Dissertation Report

on

INVESTIGATION OF ENERGY ANALYSIS OF DI DIESEL ENGINE AND ITS COMPARATIVE STUDY WITH MODIFIED HARDWARE PARAMETERS OF DIESEL ENGINE

Submitted in partial fulfilment of the requirement for the award of degree

Of

Master of Technology

in

Mechanical Engineering

By

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PROPOSED TOPIC: Investigation of Exergy Analysis of DI Diesel Engine and its Comparative study with Modified Hardware Parameters of DI Diesel Engine: The Modelling Approach.

Qualitative Assessment of Proposed Topic by PAC		
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2	Project Feasibility: Project can be timely carried out in-house with low-cost and available resources in the University by the students.	7.33
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4	Project Supervision: Project supervisor's is technically competent to guide students, resolve any issues, and impart necessary skills.	7.00
5	Social Applicability: Project work intends to solve a practical problem.	6.00
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Overall Remarks: Approved

PAC CHAIRPERSON Name: 12174: Gurpreet Singh Phull **Approval Date:** 10 Oct 2016

CERTIFICATE

I hereby certify that the work being presented in the dissertation entitled "**Investigation of Energy Analysis of DI Diesel Engine and its Comparative Study with Modified Hardware Parameters of Diesel Engine**" in partial fulfilment of the requirement of the award of the Degree of master of technology and submitted to the Department of **Mechanical Engineering of Lovely Professional University, Phagwara**, is an authentic record of my own work carried out under the supervision of **Mr.Sumit Kanchan**, **Assistant Professor, Department of Mechanical Engineering, Lovely Professional University**. The matter embodied in this dissertation has not been submitted in part or full to any other University or Institute for the award of any degree.

9th May, 2017

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This is to certify that the above statement made by the candidate is correct to the best of my knowledge.

9th May, 2017

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Signature of Examiner

ABSTRACT

For the efficiency of the diesel engine heat losses play a major role. Thermally induced mechanical stress compromises the reliability of engine components due to the heat transfer phenomena in the engine. For the development of the engine, the ability to predict the heat transfer in the engines is needed. Today, predictions are increasingly being done with numerical simulations at an ever earlier stage of engine development. These methods must be based on the understanding of the principles of heat transfer.

This work presents the principle of thermodynamics and heat transfer totally. The test was conducted on a 4-stroke diesel engine with four cylinders. It is a BSII without EGR system having inline overhead valve and water cooled direct injection. First the engine had run without doing any modification to it and the all the heat transfer analysis has been done by the heat balance sheet. Then this engine is modified to analyze the heat transfer through the various section of the engine by doing the hardware changes in it. Again, a new heat balance sheet is made based on the test on the modified engine. The modifications to the engine are injection pressure (OEM: 240bar) to 220 & 200bar, auxiliary air supplied (0.50-0.55bar), injection timing (29°CA BTDC to 24°CA BTDC), air throttling (0-50%), post injection (5°CA BTDC) and intake air temperature (350°C). The heat transfer analysis after all the modifications done have been done and its was found that as the Hardware parameters were changed (increased or decreased) from the OEM values it results in increase in value of Fuel consumption and more heat loss from exhaust gases. Therefore, the thermal efficiency of the engine decreases. The repeatability test has been done with various parameters and the conclusion is confirmed after every test.

ACKNOWLEDGEMENT

Gratitude is the hardest of emotions to express often one does not find adequate words to convey the feeling of gratefulness. There were many people whom we would like to thank for their support. We sincerely feel that credit of this work could not be narrowed down to any one individual, by which we have achieved its completion.

We take this opportunity to express profound gratitude and deep regards to our project mentor **Mr.Sumit Kanchan** for his exemplary guidance, monitoring and constant encouragement throughout the course of this efficiency appraisal for catalytic converter project. The blessing, help and guidance given by him time to time will carry us a long way in the journey of life on which we are about to embark. We are thankful to him, who had to bear a heavy load of responsibility and concern in bringing this project to a successful end, indeed in selfless spirit. We further extend our heartiest gratitude to our friends who constantly suggested us to improve our work and standard for this project. Their advices were immensely valuable.

We further thank all the people who were directly or indirectly connected to the project without them the work would not have been in the conclusive state.

Bharat Bhusan Patra

11206104

DECLARATION

I, Bharat Bhusan Patra, student of M.Tech Mechanical Engineering hereby declare that the Project titled “**Investigation of Energy Analysis of DI Diesel Engine and its Comparative Study with Modified Hardware Parameters of Diesel Engine**”which is submitted by me to Department of Mechanical Engineering, Lovely Faculty of Technology & Sciences, Lovely Professional University Phagwara, Punjab in partial fulfillment of requirement for the award of degree of Master of Technology in “Mechanical Engineering”, has not been previously formed the basis for the award of any degree, diploma or other similar title or recognition.

Signature………………………..

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Chapter 1: Introduction

1.0 Overview

This chapter contains about detail description about the losses of engine. This chapter is divided into six sections which separately discussed about the all losses which occur in the engine.

1.1 Energy balance

In an engine fuel is used for the energy generation. That energy is utilizing to generate heat energy for supply energy to the engine. However, a little amount of energy is only used to transform the desired work rest of all is wasted in other parts. There are two main areas where the heat not utilizing for the special application, first one is heat carried away by exhaust gasses and the cooling system.

For the perfect calculation and making of heat balance sheet a no of tests should be included like friction losses, speed and load and fuel consumption like phenomenon. The exhaust temperature and the cooling system will also be measured for making perfect balance sheets. There are some small losses also occurring but we neglect all those for making the balance sheet. Due to cooling system water tubes and their friction phenomenon also a major heat distribution for the measurement of heat balance sheet. There are no of losses listed below.

1.2 Types of engine losses

In the engine system, there are three major areas were the losses happening with a wide range.

1. Through friction
2. Through cooling
3. Through exhaust
4. Losses to surrounding

The figure below (Fig1.1) describes the detailed description about the various heat losses from the diesel engine.

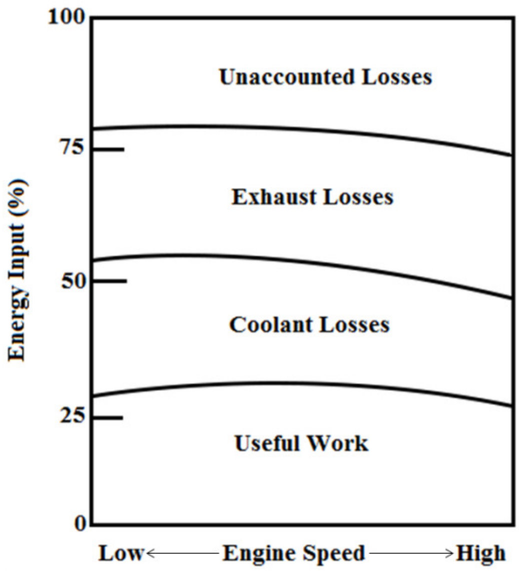

Figure 1.1 Schematic diagram of distribution of heat

1.2.1 Friction losses

As the name shows that the losses happen due to friction phenomenon. The power produce in engine system then it is used to circulate in the whole engine body system. The power is utilizing to produce motion like sliding and rotary motions. That friction in piston and bearings as well as Cram shaft and oil pump also. These create some losses which come under the friction losses. These losses are combined together and then stimulated for the friction power. This heat or friction power is utilizing for making some desired work. The friction heat also reduced and put under the control by the coolant and the oil.

1.2.2 Cooling losses

Cooling system is a basic phenomenon of the engine system. Due to energy generation and the heat production of the engine radiation reaches to 3 to 5% of the total heat supplied. So, we use a coolant for the heat carry. But it is a type of dead loss occurs in the engine system because the little amount of energy also spent for removing heat from the engine system. So, we can’t stop to ignoring the cooling losses as the engine power also use to remove that heat energy. It is a primary objective to reduce the losses by the engineer which can be happened by the engine system and their phenomenon.

1.2.3 Exhaust losses

The engine consists of a no of interrelated components used for the exhaust purpose from the combustion chamber. The combustion chamber contains different type of parts like exhaust manifold, pipes of exhaust, turbocharger etc. these are all used to make a good exhaust system for the combustion chamber.

The losses due to exhaust system were measured by the products of the combustions. From the exhaust gasses these can be go through the different sections of the combustions chamber and make a distribution among them. After that the amount of energy to drag out the exhaust gasses was considered as losses. Due to incomplete combustion, the excess amount of exhaust gasses will have generated because there are no possibilities to create or make the compete combustion cause of we can’t achieve the actual and perfect air/fuel mixture.

Due to excess making of exhaust gasses, the exhaust system works very sufficiently and combust some energy from the engine system to drag out the exhaust gasses and the incomplete combustion also very harmful for the engine system and affect the efficiency of the engine and the expensive medium.

However, the amount of energy required for the reducing and exhausting the gasses from the engine system in considered as the exhaust losses. Therefore, it is required to reduce the exhaust losses to maintain the good air/fuel mixture.

1.2.4 Losses to surrounding

Heat loses to surrounding from the diesel engine in the form of radiation. When the engine runs, it dissipates heat to the surrounding through the engine walls in the form of radiation. To calculate the heat loss to the surrounding in the form of radiation the proper geometry of the engine is most needed, which is quite difficult.

1.3 Nature of exhaust losses

In this study of the losses, exhaust losses were the most preferable losses among all the losses. It is generated due to high engine speed. When the engine throttle valve is open broadly the speed is increases and when is closed then the engine stop producing the energy. Due to this more energy will losses for exhausting the gasses. If the valve opens widely then the fresh air inserted and made the good mixture. As the mixture is burned in the combustion chamber the temperature will increase. Therefore, we try to keep a good air/fuel mixture to reduce the heat in the combustion chamber.

Chapter 2: Literature review

This section represents the summery works carried out by different researcher on the basis of heat loss, exergy analysis and energy analysis.

N.M. Al-Najemet al.[1]considered on the first and second law of thermodynamics. The availability energy analysis mainly based on the second law of thermodynamics. By this the losses of heat energy is identified. In this research work it is shown that about 50% of the chemical energy is destroyed due to unaccounted factors. About 15% is lost through the cooling water or exhaust. From the research the result has came out that about 50% of losses take place due to cooling and exhaust gas of the total input fuel energy. It is suggested that this amount of losses energy can be utilized in a heat recovery system for producing the power. The second law of thermodynamics is mainly to determine the kind of application.

Here the first and second law of thermodynamics is used to quantity and quality of energy in the diesel engine. The major part of the losses owe wasted heat energy are due to unaccounted factors. It is shown that about 30% of the total useful energy is lost. The analysis of exergy is allowed to develop allowed a systematic approach that can be used to identify sites of real losses of valuable energy in thermal device.

Jamil Ghojelet al.[2]analyzed that in the internal combustion engine the diagnosing process is done by using cylinder pressure, delivery pressure of fuel, the injector needle lift data is well established and widely used procedure in engine and fuel performance test. In this research work the output includes the apparent heat release, burning rate of fuel, the fraction of fuel burned, average gas temperature, indicated parameters, heat losses.

To calculate the heat release characteristics of diesel engine a simple single-zone model is developed by using diesel oil emulsions and standard diesel fuels. For quick evaluation and interpretation of the performance of engines with different configurations or fuels this model is very much suitable. It is also very much useful for the monitoring the real time engine heat release characteristics in the purpose of diagnostic.

Samad Jafarmadar[3]in this research work the energy and exergy analysis are done in pre and main chamber of a Lister 8.1indirect diesel engine and the engine with BMEP=2.96 bar and 5.96 bar, load with 50% and with full load. The maximum torque will be at 730 rpm. The energy analysis is done by the help of the CFD code during a closed engine cycle.

Here the test is carried between two different loads. The exergy components are calculated and verified at different crank angles at different two loads. It is shown that 56% at partial load and 77% of full load of total irreversibility are related to combustion chamber. This study shows that at complex chamber geometry the multi-dimensional geometry can be used to get more insight into the effect of flow field on the combustion process by considering the second law of thermodynamics.

At partial load and full load condition the combustion irreversibility is 293.76% and 24.93% respectively. At full load 86% of burned fuel exergies and at partial load 89% of burned fuel exergies concern to heat loss main chamber. The amount exergy flow in throat increases up to 56%, when the load increases from partial load to full load.

Y Li, M Jiaet al.[4]in this research there are multi-dimensional models are coupled with a detailed chemical mechanism. It is for investigate the exergy and energy distribution of three different combustion regimes of I.C engine. The result of the study shows that about 50% heat release point affects the ringing intensity and fuel efficiency. The ringing intensity is used to quantify the knock level. It is shown that about 10% to 50% of heat release point dominates the ringing intensity and the piston of 90% heat release pint affects the fuel efficiency. The heat transfer losses of conventional diesel combustion are depending upon the local temperature gradient. The destruction of exergy is related to the homogeneity of inside temperature of combustion chamber and also equivalence ratio during the combustion process. It also related to combustion duration, chemical reaction.

F Payriet al.[5]in this study the experimental energy balance with the analysis regarding the path followed by the energy has been presented. The objective of the research is to identify the most important energy terms, targeted to the assessment of the most influential parameters and the variation of parameters in the performance of the engine. With the speed a load the determination of the variation range of most important energy is allowed by the analysis of the complete engine map.

In the case of the engine efficiency the variation of coolant temperature has no effect (lower than 0.5%). The heat from cylinder gas to the coolant temperature is increased and while heat to the oil and to the ambient decreases while the colder coolant temperature is used. Comparing to the achieved coolant temperature the variation of the intake air temperature has higher effects on the engine performance. The brake efficiency is improved by 1%, while the air is cooled to 35°C.This happened due to the reduction of the heat of the chamber.

The benefits we get at the rpm of 2000, while the exhaust gas losses are lowered at the rpm of 4000. Finally, this study allows performing detail evaluation and analysis of the engine energy output.

F Payriet al.[6]in this study the external heat losses of turbocharger are examined and a simplified model, in which both the convection and radiation is taken into consideration. In a turbocharger test bench the model is adjusted for two different models of turbo chargers. Later it is attached with the engine test bench. This model represents the various heats lose from the different elements of the turbochargers.

The researcher concluded that the heat losses in the small type turbochargers are neglected and the machine behavior has been predicted by the use of map of manufacturer. Due to higher temperature and larger surface area the most important heat fluxes comes out from external surface of the turbine. This is the cause of drop of the enthalpy of turbine up to its half. In the compressor side depending upon the running conditions external heat flow can be absorbed or can be lost. In the compressor side the heat is radiated by the compressor side.

J Fuet al.[7]in this research work the energy and exergy analysis of the biogas run dual fuelled diesel engine was done. The first and second law of thermodynamics is taken into consideration in this study. This shows the brake thermal efficiency, fuel consumption, exergy efficiency and different availability by varying the load on the engine and compared to the diesel value.

At the conclusion, the researcher concluded that the presence of CO_2 in the bio gas reduces the burning velocity and causes the incomplete combustion and increases the exhaust gas temperature in the dual fuel mode. The biogas has longer pilot ignition delay and high self-ignition temperature causes the delay in dual fuel combustion process. These factors lower the thermal efficiency. It is seen that if the load increases then the exergy efficiency also increases. To producing same amount of shaft work output the dual fuel mode higher fuel exergy.

Z Hanet al.[8]in transient condition the diesel engine operates almost constantly. The fuel is increased by the governor to increase the speed of engine and power. To decelerate the fuel portion is need to be decrease gradually, by which speed and engine power decreases gradually. For acceleration and deceleration transient operation analysis is performed. Acceleration shows the lag of significant turbocharger, where the parameter of exhaust gas can not instantly match increase in fueling. Also, insufficient energy is supplied to the turbocharger is the possibilities of decrease of excess air ratio up to stoichiometric values. The deceleration does not show any lag of turbocharger and as a result of that effect on engine operation efficiency.

M Özkanet al.[9] in this study according to the perspective of second law of thermodynamic the influence of pre injection strategies on thermal and energetic efficiencies are studied. In an I.C engine the fuel energy distributed among the exhaust loss, cooling loss, among the brake work. The inter-cooling losses and exhaust losses increases slightly and the cooling and block heat losses decreases slightly, when pre-injection strategies allow. Here the total heat loss is negligible, so the exergy, exergetic efficiency and thermal efficiency do not show any significant change. Exergy destruction is independent of pre-injection and it has only 6% destruction.

Oleg spitsov[10]was investigated about the specific aspects of heat transfer in the combustion chamber of internal combustion engines and possibility to directly measure the heat flux by the software implemented with the aid of Mat lab. With this software, he measured the actual and perfect picture of the flux behaviour and their properties which is continuously done in the combustion chamber. Rest of numerical approach is explained with the experimental result. The diesel engine was taking in action in this experiment.

Mirko bovo[11] stated about the heat losses in the IC engine and discussed about the major area of the efficiency and the limitations of IC engine. In this study, they also use numerical method to calculate the numeric terms. He is also giving his statement about the heat transfer phenomenon and the mechanical stress phenomenon. In today world, the numerical method simulation method is being done at every stage of engine development. This total study is explained about the specific numerical simulation of the heat transfer phenomenon.

FinolParra[12]has studied about an experimental investigation who has been explained the heat flux and temperature distribution on the walls of cylinder and cylinder head in diesel engine. In this study, the temperature was measured and heat flux was also calculated. The temperature distribution also gets affected by conduction, which was generated by piston head and piston rings. The results observed in the cylinder bores were used to develop a simple model from first principles to estimate the heat transferred from piston rings and skirt to the cylinder wall.

C.Arcoumaniset al.[13]studied about the heat transfer taking place in the combustion chamber and their defects occur in the combustion chamber. This study indicated about the complete investigation of the heat transfer occurring in the direct injection system in diesel engine. The convective heat transfer mechanism also associated with high temperature phenomenon and the combustion associated with the swirl ratio. The measured temperature of the gas and the piston wall was measured by two-colored pyrometer and fast-response thermocouples, respectively.

Chapter 3: Objective of the study

With the advent of change in technology the companies are in a habit of varying the hardware performance parameters to keep them in tune with the Sensors and ECU, without being realizing that slight change in hardware parameters, decreases the thermal efficiency of the engine. Effort has to be made to identify and highlight that too much change in the value of hardware parameters (Injection Pressure and Injection Timing), does affects the performance of an engine. The accessories sensors like the pressure sensors, Accelerator sensor also hamper the vehicle performance. In this exertion, various hardware modifications have been incorporated in an engine and is tested for its heat transfer characteristics to test the thermal efficiency of the engine.

Chapter 4: Engine Accessories and Experimental setup

4.1 Resources required

- Engine
- 4-stroke diesel engine, turbocharged with intercooler, water cooled direct injection, in-line overhead valve
- hp=820Kw @2500rpm
- AVL coolant conditioning system
- AVL throttle actuator
- AVL oil conditioning system
- Software-AVL PUMA Automation and Control system
- AVL eddy current dynamometer

Table 4.1 Test engine specification

S.No.	Description	Diesel Engine
1.	**Type**	4-stroke Diesel engine, 4 cylinders, Water cooled direct injection, in-line overhead valve
2.	**Aspiration**	Turbocharged with inter cooler
3.	**Max Power**	88.25 kW(120ps) @2500rpm
4.	**Max Torque**	40 kgm @ 1500-1600rpm
5.	**Compression**	17:5:1
6.	**Injection Pressure**	240bar
7.	**Injection Timing**	1.36mm (±0.002mm) plunger lifts at TDS with No.1 cylinder on compression stroke
8.	**Oil Sump/System Capacity**	8.5L

Figure 4.1 Engine test bench

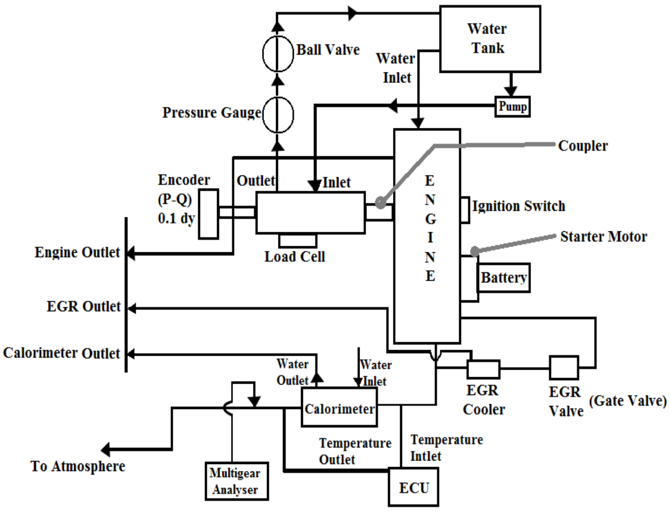

Figure 4.2 Schematic diagram of full engine setup

Figure 4.3 Fuel injector

Figure 4.4 Four cylindered diesel Engine

4.3 Investigation foundation

Before doing the experiment, the engine was disassembled and all components were cleaned. Then the measurement and assembling is done in the measurement and assembly section. Then the engine is installed on the test bench and connected to the dynamometer via connecting shaft.

4.4 Accessories description

4.4.1 Fuel conditioning unit

It is the unit in a closed circuit in which the fuel returning from the engine is directed back into the fuel feed line. The fuel feed and return lines must be linked by a by-pass immediately before the engine.

4.4.2 Coolant conditioning Unit

It is the unit in a closed system which is used for the control of temperature fluctuation in fuel system. It is directly connected to the engine.

4.4.3 Eddy Current Dynamometer

Force and torque is measured by the eddy current dynamometer. Through the coupling shaft the eddy current dynamometer is connected to the engine main shaft.

4.4.4 Fuel mass flow meter

The mass flow rate of the fuel is controlled by this device.

4.4.5 Calorie meter

In order to quantifying the losses, it is easier to used exhaust gas calorimeter, shell and tube type. Shell and tube heat exchangers consist of a series of tubes. One set of these tubes contains the fluid that must be either heated or cooled. The second fluid runs over the tubes that are being heated or cooled so that it can either provide the heat or absorb the heat required. A set of tubes is called the tube bundle and can be made up of several types of tubes: plain, longitudinally finned, etc.

4.4.6 Water jacket

A **water jacket** is a water-filled casing surrounding a device, typically a metal sheath having intake and outlet vents to allow water to be pumped through and circulated. The flow of water to an external heating or cooling device allows precise temperature control of the device.

Chapter 5: Experimental work

5.1 General

The testing was done in two phases

Phase 1: In the phase 1 the engine was run without doing any modification and all the necessary data were noted down.

Phase 2: In the phase 2 the engine was run by doing the all modification to the engine and necessary data were noted down.

5.1.1 Description of phase1:

First the engine was run at three different rpm (1100-1300rpm) and torque was kept constant at 350kN. Then the different parameters were noted down.

The parameters which were noted down are as follows

1. The mass flow rate of fuel to the engine (in kg/hr)
2. The mass flow rate of cooling water through the water jacket
3. The inlet and outlet temperature of the cooling water
4. The inlet and outlet temperature of the cooling water through the calorie meter
5. The inlet and outlet temperature of the exhaust gas through the calorie meter

After the readings had been noted the data were putted into the formula to find out the heat losses, break power, indicated power and friction loss. Then the heat balanced sheet was made.

5.1.2 Description of phase2:

In this phase, the hard ware modification was done and then required readings were noted down. Following modifications were done:

1. Change in injection pressure:

To change the injection, pressure the shim was inserted to the fuel injector. As shim was inserted the width of the shim increased, hence the fuel inside the injector travelled a long distance and caused more turbulence inside the injector. Therefore, injection pressure reduced. The original injection pressure was 240bar and the modified injection pressure was 200bar.

2. Change in Injection Timing:

Injection timing was changed from 29° CA BTDC to 24° CA BTDC**.**

3. Compressing the air supply:

The compression of air was done by the help of banjo washer, the diaphragm valve attached to the fuel injection pump and it is coupled with the intake manifold to the engine through the line of fuel delivery, injector. Due to this inside the combustion chamber a swirl was produced. In the combustion chamber the diaphragm associated with the intake manifold and showed the effect. In the combustion chamber at low intake pressure low amount of fuel supply took place. The compressed gas removed this variable behavior of fuel supply. The air is compressed from **0.50bar to 0.55bar**

4. Throttling of air:

After compression through turbocharger, the compressed air entered into the intercooler. Then at high pressure it entered into the manifold region. Between the manifold and inter cooler the throttling valve was introduced. The throttling of air was done up to **50%**. By this the air entered into the combustion chamber with some restriction and created little turbulence.

5. Post injection:

After the main injection, a shorter injection was done to the combustion chamber. By this it affected the combustion and emission. The post injection was done **5° BTDC**

6. Change in air intake temperature:

Here the air was pre-heated before it mixed with the fuel. As the pre-heated air mixed with fuel the ignition temperature of the air-fuel mixture was increased. It showed the effect in the final result of heat losses. The air is heated up to the temperature **350℃**

5.2 Combination of changes

In the above the modification is described. After doing the modification the **combination** of all these changes were done. The combinations were done in the five categories.

The combinations are as follows:

Combination-1: Injection Pressure + Injection Timing**Combination**-2: Injection Pressure + Injection Timing + Compressed air

Combination-3: Injection Pressure + Injection Timing + Compressed air + Air throttling

Combination-4: Injection Pressure + Injection Timing + Compressed air + Air throttling + Post injection

Combination-5: Injection Pressure + Injection Timing + Compressed air + Air throttling + Post injection + Air intake temperature

5.2.1 Combination 1: Injection Pressure + Injection Timing

As described above the injection pressure was decreased from 240bar to 200bar and the injection timing is set to (29°CA BTDC to 24°CA BTDC). The both modification were done at the same time and remains changes were not done.

Table 5.1 Combination 1

S.No.	Operating Parameters	Operating Values
1	Speed	1100-1300rpm
2	Torque (@full throttle)	350Nm
3	Oil pressure	3bar
4	Air intake temperature	25-35°C
5	Injection Pressure	200bar
6	Injection Timing	24°CA BTDC

5.2.3 Combination 2: Injection Pressure + Injection Timing + Compressed air

In this combination, the changes in air supply (compressed gas) were added to the **combination1.** Hence the new combination is with injection pressure 240bar to 200bar, injection timing is set to (29°CA BTDC to 24°CA BTDC) and the compressed air supply is 0.55bar.

Table 5.2 Combination 2

S.No.	Operating Parameters	Operating Values
1	Speed	1100-1300rpm
2	Torque(@full throttle)	350Nm
3	Oil pressure	3bar
4	Air intake temperature	25-35℃
5	Injection Pressure	200bar
6	Injection Timing	24°CA BTDC
7	Compressed air supply	0.55bar

5.2.4 Combination 3: Injection Pressure + Injection Timing + Compressed air + Air throttling

In this combination with the combination 2 throttling of air were done. Hence the new set parameters are injection pressure 240bar to 200bar, injection timing is set to (29°CA BTDC to 24°CA BTDC), the compressed air supply is 0.55bar and supply air is throttled up to 50%

Table 5.3 Combination 3

S.No.	Operating Parameters	Operating Values
1	Speed	1100-1300rpm
2	Torque(@full throttle)	350Nm
3	Oil pressure	3bar
4	Air intake temperature	25-35℃
5	Injection Pressure	200bar
6	Injection Timing	24°CA BTDC
7	Compressed air supply	0.55bar
8	Air throttling	50%

5.2.5 Combination 4: Injection Pressure + Injection Timing + Compressed air + Air throttling + Post injection

In this combination, the 5° BTDC post injection as added to previous combination 3. Hence the new combination is injection pressure 240bar to 200bar, injection timing is set to (29°CA BTDC to 24°CA BTDC), the compressed air supply is 0.55bar, supply air is throttled up to 50% and post injection to 5° BTDC.

Table 5.4 Combinations 4

S.No.	Operating Parameters	Operating Values
1	Speed	1100-1300rpm
2	Torque(@full throttle)	350Nm
3	Oil pressure	3bar
4	Air intake temperature	25-35℃
5	Injection Pressure	200bar
6	Injection Timing	24°CA BTDC
7	Compressed air supply	0.55bar
8	Air throttling	50%
9	Post injection	5° BTDC

5.2.6 Combination 5: Injection Pressure + Injection Timing + Compressed air + Air throttling + Post injection + Air intake temperature

The last change was done by changing the air intake temperature. Here the was pre-heated up to 350℃ and then this change was added to **combination 4**and new combination was formed with injection pressure 240bar to 200bar, injection timing is set to (29°CA BTDC to 24°CA BTDC), the compressed air supply is 0.55bar, supply air is throttled up to 50%, post injection to 5° BTDC and air intake temperature is 350℃.

Table 5.5 Combination 5

S.No.	Operating Parameters	Operating Values
1	Speed	1100-1300rpm
2	Torque(@full throttle)	350Nm
3	Oil pressure	3bar
4	Air intake temperature	25-35℃
5	Injection Pressure	200bar
6	Injection Timing	24°CA BTDC
7	Compressed air supply	0.55bar
8	Air throttling	50%
9	Post injection	5° BTDC
10	Air intake temperature	350℃

After made all the combinations of changes of hardware required data were noted down to find the heat losses, brake power, indicated power and friction losses. Then the heat balance sheet was made for all the combinations and one more heat balance sheet was made for the original condition of the engine (without any modification).

Chapter 6: Observation and Calculations

6.1 Calculation of total heat input (Q_T)

According to experiment the data require to calculate the total heat input was noted down at the time of experiment. To calculate the total heat input ($\boldsymbol{Q_T}$) mass flow rate of fuel ($\boldsymbol{m_f}$) is required and the calorific value of fuel ($\boldsymbol{CV_f}$).

The formula used to calculate the Total heat input is

$$\boldsymbol{Q_T = fc \times CV_f}\,\mathrm{kW}$$

Where

Q_T=Total heat input to the engine in kW

fc=Fuel consumption (diesel) in kg/hr

CV_f=Calorific value of diesel=44800 kJ/kg K

The engine setup had fuel mass flow meter from where we got the $\boldsymbol{m_f}$at different speed of the engine (1100-1300rpm).

- **Here the torque was same for all the speed and combinations**

  Torque=350Nm

By putting into the above formula according to observed data we got the $\boldsymbol{Q_T}$**.**

At constant Torque=350Nm (At every speed and combination)

Table 6.1 Combination wise total heat input

Combinations	Fuel consumption (in kg/hr)			Total heat input (Q_T) (in kW)		
	1100rpm	1200rpm	1300rpm	1100rpm	1200rpm	1300rpm
Without any modification	10.50	10.55	10.57	130.62	131.24	131.49
1	11.53	11.55	11.57	143.48	143.7	143.93
2	11.69	11.71	11.74	145.42	145.67	146.30
3	11.75	11.83	11.85	146.17	147.16	147.41
4	11.80	11.91	11.95	146.80	148.16	148.65
5	11.90	11.95	12.00	148.03	148.65	149.28

6.2 Calculation of brake power, indicted power and friction losses

6.2.1 Calculation of brake power (bp)

Brake power was calculated by the data speed and torque as the formula given below

$$bp = \frac{2\pi NT}{60000}\,\text{kW}$$

Where N=Speed

T=Torque

6.2.2Calculation of indicated power (ip)

Here to calculate the indicated power the following formula was use

$$\eta_m = \frac{bp}{ip}$$

$$or$$

$$ip = \frac{bp}{\eta_m}\,\text{kW}$$

Where η_m= Mechanical efficiency (Which is assumed to 0.6)=0.6

6.2.3 Calculation of friction losses (f_L)

The friction loss is the difference between indicated power and brake power.

Hence the formula used is

$$\boldsymbol{f_L = ip - bp} \text{ kW}$$

Following tables are the calculated value of $\boldsymbol{bp}$, $\boldsymbol{ip}$, $\boldsymbol{f_L}$ with all types of combinations at three different speeds (1100-1300rpm) and at constant torque.

At Speed=1100rpm and Torque=350Nm

Table 6.2Combination wise BP, IP, FL

Combination	Brake power (in kW)	Indicated power (in kW)	Friction loss (in kW)
Without any modification	40.29	67.16	26.87
1	40.29	67.16	26.87
2	40.29	67.16	26.87
3	40.29	67.16	26.87
4	40.29	67.16	26.87
5	40.29	67.16	26.87

At Speed=1200rpm and Torque=350Nm

Table 6.3 Combination wise BP,IP,FL

Combination	Brake power (in Kw)	Indicated power (in kW)	Friction loss (in kW)
Without any modification	43.96	73.26	29.3
1	43.96	73.26	29.3
2	43.96	73.26	29.3
3	43.96	73.26	29.3
4	43.96	73.26	29.3
5	43.96	73.26	29.3

At Speed=1300rpm and Torque=350Nm

Table 6.4 Combination wise BP,IP,FL

Combination	**Brake power (in kW)**	**Indicated power (in kW)**	**Friction loss (in kW)**
Without any modification	47.62	79.37	31.75
1	47.62	79.37	31.75
2	47.62	79.37	31.75
3	47.62	79.37	31.75
4	47.62	79.37	31.75
5	47.62	79.37	31.75

6.3 Calculation of heat loss through cooling medium (water) (Q_W)

The heat loss through the cooling medium is depend upon the mass flow rate of the cooling water through the water jacket, temperature difference between inlet and outlet temperature of cooling water through the water jacket and specific heat of the water

The formula used to calculate heat loss through cooling medium (water) is

$$\boldsymbol{Q_W = m_w C_p \Delta T}\text{kW}$$

Where

Q_W=Heat loss through the cooling water

m_w= Mass flow rate of water through the water jacket=0.490625 kg/sec

C_p=Specific heat of water at constant pressure=4.187 kJ/kg K

ΔT=Temperature difference between inlet and outlet temperature of the Water jacket

To find the $\boldsymbol{\Delta T}$the water inlet temperature ($\boldsymbol{T_{wi}}$) and the outlet temperature ($\boldsymbol{T_{wo}}$) were observed by the temperature sensors.

- **The water inlet temperature was set to the 25℃ at every speed and modification of the engine.**

Following table is the readings of the water jacket inlet and outlet temperature

At constant Torque=350Nm (At every speed and combination)

Table 6.5 Inlet and outlet temperature of cooling water in water jacket

Combinations	T_{wi} (in ℃)			T_{wo} (in ℃)		
	1100rpm	1200rpm	1300rpm	1100rpm	1200rpm	1300rpm
Without any modification	25	25	25	28.30	28.70	29.67
1	25	25	25	29.04	31.14	33.37
2	25	25	25	29.62	31.7	33.86
3	25	25	25	30.02	31.18	34.37
4	25	25	25	30.44	32.78	34.90
5	25	25	25	31.00	33.18	35.54

After getting all the temperature values at all conditions the heat loss through the cooling medium was calculated.

The following table shows Q_W

At constant Torque=350Nm (At every speed and combination)

Table 6.6 Cooling medium heat loss

Combinations	Q_w(in kW)		
	At 1100rpm	At 1200rpm	At 1300rpm
Without any modification	06.7789	08.2169	09.5933
1	08.2991	12.6130	7.19390
2	09.4905	13.7634	18.2005
3	10.3122	14.7494	19.2482
4	11.1750	15.9819	20.3369
5	12.0542	16.8036	21.6516

6.4 Calculation of heat loss through the exhaust gas (Q_{ex})

The formula used to calculate heat loss through exhaust gas is as follows:

$$Q_{ex} = m_{ex} C_{Pex} \Delta T \text{ kW}$$

Where

m_{ex}=Mass flow rate of exhaust gas (Observed from the calorie meter)

C_{Pex}=Specific heat of exhaust gas at constant pressure =1.063 kJ/kg K

ΔT=Diference between exhaust gas inlet and outlet (Observed from the thermocouple temperature sensors)

Mass flow rate of exhaust gas varied at different speed and different combinations.

At constant Torque=350Nm (At every speed and combination)

Table 6.7 Exhaust gas mass flow rate

Combinations	m_{ex} (in kg/sec)		
	At 1100rpm	**At 1200rpm**	**At 1300rpm**
Without any modification	0.5087	0.7621	0.7537
1	0.9084	1.1927	1.4506
2	0.9621	1.2558	1.5093
3	1.0036	1.3072	1.5442
4	1.0775	1.4193	1.6307
5	1.1587	1.4046	1.7658

Then according to the formula written for Q_{ex} the following table was made

At constant Torque=350Nm (At every speed and combination)

Table 6.8 Heat loss through exhaust gas

Combinations	Q_{ex} (in kW)		
	At 1100rpm	**At 1200rpm**	**At 1300rpm**
Without any modification	07.6007	10.2300	10.3117
1	12.3238	15.6844	19.0204
2	13.2491	16.5142	19.8221
3	14.0906	17.6256	20.7877
4	15.1174	18.8363	21.5673
5	16.1032	19.9043	22.8004

6.5 Calculation of unaccounted heat loss (Q_{unac})

The heat losses to the surrounding in the form of radiation and the heat losses due to some unknown factors undergo in the section of unaccounted loss. It is the difference between the **total heat inputs** and **sum of all the heat energy used and loss.**

Unaccounted loss is shown below the table

Table 6.9 Unaccounted heat loss

Combinations	Unaccounted heat loss (Q_{unac}) (in kW)		
	1100rpm	**1200rpm**	**1300rpm**
Without any modification	49.0804	39.5331	32.2150
1	55.6971	42.1426	28.3457
2	55.5204	42.1324	28.9074
3	54.6072	41.5250	28.0041
4	53.3476	40.0818	27.3758
5	52.7126	38.6821	25.4280

Chapter 7: Result and Discussion

In the previous chapter the observation table was made and all the required data were putted into the tables. Then the data were applied in the formulas and the calculation part was done. Then the heat balance sheet was made based on all the combinations from 1100rpm to1300rpm.

7.1 heat balance sheet

1. At Speed=1100rpm
   Torque=350Nm

Table 7.1 Heat balance sheet at 1100rpm

Combinations	Fuel consumption (kg/hr)	Q_T (kW)	bp (kW)	ip (kW)	f_L (kW)	Q_W (kW)	Q_{ex} (kW)	Q_{unac} (kW)
Without any modification	10.50	130.62	40.29	67.16	26.87	06.7789	07.6007	49.0804
1	11.53	143.48	40.29	67.16	26.87	08.2991	12.3238	55.6971
2	11.69	145.42	40.29	67.16	26.87	09.4905	13.2491	55.5204
3	11.75	146.17	40.29	67.16	26.87	10.3122	14.0906	54.6072
4	11.80	146.80	40.29	67.16	26.87	11.1750	15.1174	53.3476
5	11.90	148.03	40.29	67.16	26.87	12.0542	16.1032	52.7126

2. At Speed=1200rpm
   Torque=350Nm

Table 7.2: Heat balance sheet at 1200rpm

Combinations	Fuel consumption (kg/hr)	Q_T (kW)	bp (kW)	ip (kW)	f_L (kW)	Q_W (kW)	Q_{ex} (kW)	Q_{unac} (kW)
Without any modification	10.55	131.24	43.96	73.26	29.3	08.2169	10.2300	39.5331
1	11.55	143.7	43.96	73.26	29.3	12.6130	15.6844	42.1426
2	11.71	145.67	43.96	73.26	29.3	13.7634	16.5142	42.1324
3	11.83	147.16	43.96	73.26	29.3	14.7494	17.6256	41.5250
4	11.91	148.16	43.96	73.26	29.3	15.9819	18.8363	40.0818
5	11.95	148.65	43.96	73.26	29.3	16.8036	19.9043	38.6821

3. At Speed=1300rpm

   Torque=350Nm

Table 7.3 Heat balance sheet at 1300rpm

Combinations	Fuel consumption (kg/hr)	Q_T (kW)	bp (kW)	ip (kW)	f_L (kW)	Q_W (kW)	Q_{ex} (kW)	Q_{unac} (kW)
Without any modification	10.57	131.49	47.62	79.37	31.75	09.5933	10.3117	32.2150
1	11.57	143.93	47.62	79.37	31.75	17.1939	19.0204	28.3457
2	11.74	146.30	47.62	79.37	31.75	18.2005	19.8221	28.9074
3	11.85	147.41	47.62	79.37	31.75	19.2482	20.7877	28.0041
4	11.95	148.65	47.62	79.37	31.75	20.3369	21.5673	27.3758
5	12.00	149.28	47.62	79.37	31.75	21.6516	22.8004	25.4280

In this heat balanced sheet the consumption of fuel is increasing from first combination to fifth combination. Therefore, the total heat input is also increasing according combinations. The losses through the cooling water, exhaust gas and the unaccounted loss are increasing as the combinations of modification are added one by one from (Injection Pressure + Injection Timing) to (Injection Pressure + Injection Timing + Compressed air + Air throttling + Post injection + Air intake temperature).

7.2 Comparison

From the heat balanced sheets shown in this chapter, we got the result of heat losses, brake power, indicated power and friction loss in the original condition as well as in the all types of combinations. Now it is possible to compare the results between original condition engine (without modified engine) and modified engine on the basis of heat loss at the speed between 1100rpm to 1300rpm.

7.2.1. At the speed of 1100rpm and Torque at 350Nm

Table 7.4 Comparison 1

Combination	**Fuel consumption (kg/hr)**	**Q_T (kW)**	**bp (kW)**	**ip (kW)**	**f_L (kW)**	**Q_W (kW)**	**Q_{ex} (kW)**	**Q_{unac} (kW)**
Without any modification	10.50	**130.62**	40.29	67.16	26.87	**06.7789**	**07.6007**	**49.0804**
1	11.53	**143.48**	40.29	67.16	26.87	**08.2991**	**12.3238**	**55.6971**

Due to change in injection pressure to 200bar from 240bar and the change in injection timing 29°CA BTDC to 24°CA BTDC the fuel consumption rate increased to **1.03 kg/hr**, thereby resulting in average temperature of the combustion process. Hence the total heat input also increased to **12.86kW**. As a result, it did not affect **bp, ip and f_L,** they remain same as the without modification tested value. But the heat loss through the cooling water and exhaust gases increased to **1.5201 kW and 4.7231kW** compared to the losses in the original condition engine. The unaccounted heat loss also increased to **6.6167 kW.**

Table 7.5 Comparison 2

Combination	**Fuel consumption (kg/hr)**	**Q_T (kW)**	**bp (kW)**	**ip (kW)**	**f_L (kW)**	**Q_W (kW)**	**Q_{ex} (kW)**	**Q_{unac} (kW)**
Without any modification	10.50	**130.62**	40.29	67.16	26.87	**06.7789**	**07.6007**	**49.0804**
2	11.69	**145.42**	40.29	67.16	26.87	**09.4905**	**13.2491**	**55.5204**

Due to change in injection pressure to 200bar from 240bar and the change in injection timing 29°CA BTDC to 24°CA BTDC and air was compressed up to 0.55bar by which the fuel consumption rate increased to **1.19 kg/hr**. Hence the total heat input also increased to **15 kW**. As a result, it did not affect **bp, ip and f_L,** they remain same as the without modification tested value. But the heat loss through the cooling water and exhaust gases increased to **2.7116 kW and 5.6484 kW** respectively compared to the losses in the original condition engine. The unaccounted heat loss also increased to **6.44 kW.**

Table 7.6 Comparison 3

Combination	Fuel consumption (kg/hr)	Q_T (kW)	bp (kW)	ip (kW)	f_L (kW)	Q_W (kW)	Q_{ex} (kW)	Q_{unac} (kW)
Without any modification	10.50	**130.62**	40.29	67.16	26.87	**06.7789**	**07.6007**	**49.0804**
3	11.75	**146.17**	40.29	67.16	26.87	**10.3122**	**14.0906**	**54.6072**

Due to change in **injection pressure to 200bar from 240bar and the change in injection timing 29°CA BTDC to 24°CA BTDC, air was compressed up to 0.55bar and air was throttled up to 50%,** by which the fuel consumption rate increased to **1.25 kg/hr**. Hence the total heat input also increased to **15.55 kW**. As a result, it did not affect **bp, ip and f_L,** they remain same as the without modification tested value. But the **heat loss through the cooling water and exhaust gases** increased to **3.5333 kW and 6.4899kW** respectively compared to the losses in the original condition engine. The **unaccounted heat loss** also increased to **5.5268 kW.**

Table 7.7 Comparison 4

Combination	Fuel consumption (kg/hr)	Q_T (kW)	Bp (kW)	ip (kW)	f_L (kW)	Q_W (kW)	Q_{ex} (kW)	Q_{unac} (kW)
Without any modification	10.50	**130.62**	40.29	67.16	26.87	**06.7789**	**07.6007**	**49.0804**
4	11.80	**146.80**	40.29	67.16	26.87	**11.1750**	**15.1174**	**53.3476**

Due to change in **injection pressure to 200bar from 240bar and the change in injection timing 29°CA BTDC to 24°CA BTDC, air was compressed up to 0.55bar, air was throttled up to 50% and post injection of 5° BTDC** by which the fuel consumption rate increased to **1.30 kg/hr**. Hence the total heat input also increased to **16.18 kW**. As a result, it did not affect **bp, ip and f_L,** they remain same as the without modification tested value. But the **heat loss through the cooling water and exhaust gases** increased to **4.3961 kW and 7.5167 kW** respectively compared to the losses in the original condition engine. The **unaccounted heat loss** also increased to **4.2672 kW.**

Table 7.8 Comparison 5

Combination	Fuel consumption (kg/hr)	Q_T (kW)	bp (kW)	Ip (kW)	f_L (kW)	Q_W (kW)	Q_{ex} (kW)	Q_{unac} (kW)
Without any modification	10.50	**130.62**	40.29	67.16	26.87	**06.7789**	**07.6007**	**49.0804**
5	11.90	**148.03**	40.29	67.16	26.87	**12.0542**	**16.1032**	**52.7126**

Due to change in **injection pressure to 200bar from 240bar and the change in injection timing 29°CA BTDC to 24°CA BTDC, air was compressed up to 0.55bar, air was throttled up to 50%, post injection of 5° BTDC and air was pre heated up to 350℃** by which the fuel consumption rate increased to **1.49 kg/hr**. Hence the total heat input also increased to **17.41 kW**. As a result, it did not affect **bp, ip and f_L,** they remain same as the without modification tested value. But the **heat loss through the cooling water and exhaust gases** increased to **8.5025 kW and 5.2753 kW** respectively compared to the losses in the original condition engine. The **unaccounted heat loss** also increased to **3.6322 kW**.

7.2.2. At the speed of 1200rpm and Torque at 350Nm

Table 7.9 Comparison 6

Combination	Fuel consumption (kg/hr)	Q_T (kW)	bp (kW)	ip (kW)	f_L (kW)	Q_W (kW)	Q_{ex} (kW)	Q_{unac} (kW)
Without any modification	10.55	**131.24**	43.96	73.26	29.3	**08.2169**	**10.2300**	**39.5331**
1	11.55	**143.70**	43.96	73.26	29.3	**12.6130**	**15.6844**	**42.1426**

Due to change in **injection pressure to 200bar from 240bar and the change in injection timing 29°CA BTDC to 24°CA BTDC** by which the fuel consumption rate increased to**1 kg/hr**. Hence the total heat input also increased to **12.46 kW**. As a result, it did not affect **bp, ip and f_L,** they remain same as the without modification tested value. But the **heat loss through the cooling water and exhaust gases** increased to **4.43961 kW and 5.4544kW** respectively compared to the losses in the original condition engine. The **unaccounted heat loss** also increased to **2.6095 kW**.

Table 7.10 Comparison 7

Combination	Fuel consumption (kg/hr)	Q_T (kW)	bp (kW)	ip (kW)	f_L (kW)	Q_W (kW)	Q_{ex} (kW)	Q_{unac} (kW)
Without any modification	10.55	**131.24**	43.96	73.26	29.3	**08.2169**	**10.2300**	**39.5331**
2	11.71	**145.67**	43.96	73.26	29.3	**13.7634**	**16.5142**	**42.1324**

Due to change in injection pressure to 200bar from 240bar and the change in injection timing 29°CA BTDC to 24°CA BTDC and air was compressed up to 0.55bar by which the fuel consumption rate increased to **1.16 kg/hr**. Hence the total heat input also increased to **14.43 kW**. As a result, it did not affect **bp, ip and f_L,** they remain same as the without modification tested value. But the heat loss through the cooling water and exhaust gases increased to **5.5465 kW and 6.2842 kW** respectively compared to the losses in the original condition engine. The unaccounted heat loss also increased to **2.5993 kW**.

Table 7.11 Comparison 8

Combination	Fuel consumption (kg/hr)	Q_T (kW)	bp (kW)	ip (kW)	f_L (kW)	Q_W (kW)	Q_{ex} (kW)	Q_{unac} (kW)
Without any modification	10.55	**131.24**	43.96	73.26	29.3	**08.2169**	**10.2300**	**39.5331**
3	11.83	**147.16**	43.96	73.26	29.3	**14.7494**	**17.6256**	**41.5250**

Due to change in **injection pressure to 200bar from 240bar and the change in injection timing 29°CA BTDC to 24°CA BTDC, air was compressed up to 0.55bar and air was throttled up to 50%,** by which the fuel consumption rate increased to **1.28 kg/hr**. Hence the total heat input also increased to **15.92 kW**. As a result, it did not affect **bp, ip and f_L,** they remain same as the without modification tested value. But the **heat loss through the cooling water and exhaust gases** increased to **6.5325 kW and 7.3956 kW** respectively compared to the losses in the original condition engine. The **unaccounted heat loss** also increased to **1.9919 kW.**

Table 7.12 Comparison 9

Combination	Fuel consumption (kg/hr)	Q_T (kW)	bp (kW)	ip (kW)	f_L (kW)	Q_W (kW)	Q_{ex} (kW)	Q_{unac} (kW)
Without any modification	10.55	**131.24**	43.96	73.26	29.3	**08.2169**	**10.2300**	**39.5331**
4	11.91	**148.16**	43.96	73.26	29.3	**15.9819**	**18.8363**	**40.0818**

Due to change in **injection pressure to 200bar from 240bar and the change in injection timing 29°CA BTDC to 24°CA BTDC, air was compressed up to 0.55bar, air was throttled up to 50% and post injection of 5° BTDC** by which the fuel consumption rate increased to **1.36 kg/hr**. Hence the total heat input also increased to **16.92 kW**. As a result, it did not affect **bp, ip and f_L,** they remain same as the without modification tested value. But the **heat loss through the cooling water and exhaust gases** increased to **7.7650 kW and 8.6063 kW** respectively compared to the losses in the original condition engine. The **unaccounted heat loss** also increased to **0.5487 kW**.

Table 7.13 Comparison 10

Combination	Fuel consumption (kg/hr)	Q_T (kW)	bp (kW)	ip (kW)	f_L (kW)	Q_W (kW)	Q_{ex} (kW)	Q_{unac} (kW)
Without any modification	10.55	**131.24**	43.96	73.26	29.3	**08.2169**	**10.2300**	**39.5331**
5	11.95	**148.65**	43.96	73.26	29.3	**16.8036**	**19.9043**	**38.6821**

Due to change in **injection pressure to 200bar from 240bar and the change in injection timing 29°CA BTDC to 24°CA BTDC, air was compressed up to 0.55bar, air was throttled up to 50%, post injection of 5° BTDC and air was pre-heated up to 350℃** by which the fuel consumption rate increased to **1.40 kg/hr**. Hence the total heat input also increased to **17.41 kW**. As a result, it did not affect **bp, ip and f_L,** they remain same as the without modification tested value. But the **heat loss through the cooling water and exhaust gases** increased to **8.5867 kW and 9.6743 kW** respectively compared to the losses in the original condition engine. The **unaccounted heat loss** also decreased to **0.851 kW**.

7.2.3. At the speed of 1300rpm and Torque at 350Nm

Table 7.14 Comparison 11

Combination	**Fuel consumption (kg/hr)**	**Q_T (kW)**	**bp (kW)**	**ip (kW)**	**f_L (kW)**	**Q_W (kW)**	**Q_{ex} (kW)**	**Q_{unac} (kW)**
Without any modification	10.57	**131.49**	47.62	79.37	31.75	**09.5933**	**10.3117**	**32.2150**
1	11.57	**143.93**	47.62	79.37	31.75	**17.1939**	**19.0204**	**28.3457**

Due to change in **injection pressure to 200bar from 240bar and the change in injection timing 29°CA BTDC to 24°CA BTDC** by which the fuel consumption rate increased to**1 kg/hr**. Hence the total heat input also increased to **12.44 kW**. As a result, it did not affect **bp, ip and f_L,** they remains same as the without modification tested value. But the **heat loss through the cooling water and exhaust gases** increased to **7.598 kW and 8.7087 kW** respectively compared to the losses in the original condition engine. The **unaccounted heat loss** also decreased to **3.8693 kW**.

Table 7.15 Comparison 12

Combination	**Fuel consumption (kg/hr)**	**Q_T (kW)**	**Bp (kW)**	**ip (kW)**	**f_L (kW)**	**Q_W (kW)**	**Q_{ex} (kW)**	**Q_{unac} (kW)**
Without any modification	10.57	**131.49**	47.62	79.37	31.75	**09.5933**	**10.3117**	**32.2150**
2	11.74	**146.30**	47.62	79.37	31.75	**18.2005**	**19.8221**	**28.9074**

Due to change in injection pressure to 200bar from 240bar and the change in injection timing 29°CA BTDC to 24°CA BTDC and air was compressed up to 0.55bar by which the fuel consumption rate increased to **1.17 kg/hr**. Hence the total heat input also increased to **14.81 kW**. As a result, it did not affect **bp, ip and f_L,** they remain same as the without modification tested value. But the heat loss through the cooling water and exhaust gases increased to **8.6072 kW and 9.5104 kW** respectively compared to the losses in the original condition engine. The unaccounted heat loss also increased to **3.3076 kW**.

Table 7.16 Comparison 13

Combination	Fuel consumption (kg/hr)	Q_T (kW)	bp (kW)	ip (kW)	f_L (kW)	Q_W (kW)	Q_{ex} (kW)	Q_{unac} (kW)
Without any modification	10.57	**131.49**	47.62	79.37	31.75	**09.5933**	**10.3117**	**32.2150**
3	11.85	**147.41**	47.62	79.37	31.75	**19.2482**	**20.7877**	**28.0041**

Due to change in **injection pressure to 200bar from 240bar and the change in injection timing 29°CA BTDC to 24°CA BTDC, air was compressed up to 0.55bar and air was throttled up to 50%,** by which the fuel consumption rate increased to **1.28 kg/hr**. Hence the total heat input also increased to **15.92 kW**. As a result, it did not affect **bp, ip and f_L,** they remain same as the without modification tested value. But the **heat loss through the cooling water and exhaust gases** increased to **9.6549 kW and 10.476 kW** respectively compared to the losses in the original condition engine. The **unaccounted heat loss** also decreased to **4.2109 kW**.

Table 7.17 Comparison 14

Combination	Fuel consumption (kg/hr)	Q_T (kW)	bp (kW)	ip (kW)	f_L (kW)	Q_W (kW)	Q_{ex} (kW)	Q_{unac} (kW)
Without any modification	10.57	**131.49**	47.62	79.37	31.75	**09.5933**	**10.3117**	**32.2150**
4	11.95	**148.65**	47.62	79.37	31.75	**20.3369**	**21.5673**	**27.3758**

Due to change in **injection pressure to 200bar from 240bar and the change in injection timing 29°CA BTDC to 24°CA BTDC, air was compressed up to 0.55bar, air was throttled up to 50% and post injection of 5° BTDC** by which the fuel consumption rate increased to **1.38 kg/hr**. Hence the total heat input also increased to **17.16 kW**. As a result it did not affect **bp, ip and f_L,** they remains same as the without modification tested value. But the **heat loss through the cooling water and exhaust gases** increased to **10.7436 kW and 11.2556 kW** respectively compared to the losses in the original condition engine. The **unaccounted heat loss** also decreased to **4.8392 kW**.

Table 7.18 Comparison 15

Combination	**Fuel consumption (kg/hr)**	**Q_T (kW)**	**bp (kW)**	**ip (kW)**	**f_L (kW)**	**Q_W (kW)**	**Q_{ex} (kW)**	**Q_{unac} (kW)**
Without any modification	10.57	**131.49**	47.62	79.37	31.75	**09.5933**	**10.3117**	**32.2150**
5	12.00	**149.28**	47.62	79.37	31.75	**21.6516**	**22.8004**	**25.4280**

Due to change in **injection pressure to 200bar from 240bar and the change in injection timing 29°CA BTDC to 24°CA BTDC, air was compressed up to 0.55bar, air was throttled up to 50%, post injection of 5° BTDC and air was pre-heated up to 350℃** by which the fuel consumption rate increased to **1.43 kg/hr**. Hence the total heat input also increased to **17.79 kW**. As a result, it did not affect **bp, ip and f_L,** they remain same as the without modification tested value. But the **heat loss through the cooling water and exhaust gases** increased to **12.0583 kW and 12.4887 kW** respectively compared to the losses in the original condition engine. The **unaccounted heat loss** also decreased to **6.787 kW**.

Chapter 8: Conclusion and Future Scope

8.1 Conclusion

With the advent of change in technology the companies are in a habit of varying the hardware performance parameters to keep them in tune with the Sensors and ECU, without being realizing that slight change in Hardware parameters, decreases the thermal efficiency of the engine. The accessories sensors like the pressure sensors, Accelerator sensor also hamper the vehicle performance. Keeping in this line the following conclusions were made after testing the changes in various hardware parameters:

1. It was observed that with the increase in Injection pressure (240 to 200bar) and Injection timing (29° CA BTDC to 24°CA BTDC), the total amount of heat loss through exhaust gases increases by 53.31%. This resulted in decrease in thermal efficiency by 8.67%. Thus, decreasing the Injection Pressure and Injection timing is not suggested since it decreases the thermal efficiency.
2. It was observed that with the increase in Injection pressure (240 to 200bar), Injection timing (29° CA BTDC to 24°CA BTDC) and compression of air up to 0.55bar, the total amount of heat loss through exhaust gases increases by 61.42%. This resulted in decrease in thermal efficiency by 9.90%. Thus, decreasing the Injection Pressure and Injection timing is not suggested since it decreases the thermal efficiency.
3. It was observed that with the increase in Injection pressure (240 to 200 bar), Injection timing (29° CA BTDC to 24°CA BTDC), compression of air up to 0.55bar and air was throttled up to 50%, the total amount of heat loss through exhaust gases increases by 72.29%. This resulted in decrease in thermal efficiency by 10.81%. Thus, decreasing the Injection Pressure and Injection timing is not suggested since it decreases the thermal efficiency.
4. It was observed that with the increase in Injection pressure (240 to 200 bar), Injection timing (29° CA BTDC to 24°CA BTDC), compression of air up to 0.55bar, air was throttled up to 50% and post injection of 5° BTDC the total amount of heat loss through exhaust gases increases by 84.12%.

This resulted in decrease in thermal efficiency by 12.89%. Thus, decreasing the Injection Pressure and Injection timing is not suggested since it decreases the thermal efficiency.

5. It was observed that with the increase in Injection pressure (240 to 200 bar), Injection timing (29° CA BTDC to 24°CA BTDC), compression of air up to 0.55bar, air was throttled up to 50% and post injection of 5° BTDC the total amount of heat loss through exhaust gases increases by 83.93%. This resulted in decrease in thermal efficiency by 11.71%. Thus, decreasing the Injection Pressure and Injection timing is not suggested since it decreases the thermal efficiency.

8.2 Future scope

It has been concluded that with the change in various hardware parameters there is an increase in thermal efficiency and fuel consumption. The possibilities of exergy analysis on factors considered in this exertion could be found for finding the optimum levels of hardware parameter change.

Chapter 9: Trouble Shooting

No work can be done without facing any problems. It is the best source of learning and exploring many things. Many problems were faced during commissioning of the project. These were listed as shown in table below:

S. No	Problem Faced	Causes of problem found	Remedy employed.
1.	**Less amount of Fuel was transferring to the FIP (Fuel injection pump)**	a. Fuel Filter clocked. b. Fuel injection line blocked.	Fuel injection line was found damaged. The line was changed.
2.	**Exhaust Line was damaged.**	a. Improper designing of the exhaust line. b. High amount of vibration. c. Heavier exhaust line.	The entire exhaust line was redesigned and this time lighter material was chosen (Stainless steel with nickel coating).
3.	**Fuel temperature control unit not working.**	a. Blockage in water flow in piston movement of the unit (drainer) b. Bad quality of water.	The drainer was opened and was cleaned with the MTO.
4.	**Torque reading was showing +20 Nm even when the engine is stopped.**	a. Problem in the load cell of the dynamometer. b. Dynamometer needs calibration.	The load cell was found faulty and thus replaced with new one. The dynamometer was then calibrated.
5.	**High amount of vibration in the engine.**	a. Dampers out of performance. b. Improper balancing.	The engine was tested for vibration test. The area of high vibration was found and thus the proper damper was provided.

6.	**The exhaust manifold studs were broke down due to high amount of vibration in the exhaust line.**	a. Improper designing of the exhaust line. b. High amount of vibration. c. Heavier exhaust line.	The entire exhaust line was redesigned and the lighter material was chosen.
7.	**Oil sump was found leaking.**	a. High vibration. b. Gaskets torn. c. Crack	New oil sump was ordered and was replaced with the new one.
8.	**Blow by gas line was found filled with Engine oil.**	a. High amount of engine oil in the crankcase.	All the lubricating oil was drained off and was again filled.
9.	**Temperature of the Oil gallery was high i.e., 106^{o}C**	a. Improper insulation of the exhaust line. b. Low bend exhaust line design.	The exhaust line was properly insulated with glass wool. Thermocouple temperature was controlled (65^{o}C).
10.	**Temperature of the exhaust was more than 650^{o}C.**	a. Faulty Turbocharger. b. Setting of turbocharger was altered.	The turbocharger was opened and was found improper installation of waste gate. Thus it was properly installed and the temperature was controlled (580^{o}C).
11.	**Exhaust line breakdown during operation.**	a. Improper designing of the exhaust line. b. Improper insulation of the exhaust line. c. Improper material of the exhaust line.	The mild steel exhaust line is brought from the market, and is welded. The temporary arrangement was made.
12.	**Oil conditioning unit was delivering desired**	a. Any clogging might have occurred which is opposing the oil to	The staife valve was found out of function. The staife valve was changed.

	amount of temperature.	reach the engine. b. Heaters not working. c. Staife valve not working.	Transformer was also changed. It was found that staife valve was out of function because of transformer improper functioning.
13.	**Fuel mass flow meter was not delivering the fuel.**	a. Solenoid not working. b. Air bubbles in the fuel lines. c. No venting.	It was found that the piece of Teflon tape has entered the solenoid valve when opened. The foreign substance was removed.
14.	**Air bubbles in the fuel delivery line.**	a. Fuel mass flow meter not delivering fuel.	The fuel mass flow meter was checked and was found that 24V was not getting to the solenoid valve. The reason was that the maintenance limit of the fuel mass flow meter has been reached. The limits were increased.

Table 9.1: The various problems and their trouble shooting employed during the accomplishing of the thesis.

Chapter 10:References

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