AN INTERATED HEAT PUMP THIN FILM SCRAPED SURFACE HEAT EXCHANGER SYSTEM FOR CONCENTRATION OF MILK

THESIS SUBMITTED TO THE
NATIONAL DAIRY RESEARCH INSTITUTE, KARNAL
IN PARTIAL FULFILMENT OF THE REQUIREMENT
FOR THE DEGREE OF

DOCTOR OF PHILOSOPHY
IN
DAIRY ENGINEERING

BY:
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KARNAL -132001(HARYANA), INDIA

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CERTIFICATE

This is to certify that the thesis entitled **"AN INTEGRATED HEAT PUMP THIN FILM SCRAPED SURFACE HEAT EXCHANGER SYSTEM FOR CONCENTRATION OF MILK"** submitted by Mr. **RAJ KUMAR KOHLI** in partial fulfilment of the requirement for the award of the degree of **DOCTOR OF PHILOSOPHY in DAIRY ENGINEERING** of the National Dairy Research Institute (Deemed University), Karnal (Haryana), India, is a bonafide research work carried out by him under my supervision and guidance and no part of the thesis has been submitted for any other degree or diploma.

(H. ABICHANDANI)
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(R.K. KOHLI)

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LIST OF FIGURES

NOMENCLATURE

A	:	Heat transfer area, m^2
B	:	Number of blades
COP	:	Coefficient of performance
C_p	:	Specific heat, kJ/kg K
D	:	Inside diameter of heat exchanger, m
g	:	Acceleration due to gravity, m/s^2
g_c	:	Newton's law conversion factor
h_o	:	Refrigerant side film coefficient, W/m^2 K
h_s	:	Scraped film heat transfer coefficient, W/m^2 K
K	:	Thermal conductivity of fluid, W/m K
L	:	Length of heat exchanger, m
$\dot{M}$	:	Mass flow rate, kg/s
$\dot{M}_v$	:	Evaporation rate, kg/s
N	:	Rotor speed, RPS
P_a	:	Absolute pressure in the system, mm of Hg
PER	:	Primary energy ratio
Pd	:	Discharge pressure of refrigerant, kg/cm^2
P_s	:	Suction pressure of refrigerant, kg/cm^2
P_S	:	Power input to rotor, W
P_T	:	Total power input to system, KW
S_i	:	Per cent total solids in feed
S_o	:	Per cent total solids in product at outlet
S	:	Per cent average total solids, $(S_i + S_o)/2$
T_{DR}	:	Refrigerant discharge temperature, ^oC

contd.....

To_C : Fluid temperature at outlet of condenser/SSHE, ^oC

To_E : Fluid temperature at outlet of evaporator/milk cooler, ^oC

U_o : Overall heat transfer coefficient based on outside area of heat exchanger, W/m^2 K

V : Velocity of fluid, m/s

V_c : Circumferential velocity of rotor blades, m/s

W_B : Weight of one blade, N

Greek Letters

μ : Coefficient of viscosity, Pa-s.

ρ : Density, kg/m^3

σ : Surface tension, N/m

η : Efficiency

Subscripts

a : Absolute

^oC : Outlet of condenser

d : Discharge

^oE : Outlet of evaporator

R : Refrigerant

m : Material

BM : Buffalo whole milk

SM : Skim milk

W : Water

O : Outside

L : Low

H : High

<u>Dimensionless Groups</u>

B_F : Blade factor, $\dfrac{B\ W_B\ g_c}{\mu\ N\ D^2}$

N_u : Nusselt numbner, $\dfrac{h_s\ D}{K}$

Pr : Prandtl number, $\dfrac{\mu\ c_p}{K}$

Re_f : Film Reynolds Number, $\dfrac{\dot{M}}{\pi\ D\ \mu}$

Re_R : Rotational Reynolds Number, $\dfrac{D^2\ N\ \rho}{\mu}$

We : Weber Number, $\dfrac{\rho N^2 D^3}{\sigma g_c}$

0. SUMMARY

0.1 The dairy and food industry is ranked among first three industries from the angle of energy consumption. Considering the present day energy crunch, every effort is made to use it in the most prudential manner.

0.2 The potential of heat pumps in dairy and food industry has since been realized with a view point of energy conservation. Process engineers frequently come across situations of heat transfer to viscous and heat sensitive products which are normally treated using a special type of heat exchanger, known as thin film scraped surface heat exchanger.

0.3 Concern about the increased energy costs and quality of the end product promoted this study as one of the technique.

0.4 In the present study, a concept of integrated heat pump - thin film scraped surface heat exchanger system was evolved for concentration of milk. The literature survey revealed many inadequacies regarding optimum engineering design of such a system.

0.5 The present system consisted of a liquid concentrator, liquid cooler, vapour condenser, vapour separator, condensate collecting device, refrigerant compressor, vacuum rotary seal

with vacuum pump, along with other pumps, measuring instruments and piping etc. The rotor was povided with variable clearance staggered blades. The number of blades could be changed from two to eight.

0.6 Two hundred thirty six (236) trials were conducted for evaporation of water, skim milk, and buffalo whole milk under sub-atmospheric conditions. The data generated were analysed in HCL-4 computer. Nusselt type correlation was developed for scraped film heat transfer coefficients. Box Wilson model (polynomial equation) was applied to optimize the process variables for overall heat transfer coefficient and power consumption. On the basis of observations, a concept was evolved to explain the usefulness of staggering the blades in enhancing heat transfer and reducing power consumption. Staggering of blades caused more turbulence due to more frequent interception of fluid film and their overlaping at the centre of the rotor shaft and also because of abrupt change in axial velocity of fluid film when it covered half the effective length of heat exchanger.

0.7 The thermal performance of heat pump was evaluated and the coefficient of performance was found to be 3.5 under optimum operating conditions (number of blades = 6, rotor speed = 3.42 RPS, mass flow rate of buffalo whole milk = 16.67×10^{-3} kg/s.

0.8 In the present system, two processes, viz., evaporation of milk and cooling of concentrated milk are integrated. This concept is very attractive from the point of view of energy saving and other advantages. The integrated heat pump - thin film scraped surface heat exchanger system appears to have an excellent potential for processing of milk and similar food products due to its inbuilt characteristic of maintaining overall heat transfer coefficient. Also it offers a scope of recovery of flavours and other volatile components.

0.9 The result of series of experiments should provide a theoretical basis for design of integrated heat pump - thin film SSHE system for concentrating milk and other similar food products.

1. INTRODUCTION

1.1 In the wake of energy crunch, the industrial sector is increasingly becoming conscious of energy conservation and recovery. Many dairy and food processes are highly energy intensive and, therefore, the use of energy conservation technology in these industries is gaining wide acceptance.

1.2 One device which can make a significant contribution to conserve energy in food industry is heat pump. It pumps heat energy from a source at low temperature and elevates it to high temperature. In the early 19th century, the idea of heat pump was evolved from the Carnot cycle. William Thomson (later known Lord Kelvin), was the first to propose a practical heat pump system. The heat pumps have been in commercial use in United States of America and United Kingdom since 1927.

1.3 A heat pump is a thermal amplifier which raises the temperature of low grade heat energy to a more useful higher level employing a relatively small amount of high grade energy. It is primarily a refrigerating machine and works as a heat extraction device at the evaporator and simultaneously as a heat supplying device at the condenser. Thus a heat pump can be used advantageously in a process which simultaneously requires heating and cooling. Fig. 1.1 shows layout of a mechanical vapour

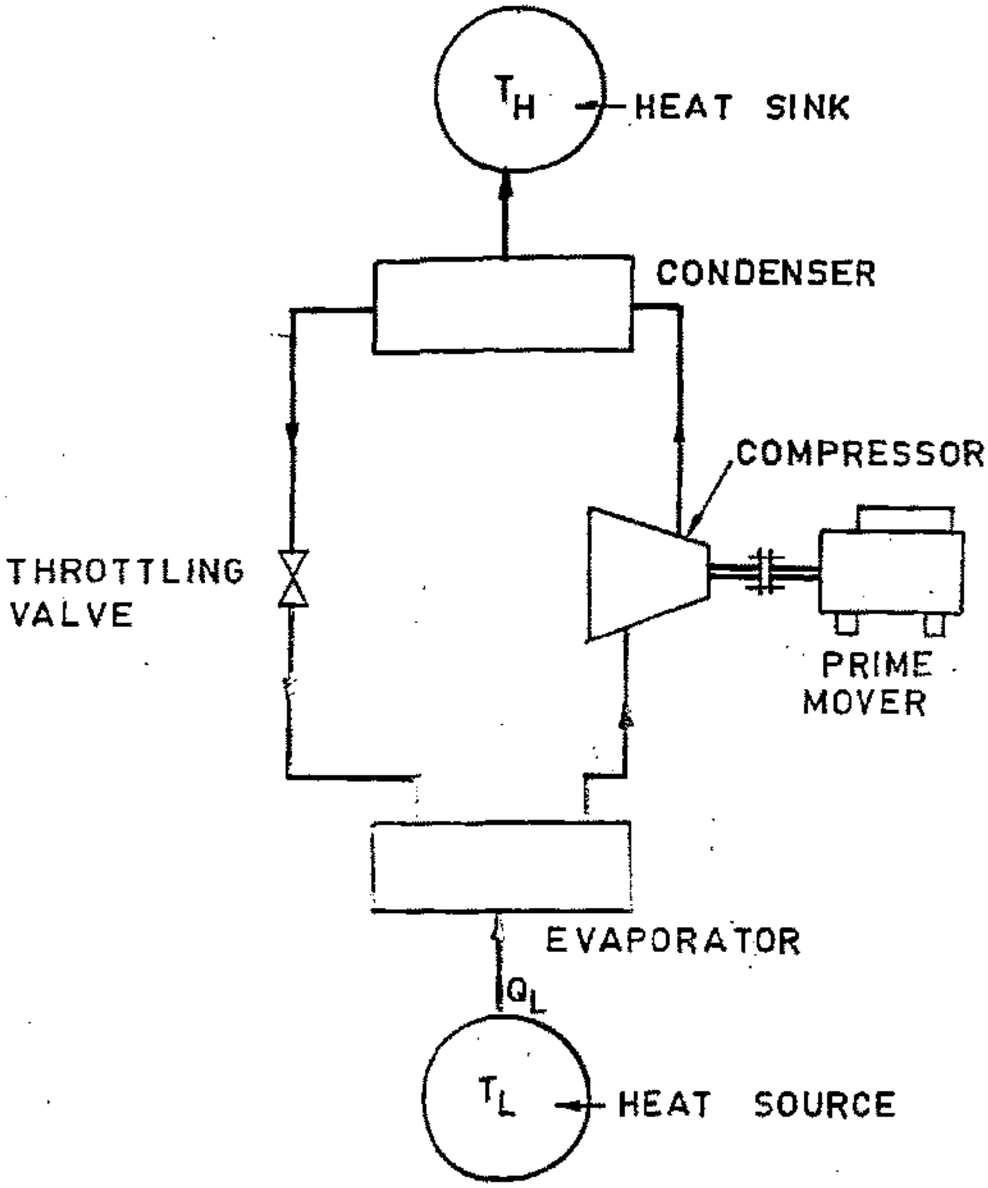


Fig. 1.1. LAYOUT OF MECHANICAL VAPOUR COMPRESSION CYCLE HEAT PUMP.

compression cycle heat pump. The sources of low grade energy are water, air, solar and waste heat from different processes.

1.4 Despite the availability of different thermodynamic cycles such as mechanical vapour compression cycle (MVCC), the absorption cycle, three fluid vapour absorption cycle and the Stirling cycle, etc., the most common types of heat pump cycle is MVCC. The range of working fluids is very wide. Any fluid which can be made to evaporate between 1 bar and 20 bar at useful temperatures is useful.

1.5 The effectiveness of a machine is usually expressed in terms of efficiency. The effectiveness of a refrigeration cycle is expressed as coefficient of performance (COP). It is defined as the ratio of desired effect to work required to produce that effect, the effect and work both being expressed in thermal units. Since the purpose here is to utilize heat at condenser, therefore, COP is the ratio of heat energy at condenser to work input. The COP is considerably greater than unity, often ranging from three to six. It means that energy output is three to six times the input.

1.6 The compressor is a critical component of the mechanical vapour compression heat pump. Numerous types of compressors suitable to vapour compression cycle are rotary vane, reciprocating, screw and centrifugal compressors. Selection of compressor depends upon the size, capacity of system and the working fluid.

2. REVIEW OF LITERATURE

The review of literature presents recent trends pertaining to development and application of heat pumps. Also the heat transfer and hydrodynamic aspects of mechanically formed thin film heat exchangers are delineated as under:

2.1 HEAT PUMPS

2.1.1 Advent of heat pumps

2.1.2 Heat pump cycles

2.1.3 Heat sources

2.1.4 Heat pump circuits

2.1.5 Working fluids

2.1.6 Compressors

2.1.7 Primemovers

2.2 HEAT TRANSFER AND HYDRODYNAMICS IN THIN FILM SSHE

2.2.1 Mechanisms of heat transfer with phase change

2.2.2 Power requirement

2.3 ENERGY ANALYSIS

2.4 APPLICATIONS OF HEAT PUMP IN DAIRY AND FOOD INDUSTRIES

2.1 HEAT PUMPS

2.1.1 ADVENT OF HEAT PUMPS

Lord Kelvin was the first to propose a practical heat pump. It was claimed that the heat pump was able to produce heat using only three per cent of the energy which would be needed for direct heating. Haldane (1930) reported on the installation and tests on a domestic heat pump in Scotland. This unit provided hot water and space heating, using air as the heat source. The first commercial heat pump was commissioned in Europe during 1938-39. This heat pump used river water as the heat source and a rotary compressor with R-12 as working fluid (Engle, 1978). The first heat pump was installed in the United Kingdom for heating a building (Sumner, 1953). The second heat pump installation was at Royal Festival Hall in the U.K. (Montagen and Ruckley, 1954). Reay and Macmichael (1979) summerised the heat pump installations all over the Switzerland and USA. The rapid growth in use of heat pump in USA started from 1971 onwards.

2.1.2 HEAT PUMP CYCLES

2.1.2.1 The mechanical vapour compression is commonly employed in heat pumps. In 1824, Carnot was the first to use a thermodynamic cycle to describe a process. The cycle conceived by him remained as the fundamental measure against which heat pump performance is judged because it consists of reversible processes. The term 'Rankine cycle' is sometimes used to explain both a power and mechanical vapour compression cycles. In case a Rankine cycle heat engine is used to drive heat pump, one can be termed as a Rankine/Rankine cycle. This combination is particularly intriguing when a common working fluid is used for the two cycles for economy.

2.1.2.2 The absorption refrigeration system (ARS) is a heat operated unit whereas mechanical vapour compression cycle (MVCC) uses the energy in the form of mechanical work. The MVCC requires a compressor where as ARS requires an absorber, generator and a pump. The MVCC can be employed on a wide range of capacity as compared to ARS.

2.1.2.3 Apart from these cycles, there are few other cycles for example, Stirling cycle and Brayton cycle. The scope and application of these cycles is also unlimited.

2.1.3 HEAT SOURCES

Heat pumps can make use of effective free heat sources and their derived sources.

2.1.3.1 Setty (1979) reported a brief analysis on utilization temperature range for heat pumping. Utilization temperature range is the optimum temperature range at which heat can be absorbed or rejected for providing heating or cooling simultaneously. The source in temperature range of 322° to 422°K is not economical for heat pumping even if it is available free. The utilization temperature range of 283° to 322°K is most suitable for heat pumping.

2.1.3.1 <u>Air</u>

Most commercially available heat pumps use air as the heat source. Air source heat pumps working on the MVCC also use air as the heat sink. The variation in temperature and the possibility of formation of frost must be considered in designing air source heat pump. The fall in out-door temperature is directly proportional to capacity of heat pump. However, the out-door air is available in abundance and at low cost, but its major draw back is that, it is normally coldest when maximum heat has to be extracted from it and hottest when maximum heat has to be rejected to it.

2.1.3.1.1 Heap (1975) reported that the COP reduces for a typical air source heat pump, as a function of the outside ambient air temperature. The COP ranged from 1.8 to 2.8 with ambient temperature of 268°K to 293°K. MacDonald (1985) reported for air-to-air heat pump having average COP of 2.51. MacDonald (1986) also reported for a dual heat pump system that average COP was 1.2, ranging on a daily basis from 0.75 to 1.8. He also proposed modification which included fin spacing, orientation of air flow over the barn evaporator coil and an improved heat storage system.

2.1.3.1.2 Sasaki (1989) analysed the heating efficiency and coefficient of performance of an air type heat pump with a 7.5 KW compressor used in a green house having 352 m^2 in area. The air temperature in the green house was maintained at 13°-15°C. The maximum difference between the temperature inside the green house and the ambient air temperature was 17°C. When the ambient air temperature decreased markedly, the heat pump was operated frequently to remove the frost covering the outer part of the instrument. When the defrosting operations were carried out too frequently, the air temperature in the green house could not be maintained at 15°C. When the set point of the night temperature was reduced from 16 to 13 to 10°C the air temperature in the green house could easily be maintained at the set point. Therefore, it was considered that the modification of the set point of the night temperature was the most suitable method to control the temperature by the air to air type heat pump. The values of the COP of the heat pump ranged from 2.4 to 3.2 and there was a high correlation between the values of the ambient air temperature and the COP. Electricity charges were almost 40 per cent of the heavy oil charges.

2.1.3.2 Water

2.1.3.2.1 Among the various water sources, city mains would be an ideal as a heat source for a heat pump as it is continuously available in a fixed range of temperatures. The use of water as a heat source was tried by Sumner (1976). Slightly warm water is more attractive. Ground water also used as source of heat. Fordsmand (1978) reported that the normal heat removal rate for an evaporator coil located in the ground is between 20 to 25 w/m. Values as low as 10 w/m were obtained by Schar (1978), the maximum being 50-60 w/m, as reported by Von Coube (1978) in Germany. Biest *et al.* (1985) reported COP of 3.5 to 4 for water - water heat pump using cooled underground water.

2.1.3.2.2 Groote *et al.* (1985) used three water - water heat pumps and achieved COP of system ranged from 3.4 - 4.4.

2.1.3.3 Solar

2.1.3.3.1 Solar energy may be harnessed by heat pump employing solar collectors. Bridgers (1967) described heat pump with solar collectors and obtained a temperature of 260.8°K which was less than the designed temperature. McVeigh (1977) gave a comprehensive treatment of solar collectors design and applications. The use of solar collectors alongwith ground water was studied by Fordsmand (1978). The conclusion was that it was impractical to install a solar collector if the size was less than $3m^2$/KW loss of heat from the dwelling. For a collector area of $30m^2$ and a ground coil area of only 100 m^2, a COP of 3.4 was achieved.

2.1.3.3.2 Jardine (1976) reported that storage temperature had significant influence on the heat pump coefficient of performance. If variation in storage temperature was between 5 to 45°C, the influence on COP was found to be in the range of 2 to 4. The relationship found was linear in nature.

2.1.3.3.3 Hanby (1977) indicated that air solar collectors of some considerable size are needed to be technically effective. He also reported that the influence of heat storage, both on overall COP and on system cost, is also a factor of prime importance.

2.1.3.3.4 Corman *et al.* (1974) studied a system using heat storage, with the heat pump evaporator directly immersed in the storage medium. This work was of particular interest as regards the effect on COP with varying a number of parameters with solar collectors and thermal storage. They noticed that the increase in performance with storage was considerable. With a storage of 3400 kg with 2 plate collector COP improved from 2.3 to 5.5 with an collector area of 25 to 150 m^2.

2.1.3.3.5 Cottingham (1979) studied heat pump with solar energy as heat source. He concluded that COP was higher than 4 at even 266.4°K evaporator temperature. Efficiency of collectors were also studied and it was found to be almost constant over a good range of intercept, that is, 0.0 to 0.16°C-m^2/w.

2.1.3.3.6 Setty (1979) made economic study of solar assisted heat pump and found that for temperature range of 294°K to 322°K cheaper flat plate collectors are necessary.

2.1.3.3.7 Andrews (1978) reported the solar collector as having a straight line performance characteristic relating efficiency and the ΔT/I ratio, where ΔT was the ambient to mean collector temperature difference and I was the input incident solar radiation.

2.1.3.4 Waste Heat

2.1.3.4.1 Smith (1976) and Freund (1978) used heat pumps to recover waste heat from several processes. Kolbusz (1974) claimed the waste heat recovered from a condenser also served as the heat pump evaporator and was used by the steam-driven heat pump as the first stage in heating water. He showed that a COP of 6.6 was achieved. A heat source at 28°C in the system was able to supply water at 70°C via heat pump.

2.1.3.5 Heat Recovery from Refrigeration Plant

2.1.3.5.1 Bowen (1976) tried to recover heat from condenser of refrigeration plant. The headquarters of the Merseyride and MANWEB possesses the first large scale fully heat recovery system, Anon (1969).

2.1.4 HEAT PUMP CIRCUITS

The basic heat pump circuits which are commonly used are of four designs.

2.1.4.1 Air to air

In this system atmospheric air is used as a source of heat and for absorbing heat from the condenser. The heat pump performance decreases and the heating requirement increases as the outside air temperature drops. Defrosting becomes a problem at operating temperatures below freezing. If the air temperature drop is kept low, large quantities of air must be handled. This involves large equipments and possible noise problems thereby limiting its use as described by Jordon and Priestor (1973).

2.1.4.2 Water to air

This design is of two types:

2.1.4.2.1 Water to air

Here the atmospheric water is used as a source of heat and air is used to carry away this heat.

2.1.4.2.2 Air to water

Outdoor air is used as the source of heat and water is used as the transferring medium between the air going to the conditioned space and refrigerant coming to water coil.

2.1.4.3 Water to water

In this system, water is used as a source of heat and also as a medium of heat transfer.

2.1.4.3.1 The air to air and water to air circuit arrangements are most simple in construction and economical in operation. Dunn and Reay (1978) described the heat pipe as a device which, using an evaporation - condensation cycle, transfers heat isothermally.

2.1.4.3.2 In water to water and air to liquid circuits have a fixed refrigerant circuit in which refrigerant flows in one direction only. This arrangement reduces the leakages and also problem of oil return to the compressor. Simplicity in equipment design is achieved in these two cases because the condenser and the evaporator are not required to serve the dual purpose (Domkundwar, 1980).

2.1.5 WORKING FLUIDS

2.1.5.1 Any substance capable of absorbing heat can be used as working fluid for heat pump. Jordon and Priestor (1973) reported that desirable refrigerants are those which possess chemical, physical and thermodynamic properties that permit their efficient application and service in practical design. The common refrigerants such as air, ammonia, carbon dioxide and the freon group etc. can be compared on the basis of above properties. Effect on perishable material is also an important factor in Dairy and Food Industry. Freon vapours have no effect on dairy products, meats, vegetables, flowers and plant life etc. Many foods when exposed to ammonia become unplatable. Freon group refrigerants are non-flamable and oil miscible. The freon refrigerants can be used with practically all metals without danger of corrosion and ammonia can be used only with ferrous materials.

2.1.5.2 Holland *et al.* (1982) compiled an exhaustive work derived from thermodynamic design data for 21 working fluids.

2.1.5.3 Cole and Pietsch (1973) observed that the system performance could be optimized with a precise amount of refrigerant charge. Overcharge or under-charge would damage the system. Jebson and Lascelles (1977) used F-12 for pasteurization of milk. Milk Marketing Board's dairy at Bamber Bridge Lane, U.K. utilised R-12 and R-114. The systems were used for producing hot water from waste heat at temperatures between 300°K and 350°K upgrading it to 333°K and 383°K. For the lower temperature lifts R-12 and for higher temperatures R-114 were used (Anon, 1981).

2.1.5.4 Grooves (1981) studied coefficient of performance of R-11, R-12 and R-22 and found the range of COP as 4.4 to 5.5, the highest for R-11 and the lowest for R-22. But R-11 is used only in centrifugal compressors because of its large specific volume. The R-12 is used in high ambient temperatures where condensing pressure must be kept low enough to prevent overheating of the compressor. R-22 is used in the low ambient temperatures.

2.1.5.5 Griswold and Hellickson (1984) used R-22 for milk cooling. They sized the components for R-12 and R-22 because of their superiority for high temperature applications. Performance data indicated that in-series operation of a desuperheater energy recovery system and water-to-water heat pump had excellent energy conservation potential on dairy farms. Electrical energy consumption for tempering low and high grade process hot water decreased from 110.0 to 51.3 kwh/1000 litre of water, an overall reduction of 53 per cent Pay back period was 12.5 years. Combining energy savings with system cost reduction was able to make pay back period of 5 years.

2.1.5.6 Ewert (1985) used R-22 for a single unit dual-source (SUDS) heat pump evaporator. The SUDS combines two separate evaporators into a single evaporator. The system showed that the uncertainity in the evaporator loading to be ± 15 per cent.

2.1.5.7 Hama and Tani (1984) used heat pump with a two-stage cascade refrigeration system with two different refrigerants (R-12 and R-22). They obtained hot water at 353°K.

2.1.5.8 Balakumar *et al.* (1985) also used a cascade heat pump using R-11 and R-12 refrigerants with vapour compression system. Their experiments showed that R-11 condensing temperatures in the range of 338°K to 368°K could be achieved. Corresponding values for R-12 varied from 271°K to 281°K. Overall COP of 1.2 to 1.7 was obtained.

2.1.5.9 Patwardhan (1987) derived and tested a criterian for determining whether the isentropic compression of saturated vapours led to superheat or condensation. This criterian needed only values of critical temperature, the isentric factor and liquid specific heat. This criterian could be used for evaluating a heat pump working fluid when detailed thermodynamic data are not available.

2.1.5.10 Patwardhan and Patwardhan (1987) also studied COP of the heat pumps with eight different working fluids and evolved a simplified procedure for estimation of COP The results indicated that the procedure could predict COP values within three to four per cent.

2.1.5.11 Patwardhan *et al.* (1987) reported the use of nonzaeotropic mixtures of R-11, R-12 and R-114. It was found that R-11/R-12 mixture is better than pure R-11. The improvement in COP was found to be more pronounced for R-11/R-12 mixture than for the R-12/R-114 mixture.

2.1.5.12 Devotta (1988) described the factors for selecting a working fluid with reference to thermodynamic, thermophysical and safety aspects. Properties with reference to heat pump applications for a few halocarbons were also described.

2.1.5.13 The operating temperatures for different refrigerants are described by Reay and Macmichael (1979) and Devotta (1988). The data are useful for proper selection of refrigerants.

2.1.6 COMPRESSORS

Following compressors are commonly used in heat pumps depending upon the refrigerant;

(1) Rotary vane
(2) Reciprocating
(3) Screw
(4) Centrifugal

2.1.6.1 The compressors mentioned above are well tried and universally accepted in domestic and industrial refrigeration. Reay and Macmichael (1979) described certain operational differences between the refrigerator and the heat pump. These were:

(a) Heat pumps normally operate at the higher condensing temperatures and greater pressure ratios than refrigerators. This puts more stress on both refrigerant and compressor.

(b) Heat pumps generally operate for many more hours per annum at their maximum duty than refrigerators or air-conditioning units (except in domestic applications) and this also tests the compressors capabilities.

(c) Because heat pumps must compete commercially with other heating systems such as direct gas or electric heating, on packaged boilers, there is great pressure to reduce the initial cost of the system.

The combined effect of these three requirements set a daunting task for the manufacturers because of high condensing temperatures.

2.1.6.2 Hodgett (1988) gave classification of compressors for working fluids viz., NH_3, R-12, R-22, R-500 and R-114, evaporating and condensing limiting temperatures, and refrigerant

2.1.7.3 Reay and Macmichael (1979) described that internal combustion engines are used in remote areas where electric supply is not available. Other power sources such as wind mill, direct-water power, and animal power could also be utilized depending upon feasibility.

2.1.7.4 Hodgett (1988) reported that the efficiency of the prime mover was of fundamental importance to the overall efficiency of the heat pump. He described that normal starting torque motors fitted with the appropriate starter units can be used for large centrifugal compressors. Most of the other types of compressors can be driven by normal starting torque motors although they are usually started in an unloaded conditions.

2.2 HEAT TRANSFER AND HYDRODYNAMICS IN THIN FILM SCRAPED SURFACE HEAT EXCHANGERS (SSHE)

Design of heat exchangers for heat pumps is a major subject. The design should be such that optimum heat exchange conditions are achieved.

Heat transfer to viscous and heat sensitive milk products is a problem for process engineers in dairy industry. Due to scaling and foaming tendencies of these products, severe fouling results when they are processed in conventional heat exchangers. This imposes restriction on operational temperatures and time. It is thus essential to remove the scale rapidly from heating surface. The thin film-scraped surface heat exchangers (SSHE) is an ideal choice for such applications.

Fischer (1965) described the applications of agitated thin-film heat exchangers. The processes like concentration, distillation, fractionation, stripping and deodorization can be effectively carried in these type of heat exchangers.

Skoczylas (1970) claimed that a thin film scraped surface heat exchanger with rotor having swinging blades has wider application compared to one with fixed blades. Swinging blade type of SSHE was reported to be capable of delivering final product in the form of solid as reported by Hauschild (1969).

Arlidge (1983) reported about a wiped film evaporator with slotted-blade rotor. The blades, with their regularly spaced slots, push a bow wave of fluid ahead, and thus are continuously encountering fluid fed from feed weir and fluid left by preceeding blade. Average film thickness is a function of slot width, depth and spacing. It has been stressed that this evaporator is an ideal one for specialized evaporation processes where the product is heat sensitive and/or viscous. Freeze and Glover (1979) summarized several unique performance characteristics of mechanically agitated thin film evaporators.

2.2.1 MECHANISMS OF HEAT TRANSFER WITH PHASE CHANGE (EVAPORATION)

2.2.1.1 Two different types of mechanism of heat transfer have been suggested for a flowing, boiling and mechanically agitated film. According to first mechanism which occurs at low values of specific heat flux, conduction heat transfer plays a major role across a vapour film combined with evaporation of more volatile component at the surface. The internal friction, force of gravity and peripherial forces created by the effect of rotation are the main cause of flow of this film as reported by Kramers *et al.* (1955).

2.2.1.2 The second mechanism describes that heat transfer takes place because of vapour bubbles formed at the heat transfer surface or at overheated spots in-side the film. Higher values of specific heat flux and turbulent flow of film contribute to this mechanism. The flow of the film may be regarded as the resultant of the forces of internal friction, the force of gravity and the peripherial force affected also by the formation and release of vapour bubbles (Ziolkowski and Skoczylas, 1965).

2.2.1.3 Dieter (1960), Krischbaum and Dieter (1958), Leniger and Veldslra (1959) and Schneider (1955) made studies on dynamically loaded wiper (Samby) and fixed clearance wiper (luwa) evaporators. They observed, in general, that the temperature difference (ΔT) between heating medium and working fluid, and axial flow rate of working fluid had little effect on heat transfer coefficient. Following correlation resulted from the observations of Krischbaum and Dieter (1958):

$$h_s = 437 \frac{N^{0.33} \rho \; K}{\mu} \; W/m^2k \qquad \text{------------(2.1)}$$

Their study also showed that scraped surface heat transfer coefficient decreased with increasing Δ T at constant rotor speed, low flow rates and high values of Δ T.

2.2.1.4 Bressler (1958) reported an optimum speed beyond which an increase in rotor speed had negligible effect upon heat transfer coefficient.

2.2.1.5 Reay (1963) carried out experiments to study the relationship between blade clearance and the thermal performance in thin film heat exchanger with various fixed clearance blades. He observed that heat transfer coefficient reduced by 20 per cent when clearance was doubled and when clearance was decreased by 80 per cent, heat transfer coefficient increased by 20 per cent. When tip clearance is reduced to a specific value, however, higher fluid flow does not have any effect on heat transfer coefficient, after the whole surface becomes wetted.

2.2.1.6 Bott and Sheikh (1966) conducted experiments on scraped surface vertical heat exchanger using mixture of boiling water and glycerol. The feed was maintained at or near to boiling temperature. The water trials showed that as rotor speed was increased, heat transfer coefficient had less effect of film Reynolds number. Also film coefficient was nearly independent of temperature difference, at higher rotor speeds. The scraped surface film coefficient for boiling water increases with speed of rotor in accordance with the following correlation:

$$h_s \propto N^{0.37} \qquad \text{----------(2.2)}$$

which is in agreement with Dieter and Krischbaum who showed $N^{0.33}$.

Following correlation was obtained for heat transfer during boiling of glycerol:

$$Nu = 0.65\ (Re_f)^{0.25}\ (Re_R)^{0.43}\ (Pr)^{0.3}\ (B)^{0.33} \qquad \text{----------(2.3)}$$

All the physical properties were taken as mean of entry and exist conditions.

2.2.1.7 Skoczylas (1970) obtained experimental heat transfer data on a hinged blade wiped film evaporator. He also used data of Krischbaum and Dieter (1958). The working fluids were water, methanol and ethyl glycol solutions. The following generalized correlation was developed by him:

$$Nu = 3103\ (Re_f)^{-0.988}\ (Re_R)^{0.404}\ (Pr)^{-1.053}\ (B)^{-0.326}\ (We)^{0.494} \qquad \text{----------(2.4)}$$

2.2.1.8 Gudheim and Donovan (1957) compared the heat transfer coefficient of vertical straight sided thin film evaporator with horizontal reverse tapered (1°) for low viscosity fluids. They concluded that the product side film coefficient in horizontal tapered heat exchanger was superior to straight vertical heat exchanger. They also studied the effect of wall thickness, viscosity and vacuum inside the heat exchanger on heat transfer coefficient. Holdt (1978) reported that processing of dairy products in SSHE resulted in extension of its shelf life.

2.2.1.9 Abichandani *et al.* (1987) have published an extensive review on the heat transfer and hydrodynamics in thin film scraped surface heat exchangers.

2.2.1.10 Abichandani and Sarma (1991) studied the evaporation of water, milk and cream in horizontal thin film scraped surface heat exchanger. The data generated in 108 experiments were processed in a computer to fit a Box Wilson model in the form:

$$\begin{aligned} P = {} & 189.67 - 3.86(S) - 36.42(V_C) - 16.62(B) \\ & + 0.09\,(S)^2 + 16.56\,(V_C)^2 + 1.35\,(B)^2 \\ & - 1.24\,(S)\,(V_C) - 0.41\,(S)\,(B) + 16.51\,(V_C)\,(B) \end{aligned} \quad \text{----------(2.5)}$$

In developing above correlation, mass flow rate was excluded as it was observed that effect of increasing mass flow rate on power consumption was insignificant. Further it was seen that for a given rotor speed, the power consumption for water was higher than that of more viscous fluids viz. buffalo milk and cream because of water having higher surface tension compared to milk and cream. It was concluded that the optimum range of rotor speed with respect to overall heat transfer coefficient and power consumption was 2.67 to 3.56 m/s. Between 3.56 to 4.45 m/s the increase in U_o was just one per cent while increase in power consumption was 30 per cent.

2.2.1.11 Concentration of milk to high solids in thin film SSHE results in continuous change of physical properties of the product. Further, the product which is initially Newtonian, gradually attains pseudo-plastic properties as it moves along the surface. To take into consideration these changes, Dodeja *et al.* (1990) developed the following correlation for concentration of milk at atmospheric pressure.

$$Nu = 6615.06\ (NRe)_G^{0.1331}\ (Pr)_G^{0.0764}\ \left(\frac{\Delta T}{T_s}\right)^{0.2843} \qquad (2.6)$$

The scraped film heat transfer coefficient increased with increasing steam temperature until the product attained about 45 per cent solids. At higher concentration beyond this level, the h_s decreased with increasing steam temperatures.

It was also noticed that milk could be concentrated in a single pass to about 70 per cent total solids in horizontal thin film SSHE. The product obtained was free from defects such as browning etc. Thus, horizontal thin film SSHE is probably the most versatile of all types of heat exchangers as it can handle wide range of products.

2.2.2 POWER REQUIREMENT

Both rigid and hinged agitator are responsible for clearance between the agitator and internal surface in thin film heat exchangers. The blades are forced towards the heat exchanger wall by springs or centrifugal action, in variable clearance heat exchangers. The clearance varies with operating conditions. Internal friction or turbulence within the liquid is the main source of consumption of energy induced by the rotating blade.

2.2.2.1 The application of slipper bearing theory to compute the clearance power (P_C) requirement of a centrifugally loaded beveled - edge agitator was studied by Kern and Karakas (1959). They developed the following relationship which holds good as θ tends to zero.

$$P_C \propto \dot{M} D^2 N^3 \tan \theta \qquad \text{----------(2.7)}$$

where

θ = angle between blade and tangent to the vessel.

Here the lift force due to pressure increase under the bearing was not accounted.

2.2.2.2 Norton (1942) and Streeter (1958) derived the following expression on the basis of slipper bearing theory accounting lift force due to pressure increase:

$$P_c = \pi^2 D^2 N^2 \mu L \left[\frac{t}{c}\right] \left[\frac{1}{a-1}\right] \left[4 \ln a - \frac{6(a-1)}{a+1}\right] \text{ ft lb}_f\text{/sec} \qquad \text{----------(2.8)}$$

Equating the lift force on the bearing to the centrifugal force relates, M, c and a

$$M = \frac{3\mu L}{N} \left[\frac{t}{c}\right]^2 \left[\frac{1}{a-1}\right]^2 \left[\ln a - \frac{2(a-1)}{a+1}\right] \qquad \text{----------(2.9)}$$

where, t = Circumferential thickness of agitator nearest heat exchanger wall, ft.

M = Mass of agitator, lbm.

a = $\dfrac{C_m}{c}$

C = Minimum clearance between agitator and heat exchanger, ft.

C_m = Maximum clearance between agitator and heat exchanger, ft.

D = Diameter of heat exchanger, ft.

2.2.2.3 The energy consumed by internal friction is approximately 1615-3230 W/m^2 in commercial equipments as stated by Mutzenburg (1965). This is equivalent to the drop in pressure of a liquid flowing at 4-8 m/s through 2.5×10^{-2} m I.D. tube with a surface area of 9.3×10^{-2} m^2. These velocities are considerably higher than actual velocities in tube heat exchangers and explain the high heat transfer values in agitated thin film evaporators.

2.2.2.4 Abichandani and Sarma (1988) reported the power requirements in horizontal thin film SSHE in form of correlation.

The functional relationship between power requirement and physical parameters can be shown as:

$$P \propto (\sigma)^{0.74} (\rho)^{0.26} (\mu)^{-0.12} (N)^{1.4} (D)^{1.54} (L)^{1.0} (BW_B)^{0.12} \qquad \text{----------(2.10)}$$

This clearly establishes the role of surface tension in contrast to density and viscosity of fluid. The power consumption did not increase significantly with an increase in mass flow rate. However, it increased, with increasing speeds and with the number of blades.

2.2.2.5 Abichandani and Sarma (1989) also developed an expression for power requirement during evaporation of milk in horizontal thin film SSHE.

2.3 ENERGY ANALYSIS IN HEAT PUMPS

2.3.1 Bridgers (1967) described economic cost analysis of air conditioning systems running with electric heat pump and steam. It was estimated that a net saving of 10 per cent was in the case of electrical heat pump as compared to steam heating. Comparison between oil, gas and electrical energy used was also done over a period of 15 years. It was found electric source of energy is much cheaper as compared to oil and gas. It was noticed the oil source is almost two times costlier than electric source of energy. Coefficient of performance for refrigeration machine and heat pump were also studied and found them 4.35 and 5 respectively.

It was observed that the heat recovery from refrigeration system of plastic injection moulding machine resulted in annual energy saving amounting to £ 15000 (Anon, 1976).

2.3.2 Petterson and Wells (1977) studied energy saving in distillation process. The conventional distillation column consisted, a reboiler and condenser. Comparison of heat pump and alternative systems in distillation were studied. The distillation system which was first developed as a simple tower with a water cooled condenser and a steam heated reboiler and then treated with incorporation of heat pump. Among the changes brought about by use of a heat pump were a reduction in tower size and elimination of the condenser, replaced by heat pump. Operating and capital costs were reduced significantly. A saving of 25 per cent in cost was observed with heat pump and it was possible due to using conventional waste recovery techniques, particularly when recovered heat was used to replace steam heating.

2.3.3 Jebson and Lascelles (1977) observed that the application of heat pumps to evaporators would result in a saving of at least 35 per cent of energy currently used to evaporate milk. The application of heat pumps to pasteurization of milk was also described, estimating a net saving of energy by 37 per cent possibility of heat pump applications were also described for heating of casein wash water which interm gave a saving of 25 per cent.

2.3.4 Morlok and Kopp (1978) studied the possibilities of energy recovery in dairy plants and their economic effects. Operating costs of heat recovery by a motor-driven heat pump using cooling water at 35°C as the heat source (cooling by 10°C) in the production of hot water at 70°C were examined and calculated to be about 4.6 pfennig/kWh electricity (energy) costs accounting for 86 per cent. By the use of gas engines energy costs could be reduced to about 1.4 pfennig/kWh and operating costs to 2.3 pfennig. It was further emphasized that for economical use of the heat pump in dairies the temperature difference should be small and the heating should be to a relatively low absolute temperature.

2.3.5 Vries (1979) discussed the use of heat pumps for saving energy on dairy farms by heating water with heat extracted from milk. In the light of laboratory results with five heat pumps and a 12 months trial on four farms, it was observed that heat pumps can be economically justified for farms with minimum of 35 cows if electricity charges are f0.20/kWh. Savings obtainable ranged from 12 to 20 kWh/m^3 milk, at normal levels of water usage. Some problems which may arise with heat pumps were pointed out e.g. problems of corrosion of the boiler-condenser, and adverse effects on the compressor.

2.3.6 It was estimated that 10 per cent savings were effected in energy use in the French dairy industry since 1973. Examples of energy economics actually achieved in dairy factories and techniques experimented with were discussed briefly, i.e. for recovery of heat in the exhaust air of spray drying plants, and use of the heat pump for conditioning cheese stores which because of temperature levels used, was considered to be an ideal application (Anon, 1979).

2.3.7 Calm (1979) explained that heat energy used to heat and cool the buildings in U.S.A. is 26 per cent approximately. He explained 20 to 50 per cent energy can be saved by using heat pump centred system.

2.3.8 A comparison of effective energy utilization of heat pumps driven by an electric motor and internal combustion engine has been shown by Reay and Macmichael (1979). It was observed that electric driven heat pump have a lower capital cost than gas engine driven heat pump but the later having a more effective utilization of primary energy resources.

2.3.9 Branwett (1979) discussed basic principles of heat pumps together with their potential use in malting operation. Results showed that use of electric motor-powered heat pumps was uneconomical as compared to combustion engine drive which saved 18 per cent in primary fuel energy. Diagrams for calculation of the economics of heat pump installation were also discussed.

2.3.10 Dallinga (1979) developed a system of heat recovery from control heat storage tanks from which the water was heated in the condensers of the refrigeration plant. In the two fre-heaters water is heated to about 64°C and 55°C and stored in tanks with capacities of 410 and 420 litres. Laboratory tests simulating conditions on farm with about 50 cows, using a 24-24 litre milk tank, were carried out to determine how the efficiency of the two heat pumps was affected by the quantity of milk, the ambient temperature, and the temperature of the feed water. Energy savings with each system were calculated from results of a standard test in which four successive milkings, each 16.7 per cent of the volume of the milk tank, were cooled and stored over a 48 h period, ambient temperature being 20°C and feed water temperature 11°C. Both systems were compared with a conventional air-cooled milk tank having an average coefficient of performance 2.5. The refrigeration coefficient for both fre-heater was 2.25 and 2.52 and net energy savings obtainable were 16.8 and 17.2

kWh/m^3 cooled milk. Based on a five year pay back period and an electricity price of f 0.20/kWh, one fre-heater appeared to be economically justified for a farm with 38 cows and other for a farm with 48 cows and annual milk production of 5500 l/cow.

2.3.11 Elwell *et al.* (1980) developed a system for substituting alternative energy sources for heating of water in dairy farms. The system recovered more than 168,800 kJ/day of refrigeration compressor reject heat, while consuming 16 per cent less electricity to cool the milk in a 120 cow herd.

2.3.12 Vallot (1981) observed on a heat pump installed in France comprised of two circuits a chilled water production circuit with a 'Trane' motor compressor connected to a conventional evaporator and with R-22 as refrigerant and a circuit producing hot water at 95°C, with the conventional condenser, 'Trane' compressor and R-114 refrigerant. It was calculated that the pump recovering 2560000 kcal/day and operating 6000 h/year (300 days of 20 hours) could save 45,000 f/yr compared with a conventional boiler burning 334.7 l heavy fuel oil daily.

2.3.13 Groves (1981) compared the COP of R-11, R-12 and R-22 and also found that for 122 kWh (cost @ 3.5 p/kWh) electrical energy input, the heating effect of 208 kWh (cost 1.51P/kWh) could be obtained. Thus heat pump was found from energy and cost saving angles as an excellent system.

2.3.14 Christensen (1982) worked on vacuum evaporators with heat pump. It was observed that a three effect evaporator with a heat pump gives energy savings of 94 per cent and water savings of 98 per cent compared with single effect evaporator used with a heat pump. These saving were halved if the single effect evaporator was combined with a steam vapour re-compressor.

2.3.15 Hogan *et al.* (1983) used heat pump for low-temperature grain drying. He compared energy cost scenario for electricity, LP gas and natural gas and found electricity to be the cheapest among all. It was found that the heat pump system consumed 39 per cent less electricity than the conventional resistance system. The heat pump system drying energy requirement was 0.557 kWh/kg of water removed as compared to 0.905 kWh/kg by the conventional system. The electric demand of the heat pump was 52 per cent less than the conventional system.

2.3.16 Blandini *et al.* (1983) made energy comparison using heat pump for hot water requirements. They observed a reduction in energy consumption by 60 to 78 per cent.

2.3.17 Gunkel *et al.* (1983) did experimental and simulation study of a wind driven heat pump system and demonstrated the feasibility of heating water and cooling milk with a wind turbine. Operation of a heat pump system directly coupled to a commercial cycloturbine was simulated on a digital computer for a period of nine months. The result showed an average reduction of

45.29 per cent in the total electrical energy consumption on a typical 50 cows dairy farm.

2.3.18 Griswold and Hellickson (1984) worked on heat recovery using a water-to-water heat pump in milking parlours. A combination of desuper heater heat recovery unit and a specialized water-to-water heat pump were installed for a 130 head dairy to heat process water. As operated, the system affected a 53 per cent savings in water heating energy consumption as compared to electric water heater. It was also suggested the proposed optimized operation of the system may allow a 90 per cent reduction in required energy consumption.

2.3.19 In France, in a cheese factory, heat from evaporator condensates at 55°C and the refrigeration unit of a lactose crystallization plant at 40°C was used to heat water to 90°C. About 50 per cent of energy requirement in cheese making was for heating water and a heat pump cost was recovered in 2-3.5 years (Anon, 1984).

2.3.20 Cunney *et al*. (1985) tried an engine driven heat pump used in a barely drying/storage system. They observed 10-15 percent energy saving over conventional drying.

2.3.21 A heat recovery plant, with four water/water heat pumps for heating the factory and providing hot water, was installed in dairy factory. The installation brought about an 18 per cent decrease in oil usage in the 2nd half of the year compared with the same period in the previous year, despite the fact that increased production had increased oil usage by 13.5 per cent, in the 1st half of the year. It was calculated that the plant gave a real cost reduction of DM 45000 in the second half of previous year, giving it an estimated pay back period of 7-8 years (Wissel, 1985).

2.3.22 Molina and Mascle (1986) compared energy consumption of two malt drying processes: (i) The traditional process in which the heat exchanger was fed with hot water from a gas or oil boiler and (ii) a heat pump system in which low-temperature heat expelled from an air/air heat exchanger was used to heat incoming air. They indicated that the malt drying by heat pump reduced energy consumption from 705 to 213 kWh/l of malt. On this basis, it was calculated that the capital cost of the heat pump system could be paid for in energy savings within two years.

2.3.23 Gioco (1987) studied physical properties of rice related to the use of heat-pumps in improving the thermal efficiency of driers: Drying was done keeping in mind temperature less than 40°C with respect to reduce energy consumption. Two systems using heat pumps were tried. Use of heat pump in an experimental static drier reduced specific energy consumption from 5814 to 2616 kJ/kg of water removed (by 55 per cent) and in a pilot-scale dynamic farm drier from 7510 to 3938 kJ/kg (by 48 per cent). Savings in primary energy consumption (allowing for generation and distribution of the electric power required) were of the order of 20 per cent. Both machines resulted in simple operation, although the static design gave fewer dust filteration problems.

2.3.24 Colliver *et al.* (1987) studied electrical energy consumption and demand reduction using heat pump controls. They used microprocessor-based thermostats and outdoor heat pump thermostats. The major results obtained were: (i) the microprocessor based setback thermostats significantly reduced the total energy used by 20.4 per cent and the maximum demand, and (ii) the outdoor heat pump thermostats significantly reduced the total energy used by 7.9 per cent and the maximum demand.

2.3.25 Stinson *et al.* (1987) studied the performance and economics of a Dairy refrigeration heat recovery unit. They pointed out that increasing the condenser pressure from 6.5 bar to 12 bar increased the gross heat recovery from 15.1 MJ/dm^3 to 29.2 MJ/dm^3 for the water cooled system (compared with a change of 13.7 MJ/dm^3 to 24.5 MJ/dm^3 for the air cooled system). The maximum net heat recovery after subtracting the extra compressor energy consumption was 19.1 MJ/dm^3 and 11.5 MJ/dm^3 for water and air cooled systems respectively.

2.3.26 Leshin *et al.* (1987) worked on energy saving systems for milk cooling. The technical and economical characteristics of standard milk-cooling systems used in the USSR were compared with those of systems employing heat pumps or ice banks. In the ice bank systems, which reduced electricity consumption by about 50 per cent, outside air was used to chill the cooling water in winter, without the need for refrigeration units. In other seasons the refrigeration units were switched on automatically when the temperature of water exceeds 2°C. The heat extracted from the milk was used to warm up drinking water for the cows.

2.3.27 Use of slurry heat and solar energy as a heat source in heat pumps were tried by Zlotowki and Steinbruck (1987). The results of studies on two heat pump systems using small heat pumps were as: one system used slurry heat and was installed in a 770 head cattle fattening unit and one incorporated energy absorbers with a 40 m^2 surface area. Energy savings of 24 and 56 per cent were achieved compared with coal and electricity in the first system and 30 and 60 per cent in the 2nd system.

2.3.28 Fainzil'ber (1988) worked on saving energy in pasteurization of milk. This study analysised possibilities of saving energy on USSR dairy farms by using a supplementary heat pump to transfer heat from the milk cooling system to the pasteurizer. The heat pump is of a type used in water-distilling systems and could give energy savings of upto 90 per cent when milk is pasteurized at 90°C.

2.3.29 Holland (1988) gave a procedure for estimating the economics of heat pump with MVCC. This is based mainly on COP of the system. Payback period was also taken into consideration.

2.3.30 Gyanutis and Milyunas (1989) used heat pump in drying grain in order to reduce energy consumption. During tests on two air-air heat pumps when drying grain by forced ventilation, it was found that the use of the heat pump increased the moisture absorption capacity of the air to 4.5 g/kg and the energy consumption for the evaporation of 1 kg moisture from the grain was 0.451 kWh/kg (1624 kJ/kg).

2.4 APPLICATIONS OF HEAT PUMP IN DAIRY AND FOOD INDUSTRY

The application of heat pumps in Dairy and Food industry covers essentially the waste heat recovery from refrigeration, drying, evaporation and effluents.

2.4.1 Davis (1949) reported that an experimental grain dryer was constructed and examined over a range of conditions, partly to verify some theoretical studies carried out earlier. The dryer contained a grain bin with a floor area of 1.3 m^2. The heat pump used a 570 W electric drive, with R-12 as refrigerant. A centrifugal blower of 380 W was used for blowing air. A water-cooled coil was installed for controlling the maximum drying temperature. It was found that the power consumption in Kilowatt-hours per kilogram of water evaporated agreed very closely with the theoretical values and tests showed that the minimum cost occured, at an airflow of between 800 m^3/h and 1000 m^3/h. The value also varied with the drying air temperature, and expressed in terms of specific moisture extraction rate, a figure of 0.28 kWh/kg was achieved at 316°K and 0.27 kWh/kg at 327°K.

2.4.2 Work in the United States, as quoted by Charity (1952) on the conversion of farm milk coolers which conventionally used air-cooled condensers to fulfil a dual role as milk coolers and water heaters indicated useful savings when electric heating was used on the farm for heating of wash water. With the modified arrangement, a condenser coil placed within an insulated water tank could be used to preheat mains water before it passed to conventional water heaters. It was calculated that approximately 250 litres of water could be heated from 289°K to 319°K.

2.4.3 Hodgett (1976) analysed the state of the art in dryer technology and also calculated the amount of water removed in industrial processes of chemical and food industry specially in milk. Whitehead and Roley (1976) suggested some interesting proposals involving the use of waste heat recovery systems, including heat pumps in industries where process steam requirements are high.

2.4.4 A study carried out in France as described by Laroche and Solignac (1976) using water vapour compression in dryers, in association with a conventional heat pump vapour compression cycle. An efficiency of 1000 kJ/kg water evaporated (3.6 kg/kWh) was claimed.

2.4.5 The milpro heat pump system (UK Patent, 1977) includes water treatment facilities and it is claimed that this unit provides all the heat required for operation of a bottle washing machine. A conventional heat exchanger is used to recover heat from final detergent tank, this heat being used to raise the temperature of the pre-rise spray water. Milpro claims that the heat pump and filter system can reduce consumption to 70 kW and 2600 litres per hour, which can run either on natural gas or electricity.

2.4.6 Jebson and Lascelles (1977) reported that an ammonia compressor was employed and the heat from compressed gas was used to evaporate the milk in the first effect of multiple effect evaporator. An additional evaporator after the final effect condensed the vapours from this effect and evaporated ammonia prior to recompression. The application of heat pumps resulted in saving of 35 per cent energy. It was also pointed out that thermal energy savings in pasteurization and heating casein wash water were 37 per cent and 27 per cent respectively.

2.4.7 Lascelles and Jebson (1975) studied the application of heat pumps to the spray drying of concentrated milk. A method has been explained for saving heat energy upto 45 per cent.

2.4.8 Uges (1978) reported that the 'Fre-heater' heat pump was originally designed for milk cooling and enabled the production of hot water at 335°K or 328°K on a commercial scale, without the use of additional energy. All energy withdrawn from milk as well as the energy added to the compressor, were utilised. Practical experience of about two years proved that the 'Fre-Heater' was almost trouble free. The system operated on the principle of thermosyphon. Provisions were also made for automatic draining of surplus water. The average condensing temperatures were 323°K to 328°K. The influence of the feed water temperature between 281°K and 289°K as well as draining of hot water, did not affect cooling capacity.

2.4.9 Wehrung (1978) installed a unit of 400 l capacity on a milk tank. The heat removed from milk (otherwise normally lost) was drawn into a heat recuperator by an exchange system involving rare gases with change of phase. In the tank, milk was cooled from 305°K to 277°K. The heat removed from milk added to the energy consumed by cooling process itself (motor) heated the water from 283°K to 323°K. One litre water at 323°K was obtained from one litre of cooled milk. An additional 1.5 kW resistance heater with a thermostat in the circuit gave warmer water upto 358°K.

2.4.10 Two dairies in U.K. were using integrated combined heat and water recovery systems em-bodying heat pumps at milk marketing Board's Bamber Bridge dairy near Preston and a dairy at walsall. Both installations were employing Westinghouse templifier. These heat pumps used centrifugal compressors together with shell & tube heat exchanger as condenser capable of generating useful temperature upto 383°K to 388°K. The working fluid was R-114. The source water temperature ranged between 300°K to 350°K and upgraded to 333°K to 383°K. A COP of 4.4 was obtained when source water was at 308°K and outlet water temperature was 338°K. COP was studied over a temperature range of 293°K to 353°K for source water and for 333°K to 373°K as outlet water temperature. Comparison in cost analysis was also done (Anon, 1981).

2.4.11 Hogan *et al.* (1983) reported the development and testing of a low temperature heat pump grain dryer. Corn in the heat pump bin was dried from initial moisture content of 22.9 to 14.2 per cent. Energy auditing and economical analysis were done. The heat pump consumed 39 per cent less energy.

2.4.12 Griswold and Helickson (1984) used a combination de-superheater heat recovery system. A water-to-water heat pump was installed in 130-herd dairy to heat water. A 53 per cent savings in water heating energy consumption was reported. They also proposed a modified system to save energy upto 91 per cent.

2.4.13 Ajupov *et al.* (1984) reported automatic control of a heat pump for pasteurisation and cooling of milk. The system was producing water between 363°K and 368°K at the outlet of condenser at the second stage of heat pump while pasteurizing the milk.

2.4.14 Keith and Mittal (1985) reported employing heat pump in vegetable canning industry. 'Carrier- 30 Hm' heat pump was used. The results revealed that supply of 65 per cent of simulated total thermal load was achieved using a heat pump system with an intermediate temperature of 313.6°K. Computerised simulation was of the refrigeration system to meet the requirements of the cooling regulations was independent of type of condenser used but was dependent upon condenser pressure, environmental heat load, milk inlet and final temperature.

2.4.19 A four stage heat pump recovered waste heat and upgraded it to produce hot water at 338°K to feed the malting process. System also used passive air-to-air heat recovery. Bypass air was used through the germination beds. Overall COP was four (Anon, 1987).

2.4.20 Skevington (1987) reported the performance of two heat pumps used by Australian Department of Agril. The first unit was used for drying apple crisps. A COP of 7.5 was achieved. The second unit was for the production of steam under a partial vacuum for use in the low temperature deodorization of mutton and the COP obtained was 2.3.

2.4.21 Ting and Roberts (1988) worked on dehumidifier assisted drying of soyabeans. A flexible and modular soyabean drying system consisted of ambient air drying and heat pump. The module was an bin batch drying/storage unit. The module with a capacity of 21.24 m^3 of soybeans was used to dry 14.16 m^3 of material. The drying air was conditioned with the help of heat pump. During the period when ambient air drying was not possible, the system was used to dry resulted drying from 14.5 per cent m.c. (w.b.) to an average of 11.3 per cent in 74 hours.

2.4.22 Undried grain was stored in an air-tight store and then dried with a heat pump. Moller and Nilsson (1988) noticed air leakage of 10 litres/minute at 100 Pa pressure occurred in the silo but no effect on grain quality was observed. A shorttime of 12-15 days drrying was compared with hot air from drying and found energy consumption ratio of fan and compressor as 3:1, sharing an energy saving potential with heat pump.

2.4.23 Kyritsis *et al.* (1989) worked on green house heating with the help of heat pump. They used a water-to-air heat pump, and a solar assisted air-to-air heat pump for heating a 1000 m^2 green house for tomatoes. Results indicated that the minimum inside temperature was 13°C and 10°C for water-to-air and solar assisted air-to-air heat pumps respectively, when outside temperature was 3°C. Yields were similar in quantity, quality and timing. The electricity consumption of solar-assisted heat pump represented 56-68 per cent electricity demand of the water-to-air heat pump and also total cost was 83 per cent of water-to-air pump, thus solar assisted pump was economical as compared to air-to-air heat pump.

2.4.24 Laville-studer (1990) applied heat pump concept for economical running of artificial hay driers. Operating costs of heat pumps and air dehumidifiers for hay drying were analysed and compared with convenitional systems. It was concluded that cold ventilation and collector units were economical but that heat pumps and air dehumidifiers were less economical than floor drying in turns of total production cost at low level.

2.4.25 Baumgartner (1990) worked on air-air heat pump for drying of maize. A batch drier with air-air heat pump was used to dry chopped maize and sugarbeet. Experiments were conducted with 32-40 and 26-27 per cent DM (2.9-3.5 and 3.2 t respectively) maize and 2.1 t sugarbeet. Results showed that the system was suitable for drying one t wet maize at 30 per cent DM per day but was not recommended for sugarbeet due to high energy requirements of the order of 200 kWh/100 kg dried crop. Average electricity consumption for maize drying was 57.8 kWh/100 kg of dried crop with average water removal of 27.3 kg/h.

2.5 INFERENCES DRAWN FROM SURVEY OF LITERATURE

1. Engineering studies carried on heat pumps are inadequate.

2. Studies on heat pumps employing horizontal thin film SSHE have not been attempted.

3. Twin evaporator heat pumps for multiple heat recovery are not tried. The system appears to be far more promising.

4. The heat transfer correlations reported are for heating/evaporation and cooling at atmospheric pressures. There is need to develop mathematical models for sub-atmospheric pressure conditions.

5. There is a need to develop simplified correlation for power requirement in horizontal thin film SSHE operating under vacuum.

6. Energy auditing and analysis of the proposed system needs to be done.

3. PLAN OF STUDY

3.1 IDENTIFICATION OF THE PROBLEM AND OBJECTIVES

3.1.1 The review of literature indicated that in recent years there has been growing concern over various dairy and food proccesses being highly energy intensive. In such a situation application of heat pump offers on excellent solution. The inferences drawn from the literature led to the study with following objectives:

3.1.1.1 Design of heat pump system comprising of reciprocating compressor, horizontal thin film scraped surface heat exchanger (liquid concentrator), thermostatic expansion valve, vapour separator, vapour condenser, vapour condensate collecting device, liquid cooler, vacuum pump, and inter connecting piping and valves. (mainly functional design)

3.1.1.2 Study the performance of thin film scraped surface heat exchanger working as condenser of the heat pump for concentration of milk.

3.1.1.3 Development of mathematical models for heat transfer coefficients during phase change and power consumption.

3.1.1.4 Evaluation of coefficient of performance of the system.

3.1.1.5 Energy auditing and analysis of the integrated system.

3.2 TECHNICAL PROGRAMME

In order to achieve the above objectives the following technical programme was formulated:

3.2.1 To evolve suitable heat pump system for concentration of milk.

3.2.2 Design and fabrication of a straight sided horizontal thin film SSHE having inner shell of stainless-steel and a Jacket of mild steel to work as Refrigerant condenser /liquid concentrator. The rotor was designed in such a way that number of blades could be changed and staggered. The blades were of variable clearance type.

3.2.3 Design and fabrication of one horizontal pipe-in-pipe heat exchanger having inner pipe of stainless steel and outer pipe of mild steel to work as vapour condenser (Refrigerant evaporator-1).

3.2.4 Design and fabrication of straight sided horizontal thin film SSHE having inner shell of stainless steel and outer jacket of mild steel to work as liquid cooler (Refrigerant evaporator-2). The rotor was provided with four variable clearance blades.

3.2.5 Design and fabrication of vapour separator.

3.2.6 Controlling the speed of rotor by means of stepped pulley and belt.

3.2.7 Maintaining the system under vacuum.

3.2.8 Incorporating condensate collecting device using plexy glass pipe and suitable valves.

3.2.9 Providing instrumentation for measuring and recording the process variables.

3.2.10 Adopting dimensionless analysis and statistical techniques to develop correlations for heat transfer coefficients and power consumption.

3.2.11 Estimation of coefficient of performance of the system.

3.2.12 Energy auditing and analysis for the entire system.

3.3 PROCESS VARIABLES

It was proposed to study the heat transfer and hydrodynamic aspects during evaporation in the integrated heat pump - thin film SSHE system by varying the following parameters:

3.3.1 Rotor speed varied in the range of 3.42 to 6.97 RPS. (3.42, 5.00, 6.00 and 6.97 RPS)

3.3.2 Number of blades were taken as 2, 4, 6 and 8.

3.3.3 Mass flow rate of working liquid was varied between 0.347 x 10^{-2} to 2.5 x 10^{-2} kg/s. (3.47, 5.56, 6.94, 11.11, 16.67, 22.22, 25.00 x 10^{-3} kg/s)

3.3.4 Working liquids used were water, skim milk and buffalo whole milk.

3.4 EXPERIMENTAL SET-UP

An experimental set-up consisted of the following components was developed (Ref. Plate 1).

3.4.1 Feed tank

3.4.2 Horizontal thin film - scraped surface heat exchanger (liquid concentrator)

3.4.3 Vapour Separator

3.4.4 Screw pump (product pump)

3.4.5 Horizontal thin film - scraped surface heat exchanger (liquid cooler)

3.4.6 Pipe-in-pipe heat exchanger (vapour condenser)

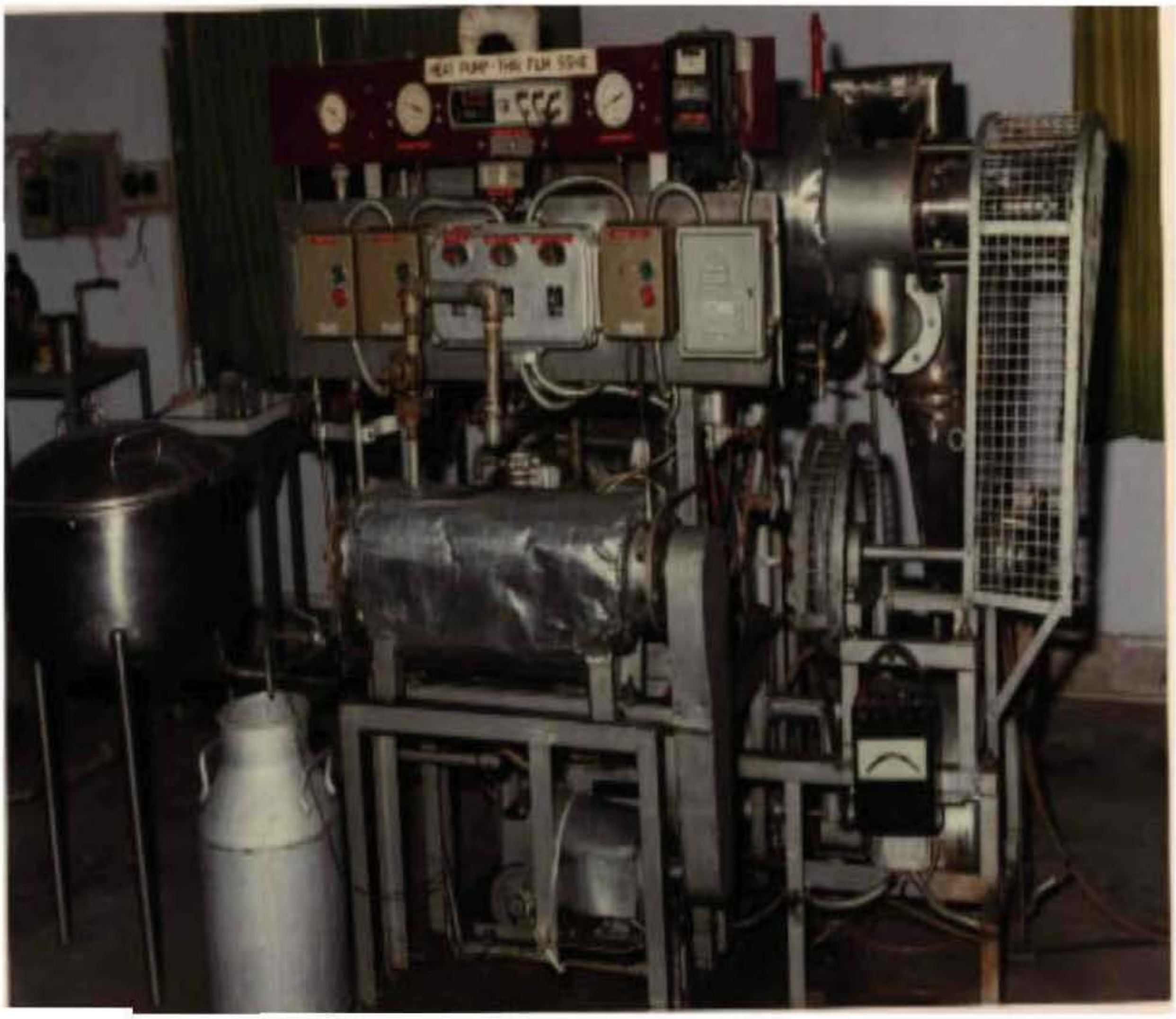


PLATE 1 : EXPERIMENTAL SET-UP

3.4.7 Condensate collecting device

3.4.8 Drive mechanism

3.4.9 Vacuum pump

3.4.10 Rotary vacuum seal

3.4.11 Refrigerant compressor

3.4.12 Expansion valve

3.4.13 LP/HP cut out

3.4.14 Instruments for measuring process variables.

3.4.1 FEED TANK

It was 80 l capacity stainless steel tank with dished bottom to ensure self draining of working liquid.

3.4.2 HORIZONTAL THIN FILM SSHE (LIQUID CONCENTRATOR)

The heat exchanger shell was of 304 stainless steel having $20x10^{-2}$ m I.D., $0.5x10^{-2}$ m wall thickness and $135x10^{-2}$ m overall length. It was provided with mild steel jacket of $27.6x10^{-2}$ m I.D., $1x10^{-2}$ m wall thickness, and $100x10^{-2}$ m length which was also the effective heated length. The heat exchanger was insulated with glass wool of 5×10^{-2} m thickness to minimise convective and radiative losses. An aluminium sheet was used for covering the insulation. The end covers were fabricated from 0.6×10^{-2} m thick 304 stainless steel sheet and flanged with the heat exchanger by means of S.S. nuts and bolts. At one end of the top of heat exchanger an inlet of size I.D. 1.25×10^{-2} m was provided. On the opposite end an outlet of size I.D. 10×10^{-2} m was provided for liquid and vapours. On this end cover, a vacuum seal housing was bolted with S.S. studs.

The rotor assembly was made of solid S.S. shaft of 4×10^{-2} m diameter. Three circular plates with eight slots on each were bolted on the shaft. The flat blades each of size 6.35×10^{-1} m length and 2.5×10^{-2} m width could be hinged to the slotted plates (Ref. Plate 2). A provision was made to change the number of scraping blades upto 16 with staggering positions. Configuration of rotor with staggered blades is shown in Fig. 3.1. The one end of rotor shaft was mounted on sealed ball bearing. This bearing was housed in a three legged spider assembly fixed with S.S. nut and bolts on the inner surface of the heat exchanger. The other end of the rotor was connected to a coupler made of stainless steel. The coupler shaft of 1.2×10^{-2} m diameter was mounted through a ball bearing and a rotory vacuum seal. A safety valve was fitted on the jacket. The mechanical design of heat exchanger is given in Appendix-5.

PLATE 2 : ROTOR WITH STAGGERED BLADES

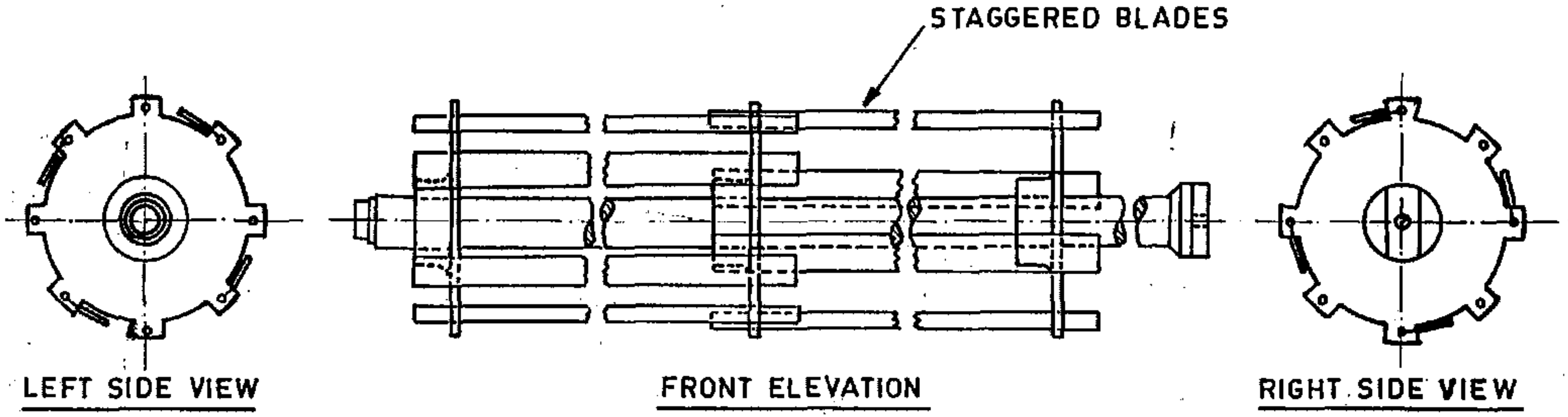


FIG. 3.1 CONFIGURATION OF ROTOR WITH STAGGERED BLADES

3.4.3 VAPOUR SEPARATOR

Cyclones for removing liquid from vapour offer one of the least expensive means both from an operating and an economics view points. The cyclone separator was designed to separate vapours from liquid product assuming vapour velocity to be 4.33 m/s. The mixture of vapour and liquid enters tangentially in a cylindrical chamber at top end. The vapours leave through central opening and liquid through bottom of the conical chamber (Ref. Plate 3). The separator was made of 304 S.S. material having thickness 0.5×10^{-2} m. The vapour separator is shown in Fig 3.2.

3.4.4 SCREW PUMP (PRODUCT PUMP)

It was a screw type positive displacement pump made of stainless steel (S.S.). Its maximum discharge rate was 0.027 lps. The discharge of the pump was regulated through S.S. valve at the suction side of the pump.

3.4.5 THIN FILM - SSHE (LIQUID COOLER)

It was a horizontal thin film SSHE fabricated from 10.16×10^{-2} m nominal diameter S.S. pipe with inner diameter 9.52×10^{-2} m, wall thickness 0.32×10^{-2} m and overall length 76.2×10^{-2} m. This pipe was enclosed by S.S. jacket of 15.24×10^{-2} m I.D. and 66×10^{-2} m effective length.

PLATE 3 : VAPOUR SEPARATOR

The jacket was insulated with glass wool of 2.54×10^{-2} m thickness to minimise the convective and radiative losses. Two ends of inner pipe of the heat exchanger were provided with flanges by means of nut and bolts.

The scraper assembly consisted of solid S.S. shaft of 1.27×10^{-2} m diameter on which supports for holding blades were welded. Four blades were fixed to these supports with the help of half threaded bolts to enable them to have swinging action. The scraper assembly was mounted on pin roller bearings housed at both ends. A seal was provided in each housing to prevent the leakage of working fluid. The driven end of shaft was coupled to rotor assembly by self locking coupler.

The concentrated liquid was pumped to this heat exchanger where it was cooled by rejecting heat to the evaporating refrigerant.

3.4.6 PIPE-IN-PIPE HEAT EXCHANGER (VAPOUR CONDENSER)

It was a horizontal concentric pipe (pipe-in-pipe) heat exchanger for condensing vapours generated in thin film SSHE during evaporation of liquid and separated in vapour separator. It consisted of an inner S.S. pipe of 9.52×10^{-2} m I.D. and 76.2×10^{-2} m length. The outer pipe of S.S. was of 16.50×10^{-2} m inside diameter and 66×10^{-2} effective length. The outer pipe was insulated with glasswool of 2.5×10^{-2} m thickness to minimise heat losses. The two ends of inner pipe were closed with flanges by means of nut & bolts.

One end of flange plate was welded to vapour inlet pipe of 10×10^{-2} m I.D. and the other end of flange plate to 2.54×10^{-2} m m.s. pipe. This pipe was connected to the vacuum pump through a riser to avoid condensate entering into vacuum pump. The bottom of inner pipe end was connected to condensate collecting device.

The vapours enter the heat exchanger and condensed by rejecting heat to the evaporating refrigerant.

3.4.7 CONDENSATE COLLECTING DEVICE

A plexy glass pipe of 10×10^{-2} m I.D. and 0.3 m length was used to collect condensate from pipe-in-pipe heat exchanger (vapour condenser). The condensate collecting pipe was graduated for measuring the quantity of condensate (Ref. Plate 4).

3.4.8 DRIVE MECHANISM

3.4.8.1 The drive to rotor assembly of horizontal thin film SSHE (liquid concentrator) consisted of one 3 h.p., 3 phase, 1400 rpm induction motor, intermediate shaft and stepped V-grooved pulley. The speed of rotor was varied with the help of intermediate shaft and stepped pulley. The mechanism was capable to provide speed in four steps viz. 3.42, 5, 6 and 6.97 RPS.

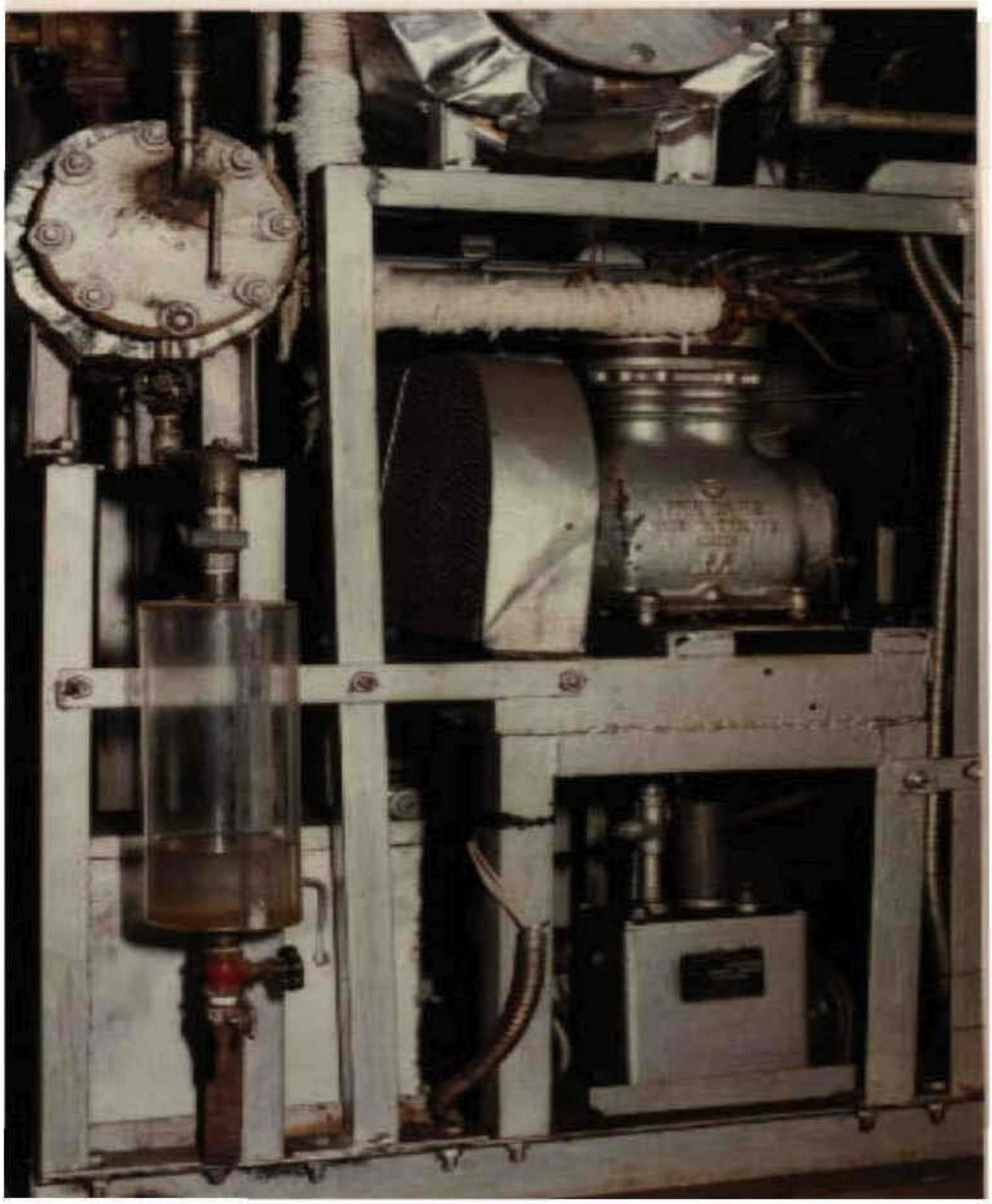


PLATE 4 : CONDENSATE COLLECTING DEVICE

3.4.8.2 The drive to the rotor of other horizontal thin film SSHE (liquid cooler) was given with the help of a motor and a V-belt pulley arrangement. The rotor was driven with a single phase induction motor. The drive system was selected to run the rotor at 9.5 RPS.

3.4.9 VACUUM PUMP

A dry vacuum pump was used to create vacuum in the entire system. To ensure that the vacuum pump handles only air and non condensables, vapour condenser (pipe-in-pipe heat exchanger) was used before the inlet of vacuum pump. Also a riser of 30×10^{-2} m length was provided to prevent the flow of condensate into the suction line of vacuum pump. A vacuum in the range of 625 to 715 mm of Hg was maintained in thin film SSHE (liquid concentrator). Other specifications of the vacuum pump are as follows:

Type	:	Two stage rotary pump
RPM	:	450
HP	:	1/3
Make	:	BASYNTH
Driven by motor	:	Single phase
RPM	:	1440
Amp	:	3
Volts	:	220/240 V

3.4.10 ROTARY VACUUM SEAL

Dura rotary double seal was designed and fabricated by M/s Durametallic (India) Limited for horizontal thin film SSHE for accommodating shaft of 12×10^{-3} m diameter. The seal was capable to withstand temperature upto 423°K. The seal in the S.S. housing was mounted on end plate of the driven side heat exchanger. The seal was cooled by circulating tap water.

3.4.11 REFRIGERANT COMPRESSOR

It was a single acting, twin cylinder vertical, air cooled compressor. It had bore of 5.4×10^{-2} m and stroke of 6.4×10^{-2} m. The speed of compressor was 11.66 RPS. Other specifications are as follows:

Make:	FRIGIDAIRE ENGLAND		
Capacity:	1.5 TR		
Motor:	HP	:	2
	RPM	:	1430
	Amps	:	3.2
	Make	:	Jyoti Ltd. Baroda
	Phase	:	Three

3.4.12 EXPANSION VALVE

This is one of the basic components of the refrigeration system. The function performed by expansion valve includes the reduction in temperature and pressure of the refrigerant coming from the condenser (liquid concentrator), and also to regulate the flow of the refrigerant as per load on the evaporators.

Type	:	Thermostatic Expansion valve
Orifice	:	5.6 mm
Inlet	:	9.5 mm
Outlet	:	15.8 mm
Tube length	:	1525 mm
Maximum pressure	:	3.8 kg/cm^2
Make	:	Detriot Refg. components

3.4.13 LOW PRESSURE/HIGH PRESSURE CUT OUT

High pressure cutout	:	5-25 kg/cm^2
Low pressure cutout	:	0-3 kg/cm^2
Differential cutout	:	5-25 kg/cm^2
Make	:	Ranco

3.4.14 INSTRUMENTS FOR MEASURING PROCESS VARIABLE

Following instruments were mounted on the experimental set-up to measure and control the process variables mentioned:

3.4.14.1 Rotameter

To measure the flow rate of working fluid in kg/min.

Range	:	0-90 kg/h
Make	:	Instrtumentation Engineers
S.G.	:	1.00
Accuracy	:	± 0.5

3.4.14.2 Digital temperature indicator

To measure the temperatures at various locations of experimental set-up.

Range	:	0-200°C
Resolution	:	0.1°C
Make	:	Century Instruments Pvt. Ltd.
Accuracy	:	± 0.5% ± 1 digit
Number of channels	:	six

3.4.14.3 Vacuum gauge

To measure the vacuum in the system.

Type	:	Bourdan tube
Range	:	0-760 mm Hg

3.4.14.4 Pressure gauges

To measure the suction and discharge pressure of refrigerant (F-12).

Type	:	Bourdan tube
Range	:	1-12.5 kg/cm^2 (Suction)
Range	:	1-25 kg/cm^2 (Discharge)

3.4.14.5 Wattmeter

An electrodynamo meter type to measure power input to the rotor of thin film SSHE (liquid concentrator). Since the load was balanced, therefore only one wattmeter was incorporated in the circuit. Total power input to rotor was computed by multiplying the observed reading with an appropriate multiplying factor.

Range	:	0-2500 W
Accuracy	:	$\pm$ 1
Make	:	Automatic Elect. Pvt. Ltd.

3.4.14.6 Energy meter

Total energy input to the system was measured by connecting the energy meter in supply line.

Capacity	:	3 x 60 A 50 ∿
Range	:	0-1 kWh
Make	:	Volkart Brothers Engineers

3.5 EXPERIMENTAL PROCEDURE

The schematic diagram of experimental set-up is shown in Fig. 3.3.

3.5.1 EVAPORATION OPERATION

The water at near saturation temperature pertaining to absolute pressure in the system (44-50°C) was filled in the feed tank (1). The vacuum pump was switched-on. The regulating valve (V_1) was adjusted until the desired flow rate was achieved. Subsequently the motor of refrigerant compressor was started after adjusting LP/HP cutout. Then the drive mechanism was switched on to run the rotor of thin film SSHE (liquid concentrator). Due to centrifugal action of blades, the liquid gets spread over the heat transfer surface in the form of a thin film. When sufficient water was collected in the vapour separator then product pump (10) was started to pump the hot water into the liquid cooler and its motor was started to run the rotor. The cooled water was collected in the can. When the steady state was attained, the process variables were recorded. The procedure was repeated for skim milk and buffalo whole milk. The observations are recorded in Table 1 to 12 (Appendix 6-8). The temperatures at inlet and outlet of both thin film SSHE (liquid concentrator and liquid cooler) were measured by temperature sensors inserted in respective thermo-wells.

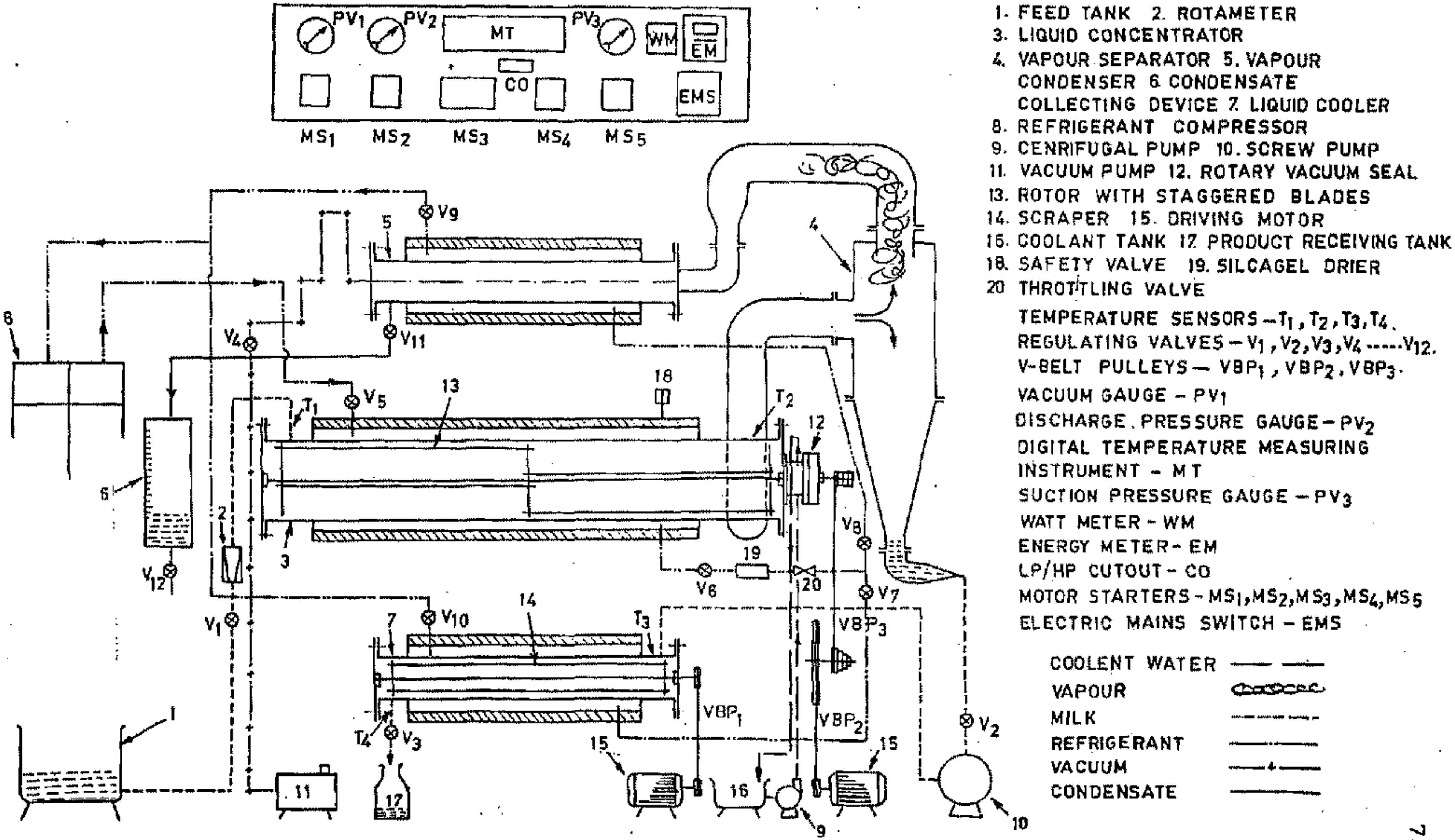


FIG. 3·3. SCHEMATIC DIAGRAM OF INTEGRATED HEAT PUMP-THIN FILM SCRAPED SURFACE HEAT EXCHANGER SYSTEM FOR CONCENTRATION OF MILK

3.6 SANITIZATION OF EXPERIMENTAL SET-UP

The experimental set-up was sanitized by the following sequence of operations:

3.6.1 The clean luke warm water was run in the entire system.

3.6.2 Sodium hydroxide solution of one per cent concentration and temperature around 65-70°C was recirculated for 20 minutes.

3.6.3 After draining sodium hydroxide solution, the set-up was rinsed with water at 65-70°C for 10 minutes.

3.6.4 Nitric acid of one per cent concentration at 65-70°C was recirculated for 20 minutes.

3.6.5 Nitric acid was drained and equipment was again rinsed thoroughly with water at 65-70°C for 10 minutes.

3.7 MATHEMATICAL MODELS

The mechanisms of heat transfer and the flow phenomenon in horizontal thin film SSHE are extremely complex. Because of large number of process variables affecting the hydrodynamics and heat transfer processes, exact mathematical approach may not be feasible. The alternative approach could be the development of various dimensionless groups using Buckinghm-Pi theorem which then can be correlated by appropriate statistical techniques.

3.7.1 EVAPORATION OPERATION

3.7.1.1 Calculation of overall heat transfer coefficient (Uo)

The liquid was pumped into the thin film SSHE (liquid concentrator) at near to saturation temperature. The entire heat transferred by the refrigerant was assumed to be causing evaporation.

Thus:

$$Q = U_o A_o (\Delta T) \qquad \text{----------(3.1)}$$

Where:

A_o = Outside area of heat exchanger

ΔT = $T_R - T_L$

T_L = Saturation temperature of process fluid corresponding to sub-atmospheric pressure

T_R = Temperature of refrigerant corresponding discharge pressure

But $Q = \dot{M}_V L_V$

L_V = Latent heat of vaporization of water at sub-atmospheric pressure

$\dot{M}_V$ = Rate of evaporation

3.7.1.2 Calculation of Refrigerant-side film coefficient (ho)

Refrigerant side heat transfer coefficients, for condensing refrigerant on horizontal pipe, were calculated assuming film type condensation, which is the basis of Nusselt theory (McAdams, 1963).

$$h_o = 1.51\ \phi \left[\frac{4\zeta}{\mu_R}\right]^{-1/3} \qquad \text{----------(3.2)}$$

where ζ = Mass flow rate of liquid refrigerant from the lowest point divided by heated length

$$\phi = \left[\frac{K_R^3 \quad \rho_R \quad g}{\mu_R}\right]^{1/3} \qquad \text{----------(3.3)}$$

where K_R = Thermal conductivity of liquid refrigerant

ρ_R = Density of liquid refrigerant film

μ_R = Absolute viscosity of liquid refrigerant film

All the above properties were corresponding to discharge temperature of refrigerant.

3.7.1.3 <u>Calculation of scraped film heat transfer coefficient (hs)</u>

The individual scraped film heat transfer coefficients were estimated on the basis of resistance concept, taking into account the system geometry and material used.

The resistance correlation is:

$$\frac{1}{h_s} = \frac{D_i}{D_o}\frac{1}{U_o} - \frac{D_i}{D_o}\frac{1}{h_o} - \frac{D_i}{2K_m}\ln\frac{D_o}{D_i} \qquad (3.4)$$

where,

D_i = Inside diameter of heat exchanger (20×10^{-2} m)

D_o = Outside diameter of heat exchanger (21×10^{-2} m)

K_m = Thermal conductivity of wall material. In this case it is 304 SS. The value is 16.27 W/mk.

3.7.2 DIMENSIONLESS CORRELATIONS FOR h_s

The scraped film coefficient can be considered as a function of the following parameters:

3.7.4.1 Physical properties : ρ, μ

3.7.4.2 Thermal properties : K, C_p

3.7.4.3 Kinematic variables : $\dot{M}, N$

3.7.4.4 Geometrical factors : D, L, B

It can be expressed as:

$$h_s = f\ (\dot{M},\ D,\ L,\ \rho,\mu\ ,\ K,\ C_p,\ N,\ g_c,\ B) \qquad \text{----------(3.5)}$$

Appliction of Pi - theorem gives:

$$h_s = f\ (\dot{M},\ D,\ L,\ \rho,\mu\ ,\ K,\ C_p,\ N,\ g_c,\ B)$$

or $\quad f\ (h_s,\ \dot{M},\ D,\ L,\ \rho,\mu\ ,\ K,\ C_p,\ N,\ g_c,\ B) = 0$

Total number of variables = 11

Number of dimensionless groups = 11-5 = 6

$$f\ (\ \pi_1, \pi_2, \pi_3, \pi_4, \pi_5, \pi_6\ ) = 0$$

Number of primary dimensions = 5

Abichandani (1985) developed following correlation using Bukingham Pi-theorem:

$$Nu = C\ (Re_f)^{a_1}\ (Re_R)^{b_1}\ (Pr)^{c_1}\ (B_F)^{d_1}\ (L/D) \qquad \text{----------(3.6)}$$

where,

(L/D) is constant in this case.

3.7.3 CORRELATIONS FOR OVERALL HEAT TRANSFER COEFFICIENT AND POWER REQUIREMENT (BOX WILSON MODEL)

Box-Wilson experimental designs are a series of tests for characterizing a physical mechanism. For this purpose a general series of experiments have been developed which efficiently serve as a basis for deriving mathematical model of the process. In general the model, which is called a response function, includes three types of terms in addition to the intercept constant:

1. Linear terms in each of the variables say X_1, X_2 --------X_p

2. Squared terms in each of the variables say X_1^2, X_2^2 ------- X_p^2

3. Cross-product or direct - order interaction terms for each paired combination X_1 X_2, X_1 X_3 ------------X_{p-1} X_p

The number of terms for P variables is therefore (P+1)·(P+2)/2.

While the general form of the model is not unique (for example, terms like X_1^2 X_2 or X_1 X_2 X_3 could be included - however, this form does not include the first (P+1) (P+2)/2 terms of the Taylor series expansion in P variables), it does have convenient statistical, computational and interceptive properties and is therefore convenient to use in this form.

Most physical situations can usually be approximated by a quaderatic function over a reasonable range of the variables.

For this reason the Box-Wilson experimental designs are very useful in the study of industrial applications. The processes involving more than three variables, the number of tests become excessive with a factorial design and therefore Box-Wilson experiments can be effectively used because they required a relatively small number of tests and have convenient computing properties.

Moreover, the model being easily differentiable, can be used for optimizing any parameter. Hoerl (1959) and Perry *et al.* (1963).

3.7.3.1 <u>Over all heat transfer coefficient (Uo)</u>

Abichandani (1985) developed a correlation for overall heat transfer co-efficient involving all the process variables for atmospheric conditions. Here it is proposed to develop a correlation under sub-atmospheric conditions, thus:

$$U_o \doteq f\ (\dot{M},\ Pr,\ V_c,\ B) \qquad \text{----------(3.7)}$$

then a polynomial model would be:

$$\begin{aligned} U_o = {} & a_0 + a_1\ (\dot{M}) + a_2\ (Pr) + a_3\ (V_c) + a_4\ (B) \\ & + \quad a_{11}\ (\dot{M})^2 + a_{22}\ (Pr)^2 + a_{33}\ (V_c)^2 + a_{44}\ (B)^2 \\ & + \quad a_{12}\ (\dot{M})\ (Pr) + a_{13}\ (\dot{M})\ (V_c) + a_{14}\ (\dot{M})\ (B) \\ & + \quad a_{23}\ (Pr)\ (V_c) + a_{24}\ (Pr)\ (B) \\ & + \quad a_{34}\ (V_c)\ (B) \end{aligned}$$

3.7.3.2 Power requirement (P)

The physical properties of working fluid, kinematic variables, and geometrical factors govern the power requirement to drive the rotor of thin film-SSHE.

Thus, a polynomial model involving process variables such as Vc, B and S would be:

$$
\begin{aligned}
P = {} & a_0 + a_1 (V_c) + a_2 (B) + a_3 (S) \\
& + a_{11} (V_c)^2 + a_{22} (B)^2 + a_{33} (S)^2 \\
& + a_{12} (V_c)(B) + a_{13} (V_c)(S) \\
& + a_{23} (B)(S)
\end{aligned}
$$

3.8 PERFORMANCE OF HEAT PUMP

Primary energy ratio is applied to heat pumps which takes into account not only the heat pump COP but also the conversion efficiency of primary fuel (e.g. oil, gas, coal, solar etc.)

The PER is defined as:

$$
\text{PER} = \frac{\text{Useful heat delivered by heat pump}}{\text{Primary energy consumed}}
$$

PER is also = $\eta \times \text{COP}$

3.9 ENERGY AUDITING

The procedure for energy auditing is delineated in Chapter (Results and Discussion). The net energy saving has been compared between conventional system and heat pump - thin film SSHE system by accounting all primary energy sources.

4. RESULTS AND DISCUSSION

In this chapter, an attempt has been made to present the data and their analysis on the following aspects:

4.1 Heat transfer during evaporation under subatmospheric pressure.

4.2 Power requirement to drive the rotor of horizontal thin film SSHE (liquid concentrator). This rotor was provided with staggered blades.

4.3 Performance of heat pump.

4.4 Energy analysis.

4.1 HEAT TRANSFER DURING EVAPORATION UNDER SUB-ATMOSPHERIC PRESSURE

4.1.1 SCRAPED FILM COEFFICIENT; h_s

In all 236 trials were conducted to study the evaporation of water, skim milk and buffalo whole milk under vacuum. The data were grouped in accordance with the correlation developed earlier for Nusselt number, viz.,

$$Nu = C\,(Re_f)^{a_1}\,(Re_R)^{b_1}\,(Pr)^{c_1}\,(B)^{d_1}$$

The derived data were processed in a computer, HCL-4 to fit in the Cobb-Douglas model. The values of intercept constant c and the coefficients a_1, b_1, c_1 and d_1 were obtained. The regression analysis was made by employing the method of least squares. The correlation obtained was

$$Nu = 0.7959\ (Re_f)^{0.548}\ (Re_R)^{0.082}\ (Pr)^{-0.182}\ (B)^{0.667} \quad \text{----------} \ (4.1)$$

$$2.19 \leq Nu \leq 1438$$
$$4.17 \leq Re_f \leq 74$$
$$119379 \leq Re_R \leq 508552$$
$$2.5 \leq Pr \leq 9.36$$
$$2 \leq B \leq 8$$

The correlation co-efficient was 0.7791.

The effect of each derived parameter (dimensionless number), viz., film Reynold number, rotational Reynold number, Prandtl number and number of blades on scraped film heat transfer co-efficient, h_s is delineated as under.

The HCL-4 computer was used to do error analysis of 236 data sets. Fig. 4.1 illustrates plot of Nu (empirical) versus Nu (experimental) for selected data sets. The 45°-line corresponds to Nu (empirical) = Nu (experimental).

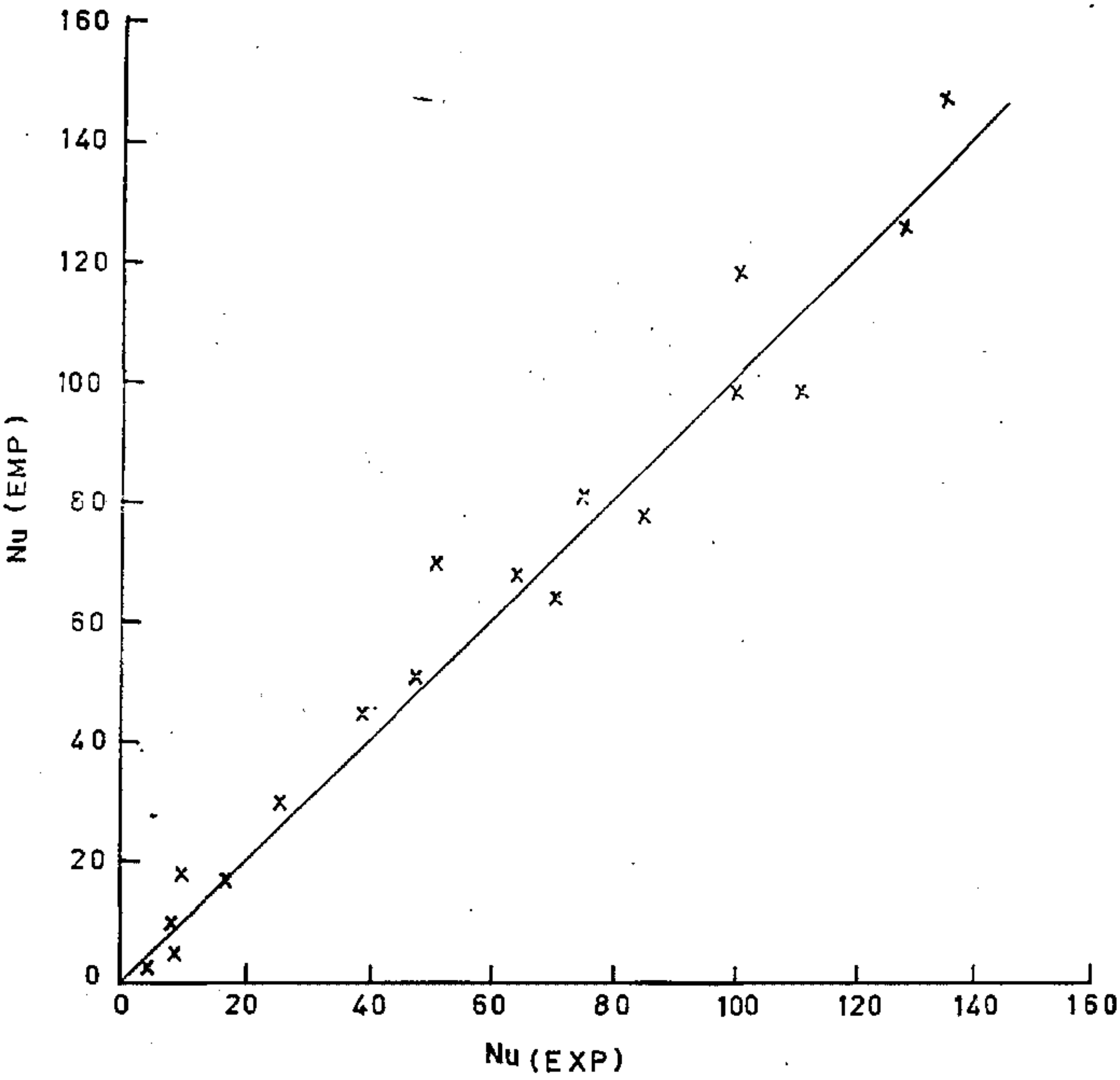


FIG. 4.1. Nu (EXPERIMENTAL) Vs Nu (EMPIRICAL)

4.1.1.1 Effect of film Reynolds number, Re_f on h_s

4.1.1.1.1 The Fig. 4.2 explains the influence of mass flow rate and hence film Reynolds number on scraped film heat transfer coefficient h_s for water, skim milk and buffalo whole milk with 6 blades and at V_c = 2.18 m/s rotor speed. Similar trend was observed at other combinations of number of blades and rotor speeds. It is obvious that h_s increases with increasing Re_f. Further, it can be seen that under similar conditions of mass flow rate etc., the value of h_s is highest in case of water followed by skim milk and buffalo whole mik. This could be due to the viscosity, which is lowest for water followed by skim milk and buffalo whole milk. Also it is evident from equation (4.1) that the index of Re_f is 0.548 which is quite high. This shows that the effect of Re_f on h_s is highly significant. Similar effect of film Reynolds number (generalised Reynolds number) on h_s for horizontal thin film scraped surface heat exchanger was reported by Dodeja (1990).

4.1.1.1.2 In order to explain the effect, that is, increase of h_s with mass flow rate and hence Re_f, the basic mechanism governed by rotating blades on the fluid flow forming thin film along with heat transfer was proposed by Abichandani and Sarma (1991) (Refer to Fig. 4.3).

4.1.1.1.3 The fluid enters the thin film SSHE (liquid concentrator) initially at saturation temperature tends to partly flash and remain at the bottom portion of the heat exchanger.

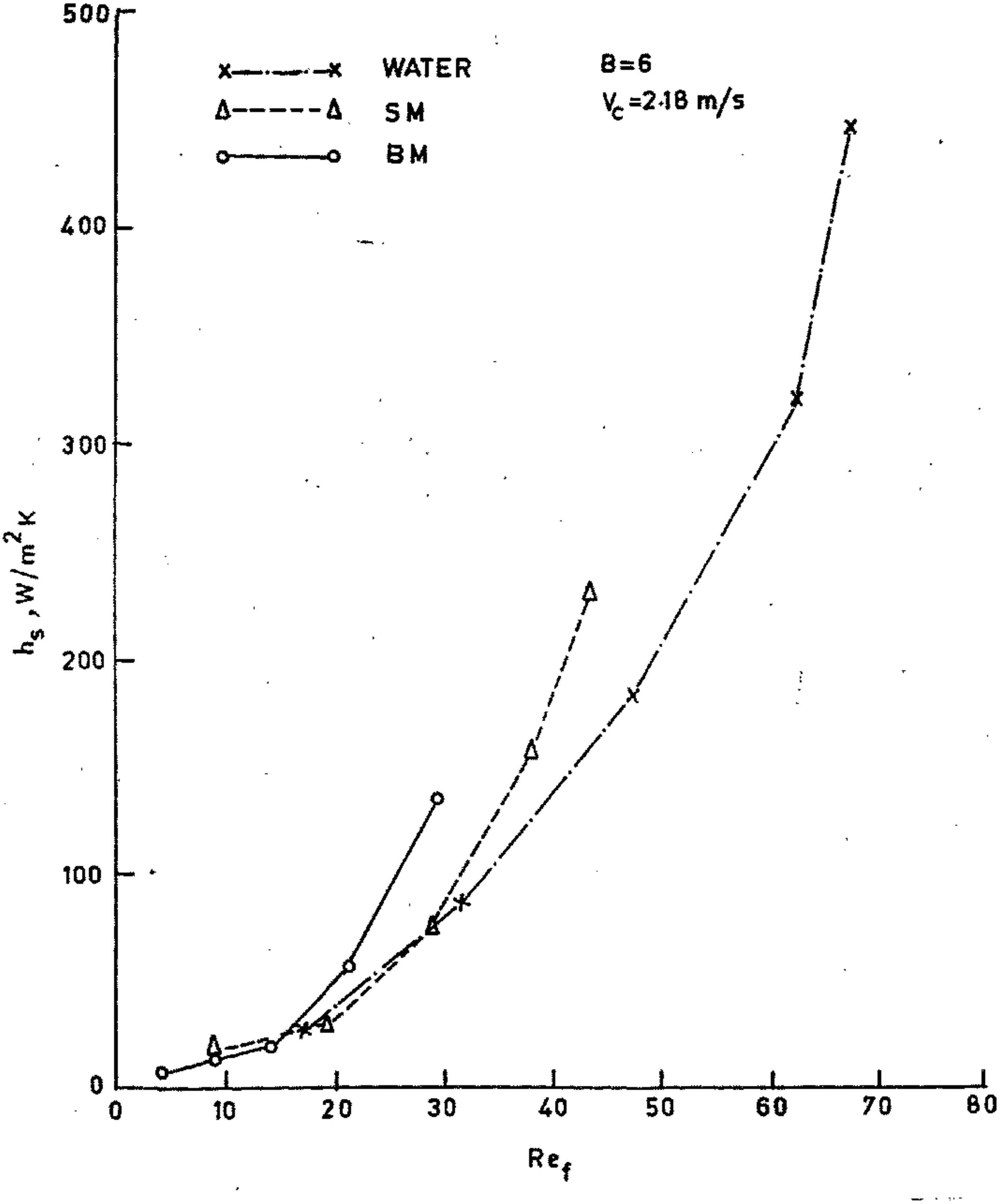
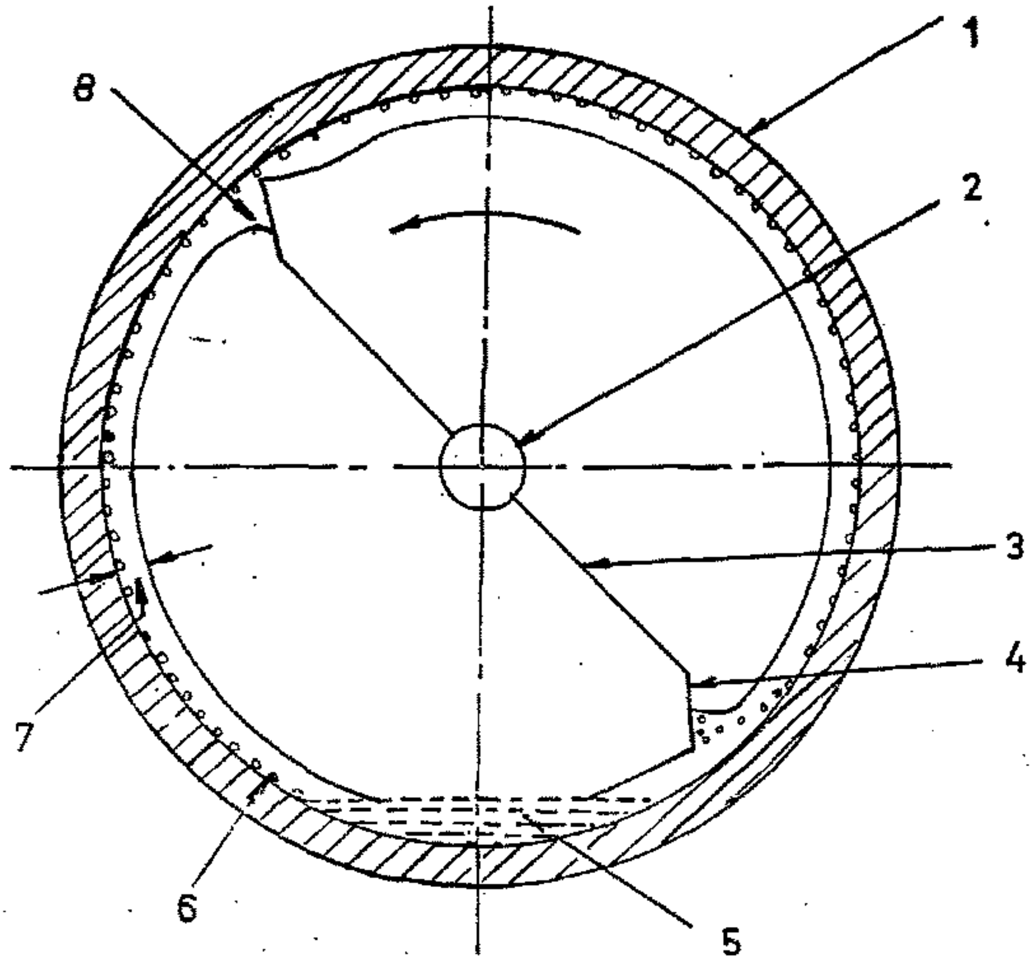


FIG. 4.2. EFFECT OF Re_f ON h_s



1. HEAT EXCHANGER WALL	2. ROTOR SHAFT
3. ROTOR ARM	4. BLADE
5. LIQUID POOL	6. BUBBLES
7. LIQUID FILM	8. LIQUID FILLET

Fig. 4.3 EVAPORATION FROM FILM FORMED IN SSHE

The action of blades is to scoop a certain amount of fluid from the liquid pool and spread in over the internal surface of heat exchanger. At any particular instant, the fluid picked up and carried away by the blade is partly in the shape of a film behind the blade and partly in form of fillet in front of the blades. The action of blade which is similar to that of a plough makes part of the fluid in the film to mix with that in the fillet and at the same time also restore film thickness by providing an equal amount of fluid to squeeze past the tip of the blade.

Since the fluid is subjected to sub-atmospheric conditions and the fluid at the interface of the film and heating surface is already at saturaion temperature, bubbles are formed partially the fluid in film to attain superheat by blade action is also possible.

The bubbles gradually cover the whole surface and if allowed and cause additional barrier to heat transfer. As ealier discussed, the blade action causes the fluid and the bubbles in the film to mix with that present in the fillet. Because the temperature of liquid in the fillet is also saturated as it enters the heat exchanger, the bubbles leave the fillet. The formation of bubbles continues until they are once again replaced/dislodged by the blade following from behind.

The increase in mass flow rate makes the fillet volume to increase after the film attains its final thickness. Consequently, the fluid making contact with the surface of heat exchanger or with the fluid film increases. This gives the high heat transfer rates, so higher value of h_s.

4.1.1.1.4 In case of vertical thin film scraped surface heat exchangers, Kramers *et al.* (1955), Ziolkowskia and Skoczylas (1965) and Bott and Sheikh (1966) also suggested the possibilities of bubble formation and the fluid in the film having superheat.

4.1.1.1.5 Therefore, it can be inferred that no matter whether evaporation is carried out at atmospheric pressure or under vacuum, the scraped film heat transfer coefficient, h_s increases with increasing Re_f.

4.1.1.2 <u>Effect of rotational Reynolds number, Re_R on h_s</u>

4.1.1.2.1 The Fig. 4.4 illustrates the influence of the peripherial rotor speed and hence the Rotational Reynolds number on the scraped film heat transfer coefficient, h_s for water, skim milk and buffalo whole milk. As evident, the h_s increases with increasing rotor speed.

However, it is noticed from the figure that in case of water, h_s increased proportionately as the rotor speed increased from 2.18 to 4.45 m/s. But in case of buffalo milk and skim milk, the increase in h_s is marginal with increase in speed above 2.18 m/s.

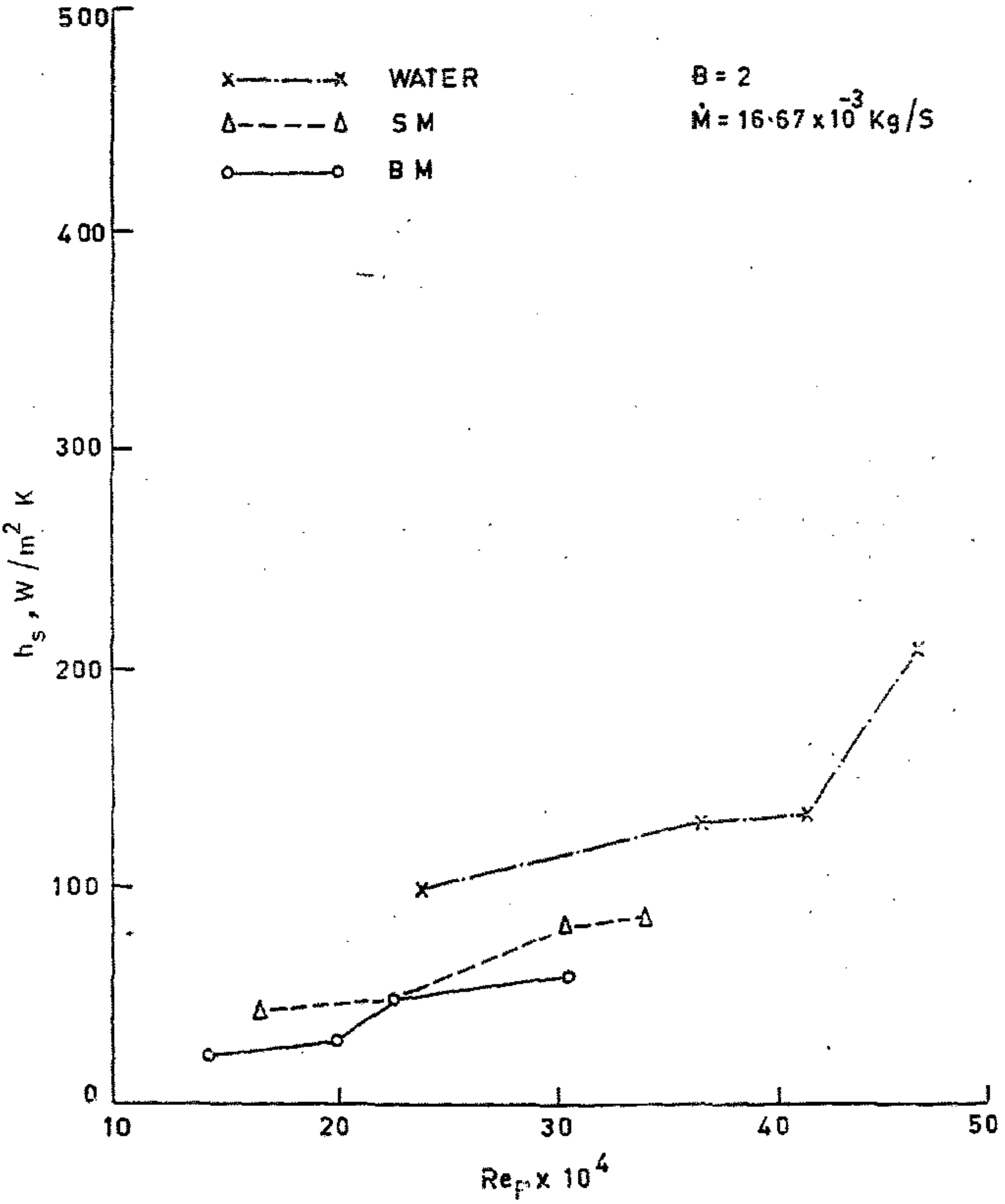


FIG. 4·4 EFFECT OF Re_R ON h_s

The increase of h_s with the rotor speed under sub-atmospheric conditions can be attributed to high turbulence in the fillet and higher frequency of its mixing with liquid pool. Both these factors play important role to cause higher rates of heat transfer. Also the contact period of the fluid film with heat exchanger surface reduces with increase in rotor speed preventing the heat transfer rates from rising as rapidly. This provides justification that as rotor speed increases beyond 2.18 m/s, the increase in h_s is marginal. However, in case of water, increase in h_s is practically proportional to the rotor speed. This could be due to negligible change in viscosity.

4.1.1.2.2 Bressler (1958) made also similar observation in vertical thin film SSHE that there was an optimum rotor speed beyond which an increase in speed had little effect upon heat transfer.

4.1.1.3 Effect of number of blades, B on h_s

4.1.1.3.1 As explained earlier, the rotor was provided with staggered blades in such a way that the angle between two corresponding blades in opposite segments was always 180°. The purpose of staggering the blades was to take advantage by two blades, each covering half the heating length, similar to that given by two full blades each covering the entire heating length of the heat exchanger.

4.1.1.3.2 The equation 4.1 shows the effect of number of blades on the scraped film heat transfer coefficient h_s. The equations predict that,

$$h_s \propto (B)^{0.667} \qquad \text{---------- (4.2)}$$

4.1.1.3.3 The Fig. 4.5 explains the effect of number of blades on h_s for water, skim milk and buffalo whole milk. It is evident that as the number of blades are increased, the h_s increases.

The increase in number of blades causes effects similar to those observed by increasing rotor speed, viz., high degree of turbulence in the fillet and highest frequency of its mixing with the liquid pool. Above all, increase in B also causes a larger quantity of fluid in the form of fillets which further increases the heat transfer rates. However, increase in B or V_c results in higher power consumption.

4.1.1.3.4 Bott and Romero (1963) also reported similar effect for the number of blades on h_s for vertical thin film scraped surface heat exchanger.

4.1.2 OVERALL HEAT TRANSFER COEFFICIENT, U_o

As explained in the chapter on plan of study that the best model in the case of overall heat transfer coefficient could be in the form of polynomial similar to Box Wilson model. Experiments on evaporation of water, skim milk and buffalo whole milk were conducted and data were obtained for different operating conditions like mass flow rate, Prandtl number,

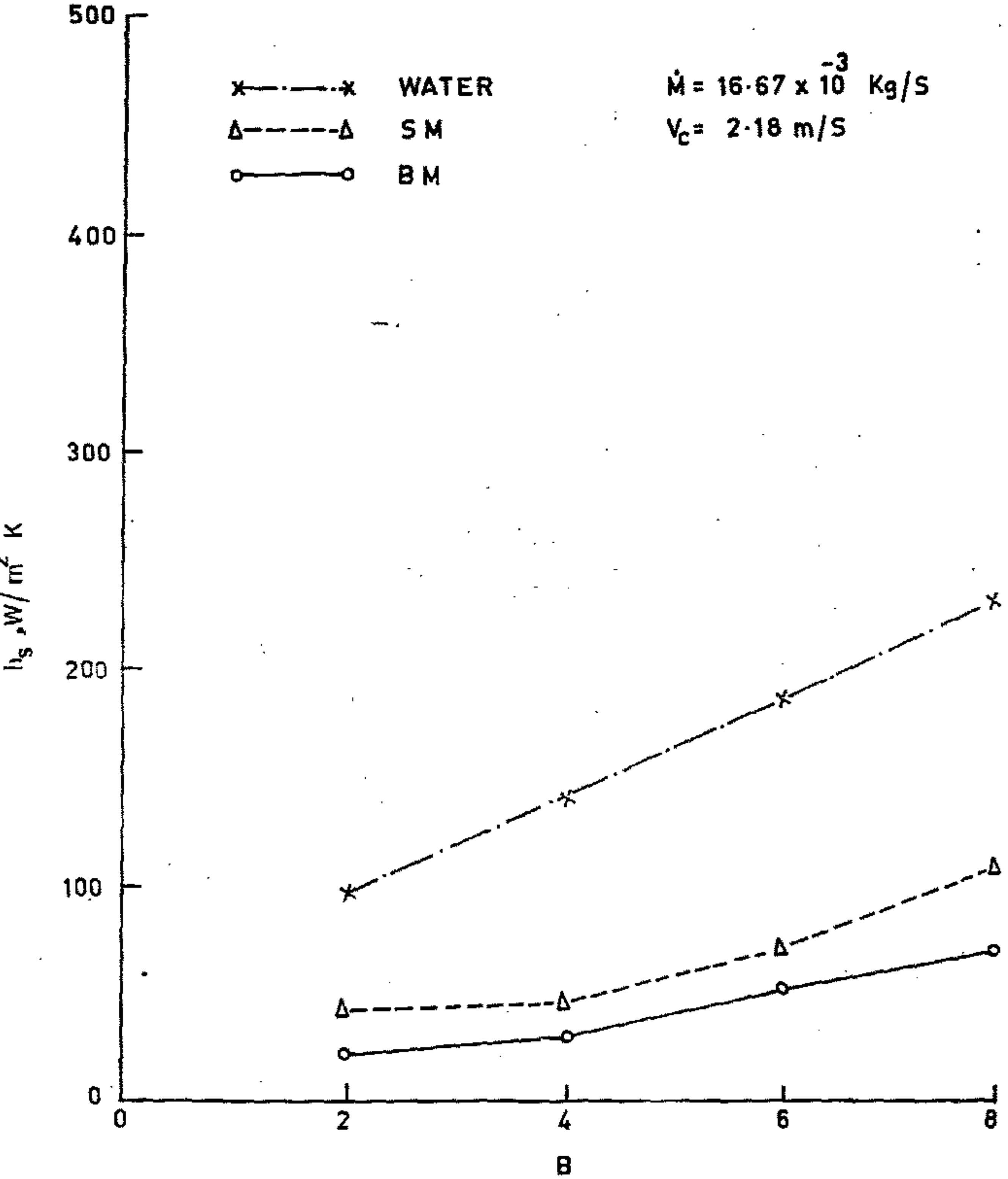


FIG. 4·5 EFFECT OF B ON h_s

peripherial speed of rotor and number of blades. An expression of following form was obtained:

$$U_o = f(\dot{M}, Pr, V_c, B)$$

The data were processed in HCL-4 computer using linear regression technique. The following prediction equation was obtained:

$$\begin{aligned} U_o = {} & 0.3142 + 23663.1\,(\dot{M}) - 0.616\,(Pr) - 30.983\,(V_c) + 5.137\,(B) \\ & -714266.56\,(\dot{M})^2 - 1.82\,(Pr)^2 + 5.67\,(V_c)^2 - 6.0869\,(B)^2 \\ & -250.46\,(\dot{M})(Pr) - 884.15\,(\dot{M})(V_c) - 159.41\,(\dot{M})(B) + \\ & +2.084\,(Pr)(V_c) + 0.863\,(Pr)(B) \end{aligned} \qquad \text{----------} \; (4.3)$$

$$3.42 \times 10^{-3} \leq \dot{M} \leq 25 \times 10^{-3} \text{ kg/s}$$

$$2.5 \leq Pr \leq 9.36$$

$$2.18 \leq V_c \leq 4.45 \text{ m/s}$$

The prediction equation (4.3) can be effectively used in optimizing all the process variables. For instance, the optimum rotor speed V_c is obtained by setting $\frac{\partial u_o}{\partial V_c} = 0$ and determining the value of V_c for which $\frac{\partial^2 u_o}{\partial V_c^2} < 0$. The procedure can be repeated for determining the optimum values of other variables.

Error analysis of 236 data sets for all the three experimental fluids was done with the help of HCL-4 computer.

Fig.4.6 shows plot of $[U_o]$ empirical versus $[U_o]$ experimental for selected data sets. The 45°-line corresponds to $[U_o]$ empirical = $[U_o]$ experimental. It can be noticed that most of the data points lie near to this line.

The effect of process variables on overall heat transfer coefficient, U_o is discussed as under.

4.1.2.1 Effect of mass flow rate, $\dot{M}$ on overall heat transfer coefficient, U_o

4.1.2.1.1 The Fig. 4.7 describes the effect of $\dot{M}$ on U_o at rotor speed of 2.18 m/s and number of blades as 6 for various liquids, viz., water, skim milk and buffalo whole milk. Similar effect was observed for other conditions of rotor speed and number of blades.

It can be seen that the mass flow rate has a profound effect on the overall heat transfer coefficient. For equal mass flow rate, the value of U_o is highest in case of water and lowest in case of buffalo milk.

4.1.2.1.2 Abichandani and Sarma (1991) studied the evaporation of water and milk at atmospheric pressure using steam as heating medium. It was reported that increase in mass flow rate caused U_o to increase for milk only, and not for water.

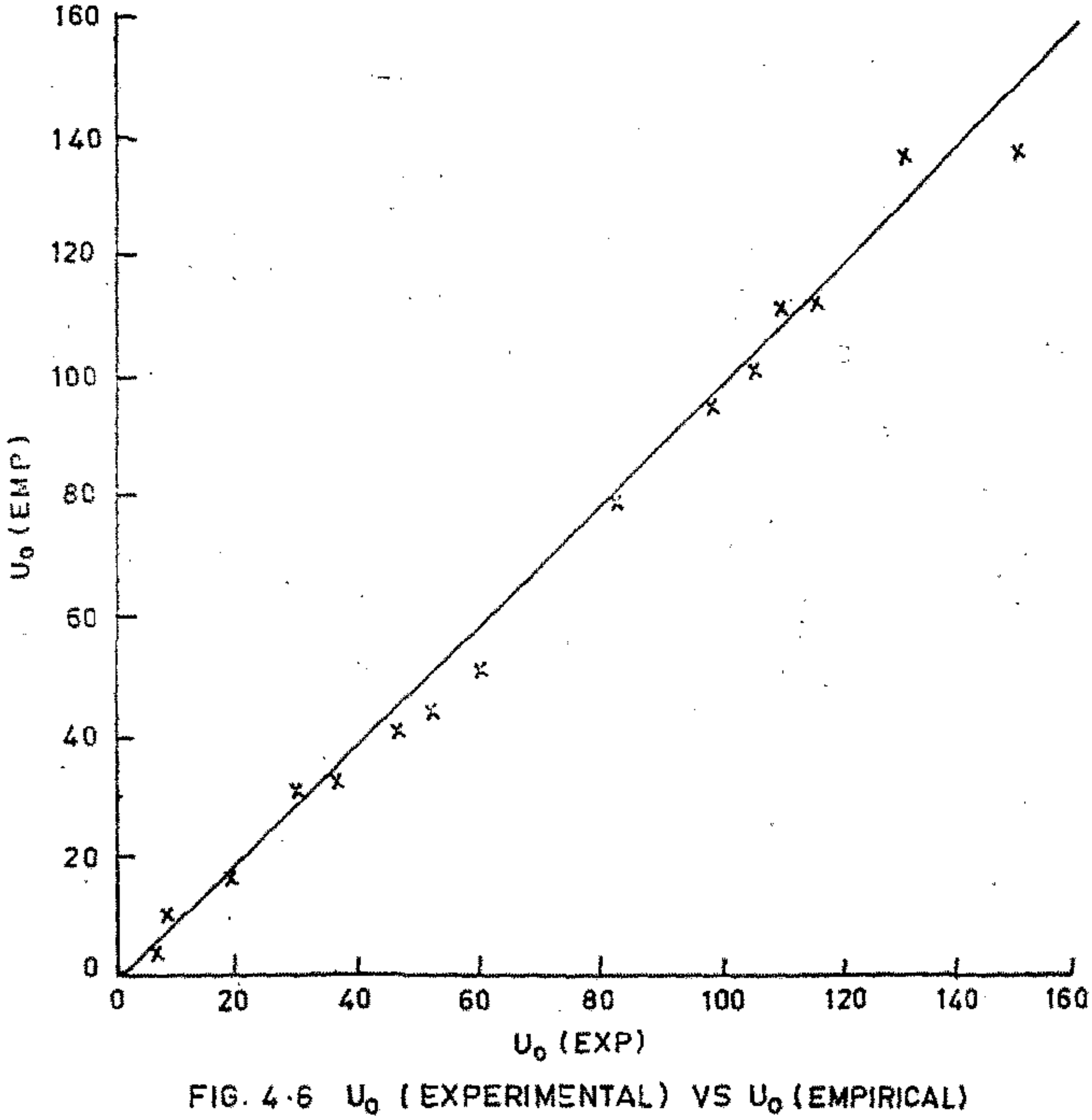


FIG. 4·6 U_0 (EXPERIMENTAL) VS U_0 (EMPIRICAL)

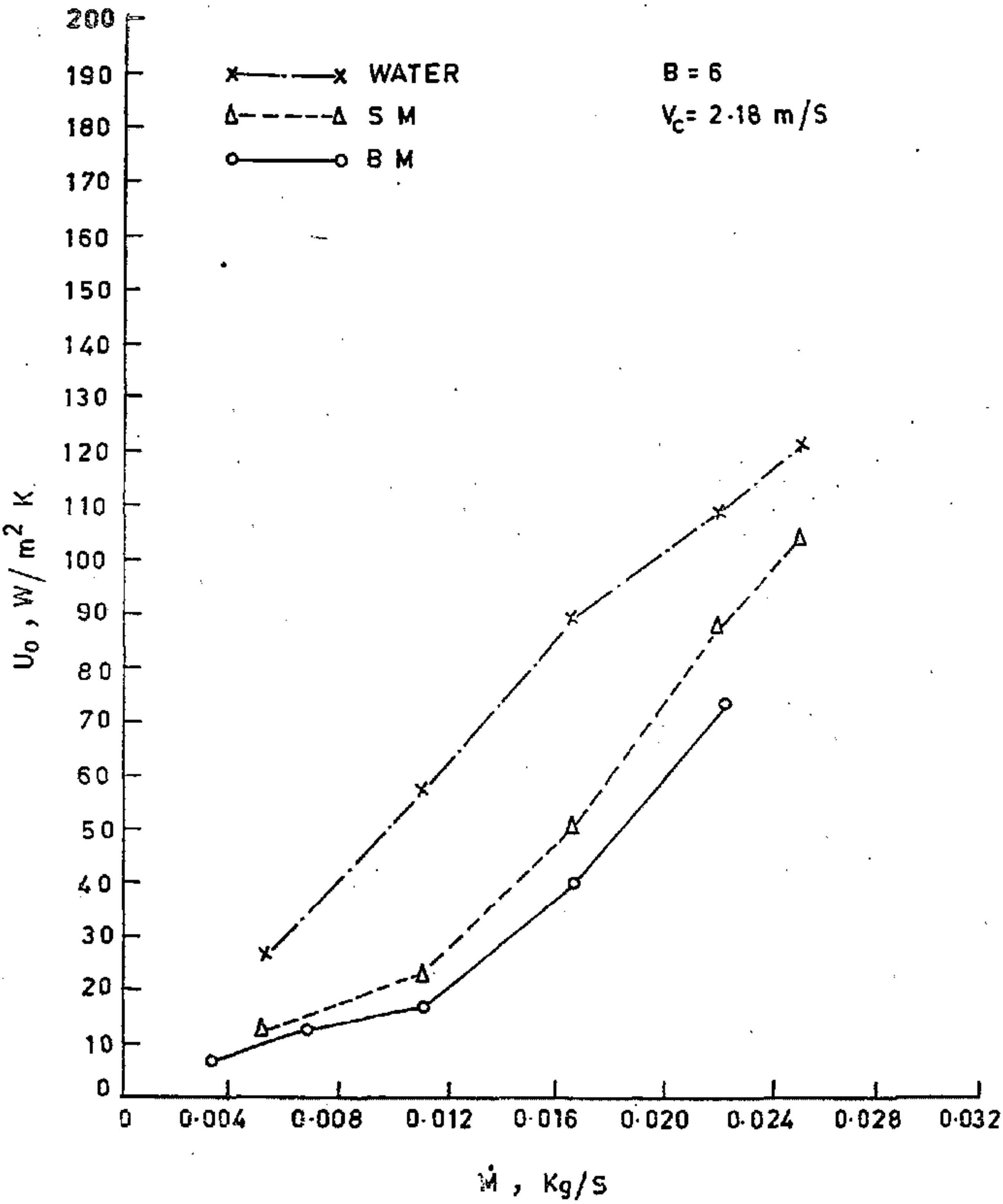


FIG. 4·7. EFFECT OF Ṁ ON U_o

4.1.2.1.3 From the above observations, it can be inferred that the mass flow rate has more pronounced effect on U_o when evaporation is carried out under vacuum than at atmospheric pressure.

4.1.2.2 <u>Effect of rotor speed</u>, V_c <u>on overall heat transfer coefficient</u>, U_o

4.1.2.2.1 The Fig. 4.8 illustrates the effect of the rotor speed, V_c on the overall heat transfer coefficient. Generally, U_o increased with increase in rotor speed. But in particular for skim milk and buffalo whole milk, the increase was appreciable upto rotor speed of 3.83 m/s. In case of water, the increase in U_o was gradual.

Increasing the speed of rotor caused more turbulence and faster dislodging of the bubbles. These factors enhanced the rate of evaporation.

4.1.2.2.2 It was reported by Abichandani and Sarma (1991) that in case of milk, increase in U_o was rapid with increase in rotor speed as compared to that in water. However, increase in U_o beyond the rotor speed of 2.67 m/s was just marginal. It was further mentioned that under similar operating conditions, the U_o values for milk were less than those of water. This is in agreement with the present findings.

4.1.2.2.3 Bressler (1958) made observations on vertical thin film scraped surface heat exchanger that there was an optimum

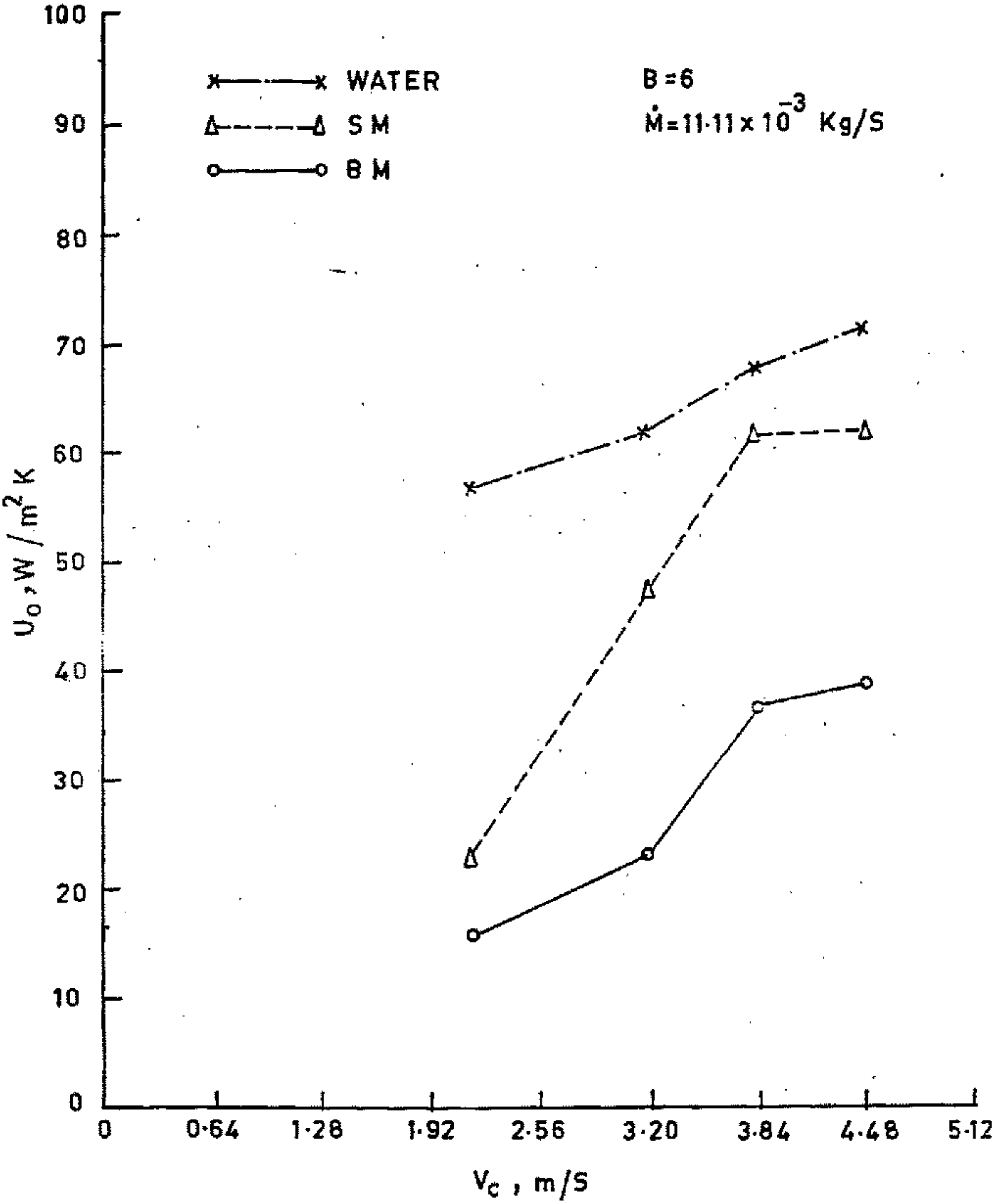


FIG. 4·8 EFFECT OF V_c ON U_o

rotor speed beyond which an increase in speed had little effect upon heat transfer.

4.1.2.3 Effect of number of blades, B on overall heat transfer coefficient, U_o

4.1.2.3.1 The Fig. 4.9 shows the effect of B on U_o for all the fluids, viz., water, skim milk and buffalo whole milk. In case of water after six blades in staggered position, the U_o did not increase significantly. However, in case of milk it increased with increasing number of blades. But beyond six blades, the advantage of increased U_o values, specially for skim milk, was far less than the power consumption. It can be seen that average increase in U_o was 33 per cent when the number of blades increased from six to eight while the average power consumption rose to 66 per cent. Therefore, it is of no advantage to increase the number of blades beyond six.

4.2 POWER REQUIREMENT

The power input to the rotor of horizontal thin film SSHE (liquid concentrator) is one of the deciding factor in judging the suitability of its application. The power required to drive the rotor is dissipated in the following ways.

1. To overcome viscous and surface tension forces to form the film on the heating surface of SSHE.

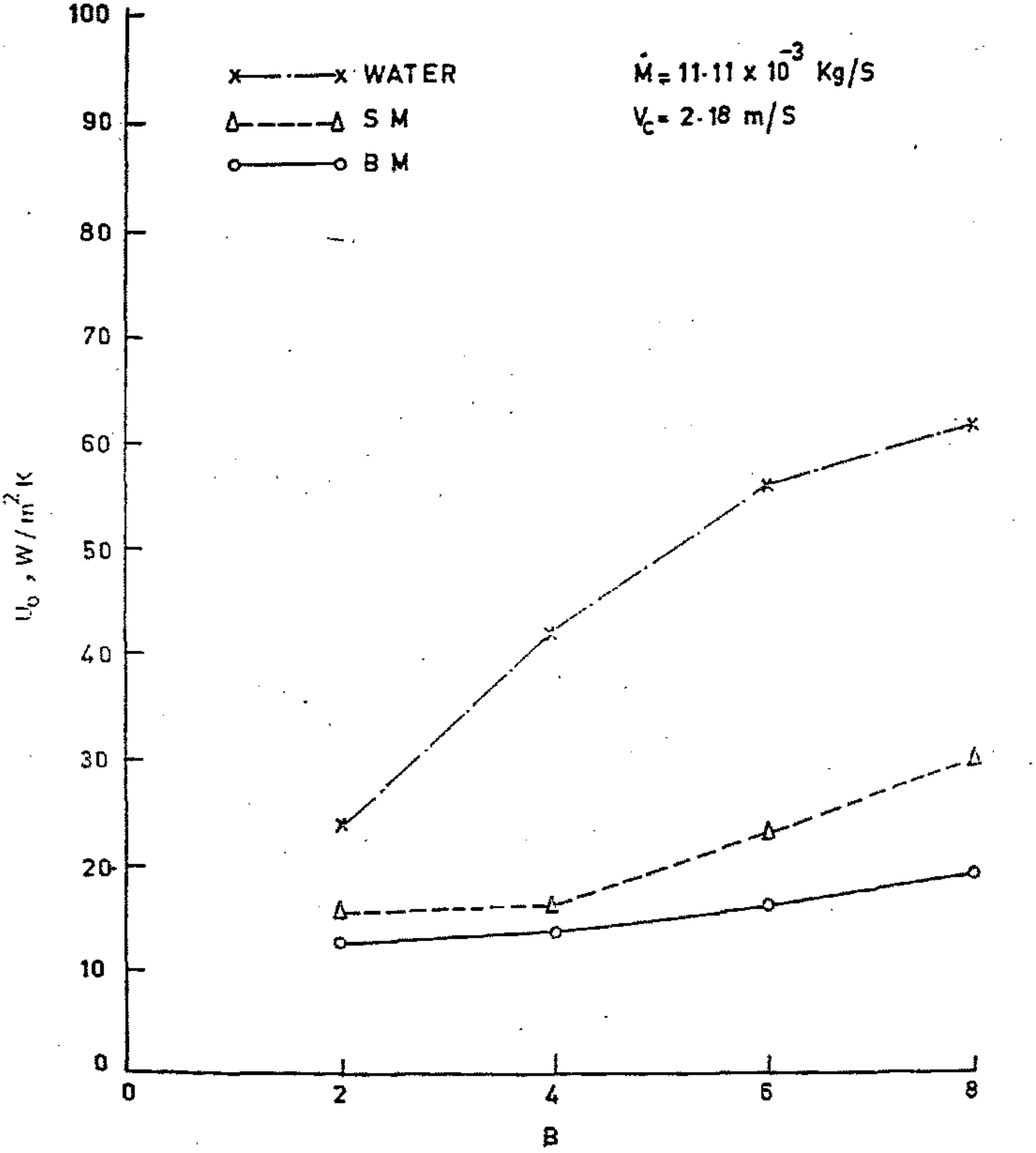


FIG. 4·9 EFFECT OF B ON U_o

2. To overcome the inertial forces of the rotor.

3. To accelerate the fillet to the rotor speed.

4. To agitate the product film.

Norton (1942), Streeter (1958), Kern and Karakas (1959) and Bhattacharya (1970) carried out the work on power consumption in thin film SSHE. The slipper bearing theory was the base for most of the work which is not enough for predicting power consumption in horizontal thin film scraped surface heat exchanger.

Only the viscosity of process fluid was assumed to have the dominating effect on power consumption. In thin film SSHE, where a new surface is generated in every revolution of the rotor, surface tension should play an important role in power consumption.

Abichandani and Sarma (1988) observed that the effect of surface tension was highly pronounced for determining power consumption. Besides surface tension, number of blades and rotor speed also greatly influence the power consumption.

The data on power consumption were taken during the evaporation studies. As mentioned earlier, Box Wilson model can be effectively used in the study of industrial applications. Therefore, it was proposed to develop correlation for power consumption involving process variables, viz. circumferential speed of rotor, number of blades and per cent average total solids, i.e., $P = f(V_C, B, S)$.

The data were processed in HCL-4 computer to fit in the correlation of form:

$$P = 1.023 + 1.215\,(V_C) - 0.729\,(B) + 1.026\,(S) - 0.455\,(V_C)^2 + 0.348\,(B)^2 + 0.010\,(S)^2 + 0.121\,(V_C)(B) - 1.013\,(V_C)(S) - 0.089\,(B)(S) \quad \text{----------- (4.4)}$$

$$2.18 \leq V_C \leq 4.45 \text{ m/s}$$
$$2 \leq B \leq 8$$
$$0 \leq S \leq 20\%$$

Error analysis of 236 data sets was done with the help of computer. Fig. 4.10 shows plot of P (empirical) versus P (experimental) for selected data sets. The 45°-line corresponds to P (empirical) = P (experimental).

Mass flow rate has not been accounted while developing the correlation for the power consumption because as the fraction of fluid present in the form of fillet or the film is independent of the mass flow rate, once the steady film thickness is attained. In other words, the fillet volume would certainly depend upon blade geometry and the power required to accelerate it to the rotor speed remains constant and does not vary with mass flow rate. Also, the power required to accelerate the fillet in

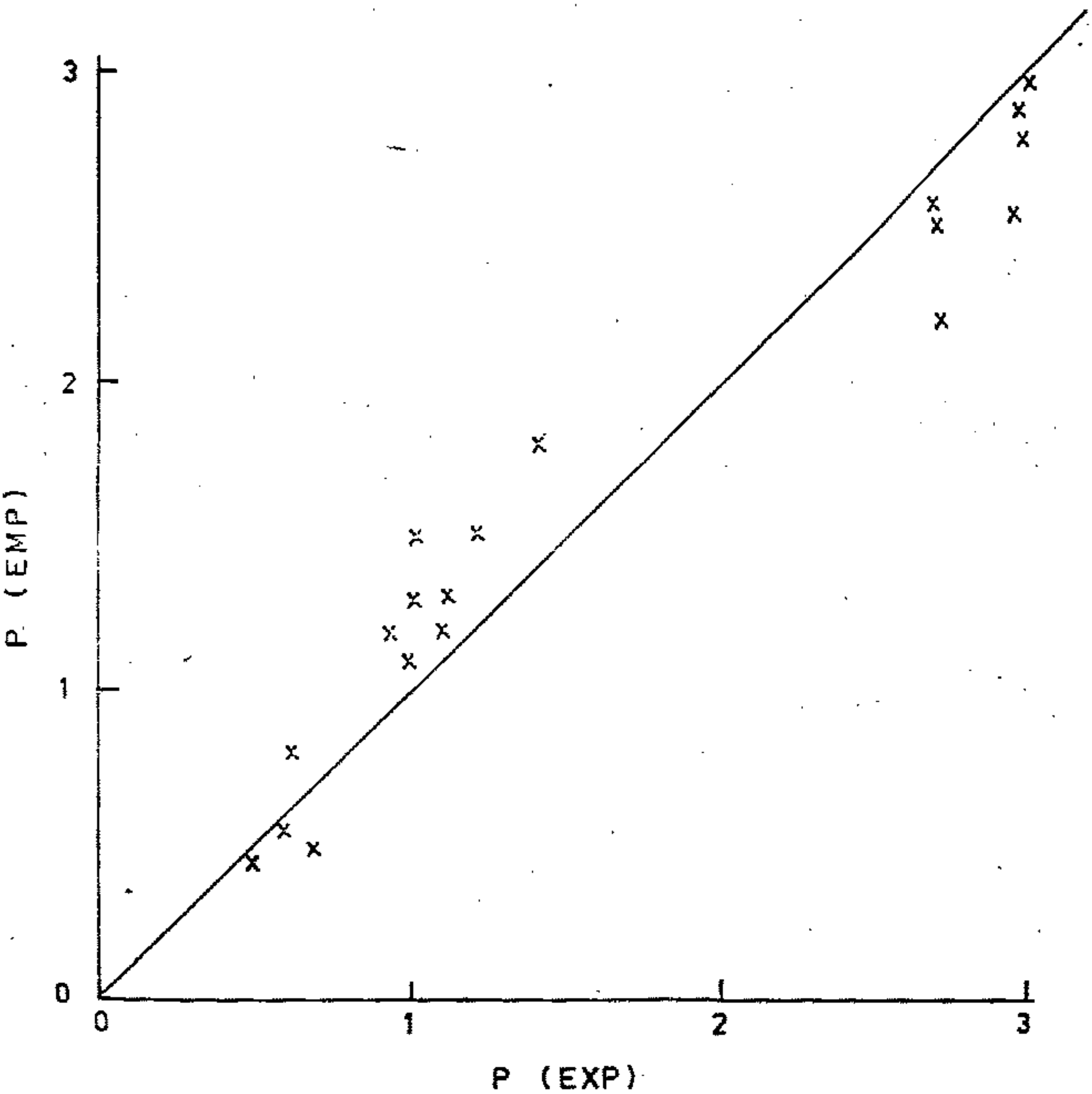


FIG. 4·10 P (EXPERIMENTAL) VS P (EMPIRICAL) comparison to that of the interial forces of rotor should be pronounced. The Fig. 4.11 explains this point.

Abichandani and Sarma (1989) made similar observations using full length blades. Therefore, it can be concluded that whether the blades are full length or staggered, the mass flow rate has no effect on power consumption.

The graphs have also been plotted between the rotor power consumption and other relevant parameters to acquire more insight.

4.2.1 EFFECT OF ROTOR SPEED, V_c

4.2.1.1 The Fig. 4.12 illustrates the effect of rotor speed on power consumption. In case of all fluids, viz., water, skim milk and buffalo whole milk, the power consumption increased with increasing rotor speed essentially due to increased rotor inertia. For example, with six staggered blades, for buffalo whole milk the power consumption at 2.18 m/s was 192 W and increased to 336 W at 4.45 m/s.

The corresponding increase in U_o was from 40 to 80 W/m^2 °K. Thus, power consumption increased by 75 per cent when rotor speed increased by 103 per cent. While the overall heat transfer coefficient, U_o, increased by 100 per cent.

4.2.1.2 It has been reported that during evaporation of buffalo

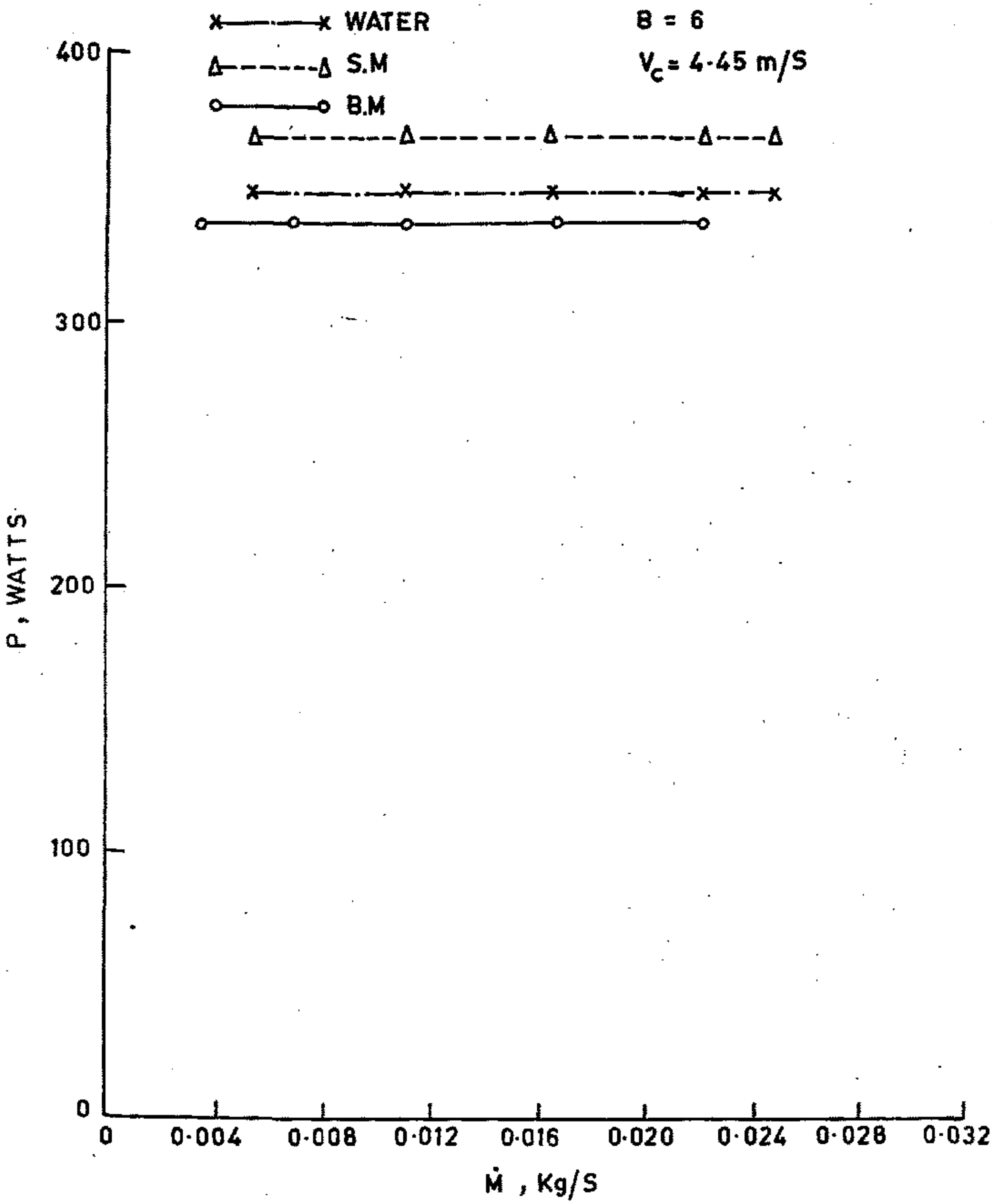


FIG. 4·11 EFFECT OF $\dot{M}$ ON P

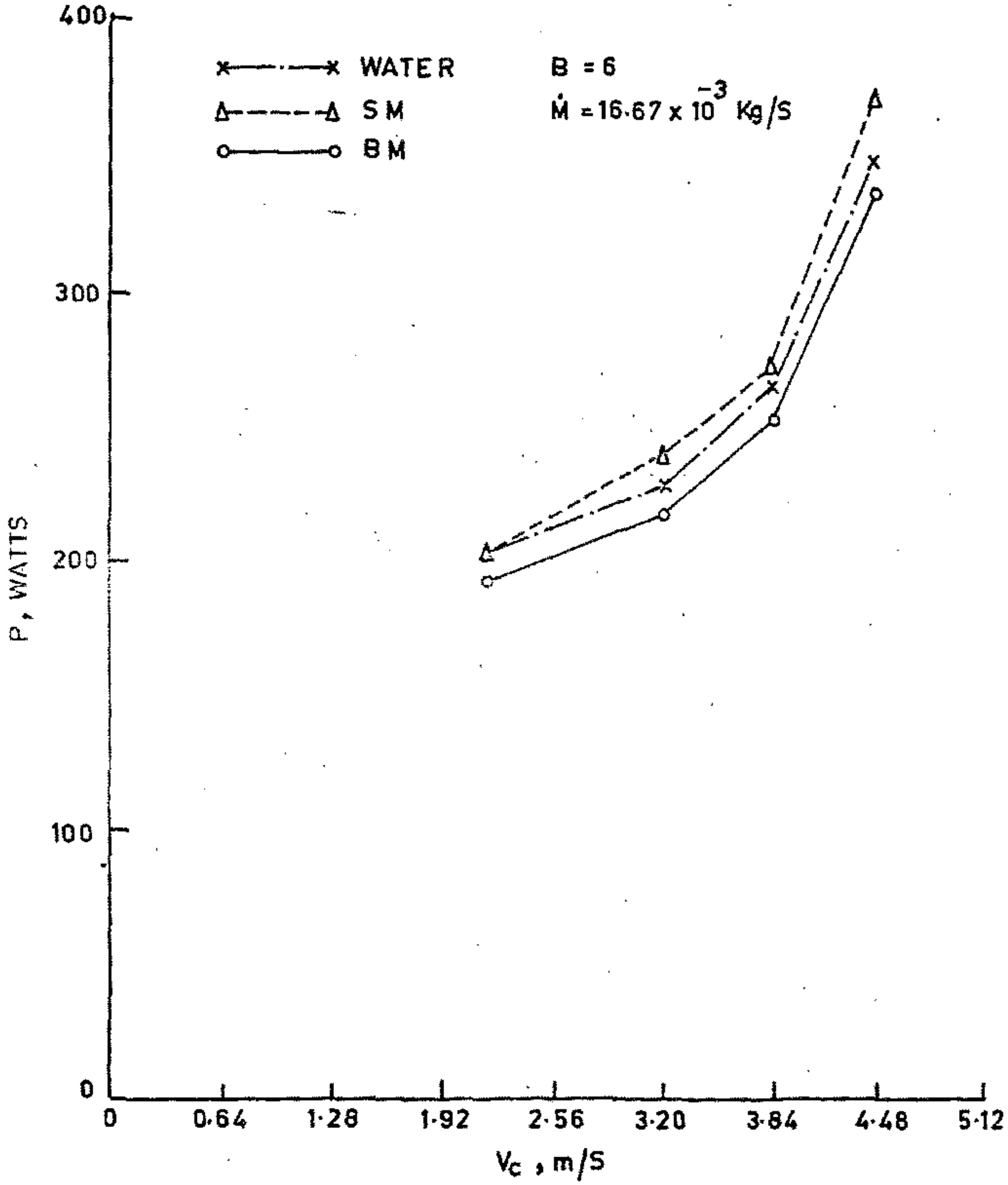


FIG. 4.12 EFFECT OF V_c ON P

whole milk at atmospheric pressure in horizontal thin film SSHE using steam as heating medium and the rotor with six full length blades, when rotor speed was increased by 100 per cent, the power consumption rose by 114 per cent, while Uo increased just by 28 per cent (Abichandani and Sarma, 1989 and 1991).

4.2.1.3 Thus, it can be seen that the use of staggering blades instead of full length blades is highly advantageous from view points of heat transfer and hydrodynamics in horizontal thin film SSHEs. This could be attributed to high turblulance created due to more frequent interception of fluid film by the staggered blades and their overlapping at the centre of the shaft and also due to abrupt change in axial velocity of fluid film when it covers half the effective length of heat exchanger.

4.2.1.4 Dodeja *et al.* (1991) had reported that staggering of blades in similar heat exchanger improved Residence Time Distribution which was the result of interception of inlet-outlet path at higher frequency and better mixing.

4.2.2 EFFECT OF NUMBER OF BLADES, B

4.2.2.1 The Fig. 4.13 shows the effect of the number of blades on the power consumption. Increasing the number of blades resulted in increased power consumption mainly because of increased inertia of rotor. It could be noticed that in case of skim milk, beyond six blades the increase in power consumption was very rapid.

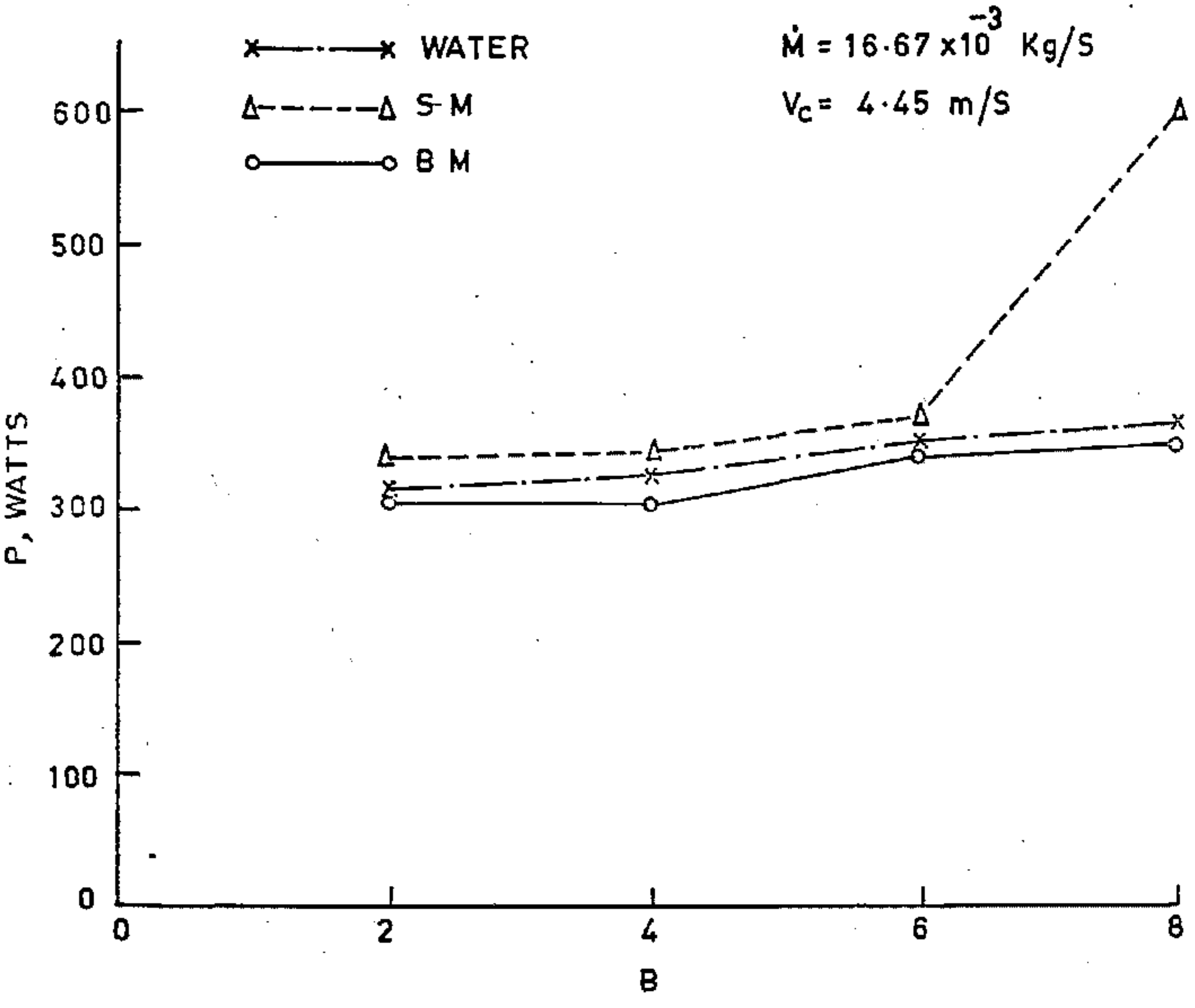


FIG. 4·13 EFFECT OF B ON P

4.2.2.2 The gain in terms of increased values of overall heat transfer coefficient, U_o in relation to increased power consumption was quite significant in case of all the three liquids and specially for buffalo whole milk when the number of blades was increased from two to six. However, beyond six blades, the increase in U_o was just marginal. Therefore, it is of no advantage to operate the heat exchanger with more than six staggered blades. The reasons furnished for increased V_o values with rotor speed should be valid in the present case also.

4.3 THERMAL PERFORMANCE

The coefficient of performance and primary energy ratio are the two parameters which judge the thermal performance of the heat pump system.

4.3.1 PRIMARY ENERGY RATIO (PER)

4.3.1.1 The COP gives a measure of the usefulness of the heat pump unit in producing large amounts of heat from a small amount of work. It does not, however, express the fact that energy available as work is normally more valuable than energy available as heat. This becomes apparent when one tries to decide how best to drive the compressor. If one uses an electric motor, one has to use power which is inefficiently generated; if some form of heat engine is used, one is only able to harness some of the heat available in the fuel as work. Ideally, a heat pump where free work is available should be contemplated, e.g., wind or water power, but this is not always possible.

In order to assess different heat pump systems using compressor drives from different fuel or energy sources, the PER or primary energy ratio is applied. The PER takes into account not only the heat pump COP but also the efficiency of conversion of the primary fuel (e.g., oil, gas, coal or solar heat) into the work, which drives the pump.

The PER is defined as follows:

$$PER = \frac{\text{Useful heat delivered by heat pump}}{\text{Primary energy consumed}}$$

It is often possible to use an alternative definition when a heat engine with thermal efficiency η is used to drive a heat pump compressor. In this case:

$$PER = \eta \times COP$$

When using a heat pump for process heating or in any application where the only benefit is from the supply of heat, then the PER shows how much profit the heat pump will make as compared with a conventional boiler (for hot water or steam heating) or compared with directly fired heating.

4.3.1.2 The present system, that is, an integrated heat pump - thin film SSHE system for concentration of milk operated between average suction pressure of 1.7 bar absolute and 15.5 bar absolute discharge pressure. The COP of heat pump worked out to be 3.5.

Using the electricity generation efficiency of 30 per cent, we obtain for electric drive case,

$$\begin{aligned} PER &= 3.5 \times 0.3 \\ &= 1.05 \end{aligned} \qquad \text{----------} \; (4.5)$$

This can be directly compared with an indirectly fired heating system, using a boiler for which the efficiency (or PER) may be only 0.7 or 0.8. The comparison shows that in this case the heat pump gives about 30 to 50 per cent more heat energy output per unit of fuel consumed.

4.3.1.3 Reay and Macmichael (1979) have described a heat pump prime-moved by diesel engine (40 per cent efficiency) where in 35 per cent of the primary energy was assumed to be recoverable from exhaust gases. The PER of such a system was calculated to be 1.59. When this figure is compared with an indirectly-fired heating system as mentioned above, then the Diesel engine driven heat pump gives twice the heat output per unit of fuel consumed.

The power required to drive a heat pump can come from a variety of different primary sources such as electric motors, internal combustion engines (Petrol and Diesel engines), gas turbines and reciprocating engines. They are either reliable but inefficient (in terms of primary energy) like electric motors or efficient like internal combustion engines but either unreliable (in terms of the duty required of them in this context) or expensive.

The reasons for the widespread use of electric drive are simple: the capital cost is low and the maintenance required is also low. While greater capital outlay may be permitted if lower fuel bills are promised, it is not so easy to sell a system such as a gas engine-driven heat pump with higher maintenance demands and as long as this remains true, the electric motor will maintain its supremacy.

4.4 ENERGY ANALYSIS

In this section, the energy required to evaporate a unit mass of water in conventional triple effect evaporator is considered to that in the integrated heat pump - thin film SSHE system. The sample calculation is shown as under:

4.4.1 CONENTIONAL SYSTEM

(a) Specification of boiler

Capacity	:	2000 kg/h at 10 ata gauge pressure
Fuel oil consumption	:	160 lph (LDO)
Blower motor	:	7.5 HP
Feed water pump motor	:	15.0 HP

(b) Specification of triple effect evaporator with thermo-vapour recompressor (TVR) fitted in second stage

Capacity	:	440 kg/h water evaporation
Steam requirement	:	170 kg/h at 7.0 ata gauge pressure
Feed and product pump motors	:	8 KW
Initial TS content in milk	:	15%
Final TS content in milk	:	40 to 45%

4.4.1.1 Energy required to generate unit mass of steam in boiler

Calorific value of LDO = 43095 kJ/l

$\therefore$ Rate of thermal energy by burning fuel $= 160 \left(\frac{l}{h}\right) \times 43095 \left(\frac{kJ}{l}\right)$

$= 6895.2$ MJ/h

Assuming efficiency of motor pump assembly = 0.75

Electrical energy input for one hour $= 22.5\ (HP) \times 0.75 \left(\frac{KW}{HP}\right) \times 1(h)$

$= 16.875$ kWh

Considering conversion efficiency of thermal energy into electrical energy (PER) as 30 per cent:

Thermal energy equivalent to electrical energy =

$$\frac{16.875\ (kWh)}{0.3} \times 3.6 \times 10^3 \left(\frac{kJ}{kWh}\right)$$

$= 2,02,500$ kJ

$\therefore$ Total thermal energy required to produce

$$1 \text{ kg steam} = \frac{6895200 + 202500}{2000}$$

$= 3550$ kJ

4.4.1.2 Energy input to the triple effect evaporator with TVR system

(i) Steam requirement

The economy of evaporation (mass of steam required to evaporate unit mass of water) in triple effect evaporator with

$$\text{TVR} = \frac{170}{440} = 0.386$$

∴ Thermal energy required to evaporate

1 kg of water = 0.386 x 3550
= 1371 kJ

(ii) Thermal energy equivalent of electrical energy input to feed and product pump motors (assuming PER as 0.3)

$$\text{for one hour} = \frac{8\ (\text{KW})}{0.3} \times 1\ (\text{h}) \times 3.6 \times 10^{3}\ \left(\frac{\text{kJ}}{\text{kWh}}\right)$$

$$= 96{,}000\ \text{kJ}$$

∴ Thermal energy input for 1 kg water

$$\text{evaporation} = \frac{96000}{440}$$

$$= 218.2\ \text{kJ}$$

$$\simeq 218\ \text{kJ}$$

∴ Total thermal energy required to evaporate

1 kg of water = 1371 + 218

= 1589 kJ

4.4.2 INTEGRATED HEAT PUMP - THIN FILM SSHE SYSTEM

4.4.2.1 Optimum operating conditions for concentrating buffalo whole milk

Rotor speed	:	3.42 RPS
Mass flow rate of buffalo whole milk	:	16.67×10^{-3} kg/s

Absolute pressure in the liquid concentrator : 70 mm of Hg

Rate of water evaporation : 1.44 kg/h

Total power consumed by the system : 1 KW

Electrical energy consumed during processing of 5 kg buffalo

$$\text{whole milk} = 1 \times \frac{5}{80}$$

$$= 0.0625 \text{ kWh}$$

Total equivalent thermal energy consumed by the system for evaporation of unit mass of water

$$= \frac{0.0625\ (\text{kWh})}{0.3} \times 3.6 \times 10^3 \left(\frac{\text{kJ}}{\text{kWh}}\right) \div 0.09 \left(\frac{\text{kg}}{\text{h}}\right)$$

$$= 8333.33 \text{ kJ}$$

(assuming conversion efficiency of 30%)

Rate of thermal energy recovered from concentrated milk

$$= \dot{M}.Cp.\Delta T$$

$$= 78.56 \left(\frac{\text{kg}}{\text{h}}\right) \times 3.851 \left(\frac{\text{kJ}}{\text{kg }^\circ\text{C}}\right) \times 25\ (^\circ\text{C})$$

$$= 7563.4 \frac{\text{kJ}}{\text{h}}$$

Rate of thermal energy recovered from condensing vapours in vapour condenser

$$= 1.44 \left(\frac{\text{kg}}{\text{h}}\right) \times 2389 \left(\frac{\text{kJ}}{\text{kg}}\right)$$

$$= 3440.3 \text{ kJ/h}$$

(Latent heat of evaporation at 70 mm of Hg absolute pressure is $2389 \frac{\text{kJ}}{\text{kg}}$).

Total thermal energy recovered from the system

$= 7563.4 + 3440.3$

$= 11003.7$

11004.0 kJ/h

Thermal energy recovered per unit mass of water evaporation

$$= \frac{11004}{1.44}$$

$= 7641.66$ kJ

$\therefore$ Net energy consumed for unit mass of evaporation

$= 8333.33 - 7641.66$

$= 691.67$ kJ

$\simeq 692$ kJ

Thus, it is observed that the energy input to heat pump for evaporating unit mass of water is far less as compared to that in the conventional multiple effect evaporator system.

4.5 ADVANTAGES OF THE INTEGRATED HEAT PUMP THIN FILM SCRAPED SURFACE HEAT EXCHANGER SYSTEM OVER CONVENTIONAL MULTIPLE EFFECT LONG TUBE FALLING FILM EVAPORATOR SYSTEM

4.5.1 Milk is evaporated in single stage horizontal thin film SSHE and, therefore, there are no scaling and foaming problems.

4.5.2 The value of overall heat transfer coefficients remain fairly constant and hence there would be less interruptions in production schedule.

4.5.3 The system is operated by single energy source only, that is, electrical energy and, therefore, no separator boiler room facility is required. Also no hazards of boiler act and regulations.

4.5.4 Practically no cooling water is needed but in the conventional system large quantity of water is required to condense vapour and maintain vacuum. Moreover, the volume of water vapour at low evaporation temperature is so great that it is not economical to compress the vapour directly. Mechanical vapour recompressors (MVR) become so huge that the savings actually made in primary steam use is more than offset by the friction losses, maintenance and carrying charges upon the large investment necessary. But in the heat pump system instead of attempting to compress the high volume water vapour, first the heat is transferred from the water vapour to the boiling refrigerant, thereby converting it to the vapour of low specific volume which is then easily compressed.

4.5.5 The final product leaves the system in chilled condition while in the conventional system the product leaves the last effect at temperature of 45° to 50°C, thereby, requires extra refrigeration equipment.

4.5.6 The design is very compact and hence space requirement is quite less. Also, it does not require multistorey construction.

4.5.7 Relatively less quantities of detergents and sanitizers are required as there is no scale formation.

4.5.8 The system offers an excellent scope for recovery of flavours and other volatile components.

4.5.9 The system can be operated under very low absolute pressures because the vapours are conveniently condensed by the application of refrigeration. Therefore, it is very suitable for concentrating heat labile food products.

4.5.10 The heating medium, that is, condensing refrigerant is at relatively low temperature. Therefore, less corrosion of metal surfaces is expected.

4.5.11 The system is highly energy efficient because it works on the thermodynamic principle of utilizing large quantity of waste/low grade energy after its upgrading by supplying small amount of high grade energy.

5. CONCLUSIONS

5.1 The research efforts in the present study were directed to:

5.1.1 develop an integrated heat pump - thin film scraped surface heat exchanger system.

5.1.2 develop correlations for heat transfer coefficient during evaporation of water, skim milk and buffalo whole milk in horizontal thin film SSHE having a rotor with staggered variable clearance blades.

5.1.3 develop prediction equations for evaluating overall heat transfer coefficients.

5.1.4 develop correlations for power consumption.

5.1.5 evaluate thermal performance of the system.

5.1.6 analyse the system from energy input view point.

5.2 An experimental set up was developed, designed and fabricated, in order to achieve the above objectives.

5.3 The data generated from the experiments were analysed in HCL-4 computer and the following correlations were developed.

5.3.1 EVAPORATION

5.3.1.1 The correlation obtained for the scraped film heat transfer coefficients was of the form:

$$Nu = 0.7959\ (Re_f)^{0.548}\ (\tilde{Re}_R)^{0.082}\ (Pr)^{-0.182}\ (B)^{0.667} \qquad \text{---------(4.1)}$$

$$4.17 \leq Re_f \leq 74$$
$$119379 \leq Re_R \leq 508552$$
$$2.5 \leq Pr \leq 9.36$$
$$2 \leq B \leq 8$$

5.3.1.2 The expression developed for optimizing the operating parameters of the SSHE was in the Box Wilson form:

$$\begin{aligned} U_O = {} & 0.3172 + 23660.02\ (\dot{M}) - 0.621\ (Pr) - 31.992\ (V_c) + 5.133\ (B) \\ & - 714206.94\ (\dot{M})^2 - 1.82\ (Pr)^2 + 6.06\ (V_c)^2 - 0.0865\ (B)^2 \\ & - 250.30\ (\dot{M})\ (Pr) - 913.73\ (\dot{M})\ (V_c) - 159.36\ (\dot{M})\ (B) \\ & + 2.153\ (Pr)\ (V_c) + 0.863\ (Pr)\ (B) \\ & + 0.309\ (V_c)\ (B) \qquad \text{---------(4.3)} \end{aligned}$$

$$3.42\times10^{-3} \leq \dot{M} \leq 25\times10^{-3} \text{ kg/s}$$

$$2.5 \leq Pr \leq 9.36$$

$$2.18 \leq V_c \leq 4.45 \text{ m/s.}$$

5.3.3 POWER REQUIREMENT

The prediction equation for determining power input to the rotor was also in the form of Box Wilson model.

$$P = 2.934 - 1.360\ (V_c) + 0.561\ (B) - 0.1424\ (S)$$

$$+ 0.1311\ (V_c)^2 - 0.0117\ (B)^2 - 0.00318\ (S)^2$$

$$- 0.0726\ (V_c)\ (B) - 0.0068\ (V_c)\ (S)$$

$$- 0.01093\ (B)\ (S) \qquad \text{---------(4.4)}$$

$$2.18 \leq V_c \leq 4.45 \text{ m/s.}$$

$$2 \leq B \leq 8$$

$$0 \leq S \leq 20\%$$

5.4 The engineering design of such horizontal thin film SSHE can be evolved by using above mentioned correlations.

5.5 As the film Reynolds number, Re_f increases the scraped film heat transfer coefficient, h_s increases. In horizontal thin film scraped surface heat exchanger Re_f has greater effect on h_s. Also no matter whether evaporation is carried out at atmospheric pressure or under vacuum, the h_s increases with increasing film Reynolds number.

5.6 The rotational Reynolds number, Re_R has a positive influence on h_s.

5.7 Scraped film heat transfer coefficient increases with increase in number of staggered blades.

5.8 Mass flow rate, $\dot{M}$ has a profound effect on the overall heat transfer coefficient, U_o Under identical mass flow rate conditions, the value of U_o is highest in case of water and lowest in case of buffalo whole milk. Further $\dot{M}$ has more pronounced effect on U_o when the evaporation is carried under vacuum as compared to that when evaporation is done at atmospheric pressure.

5.9 Overall heat transfer coefficient increased with increase in rotor speed. But in particular for skim milk and buffalo whole milk, the increase was appreciable upto rotor speed of 3.83 m/s. In case of water the increase in U_o was gradual.

5.10 The overall heat transfer coefficient, U_o increases with increase in number of blades in case of milk. But in case of water after six blades in staggered position the U_o did not increase significantly. Average increase in U_o was 33 per cent when the number of blades increased from six to eight while the average power consumption rose to 66 per cent. Therefore, it is of no advantage to increase the number of blades beyond six.

5.11 The mass flow rate, $\dot{M}$ has no effect on power consumption whether the blades are full length or staggered.

5.12 The power consumption increased with increasing speeds.

5.13 Increasing the number of blades resulted increased power consumption.

5.14 The gain in terms of increased values of overall heat transfer coefficients, U_o in relation to increased power consumption was quite significant when the number of blades was increased from two to six. However, beyond six blades, the increase in U_o was just marginal.

5.15 The use of staggering blades instead of full length blades is highly advantageous from view point of heat transfer and hydrodynamics in horizontal thin film SSHE.

5.16 Staggering of blades causes high turbulance created due to more frequent interception of fluid film and their overlaping at the centre of the shaft and also because of abrupt change in axial velocity component of fluid film when it covers half the effective length of heat exchanger.

5.17 The coefficient of performance of the heat pump was 3.5. It gave 30 to 50 per cent more heat energy out put when compared to boiler.

5.18 Under optimum operating conditions (Number of blades = 6, Rotor speed = 3.42 RPS, Mass flow rate of buffalo whole milk = 16.67×10^{-3} kg/s), the net savings in thermal energy was found to be 897 kJ/kg for water evaporation when compared with triple effect evaporator with TVR system.

5.19 Process integration is a concept which is very attractive from the point of view of energy saving and other advantages. The integrated heat pump - thin film scraped surface heat exchanger system appears to have an excellent potential for processing of milk and similar food products due to its inbuilt characteristic of maintaining overall heat transfer coefficient. Also it offers a scope of recovery of flavours and other volatile components.

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APPENDIX-1

A.1 PHYSICAL PROPERTIES OF DICHLORODIFLUOROMETHANE (FREON-12)

A.1.1 ABSOLUTE VISCOSITY

Absolute viscosity of F-12 was determined from correlation given below. The correlation is developed for various temperatures, data obtained from Carrier (1965).

$$\mu_R = 0.02916 - 0.0001634\ T + 0.000000723\ T^2$$

$$r^2 = 0.9901$$

where, μ_R = absolute viscosity, Pa.-s.

T = temperature, $^{\circ}C$

The regression equation is fitted within temperature range of 15.6° to $93.3^{\circ}C$.

A.1.2 DENSITY

Density of refrigerant (F-12) was estimated from regression equation given below. The equation is developed for various temperatures taken from ASHRAE (1961) and Jordon and Priester (1973).

$$\rho_R = 1392.4207 - 2.9521\ T - 0.0143\ T^2$$

$$r^2 = 1.0000$$

where, ρ_R = density, kg/m^3

T = temperature, $^\circ C$

The regression equation is fitted within temperature range of 10° to $64.4^\circ C$.

A.1.3 THERMAL CONDUCTIVITY

Thermal conductivity of F-12 had been estimated from correlation given below which was developed for various temperatures given by Perry (1950).

$$K_R = 0.097890243 - 0.00039150412\ T$$

$$r^2 = 0.9992$$

where, K_R = thermal conductivity, W/mk

T = temperature, $^\circ C$

The regression equation is fitted within temperature range of 15.6° to $82.2^\circ C$.

APPENDIX-2

A.2 PHYSICAL PROPERTIES OF BUFFALO WHOLE MILK

A.2.1 ABSOLUTE VISCOSITY

Absolute viscosity of buffalo whole milk was estimated from correlation given below. The correlation is developed from graph between absolute viscosity versus temperature, obtained from Kessler (1981).

$$\mu_{BM} = 0326 \times 10^{-2} - 0.5957 \times 10^{-4}\ T + 0.32143 \times 10^{-6}\ T^2$$

$$r^2 = 0.9977$$

where, μ_{BM} = absolute viscosity, Pa.-s

T = temperature, ^{o}C

The regression equation is fitted within temperature range of 20^{o} to $100^{o}C$.

A.2.2 DENSITY

Density of buffalo whole milk was determined from correlation given below (Agarwala, 1973):

$$\rho_{BM} = 0.9861\ (S)^{0.045} + \frac{0.004}{e^{1.32(55-T)} \cdot e^{1.32(T-55)}}$$

$$- 0.55 \times 10^{-3} \times T$$

where, ρ_{BM} = density, g/cm^3
T = temperature, ^oC
S = average total solids, %

A.2.3 THERMAL CONDUCTIVITY

Thermal conductivity of buffalo whole milk was determined from the following relationship (Fernadez-Martin and Mohtes, 1972):

$$K_{BM} = (b_o + b_1S) + (b_2 + b_3S)\ T + (b_4 + b_5S)\ T^2$$

where, K_{BM} = thermal conductivity, Cal $cm/cm^2\ s^oC \times (10^4)$

b_o = 13.21
b_1 = -0.0768
b_2 = 0.077
b_3 = -0.00135
b_4 = -0.000507
b_5 = 0.0000121
T = temperature, ^oC
S = average total solids, %

A.2.4 SPECIFIC HEAT

Specific heat of buffalo whole milk has been taken constant as 3851.4 J/kg ^oK (Farrel, 1967).

APPENDIX-3

A.3 PHYSICAL PROPERTIES OF SKIM MILK

A.3.1 ABSOLUTE VISCOSITY

Viscosity of skim milk was estimated from correlation given below. The correlation is developed from graph between absolute viscosity versus temperature obtained from Kessler (1981).

$$\mu_{SM} = 0.27 \times 10^{-2} - 0.5046238 \times 10^{-4}\ T + 0.27678516 \times 10^{-6}\ T^2$$

$$r^2 = 0.9926$$

where, μ_{SM} = absolute viscosity, Pa.-s
T = temperature, ^{o}C

The regression equation is fitted within temperature range of 20^{o} to $100^{o}C$.

A.3.2 DENSITY

Density of skim milk was determined from correlation given below (Agarwala, 1973):

$$\rho_{SM} = 0.9861\ (S)^{0.045} + \frac{0.004}{e^{1.32(55-T)} + e^{1.32(T-55)}} - 0.55 \times 10^{-3} \times T$$

where, ρ_{SM} = density, g/cm^3
T = temperature, ^{o}C
S = average total solids, %

A.3.3 THERMAL CONDUCTIVITY

Thermal conductivity of skim milk was determined from the following relationship (Fernadez-Martin and Mohtes, 1972):

$$K_{SM} = (b_o + b_1 S) + (b_2 + b_3 S)\ T + (b_4 + b_5 S)\ T^2$$

where, K_{SM} = thermal conductivity, Cal cm/cm^2 s^{o}C x 10^4

b_o = 13.21
b_1 = -0.0768
b_2 = 0.077
b_3 = -0.00135
b_4 = -0.000507
b_5 = 0.0000121
T = temperature, ^{o}C
S = average total solids, %

A.3.4 SPECIFIC HEAT

Specific heat of skim milk has been taken as 3997.96 J/kg^{o}k (Charm, 1963).

APPENDIX-4

A.4 PHYSICAL PROPERTIES OF WATER

A.4.1 ABSOLUTE VISCOSITY

Absolute viscosity of water was estimated from the correlation given below as cited by Bingham (1922):

$$\frac{10}{\mu_W} = 2.1482 \left[(T-8.435) + \sqrt{8078.4 + (T-8.435)^2}\right] - 120$$

where, μ_W = absolute viscosity, Pa.-s

T = temperature, ^{o}C

A.4.2 DENSITY

Density of water was taken as a constant value equal to:

$$\rho_W \times 10^{-3} = 0.985$$

where, ρ_W = density, kg/m^3

A.4.3 THERMAL CONDUCTIVITY

Thermal conductivity of water was obtained from relationship given below (Abichandani, 1985):

$$K_W = 0.594 + 0.863 \times 10^{-3}\ T$$

where, K_W = thermal conductivity, $\frac{W}{mk}$

T = temperature, ^{o}K

A.4.4 SPECIFIC HEAT

Specific heat of water has been taken as 4.18 kJ/kg^{o}k (Abichandani, 1985).

APPENDIX-5

A.5 MECHANICAL DESIGN OF THIN FILM SSHE (LIQUID CONCENTRATOR)

Since the shell of horizontal thin film SSHE (liquid concentrator) was being subjected to high pressure from outside (in the jacket) and sub-atmospheric pressure from inside. The mechanical design was very important.

A.5.1 DESIGN OF THE SHELL

The shell was designed on the basis of external pressure as suggested by Hesse and Rushton (1964).

The working pressure P is specified to be one-fifth of the collapsing pressure, P_c.

The thickness of pipe was calculated as follows:

$$P_c = 5P = \frac{2\,\delta\,y\,t}{1.05\,D}$$

where,
y = yield point
P_c = collapsing pressure
P = working pressure
D = diameter of pipe

$$P_c = 5P = \frac{2\,\delta\,y\,t}{1.05\,D}$$

$$t = \frac{1.05\,D \times 5P}{2\,\delta\,y}$$

Assuming $D = 8"$

$$= \frac{1.05 \times 8 \times 5 \times 450}{2 \times 27500}$$

$= 0.344"$
$= 8.74 \times 10^{-3}$ m

Say thickness $= 9 \times 10^{-3}$ m
More safe thickness would be $= 1 \times 10^{-2}$ m

Now t/D and L/D ratios were observed for particular pressure from graph for calculation of maximum length.(Hesse and Rushton, 1964)

$$\text{Thus } \left.\begin{aligned} t/D &= 0.04 \\ L/D &= 12.5 \end{aligned}\right] \text{ At } P = 450 \text{ PSI}$$

$t = 0.04 \times 20.32 = 8 \times 10^{-3}$ m
$L = 12.5 \times 20.32 = 2.54$ m

Hence, we can take thickness $= 1 \times 10^{-2}$ m and maximum length $= 2.5$ m.

A.5.2 DESIGN OF JACKET

The design of jacket was based on the equations suggested/ quoted by Harvey (1969). The portion was subjected to internal pressure and thickness of jacket was calculated by the following equations:

$$\sigma_1 = \frac{Pr}{2\,h}$$

$$\sigma_2 = \frac{Pr}{h}$$

where, σ_1 = longitudinal or meridional stress

σ_2 = hoop stress

h = thickness

r = internal radius

P = working pressure

Assuming diameter of pipe = 10"

$$\text{Now,} \qquad h = \frac{Pr}{\sigma_2}$$

$$= \frac{450 \times 5 \times 5}{27500}$$

$$= 0.41''$$

$$= 1.04 \text{ cm}$$

$$h = 1 \times 10^{-2} \text{ m}$$

Hence, the thickness of jacket pipe can be taken as 1×10^{-2}m

A.6 TABLE 1. EXPERIMENTAL FLUID - WATER

β 2

Sl. No.	N	$\dot{M}$	To_c	To_E	TD_R	P_3	P_T	P_S	$\dot{M}_v$	P_d	P_s	U_o	h_o	h_s
	RPS	$\frac{kg}{s} \times 10^{-3}$	°C	°C	°C	mm of Hg	KW	W	$\frac{kg}{s} \times 10^{-6}$	bar	bar	$\frac{W}{m^2k}$	$\frac{W}{m^2k}$	$\frac{W}{m^2k}$
1.	3.42	5.56	48.0	30.0	58.4	85	1.1	204	33.33	14.0	0.75	11.60	192.50	13.01
		11.11	48.2	30.0	58.4	85	1.1	204	66.66	14.0	0.60	23.65	199.14	28.42
		16.67	48.6	30.5	55.2	90	1.1	204	116.66	13.0	0.50	63.87	208.80	99.56
		22.22	49.0	30.7	56.8	90	1.1	204	177.77	13.5	0.50	82.37	206.60	150.45
		25.00	-	-	-	-	-	-	-	-	-	-	-	-
2.	5.00	5.56	49.5	18.5	62.8	90	1.7	216	50.00	15.5	0.90	13.59	185.09	15.47
		11.11	50.0	18.0	62.8	90	1.7	216	122.22	15.5	1.00	34.52	181.39	45.38
		16.67	51.0	15.0	58.4	100	1.7	216	150.00	14.0	0.90	73.11	190.47	129.42
		22.22	51.1	17.0	58.4	100	1.7	216	220.00	14.0	0.90	108.70	190.47	288.70
		25.00	51.3	18.0	62.8	100	1.7	216	375.00	15.5	1.00	117.52	181.39	391.58
3.	6.00	5.56	49.0	23.0	60.0	90	1.9	252	49.50	14.5	0.75	14.57	190.44	16.65
		11.11	49.0	23.0	58.4	90	1.9	252	111.11	14.0	0.75	42.71	192.50	58.66
		16.67	48.1	23.3	56.8	85	1.9	252	183.33	13.5	0.50	76.26	206.50	132.04
		22.22	49.0	22.8	56.8	90	1.9	252	244.44	13.5	0.60	113.26	201.53	296.02
		25.00	49.0	23.5	58.4	90	1.9	252	350.00	14.0	0.50	134.57	204.12	473.56
4.	6.97	5.56	46.3	26.0	55.2	80	2.2	312	55.55	13.0	1.00	22.61	190.90	27.15
		11.11	46.2	26.0	55.2	80	2.2	312	122.22	13.0	1.00	49.20	190.90	71.11
		16.67	46.2	23.0	55.2	80	2.2	312	233.33	13.0	1.00	93.95	190.90	206.48
		22.22	46.0	26.7	55.2	80	2.2	312	290.00	13.0	1.00	114.21	190.90	327.82
		25.00	46.0	26.5	55.2	80	2.2	312	375.00	13.0	1.00	147.20	190.90	862.33

A.6 TABLE 2. EXPERIMENTAL FLUID - WATER

N = 4

Sl. No.	N	$\dot{M}$	T_{o_C}	T_{o_E}	T_{D_R}	P_a	P_T	P_S	$\dot{M}_v$	Pd	Ps	U_o	h_o	h_s
	RPS	$\frac{kg}{s} \times 10^{-3}$	°C	°C	°C	mm of Hg	KW	W	$\frac{kg}{s} \times 10^{-6}$	bar	bar	$\frac{W}{m^2 k}$	$\frac{W}{m^2 k}$	$\frac{W}{m^2 k}$
1.	3.42	5.56	45.0	31.2	58.4	80	1.1	204	66.66	14.0	0.60	18.03	199.14	20.95
		11.11	46.0	31.6	58.4	90	1.1	204	144.44	14.0	0.60	42.21	199.14	57.22
		16.67	46.4	32.0	56.4	90	1.1	204	216.66	13.4	0.60	78.50	202.01	140.59
		22.22	46.4	32.1	56.8	90	1.1	204	311.11	13.5	0.60	108.40	201.53	266.30
		25.00	46.4	32.2	56.8	80	1.1	204	374.99	13.5	0.60	130.65	201.53	442.68
2.	5.00	5.56	49.0	21.2	58.4	90	1.6	216	55.55	14.0	1.00	21.31	186.66	25.46
		11.11	48.1	21.3	58.4	85	1.6	216	133.33	14.0	1.00	47.11	186.66	67.53
		16.67	49.3	21.9	58.4	90	1.6	216	225.00	14.0	1.00	89.36	186.66	190.26
		22.22	48.2	22.0	58.4	90	1.6	216	325.00	14.0	1.00	115.16	186.66	348.66
		25.00	47.6	21.8	58.4	85	1.6	216	400.00	14.0	1.00	134.03	186.66	586.95
3.	6.00	5.56	45.6	27.7	56.8	80	1.9	252	72.22	13.5	0.50	23.36	206.57	27.89
		11.11	45.7	28.5	58.4	80	1.9	252	180.00	14.0	0.60	51.35	199.14	74.27
		16.67	44.4	25.0	56.8	70	1.9	252	311.11	13.5	1.00	91.14	188.90	195.68
		22.22	45.8	26.4	58.4	80	1.9	252	444.44	14.0	0.60	127.82	199.14	423.10
		25.00	45.2	26.0	57.4	75	1.9	252	488.88	13.7	0.50	145.27	203.44	635.02
4.	6.97	5.56	46.1	30.6	55.2	80	2.2	324	66.66	13.0	0.40	26.55	214.40	32.13
		11.11	47.3	30.8	56.8	95	2.2	324	144.44	13.5	0.75	55.00	194.81	82.49
		16.67	45.8	23.8	55.2	75	2.2	324	263.33	13.0	0.90	101.56	194.79	238.75
		22.22	46.8	26.0	56.8	80	2.2	324	361.11	13.5	0.90	130.85	192.76	490.71
		25.00	47.4	28.0	58.4	85	2.2	324	464.44	14.0	0.75	152.79	192.50	1005.10

A.6 TABLE 3. EXPERIMENTAL FLUID - WATER

B - 6

Sl. No.	N (RPS)	$\dot{M}$ ($\frac{kg}{s} \times 10^{-3}$)	To_c (°C)	To_R (°C)	TD_R (°C)	P_a (mm of Hg)	P_T (KW)	P_S (W)	$\dot{M}_v$ ($\frac{kg}{s} \times 10^{-6}$)	Pd (bar)	Ps (bar)	U_o ($\frac{W}{m^2k}$)	h_o ($\frac{W}{m^2k}$)	h_s ($\frac{W}{m^2k}$)
1.	3.42	5.56	51.3	29.0	61.4	100	1.1	204	72.22	15.0	1.00	25.80	183.01	31.84
		11.11	49.3	28.0	59.2	85	1.1	204	155.55	14.2	1.00	56.85	185.80	88.33
		16.67	49.0	20.6	58.4	85	1.1	204	233.33	14.0	0.80	89.83	194.61	184.90
		22.22	48.5	23.0	58.4	85	1.1	204	300.00	14.0	1.25	109.66	179.87	323.60
		25.00	46.0	26.0	59.2	80	1.1	204	466.66	14.2	1.25	121.65	179.04	447.58
2.	5.00	5.56	50.0	26.5	60.0	90	1.8	228	72.22	14.5	1.45	26.10	174.92	32.53
		11.11	50.3	24.0	60.0	90	1.8	228	166.66	14.5	1.25	62.11	177.94	103.34
		16.67	46.0	19.1	56.8	80	1.8	228	305.00	13.5	1.50	102.33	177.48	274.67
		22.22	48.0	21.1	58.4	85	1.8	228	349.99	14.0	1.25	117.43	179.89	397.52
		25.00	48.0	22.0	60.0	85	1.8	228	450.00	14.0	1.10	135.72	189.89	588.36
3.	6.00	5.56	53.0	23.0	61.4	110	2.0	264	77.77	15.0	1.00	33.35	183.01	43.39
		11.11	52.5	23.1	61.4	85	2.0	264	166.66	15.0	1.30	68.53	176.35	122.07
		16.67	46.0	20.0	55.2	80	2.0	264	300.00	13.0	1.00	118.16	190.90	360.82
		22.22	46.1	20.6	56.8	80	2.0	264	366.66	13.5	1.10	124.17	187.09	438.66
		25.00	46.2	21.3	60.0	80	2.0	264	510.00	14.5	1.25	133.91	177.94	684.91
4.	6.97	5.56	48.8	27.1	57.6	85	2.5	348	83.33	13.7	1.20	34.26	180.96	44.98
		11.11	48.4	23.2	56.8	85	2.5	348	166.66	13.5	1.10	71.79	187.09	126.97
		16.67	44.3	13.5	53.6	70	2.5	348	333.33	12.5	1.00	130.19	192.60	482.93
		22.22	44.1	14.3	54.4	70	2.5	348	425.00	12.7	1.00	149.88	191.75	919.37
		25.00	43.6	15.6	56.0	70	2.5	348	599.99	13.2	1.00	160.25	190.90	1528.12

A. 6 TABLE 4. EXPERIMENTAL FLUID - WATER

R R

Sl. No.	N	$\dot{M}$	To_C	To_E	To_R	P_a	P_T	P_S	$\dot{M}_V$	Pd	Ps	U_o	h_o	h_s
	RPS	$\frac{kg}{s} \times 10^{-3}$	°C	°C	°C	mm of Hg	KW	W	$\frac{kg}{s} \times 10^{-6}$	bar	bar	$\frac{W}{m^2k}$	$\frac{W}{m^2k}$	$\frac{W}{m^2k}$
1.	3.42	5.56	50.0	28.3	60.0	90	1.1	204	72.22	14.5	0.80	26.10	192.53	32.01
		11.11	50.2	26.1	60.0	90	1.1	204	166.66	14.5	1.00	61.47	184.66	99.69
		16.67	47.5	20.5	56.8	85	1.1	204	250.00	13.5	1.50	97.28	177.48	242.47
		22.22	48.7	22.0	58.4	100	1.1	204	299.99	14.0	1.40	111.55	176.82	350.66
		25.00	46.4	22.7	60.0	85	1.1	204	466.66	14.5	1.25	124.17	177.94	495.56
2.	5.00	5.56	45.3	30.0	55.2	75	1.8	240	72.22	13.5	0.70	26.64	196.87	32.67
		11.11	45.6	30.1	55.2	75	1.8	240	166.66	13.5	0.80	62.96	199.03	99.70
		16.67	44.6	27.6	55.2	70	1.8	240	333.33	13.0	1.00	114.23	190.90	328.00
		22.22	45.0	27.1	55.2	70	1.8	240	350.00	13.0	1.00	124.64	190.90	425.10
		25.00	43.9	28.4	55.9	70	1.8	240	469.99	13.2	0.80	142.27	197.89	632.23
3.	6.00	5.56	43.1	28.0	51.9	70	2.2	300	77.77	12.0	0.00	33.15	245.54	42.09
		11.11	44.8	27.0	53.2	70	2.2	300	177.77	12.4	0.00	76.88	243.54	122.36
		16.67	44.3	21.5	53.6	70	2.2	300	333.33	12.5	0.20	130.19	226.16	356.58
		22.22	43.4	21.5	53.6	70	2.2	300	411.11	12.5	0.00	146.41	242.66	438.53
		25.00	40.2	21.4	53.6	70	2.2	300	611.11	12.5	0.00	165.66	242.66	655.97
4.	6.97	5.56	49.9	28.6	58.4	80	2.6	360	77.77	14.0	0.50	33.15	204.12	42.09
		11.11	51.0	27.4	60.0	85	2.6	360	177.77	14.5	0.80	71.49	192.53	123.91
		16.67	46.0	18.8	55.2	80	2.6	360	333.33	13.5	1.40	131.29	180.83	592.59
		22.22	49.3	18.5	60.0	85	2.6	360	421.11	14.5	1.25	142.42	177.94	966.17
		25.00	46.2	21.1	60.0	80	2.6	360	588.88	14.5	1.20	154.64	177.94	1970.69

A.7 TABLE 5. EXPERIMENTAL FLUID - SKIM MILK

H 2

Sl. No.	N	$\dot{M}$	S_I	S_O	To_C	To_E	TD_R	P_a	P_T	P_S	$\dot{M}_V$	Pd	Ps	U_O	h_O	h_S
	RPS	$\frac{kg}{S} \times 10^{-3}$	%	%	°C	°C	°C	mm of Hg	KW	W	$\frac{kg}{S} \times 10^{-6}$	bar	bar	$\frac{W}{m^2 k}$	$\frac{W}{m^2 k}$	$\frac{W}{m^2 k}$
1.	3.42	5.56	9.44	9.89	48.0	30.0	61.4	85	1.0	192	25.00	15.0	0.75	6.75	188.13	7.37
		11.11	9.44	9.90	48.1	30.0	60.0	85	1.0	192	51.11	14.5	0.50	15.54	201.94	17.77
		16.67	9.44	9.94	49.0	30.0	58.4	85	1.0	192	83.33	14.0	0.75	32.08	192.50	40.92
		22.22	9.44	10.24	49.3	31.8	58.4	85	1.0	192	177.78	14.0	0.75	70.70	192.50	121.68
		25.00	9.44	10.39	49.0	31.6	60.0	85	1.0	192	237.50	14.5	0.50	78.13	201.94	139.50
2.	5.00	5.56	7.11	7.58	43.0	29.8	55.2	65	2.0	228	26.11	13.0	0.50	7.77	208.75	8.50
		11.11	7.11	7.59	45.8	30.0	55.2	75	2.0	228	53.33	13.0	0.50	20.57	208.75	24.14
		16.67	7.11	7.60	47.0	30.0	55.2	80	2.0	228	81.66	13.0	0.60	36.09	203.66	46.72
		22.22	7.11	7.91	47.0	31.0	55.2	80	2.0	228	177.78	13.0	0.50	78.56	208.75	137.82
		25.00	7.11	8.11	46.5	31.0	55.2	80	2.0	228	250.00	13.0	0.50	104.13	208.75	233.70
3.	6.00	5.56	8.75	9.11	49.0	25.0	56.8	85	2.0	252	20.00	13.5	1.00	9.28	188.90	10.28
		11.11	8.75	9.11	51.4	27.0	56.8	100	2.0	252	40.00	13.5	1.00	26.72	188.90	33.01
		16.67	8.75	9.09	51.4	27.0	55.2	100	2.0	252	56.66	13.0	1.00	53.79	190.90	80.53
		22.22	8.75	9.25	51.7	26.4	56.8	100	2.0	252	111.11	13.5	1.00	78.58	188.90	147.52
		25.00	8.75	9.30	52.0	26.6	56.8	100	2.0	252	137.50	13.5	0.75	103.33	194.81	248.20
4.	6.97	5.56	9.12	9.45	46.7	35.5	53.6	80	2.4	336	18.33	12.5	0.60	9.63	205.48	10.64
		11.11	9.12	9.48	48.7	36.2	53.6	85	2.4	336	40.00	12.5	0.50	25.90	210.62	31.30
		16.67	9.12	9.50	49.5	36.5	53.6	90	2.4	336	63.33	12.5	0.50	55.90	210.62	81.86
		22.22	9.12	9.75	46.7	36.5	51.9	80	2.4	336	140.00	12.0	0.25	97.11	225.32	189.37
		25.00	-	-	-	-	-	-	-	-	-	-	-	-	-	-

A.7 TABLE 6. EXPERIMENTAL FLUID - SKIM MILK

B - 4

Sl. No.	N (RPS)	$\dot{M}$ ($\frac{kg}{s} \times 10^{-3}$)	S_I (%)	S_O (%)	To_C (°C)	To_R (°C)	TD_R (°C)	P_v (mm of Hg)	P_T (KW)	P_S (W)	$\dot{M}_v$ ($\frac{kg}{s} \times 10^{-6}$)	Pd (bar)	Ps (bar)	U_o ($\frac{W}{m^2k}$)	h_o ($\frac{W}{m^2k}$)	h_s ($\frac{W}{m^2k}$)
1.	3.42	5.56	9.60	10.05	46.9	33.0	60.0	80	1.2	204	25.00	14.5	0.50	6.91	201.94	7.52
		11.11	9.60	10.05	40.7	33.1	60.0	90	1.2	204	50.00	14.5	0.75	15.98	190.44	18.42
		16.67	9.60	10.20	49.4	33.0	60.0	90	1.2	204	100.00	14.5	0.80	34.10	192.53	44.10
		22.22	9.60	10.60	49.3	33.2	60.0	90	1.2	204	222.22	14.5	0.90	75.06	188.43	136.34
		25.00	9.60	10.60	48.6	33.0	58.4	90	1.2	204	250.00	14.0	0.75	92.21	192.50	197.01
2.	5.00	5.56	9.70	10.07	46.3	29.1	55.2	80	1.8	240	20.55	13.0	0.75	8.35	196.87	9.18
		11.11	9.70	10.17	46.8	29.0	55.2	80	1.8	240	52.22	13.0	0.70	22.53	196.87	26.93
		16.67	9.70	10.05	49.3	29.5	53.6	90	1.8	240	58.33	12.5	0.60	49.03	205.48	69.02
		22.22	9.70	10.15	49.3	30.0	53.6	90	1.8	240	100.00	12.5	0.60	84.05	205.48	156.36
		25.00	9.70	10.52	48.0	29.6	55.2	85	1.8	240	205.00	13.0	0.70	103.03	196.87	243.53
3.	6.00	5.56	10.13	10.63	36.7	28.7	48.5	50	2.1	264	27.78	11.0	1.40	8.60	188.47	9.49
		11.11	10.13	11.13	37.1	28.0	49.4	45	2.1	264	111.11	11.2	1.40	33.08	187.30	42.72
		16.67	10.13	11.43	39.8	28.0	51.9	55	2.1	264	216.66	12.0	0.50	65.34	213.12	101.96
		22.22	10.13	11.23	39.7	28.0	50.3	55	2.1	264	266.66	11.5	0.50	91.79	214.97	168.20
		25.00	10.13	11.43	39.2	27.1	50.3	55	2.1	264	325.00	11.5	0.50	106.84	184.69	289.64
4.	6.97	5.56	9.60	10.10	38.6	30.4	48.5	50	2.6	348	27.78	11.0	0.00	10.25	250.61	11.26
		11.11	9.60	10.60	38.7	30.4	48.5	50	2.6	348	111.11	11.0	0.00	41.45	250.61	52.97
		16.67	9.60	10.60	43.7	30.4	51.9	70	2.6	348	166.66	12.0	0.10	73.83	236.61	116.62
		22.22	9.60	10.60	41.8	30.4	50.3	60	2.6	348	222.22	11.5	0.00	95.75	247.68	170.83
		25.00	9.60	10.80	40.7	30.4	48.5	60	2.6	348	300.00	11.0	0.00	140.13	250.61	371.57

A.7 TABLE 7. EXPERIMENTAL FLUID - SKIM MILK

B = 6

Sl. No.	N (RPS)	$\dot{M}$ ($\frac{kg}{s} \times 10^{-3}$)	S_I (%)	S_o (%)	To_C (°C)	To_R (°C)	TD_R (°C)	P_a (mm of Hg)	P_T (KW)	P_S (W)	$\dot{M}_v$ ($\frac{kg}{s} \times 10^{-6}$)	Pd (bar)	Ps (bar)	U_o ($\frac{W}{m^2 k}$)	h_o ($\frac{W}{m^2 k}$)	h_s ($\frac{W}{m^2 k}$)
1.	3.42	5.56	8.52	9.23	46.6	30.0	58.4	85	1.1	204	39.45	14.0	0.20	12.10	219.18	13.50
		11.11	8.52	9.27	47.0	29.0	60.0	80	1.1	204	83.00	14.5	0.50	23.14	201.94	27.67
		16.67	8.52	9.52	46.6	26.4	58.4	80	1.1	204	166.67	14.0	0.50	50.80	204.12	72.59
		22.22	8.52	9.80	47.2	26.0	50.4	80	1.1	204	204.44	14.0	0.20	87.34	219.18	159.76
		25.00	8.52	9.72	48.0	27.0	58.4	85	1.1	204	300.00	14.0	0.20	104.39	219.18	233.30
2.	5.00	5.56	8.99	9.90	51.0	30.4	62.8	100	1.8	240	50.56	15.5	0.20	15.44	212.99	17.57
		11.11	8.99	10.00	54.3	31.0	62.8	110	1.8	240	112.22	15.5	0.40	47.54	203.72	66.43
		16.67	8.99	10.10	52.8	26.7	61.4	105	1.8	240	185.00	15.0	0.80	77.51	190.81	142.93
		22.22	8.99	10.29	55.1	27.0	62.8	120	1.8	240	288.88	15.5	0.50	134.78	198.35	508.93
		25.00	8.99	10.69	55.0	27.0	62.8	105	1.8	240	425.00	15.5	0.50	152.67	198.35	879.62
3.	6.00	5.56	9.41	10.51	49.4	31.0	58.4	90	2.1	276	61.62	14.0	0.20	24.55	219.18	29.29
		11.11	9.41	10.51	53.0	31.0	60.0	110	2.1	276	122.22	14.5	0.40	62.86	207.40	97.47
		16.67	9.41	10.61	51.8	28.0	60.0	100	2.1	276	200.00	14.5	0.75	87.98	168.19	205.64
		22.22	9.41	11.21	51.0	29.0	61.4	100	2.1	276	400.00	15.0	0.50	138.73	200.13	533.59
		25.00	9.41	11.21	51.0	31.0	61.4	100	2.1	276	450.00	15.0	0.50	156.08	200.13	958.53
4.	6.97	5.56	9.08	10.08	51.0	30.0	58.4	100	2.7	372	55.56	14.0	0.50	27.09	204.12	33.13
		11.11	9.08	10.42	51.3	30.0	60.0	100	2.7	372	148.88	14.5	1.00	61.73	184.66	100.29
		16.67	9.08	10.48	52.0	26.0	60.0	100	2.7	372	233.33	14.5	0.50	105.20	201.94	247.70
		22.22	9.08	10.68	51.3	27.0	60.0	100	2.7	372	355.55	14.5	0.75	147.42	190.44	862.38
		25.00	9.08	10.78	51.0	27.1	60.0	90	2.7	372	425.00	14.7	1.00	153.61	184.66	1346.42

A.7 TABLE 8. EXPERIMENTAL FLUID - SKIM MILK

B - 8

Sl. No.	N	$\dot{M}$	S_I	S_o	To_c	To_E	TD_R	P_a	P_T	P_S	$\dot{M}_v$	Pd	Ps	U_o	h_o	h_s
	RPS	$\frac{kg}{s} \times 10^{-3}$	%	%	°C	°C	°C	mm of Hg	KW	W	$\frac{kg}{s} \times 10^{-6}$	bar	bar	$\frac{W}{m^2 k}$	$\frac{W}{m^2 k}$	$\frac{W}{m^2 k}$
1.	3.42	5.56	7.69	8.84	46.5	34.1	55.2	80	1.8	240	63.89	13.0	0.00	26.62	240.52	31.73
		11.11	7.69	8.26	47.5	33.0	55.2	80	1.8	240	63.74	13.0	0.00	30.00	240.52	36.38
		16.67	7.69	8.59	49.0	30.5	56.8	90	1.8	240	150.00	13.5	0.30	69.50	215.19	111.39
		22.22	7.69	9.39	48.0	32.0	56.8	85	1.8	240	283.34	13.5	0.20	131.30	222.88	396.39
		25.00	7.69	9.59	46.5	33.0	56.8	80	1.8	240	475.00	13.5	0.00	167.10	239.14	705.65
2.	5.00	5.56	9.04	10.10	51.3	33.0	59.2	100	2.3	300	58.89	14.2	1.00	26.89	185.80	33.35
		11.11	9.04	10.30	51.5	33.0	60.0	100	2.3	300	140.00	14.5	0.80	59.41	192.53	92.77
		16.67	9.04	10.35	49.0	30.0	56.8	90	2.3	300	218.33	13.5	0.00	101.16	238.00	195.71
		22.22	9.04	10.60	50.0	31.0	60.0	90	2.3	300	346.66	14.5	0.70	130.52	190.94	497.71
		25.00	9.04	10.94	49.0	31.7	60.0	90	2.3	300	475.00	14.5	0.70	156.07	190.94	1222.57
3.	6.00	5.56	7.71	8.81	46.6	33.0	53.6	90	2.8	384	61.12	12.5	0.00	31.65	271.83	38.05
		11.11	7.71	9.01	47.6	33.0	55.2	80	2.8	384	144.45	13.0	0.00	68.86	240.52	104.54
		16.67	7.71	8.91	49.6	32.5	56.8	90	2.8	384	200.00	13.5	0.50	100.39	206.57	218.50
		22.22	7.71	9.11	49.7	32.5	56.8	90	2.8	384	311.11	13.5	0.30	158.37	215.19	812.20
		25.00	7.71	9.51	46.5	33.0	55.2	80	2.8	384	450.00	13.0	0.20	187.43	224.16	1876.89
4.	6.97	5.56	9.47	10.67	45.3	32.0	51.9	70	3.3	600	66.67	12.0	0.00	36.69	245.54	45.93
		11.11	9.47	10.47	46.8	31.4	51.9	80	3.3	600	111.11	12.0	0.00	78.94	245.54	126.80
		16.67	9.47	10.77	46.6	31.2	51.9	80	3.3	600	216.66	12.0	0.00	148.13	245.54	444.27
		22.22	9.47	11.47	45.3	31.5	54.4	70	3.3	600	444.44	12.7	0.00	177.41	244.17	856.19
		25.00	9.47	11.57	45.3	31.2	54.4	70	3.3	600	525.00	12.7	0.00	209.57	244.17	2905.54

A.8 TABLE 9. EXPERIMENTAL FLUID - BUFFALO MILK

n = 2

Sl. No.	N	$\dot{M}$	S_I	S_o	To_c	To_E	TD_E	P_v	P_T	P_g	$\dot{M}_v$	P_d	P_s	U_o	h_o	h_s
	RPS	$\frac{kg}{s} \times 10^{-3}$	%	%	°C	°C	°C	mm of Hg	KW	W	$\frac{kg}{s} \times 10^{-6}$	bar	bar	$\frac{W}{m^2 k}$	$\frac{W}{m^2 k}$	$\frac{W}{m^2 k}$
1.	3.42	3.47	17.7	18.2	51.0	29.5	62.8	100	1.0	192	19.10	15.5	0.25	5.84	209.71	6.32
		6.94	18.0	18.3	51.0	30.0	61.4	100	1.0	192	20.83	15.0	0.25	7.22	211.59	7.86
		11.11	15.8	16.0	53.6	31.0	60.0	110	1.0	192	22.22	14.5	0.50	12.50	201.94	14.04
		16.67	18.5	18.8	51.0	31.5	60.0	100	1.0	192	50.00	14.5	0.30	20.03	213.50	23.37
		22.22	-	-	-	-	-	-	-	-	-	-	-	-	-	-
2.	5.00	3.47	18.5	19.1	48.7	28.9	60.0	90	1.5	204	20.83	14.5	0.25	6.66	213.50	7.23
		6.94	18.5	19.0	49.0	29.0	61.4	90	1.5	204	34.72	15.0	0.25	10.12	211.59	11.19
		11.31	18.5	19.4	49.0	30.0	56.8	90	1.5	204	100.00	13.5	0.50	19.45	206.57	22.69
		16.67	18.5	19.4	48.9	30.0	58.4	90	1.5	204	150.00	14.0	0.50	23.95	204.12	28.75
		22.22	18.5	19.3	48.9	30.2	61.4	90	1.5	204	177.77	15.0	0.25	51.81	211.59	73.63
3.	6.00	3.47	18.5	18.9	50.0	28.0	56.8	125	1.8	240	13.88	13.5	1.25	7.33	182.03	8.04
		6.94	18.5	18.8	51.6	34.5	56.8	100	1.8	240	20.83	13.5	0.75	14.45	194.81	16.46
		11.11	18.5	19.0	46.0	35.5	55.2	75	1.8	240	55.55	13.0	0.25	21.88	220.71	25.70
		16.67	18.5	19.0	47.0	37.0	55.2	80	1.8	240	83.33	13.0	0.75	36.82	196.87	48.24
		22.22	-	-	-	-	-	-	-	-	-	-	-	-	-	-
4.	6.97	3.47	18.5	19.3	54.0	33.6	64.2	115	2.3	312	127.77	16.0	1.00	7.86	179.73	8.65
		6.94	18.5	19.3	54.0	34.0	64.2	115	2.3	312	55.55	16.0	1.00	16.37	179.73	19.01
		11.11	18.5	19.1	53.0	32.0	60.0	110	2.3	312	61.11	14.5	1.00	23.16	184.66	28.03
		16.67	18.5	19.2	53.1	32.0	60.0	110	2.3	312	116.66	14.5	0.75	42.00	190.44	57.55
		22.22	18.5	19.3	50.2	34.0	62.8	110	2.3	312	177.77	15.5	1.00	50.73	183.01	75.35

A.8 TABLE 10. EXPERIMENTAL FLUID - BUFFALO MILK

B - 4

Sl. No.	N	$\dot{M}$	S_i	S_o	To_c	T_{og}	TO_H	P_v	P_T	P_S	$\dot{M}_v$	P_d	P_s	U_o	h_o	h_s
	RPS	$\frac{kg}{s} \times 10^{-3}$	%	%	°C	°C	°C	mm of Hg	KW	W	$\frac{kg}{s} \times 10^{-6}$	bar	bar	$\frac{W}{m^2 k}$	$\frac{W}{m^2 k}$	$\frac{W}{m^2 k}$
1.	3.42	3.47	15.8	16.5	47.9	28.0	60.0	85	1.0	192	24.30	14.5	0.50	6.28	201.94	6.80
		6.94	15.8	16.2	50.1	28.2	60.0	90	1.0	192	27.77	14.5	0.50	8.36	201.94	9.10
		11.11	15.8	16.1	51.7	28.0	60.0	100	1.0	192	38.88	14.5	0.25	14.03	213.50	15.70
		16.67	15.8	16.2	51.0	28.1	60.0	100	1.0	192	66.66	14.5	0.25	21.86	213.50	25.77
		22.22	15.8	16.4	49.5	27.9	61.4	100	1.0	192	133.33	15.0	0.25	40.41	211.59	53.28
2.	5.00	3.47	18.1	18.9	50.2	30.0	64.2	90	1.5	204	27.77	16.0	1.00	7.17	179.73	8.86
		6.94	18.1	18.7	49.6	28.1	62.8	90	1.5	204	41.66	15.5	1.00	11.40	181.39	12.82
		11.11	18.1	18.7	46.0	24.7	58.4	75	1.5	204	66.66	14.0	1.00	19.49	186.66	23.00
		16.67	18.1	18.7	48.9	26.0	60.0	90	1.5	204	100.00	14.5	1.00	29.23	184.66	36.86
		22.22	18.1	19.0	49.7	28.0	61.4	90	1.5	204	200.00	15.0	0.90	61.78	186.75	99.83
3.	6.00	3.47	18.0	18.8	51.8	26.4	64.2	100	1.8	240	27.77	16.0	1.00	8.08	179.73	8.90
		6.94	18.0	18.9	51.4	27.3	62.8	100	1.8	240	62.50	15.5	1.00	17.11	181.39	19.95
		11.11	18.0	18.9	45.9	24.0	58.4	75	1.8	240	100.00	14.0	1.00	29.23	186.75	36.78
		16.67	18.0	18.8	52.0	26.4	62.4	100	1.8	240	133.33	15.5	1.00	43.41	181.39	61.01
		22.22	18.0	19.0	49.4	25.5	61.4	90	1.8	240	222.22	15.0	1.00	68.65	183.01	119.48
4.	6.97	3.47	18.1	18.7	53.6	31.8	62.8	110	2.3	312	20.83	15.5	0.70	8.15	187.06	8.97
		6.94	18.1	18.8	53.4	32.0	61.9	110	2.3	312	48.61	15.2	0.90	20.59	186.27	24.30
		11.11	18.1	18.7	53.3	32.0	60.0	110	2.3	312	66.66	14.5	0.40	35.83	188.43	47.11
		16.67	18.1	18.8	53.0	33.3	61.4	110	2.3	312	116.66	15.5	1.00	50.01	183.01	73.85
		22.22	18.1	19.3	50.0	34.0	62.8	100	2.3	312	266.66	15.5	0.75	75.01	188.73	136.04

A.8 TABLE 11. EXPERIMENTAL FLUID - BUFFALO MILK

Sl. No.	N (RPS)	$\dot{M}$ ($\frac{kg}{s} \times 10^{-3}$)	S_I (%)	S_o (%)	T_{o_C} (°C)	T_{o_R} (°C)	T_{D_R} (°C)	P_a (mm of Hg)	P_T (KW)	P_S (W)	$\dot{M}_v$ ($\frac{kg}{s} \times 10^{-6}$)	P_d (bar)	P_s (bar)	U_o ($\frac{W}{m^2 k}$)	h_o ($\frac{W}{m^2 k}$)	h_s ($\frac{W}{m^2 k}$)
1.	3.42	3.47	18.2	19.1	44.2	23.0	60.0	70	1.0	192	31.25	14.5	1.00	6.75	184.66	7.37
		6.94	18.2	18.8	44.0	23.0	56.8	70	1.0	192	41.66	13.5	0.90	11.82	192.76	13.27
		11.11	18.2	18.9	44.1	21.5	61.4	70	1.0	192	77.77	15.0	1.25	16.33	176.35	19.09
		16.67	18.2	19.4	44.0	19.0	62.8	70	1.0	192	200.00	15.5	1.25	39.92	174.78	55.22
		22.22	18.2	20.0	43.9	23.5	64.2	70	1.0	192	400.00	16.0	1.25	72.29	173.19	135.58
2.	5.00	3.47	16.4	17.2	52.7	31.7	64.2	100	1.7	216	27.77	16.0	1.00	8.71	179.73	9.63
		6.94	16.4	16.9	53.1	31.0	62.8	110	1.7	216	34.72	15.5	0.75	12.89	187.06	14.60
		11.11	16.4	17.0	51.0	23.7	61.4	100	1.7	216	66.66	15.0	1.25	23.12	176.35	28.17
		16.67	16.4	17.1	53.6	24.8	64.2	110	1.7	216	116.66	16.0	1.25	39.63	173.19	75.84
		22.22	16.4	18.1	50.4	29.4	64.2	110	1.7	216	377.77	16.0	1.00	98.57	179.73	246.11
3.	6.00	3.47	17.3	18.2	40.0	19.5	51.9	55	2.0	252	31.25	12.0	0.00	9.58	245.54	10.50
		6.94	17.3	18.2	40.0	19.0	51.9	55	2.0	252	62.50	12.0	0.10	19.16	236.61	22.03
		11.11	17.3	18.4	40.0	17.0	51.9	55	2.0	252	122.22	12.0	0.20	37.47	228.84	54.77
		16.67	17.3	18.7	40.0	17.0	51.9	55	2.0	252	233.33	12.0	0.20	71.54	228.84	112.98
		22.22	17.3	19.3	40.0	18.3	53.6	55	2.0	252	444.44	12.5	0.20	119.24	226.16	287.67
4.	6.97	3.47	18.0	18.9	42.1	19.7	53.6	60	2.4	336	31.25	12.5	0.10	10.84	233.83	11.97
		6.94	18.0	19.0	44.7	22.1	55.2	70	2.4	336	69.44	13.0	0.20	24.02	224.16	28.48
		11.11	18.0	19.0	43.3	20.4	53.6	65	2.4	336	111.11	12.5	0.20	39.24	226.15	50.61
		16.67	18.0	19.4	44.1	22.3	53.6	70	2.4	336	233.33	12.5	0.20	79.96	226.16	135.13
		22.22	18.0	20.0	43.0	22.4	55.2	65	2.4	336	444.44	13.0	0.20	132.53	224.16	379.12

A.8 TABLE 12. EXPERIMENTAL FLUID - BUFFALO MILK

Sl. No.	N (RPS)	$\dot{M}$ ($\frac{kg}{s} \times 10^{-3}$)	S_I (%)	S_o (%)	To_c (°C)	To_E (°C)	TD_R (°C)	P_a (mm of Hg)	P_T (KW)	P_S (W)	$\dot{M}_v$ ($\frac{kg}{s} \times 10^{-6}$)	Pd (bar)	Ps (bar)	U_o ($\frac{W}{m^2k}$)	h_o ($\frac{W}{m^2k}$)	h_s ($\frac{W}{m^2k}$)
1.	3.42	3.47	16.0	16.8	44.1	29.5	56.8	70	1.0	192	27.77	13.5	0.00	7.94	238.00	8.64
		6.94	16.0	16.6	44.2	29.4	56.8	70	1.0	192	41.66	13.5	0.00	12.01	238.00	13.34
		11.11	16.0	16.6	44.1	28.0	56.8	70	1.0	192	66.66	13.5	0.80	19.06	196.95	22.30
		16.67	16.0	17.0	44.2	27.9	56.8	70	1.0	192	166.66	13.5	0.50	48.05	206.57	67.07
		22.22	16.0	17.3	44.1	28.1	56.8	70	1.0	192	288.88	13.5	0.40	82.63	212.16	146.50
2.	5.00	3.47	17.8	18.7	50.8	34.3	62.8	100	1.8	228	31.25	15.5	0.70	9.39	187.06	10.41
		6.94	17.8	18.6	51.0	34.0	64.2	100	1.8	228	55.55	16.0	1.00	15.18	179.73	17.50
		11.11	17.8	18.7	51.0	27.3	64.2	100	1.8	228	100.00	16.0	1.40	27.32	171.70	34.46
		16.67	17.8	19.3	50.8	29.4	65.5	100	1.8	228	250.00	16.5	1.25	61.35	169.66	104.05
		22.22	17.8	19.5	50.8	31.0	65.5	100	1.8	228	377.77	16.5	1.25	92.70	169.66	318.35
3.	6.00	3.47	18.0	18.8	48.9	26.1	58.4	90	2.1	276	27.77	14.0	1.10	10.50	184.86	11.73
		6.94	18.0	18.8	48.1	25.7	56.8	85	2.1	276	55.55	13.5	1.00	23.10	188.90	27.86
		11.11	18.0	19.3	42.8	23.0	53.6	65	2.1	276	144.44	12.5	0.00	48.66	242.66	65.16
		16.67	18.0	19.3	44.0	24.1	53.6	70	2.1	276	216.66	12.5	0.00	81.98	242.66	135.26
		22.22	18.0	19.6	43.9	24.5	53.6	70	2.1	276	355.55	12.5	0.00	133.43	242.66	343.27
4.	6.97	3.47	16.6	17.4	42.0	27.0	50.3	60	2.6	348	27.77	11.5	0.00	12.19	247.68	13.52
		6.94	16.6	17.4	43.0	27.0	50.3	65	2.6	348	55.55	11.5	0.00	27.68	247.68	33.04
		11.11	16.6	17.7	40.4	27.0	48.5	60	2.6	348	122.22	11.0	0.00	54.98	250.61	75.63
		16.67	16.6	18.0	40.4	28.0	48.5	60	2.6	348	233.33	11.0	0.00	104.96	250.61	201.05
		22.22	16.6	18.4	40.6	28.1	50.3	60	2.6	348	400.00	11.5	0.00	150.25	247.68	455.86

VITA

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THESIS

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MAJOR FIELD

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Higher Secondary and Pre-Engineering from Panjab University, Chandigarh in 1966 and 1967. B.Sc. in Agricultural Engineering from Punjab Agricultural University, Ludhiana in 1973; M.Sc. in Dairy Engineering from Kurukshetra University in 1979.

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January to April, 1973, Production Engineer, National Engineers Corporation, Sangrur (Punjab); August to December, 1973, SCI Soil Conservation Department (Punjab); December to April, 1974, AE, FSC, Haryana Agro-Industries Corporation Ltd.; 1974 onward in Dairy Engineering Division at National Dairy Research Institute, Karnal and was inducted in ARS Cadre in 1975; as Scientist with teaching and research assignments in Dairy Engineering. Published 25 papers.