Performance Evaluation and Wear Assessment of Diesel Engine used in Field Application

THESIS

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In

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(Farm Machinery and Power Engineering)

By

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2018

CERTIFICATE - I

This is to certify that the thesis entitled ***"*****Performance Evaluation and Wear Assessment of Diesel Engine used in Field Application*****"*** *submitted in partial fulfillment of the requirement for the degree of* ***MASTER OF TECHNOLOGY*** *in* ***Agricultural Engineering (Farm Machinery and Power Engineering)*** *of Jawaharlal Nehru Krishi Vishwa Vidyalaya, Jabalpur is a record of the bonafide research work carried out by* ***Mr. Kunal Bhelave*** *under my guidance and supervision. The subject of the thesis has been approved by the Student's Advisory Committee and the Director of Instructions.*

All the assistance and help received during the course of the investigation has been acknowledged by him.

Place: Jabalpur
Date: / / 2018

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Declaration and Undertaking by the Candidate

I, Kunal Bhelave S/o Shri Rajkumar Bhelave, Certify the Work embodied in thesis ***"*****Performance Evaluation and Wear Assessment of Diesel Engine used in Field Application*****"*** is my own first hand bonafide work carried out by me under the guidance of Dr. A.K. Shrivastava, Department of Farm Machinery and Power Engineering, College of Agricultural Engineering, Jawaharlal Nehru Krishi Vishwa Vidyalaya, Jabalpur during 2016-18.

The matter embodied in the thesis has not been submitted for the award of any other degree/diploma. Due credit has been made to all the assistance and help.

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Place: Jabalpur

Date: **Kunal Bhelave**

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List of Abbreviations

Abbreviations	Stands for
%	Percentage
AC	Alternating current
ADC	Analog to digital converter
bhp	Brake horse power
BP	Brake power
BSFC	Brake specific fuel consumption
^{o}C	Centigrade
CI	Internal Combustion
Cp	Center of pressure
CV	Calorific value
D	Diameter
DC	Direct current
DI	Direct ignition
dept	Department
e.g.	Exempli Gratia (for example)
eq.	Equation
et al.	And others
etc.	Etceteras (and so on)
Fig.	Figure
g	Gram
H_{exh}	Heat exhaust
h	Hour
Hz	Hertz
I/P	Injection pressure
IARI	Indian Agriculture Research Institute
ICAR	Indian Council of Agriculture Research
JNKVV	Jawaharlal Nehru Krishi Vishwa vidyalaya
K	Dynamometer constant
kg	Kilogram
kN	kilo Newton
kW	Kilowatt
L	Liter
LED	Light emitting diode
LPH	Liters per hour
LPM	Liters per minute
Lub	Lubrication
m	Meter
mA	Mill ampere
mep	Mean effective pressure
M_f	Flow rate
Min	Minute
mm	Millimeter
MP	Madhya Pradesh
MS	Mild steel
N	Newton

Abbreviations	Stands for
nBTH	Brake thermal efficiency
nos	Number of system
PC	Personal computer
PPR	Polypropylene
Pr	Pressure
RPM	Revolution per minute
SFC	Specific Fuel Consumption
T1	Calorimeter exhaust gas inlet temperature
T2	Calorimeter exhaust gas outlet temperature
T3	Calorimeter water inlet temperature
T4	Calorimeter water outlet temperature
T5	Engine cooling water inlet temperature
T6	Engine cooling water outlet temperature
T8	Ambient temperature
TDC	Top dead center
TFC	Total fuel consumption
USB	Universal serial bus
V	Volt
W	Brake load
ρ	Density

CHAPTER-I
INTRODUCTION

INTRODUCTION

Agriculture is the backbone of Indian economy and majority of the Indian population depends on agriculture. Technological improvements in Indian agriculture since mid sixties have brought about revolutionary increase in agricultural production. Food grains productivity in India has increased from 0.710 t/ha in 1960-61 to 2.11 t/ha in 2013-14, total farm power availability has increased from 0.296 kW/ha to 2.02 kW/ha during the same period. Thus, food grains productivity is positively associated with unit power availability in Indian agriculture. The different sources of power available on Indian farm for doing various mobile and stationary operations are mobile power viz. human, draught animals, tractors, power tillers and self propelled machines whereas, for stationary power that is diesel/oil engines for pump sets, threshers, rice hullers, sprayers etc. The population of stationary prime movers has increased tremendously since the green revolution. Diesel engine population in the country increased about 37 times between 1960-61 and 2013-14. Diesel engine availability registered an increasing trend from 1.62 units/1000 ha of net cultivated area to 59.67 units/1000 ha during the period of 1960-61 to 2013-14. Power from diesel engines increased from 0.009 kW/ha in 1960-61 to 0.247 kW/ha in 2000-01 and 0.335 kW/ha in 2013-14 (Singh et al., 2014).

The use of 5 hp stationary diesel engines is predominant for irrigation of kharif and rabi crop both in rainfed area, where various government and NGO schemes built many type of water harvesting structures are already available for many years along with local natural water bodies. The government also providing diesel subsidy in terms of irrigation pump sets to the farmers and on farm pond (Balram talab) under MANREGA schemes for irrigating the lands. Presently diesel engine applications are too broad as compare to petrol engines. The 5 hp engines is being used for diverse applications other than agriculture as a power source in stationary applications as well as local mobile applications as “jugad’’ vehicle in rural areas. These engines are also being used extensively in civil constructions machinery and electrical generators. There are many medium and small scale industry in Gujarat and Uttar Pradesh producing engines with ISI mark. It has been observed that these engines used for years after rebuilt/repair at local workshops again used as an easily available mechanical source of farm power. These engines performance in terms of power development, thermal and mechanical efficiency may vary depending upon severity of operation and poor maintenance. Due to that exhaust emissions from these engines also unnoticed and causes severe health hazard to the workers at the engine operation site. Increasing the thermal efficiency and reducing fuel consumption is a big challenge in front of engine manufacturers.

The engine performance is a function of the millage of the engines. As the numbers of years usage increases, the performance of the engine also reduces due to numbers of factors like mechanical losses and wear and tear. The importance of proper maintenance of engine by proving that there are considerable losses even in well maintain engines was stressed (Shrinivasan et al., 2014).

The basic task in the design and development of engine was always to reduce the cost of production and improve the efficiency and power output. In order to achive the above task, the development engineer has to compare the engine develop with other engine in terms of its output and efficiency (Ganeshan, 2003). The performance of diesel engine is being assessed in the laboratory under controlled condition mostly with the help of eddy current dynamometer of suitable size. In the proposed study it was opined that there is need to assess the performance of engines presently being used in the field condition particularly the power develop, brake specific fuel consumption at different speed. It was also felt that these engine should also be checked for loss of power if any, by heat balance method and there after inspection of actual physical dimensions of diesel engine components responsible for efficient operation.

By keeping in view of above the study on three different aged engine to test with following objectives.

Objectives of the present research are as follows

1. Performance and evaluation of stationary diesel engines on engine test rig in laboratory condition by using electrical dynamometer.
2. Analysis of test result by heat balance method and wear assessment of the stationary diesel engines.

CHAPTER-II
REVIEW OF LITERATURE

REVIEW OF LITERATURE

In general, the characteristics of diesel engine may be described in terms of its performance. Performance characteristics are primarily concerned with its capability to develop required Break horse power with certain efficiency. Performance of a stationary diesel engine depends on many machine and environmental factors. There are many type of diesel engines is being used with different features and modified according to the application. The performance evaluation of stationary and automotive diesel engines are carried out by testing them on a standard test rig. The usefulness and wear out characteristics of diesel engines also depends upon the operating condition and load. In past, many researchers have worked on these aspects. This chapter reviews the different methods, models and parameters adopted worldwide in various conditions, which are relevant to the present study. These are given in following subheadings:

2.1 Performance and evaluation of diesel engine.

2.2 Analysis of test results and wear assessment of the diesel engines

2.1 Performance and evaluation of diesel engine

Fang (1949) used various vegetable oils as fuel in a diesel engine and found incomplete combustion to be a problem when these fuels were used in a cold engine. Starting and stopping the engine on diesel fuel and heating the injector and fuel lines were recommended so, that the vegetable oils could be better atomized to burn more completely.

Geyer *et al.* (1984) compared emission characteristics of a single cylinder, DI, diesel engine on diesel, cottonseed oil, sunflower oil, methyl ester of cottonseed oil and methyl ester of sunflower oil. The primary objective was to assess aldehyde emissions and the potential health effects of particulate emissions when operating a diesel engine with vegetable oils. The engine was operated at 2400 rpm and load conditions of 1/3, 2/3 and full rack. The emissions of CO, He, NOx, total aldehyde as well as individual aldehyde concentrations were measured. It was reported that NOx was significantly higher for methyl esters at all rack settings. The emission of total aldehyde increased dramatically on vegetable oils compared to diesel. The amount of form aldehyde increased with rack settings and was consistently lower for methyl esters than for diesel. The diesel had a larger increase in aldehydes with rack setting while vegetable oils did not. Overall, the aldehydes averaged 12% for diesel and 31% for vegetable oils.

Alfuso *et al.* (1993) carried emission tests of a direct injection, compression ignition diesel on rapeseed methyl ester. It was reported that rapeseed methyl ester promoted a rise in NOx, decrease in HC and CO and reduced smoke. Particulate matter produced by rapeseed methyl ester in transient cycles was higher than that obtained with diesel fuel. The contribution of soluble organic fractions was reported to be higher at low loads. The rapeseed methyl ester produced more soluble organic fractions and particulate matter than the diesel fuel at lighter loads while as in proximity of full load the trend was the opposite. The emission of hydrocarbon from diesel was higher at lighter loads and lower at higher loads. Carbon monoxide emission was about the same at different loads on diesel but it reduced with rapeseed methyl ester and nitrogen oxide production was generally higher on biofuel.

Gupta (1994) conducted experiments on a Kirloskar AV-I, single cylinder, direct injection, water cooled, naturally aspirated, 4-stroke cycle constant speed diesel engine having rating of 3.7 kW at 1500 rpm. The study revealed that, at injection timing of 20° BTDC for rice bran oil esters, brake thermal efficiency was found to be 27.07, 29.48 and 28.87% at loads of 76, 98.5 and 107% respectively. In case of rice bran oil ester, highest brake thermal efficiency was observed in general at injection timing of 20° BIDC. Further, the lowest brake specific fuel consumption was reported to be 1.4066, 0.5571, 0.3885, 0.3394, 0.3116 and 0.3183 kg/kW-h at load levels of 8.5, 25.4, 50.7, 7.6, 98.5 and 107% respectively for rice bran oil ester.

King (1995) operated a conventional agricultural tractor on a blend of degummed and filtered rape seed oil with diesel mixed in 15: 85 proportions. A 400 hours' test was conducted to examine feasibility of using degummed and filtered rapeseed oil as diesel fuel extender for direct injection engines. The results indicated that the output power and torque were 2.5% lower on the test fuel compared with diesel. Brake specific fuel consumption was 1.5 percent higher on the test fuel. Analysis of the engine lubricating oil for both diesel and test blend showed no abnormal and elemental composition changes in the oil for the test fuel as well as the diesel.

Raghuwanshi *et al.* (2012) studied the failure analysis of internal combustion engine valves. For the study of fatigue life I.C. engine valves were subjected to repeated cyclic loading due to valve train dynamics. Repeated loading was resulted in materials failing well below the yield strength. It was found that the predominant cause of failure of valves of internal combustion engine was fatigue. The valves were subjected to high temperature, cyclic loading, impact loading, erosion-corrosion and high pressure inside the cylinder, a combined S-N (max. stress v/s number of cycles) curve was prepared. Such a curve was used to helps in comparing the fatigue failure for different materials at different high temperatures and was also assisted the researchers in developing the valve materials.

Kunar *et al.* (2013) studied on measurement and evaluation of reliability, availability and maintainability of a diesel locomotive engine. It was found that worker inefficiency, overloading, degradation of the system components with time are mainly the contributing factors for the failures of railway diesel locomotive engine. From the reliability prediction of the engine components it was found that the reliability of the components ranging from 59% to 68%, where maximum reliability was found in the case of transmission cable (i.e. 67.9%) and minimum reliability found in case of vehicle and structure (i.e. 59%). It was analyzed that major causes of failure include care less and in experienced handling, improper and excessive loading, environmental, care less operation and degradation.

Srinivasan *et al.* (2014) conducted experiment on testing and analysis of the correlation between engine performance and engine life. The engine was coupled to air cooled dynamometer with a maximum power-rating of 14.7 kW. It was observed during the test that the power loss of 30% in both a single cylinder SI engine of 3.8 kW, 3600 RPM and twin cylinder CI engine of 3.8 kW, 1500 RPM. The major issues were wear and pitting of bearings due to foreign partials, defective bearings seals, soot deposition, valve leakage, blow by loses due to wear in piston rings and low injection pressure.

Totala *et al.* (2014) analyzed various engine parameters in multicylinder diesel engines corresponding to fuel injection timings. They have conducted five tests on four-cylinder diesel engines had been carried out at five different fuel timings. It was at 9^o, 11^o, 13^o, and 15^o. BTDC (before top dead center) setting of fuel pump. Five mode tests was consists of testing the engine at five different load conditions at constant speed at 100% load, 75% load, 50% load, 25% load and 10% load. For that setting first five mode of test at constant 1500 rpm was conducted. This five mode tests consist of measurement of power, fuel and lubricating oil consumption. Engine was fitted with fuel injection pump for setting at 7 deg. before top dead center as load was decreased. A/F ratio was increased which is true for any type of fuel pump setting that is 7deg; 9 deg; 11deg; 13deg and15 deg. BTDC. There was very sharp rise in A/F ratio. And as advance of the fuel timing was increased i.e. from 7^o to 15^o, the A/F ratio goes on decreasing position. As load was reduced the fueling also found to be reduced i.e. fuel taken up by the cylinder per stroke reduces. As the advance was increased the fueling was found to be reduced. As the load was decreased, the SFC was increased and it was very sharp from 25% load to 10% load. It was nearly constant for 100% load to 50% load and the variation was slightly more from 50% load to 25% load.

Reddy *et al.* (2015) studied that effect of compression ratio on the performance of diesel engine at different loads**.**The engine performance study Included brake power, indicated power, frictional power, brake mean effective pressure, brake thermal efficiency, indicated thermal efficiency, mechanical efficiency, volumetric efficiency, specific fuel consumption, air fuel ratio, heat balance and combustion analysis. It was found that the on increasing compression ratio of the engine, brake specific fuel consumption was decreased. The result was also concluded that when the load was increased at any particular compression ratio then the specific fuel consumption decreased from 0.51 kg/kW at 3.31 kg load and we get lowest SFC 0.29, 0.29, 0.27, 0.24 for compression ratios 14, 15, 16 and 18 respectively at load (12 kg). At the lower sides of the compression ratios, the fuel consumption was high due to incomplete combustion of the fuel. The maximum fuel consumption was measured at compression ratio of 14. It was also found that on increasing the load at any particular compression ratio then brake power was increased and statistical data shows that brake power increased from 0.97KW at 3.31 kg load to 3.44, 3.48,3.45, 3.46 KW for compression ratio 14, 15, 16 and18 respectively at 12 kg load. The maximum brake thermal efficiency was obtained at a compression ratio of 18, due to the superior combustion and better intermixing of the fuel. The least brake thermal efficiency was obtained at a compression ratio of 14.

Desai and Shivkumar (2015) determined the efficiency of single cylinder diesel engine by introducing oxygen through air intake and preparing heat balance sheet. It was found that speed of engine increases with load because of enriching air intake by oxygen and specific fuel consumption was reduced at different load conditions. The break power and break thermal efficiency was found to be increased satisfactorily at varying speed and load as compare to normal combustion process.

Sharma *et al.* (2015) studied artificial neural networking (ANN) based modeling of performance and emission characteristics of diesel engine fueled with polanga biodiesel at different injection pressures. The Experimental data in the proposed ANN was obtained at a constant speed and full load condition. BSFC results for different biodiesel-blended diesel fuels and injection pressures at constant load. A decrease in injection pressure increased the BSFC values compared to original injection pressure of 180 bar for all the blends. The BSFC was resulted for different biodiesel-blended diesel fuels and injection pressures at constant load. A decrease in injection pressure increased the BSFC values compared to original injection pressure of 180 bar for all the blends due to decreasing injection pressure, fuel particle diameters will enlarge and ignition delay period during the combustion will increase. This situation was caused an increase in the BSFC. Minimum BSFC for BD10 was 0.226 kg/kW-hr at 180 and 200 bar and with increase in injection pressure from 200 to 240 bar, the BSFC is increased to 0.36 kg/kW-hr. It is found that the BSFC is decreased with increase in injection pressure up to 200 bar. An ANN model was developed based on standard back- propagation algorithm for the engine. Multi-layer perception network (MLP) was used for non-linear mapping between the input and output parameters. Different activation functions and several rules were used to assess the percentage error between the desired and the predicted values. It was observed that the ANN model can predict the engine performance and exhaust emissions quite well with correlation coefficient (R) 0.99998, 0.9999, 0.99998, 0.9999, 0.9958, 0.9993, 0.9999 for the brake specific fuel consumption, brake thermal efficiency, exhaust gas temperature, NOx, CO, smoke and UBHC emissions, respectively.

2.1 Analysis of test results and reliability/ wear assessment of the diesel engines

Bora *et al.* (2010) experimented on wear and tear analysis of a single cylinder diesel engine using karanja biodiesel (B20) after 512 hours. It was found that BSFC for B20 karanja slightly higher than diesel due to lower calorific value of B20K than diesel. The difference in change in viscosity of lubricating oil using diesel and B20K is found negligible. Lower level of iron concentration indicates low rate of engine wear in case of B20K fuel. It was concluded that by mixing 20% bio diesel in diesel, no adverse effect was observed for engine component, lubricating oil and emission performance of engine.

Smedley (2002) reported that the piston ring pack is the largest single contributor for friction power losses in modern internal combustion engine. In this study reduce friction design were developed with the help of modeling tool and tested on a full scale engine. Additional design was also developed to eliminate adverse effects such as increased oil consumption, blow by or wear that might accompany changes in ring designs to reduce friction.

Sengar *et al.* (2016) studied the performance evaluation of a diesel engine with varying shims configuration for different blends of diesel and ethanol. They reported that the shim thickness reduction leads to injection advance which is the case required for the use of Ethanol as fuel in C.I. engines. The experiments were conducted on a direct Injection, 553 cc diesel engine of 5 HP. Experimental tests were carried out to study the configuration on the basis of five performance parameters (Mf, AFR, BSFC, Volumetric and Thermal efficiency). The test results was found that the 4 shims (0.66 mm) configuration and 6 shims (1.535 mm) configuration are most suitable for pure diesel and no shims (0 mm) configuration for E20blend. And increase in performance can be further improved by use of lubricates and igniters along with the change in coolant circulation rate to maintain the optimum working temperature of engine. It was also found that CI engine with less compression ratio are not compatible to run on higher blends of ethanol (E60 and above).

Yadav (2015) reviewed on lubrication and wear in IC engine using acoustic emission. The lubrication in the engine was successfully monitored based on the acoustic emission technique generated by captured using an AE wide band transducer that was bonded outside the engine block and located at the lower part of the bottom dead centre (BDC). The acoustic emission technique was used to assist the owner of the engine in determining the engine oil's condition before replacing it. In engine monitoring of the proper lubrication oil supply the cylinder is essential in achieving better efficiency for the engine in term of power, flue, and consumption. The engine is running at high speed in which high fuel consumption to the micro-80AE were mounted on the outer skirt of the engine cylinder, The acoustic emission was converted into digital form and was fed to a PC with a custom developed Lab view program acting as the interface by measurement instrumentation. The engine was coupled with an eddy current dynamometer and a controller. Engine speed and load were controlled by varying excitation current to the eddy current dynamometer. The crank angle en-coder connects with the crankshaft to measure angular position within a single turn the lubrication used on parts of the engine were measured using accelerometers with the sensitivity.

Wielligh (2009) studied engine problem due to poor quality of diesel fuel. It was found that about 30% of engine failures were caused by combustion related problems. In most causes the fuel was a poor quality and resulted in damaged to injection pump and the injector themselves. The lubricity, particle contamination, water contamination and diesel fuel temperature were observed. Due to the lower viscosity of oil often varying failure were observed. A great number of test were done on fuel which came from engines that failed due to fuel problem and it was determined that a minimum load 700N is required from the fuel to ensure no injector problem due to lubricity. It was found that during transporting and handling fuel other get contaminated due to dust that was resulted into were in pump and injector. It was found that whenever micro droplets of water enter the injection system scuffing of the needle in the injector occurs almost immediately.

CHAPTER-III
MATERIAL AND METHODS

MATERIAL AND METHODS

This chapter deals with the details about the study of physical specification and performance parameter of Indian stationary diesel engine. It includes theoretical consideration and method for testing the diesel engine to describe engine performance. The parameters that affect break horse power for testing stationary diesel engine under laboratory condition are also studied. The technical plan, experimental procedure, experimental or test set-up and instrumentation for conducting test of diesel engine under the laboratory conditions. To record observations and analysis of different performance characteristics parameters, a research plan was finalized to achieve the desired objectives in three steps as given below:

Step I – Selection of the stationary diesel engine and preparation of test rig.

Step II – Stationary diesel engine test for performance and evaluation.

Step III - Analysis of test result and wear assessment of these engines.

3.1 Selection of the stationary diesel engine and preparation of test rig

3.1.1 Selection of stationary diesel engine

The single cylinder stationary diesel engines of three different horse power ranges were selected randomly from the work site. The engine were belongs to different make and model which are coded as engine A, B and C for the study purpose the brief specifications of this engines are given in table 3.1.

Table 3.1 Selection of stationary diesel engine

S. No.	Particular specification	Engine A	Engine B	Engine c
1.	Engine type	Air cooled	Water cooled	Water cooled
2.	Number of cylinder	1	1	1
3.	Maximum power, kW	3.72	7.45	5.2
4.	Engine speed, RPM	1500	1500	1500
5.	Bore, mm	87.5	102	85
6.	Stroke, mm	80	116	110
7.	Compression ratio	16.7:1	17.5:1	17:1
8.	Sp. Fuel consumption, g/bhp-hr	169	190	185
9.	Lub. Oil sump capacity (at higher level on dipstick), L	3.7	3.7	3.7
10.	Governing	Class "B1"	Class "A2/B1"	Class "A2/B1"
11.	Over loading capacity of engine	10%	10%	10%
12.	Crank shaft center height, mm	203	203	203
13.	Torque at full load (crank shaft drive), kg-m	2.387	4.775	3.342
14.	Cubic capacity, L	0.553	0.948	0.661

The above engine was having different specifications and working conditions. As per the information available it was presumed that the engines result A, B and C are completed different life span. It was coded as engine A, B and C in the increasing order of working hour or life.

3.1.2 Stationary diesel engine test rig

The engine test rig was established in the farm machinery testing center, dept. of farm machinery and power engineering, college of agricultural engineering, JNKVV Jabalpur. It consists of structure (figure 3.1) with following instruments.

1. Water cooled Eddy Current Dynamometer.
2. Calorimeter with valves and temperature sensor.

The test rig panel (figure 3.2) consists of following items

- Slow ADC
- Fast ADC
- Torque Indicator/Controller
- Switches
- Air Flow Transducer
- Water Flow Transducer
- Electrical Connectors and Wiring
- Fuel Tank
- Engine Fuel Consumption Meter
- Manometer
- Air Box

The system structure is a self contained made of MS and the piping is rustproof PPR type. All instruments are duly assembled and wired in one structure.

On the front side of the panel there are various switches as shown in Plate 3.4

1. **Mains:** This switch is for switching on 230 V AC supply to the trainer.
2. **Fuel Override:** This switch is for manually filling fuel in the fuel tank. (Overriding the auto fuel fill arrangement)
3. **Torque Indicator/Controller:** This type of controller is used to apply load on the engine. You can observe applied load to engine on Slow ADC as well as on software and Torque Indicator/Controller.
4. **230V AC Supply:** Switch ON this switch on Torque Indicator/Controller to give 230V supply to Torque Indicator/Controller.

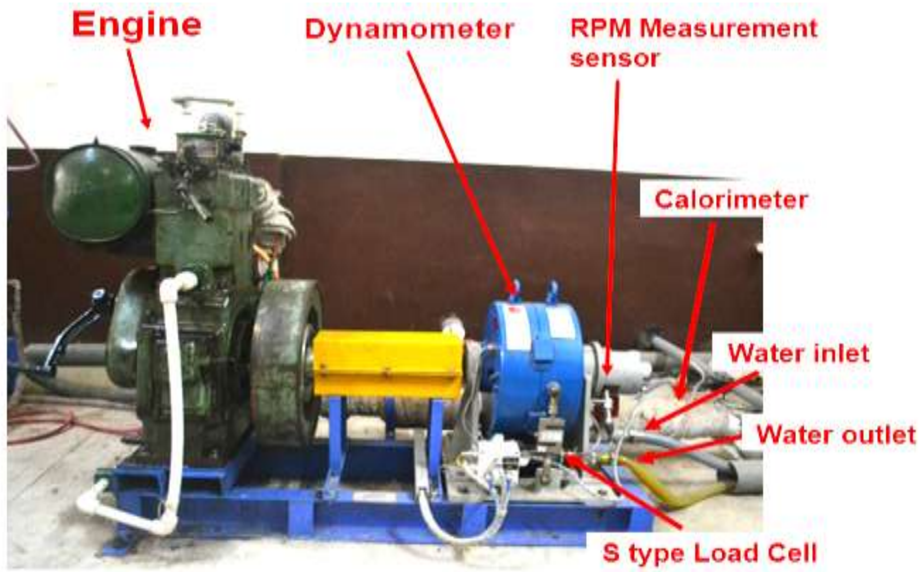

Plate 3.1: Engine with Eddy Current Dynamometer

Plate 3.2: Control Panel

Plate 3.3: During Data Logging

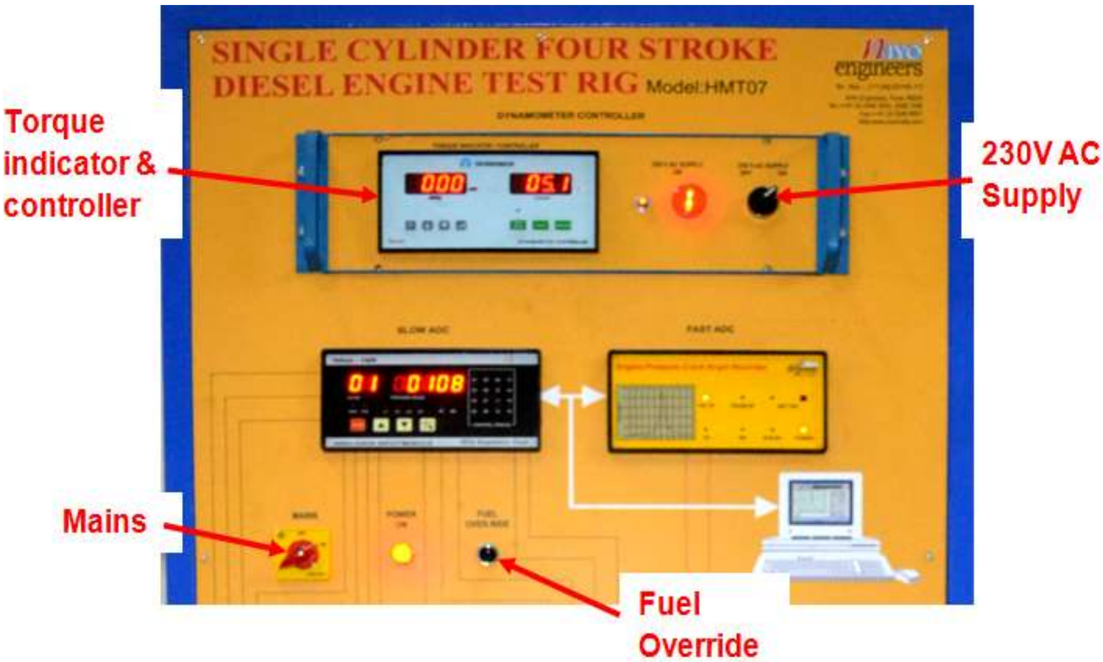

Plate 3.4: Various Switches on Panel

3.1.3 The test rig instrumentation for measuring various inputs/outputs for diesel engine performance

All the instrumentation incorporated was PC based as well as with local indicators. The various factors were to be measured as follows:

a. Fuel measurement

This was done by using a specially designed arrangement suitable for all kinds of petrol and diesel engines. This instrument was placed on the control panel. The amount of fuel consumed was determined by software/hardware combination by deducting the initial reading from final reading at a regular pre-determined interval.

$$TFC = \frac{FC_2 - FC_1}{1000} \times \frac{60}{t} \text{ ....eq. (3.1)}$$

Where,

TFC = Total fuel consumption;

FC_2 = Initial reading;

FC_1 = Final reading; and

t = Time interval.

b. Air flow measurement

Air flow is measured using an air box, orifice and manometer arrangement. The inlet of the air suction box consists of orifice housed in the orifice flanges.

Pressure difference across the orifice is read on the manometer. The outlet of the air suction box goes to the engine through a flexible hose for air suction.

$$(W_{air}) = Qact \times \rho - air \times 60 \text{ ....eq. (3.2)}$$

Where,

W_{air} = Weight of air drawn into the cylinder;

Q_{act} = Actual volume; and

ρ-air = Air pressure.

c. Exhaust gas heat loss measurement

Exhaust gases from the engine passes through the flexible hose to the calorimeter. The calorimeter is mounted on a stand and supports. Exhaust gas enters into the calorimeter through the calorimeter exhaust gas inlet. Heat is exchanged by circulating water through a copper pipe in the calorimeter. Sensors mounted at various position measures the temperatures at that point.

$$H_{exh} = \frac{T_1 - T_5}{T_1 - T_2} \times \frac{m_a}{60} \times C_p (T_4 - T_3) \text{ kW} \quad \text{...............eq. (3.3)}$$

Where,

T1 = Temperature of exhaust gas calorimeter in

T2 = Temperature of exhaust gas calorimeter out

T3 = Temperature of water calorimeter in

T4 = Temperature of water calorimeter out

T5 = Temperature of water engine in

d. Temperature measurement

The temperature at different points is measured and displayed on PC. The points are,

1. Calorimeter exhaust gas inlet temperature (T1)
2. Calorimeter exhaust gas outlet temperature (T2)
3. Ambient temperature (T8)
4. Calorimeter water inlet temperature (T3)
5. Calorimeter water outlet temperature (T4)
6. Engine cooling water inlet temperature (T5)
7. Engine cooling water outlet temperature (T6)

e. Speed measurement

The speed of the engine widely used in the computation of power, design and development. For accurate and continues measurements of speed a magnetic pick-up placed near a toothed wheel coupled to the engine shaft was used. The magnetic pick-up produces a pulse for every revolution and a pulse counter was accurately measured the speed. It was displayed on PC. The sensor gives one pulse per revolution and the frequency of these pulses is directly proportional to the speed.

f. Load Measurement

A load cell is a transducer that is used to create an electrical signal whose magnitude is directly proportional to the force being measured.

Load cell is mounted on the dynamometer Load applied to engine can be seen on Slow ADC as well as on software

g. Data Acquisition software

Data acquisition software was provided with the engine test rig which was coded by the manufacturer of test setup as per the requirement of engine testing.

h. P-θ Measurement

Pressure sensor along with signal conditioner is used for cylinder pressure measurement. Angle and TDC are marked by encoder.

Pressure Sensor: It generates an electrical voltage in response to pressure. Encoder**:** It generates pulse as per crank angle. It also generates pulse for TDC.

3.1.4 Eddy Current Dynamometer

These machines make use of the principle of electro-magnetic induction to develop torque and dissipate power shown in figure 3.1. A toothed rotor of high-permeably steel rotates with a fine clearance between water-cooled steel loss plates. A magnetic field parallel to the machine axis is generated by two annular coils and motion of the rotor gives rise to changes in the distribution of magnetic flux in the loss plates. This in turn gives rise to circulating eddy currents and the dissipation of power in the form of electrical resistive losses. Energy is transferred in the form of heat to cooling water circulating through passages in the loss plates, while some cooling is achieved by the radial flow of air in the gaps between rotor and plates. Figure 3.1 shows the view of eddy current dynamometer.

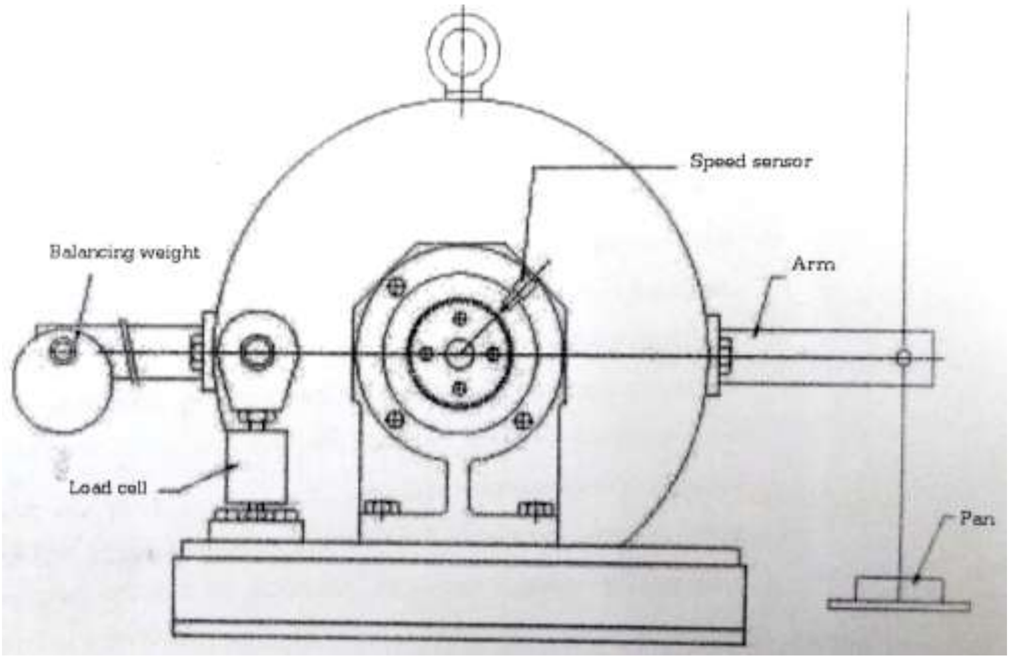

Figure 3.1: Eddy current dynamometer

3.1.5 Brief specifications of instruments

1. Dynamometer
    - Dynamometer Model: TMEC 10
    - Water cooled Eddy Current Dynamometer
    - Normal Rating: 10HP/ 7.5kW @ 1500 RPM
    - Minimum Water Pressure: 1.6 bar
    - Cooling Water Flow Rate: 14 to 15 L/min
2. Air Flow Transducer
    - Orifice with Manometer
3. Water Flow-rate Transmitter
    - Wheel type
    - Range: 0-1000 LPH
    - Output: 4-20 mA
4. Fuel Consumption meter
    - Output: 4-20mA

- Range: 112gm

5. Load Cell
   - Strain gauge, S type
   - 0 to 150 kg
   - Full Scale output: 3mV/V
   - Operating Mode: Compression/tension
   - Threading: M10

6. Load Cell Transmitter
   - Range: 0 to 270 Nm
   - Output: 4 to 20 mA

7. Pressure transmitter Sensor (P-θ and P-V arrangement)
   - Type: Piezo electric
   - Measuring Range: 0 to 100 bar
   - Output: 0 to 5 V

8. Encoder (P-θ and P-V arrangement
   a. Crank angle marker:
      - 360 PPR
      - TDC marker
      - 5 V DC supply

9. Data Interface Module
   - Slow ADC
     - No. of channels: 16 nos.
     - Supply voltage: 230 VAC ± 10%, Single phase, 50Hz
     - Display: Seven Segment LED display, 2 digit for channel number, 4 digit for pressure value
     - 4 Channels for Thermocouple (K type)
     - 4 channels for Pt 100
     - 8 channels for 4 to 20 mA
   - Fast ADC
     - No. of channels: 3 nos.
     - Supply voltage: 230 VAC ± 10%, single phase, 50Hz
     - Communication: RS485, Baud Rate: 38400
     - Resolution: 12 bit

3.1.6 Installation and operating procedure of setup for testing diesel engine

1. Mounted the engine along with base frame on a plane surface
   - The site was good ventilation.
   - Adequate distances (about 1 m) were being kept between the trainer and the nearest wall/walls.
   - Clean, pressurized water supply point were available near the engine.
   - 230V AC supply point was available near the engine.
   - Water drain arrangement was available near the engine.
2. Control panel was kept inside of the engine.
3. Four hoses were arranged.
   - 1/2’’ for water inlet of calorimeter.
   - 1/2’’ for water outlet of calorimeter
   - 1/2’’ for water inlet of engine.
   - 1/2’’ for water outlet of engine.
4. Connected the water supply to the inlet provided for the cooling of engine and calorimeter. Connected the drainpipe to the outlet of the engine and calorimeter. Started the water supply. That was the water flows through the drainpipe.
5. Before started the engine, ensure that all the switches are in OFF Position checked.
6. Connected communication cables to USB port of the PC correctly.
7. Filled 5 L of fuel in the Fuel Tank. Capacity of the tank was 7 L
8. Opened the manual valve of fuel provided on panel and were fill the fuel measurement tube completely.
9. Never exceeded 70g on the Fuel Indicator.
10. Switch ON Mains switch provided on panel.
11. Switch ON 230V AC Supply switch on Torque Indicator/Controller.
12. Fuel override the auto fuel filled cycle were using Fuel Override switch provided on the panel. However, it was not used with care this result was in overflow of fuel from the Fuel Consumption Unit.

13. Started the water supply to the engine and calorimeter.
    - Calorimeter: 2 LPM (approximately)
    - Engine: 5 LPM (approximately)
14. Started the engine. Refer to the section "How to Start/Stop engine" for more details.
15. Ensure that the engine was run under no load condition and at idling speed for about a minute to ensure the engine stabilize.
16. Started the software that was installed and enter the input parameter required for the particular specification diesel engine on test rig. As shown in table 3.2.
17. All the parameters were seen in the Observation Table on PC. Verified that all the parameters used in the system were displayed.
18. Checked the engine was loaded. Loaded the engine gradually by using Torque Indicator/Controller provided on control panel. Refer to the section "How to Load/Unload Engine" observe load on slow ADC, it should increase as load increases. For the slow ADC the parameters and ranges are given in the table 3.3.
19. All the parameters were seen in the observation table on PC as well as on slow ADC. Ensure that readings displayed on PC matched with readings displayed on slow ADC.
20. To acquire that the data wait for the reading of 'Fuel Consumption' (on PC) to changed once. Pressed LOG CURRENT DATA button. Now the all the data was stored.
21. Unload the engine. Refer to the section "How to Load/Unload Engine".
22. Stopped the engine. Refer to the section "How to Start/Stop engine" for more details.
23. Closed Manual valve of fuel provided on panel.
24. Switch OFF the switch 230 V. AC supply on Torque Indicator or Controller.

25. Switch OFF the mains switch.

26. With the help of all steps successfully completed stationary diesel engine tested with all parameters.

Table 3.2 Input Parameters and values required for the engine test rig are as follows

Parameters	Engine (A)	Engine (B)	Engine (C)
Type of Engine	4 Stroke	4 Stroke	4 Stroke
No. of Cylinders	1 Cylinder	1 Cylinder	1 Cylinder
Cylinder diameter (D)	87.5 mm	102 mm	85 mm
Cylinder Stroke (L)	80 mm	116 mm	110 mm
Specific heat of water (Cp)	4.187 kJ/kg ^oC	4.187 kJ/kg ^oC	4.187 kJ/kg ^oC
Density of fuel (diesel)	0.832 kg/L	0.832 kg/L	0.832 kg/L
Calorific value of fuel used (cv)	44000 kJ/kg ^oC	44000 kJ/kg ^oC	44000 kJ/kg ^oC
Density of air (ρ)	1.170 kg/m^3	1.117 kg/m^3	1.117 kg/m^3

Table: 3.3. For slow ADC the parameters and ranges are given as below

Channel No.	Parameter	Range
1.	Temperature of exhaust gas, inlet of calorimeter (T1)	0-650^{o} ^{o}C
2.	Temperature of exhaust gas, outlet of calorimeter (T2)	0-650^{o} ^{o}C
3.	Ambient temperature (T8)	0-100^{o} ^{o}C
4.	Temperature of water, inlet of calorimeter (T3)	-50-400^{o} ^{o}C
5.	Temperature of water, outlet of calorimeter (T4)	-50-400^{o} ^{o}C
6.	Temperature of cooling water, inlet of engine (T5)	-50-400^{o} ^{o}C
7.	Temperature of cooling water, outlet of engine (T6)	-50-400^{o} ^{o}C
8.	Engine speed (N)	0-2000 rpm
9.	Engine load (W)	0 to 270 Nm
10.	Fuel consumption	0-112 gm/s
11.	Air flow rate (Q actual)	0-0.01 m^3/sec
12.	Engine water flow rate (mwe)	0-16.66 kg/min
13.	Calorimeter water flow rate (me)	0-16.66 kg/min

3.1.7 Procedure for adjusting TDC marker

Their peak of P- 9 curve repeated appears to be shifted from the desired/expected position; it was corrected by adjusting the encoder TDC pulse with respect to TDC of the engine shown in Plate 3.5.

To adjust

- Bring that piston to TDC. To did was follows the steps given below.
- On the flywheel of the engine there was one notch which reads.
- By rotating the flywheel adjusted the notch with TDC.

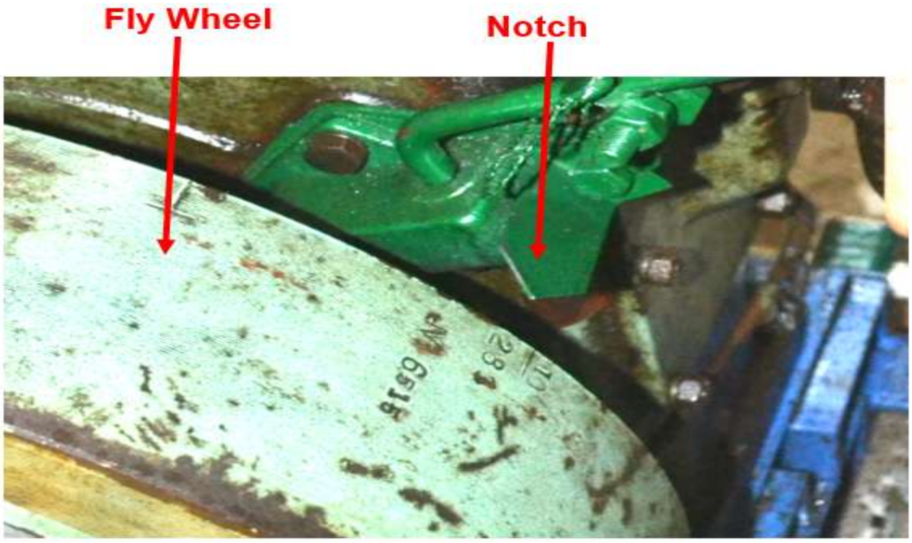

Plate 3.5: TDC Marker

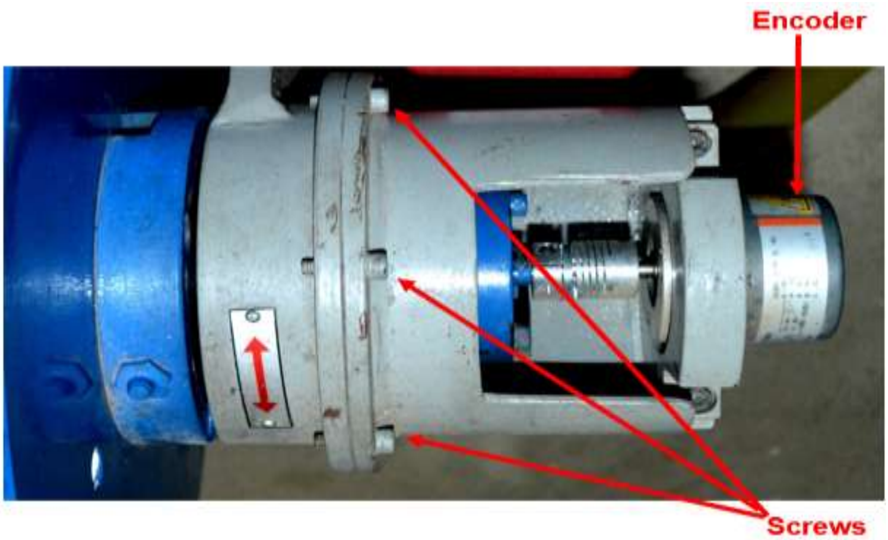

Plate 3.6: RPM Encoder

- Now loosen the encoder mounted screws shown in plate 3.6.
- Switch ON was mains supply to the panel.
- Observed the LED TDC I/P on fast ADC mounted on the panel shown in plate 3.7.
- Observed the LED TDC I/P on Fast ADC mounted on the panel.
- Gradually rotated the encoder. Stopped rotating the encoder exactly at the point where LED TDC I/P goes OFF.
- Tightened the screws holding the encoder so that the position was not changed.
- Then the encoder TDC pulse was aligned with respected to TDC of the engine.

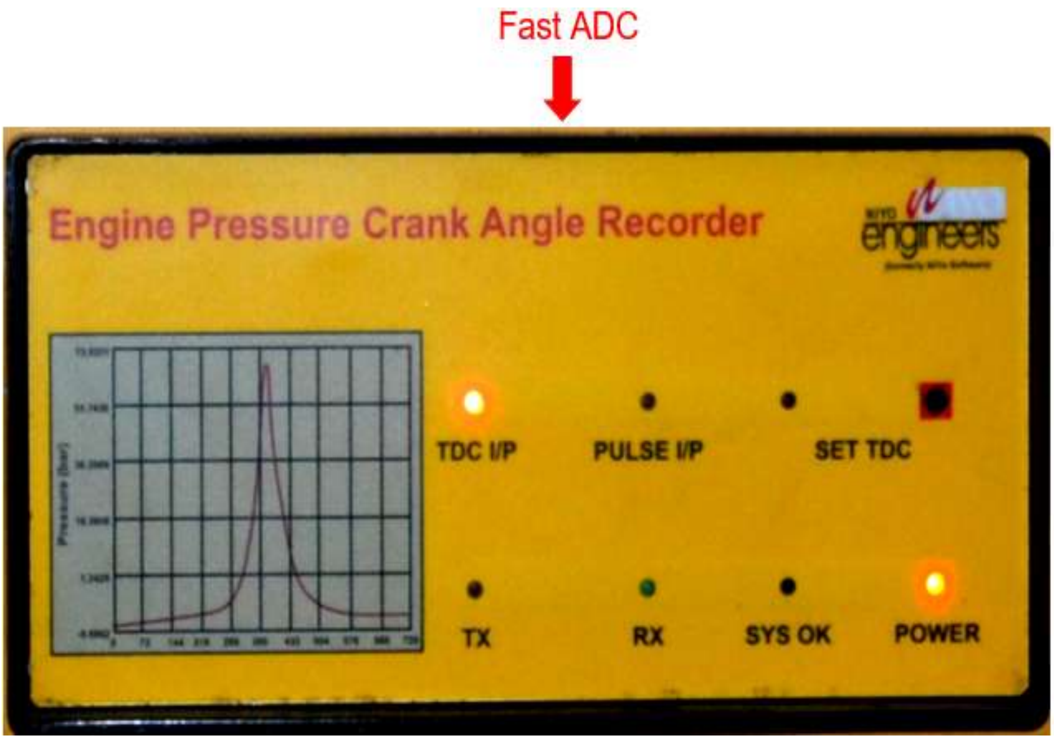

Plate 3.7: Fast ADC

3.1.8 Operating procedure of the diesel engine

To start the engine

1. Ensure that sufficient fuel was present in the fuel tank. Opened the manual valve of the fuel supply shown in plate 3.8.

2. Switched ON mains switch provided on control panel.

3. Ensure that required water flow rate was maintained for the system. water flow requirements are:

   - Calorimeter:2 LPM (approximately)

   - Engine: 5 LPM (approximately)

4. Ensure that the engine was not loaded.

Plate 3.8: Start Engine

5. Started the engine by cranking the shaft. Followed the given procedure to start the engine. Use the handle provided along with engine to crank the engine.

6. Rotated the engine shaft with the handle in clockwise direction as shown in the figure and turned the flywheel fast. When the flywheel attains a good speed, pushed the decompressed lever down. The engine will fire. Removed the handle immediately.

7. The first attempt was failed, repeated the above procedure. In the beginning 2-3 attempts were required.

8. The engine was stabilized for 10-15 min.

To stop the engine

1. Ensure that the engine was fully unloaded.
2. Pressed the lever towards the direction shown in figure.
   Hold it in that position tilled the engine was stopped.
   Switched OFF the water supply after 5 min shown in plate 3.9.

Plate 3.9: Stop Engine

3.1.9 LOAD/UNLOAD procedure of the engine

- Did not load the engine without adequate water supply.
- Water flow requirements were,
    1. Calorimeter: 2 LPM (approximately)
    2. Engine: 5 LPM (approximately)

Keys

F ➔ Function

△ ➔ Increase

▽ ➔ Decrease

↵ ➔ Enter/OK

3. These keys were used for factory setting. It was used for load the stationary diesel engine shown in Plate 3.10.

Plate 3.10: Load and Unload Engine

Run/Stop ➔ This worked was a toggle between RUN and STOP. LED was off when stop mode, LED was ON when in RUN mode.

LOAD ➔ increased the set value of load.

There was segment display.

A. DISPLAY: This was for factory setting. In normal use, this was not of any significance.

B. TORQUE: Torque indicator was displayed the load value in Nm, excepted when LOAD/UNLOAD key was pressed. When one of these keys was pressed the display shows that the set value of torque in Nm.

1. Switched ON that the supply of the control panel and also the Torque Indicator or Controller.

2. Ensured that RUN/STOP LED was OFF.

3. Pressed UNLOAD to make load sitting to 0 Nm.

4. Started the engine. (Refer to the section "How to start and stop the engine" for more details)

5. Display TORQUE showed the load that was approximately 0 Nm.

6. Press and hold LOAD key to increase the set value of torque.

7. UNLOAD key was pressed to reduce torque.

3.2 Stationary diesel engine test for performance and evaluation

3.2.1 Method for testing stationary diesel engines on test rig

We were used diesel to run our engine. Our engine is governor controlled; it takes the diesel fuel in accordance with its need. Observed performance parameters in every case & try to determine the suitability of diesel for improved performance in pure fuel condition.

Started water supply then engine and switch on every accessory such as, temperature indicators and let the engine run for 20 min so, that it can achieve its steady state. In the first set we run the engine on pure diesel fuel purchased from a Petrol Pump. We, initially run the engine at Zero load at take readings of all the parameters required for our performance and emission results. Following parameters are to be noted:

1. Air Velocity in the air passage (m/s)
2. Fuel consumption (kg/min)
3. Temperatures (^oC)

T1 = Calorimeter exhaust gas inlet temperature

T2 = Calorimeter exhaust gas outlet temperature

T3 = Calorimeter water inlet temperature

T4 = Calorimeter water outlet temperature

T5 = Engine cooling water inlet temperature

T6 = Engine cooling water outlet temperature

T8 = Ambient temperature

4. Load on the engine applied by the belt dynamometer (kg.).

5. Water flow to the engine jacket and to the exhaust calorimeter (L/min).

6. After noting down all the parameters, we apply a load of 1 kg on the engine and let it run for 15 min to achieve the steady state and then point down all the parameters again.

7. In the same manner, we increase the load to 2 kg than 4 kg and than 8 kg and note down all the parameters like before.

8. The results of the experiment are tabulated. These are our reference values.

9. After all the values are obtained, these values are scrutinized for the preparation of final results and for comparing the performance parameters.

3.2.2 Important Engine Characteristics

Some basic relationships and the parameters commonly used to characterize engine operation are developed. The factors important to an engine user are:

- The engine's performance over its operating range.
- The engine's fuel consumption with in this operating range and the cost of the required fuel.
- The engine's noise and air pollutant emissions within this operating range.
- The initial cost of the engine and its installation.

- The reliably and durably of the engine, its maintenance requirements, and how these affect engine available and operating costs.

These factors control total engine operating costs usually the primary consideration of the user and whether the engine in operation can satisfy environmental regulations.

Engine performance is more precisely defined by:

- The maximum power (or the maximum torque) available at each speed within the useful engine operating range.
- The range of speed land power over which engine operation is satisfactory.

The following performance definitions are used:

Maximum rated power: The highest power an engine is allowed to develop for short periods of operation.

Normal rated power: The highest power an engine is allowed to develop in continuous operation.

Rated speed: The crankshaft rotational speed at which rated power is developed.

To evaluate the performance of an engine the following are the most important characteristics.

- Thermal efficiency
- Mechanical efficiency
- Indicated work per cycle
- Mean effective pressure
- Specific fuel consumption
- NF & FIA ratio
- Volumetric efficiency

- Engine specific weight/volume
- Specific emissions

a. Thermal efficiency

It is the ratio of the output in the form of useful mechanical power to the power value of the fuel consumed.

Thermal efficiency = Indicated energy/total energy supplied by the fuel …………… (eq. 3.4)

- For Otto cycle thermal efficiency is 50-54%
- For diesel cycle thermal efficiency is 32-34%
- For dual combustion cycle thermal efficiency is 42%

b. Mechanical efficiency

It is the ratio of the brake power to indicate power.

Part of the gross indicated work per cycle or power is used to expel exhaust gases and induct fresh charge. An additional portion is used to overcome the friction of the bearings, pistons, and other mechanical components of the engine, and to drive the engine accessories.

Mechanical efficiency = Brake power/Indicated power It is normally between 85-95% ...…..(eq. 3.5)

c. Indicated work per cycle

Pressure data for the gas in the cylinder over the operating cycle of the engine can be used to calculate the work transfer from the gas to the piston. The cylinder pressure and corresponding cylinder volume throughout the engine cycle can be plotted on P-V diagram. The indicated work per cycle is obtained by integrating around the curve to obtain the area enclosed on the diagram. With two stroke cycles this application is straightforward.

With the addition of inlet and exhaust strokes for the four stroke cycle, some ambiguity is introduced as two definitions of indicated output are in common use. These will be defined as:

Gross indicated work per cycle: Work delivered to piston over the compression and expansion strokes only.

Net indicated work per cycle: Work delivered to piston over the entire four-stroke cycle.

d. Mean effective pressure

It is the average pressure during the power stroke minus the average pressure during other strokes. This pressure actually forces the piston down during the power stroke.

While torque is a valuable measure of a particular engine's abiLy to do work, it depends on engine size. A more useful relative engine performance measure is obtained by dividing the work per cycle by the cylinder volume displaced per cycle. The parameter so obtained has units of force per unit area and is called the mean effective pressure (mep).

e. Specific fuel consumption

In engine tests, the fuel consumption is measured as a flow rate. A more useful parameter is the specific fuel consumption. It measures how efficiently an engine is using the fuel supplied to produce work:

$$\mathrm{SFC} = m_f / P \quad \ldots\ldots\ldots\ldots \text{(eq. 3.6)}$$

Where,

m_f is flow rate; and

P = power out put.

Low values of SFC are obviously desirable. For SI engines typical best values of brake specific fuel consumption are about 270 gm / kW and for CI engines it is about 200 gm/kW.

e. Air/Fuel and Fuel/Air ratio

In engine testing, both the air mass flow rate and fuel mass flow rate are:

Normally measured the ratio of these flow rates is useful in defining engine-operating conditions.

The normal operating range for a conventional SI engine using gasoline fuel is 11< A/F < 19 and for CI engines with diesel fuel, it is 17 < nf < 71.

f. Volumetric efficiency

It is the ratio of actual weight of air introduced by the engine on the suction stroke to the theoretical weight of air that should have been introduced by filling the piston displacement volume with air at atmospheric pressure and temperature.

The intake system - the carburetor, the throttle plate (in a spark ignition engine), intake manifold, intake valve, intake port - restricts the amount of air which an engine of given displacement can induct. The parameter used to measure the effectiveness of an engine's induction process is the volumetric efficiency. Volumetric efficiency is only used with four-stroke cycle engine, which have a distinct induction process. It is obtained as the volume flow rate of air into the intake system divided by the rate at which volume is displaced by the piston.

Engine specific weight / volume

Engine specific weight and bulk volume for a given rated power are important in many applications. Two parameters useful for comparing these attributes from one engine to another are:

Specific weight = engine weight / rated power Specific weight = engine volume / rated power

g. Specific emission

Levels of emissions of oxides of nitrogen, Carbon monoxide, unburned hydrocarbons, and particulate are important engine operating characteristics.

The concentrations of gaseous emissions in the engine exhaust gases are usually measured in parts per million or percent by volume. Normalized indicators of emissions levels are more useful, however, and two of these are in common use.

3.2.3 Losses Incurred During the Process Cycle of an IC Engine

During this process there are losses in the system.

- Thermal losses
- Frictional losses

Thermal Losses

- Radiation losses
- Cooling system losses
- Exhaust losses
- Unaccounted for (dissociation, imperfect combustion) Frictional Losses

The friction losses are introduced due to the relative motion in different components of the engine.

- The friction in between the different parts of the engine.
- Pumping losses (suction, compression and exhaust).
- Windage losses (unaccounted)

3.3 Analysis of test results and wears assessment of these engines

3.3.1 Wear assessment of the stationary diesel engines

3.3.2 Vernier caliper

The main use of the vernier caliper is to measure the internal and the external diameter of an object. The user first read the finely marked "fixed" scale reading shown in Plate 3.11.

It was used to measure inside big end diameter of piston. It was also used to measure thickness of piston ring and oil ring shown in Plate 3.12 and 3.13.

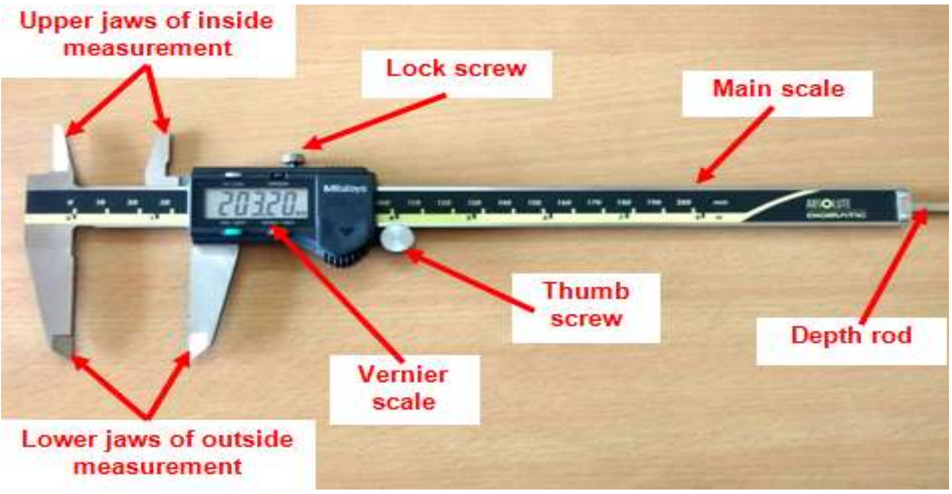

Plate 3.11: Vernier Caliper

Plate 3.12 Measurement of Inside Diameter of Big End Bearing of Piston

Plate 3.13 Measurements of Piston Ring and Oil Ring

3.3.3 Cylinder bore gauge

To measure the diameter of cylinder bore, a special tool required called Cylinder Bore Gauge. This tool measured the diameter of the cylinder with precision of about 0.01 mm shown in Plate 3.14.

- Dial gauge: This component showed the size scale with units of mm. This component served to hold the dial gauge remain silent when bore gauge was used.
- Grip: This component served as a handle when applying a bore gauge.
- Replacement rod: This was the main component of bore gauge of no 8 and it varied according to the difference in the diameter of cylinder.
- Replacement washer: It was similar to replacement rod, but replacement washer was of small thickness about 0.5 mm.
- Measuring point: In the form of a bulge, when pressed it moved the needle of dial gauge.

Plate 3.14 Cylinder Bore Gauge

Adjustment of the bore gauge cylinder

- To make the adjustment, there was need to know the normal specification of cylinder diameter; and this was served by using directly vernier caliper.

- Continued to assemble the cylinder bore gauge. Started with installation of the right replacement rod on the bore gauge. The diameter of the cylinder was 62.05 mm. So, a replacement rod with a length of 60 mm plus and replacement washer with a thickness of 3 mm was used. Total length of bore gauge used was 63.00 mm.

- After installment, the dial gauge was set to zero scale using micrometer. Position out micrometer at size 62.05 mm and the cylinder bore gauge (replacement rod parts) on timble micrometer. Then the measuring point moved resulted in the movement of needle of dial gauge. Fixed the dial gauge scale to zero position on the dial gauge needle.

Measurement of Cylinder block diameter

- Measurement was done inside the cylinder. There were two measurement positions of cylinder that was on the X and Y axes of each upper, middle and lower positions. The X axis was the line that intersected the engine horizontally. While the Y axis was the line that cut the machine vertically or elongated measurement cylinder shown in Plate 3.15.

Plate 3.15 Cylinder Bore Measurement

- Placed the cylinder bore gauge into the cylinder. This tool was shift to right and left while watching the dial indicator. Attention was given to the farthest point of moving needles. Because this point showed the difference in diameter of the cylinder with a standard diameter. The measurement was performed at each position of cylinder bore.

- Calculated the wear assessment by finding out the difference between the measurement of X axis and Y axis at each position. The tautness was measured by finding the difference in measurement of the top and bottom diameters on one axis.

3.3.4 Fuel injector pressure test

Testing of fuel injector was carried out as follows. Mounted the fuel injector in its test rig and connected with oil supply. Hands were not placed under the injector spray because high velocity oil jet can penetrate the skin and cause blood poisoning. For injector priming shut off valve was open, operated the pump lever to prime the injector, shown in Plate 3.16.

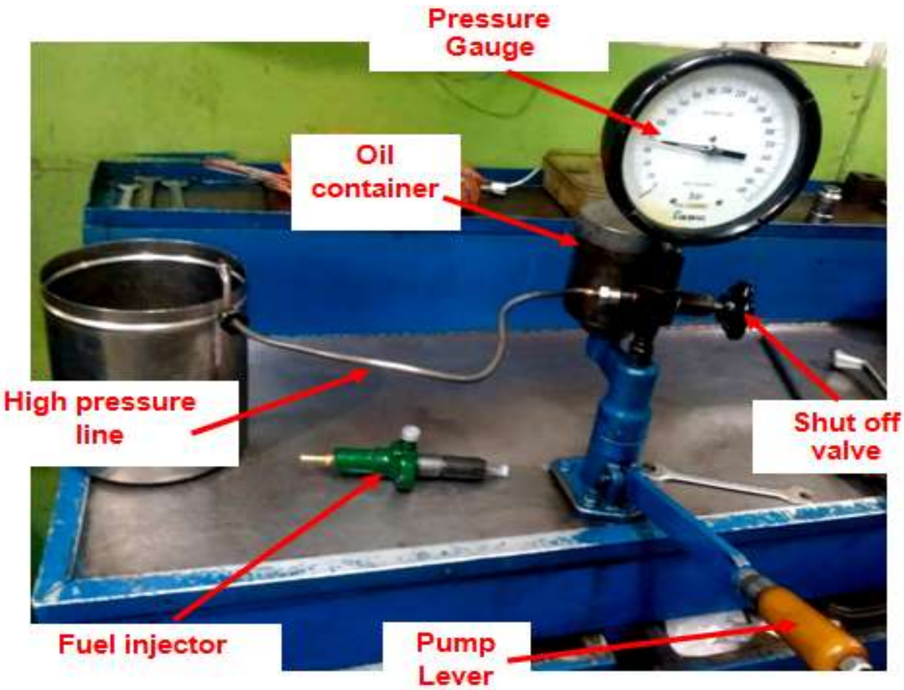

Plate 3.16 Fuel Injector Pressure Tester

Operated the pump rapidly for several strokes. The injector opened by high pitch chatter and fuel emitted in a fine cloud. After the injector opened, it was checked to make sure the pressure not fall off too quickly.

To test for the tightness between the nozzle needle and seat, the hand pump was operated slowly to increase the pressure until it was just below opening pressure. Maintained the pressure for a few seconds and ensured injector was not dripping shown in Plate 3.17.

Plate 3.17 Fuel Injector Pressure Measurement

Blanked off one of the fuel valve cooling connections and filled the injector cooling space with water or fuel, depending upon the cooling medium. Connected a low pressure air supply to the other connection. Leave the air on for a short period of time and tested for internal or external leakage. Testing of fuel injector was usually carried out at an interval of 2000 running hours for marine diesel engines.

3.3.5 Feeler gauge

Feeler gauge was a tool used to measure gap widths. Feeler gauge were mostly used in engineering to measure the clearance between two parts, shown in Plate 3.18.

Plate 3.18 Feeler Gauge

This was consisted of a number of small lengths of steel plates and different thickness with measurement mark on each piece. They were flexible enough that, to stacked together to gauge intermediate values, shown in Plate 3.19.

Plate 3.19 Gap Measurement by Feeler Gauge

A similar device with wires of specific diameter instead of flat blades was used to set the gap in piston rings and oil rings to the correct size. This was done by increasing or decreasing the gap until the gauge of the correct size fitted inside the gap.

CHAPTER-IV
RESULTS AND DISCUSSION

RESULTS AND DISCUSION

The results of the present study entitled 'Performance evaluation and wear assessment of diesel engines used in field application' are discussed in this chapter. The stationary diesel engines were tested at the Farm Machinery Testing Center, Department of Farm Machinery and Power engineering, College of Agricultural Engineering, JNKVV, Jabalpur for its performance evaluation. The wear assessment of different engine was carried out after the performance evaluation on engine test rig. Results obtained from the experiments are presented in this chapter with the help of appropriate tables and suitable figures under the following major headings:

i. Testing of different stationary diesel engines used in field application.

ii. Performance evaluation of tested stationary diesel engines.

iii. Analysis of the performance parameter by heat balance method.

iv. Wear assessment of the tested stationary diesel engines.

4.1 Testing of the different stationary diesel engines used in field application

The performance of diesel engines was measured on the engine test rig of suitable size. The engine performance in terms of power development, thermal and mechanical efficiency depends upon operating, environmental conditions and on the load being applied. In order to achieve higher efficiency and power output the engine was need to be tested and made measurements of related parameter that reflect the performance of engine. Measurements of break power were one of the most important parameter in the test schedule of the engine. It involved the determination of torque and angular speed of engine output shaft (Ganeshan 2003). The torque measuring device was an eddy current dynamometer in test rig.

The engines ware properly coupled to the dynamometer with the help of standard flanged shaft. The dynamometer rotor was driven by the engine under test and it was electromagnetically engaged to way stator. The break load was applied with the help of electrical circuit on control panel and was displayed on the screen at the setup.

The eddy current dynamometer is basically an absorption type dynamometer. The power absorbed was dissipated as heat in the cooling chamber water and the inlet and outlet water temperature was also recorded through a calorimeter.

The break power was calculated by the software as given formula:

$$BP = \frac{W \times N}{K \times 60} \text{ ……….. (eq. 3.7)}$$

Where,

K = Dynamometer constant (159.00);

W = Break load applied, kN; and

N = Engine speed, RPM.

The engine A was mounted on the 5 kW rating dynamometer with separate test rig and data acquisition system. The engine B and C was mounted on the 7.5 kW rating dynamometer with another separate test rig and data acquisition system. The engines were operated during the testing within the suitable or useful range of speed from no load engine speed to full load engine speed at maximum power. These engines are for stationary purpose and were having constant speed governor. These engines are designed for fixed speed operation that is known as rated engine speed which was 1500 rpm for all three engines. The engines were tested in the actual environmental conditions without controlling the room temperature to simulate the actual field condition where the engines actually run throughout or any time in the working hours. Hence, the results of these engines were independent and the performances obtained have direct relation with the temperature along with their physical condition.

It was noted that there was a certain engine speed within the speed range of particular engine, at which the fuel injected in the cylinder was maximum. At this point, maximum force may therefore was exerted on the piston. For all practical purposes, the torque or engine capacity to do work was also found to be maximum at this point.

The total energy input (H_{in}, kW) was calculated from fuel calorific value which results indicated horse power in the cylinder and the output energy in useful form known as brake horse power at the flywheel or crank shaft. The difference in the energy was found to be lost in friction and other system of engine by conduction and convection. It was observed that at no load speed (high idle speed) some residual torque was there which may be due to friction of internal components and inertia of rotating components of engine and dynamometer both. Therefore at this speed the engine demonstrated some brake horse power and fuel consumption to complete the above requirements. During the test the no load engine speed was also observed at the end of full load or maximum torque observations in almost similar condition to verify the initial no load observation.

4.2 Performance evaluation of tested stationary diesel engines

The power output of diesel engine at each speed within the useful range was varied and it was the maximum usable values. The total fuel consumption and specific fuel consumption was also varied with load and speed the performance of engine depends on inter relationship between power developed, speed and the specific fuel consumption at each operating condition with in the useful range of speed and load. The following factors were considered in evaluating the performance of these engines;

1. Maximum power and brake load (torque) obtained at each speed.
2. Brake specific fuel consumption, brake thermal efficiency with respect to break load.
3. Exhaust gas temperature with respect to brake load.

4.2.1 Performance of engine "A"

The engine “A” having declared rated power of 3.72 kW at 1500 rated engine speed, was tested after all preliminary services and maintenance in the lab with same physical components and dimensions. This engine was run for about 350 hours (as per the record) till the time of selection and was assumed to be 2-3 year old. The observations were taken after sufficient warm-up and stabilized condition. The observed parameters taken for study are given below in table 4.1.

Table 4.1 : Performance result of engine "A"

W (Nm)	Speed (rpm)	Power (kw)	TFC (g/min)	BSFC, g/kW-h	nBTH (%)	T1 (^oC)	T8 (^oC)
2.1	1590	0.35	40	684	11.93	129	34
3.2	1587	0.53	52	588	13.95	131	34
4.4	1584	0.73	68	558	14.65	142	37
8.6	1572	1.41	116	468	17.56	174	39
12.8	1566	2.10	156	444	18.36	204	39
17	1515	2.70	178	396	20.68	263	41
20	1503	3.10	191	366	22.13	320	42
21.5	1479	3.33	214	384	21.15	385	43
21.9	1448	3.32	221	396	20.48	402	43

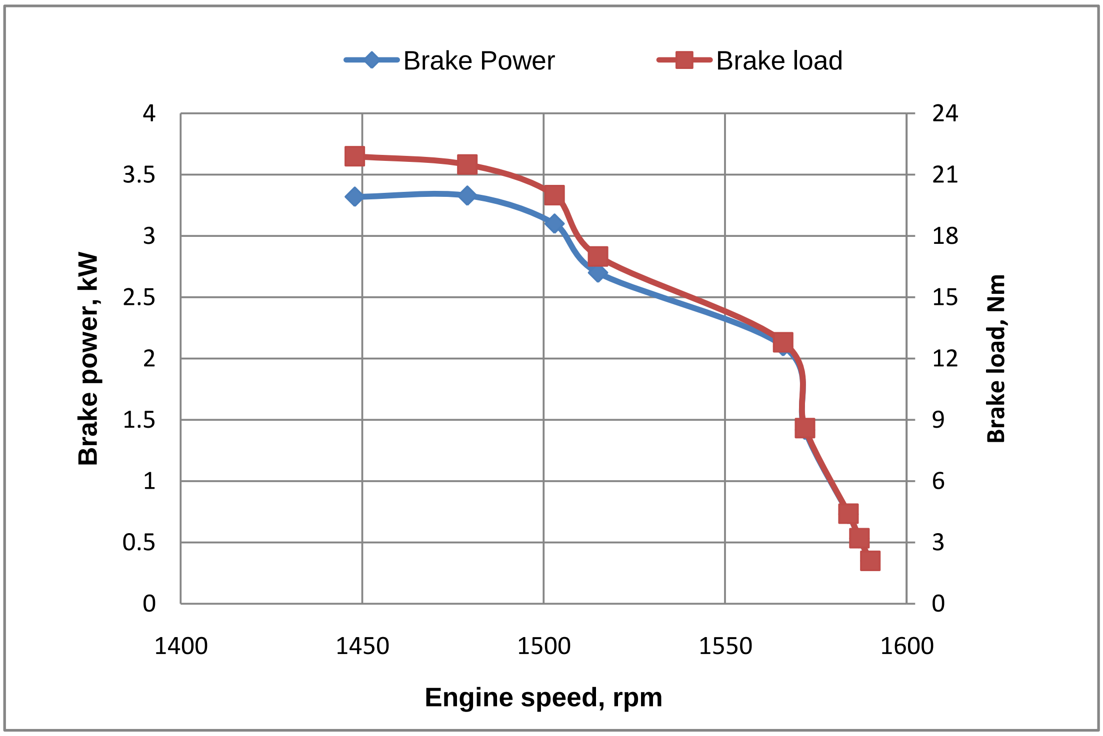

Fig. 4.1: Variation of brake power kW and brake load Nm with respect to engine speed rpm

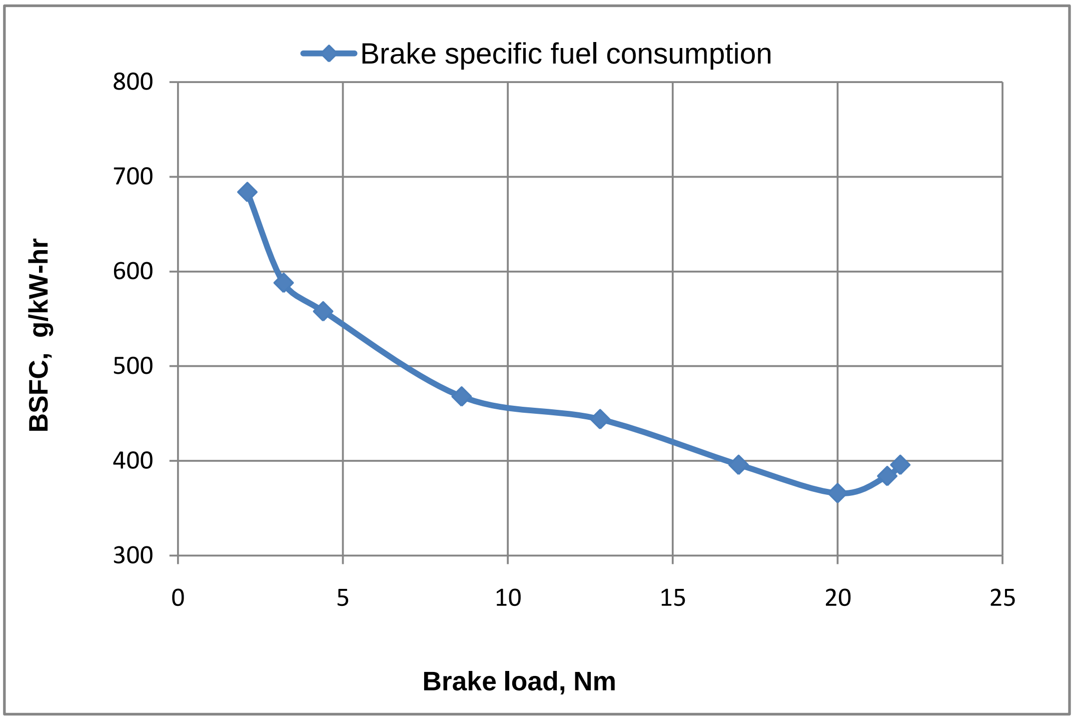

Fig 4.2: Variation of BSFC g/kW-hr. respect to brake load Nm

A) Brake power and brake specific fuel consumption

From the figure 4.1 it was found that as the brake load was increased from no load engine speed (1590 rpm) for this engine up-to the 10% over load engine rpm (1448 rpm) the engine brake power was also increased gradually. The maximum rated power was found to be 3.1 kW at the rated engine rpm of 1500 which was considered lower side as compare to the declared value of 3.72 kW. Hence, the rated power was found to be reduced by 16.7% after completing part of engine life. The further loading of engine beyond 1500 rpm brake load that was treated as 100%, the sudden drop of engine rpm was noticed and was reached up-to 1448 rpm. It may be due to governing action which was not able to maintain the rpm and load. In case of sudden over load the engine must maintain the rpm and produce sufficient power to cope-up this situation. Similarly from the figure 4.2 it was found that the BSFC was the fuel consumed by the engine per unit of power developed during the test. It can be seen from the graph that as brake power was increased, BSFC decreased to minimum (366 g/kW-h) at 1503 rpm (100%

load) and decreasing trend was found with increasing brake power. The quantity of fuel consumed was increase with engine speed. At low speed heat lose to combustion chamber walls was greater and combustion efficiency poor resulted in high fuel combustion for the power development. At the high speed friction power increases at rapid rate result in the slow increasing brake power than fuel consumption with a consequent in BSFC. The further loading of engine was resulted into increased BSFC. The observed rated BSFC was found much higher side against the declared BSFC of 230 g/kW-h. The BSFC was found to be increased by 59% after completing part of engine life. Hence, above both parameters were not matched with the declaration of performance when the engine was new.

B) Brake thermal efficiency and exhaust gas temperature

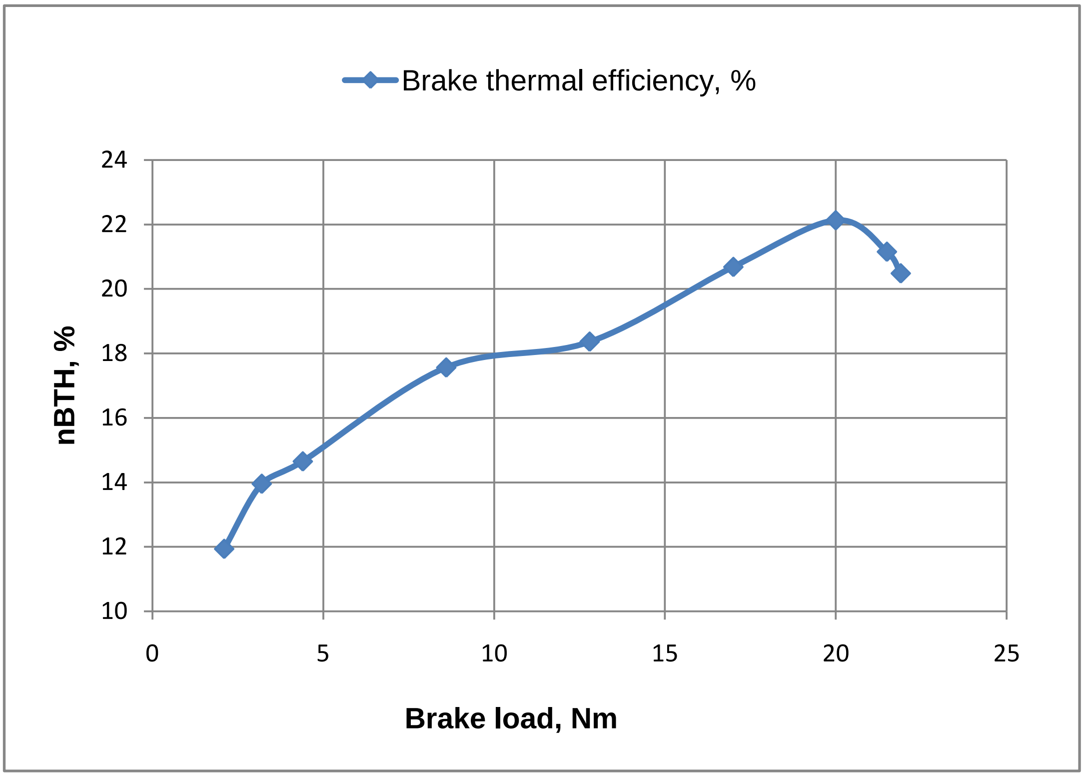

Fig 4.3: Variation of nBTH% with respect to brake load Nm

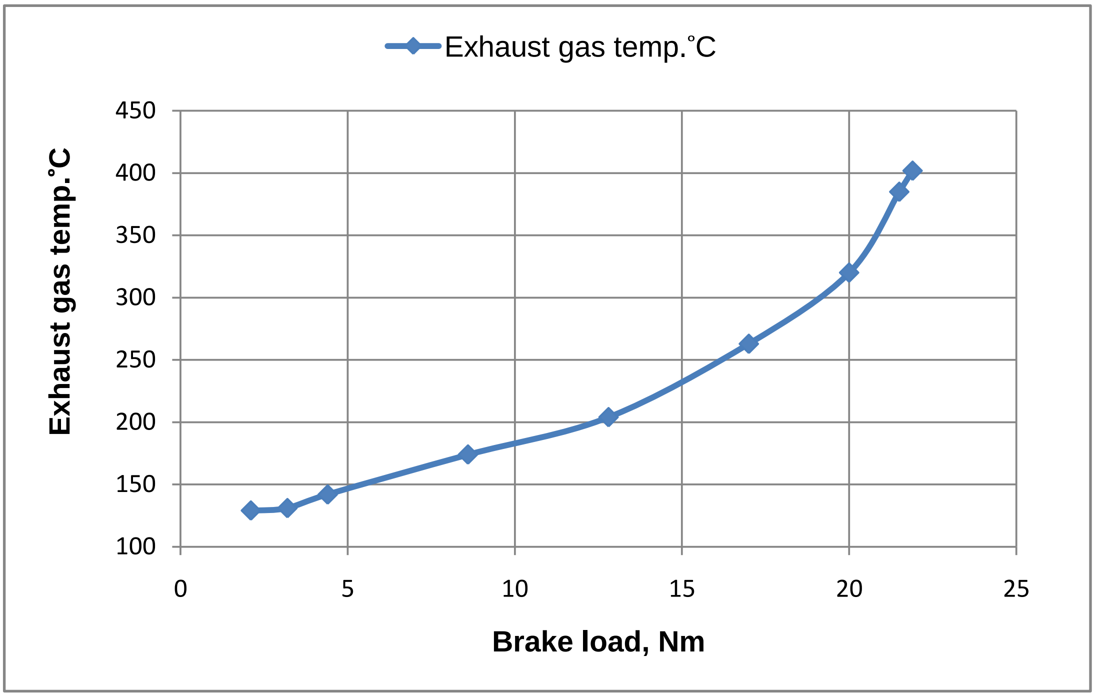

Fig 4.4: **Variation of exhaust gas temp. ^{o}C with respect to brake load Nm**

The brake thermal efficiency (BTE) gives an idea of the output generated by the engine with respect to heat supply from the fuel. In modern diesel engines the indicated thermal efficiency for compression ignition engine was estimated by the researchers about 36% and even more with high compression ratio. The figure 4.3 represented brake thermal efficiency in percentage with respect to brake load. For the engine tested the increase in brake thermal efficiency was found with increases in brake load and consequently the power. The BTE was found less (11.93%) at low load and increased at rated output of tested engine (22.13%). Further increase in load caused slightly increased in power but total fuel consumption is comparatively higher and due to this the BTE was found to be reduced (20.48%).

Exhaust gas temperature (EGT) is important parameter in engine performance check, it is indication of how hot the combustion process in the cylinder, and the amount of after burning that is occurring in the exhaust manifold. EGT is directly related to the air-fuel ratio. Higher the ratio in a diesel, higher would be the EGT. The air-fuel ratio depends on the engine speed, brake load and surrounding condition of the engine etc. from the figure 4.4 it was concluded that with increase in brake load the EGT was also increased but exponentially at higher brake load and power side. At rated performance point the EGT was 320^{o}C which may be considered optimum at 42^{o}C ambient temperature at test site. Whereas, just after this point the EGT was increased to 402^{o}C which may be due to higher intake of fuel or high heating value at this much load. High temperature inside the combustion chamber and manifold may create thermal stresses on engine parts and reduced mechanical strength.

4.2.2 Performance of engine B

The engine "B" having declared rated power of 7.4 kW at 1500 rated engine speed was tested after all preliminary service and maintenance in the lab with same physical components and dimension. This engine was run for about 1800 hours till the time of selection and was assumed to be 4-5 year old. The observations were taken after sufficient warm-up and stabilized condition. The observed parameters taken for study are given below in table 4.2.

Table 4.2 : Performance result of engine "B"

W (Nm)	Speed (rpm)	Power (kW)	TFC (g/min)	BSFC (g/kW-h)	nBTH (%)	T1 (^{o}C)	T8 (^{o}C)
1.2	1618	0.20	72	2128	3.84	123	43
2.4	1616	0.40	146	2152	3.8	130	43
5.8	1612	0.98	168	1029	7.95	156	44
6.1	1567	1.00	151	898	9.11	174	44
9.1	1552	1.48	148	600	13.64	191	46
14	1538	2.25	162	431	19	225	46
20.3	1525	3.24	220	407	20.11	257	47
24.8	1515	3.93	196	299	27.4	289	47
26.2	1506	4.13	204	296	27.65	307	47
29.6	1507	4.67	216	277	29.52	320	48
37.5	1493	5.81	279	288	28.00	365	48
37.5	1480	5.81	310	320	25.59	409	49
39.4	1473	6.07	332	328	24.97	426	50

A) Brake power and BSFC

From the figure 4.5 it was found that as the brake load was increased from no load engine speed (1618 rpm) for this engine up-to the further load engine rpm (1493 rpm) the engine brake power was also increased gradually. The maximum rated power was found to be 5.8 kW at the rated engine rpm of 1493 which was considered lower side as compare to the declared value of 7.4 kW. Hence the rated power was found to be reduced by 21.62% after completing part of engine life. The further loading of engine beyond 1500 rpm brake load that was treated as 100%, the sudden drop of engine rpm was noticed and was reached up-to 1493 rpm. It may be due to governing action which was not able to maintain the rpm and load. In case of sudden over load the engine must maintain the rpm and produce sufficient power to cope-up this situation.

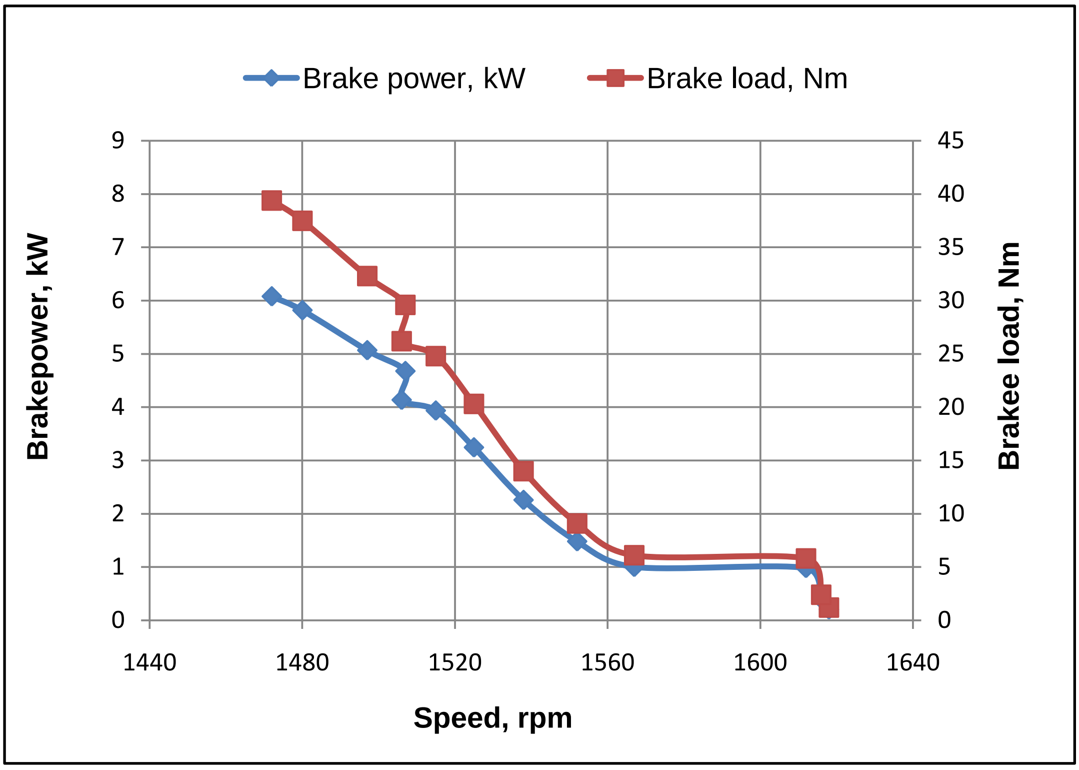

Fig. 4.5: Variation of brake power kW and brake load Nm with respect to engine speed rpm

Similarly from the figure 4.6 it was found that the BSFC was the fuel consumed by the engine per unit of power developed during the test. It can be seen from the graph that as brake power was increased, BSFC decreased to minimum (288 g/kW-h) at 1493 rpm (100% load) and decreasing trend was found with increasing brake power. The quantity of fuel consumed was increase with engine speed. At low speed heat lose to combustion chamber walls was greater and combustion efficiency poor resulted in high fuel combustion for the power development. At the high speed friction power increases at rapid rate result in the slow increasing brake power than fuel consumption with a consequent in BSFC. The further loading of engine was resulted into increased BSFC. The observed rated BSFC was found much higher side against the declared BSFC of 190 g/kW-h. The BSFC was found to be increased by 34% after completing part of engine life. Hence, above both parameters were not matched with the declaration of performance when the engine was new.

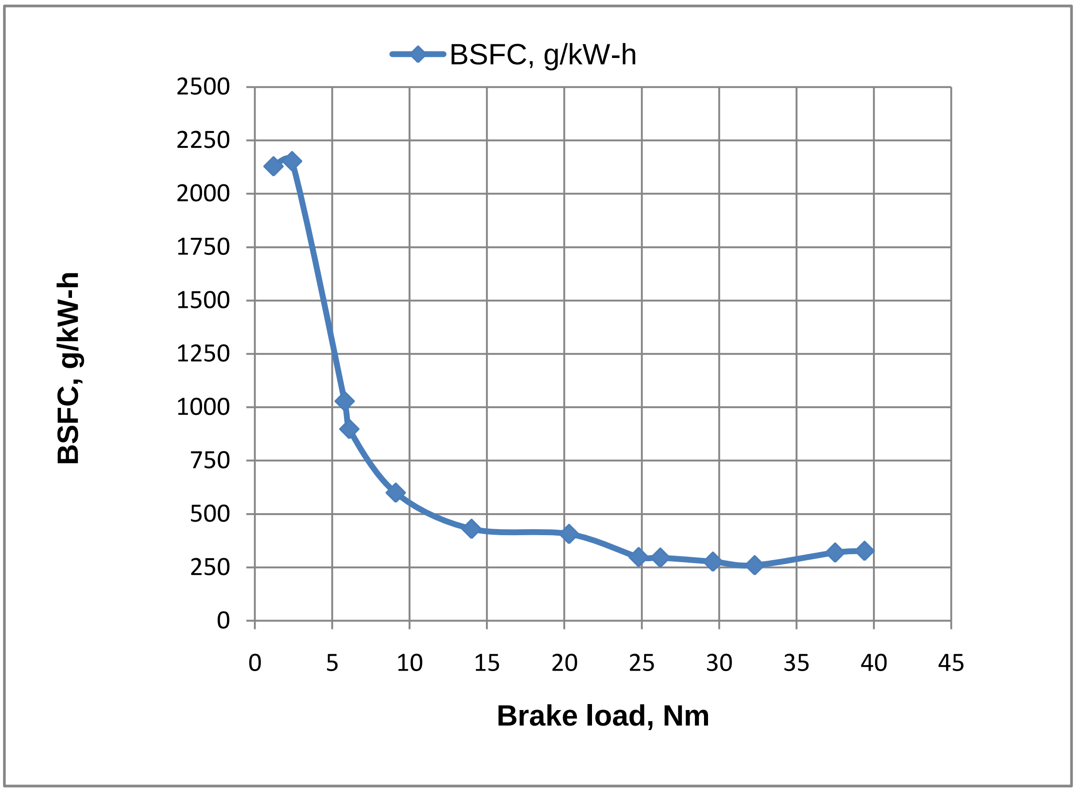

Fig 4.6: Variation of BSFC g/kW-hr. respect to brake load Nm

B) **Brake thermal efficiency and exhaust gas temperature**

The brake thermal efficiency (BTE) gives an idea of the output generated by the engine with respect to heat supply form of the fuel. In modern diesel engines the indicated thermal efficiency for compression ignition engine was estimated by the researchers about 36% and even more with high compression ratio. The figure 4.7 represented brake thermal efficiency in percentage with respect to brake load. For the engine tested the increase in brake thermal efficiency was found with increases in brake load and consequently the power. The BTE was found less (3.84 %) at low load and increased at rated output of tested engine (28%). Further increase in load caused slightly increased in power but total fuel consumption is comparatively higher and due to this the BTE was found to be reduced (32.88%).

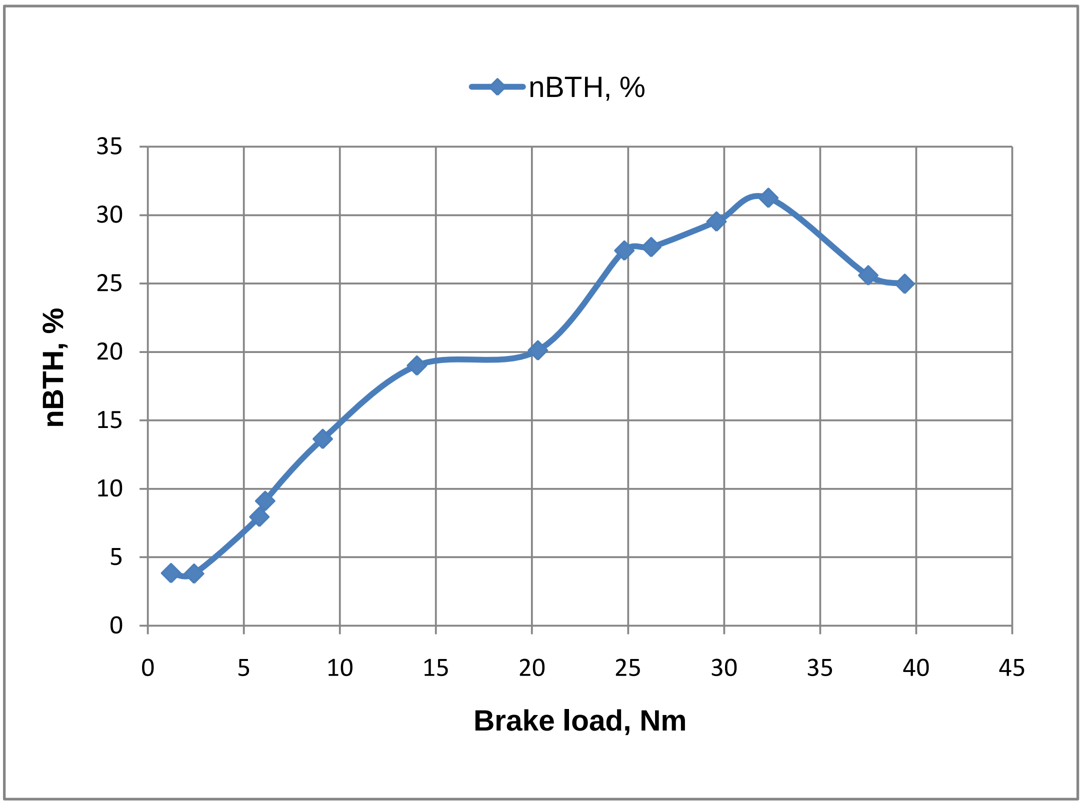

Fig 4.7: Variation of nBTH% with respect to brake load Nm

Exhaust gas temperature (EGT) is important parameter in engine performance check, it is indication of how hot the combustion process in the cylinder and the amount of after burning that is occurring in the exhaust manifold. EGT is directly related to the air-fuel ratio. Higher the ratio in a diesel, higher would be the EGT. The air-fuel ratio depends on the engine speed, brake load and surrounding condition of the engine etc. from the figure 4.8 it was concluded that with increase in brake load the EGT was also increased but exponentially at higher brake load and power side. At rated performance point the EGT was 365^{o}C which may be considered optimum at 48^{o}C ambient temperature at test site. Whereas just after this point the EGT was increased to 426^{o}C which may be due to higher intake of fuel or high heating value at this much load. High temperature inside the combustion chamber and manifold may create thermal stresses on engine parts and reduced mechanical strength.

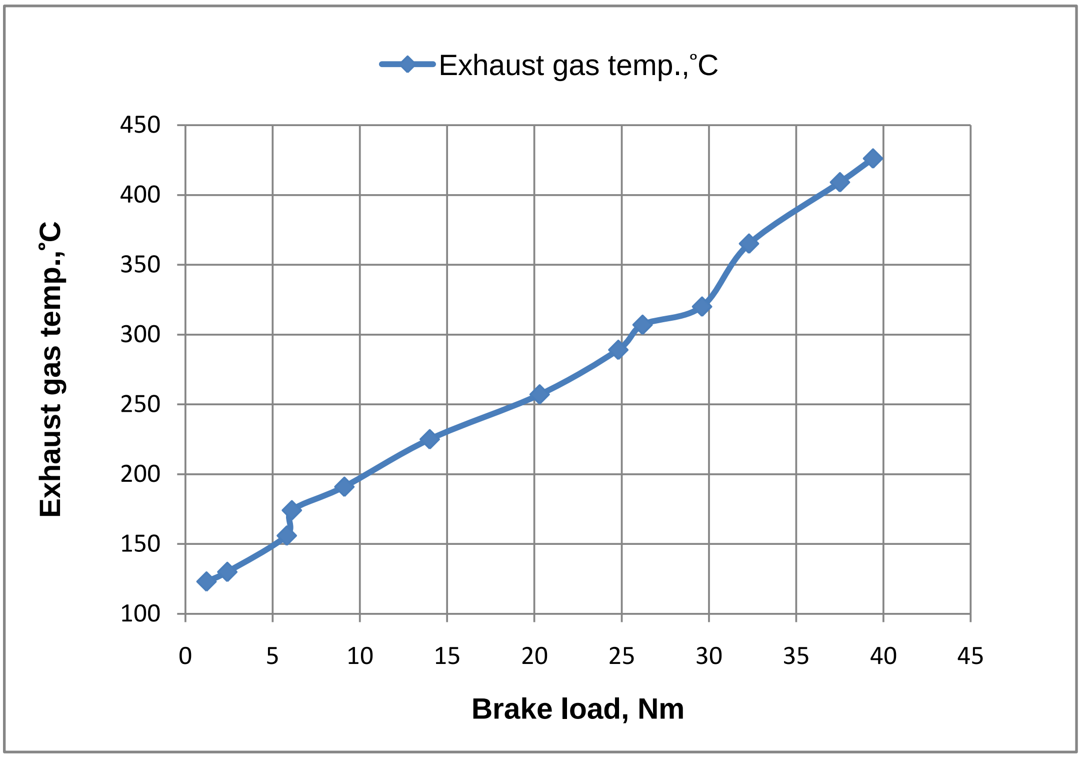

Fig 4.8: Variation of exhaust gas temp. ^{o}C with respect to brake load Nm

4.2.3 Performance of engine "C"

The engine B having declared rated power of 4.81 kW at 1500 rated engine speed was tested after all preliminary services and maintenance in the lab with same physical components and dimensions. This engine was run for about 2300 hours till the time of selection and was assumed 6-7 year old. The observations were taken after sufficient warm-up and stabilized condition. The observed parameters taken for study are given below in table 4.3.

Table 4.3: Performance result of engine "C"

W (Nm)	Speed (rpm)	Power (kW)	TFC (g/min)	BSFC (g/kW-min)	nBTH (%)	T1 (^oC)	T8 (^oC)
1.7	1568	0.27	74	1505	5.44	138	40
3.2	1557	0.52	106	1218	6.72	172	41
6.9	1531	1.10	112	607	13.48	187	42
10.7	1521	1.70	124	436	18.76	205	43
11.2	1507	1.76	126	427	19.15	216	43
14.2	1503	2.23	152	408	20.07	224	44
15.5	1488	2.41	156	387	21.14	241	45
25.8	1466	3.96	236	357	22.91	336	46
28.2	1453	4.29	276	384	21.22	365	48

A) Brake power and BSFC

From the figure 4.9 it was found that as the brake load was increased from no load engine speed (1568 rpm) for this engine up-to the further load engine rpm (1453 rpm) the engine brake power was also increased gradually. The maximum rated power was found to be 2.237 kW at the rated engine rpm of 1503 which was considered lower side as compare to the declared value of 4.81 kW. Hence the rated power was found to be reduced by 53.49 % after completing part of engine life. The further loading of engine beyond 1503 rpm brake load that was treated as 100 %, the sudden drop of engine rpm was noticed and was reached up-to 1453 rpm. It may be due to governing action which was not able to maintain the rpm and load. In case of sudden over load the engine must maintain the rpm and produce sufficient power to cope-up this situation.

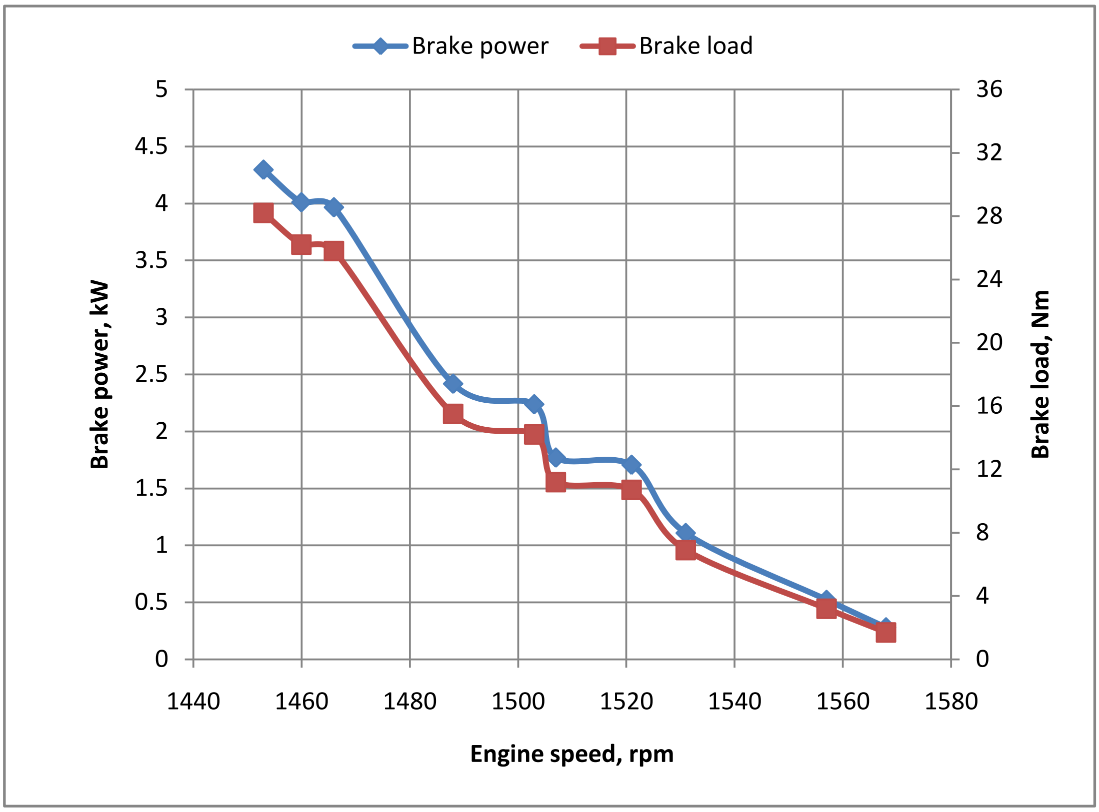

Fig. 4.9: Variation of brake power kW and brake load Nm with respect to engine speed rpm

Similarly from the figure 4.10 it was found that the BSFC was the fuel consumed by the engine per unit of power developed during the test. It can be seen from the graph that as brake power was increased, BSFC decreased to minimum (408 g/kW-h) at 1503 rpm (100% load) and decreasing trend was found with increasing brake power. The quantity of fuel consumed was increase with engine speed. At low speed heat lose to combustion chamber walls was greater and combustion efficiency poor resulted in high fuel combustion for the power development. At the high speed friction power increases at rapid rate result in the slow increasing brake power than fuel consumption with a consequent in BSFC. The further loading of engine was resulted into increased BSFC. The observed rated BSFC was found much higher side against the declared BSFC of 180 g/kW-

h. The BSFC was found to be increased by 55 % after completing part of engine life. Hence, above both parameters were not matched with the declaration of performance when the engine was new.

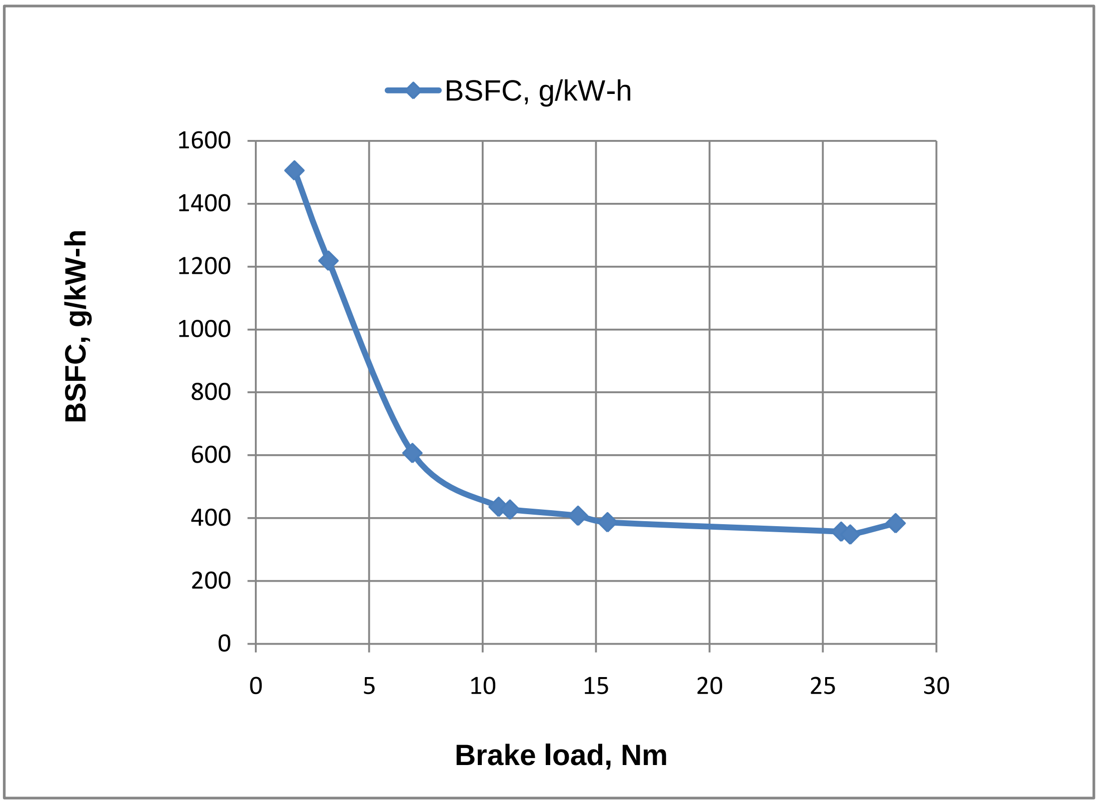

Fig 4.10: Variation of BSFC g/kW-hr. respect to brake load Nm

B) Brake thermal efficiency and exhaust gas temperature

The Brake Thermal Efficiency (BTE) gives an idea of the output generated by the engine with respect to heat supply form of the fuel. In modern diesel engines the indicated thermal efficiency for compression ignition engine was estimated by the researchers about 36 % and even more with high compression ratio. The figure 4.11 represented brake thermal efficiency in percentage with respect to brake load. For the engine tested the increase in brake thermal efficiency was found with increases in brake load and consequently the power. The BTE was found less (5.44%) at low load and increased at rated output of tested engine (20.07%). Further increase in load caused slightly increased in power but total fuel consumption is comparatively higher and due to this the BTE was found to be reduced (21.22%).

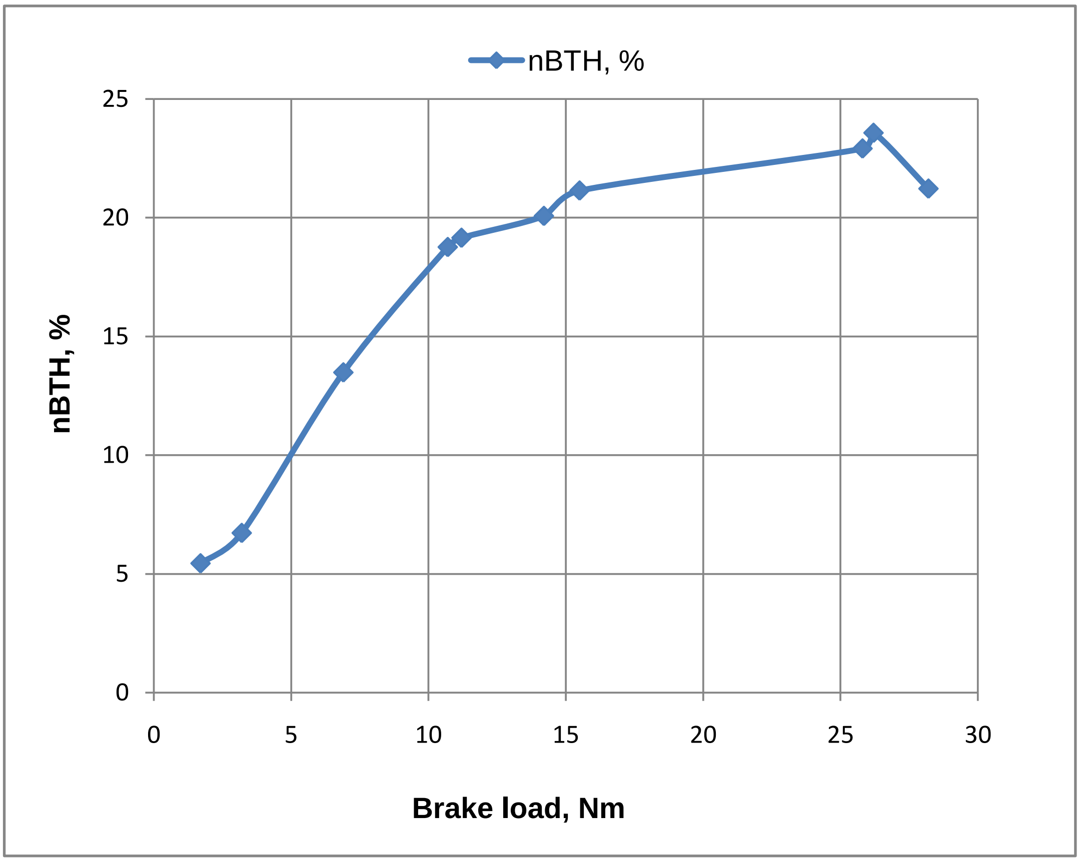

Fig 4.11: Variation of nBTH% with respect to brake load Nm

Exhaust Gas Temperature (EGT) is important parameter in engine performance check, it is indication of how hot the combustion process in the cylinder, and the amount of after burning that is occurring in the exhaust manifold. EGT is directly related to the air-fuel ratio. Higher the ratio in a diesel, higher would be the EGT. The air-fuel ratio depends on the engine speed, brake load and surrounding condition of the engine etc. from the figure 4.12 it was concluded that with increase in brake load the EGT was also increased but exponentially at higher brake load and power side. At rated performance point the EGT was 224^{o}C which may be considered optimum at 44^{o}C ambient temperature at test site. Whereas just after this point the EGT was increased to 365^{o}C which may be due to higher intake of fuel or high heating value at this much load. High temperature inside the combustion chamber and manifold may create thermal stresses on engine parts and reduced mechanical strength.

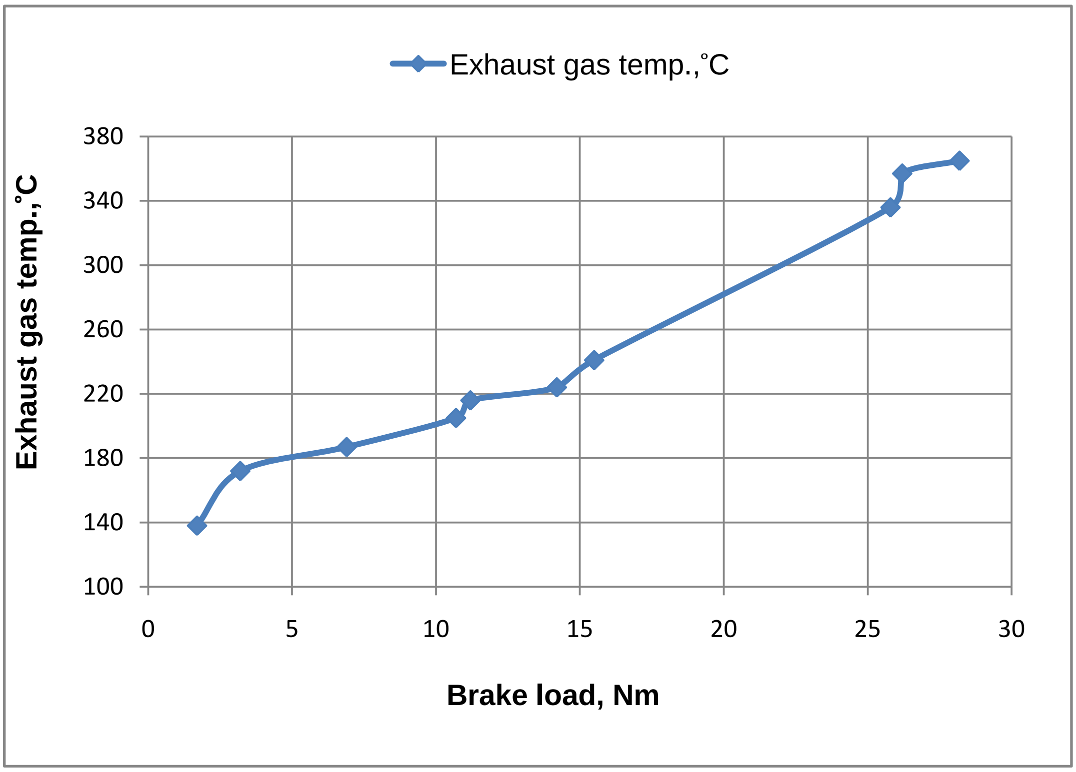

Fig 4.12: Variation of exhaust gas temp. °C with respect to brake load Nm

4.3 Analysis of the performance parameter by heat balance method

Energy supplied to an engine is the heat value of the fuel consumed. In diesel engine most heat is generated during power stroke which increases the internal temperature of the engine. To carry away the heat generated inside the engine, jackets are provided around the path through which heat travels and in these jackets coolant is filled which absorbs the heat and travels to radiator for cooling. In compression ignition engine combustion of air and fuel takes place inside the engine cylinder and hot gases are produced. The temperature of gas remains around 2200 to 2400^{o}C. At this temperature burning of lubricant oil film between moving parts and overheating may result into seizure of engine so, the temperature must be reduced at which the engine can work most efficiently. To obtain sufficient data for the preparation of heat balance sheet the test was conducted at different load. The test was carried out for the three engines of different rated horse power range. The test was helpful to observe the temperature of water used for cooling engine, exhaust gases and water flowing through calorimeter. Also the water mass flow rate through engine water jackets and calorimeter was observed. The observations were used to calculate parameters like heat input, brake power, loss of heat in cooling water, loss of heat in exhaust gases and calorimeter. These parameters were used for the preparation of heat balance sheet of the respective diesel engine.

Table 4.4: Performance result by heat balance method of engine "A"

W (Nm)	Speed (rpm)	T1 (°C)	T2 (°C)	T3 (°C)	T4 (°C)	T8 (°C)	Hin (kW)	H BP (kW)	Hexh	Hunac (kW)	TFC (kg/min)	% HBP	% Hexh	% Hunacc
1.9	1590	129	83	26.3	28.3	34	2.93	0.33	1.39	1.20	40	11.9	47.3	40.8
3.2	1587	131	88	26.2	28.3	34	3.81	0.53	1.59	1.69	52	14.0	41.7	44.3
4.4	1584	142	94	26.8	28.9	37	4.99	0.73	1.54	2.71	68	14.7	31.0	54.4
8.6	1572	174	108	26.6	29.2	39	8.07	1.41	1.79	4.86	110	17.6	22.2	60.3
12.8	1566	204	119	26.5	29.5	39	11.44	2.10	1.96	7.38	156	18.4	17.1	64.5
17	1515	263	142	26.3	30.1	41	13.05	2.7	2.34	8.01	178	20.7	17.9	61.4
20	1503	320	163	26.3	31.1	41	14.01	3.1	2.87	8.04	191	22.1	20.5	57.4
21.5	1479	376	190	24.9	30.5	43	15.69	3.33	3.37	8.99	214	21.2	21.5	57.3
21.9	1448	420	213	25	31.4	43	16.21	3.32	3.92	8.97	221	20.5	24.2	55.3

Performance of engine A in heat balance terms

From the table 4.4 and the figure 4.13 showed that the total heat energy input at rated engine speed of 1503 rpm was14.1 kW which was increased further upto 10 % overload condition at 1448 rpm. From 14.1 kW (100%) of total input the brake power was found only to be 3.1 kW (22.1%), remaining was converted to heat energy in exhaust and in some unaccounted losses i. e. 2.87 kW (20.5%) and 8.04 kW (57.4%) respectively. The figure4.13 shows that the heat energy input to the engine was increased from 2.93 kW to 16.21 kW and correspondingly the brake power was also gradually increased. It was observed that the engine lost major part of energy in the energy of exhaust (47.3%) at lower brake load than decreased considerably up to rated point (20.5%) where the fuel consumption was 191 g/min which was further increased up to 221 g/min below that speed but the power was almost constant because the energy in the exhaust was again increased up to 24.2% means part of increased fuel energy may wasted in exhaust. Since, there are always certain losses which cannot be accounted for including frictional power loss in heat balance calculations; the unaccounted heat energy loss at rated engine speed of 1503 rpm was 57.4%. The engine was air cooled so, the energy loss for air cooling system was also included in this unaccounted energy loss. It was also observed that unaccounted loss was increased in the middle brake load range than decreased after rated point which may be due to gradual increase in the temperatures and there after reduced gradient of temperatures on higher load side.

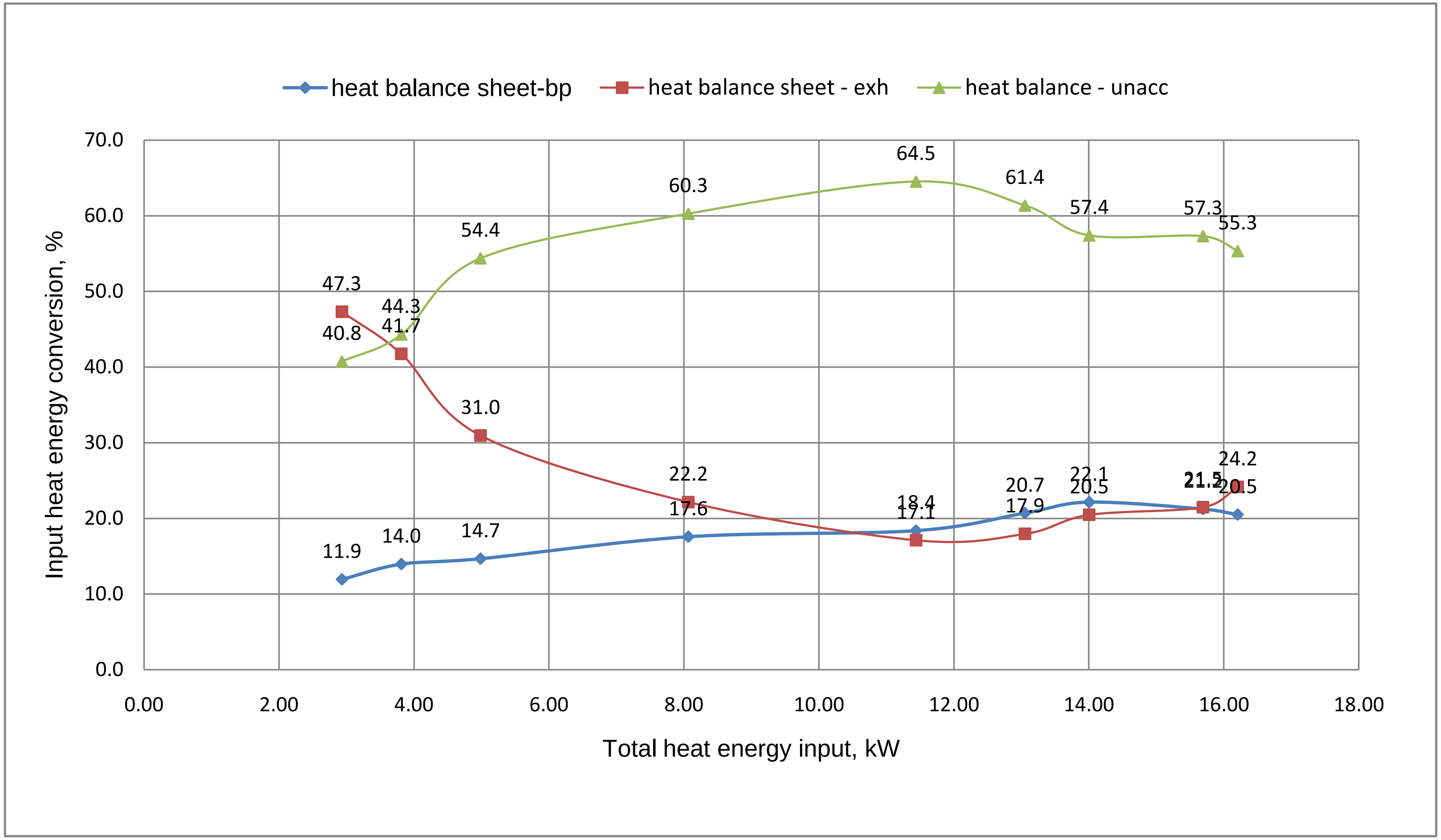

Fig 4.13: Heat balance curve for engine A

Performance of engine B in heat balance terms

The engine was also tested for heat balance observation and fuel consumption by gradually loading from no load to 110% of rated load. All the data were recorded as in performance test along with the heat value of various systems and are given in the table 4.5. The figure 4.14 showed that the total heat energy input near rated engine speed of 1500 rpm (1493, rpm) at 37.6 Nm (100% load) was 21.6 kW which was increased further up to 24.9 kW on 10 % overload condition at 1470 rpm. From 21.6 kW (100%) of total input the brake power was found only to be 5.9 kW (i.e. 27.4%), remaining was converted to heat energy in cooling system, exhaust system and unaccounted losses i.e.8.3 kW (38.4%), 5.4 kW (24.8%) and 2.04 kW (9.44%) respectively. The figure 4.14 shows that the heat energy input to the engine was increased from 5.28 kW to 20.93 kW and correspondingly the brake power was also increased proportionately. It was observed that the engine lost major part of energy in the cooling (53.6%) and exhaust (38.3 %) at lowest brake load than decreased considerably up to rated point (38.4 and 24.8 %) where the fuel consumption was 294 g/min which was further increased up to 340 g/min below that speed but the power was only increased by 0.4 kW because the energy in the exhaust and other unaccounted area was again increased that means part of increased fuel energy may wasted in exhaust and unaccounted area. Since, there are always certain losses which cannot be accounted for including frictional power loss in heat balance calculations; the unaccounted heat energy loss near rated engine speed of 1493 rpm was 9.44%. It was also observed that unaccounted loss was increased in the middle brake load range than decreased after and near rated point and was again reached to maximum of its whole range value that is 13.6%. The variation may be due to sudden increase in the temperatures and the mechanical or frictional losses in the various components at the applied load. The abnormal behavior of unaccounted loss demonstrated that the energy may be lost due to unknown internal causes due to worn or faulty system component. It was also observed that when two system losses increased just before rated speed the unaccounted loss was reduced and vice versa. Hence, it was concluded that the engine governor was not able to maintain the speed and performance near and on rated engine speed.

Table 4.5: Engine - B heat balance and fuel consumption sheet during various load application

W (Nm)	Speed (rpm)	T1 (°C)	T2 (°C)	T3 (°C)	T4 (°C)	T5 (°C)	T6 (°C)	T8 (°C)	Hin (kW)	H BP (kW)	Hexh	Hunac (kW)	Hwater (kW)	TFC (kg/min)	% HBP	% Hexh	% Hunacc	% Hwater
1.2	1615	123	83	32.4	34.6	34.7	41.2	44	5.28	0.20	2.02	0.22	2.83	72	3.84	38.30	4.26	53.6
7.6	1568	181	119	32.4	35.6	34.7	42.4	44	10.27	1.24	3.15	0.99	4.86	140	12.17	30.76	9.69	47.4
15.2	1551	219	139	32.4	36.4	32.5	42.5	44	13.57	2.47	3.93	1.21	5.94	185	18.21	29.03	8.96	43.8
23	1527	278	169	32.5	37.3	32.9	43.5	45	16.72	3.68	4.64	1.90	6.49	228	22.02	27.75	11.41	38.8
29.9	1512	332	191	32.6	38.3	33	44.6	46	18.63	4.73	5.26	1.33	7.28	254	25.44	28.28	7.17	39.1
37.6	1493	373	203	32.6	38.9	33.2	46.1	47	21.56	5.90	5.34	2.03	8.27	294	27.38	24.79	9.44	38.4
41.2	1469	418	221	32.6	40.1	33.2	47.6	47	24.93	6.34	6.35	3.38	8.84	340	25.46	25.50	13.59	35.5
5.6	1608	249	181	32.7	37.2	33.4	49	47	8.21	0.94	3.97	-1.88	5.17	112	11.49	48.34	-22.89	63.1

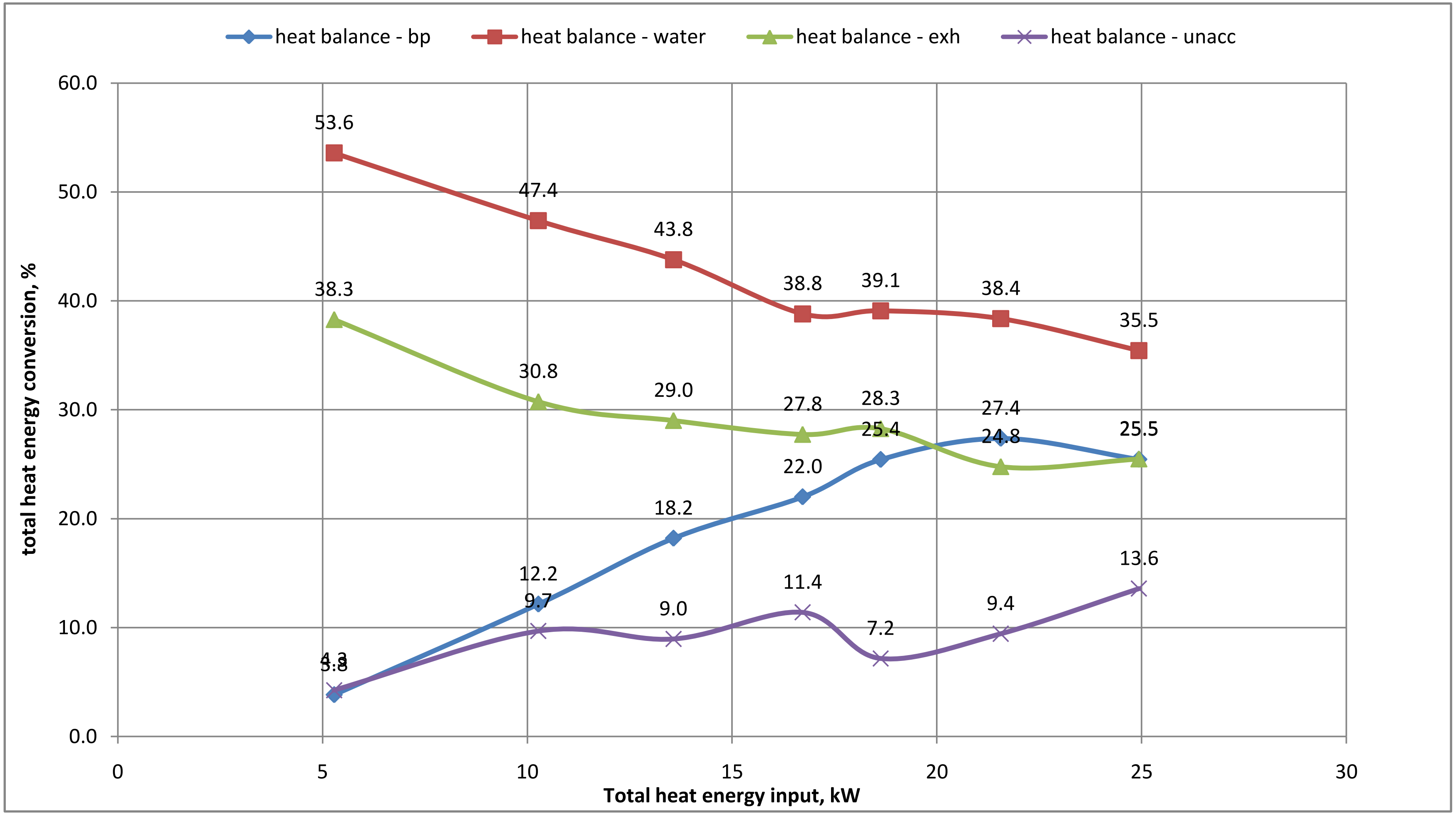

Fig 4.14: Heat balance curve for engine B

Performance of engine C in heat balance terms

The engine C was experimented for different load by taking 100% load value of 25.8 Nm at 1466 engine rpm below the rated engine speed and was considered to be operated under over load condition. Hence to assess the performance in over load condition it was tested for heat balance observation and fuel consumption by gradually loading from no load to 110% of 25.8 Nm load. All the data were recorded as in performance test along with the heat value of various systems and are given in the table 4.6. The figure 4.15 showed that the total heat energy input near rated engine speed of 1500 rpm (1495, rpm) at 20.2 Nm (80% load) was 14.01 kW which was increased further up to 26.2 kW treated as 100 % load condition near 1466 rpm. From 14.01 kW (100% energy) of total input the brake power was found only to be 4.01 kW (i.e. 23.6%), remaining was converted to heat energy in cooling system, exhaust system and unaccounted losses i.e.5.70 kW (33.5%), 2.51 kW (14.7%) and 4.79 kW (28.2%) respectively. The figure 4.14 show that the heat energy input to the engine was increased from 4.25 kW to 20.24 kW and correspondingly the brake power was also increased with variation but the maximum contribution of energy into brake power (23.6%) was observed on this assumed 100% load then dropped to 21.2%.

It was observed that the engine lost major part of energy in the cooling (14% to 35.5%) and unaccounted (28.2% to 41.8%) at throughout range of load. It was observed that the energy loss in exhaust was comparatively lower then cooling and unaccounted loss as compare to previous two tested engines. The figure shows direct relation between brake power developed and the unaccounted losses where increased losses resulted into low power development. It was also observed that unaccounted loss was remain flatter in the middle brake load range than increased again reached to is to another end maximum of its whole range value that is 34.4%. The variation may be due to sudden increase in the load and delayed response of governor in extreme range of speed of operation against the applied load. It was observed from the figure that loss in cooling reduces when the exhaust losses were dropped considerably that may be due to smaller temperature gradient in engine systems. The behavior of unaccounted loss demonstrated that the energy may be lost due to unknown internal causes due to frictional losses, incomplete combustion, and blow by, worn or faulty system component. While changing the load from one stage to next stage the governor hunting was also observed. Hence, it was concluded that the engine governing and fuel supply system was not able to maintain the load and speed consistently.

Table 4.6: Engine - C heat balance and fuel consumption sheet during various load application

W (Nm)	Speed (rpm)	T1 (^oC)	T2 (^oC)	T3 (^oC)	T4 (^oC)	T5 (^oC)	T6 (^oC)	T8 (^oC)	Hin (kW)	H BP (kW)	Hexh	Hunacc (kW)	Hwater (kW)	TFC (g/min)	% H BP	% Hexh	% Hunacc	% Hwater
3.1	1585	186	61	33.8	36.6	36.1	37.4	45	4.253	0.515	1.367	1.777	0.594	58	12.1	32.1	41.8	14.0
5.3	1572	183	62	33.8	36.6	36.1	39.3	45	5.28	0.873	1.461	0.821	1.392	72	16.5	27.7	29.4	26.4
10.3	1533	227	62	33.9	37.6	36.2	43.5	45	9.24	1.655	1.724	2.583	3.278	126	17.9	18.7	28.0	35.5
15.4	1506	257	62	33.9	38	36.2	45.2	46	12.027	2.431	2.008	3.446	4.142	164	20.2	16.7	28.7	34.4
20.2	1495	301	62	33.9	38.6	36.2	46.4	46	14.013	3.217	2.224	4.756	4.616	202	23.0	15.9	28.2	32.9
26.2	1462	366	64	34	39.4	36.3	49.1	47	17.013	4.01	2.505	4.795	5.703	232	23.6	14.7	28.2	33.5
28.2	1453	390	64	34.1	39.9	36.4	50.5	47	20.24	4.295	2.595	6.969	6.381	276	21.2	12.8	34.4	31.5

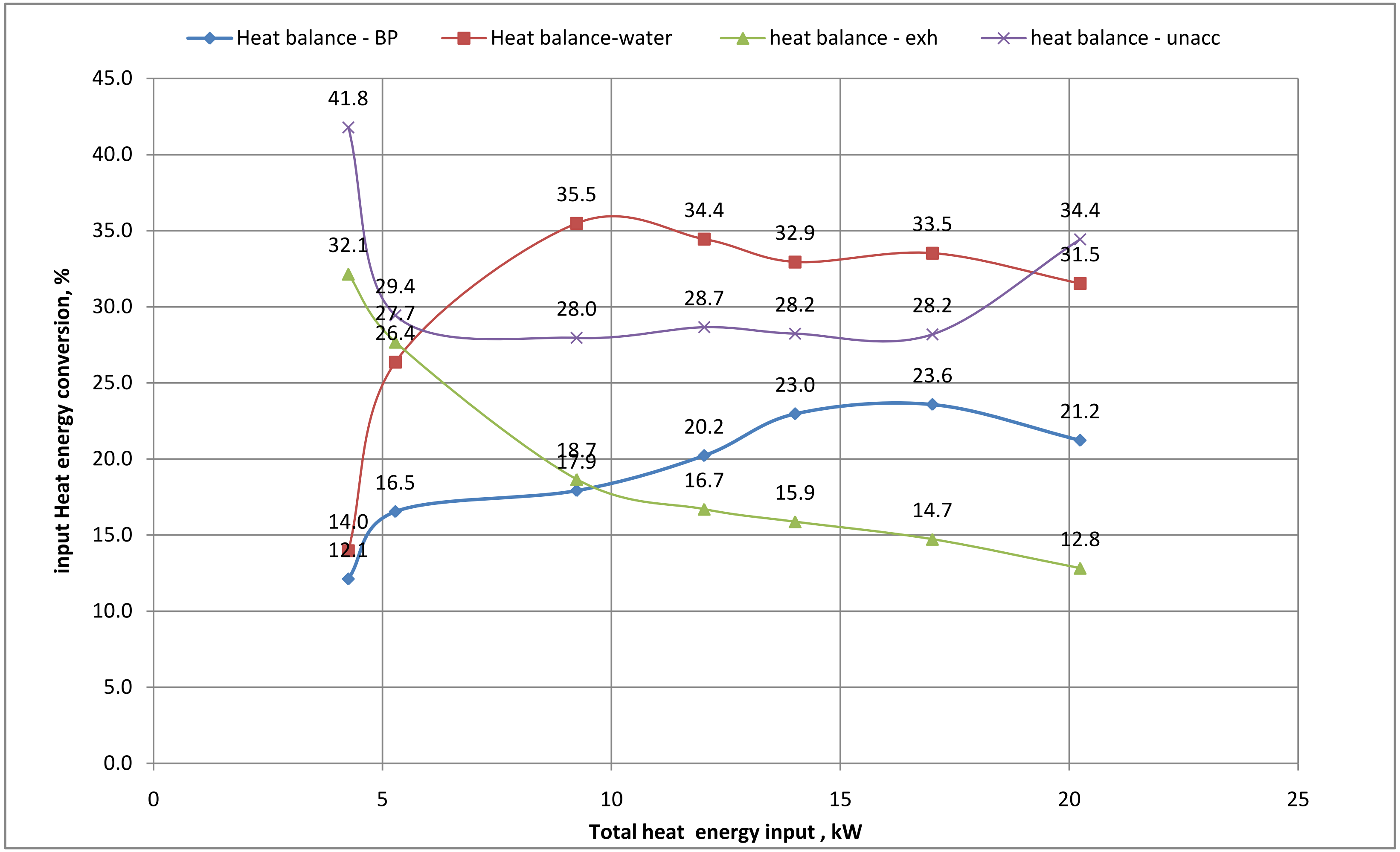

Fig 4.15: Heat balance curve for engine C

4.4 Wear assessment of the tested stationary diesel engines

The long term diesel engine operation in various field conditions and applications gradually affect the performance. It was observed that engine performance reduces due to number of factors like mechanical losses, wear and tear and poor work site conditions. The engine performs several important functions among which include lubrication, cooling, sealing, cleaning and protection against wear and tear along with corrosion. The wearing surfaces once formed and generate move particle that in term cause further wear is called the chain-reaction of wear. In internal combustion engine the wear may occur in the components by abrasion, adhesion, corrosion, brake down-of lube oil film and fatigue. The particles bridge the gap maintained by oil film, making simultaneous contact with both surface causing damage and resulting in component wear. In the technical literature for oil film thickness in diesel engine components it have been found that most of these dynamic clearances are ranges between 0-20 microns. It was necessary to quantify the power loss due to increase in millage and factor that led to power loss. The engines were completely dismantled in the laboratory and were subjected to dimensioning of various engine components. The engines were inspected for wear assessment of components before performance test. After required service and maintenance the engines were subjected to performance test at test rig.

4.4.1 Component of diesel engine “A” inspection

The engine A and B was dismantled for inspection and found in normal condition. The injector pressure was checked with pressure tester gauge and found 195 kg/cm2 which was considerably good for fuel atomization. The other system components were found in satisfactory working condition. The head gasket found in broken and leak condition which cause loss of compression. The compression loss may result in low air-fuel ratio and due to less amount of air the engine power may reduced. The engine oil in the sump was also need to be replaced and found oil level below the mark which may result in poor lubrication. The cooling fins around the block were found cooled by the blower and was radiated hot air around the engine which may be due to the design of engine. The engine components were measured and all the clearances were found within the operational limits.

The following table for component inspection was prepared for wear assessment of component of two diesel engines, namely A and C.

1. Cylinder bore

No.	Cylinder bore diameter, (mm)						Max. permissible wear limit, (mm)
	Top position		Middle position		Bottom position		
	Thrust side	Non-thrust side	Thrust side	Non-thrust side	Thrust side	Non-thrust side	
1.	87.65	87.62	87.69	87.65	87.73	87.65	87.80

2. Piston

	Piston diameter, (mm)				Clearance between piston cylinder liner at the skirt of the piston on thrust side (mm)	Max. permissible wear limit, (mm)
	Top(above top compression ring)		At skirt			
	Thrust side	Non-Thrust side	Thrust side	Non-Thrust side		
1.	86.75	86.70	87.35	87.33	0.34	0.50

3. Ring end gap

Rings	Ring end gap,(mm)			Max. permissible end gap
	Top	Middle	Bottom	
1st	0.30	0.40	0.35	0.80
2nd	0.35	0.45	0.45	
3erd	0.35	0.45	0.45	
Oil ring	0.30	0.35	0.40	

4. Ring side clearance

Rings	Ring side clearance, (mm)	Max. permissible clearance limit, (mm)
1st Comp. ring	0.04	0.20
2nd Comp. ring	0.05	
3rd comp. ring	0.05	
Oil ring	0.04	

5. Main bearing

Bearing No.	Diametrical Clearance (mm)	Crankshaft end float (mm)	Max. permissible clearance limit,(mm)	
			Diametrical Clearance	Crankshaft and Float
1.	0.12	0.25	0.20	0.3

6. Big end bearings

Bearing No.	Clearance, (mm)		Max. permissible clearance limit, (mm)	
	Diametrical	Axial	Diametrical	Axial
1.	0.17	0.3	0.35	0.5

4.4.2 Component of diesel engine "C" inspection

1. Cylinder bore

No.	Cylinder bore diameter, (mm)						Max. permissible wear limit, (mm)
	Top position		Middle position		Bottom position		
	Thrust side	Non-thrust side	Thrust side	Non-thrust side	Thrust side	Non-thrust side	
1.	85.08	85.06	85.08	85.07	85.10	85.08	85.125

2. Piston

S.No.	Piston diameter, (mm)				Clearance between piston $ cylinder liner at the skirt of the piston on thrust side (mm)	Max. permissible wear limit, (mm)
	Top(above top compression ring)		At skirt			
	Thrust side	Non-Thrust side	Thrust side	Non-Thrust side		
1.	84.53	84.56	84.52	84.54	0.58	0.50

3. Ring end gap

Rings	Ring end gap,(mm)			Max. permissible end gap limit,(mm)
	Top	Middle	Bottom	
1st	0.80	0.90	0.90	1.0
2nd	0.75	0.85	0.85	
3erd	0.80	0.85	0.90	
Oil ring	0.80	0.85	0.90	

4. Ring side clearance

Rings	Ring side clearance, (mm)	Max. permissible clearance limit, (mm)
1st Comp. ring	0.10	0.25
2nd Comp. ring	0.10	
3rd comp. ring	0.10	
Oil ring	0.15	

5. Main bearing

Bearing No.	Diametrical Clearance (mm)	Crankshaft end float (mm)	Max. permissible clearance limit,(mm)	
			Diametrical Clearance	Crankshaft and Float
1.	Roller bearing provided	0.15	NA	0.20

6. Big end bearings

Bearing No.	Clearance, (mm)		Max. permissible clearance limit, (mm)	
	Diametrical	Axial	Diametrical	Axial
1.	0.12 – 0.18	0.20	0.25	0.25

It was found during the inspection of engines C that the fuel filter was chocked and air cleaner was also dry. The cylinder head was found very sticky with black oil and head gasket found damaged. Excessive sludge was found on valve openings with burned smoke which indicate the engine oil was not changed since long time and valve clearance was also not maintained. It was found that valve spring deteriorated and deformed with over time use. The spring rate changes over the time due to material heating and cooling under load and it causes valve malfunction which reduces efficiency. The engine manufacturer generally recommends replacement of valve spring with 3mm or more deformation. The injector pressure was also measured and found 153 kg/cm^2 against the standard required pressure of

180 kg/cm^2. Which indicate there was incomplete combustion due to poor atomization of diesel fuel inside the chamber. The cylinder block was also observed and found that excessive carbon deposition on the ridge. The piston ring groves were found filled with oil mix carbon and it seems that there may be little or no movement of the rings inside the groves. The liner or sleeve of the block found glazy and scratched which may be due to overheating and excessive wear. The clearance between cylinder liner and piston skirt at thrust side was found 0.58 mm against the discard limit of 0.50 mm which is higher side. The ring end gap was ranged from 0.75 to 0.90 mm against the maximum permissible limit of 1.0 mm which is also near to maximum limit. The ring side clearance was ranged from 0.10 to 0.15 mm which is within the limit. The diametric clearance of big end bearing was found from 0.12 to 0.18 mm and shows some ovality of crank pin which causes poor and non uniform lubrication either splash or forced. The above discussed engine components conditions are very significant from the efficient operational point of view. Hence it may be concluded that the engine C was not serviced and maintained properly. It resulted in poor performance; uneconomical operation and wear tear of engine components also.

CHAPTER-V

SUMMARY, CONCLUSIONS AND SUGGESTIONS FOR FURTHER WORK

SUMMARY, CONCLUSION AND SUGGESTIONS FOR FURTHER WORK

5.1 Summary

The stationary sources of power available on Indian farm for doing various stationary operations are diesel/oil engines for irrigation pump sets, power threshers, sugarcane crushers, rice hullers and sprayers etc. The population of stationary prime movers has increased tremendously since the green revolution. Power from diesel engines increased from 0.009 kW/ha in 1960-61 to 0.247 kW/ha in 2000-01 and 0.335 kW/ha in 2013-14. These engines are portable and useful for irrigation from water harvesting structures and local natural water bodies. These engines are being used as a versatile source of power in other non agricultural rural application such as civil construction machinery, electrical generators and processing unit. These engine performances in terms of power development, thermal and mechanical efficiency may vary depending on severity of operation and maintains practices. In the present study it was needed to assess the performance of engine being used in different working conditions. The main objective taken for study were performance evaluation of stationary diesel engine on engine test rig in laboratory and analysis of test result by heat balance method. The wear assessment of the engine was also carried out for two of three engines under experiment.

5.2 Conclusions

In past, many researchers have worked on the stationary diesel engines performance with related parameter in different aspect and these work were also reviewed for the present study. To record observations and analysis of various performance parameters these different engines were tested on two test rig at College of Agricultural Engineering Jabalpur. The wear assessment was done with the help of various measuring instruments. On the basis of the performance result as per our objective the following conclusions were drawn:

1. The maximum rated power of engine A, engine B and engine C was found to be 3.1 kW, 5.8 kW and 2.24 kW against the declared value of 3.72 kW, 7.4 kW and 4.85 kW respectively which may be due to poor governing, thermal and mechanical losses.

2. The rated brake specific fuel consumption of engine A, engine B and engine C was found to be 366 g/kW-h, 288 g/kW-h and 408 g/kW-h against the declared value of 230 g/kW-h, 260 g/kW-h and 180 g/kW-h. The fuel consumption of engine A and B are higher side and for engine C it was considerably too high. The engine C was found to be uneconomical to operate under the present physical conditions and the loss of power may be due to all

possible reasons like worm component, low injection pressure, poor compression and incomplete combustion.

3. The rated brake thermal efficiency of engine A, engine B and engine C was found to be 22.13%, 28.34% and 20.07%. The brake thermal efficiency of engine A and C was considerably low due to higher fuel consumption and loss of input energy in various system. The result of engine C is comparatively poor due to its physical condition also that may be due to its faulty maintenance practice.

4. The temperature of exhaust gases at or near rated engine rpm of 1500 for the engine A, engine B and engine C was found to be 320, 365 and 224^{o}C. The higher temperature of exhaust gases was due to higher heating value of burned fuel and that proportionately converted to brake power. The poor conversion efficiency of engine C was clearly demonstrated by the observed temperature.

5. These engines were further tested with the load from no load, 20 %, 40%, 60%, 80%, 100% and 110% for analyzing with heat balance method. The observations were used to calculate parameters like heat input, brake power, loss of heat in cooling, exhaust and unaccounted losses.

6. The engine A lost major part of energy i.e. 20.5% in exhaust and 57.4% in unaccounted losses at rated engine speed. The engine A was lower power air cooled and these unaccounted losses may be due to higher temperature gradient between fins and blown air though these losses reduced at higher load because of reduced gradient. The unaccounted losses may include a blower fan, plugging of fins with dirt and hot recirculating cooling air.

7. The engine B lost major part of energy i.e. 38.4% in cooling and 24.8% exhaust near rated engine speed of 1493 rpm. The loss in cooling the engine through water was considerably higher and that may be one of the reason of energy loss. The engine B was also shown higher exhaust gas temperature and these gases carrying away heat energy that means the cooling efficiency of the engine was not up to mark.

8. The engine C lost major part of energy i.e. 32.9% in cooling and 28.2% unaccounted heat loss. The unaccounted loss may be due to friction, incomplete combustion, blowby and faulty components of the engine. The cooling loss may be due to corrosive and atmospheric deterioration and variation in water flow rate or higher flow rate. The coolant temperature increases as it absorbs heat from cylinder valve and head.

9. The wear assessment of two engines i.e. engine A and engine C was carried out. For engine A the most of engine component were measured and

all the clearances were found within limits that mean the engine physical condition was not the prime reason of loss in performance.

10. The engine C was inspected and major part were measured. The injection pressure was 153 kg/cm^2 and most of the parts were found worn out. It may be concluded that it was not serviced and maintain properly.

5.3 Suggestions for further works

- It can be suggested that fault tree analysis may be applied for the single cylinder diesel engine.
- Measurement and evaluation of reliability, availability and maintainability can be applied for comparative study of single cylinder stationary diesel engine.

CHAPTER-VI
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BIBLIOGRAPHY

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CHAPTER-VII
APPENDICES

APPENDIX – I

Calculation sheet for heat balance during performance test of diesel engine "**A**"

1. Total fuel consumption

TFC = $\frac{FC_2 - FC_1}{1000} \times \frac{60}{t} = \frac{48-38.5}{1000} \times \frac{60}{30} = 191 \ \mathrm{g/min}$

2. Break power (BP)

BP = $\frac{W \times N}{K \times 60} = \frac{20 \times 1503}{159.0000 \times 60} = 3.15 \ \mathrm{kW}$

3. Break thermal efficiency

$$\eta_{BTH} = \frac{BP}{Cv \times \frac{TFC}{60}} \times 100\% = \frac{3.15}{44000 \times \frac{191}{60}} \times 100 = 22.48\%$$

4. Break specific fuel consumption (BSFC)

BSFC = $\frac{TFC}{BP} = \frac{191}{3.15} = 366.80$ g/kW-h

5. Total heat input

Total Heat Input (H_{in}) = $\frac{TFC}{60} \times C_v \ (\mathrm{kW}) = \frac{191}{60} \times 44000 =$ 14.01 kW

6. Heat converted to break power

H_{BP} = BP (kW) = 3.15 kW

$\%H_{BP} = \frac{BP}{H_{in}} \times 100\% = \frac{3.15}{14.01} \times 100 = 22.48\%$

7. Heat carried away by the exhaust gas

$H_{exh} = \frac{T_1 - T_8}{T_1 - T_2} \times \frac{m_a}{60} \times C_P \ (T_4 - T_3) \ \mathrm{kW} = \frac{103.0 - 46.0}{103.0 - 73.0} \times \frac{6.4200}{60} \times 4.187 \ (31.9 - 30.0) =$ 2.87 kW

H_{exh} (%) = $\frac{H_{exh}}{H_{in}} \times 100 = \frac{2.87}{14.01} \times 100 = 20.5\%$

8. Unaccounted heat loss

$$H_{unsac} = H_{in} - (H_{exh} + H_{BP}) \quad = \quad 14.01 - (2.87 + 3.15) = 8.04\text{kW}$$

$$H_{unsac}\ (\%) = H_{in}\,(\%) - (H_{exh}\ \% + H_{BP}\ \%) \quad = \quad 100 - (20.5 + 22.48) = 57.4$$

APPENDIX – II

Calculation sheet for heat balance during performance test of diesel engine "**B**"

1. Total fuel consumption

TFC = $\frac{FC_2 - FC_1}{1000} \times \frac{60}{t} = \frac{50.5 - 35.9}{1000} \times \frac{60}{30} = 292 \text{ g/min}$

2. Break power (BP)

BP = $\frac{W \times N}{K \times 60} = \frac{37.5 \times 1493}{159.0000 \times 60} = 5.813 \text{ kW}$

3. Break thermal efficiency

$$\eta_{BTH} = \frac{BP}{Cv \times \frac{TFC}{60}} \times 100\% = \frac{5.9}{44000 \times \frac{292}{60}} \times 100 = 28\%$$

4. Break specific fuel consumption (BSFC)

BSFC = $\frac{TFC}{BP} = \frac{292}{5.9} = 296$ g/kW-h

5. Total heat input

Total Heat Input (H_{in}) = $\frac{TFC}{60} \times C_v \text{ (kW)} = \frac{292}{60} \times 44000 = 21.56$ kW

6. Heat converted to break power

H_{BP} = BP (kW) = 5.9 kW

$\%H_{BP} = \frac{BP}{H_{in}} \times 100\% = \frac{5.9}{21.56} \times 100 = 27.38\%$

7. Heat carried away by the exhaust gas

$H_{exh} = \frac{T_1 - T_8}{T_1 - T_2} \times \frac{m_a}{60} \times C_P (T_4 - T_3) \text{ kW} = \frac{373 - 47}{373 - 203} \times \frac{6.4200}{60} \times 4.187 (38.9 - 32.6) =$ 5.34 kW

H_{exh} (%) = $\frac{H_{exh}}{H_{in}} \times 100 = \frac{5.34}{21.56} \times 100 = 24.79\%$

8. Heat carried away by engine cooling water

$$H_{water} = \frac{m_{wa}}{60} \times c_p\,(T_6 - T_5) = \frac{6.46}{60} \times 4.187\,(46.1 - 33.2) = 8.27kW$$

$$H_{water}\ (\%) = \frac{H_{water}}{H_{in}} \times 100 = \frac{8.27}{21.56} \times 100 = 38.4\%$$

9. Unaccounted heat loss

$$H_{unsac} = H_{in} - (H_{water} + H_{exh} + H_{BP}) = 21.56 - (8.27 + 5.34 + 5.9) = 2.05\ kW$$

$$H_{unsac}\ (\%) = H_{in}\,(\%) - (H_{water}\,\% + H_{exh}\,\% + H_{BP}\,\%) = 100 - (38.4 + 27.38 + 24.79) = 9.44\%$$

APPENDIX – III

Calculation sheet for heat balance during performance test of diesel engine “**C**”

1. Total fuel consumption

TFC = $\frac{FC_2-FC_1}{1000} \times \frac{60}{t} = \frac{48.2-38.1}{1000} \times \frac{60}{30} = 202 \text{ g/min}$

2. Break power (BP)

BP = $\frac{W \times N}{K \times 60} = \frac{20.2 \times 1495}{159.0000 \times 60} = 3.1 \text{ kW}$

3. Break thermal efficiency

$$\eta_{BTH} = \frac{BP}{Cv \times \frac{TFC}{60}} \times 100\% = \frac{3.15}{44000 \times \frac{0.0191}{60}} \times 100 = 22.48\%$$

4. Break specific fuel consumption (BSFC)

BSFC = $\frac{TFC}{BP} = \frac{202}{3.15} = 384.76$ g/kW-h

5. Total heat input

Total Heat Input (H_{in}) = $\frac{TFC}{60} \times C_v \text{ (kW)} = \frac{202}{60} \times 44000 =$ 14.01 kW

6. Heat converted to break power

H_{BP} = BP (kW) = 3.15 kW

%H_{BP} = $\frac{BP}{H_{in}} \times 100\% = \frac{3.15}{14.01} \times 100 = 22.48\%$

7. Heat carried away by the exhaust gas

$H_{exh} = \frac{T_1-T_8}{T_1-T_2} \times \frac{m_a}{60} \times C_P (T_4 - T_3) \text{ kW} = \frac{301-46}{301-62} \times \frac{6.4200}{60} \times 4.187\,(38-33.9) =$ 2.224 kW

H_{exh} (%) = $\frac{H_{exh}}{H_{in}} \times 100 = \frac{2.224}{14.01} \times 100 = 15.9\ \%$

8. Heat carried away by engine cooling water

$$H_{water} = \frac{m_{wa}}{60} \times c_p\,(T_6 - T_5) = \frac{6.46}{60} \times 4.187\,(46.4 - 36.2) = 4.616\ \mathrm{kW}$$

$$H_{water}\ (\%) = \frac{H_{water}}{H_{in}} \times 100 = \frac{4.616}{14.013} \times 100 = 32.9\ \%$$

9. Unaccounted heat loss

$$H_{unsac} = H_{in} - (H_{water} + H_{exh} + H_{BP}) = 14.013 - (4.616 + 2.224 + 3.217) = 3.12\ \mathrm{kW}$$

$$H_{unsac}\ (\%) = H_{in}\,(\%) - (H_{water}\,\% + H_{exh}\,\% + H_{BP}\,\%) = 100 - (32.9 + 15.9 + 22.48) = 28.72\%$$

CURRICULUM VITAE

Curriculum Vitae

The author of this thesis is Mr. **Kunal Bhelave**, S/o Shri Rajkumar Bhelave. He was born on 21 November 1992 at Malanjkhand District - Balaghat (M.P.).

He completed his High School & Higher Secondary Examination from Excellence School Balaghat (M.P.) in year 2008 & 2010 respectively.

He completed his Graduation Degree B.Tech. (Agricultural Engineering) from MGCGV, Chitrakoot in year 2015.

In the year 2016-17, he joined Master of Technology (Agricultural Engineering) two year degree programme in Department of Farm Machinery and Power Engineering, at College of Agricultural Engineering, JNKVV, Jabalpur.

After completing the entire prescribed course work successfully, he was submitted the thesis in "Performance Evaluation and Wear Assessment of Diesel Engines used in Field Application" in partial fulfilment of the requirements for the award of the degree of "Master of Technology".

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