capacity of 2000 lb at a pump pressure of 1000 psi. Under these conditions, determine the time required to lift the load from the ground to the maximum height of 9 ft. Assume a 10 per cent energy loss in the oil lines, hydraulic cylinder, and lift linkage.
APPENDIX A

V-BELT DRIVES FOR FARM MACHINES
(V-BELTS, DOUBLE V-BELTS, AND ADJUSTABLE-SPEED BELTS)
(ASAE STANDARD, ADOPTED JUNE, 1950)

This standard covers the application of V-belt drives to farm machines, other than tractors, and is intended primarily to include drives using V-belts that have fiber tensile members. The purpose of the standard is to provide agricultural engineers with sufficient technical data to enable them to properly apply V-belt drives to farm machines. Acceptable manufacturing tolerances, methods of measuring, and proper application are included. Use of this standard will contribute to the design of simple and economical drives that in turn will insure satisfactory service to the user.

This standard has been approved by a joint V-belt technical committee representing the Rubber Manufacturers Association and the Farm Equipment Institute, and it is supplemental to existing standards (a) for V-belts on automotive fan drives published by the Society of Automotive Engineers and (b) for multiple V-belts and sheaves on industrial drives published by the Rubber Manufacturers Association.

Standard Cross-Sections. Standard cross-sections of agricultural V-belts (designated by letter H) have been developed to cover the needs of designers and users of farm machines, and belts with these cross-sections are available from manufacturers of V-belts. Nominal dimensions of these cross-sections are shown in Fig. A-1. (Note: The dimensions and cross-sectional shapes shown in Fig. A-1 are nominal only. Actual cross-sectional dimensions of belts made by one manufacturer may differ from those of the same belt cross-section made by another manufacturer. Because of different constructions and different methods of manufacturing, the cross-sectional shapes and the included angle between the sidewalls may be different for different manufacturers. However, all standard cross-sections will operate interchangeably in standard grooves. See Tables A-4, A-5, and A-6.)
Available Lengths. Production equipment exists generally for the manufacture of agricultural V-belts, double V-belts, and adjustable-speed belts in the lengths shown in Tables A-1 and A-2. These lengths should be used if at all possible, as their use will result in the design of simplified and economical drives for farm machines.

Method of Measuring Lengths of V-Belts. The “effective length” of a V-belt depends upon the position at which the belt rides in the sheave groove as well as upon the actual length of the belt. The cross-sectional dimensions and the lengths of agricultural V-belts are to be determined on a belt-measuring fixture (Fig. A-2), consisting of two sheaves of equal diameter having
Table A-1 **EFFECTIVE LENGTHS OF AGRICULTURAL V-BELTS AND DOUBLE V-BELTS**

**Effective Lengths, in.**

<table>
<thead>
<tr>
<th>V-Belt Cross-Sections</th>
<th>Double V-Belt Cross-Sections</th>
<th>Length Tolerance, in.†</th>
</tr>
</thead>
<tbody>
<tr>
<td>HA</td>
<td>HB</td>
<td>HC</td>
</tr>
<tr>
<td>28.1</td>
<td>33.1</td>
<td>35.1</td>
</tr>
<tr>
<td>55.1</td>
<td>57.1</td>
<td>57.9</td>
</tr>
<tr>
<td>73.1</td>
<td>77.9</td>
<td>79.2</td>
</tr>
<tr>
<td>92.1</td>
<td>92.9</td>
<td>94.2</td>
</tr>
<tr>
<td>111.1</td>
<td>114.9</td>
<td>116.2</td>
</tr>
<tr>
<td>138.9</td>
<td>140.2</td>
<td>146.9</td>
</tr>
<tr>
<td>160.9</td>
<td>162.2</td>
<td>163.2</td>
</tr>
<tr>
<td>183.5</td>
<td>184.3</td>
<td>185.2</td>
</tr>
<tr>
<td>197.9</td>
<td>200.2</td>
<td>202.0</td>
</tr>
<tr>
<td>213.9</td>
<td>214.2</td>
<td>215.3</td>
</tr>
<tr>
<td>241.4</td>
<td>242.2</td>
<td>242.7</td>
</tr>
<tr>
<td>271.4</td>
<td>272.2</td>
<td>272.7</td>
</tr>
<tr>
<td>301.4</td>
<td>302.2</td>
<td>302.7</td>
</tr>
<tr>
<td>332.2</td>
<td>332.7</td>
<td>335.5</td>
</tr>
<tr>
<td>362.2</td>
<td>362.7</td>
<td>365.5</td>
</tr>
</tbody>
</table>

* Effective lengths are determined by the use of measurements described under "Method of Measuring Lengths of V-Belts" in the accompanying text.

† Length tolerance includes belt manufacturing tolerance and normal belt shrinkage.
### Table A-2 EFFECTIVE LENGTHS OF AGRICULTURAL ADJUSTABLE-SPEED BELTS (ALL CROSS-SECTIONS)

<table>
<thead>
<tr>
<th>Effective Lengths, in.</th>
<th>Length Tolerance, in.*</th>
</tr>
</thead>
<tbody>
<tr>
<td>60</td>
<td>±0.4</td>
</tr>
<tr>
<td>64</td>
<td>-0.9</td>
</tr>
<tr>
<td>68</td>
<td>±0.5</td>
</tr>
<tr>
<td>72</td>
<td>-1.0</td>
</tr>
<tr>
<td>76</td>
<td></td>
</tr>
<tr>
<td>80</td>
<td></td>
</tr>
<tr>
<td>84</td>
<td></td>
</tr>
<tr>
<td>88</td>
<td></td>
</tr>
<tr>
<td>92</td>
<td></td>
</tr>
<tr>
<td>96</td>
<td></td>
</tr>
<tr>
<td>104</td>
<td></td>
</tr>
<tr>
<td>112</td>
<td></td>
</tr>
<tr>
<td>120</td>
<td></td>
</tr>
<tr>
<td>128</td>
<td></td>
</tr>
</tbody>
</table>

*Length tolerance includes belt manufacturing tolerance and normal belt shrinkage.

### Table A-3 DATA FOR USE IN MEASURING LENGTHS OF V-BELTS

<table>
<thead>
<tr>
<th>Belt Cross-Section</th>
<th>Sheave OD, in.</th>
<th>Sheave OC, in.</th>
<th>Groove Angle, deg</th>
<th>Groove Width, in.</th>
<th>Groove Depth, in.</th>
<th>Total Tension, lb</th>
<th>Position of Belt with Respect to Top of Sheave</th>
</tr>
</thead>
<tbody>
<tr>
<td>HA, HAA</td>
<td>3.183</td>
<td>10.000</td>
<td>±0° - 20°</td>
<td>0.400</td>
<td>0.400</td>
<td>50</td>
<td>+5 x to -5 x</td>
</tr>
<tr>
<td>HB, HBB</td>
<td>4.775</td>
<td>15.000</td>
<td>32</td>
<td>0.630</td>
<td>0.580</td>
<td>65</td>
<td>+5 x to -5 x</td>
</tr>
<tr>
<td>HC, HCC</td>
<td>7.058</td>
<td>25.000</td>
<td>34</td>
<td>0.870</td>
<td>0.780</td>
<td>105</td>
<td>+5 x to -5 x</td>
</tr>
<tr>
<td>HD, HDD</td>
<td>11.141</td>
<td>35.000</td>
<td>34</td>
<td>1.250</td>
<td>1.050</td>
<td>300</td>
<td>+5 x to -5 x</td>
</tr>
<tr>
<td>HE</td>
<td>19.000</td>
<td>60.000</td>
<td>36</td>
<td>1.527</td>
<td>1.300</td>
<td>400</td>
<td>+5 x to -5 x</td>
</tr>
<tr>
<td>HJ</td>
<td>9.540</td>
<td>30.000</td>
<td>26</td>
<td>1.250</td>
<td>0.938</td>
<td>100</td>
<td>+5 x to -5 x</td>
</tr>
<tr>
<td>HK</td>
<td>9.540</td>
<td>30.000</td>
<td>26</td>
<td>1.500</td>
<td>1.000</td>
<td>225</td>
<td>+5 x to -5 x</td>
</tr>
<tr>
<td>HL</td>
<td>9.540</td>
<td>30.000</td>
<td>26</td>
<td>1.750</td>
<td>1.125</td>
<td>310</td>
<td>+5 x to -5 x</td>
</tr>
<tr>
<td>HM</td>
<td>9.540</td>
<td>30.000</td>
<td>26</td>
<td>2.000</td>
<td>1.388</td>
<td>400</td>
<td>+5 x to -5 x</td>
</tr>
</tbody>
</table>

Note 1. Grooves of master inspection sheaves shall be machined to tolerances shown in Table A-3, treated to resist wear, and checked periodically for wear.

Note 2. Any other diameters of measuring sheaves may be used provided the grooves conform (within the tolerance shown in Table A-3) to dimensions given in Tables A-4 or A-6.

Note 3. The position of belt with respect to top of the sheave groove is shown in the last column of Table A-3 (see also Fig. A-3). The approved method of interpreting the ride-out or the position of the belt with respect to the top of the sheave groove is to measure from the top of the measuring-sheave groove to the outermost part of the belt in a radial direction. For double V-belts, this dimension is from the center of the belt to the top of the sheave groove.
standard groove dimensions (see Table A-3). One of the sheaves is fixed in position, while the other is movable along a graduated scale and will have tension applied to it. Belts are to be placed on this fixture, the proper tension applied to the movable sheave, and the belt pulled around the sheaves a minimum of three revolutions of the belt. Cross-sectional dimensions are to be checked by observing the "ride" of the belt in the sheave grooves. Effective length of the belt will be determined by adding twice the center distance measured on the fixture to the product of 3.14 times the outside diameter of one sheave.

STANDARDS FOR SHEAVES USED WITH V-BELTS

Construction. Sheaves used with agricultural V-belts should be made of a material which is resistant to abrasion between the groove wall and the belt and sufficiently close-grained to take a smooth finish on the groove sidewalls.

Cast-iron sheaves should have machine-finished groove surfaces and should present a smooth surface to the belts. Sheaves formed from sheet metal should be so made that the top width and angle of the groove are constant throughout the circumference of the sheave. The gage of the sheet metal used should be such that the groove will not spread under the load imposed by the belt.

Adjustable-speed sheaves should be so designed that the movable disk is perpendicular to the axis of rotation at all times without appreciable run-out or wobble. Failure to accomplish this results in a nonuniform groove width which materially reduces belt life and may set up undesirable vibration of the belt or the machine on which it is used.

Groove dimensions with essential tolerances for agricultural V-belts are shown in Tables A-4, A-5, and A-6. The tolerance on outside diameters of sheaves should be $\pm \frac{1}{32}$ in. for sheaves having outside diameters less than 14 in. and $\pm \frac{1}{16}$ in. on sheaves having outside diameters of 14 in. or more.

DEEP-GROOVE SHEAVES

Deep-groove sheaves are used on quarter-turn drives or on other drives where the belt must enter the groove at an angle.
Table A-4  GROOVE DIMENSIONS FOR V- AND DOUBLE V-BELTS

<table>
<thead>
<tr>
<th>Belt Cross-Section</th>
<th>Effective Outside Diameter, in.</th>
<th>$A$, deg</th>
<th>$W$, in.</th>
<th>$G$, in. (min)</th>
<th>$2x$, in.</th>
</tr>
</thead>
<tbody>
<tr>
<td>HA</td>
<td>Under 2.65</td>
<td>30</td>
<td>0.485</td>
<td>0.490</td>
<td>0.250</td>
</tr>
<tr>
<td></td>
<td>2.65 to 3.24</td>
<td>32</td>
<td>0.490</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>3.25 to 5.65</td>
<td>34</td>
<td>0.494</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Over 5.65</td>
<td>38</td>
<td>0.504</td>
<td></td>
<td></td>
</tr>
<tr>
<td>HB</td>
<td>Under 3.95</td>
<td>30</td>
<td>0.624</td>
<td>0.580</td>
<td>0.350</td>
</tr>
<tr>
<td></td>
<td>3.95 to 4.94</td>
<td>32</td>
<td>0.630</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>4.95 to 7.35</td>
<td>34</td>
<td>0.637</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Over 7.35</td>
<td>38</td>
<td>0.650</td>
<td></td>
<td></td>
</tr>
<tr>
<td>HC</td>
<td>Under 5.40</td>
<td>30</td>
<td>0.864</td>
<td>0.780</td>
<td>0.400</td>
</tr>
<tr>
<td></td>
<td>5.40 to 6.39</td>
<td>32</td>
<td>0.872</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>6.40 to 8.39</td>
<td>34</td>
<td>0.879</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>8.40 to 12.40</td>
<td>36</td>
<td>0.887</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Over 12.40</td>
<td>38</td>
<td>0.895</td>
<td></td>
<td></td>
</tr>
<tr>
<td>HD</td>
<td>Under 8.60</td>
<td>30</td>
<td>1.237</td>
<td>1.050</td>
<td>0.600</td>
</tr>
<tr>
<td></td>
<td>8.60 to 10.59</td>
<td>32</td>
<td>1.248</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>10.60 to 13.59</td>
<td>34</td>
<td>1.259</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>13.60 to 17.60</td>
<td>36</td>
<td>1.271</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Over 17.60</td>
<td>38</td>
<td>1.283</td>
<td></td>
<td></td>
</tr>
<tr>
<td>HE</td>
<td>Under 12.80</td>
<td>32</td>
<td>1.496</td>
<td>1.300</td>
<td>0.800</td>
</tr>
<tr>
<td></td>
<td>12.80 to 16.79</td>
<td>34</td>
<td>1.512</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>16.80 to 24.80</td>
<td>36</td>
<td>1.527</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Over 24.80</td>
<td>38</td>
<td>1.542</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

* The values of $2x$ in Table A-4 apply to the use of V-belts; for double V-belts the pitch diameter of the sheave is equal to the effective outside diameter of the sheave. (Note: "Effective outside diameter" is explained in the accompanying text.)
<table>
<thead>
<tr>
<th>Belt Cross-Section</th>
<th>Outside Diameter, in.</th>
<th>A, deg</th>
<th>W, in. ±0.005</th>
<th>G, in. (min)</th>
<th>2x, in.</th>
</tr>
</thead>
<tbody>
<tr>
<td>HA</td>
<td>Under 2.96</td>
<td>30</td>
<td>0.568</td>
<td>0.645</td>
<td>0.560</td>
</tr>
<tr>
<td></td>
<td>2.96 to 3.55</td>
<td>32</td>
<td>0.579</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>3.56 to 5.96</td>
<td>34</td>
<td>0.589</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Over 5.96</td>
<td>38</td>
<td>0.611</td>
<td></td>
<td></td>
</tr>
<tr>
<td>HB</td>
<td>Under 4.31</td>
<td>30</td>
<td>0.720</td>
<td>0.760</td>
<td>0.710</td>
</tr>
<tr>
<td></td>
<td>4.31 to 5.30</td>
<td>32</td>
<td>0.734</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>5.31 to 7.71</td>
<td>34</td>
<td>0.747</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Over 7.71</td>
<td>38</td>
<td>0.774</td>
<td></td>
<td></td>
</tr>
<tr>
<td>HC</td>
<td>Under 6.01</td>
<td>30</td>
<td>1.028</td>
<td>1.085</td>
<td>1.010</td>
</tr>
<tr>
<td></td>
<td>6.01 to 7.00</td>
<td>32</td>
<td>1.047</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>7.01 to 9.00</td>
<td>34</td>
<td>1.066</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>9.01 to 13.01</td>
<td>36</td>
<td>1.085</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Over 13.01</td>
<td>38</td>
<td>1.105</td>
<td></td>
<td></td>
</tr>
<tr>
<td>HD</td>
<td>Under 9.43</td>
<td>30</td>
<td>1.459</td>
<td>1.465</td>
<td>1.430</td>
</tr>
<tr>
<td></td>
<td>9.43 to 11.42</td>
<td>32</td>
<td>1.486</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>11.43 to 14.42</td>
<td>34</td>
<td>1.513</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>14.43 to 18.43</td>
<td>36</td>
<td>1.541</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Over 18.43</td>
<td>38</td>
<td>1.569</td>
<td></td>
<td></td>
</tr>
<tr>
<td>HE</td>
<td>Under 13.69</td>
<td>32</td>
<td>1.751</td>
<td>1.745</td>
<td>1.690</td>
</tr>
<tr>
<td></td>
<td>13.69 to 17.68</td>
<td>34</td>
<td>1.783</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>17.69 to 25.69</td>
<td>36</td>
<td>1.816</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Over 25.69</td>
<td>38</td>
<td>1.849</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
Table A-6  GROOVE DIMENSIONS FOR ADJUSTABLE-SPEED SHEAVES

<table>
<thead>
<tr>
<th>Bell</th>
<th>Minimum Outside Diameter, in.</th>
<th>Recommended $W$, in. $\pm 0.005$</th>
<th>Maximum Lateral Movement, in. $2x$</th>
</tr>
</thead>
<tbody>
<tr>
<td>HJ</td>
<td>$8\frac{3}{4}$</td>
<td>1 3/4</td>
<td>2.71 0.98 0.37</td>
</tr>
<tr>
<td>HK</td>
<td>$10\frac{1}{2}$</td>
<td>1 3/4</td>
<td>3.25 1.18 0.45</td>
</tr>
<tr>
<td>HL</td>
<td>$12\frac{1}{4}$</td>
<td>1 3/4</td>
<td>3.79 1.39 0.52</td>
</tr>
<tr>
<td>HM</td>
<td>14</td>
<td>2</td>
<td>4.33 1.60 0.60</td>
</tr>
</tbody>
</table>

Note 1. Adjustable-speed drives should be arranged so that the belt will run in all positions with the least possible misalignment between sheaves.

Note 2. Be sure to check the rim speed of the driven sheave at its maximum rpm to see that it is within safe limits.

Note 3. Make positive provision for clearance below the bottom of the belt when the belt is operated at a minimum diameter.

**DESIGN DATA**

**Effective Outside Diameters of Sheaves and Pulleys.** The "effective" outside diameter of a sheave is the diameter at which the groove width is equal to dimension $W$ shown in Tables A-4 and A-6. This diameter will be equal to the outside diameter of the sheave if the grooves are machined exactly according to Tables A-4 and A-6. But for sheaves pressed from sheet metal, sheaves with deep grooves, etc., the effective outside diameter may be less than the actual outside diameter of the sheave.

If flat-faced pulleys or idlers are used, the effective outside diameter may be different from the actual outside diameter. If the top of the belt runs against the flat-faced pulley, the effective outside diameter may be taken as the actual outside diameter without introducing an appreciable error. If the bottom of a
V-belt rides on the pulley, the effective outside diameter will be approximately equal to the actual outside diameter plus two times the nominal thickness of the belt. If a double V-belt rides on a flat pulley, the effective outside diameter will be equal to the

actual outside diameter of the pulley plus the nominal thickness of the belt. These relationships are illustrated in the several sketches in Fig. A-4.

**Center Distance and Belt Length.** For drives with two sheaves (Fig. A-5) the relation between center distance and belt length is as follows:

\[
L = 2C + 1.57(D + d) + \frac{(D - d)^2}{4C}
\]  

(A-1)
where $L =$ effective length of belt, in inches (Tables A-1 and A-2).

$C =$ distance between centers of sheaves, in inches.

$D =$ effective outside diameter of large sheave, in inches.

$d =$ effective outside diameter of small sheave, in inches.

If sheave diameters ($D$ and $d$) and belt length ($L$) are known, the center distance between sheaves may be calculated by means of formula A-2 as follows:

$$C = a + \sqrt{a^2 - b}$$

(A-2)

where $a = -0.393(D + d)$

$$b = 0.125(D - d)^2$$

The following methods may be used for determining belt length when more than two sheaves are used on a drive (Fig. A-6).

**Method 1.** Sheaves in terms of their effective outside diameters should be accurately laid out to scale in the position desired when a new belt is applied and first brought to driving tension. The length of belt will then be the sum of the tangents and connecting arcs around the effective outside diameters of the sheaves. The length of connecting arcs can be calculated by the formula: Length of arc $= DA/115$, in which $D$ is the effective outside diameter of the sheave and $A$ is the angle.
in degrees subtended by the arc of belt contact on the sheave.

Method 2. With the drive laid out to scale, insert pins every \( \frac{3}{4} \) in. on the effective outside diameters of the sheaves in contact with the belt. Measure the length around these pins with a flexible steel tape.

Method 3. With the drive laid out to scale, measure the length of line representing the effective outside length of the belt with a map measure.

Method 4. With the drive laid out to scale, cut disks to the effective outside diameters of the sheaves, place these disks in their proper position on the drawing, and measure around them with a tape or steel wire.

Installation and Take-Up. The belt length obtained by one of the methods above will be the effective length of an ideal belt under operating tension. A belt must be so arranged that any belt within the length tolerances given in Tables A-1 and A-2 can be placed in the sheave groove without being prised over the edge. In addition, provision must be made to compensate for the change in effective length caused by the seating of the belt in the sheave groove and by the stretch of the belt during its life.

Fig. A-7. The sketches illustrate the allowances for installation and take-up in agricultural V-belts.
Installation and take-up allowances as shown in Table A-7 should be provided on every V-belt drive to ensure satisfactory operation. These allowances are in terms of effective belt length.

**Table A-7 INSTALLATION AND TAKE-UP ALLOWANCES FOR AGRICULTURAL V-BELTS**

<table>
<thead>
<tr>
<th>Effective Belt Length, in.</th>
<th>Allowance for Installation of Belt, in.*</th>
<th>Allowance for Stretch and Wear, All Sections, in.†</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>HA</td>
<td>HB</td>
</tr>
<tr>
<td>25 to 60</td>
<td>1 ¼</td>
<td>1 ½</td>
</tr>
<tr>
<td>60.1 to 80</td>
<td>1 ½</td>
<td>1 ½</td>
</tr>
<tr>
<td>80.1 to 100</td>
<td>1 ½</td>
<td>1 ½</td>
</tr>
<tr>
<td>100.1 to 114</td>
<td>13 ½</td>
<td>13 ½</td>
</tr>
<tr>
<td>114.1 to 122</td>
<td>13 ½</td>
<td>23 ½</td>
</tr>
<tr>
<td>122.1 to 130</td>
<td>2</td>
<td>2 ½</td>
</tr>
<tr>
<td>120.1 to 138</td>
<td>2½</td>
<td>2½</td>
</tr>
<tr>
<td>138.1 to 140</td>
<td>2½</td>
<td>2½</td>
</tr>
<tr>
<td>148.1 to 160</td>
<td>2½</td>
<td>2½</td>
</tr>
<tr>
<td>160.1 to 164</td>
<td>2½</td>
<td>2½</td>
</tr>
<tr>
<td>164.1 to 175</td>
<td>2½</td>
<td>2½</td>
</tr>
<tr>
<td>175.1 to 182</td>
<td>2½</td>
<td>2½</td>
</tr>
<tr>
<td>182.1 to 197</td>
<td>2½</td>
<td>2½</td>
</tr>
<tr>
<td>197.1 to 212</td>
<td>2½</td>
<td>2½</td>
</tr>
<tr>
<td>212.1 to 240</td>
<td>2½</td>
<td>2½</td>
</tr>
<tr>
<td>240.1 to 270</td>
<td>2½</td>
<td>2½</td>
</tr>
<tr>
<td>270.1 to 300</td>
<td>2½</td>
<td>2½</td>
</tr>
<tr>
<td>300.1 to 330</td>
<td>2½</td>
<td>2½</td>
</tr>
<tr>
<td>330.1 to 360</td>
<td>2½</td>
<td>2½</td>
</tr>
<tr>
<td>360.1 to 390</td>
<td>2½</td>
<td>2½</td>
</tr>
</tbody>
</table>

* Allowance for installation includes the minus manufacturing length tolerance from Tables A-1 and A-2, the difference between the length of belt under no tension and the length under measuring tension, and an amount for installing the belts over the sheave flanges without injury.

† Allowance for stretch and wear includes the plus manufacturing tolerance from Tables A-1 and A-2 as well as an allowance for the stretch and wear of the belt resulting from service on the drive.

Examples of the calculation of center distance, effective length, and installation and take-up allowances are shown in the sample drive sketches (Sheets A-1, A-2, A-3, and A-4) on pages 531-534.

**Cross Drives, Mule Drives, and Other Twisted-Belt Drives.**

The minimum tangent length for a 180° twist in a V-belt is shown in Table A-8.

- The minimum tangent length for any amount of twist other than 180° can be obtained by multiplying the minimum tangent...
Table A-8  MINIMUM TANGENT LENGTHS FOR 180° TWIST IN AGRICULTURAL V-BELTS

<table>
<thead>
<tr>
<th>Belt Cross-Section</th>
<th>Tangent Length, in. (min)</th>
</tr>
</thead>
<tbody>
<tr>
<td>HA</td>
<td>18</td>
</tr>
<tr>
<td>HB</td>
<td>22</td>
</tr>
<tr>
<td>HC</td>
<td>28</td>
</tr>
<tr>
<td>HD</td>
<td>37</td>
</tr>
<tr>
<td>HE</td>
<td>45</td>
</tr>
</tbody>
</table>

length in Table A-8 by the fraction: degrees of twist required divided by 180.

Agricultural double V- and adjustable-speed belts should never be used on drives of this kind.

**Quarter-Turn Drives.** On quarter-turn drives, the fleeting angle (angle of entry of the belt into the plane of the sheave groove) should not exceed 5°. A center distance at least \(5\frac{1}{2}\) times the diameter of the large sheave is necessary to insure this condition where a single belt is used.

**Pitch Diameter, Speed Ratio, and Belt Speed.** Pitch diameters of sheaves rather than outside diameters should be used in calculating speed ratios and belt speeds to secure necessary accuracy. When V-belts or adjustable-speed belts are used, the \(2x\) values shown in Tables A-4 and A-6 are to be subtracted from the effective outside diameters of sheaves to obtain pitch diameter. For V-belts in deep groove sheaves, use the data in Table A-5. When double V-belts are used, the effective outside diameter of the sheave is equal to the pitch diameter. Pitch diameters are used only in calculating speed ratios and belt speeds.

**Sheave Diameters.** In designing V-belt drives, it should be recognized that the use of larger sheave diameters will result in lower bearing loads and can result in the use of smaller and less expensive belt cross-sections.

**Idlers.** Idlers may be necessary on agricultural V-belt drives to provide take-up or to increase the arc of contact to obtain the required drive capacity. If an idler is needed, it should be located on the slack side of the drive. Other things that affect the location of the idler are its effectiveness in belt take-up and its effect on arcs of contact.
An idler should have its axis of rotation perpendicular to the plane of the belt strand on which it runs. The idler mounting should be heavy enough to maintain this relationship at all times.

If grooved idlers are used, the groove dimensions should be as shown in Tables A-4 and A-5. Grooved idlers for adjustable-speed belts should have the groove top width and angle shown in Table A-6, but the depth would need to be only great enough to accommodate the belt thickness.

Minimum diameters recommended for idlers are shown in Table A-9.

Table A-9 MINIMUM DIAMETERS FOR IDLERS

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>HA, HAA</td>
<td>2.75</td>
<td>2.25</td>
<td>4.25</td>
<td>1.00</td>
</tr>
<tr>
<td>HB, HBB</td>
<td>4.00</td>
<td>3.75</td>
<td>6.00</td>
<td>1.25</td>
</tr>
<tr>
<td>HC, HCC</td>
<td>5.75</td>
<td>4.75</td>
<td>8.50</td>
<td>1.50</td>
</tr>
<tr>
<td>HD</td>
<td>9.00</td>
<td>7.50</td>
<td>13.50</td>
<td>2.00</td>
</tr>
<tr>
<td>HE</td>
<td>13.00</td>
<td>11.25</td>
<td>19.50</td>
<td>2.38</td>
</tr>
<tr>
<td>HJ</td>
<td>6.75</td>
<td>5.63</td>
<td>10.00</td>
<td>2.00</td>
</tr>
<tr>
<td>HK</td>
<td>8.00</td>
<td>6.75</td>
<td>12.00</td>
<td>2.25</td>
</tr>
<tr>
<td>HL</td>
<td>9.25</td>
<td>7.75</td>
<td>13.75</td>
<td>2.50</td>
</tr>
<tr>
<td>HM</td>
<td>10.50</td>
<td>8.75</td>
<td>15.75</td>
<td>2.75</td>
</tr>
</tbody>
</table>

Specifications of V-Belt Drives. In submitting a drive design problem to the engineering departments of the different V-belt manufacturers, it is strongly recommended that complete information be given. The sketches that follow (Sheets A-1 through A-4) are suggested as examples of the data needed on the blueprint of a proposed drive.

EXAMPLES OF THE CALCULATION OF BELT LENGTH, CENTER DISTANCE, INSTALLATION AND TAKE-UP ALLOWANCES, AND INSPECTION REQUIREMENTS

Example 1. (Refer to Sheet A-1.) The drive consists of two sheaves and one of the shafts may be moved for installation and take-up. Effective diameters have been determined. The preferred center distance is about 20 in.
Sheet A-1. Typical two-sheave drive with one shaft movable for take-up. (See Sheet A-2 for inspection requirements.)

**Belt Length and Center Distance:**

1. Substitute the effective diameters and preferred center distance in formula A-1. The belt length \( L \) required will be 79.03 in.
2. The effective length of the nearest standard belt (HB belt cross-section, Table A-1) is 77.9 in.
3. This length substituted in formula A-2 will give a center distance of 19\( \frac{1}{2} \) in.

**Installation Allowance:**

1. From Table A-7, the installation allowance will be 1\( \frac{1}{2} \) in.
2. Subtract this amount from the effective belt length of 77.9 in. to get a length of 76.28 in.
3. This length substituted in formula A-2 will give a center distance of 18\( \frac{7}{16} \) in., the minimum center distance needed for installation of the belt.

**Take-Up Allowance:**

1. From Table A-7, the allowance needed for take-up is 2\( \frac{3}{8} \) in.
2. Add this amount to the effective belt length of 77.9 in. to get a maximum length of 80.28 in.
3. This length substituted in formula A-2 will give the maximum required center distance of $20^{21/2}$ in.

**Inspection Requirements:** (See Example 2.)

![Inspection Diagram]

Sheet A-2. Typical inspection diagram to be placed on each blueprint of a proposed V-belt drive.

**Example 2.** (Refer to Sheet A-2.) Fill in the inspection requirements for the belt required in Example 1.

1. Fill in values from Table A-3 as follows:

   $T = 65$ lb  
   $G = 0.580$ in.  
   $W = 0.630$ in.  
   $A = 32^\circ$  
   $OD = 4.775$ in.  

   "Ride" = $\pm 1/16$ in. ("Ride" is position of belt with respect to top of sheave groove.)

   Note also from Table A-3 that the measuring sheave $OC$ is 15,000 in.

2. From the effective length of 77.9 in., subtract 15,000 in. and divide the remainder by 2 to find $y$, or

   $y = (77.9 - 15,000)/2 = 31.45$ in.

3. From Table A-1, the length tolerance is $+0.4$, $-0.9$. The tolerance on dimension $y$ will be equal to these length tolerances divided by 2, or

   Tolerance on $y = +0.2$, $-0.45$
Example 3. (Refer to Sheet A-3.) The effective diameters have been determined. Both shafts are fixed in position, and the

center distance is $26^{3/4}$ in. An 8½-in. outside diameter flat idler will be used for take-up on the drive.

*Belt Length:*

1. Substitute the effective outside diameters of the sheaves and the fixed center distance of $26^{3/4}$ in. in formula A-1. The resulting belt length is 83.14 in.
2. Since the centers cannot be moved for installation, the shortest possible belt must go on the drive with the idler out of the way. Consequently, the installation allowance must be
added to the belt length obtained above. The installation allowance from Table A-7 is 2 3/16 in. This, added to the length 83.14 in., gives a required effective belt length of 85.2 in. Table A-1 shows this required length as a standard length.

**Take-Up Allowance:**

From Table A-7, the take-up allowance needed for this belt is 3 in. This amount added to the effective belt length of 85.2 in., gives a maximum length of 88.2 in. By one of the methods outlined above for determining belt length when more than two sheaves are used on a drive, locate the position of the idler so that it will take up this length of belt.

---

**Sheet A-4.** Double V-belt drive with four sheaves on fixed centers. Idler used for take-up.
Example 4. (Refer to Sheet A-4.) The effective diameters have been selected and shaft centers have been located approximately. All shafts will be fixed in position and belt take-up will be accomplished by means of a grooved idler pulley.

**Belt Length:**

1. With the idler in its "installation position," use one of the methods outlined above for determining belt length when more than two sheaves are used on a drive, to determine the length of belt on the drive.

2. To find the length of belt to order for the drive, add to the length obtained in step 1 the allowance for installation from Table A-7.

3. If the length of belt to be ordered is not a standard length as shown in Table A-1, it may be possible to change wheel diameters or center distances enough to accommodate a standard length.

**Take-Up Allowance:**

To the length of belt to be ordered for the drive add the allowance for take-up from Table A-7. Check the drive with the idler in its maximum take-up position to see that this length of belt can be accommodated.
APPENDIX B

APPLICATION OF HYDRAULIC REMOTE CONTROL TO
FARM TRACTORS AND TRAILING-TYPE FARM IMPLEMENTS
(ASAE-SAE STANDARD, REVISED, 1951)

FOREWORD

This standard was developed by the Advisory Engineering Committee of the Farm Equipment Institute. Adopted originally by ASAE in March, 1949, it was later revised and expanded, and approved in its present form, July, 1951, as an official ASAE standard.

APPLICATION

Hydraulic remote controls for all general-purpose and other farm tractors with a capacity up to 6000 lb maximum drawbar pull shall include a cylinder with 8-in. working stroke. Trailing-type hydraulically controlled farm implements intended for use with such tractors shall provide standard mounting points and clearance space for this cylinder.

Hydraulic remote controls for farm tractors larger than the above, up to those with a capacity of 11,000 lb maximum drawbar pull shall regularly include a cylinder with 8-in. working stroke. Trailing-type hydraulically controlled farm implements intended for use with such tractors and requiring an operating thrust within the capacity of an 8-in.-stroke cylinder, shall provide standard mounting points and clearance space for this cylinder. Manufacturers of such tractors shall, however, make available cylinders with 16-in. working stroke for use with implements for which an 8-in.-stroke cylinder is inadequate. Implements requiring a 16-in.-stroke cylinder include deep-tillage plows, five-furrow moldboard plows, heavy-duty disk plows, deep-tillage tool carriers, and offset disk harrows, 9-ft cut and over.

Hydraulic remote controls for all farm tractors with a capacity above 11,000 lb and up to 20,000 lb maximum drawbar pull shall
include a cylinder with 16-in. working stroke. Trailing-type hydraulically controlled farm implements intended for use with such tractors shall provide standard mounting points and clearance space for this cylinder.

Since most implements intended for use with tractors having a capacity over 20,000 lb maximum drawbar pull are regularly provided with one or more suitable cylinders, they impose little requirement for the interchangeable use of hydraulic cylinders. For this reason, hydraulic controls for such tractors are not considered within this standard.

**DEFINITION**

The purpose of the standard is to establish common mounting and clearance dimensions for hydraulic remote-control cylinders and trailing-type farm implements with such other specifications as are necessary to accomplish the following objectives:

(a) To permit use of any make or model of trailing-type farm implement adapted for control by an 8-in.-stroke hydraulic cylinder, with the 8-in.-stroke hydraulic cylinder furnished with any make or model of farm tractor.

(b) To permit use of any make or model of trailing-type farm implement adapted for control by a 16-in.-stroke hydraulic cylinder, with the 16-in.-stroke hydraulic cylinder furnished with any make or model of farm tractor, consistent with the maximum drawbar pull of the tractor normally required to operate the implement.

(c) To facilitate changing the hydraulic cylinder from one implement to another and decrease the possibility of introducing dirt or other foreign material into the hydraulic system, by reducing the necessity for supplemental hose lengths or piping with certain types of implements.

**STANDARD DIMENSIONS AND SPECIFICATIONS FOR HYDRAULIC REMOTE CONTROLS INCLUDING A CYLINDER WITH 8-IN. WORKING STROKE**

1. The hydraulic cylinder with hose shall be considered as part of the tractor hydraulic remote control and shall be built to standard dimensions.
2. Both single- and double-acting cylinders shall operate to raise the implements (or de-angle disk harrows) on their extending stroke. Implements requiring actuating force in both directions should be operated by a double-acting cylinder.

3. Provision shall be made on the implement to accommodate the full stroke of the hydraulic cylinder. Variable-stroke control necessary in the application of hydraulic control to some implements shall be incorporated in the cylinder or hydraulic system and applied on the retracting stroke.

4. **Cylinder Length of Stroke.** 8 in., plus \( \frac{1}{6} \) in., minus 0 in.

5. **Distance, Center to Center between Attaching Pins.** Extended, 28\( \frac{3}{4} \) in., plus \( \frac{7}{8} \) in., minus 0 in.

6. **Size of Attaching Pins.** Cylinder attaching pins shall be of 1 in. nominal diameter. Oversize tolerance, 0.005 in. maximum. Implement mountings shall provide operating clearance for 1.005-in. maximum diameter pins.

7. **Type of Ends.** Yoke on anchor and rod ends.

8. **Width of Throat.** 1\( \frac{3}{4} \) in. minimum and 1\( \frac{1}{8} \) in. maximum for bar \( \frac{7}{8} \)-in. minimum and 1-in. maximum thickness.

9. **Depth of Throat.** The anchor end of the cylinder shall provide the clearance shown in Fig. B-1. This affords clearance for a 1 × 2\( \frac{1}{2} \) in. bar through a 30° included angle, equally divided, and a 1 × 3 in. bar in a perpendicular position.

   The rod end of the cylinder shall provide the clearance shown in Fig. B-2. This affords clearance for a 1 × 2\( \frac{1}{2} \) in. bar through
10. Clearance Area on Implements. Hydraulic cylinders shall operate within the composite volume specified in Fig. B-3. Implements designed for remote-cylinder operation shall provide clearance for a cylinder of the composite volume specified in Fig. B-3.

11. Standard Hose Lengths for Remote Hydraulic Cylinders. The tractor manufacturer shall provide sufficient lengths of hose so that the hydraulic cylinder, provided with tractors having a
capacity up to 6000 lb maximum drawbar pull, is operable when the front anchor pin is located at a maximum 60-in. spherical radius from a center which is the ASAE-SAE Standard Drawbar Hitch Point (Fig. B-4).

![Hose length diagram for wheel-type tractor drawbar. 60-in. spherical radius (ASAE-SAE standard).](image1)

![Hose length diagram for wheel-type tractor drawbar. 84-in. spherical radius (ASAE-SAE standard).](image2)

Tractors with a capacity between 6000 and 11,000 lb maximum drawbar pull shall be provided by the manufacturer with sufficient lengths of hose so that the hydraulic cylinder is operable when the front anchor pin is located at a maximum 84-in. spherical radius from the drawbar hitch point.

On such tractors, for which the ASAE-SAE Standard Power-Take-Off Shaft is available, the ASAE-SAE Standard Drawbar Hitch Point shall be used, as shown in Fig. B-5. On all other track-type tractors, the SAE Standard Track-Type Drawbar Hitch Point shall apply as shown in Fig. B-6.

The implement manufacturer shall locate the hydraulic cylinder on the implement to provide allowance for cushion spring hitches, maneuverability and turning so that the implement can be operated safely without stretching or breaking the hose under any circumstances.

12. **Hose Supports.** Support required for remote cylinder hose shall be considered as part of the implement.
13. **Hose Connections to Cylinders.** Hose connections shall be such that the hose will not interfere with bars extending through the yoke on either end of the hydraulic cylinder.

14. **Operating Time at Rated Engine Speed.** 1½ to 2 sec per 8-in. stroke at rated hydraulic pressure.

**STANDARD DIMENSIONS AND SPECIFICATIONS FOR HYDRAULIC REMOTE CONTROLS INCLUDING A CYLINDER WITH 16-IN. WORKING STROKE**

1. The hydraulic cylinder with hose shall be considered as part of the tractor hydraulic remote control and shall be built to standard dimensions.

2. All hydraulic cylinders shall be double acting and shall operate to raise the implements (or de-angle disk harrows) on their extending stroke.

3. Provision shall be made on the implement to accommodate the full stroke of the hydraulic cylinder. Variable-stroke control
necessary in the application of hydraulic control to some imple-
ments shall be incorporated in the cylinder or hydraulic system
and applied on the retracting stroke.

4. **Cylinder Length of Stroke.** 16 in., plus \( \frac{1}{8} \) in., minus 0 in.

5. **Distance, Center to Center between Attaching Pins.** Extended, 47\( \frac{1}{2} \) in., plus \( \frac{1}{8} \) in., minus 0 in.

6. **Size of Attaching Pins.** Cylinder attaching pins shall be of
1\( \frac{3}{4} \) in. nominal diameter. Oversize tolerance, 0.005 in. maxi-

Fig. B-7. Yoke clearances (anchor end) for 16-in.-stroke hydraulic
cylinder.

7. **Type of Ends.** Yoke on anchor and rod ends.

8. **Width of Throat.** 1\( \frac{3}{4} \) in. minimum and 1\( \frac{1}{8} \) in. maximum
for bar 7\( \frac{3}{8} \)-in. minimum and 1-in. maximum thickness.

9. **Depth of Throat.** The anchor end of the cylinder shall pro-
vide the clearance as shown in Fig. B-7. This affords clearance
for a 1 \( \times \) 2\( \frac{3}{4} \) in. bar through a 30° included angle, equally
divided, and a 1 \( \times \) 3\( \frac{1}{4} \) in. bar in a perpendicular position.

The rod end of the cylinder shall provide the clearance as
shown in Fig. B-8. This affords clearance for a 1 \( \times \) 3 in. bar
through a 70° included angle, equally divided, and a 1 \( \times \) 4 in.
bar in a perpendicular position.

10. **Clearance Area on Implements.** Hydraulic cylinders for
tractors, up to and including those with a capacity of 11,000 lb
maximum drawbar pull, shall operate within the composite vol-
ume specified in Fig. B-9a. Hydraulic cylinders for larger tractors, up to and including those with a capacity of 20,000 lb maximum drawbar pull, shall operate within the composite volume specified in Fig. B-9b. Implements designed for remote-cylinder operation shall provide clearance for a cylinder of the composite volume specified in Fig. B-9a or Fig. B-9b consistent with maximum drawbar pull of the tractor normally required to operate the implement.

Fig. B-8. Yoke clearances (rod end) for 16-in.-stroke hydraulic cylinder.

11. Hose Lengths for Remote Hydraulic Cylinders. The tractor manufacturer shall provide sufficient hose, including a self-sealing coupling in each line, so that the hydraulic cylinder is operable when the front anchor pin is located at a maximum 96-in. spherical radius from the drawbar hitch point. On tractors for which the ASAE-SAE Standard Power-Take-Off Shaft is available, the ASAE-SAE Standard Drawbar Hitch Point shall be used as shown in Fig. B-10. On all other track-type tractors, the SAE Standard Track-Type Drawbar Hitch Point will apply as shown in Fig. B-11.

To provide for the hydraulic control of implements on which the cylinder is positioned outside of the specified 96-in. spherical radius, two lengths of supplemental hose, each including a self-sealing coupling and increasing the spherical radius by 60-in. and 96-in. increments shall be made available by the tractor manufacturer.
Fig. B-9b. Clearance model of agricultural hydraulic cylinder with 16-in. length of stroke for tractors having above 11,000 lb and up to 20,000 lb maximum drawbar pull.
The implement manufacturer shall locate the hydraulic cylinder on the implement to provide allowance for cushion spring hitches, maneuverability, and turning, so that the implement can be operated safely without stretching or breaking the hose under any circumstances.

12. **Hose Supports.** Support required for remote cylinder hose shall be considered as part of the implement.

13. **Hose Connections to Cylinders.** Hose connections shall be such that the hose will not interfere with bars extending through the yoke on either end of the hydraulic cylinder.

14. **Operating Time at Rated Engine Speed.** 3 to 5 sec per 16-in. stroke at rated hydraulic pressure.

**RECOMMENDED PRACTICE: SUPPLEMENTAL HOSE LENGTHS**

(Not a Part of the Standard)

As built at present, the position of the hydraulic cylinder on some large disk implements requires additional hose beyond that necessary for the specified spherical radius from center of draw-
bar hitch point. For implements requiring additional hose length, two supplemental hose lengths increasing the spherical radius by 60-in. and 96-in. increments shall be made available by the tractor manufacturer on special orders. When supplemental hose lengths are used, self-sealing couplings shall be provided for each of the hose lines leading from the tractor and for the supplemental hose lengths.
APPENDIX C

**DRAFT AND POWER REQUIREMENTS OF CROP MACHINES** *

<table>
<thead>
<tr>
<th>Machine</th>
<th>Normal Range</th>
<th>References</th>
</tr>
</thead>
<tbody>
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<td><strong>Tillage:</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Moldboard plow</td>
<td>5-12 psi of furrow section</td>
<td>2, 10, 13</td>
</tr>
<tr>
<td>Lister</td>
<td>400-750 lb per row</td>
<td>7, 8, 12, 13</td>
</tr>
<tr>
<td>Vertical-disk plow</td>
<td>150-350 lb per ft width</td>
<td>7, 8</td>
</tr>
<tr>
<td>Single-acting disk harrow</td>
<td>40-130 lb per ft width</td>
<td>2, 7, 10, 11, 12, 13</td>
</tr>
<tr>
<td>Tandem disk harrow</td>
<td>80-160 lb per ft width</td>
<td>2, 7, 10, 12, 13</td>
</tr>
<tr>
<td>Tandem disk harrow, 22-in.</td>
<td>170-225 lb per ft width, or 60 per</td>
<td>1, 2</td>
</tr>
<tr>
<td>diameter, 9-in. spacing</td>
<td>cent of weight</td>
<td></td>
</tr>
<tr>
<td>Spike-tooth harrow</td>
<td>30-60 lb per ft width</td>
<td>7, 10, 11, 12, 13</td>
</tr>
<tr>
<td>Spring-tooth harrow</td>
<td>75-150 lb per ft width</td>
<td>1, 7, 8, 10</td>
</tr>
<tr>
<td>Duckfoot field cultivator</td>
<td>90-160 lb per ft width</td>
<td>7, 8, 10, 12, 13</td>
</tr>
<tr>
<td>Roller</td>
<td>30-80 lb per ft width</td>
<td>10, 11</td>
</tr>
<tr>
<td>In certain southern and far western soils, draft figures ranging up to a maximum of approximately double the above have been recorded</td>
<td>9</td>
<td></td>
</tr>
<tr>
<td><strong>Planting:</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Grain drill</td>
<td>30-80 lb per ft width</td>
<td>7</td>
</tr>
<tr>
<td>Corn planter</td>
<td>80-129 lb per row</td>
<td>12, 13</td>
</tr>
<tr>
<td><strong>Cultivating:</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Rotary hoe</td>
<td>30-60 lb per ft width</td>
<td>11, 12, 13</td>
</tr>
<tr>
<td>Corn cultivator</td>
<td>22-65 lb per shovel</td>
<td>1</td>
</tr>
<tr>
<td>Spring-tine weeder</td>
<td>25-35 lb per ft width</td>
<td>12, 13</td>
</tr>
<tr>
<td>Rod weeder</td>
<td>80-110 lb per ft width</td>
<td>7, 8</td>
</tr>
<tr>
<td><strong>Harvesting:</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Mower</td>
<td>60-100 lb per ft width</td>
<td>3</td>
</tr>
<tr>
<td>Grain binder</td>
<td>65-150 lb per ft width</td>
<td>3</td>
</tr>
<tr>
<td>Thresher</td>
<td>0.8-1.2 hp per in. cylinder width</td>
<td>14</td>
</tr>
<tr>
<td>Combine, 5 and 6 ft</td>
<td>2 to 4½ (pto) hp per ft of cutter bar</td>
<td>4, 6</td>
</tr>
<tr>
<td>Combine, 8 to 12 ft</td>
<td>Engine with 2 to 3 net hp per ft of</td>
<td>Common practice</td>
</tr>
<tr>
<td></td>
<td>cutter bar</td>
<td></td>
</tr>
<tr>
<td>Corn picker, 2-row</td>
<td>2 to 5 (pto) hp per row</td>
<td>12</td>
</tr>
<tr>
<td>Stationary silage cutter</td>
<td>0.761-1.60 hp-hr per ton</td>
<td>5</td>
</tr>
<tr>
<td>Husker-shredder</td>
<td>0.25-0.35 hp-hr per bu</td>
<td>3</td>
</tr>
</tbody>
</table>

**REFERENCES**


*Crop Machines Use. *ASAE Data (revised, 1953). *


APPENDIX D

 SPECIFICATIONS FOR MARKING PLOWSHARES
 AND OTHER SOIL-WORKING SHAPES
 (ASAE STANDARD, ADOPTED JUNE, 1950)

These specifications are recommended to satisfy the universal need for identification of materials used in plowshares and other soil-working shapes. It is understood that they will be followed by manufacturers whenever present stamping and pattern equipment is replaced or modified.

1. Recommended Markings for Plowshares.

<table>
<thead>
<tr>
<th>Material</th>
<th>Description</th>
<th>Marking</th>
</tr>
</thead>
<tbody>
<tr>
<td>Soft-center steel</td>
<td>Soft-center steel is three-ply steel; the center layer is low-carbon and the outer layers are high-carbon steel</td>
<td>SC or SOFT CENTER</td>
</tr>
<tr>
<td>Carburized steel</td>
<td>Carburized steel is low-carbon steel to which surface carbon has been added in varying amounts and depths</td>
<td>CARB</td>
</tr>
<tr>
<td>Solid steel</td>
<td>Solid steel is approximately SAE 1080 steel</td>
<td>SOLID</td>
</tr>
<tr>
<td>Cast iron (a)</td>
<td>With chilled edge and point</td>
<td>CHILLED</td>
</tr>
<tr>
<td>Cast iron (b)</td>
<td>Without chilled edge and point</td>
<td>CAST</td>
</tr>
</tbody>
</table>

The use of alloy steels and other special materials in the manufacture of plowshares has not as yet progressed to the point where it is believed they warrant inclusion in these recommendations.

2. Size and Location of Markings. The size and location of markings is left to the discretion of the individual manufacturer. The location of markings should be such, however, that they will not wear off in field use.

3. Recommended Markings for Lister and Middlebuster Shares. Lister and middlebuster shares should be identified in accord with the above recommendations for plowshares.

550
4. Recommended Markings for Cultivator Sweeps, Shovels, and Other Miscellaneous Soil-Working Shapes. As solid steel is the material most commonly used in the manufacture of these shapes, no material identification is necessary. If, however, material other than solid steel is used, the shapes should be identified in accord with the above recommendations for plowshares.
APPENDIX E

Baling Wire for Automatic Balers
(ASAE Standard, Adopted May, 1952)

SCOPE AND DESIGNATION

(a) This specification shall cover annealed baling wire for automatic balers.

(b) The wire shall be furnished in two sizes of coils: 3150 ft (approximately) and 6500 ft (approximately).

STANDARD PRACTICE

Unless otherwise specified, standard practices as covered in the latest revision of the AISI (American Iron and Steel Institute) Steel Products Manual, Section 16, on carbon steel wire shall govern on all points covered by this specification.

REQUIREMENTS

Physical Properties. 50,000 to 70,000 psi tensile strength; 12 per cent minimum permanent elongation in 10-in. length.

Rewinding Practices. Wire shall be wound with a uniform tension and shall be furnished with an oil-base protective coating that will not harden or produce a gummy condition in the baler tying mechanism.

Wire shall be free from injurious heavy scale and surface imperfections.

Wire shall be free of kinks and shall be of continuous length; therefore, it shall contain no twisted splices. Welded joints shall be dressed down to wire diameter.

Wire shall uncoil from inside or outside of coil.

Outside end of wire shall bear a tag containing only the manufacturer’s identification; inside end of coil shall be bent over the tie wire at the inside diameter.

552
Dimensions. Wire shall be furnished in rewound coils, conforming to the following requirements:

<table>
<thead>
<tr>
<th></th>
<th>3150-Ft Coil</th>
<th>6500-Ft Coil</th>
</tr>
</thead>
<tbody>
<tr>
<td>Wire gage number</td>
<td>14½</td>
<td>14½</td>
</tr>
<tr>
<td>Wire diameter</td>
<td>0.076 in. ± 0.002 in.</td>
<td>0.076 in. ± 0.002 in.</td>
</tr>
<tr>
<td>Inside diameter</td>
<td>3 in. ± ¼ in.</td>
<td>8½ in. ± ¼ in.</td>
</tr>
<tr>
<td>Outside diameter</td>
<td>97½ in. ± 0, + ¼ in.</td>
<td>13½ in. ± ¼ in.</td>
</tr>
<tr>
<td>Coil width</td>
<td>3⅞ in. ± 0, + ¼ in.</td>
<td>6 in. ± ¼, - 0 in.</td>
</tr>
<tr>
<td>Length</td>
<td>3150 ft ± 1%</td>
<td>6500 ft ± 1%</td>
</tr>
</tbody>
</table>

PACKAGING

Coils shall be banded securely with six ties on the 6500-ft coil and four ties on the 3150-ft coil, evenly spaced. (Note: If wire is used as a tie, a minimum of 15 gage shall be used.)

Coils of wire shall be shipped in suitable containers providing adequate protection for shipping, storage, and distribution.

The 3150-ft coil shall be shipped two in a carton which shall be marked as follows:

Package No. 3150
ASAE Standard Baling Wire (2 coils), 14½ gage
Approximately 3150 ft per coil

The container shall also carry, in a different location, the name and/or brand name of the wire manufacturer and necessary instructions for storing and handling.

The 6500-ft coil shall be shipped one in a carton which shall be marked as follows:

Package No. 6500
ASAE Standard Baling Wire (1 coil), 14½ gage
Approximately 6500 ft

The container shall also carry, in a different location, the name and/or brand name of the wire manufacturer and necessary instructions for storing and handling.

INSPECTION AND REJECTION

Any coil not conforming to the foregoing specifications may be rejected and the manufacturer or distributor notified.
An interim coil conforming to the following specifications shall be made available for use with balers in the field at the time of adoption of this standard, as long as there is sufficient demand to justify production.

<table>
<thead>
<tr>
<th>Specification</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Wire gage number</td>
<td>14½</td>
</tr>
<tr>
<td>Wire diameter</td>
<td>0.076 in. ± 0.002 in.</td>
</tr>
<tr>
<td>Inside diameter</td>
<td>12 in. + 0, − 1 in.</td>
</tr>
<tr>
<td>Outside diameter</td>
<td>18 in. ± ¾ in.</td>
</tr>
<tr>
<td>Coil width</td>
<td>3½ in. + 0, − ½ in.</td>
</tr>
<tr>
<td>Length</td>
<td>Approximately 6500 ft ± 1%</td>
</tr>
</tbody>
</table>

This coil shall be shipped in a carton which shall be marked as follows:

Baling Wire, 18-in. diameter coil, 14½ gage
Approximately 6500 ft
For balers requiring 18-in. diameter coil

Container shall also carry, in a different location, the name and/or brand name of the wire manufacturer and necessary instructions for storing and handling.

**EFFECTIVE DATE**

The ASAE standard baling-wire coil shall be available on or before October 1, 1952.
## APPENDIX F

### LOAD AND INFLATION DATA FOR AGRICULTURAL IMPLEMENT TIRES

**(Tire and Rim Association Standard)**

Maximum speed, 20 mph

Table AI-1A and WI-1

<table>
<thead>
<tr>
<th>Tire Size</th>
<th>20 psi</th>
<th>24 psi</th>
<th>28 psi</th>
<th>32 psi</th>
<th>36 psi</th>
<th>40 psi</th>
<th>44 psi</th>
<th>48 psi</th>
</tr>
</thead>
<tbody>
<tr>
<td>3.00-7</td>
<td>175</td>
<td>195</td>
<td>210</td>
<td>230</td>
<td>245</td>
<td>260</td>
<td>275</td>
<td>290</td>
</tr>
<tr>
<td>4.00-9</td>
<td>305</td>
<td>310</td>
<td>370</td>
<td>400</td>
<td>430</td>
<td>455(4)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>4.00-12</td>
<td>450(2)</td>
<td>500</td>
<td>550</td>
<td>625</td>
<td>675(4)</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>4.00-15</td>
<td>530</td>
<td>590</td>
<td>655</td>
<td>730</td>
<td>770</td>
<td>800(3)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>4.00-18</td>
<td>585(2)</td>
<td>650</td>
<td>710</td>
<td>770</td>
<td>825</td>
<td>875(4)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>4.00-24</td>
<td>700</td>
<td>775</td>
<td>850</td>
<td>920</td>
<td>985</td>
<td>1015(4)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>5.00-15</td>
<td>600</td>
<td>765</td>
<td>810</td>
<td>910(4)</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>5.00-16</td>
<td>730</td>
<td>810</td>
<td>885</td>
<td>960(4)</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>5.50-16</td>
<td>875</td>
<td>970</td>
<td>1060(4)</td>
<td>1150</td>
<td>1230</td>
<td>1310</td>
<td>1390(0)</td>
<td></td>
</tr>
<tr>
<td>6.00-16</td>
<td>1020</td>
<td>1130(4)</td>
<td>1240</td>
<td>1340</td>
<td>1430</td>
<td>1520(6)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>6.00-20</td>
<td>1090</td>
<td>1210</td>
<td>1330</td>
<td>1440</td>
<td>1540</td>
<td>1640(6)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>6.00-24</td>
<td>1160</td>
<td>1290(4)</td>
<td>1410</td>
<td>1530</td>
<td>1640(6)</td>
<td>1740</td>
<td>1840</td>
<td>1950(8)</td>
</tr>
<tr>
<td>6.50-24</td>
<td>1470</td>
<td>1630(4)</td>
<td>1730</td>
<td>1840</td>
<td>2000(8)</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>6.75-15</td>
<td>1665</td>
<td>1835(4)</td>
<td>1935</td>
<td>2060</td>
<td>2200(4)</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>7.50-10</td>
<td>1500(4)</td>
<td>1670</td>
<td>1820</td>
<td>1970(5)</td>
<td>2110</td>
<td>2250</td>
<td>2390(8)</td>
<td></td>
</tr>
<tr>
<td>7.50-12</td>
<td>1550(4)</td>
<td>1720</td>
<td>1870</td>
<td>2040(6)</td>
<td>2190</td>
<td>2330</td>
<td>2470</td>
<td>2620</td>
</tr>
<tr>
<td>7.50-24</td>
<td>1740</td>
<td>1940</td>
<td>2120</td>
<td>2300(6)</td>
<td>2460</td>
<td>2620</td>
<td>2780</td>
<td>2940</td>
</tr>
<tr>
<td>7.50-26</td>
<td>1930</td>
<td>2150</td>
<td>2350</td>
<td>2540</td>
<td>2720</td>
<td>2890</td>
<td>3050(8)</td>
<td></td>
</tr>
<tr>
<td>7.50-30</td>
<td>1255</td>
<td>1395(4)</td>
<td>1530</td>
<td>1650</td>
<td>1785(5)</td>
<td>1915</td>
<td>1955</td>
<td>2035(8)</td>
</tr>
<tr>
<td>7.50-44</td>
<td>1850</td>
<td>2100</td>
<td>2350</td>
<td>2640</td>
<td>2840</td>
<td>3070</td>
<td>3330(8)</td>
<td>3550(10)</td>
</tr>
<tr>
<td>9.00-44</td>
<td>2740</td>
<td>3060(6)</td>
<td>3350</td>
<td>3650</td>
<td>3920</td>
<td>4170</td>
<td>4430</td>
<td>4700</td>
</tr>
<tr>
<td>11.25-24</td>
<td>3260</td>
<td>3650(8)</td>
<td>3950</td>
<td>4270</td>
<td>4520(10)</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>11.25-26</td>
<td>3370</td>
<td>3740(8)</td>
<td>4120</td>
<td>4450</td>
<td>4770(10)</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>12.75-32</td>
<td>4060</td>
<td>5000(8)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

The parentheses denote that the load is the maximum recommended for the ply rating indicated. The ply rating is an index of the strength and does not necessarily represent the number of cord plies in the tire.

* Ratings for tires with smooth tread. All other ratings are for tires with either ribbed or traction tread.

† Maximum recommended load for 4-ply tire is 305 lb at 52 psi inflation pressure.
The ASAE standard, "Power Take-Off for Farm Tractors," specifies the essential dimensions of tractor components necessary to enable manufacturers of driven machines to provide for the satisfactory hitching of their machines to any make of tractor. The successful performance of all tractor and driven-machine combinations likely to be met in field service requires consideration of many factors other than the dimensional relationships established in the aforementioned standard. Some of the more important of these factors are as follows:

1. **Instructions for the Operator.** The tractor manufacturer should provide safety instructions in a prominent place on the tractor. The driven-machine manufacturer should also provide safety instructions in a prominent place on any pto-driven machine, especially warning against operation without both tractor and driven-machine pto shielding in place. Operator's manuals for both tractor and driven machine should stress the importance of keeping these safety shields in place.

The correct position of the hitch point when operating a pto-driven machine is the position defined in the ASAE standard, "Power Take-Off for Farm Tractors." Operator's manuals for both tractor and driven machines should indicate the proper hitch-point adjustment for pto operation; and, where practical, instructions describing proper hitch adjustments should be displayed on pto-driven machines.

2. **Driven-Machine Hitch and Power-Line Design Requirements.** The hitch and power line of any pto-driven machine, when the machine is hitched to any tractor that conforms to the ASAE standard, "Power Take-Off for Farm Tractors," should provide satisfactory operation over any terrain the machine is likely to encounter. To meet any such operating conditions, provision should be made in the power line and hitch of
the driven machine to prevent any of the following from occurring:

(a) The universal joints in the power line from reaching a locking angle.
(b) The telescoping section of the power line from separating beyond the point where there is sufficient bearing to provide for proper operation.
(c) The telescoping section of the power line from shortening to a solid position.

In normal forward operation the universal joints in the power line of the driven machine should be in straight alignment as nearly as possible and should be properly indexed with respect to each other so as to maintain the torsional-load fluctuations at the lowest possible value. Extreme care should be taken to determine load fluctuations or load reversals when one or three universal joints are employed in a power line.

3. Maximum Bending Load Limitations for Power-Take-Off-Shaft Drives Employing V-Belts or Chains. The pto drive of tractors is designed primarily to transmit torsional loads. When V-belt or chain drives with the driving sheave or sprocket mounted directly on the pto shaft are used, bending loads on the shaft should be checked carefully. The total bending load imposed on the tractor pto shaft by drives of this type should not be in excess of values shown in the following table:

<table>
<thead>
<tr>
<th>Position of Load Application</th>
<th>1(\frac{3}{8})-in. Diam pto</th>
<th>1(\frac{1}{4})-in. Diam pto</th>
</tr>
</thead>
<tbody>
<tr>
<td>At the end of pto shaft</td>
<td>500 lb</td>
<td>800 lb</td>
</tr>
<tr>
<td>Between the pto shaft rear bearing and/or at the groove in the outside diameter of the pto shaft splines</td>
<td>600 lb</td>
<td>1000 lb</td>
</tr>
</tbody>
</table>

The tractor pto shaft and bearing mountings should successfully withstand the magnitude of bending loads indicated in this table.

4. Power-Line Protective Couplings and Maximum Torsional Load Limitations for Power-Take-Off Shafts. The dynamic torsional loads on pto drives should be checked carefully. Because of the large amount of kinetic energy available at the pto shaft, instantaneous torsional loads and fluctuating
operating loads far in excess of the average rated horsepower of the tractor may be transmitted. Transmittal of these excessive loads can result in premature failure of the driving parts.

Implements subject to high starting loads or plugging should be equipped with an overload protective device in the power line which will protect the drive against torsional overloads of sufficient magnitude to cause mechanical failure of either tractor or implement parts.

In consideration of the foregoing factors it is desirable for implements to conform to the following conditions:

(a) The instantaneous operating loads should not exceed 7500 lb-in. for the 1%-in. diameter shaft or 12,000 lb-in. for the 1%-in. diameter shaft under conditions where there is no reversal of load. When a repetitive reversal of load is encountered, the aforementioned load limitations must be reduced by the amount of the reverse load.

(b) When the frequency of the instantaneous load does not exceed 10 cycles per hour, implements imposing loads greater than 7500 lb-in. for the 1%-in. diameter shaft or 12,000 lb-in. for the 1%-in. diameter shaft should have a power-line protective device which does not exceed a maximum instantaneous slip value of 13,000 lb-in. for the 1%-in. diameter shaft or 26,000 lb-in. for the 1%-in. diameter shaft.

(c) The requirements in paragraphs (a) and (b) above will generally be met with a smooth-surface, frictional type of power-line protective device which does not exceed a breakaway value of 4000 lb-in. for the 1%-in. diameter shaft or 6400 lb-in. for the 1%-in. diameter shaft when checked under static conditions. Snap-type, spring-loaded jaw clutches should not be evaluated under static conditions, and therefore must comply with paragraphs (a) and (b) above.

Tractors capable of imposing inertia loads on the power line as high as those mentioned above should have a pto drive capable of transmitting a torque equivalent to that described in paragraphs (a) and (b) without failure. Tractors which are not capable of transmitting a torque of this magnitude should have a drive of sufficient strength to transmit the maximum torque they are capable of delivering to the pto drive.
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