A DISSERTATION REPORT ON

PERFORMANCE AND EMISSION STUDIES OF A TURBOCHARGED STATIONARY CI ENGINE

SUBMITTED IN PARTIAL FULFILLMENT FOR THE AWARD OF MASTER OF TECHNOLOGY IN ENERGY ENGINEERING

BY

RANA VEER PRATAP SINGH

(2013 PME 5118)

UNDER THE GUIDANCE OF

PROF. DILIP SHARMA

DEPARTMENT OF MECHANICAL ENGINEERING

DEPARTMENT OF MECHANICAL ENGINEERING

MALAVIYA NATIONAL INSTITUTE OF TECHNOLOGY

(June 2015)

MALAVIYA NATIONAL INSTITUTE OF TECHNOLOGY
DEPARTMENT OF MECHANICAL ENGINEERING

JAIPUR – 302017 (RAJASTHAN) INDIA

CERTIFICATE

This is to certify that the dissertation report entitled **"PERFORMANCE AND EMISSION STUDIES OF A TURBOCHARGED STATIONARY C.I. ENGINE"** is being submitted by **Mr. RANA VEER PRATAP SINGH (2013 PME 5118)** to the Malaviya National Institute Of Technology, Jaipur for the award of the degree of **Master of Technology in Energy Engineering** is a bona fide record of original research work carried out by him. He has worked under my guidance and supervision and has fulfilled the requirement for the submission of this thesis, which has reached the requisite standard.

Date:

Place:

Prof. Dilip Sharma

Department Of Mechanical Engineering

MNIT Jaipur

MALAVIYA NATIONAL INSTITUTE OF TECHNOLOGY
DEPARTMENT OF MECHANICAL ENGINEERING

JAIPUR – 302017 (RAJASTHAN) INDIA

DECLARATION

I **RANA VEER PRATAP SINGH** hereby declare that the dissertation entitled **"PERFORMANCE AND EMISSION STUDIES OF A TURBOCHARGED STATIONARY C.I. ENGINE"** being submitted by me towards partial fulfillment of the degree of **M. Tech (Energy Engineering)** is a research work carried out by me under the supervision of **PROF. DILIP SHARMA,** and has not been submitted anywhere else. I also certify that no part of this dissertation work has been copied or borrowed from anyone else. In case any type of plagiarism is found out, I will be solely and completely responsible for it.

Date:
Place:

Rana Veer Pratap Singh
M. Tech. (Energy Engg.)
2013 PME 5118

Acknowledgement

I would like to express my sincere gratitude to **Prof. Dilip Sharma**, for his invaluable guidance, advice and their commitment in providing me with the guidance. He provided constant encouragement and unceasing enthusiasm at every stage of the dissertation work. He has been the principal motivation behind this work and provided all kind of possible support. I am very much thankful to him for his generosity and extending maximum possible help whenever required.

I am extremely thankful to **Prof. S.L Soni**, Incharge I.C.Engine lab, MNIT Jaipur for allowing me to work on the Internal Combustion Engine available in the lab.

I also express my indebtedness to **Prof. Rakesh Jain**, Head of Mechanical Engineering department, MNIT, Jaipur for providing me all possible support and valuable advices.

I want to express my deepest gratitude to **Dr. Nirupam Rohatagi, Prof. Jyotirmay Mathur and Dr. G.D. Aggarwal** for showing me the right direction during the duration of this project.

I am also thankful to technical staff **Mr. Ramesh Chand Meena, Mr. Kolahal Prasad and Mr. Mahaveer,** Mechanical Engineering lab to provide all kind of help during my dissertation work.

I would like to thank my friends **Govind Ram Meena, Pushkar Chandra, Saurabh Kumar, Umesh Kumar Sahu** and **Manmeet Kumar** for their sincere help and moral support during the entire work

I am also immensely grateful to **Mr. Pradeep Gupta, Mr. Jitendra Sekhawat, Mr. Animesh K. Srivastava and Mr. Dheeraj Kishor Johar** for their sincere help and moral support during entire work.

Lastly, but not least I thank one and all who have helped me directly or indirectly in completion of the dissertation work.

.

(Rana Veer Pratap Singh)

ABSTRACT

To meet the engine downsizing requirements, turbocharging the engine has become an essential tool in the automobile industry. Downsizing is the replacement of a larger size engine with one of smaller size engine of higher specific power. Turbochargers used in automobiles must be able to produce a wide flow range and a high transient response. Ideally, an engine should be able to simultaneously offer high power density and low fuel consumption. The reduction in fuel consumption causes reduction of harmful exhaust gases, which is necessary to cope with the increasingly stringent emission regulations. The operating conditions for stationary diesel engines are much easier than those for automotive vehicles as these engines operate in a certain power range at fixed RPM, whereas automotive vehicle engine operate over wide range and operating conditions changes frequently. A lot of research work has been done on Turbocharging in automotive engines but for stationary diesel engines it is rare. The focus of this study was to install a turbocharger on a small stationary C.I. engine (Power rating 5 hp @1500 RPM) and compare the performance and emission characteristics of this turbocharged stationary C.I. engine with the naturally aspirated one. This report also presents the brief discussion about the significant waste energy recovery technologies and the developments in Turbocharging technology. In this work the selection of an appropriate turbocharger to match the engine existing in the lab was the key challenge. Preparation of experimental set up of small stationary diesel engine with turbocharger was a critical task. The experimental results show that the brake thermal efficiency of the engine has been increased by up to 8.5% and the volumetric efficiency by up to 32%. Considerable reduction in Hydrocarbon and NOx emissions was observed but CO, CO_2 and smoke percentage increased due to Turbocharging of the engine. The turbocharged engine's performance improved significantly compared to the naturally aspirated engine above 2 kW loading condition.

TABLE OF CONTENTS

CHAPTER-1

CHAPTER-2

CHAPTER-3

LIST OF FIGURES

LIST OF TABLES

ABBREVIATIONS

Symbols	Description
EER	Exhaust Energy Recovery
VNT	Variable Nozzle Turbine
VGT	Variable Geometry Turbocharger
BSFC	Break Specific Fuel Consumption
BMEP	Break Mean Effective Pressure
BSEC	Break Specific Energy Consumption
BTE	Break Thermal Efficiency
C I	Compression Ignition
S I	Spark Ignition
CO	Carbon Monoxide
CO_2	Carbon Dioxide
HC	Hydrocarbon
NOx	Oxides of Nitrogen
BDC	Bottom Dead Center
TDC	Top Dead Center
HP	Horse Power
RPM	Rotation per Minute

CHAPTER-1

INTRODUCTION

A naturally aspirated Internal Combustion Engine produces large amount of waste energy through exhaust manifold and finally to the environment. Typical internal combustion engine lose about 70% of the fuel energy through the engine coolant, exhaust and surface radiations. The percentage amount of energy rejected to the environment through exhaust gas which can be potentially recovered is approximately 30-40% of the energy supplied by the fuel depending on engine load. Green house effect, diminishing petroleum supplies indicating a trend for increasing fuel prices in future have motivated governments and I.C. engine manufacturers to reduce engine emissions and increase fuel economy. Various efforts have been made to improve diesel engine efficiency and fuel economy and thus, to reduce CO2 and other gaseous particulate emissions. Two ways of improvement in I.C. engine efficiencies are, 1-improving the thermodynamic efficiency of the operating cycle and 2- reducing its mechanical and thermal losses. In general it has been widely accepted that the improvement of I.C. engine efficiency using internal measures is accompanied by a significant increase of peak combustion pressure. So, the improvement in the efficiency of the operating cycle has been accomplished in diesel engines by using various internal measures such as, increased injection pressure, improvement of the air fuel mixing mechanisms etc. despite improvement in the efficiency using internal measures, I.C. engines still reject a significant portion of thermal energy to the environment through the exhaust gas. Approximately 30-35% of fuel energy is rejected to the ambience through the exhaust gas. Figure 1 shows the total energy distributions from internal combustion engines

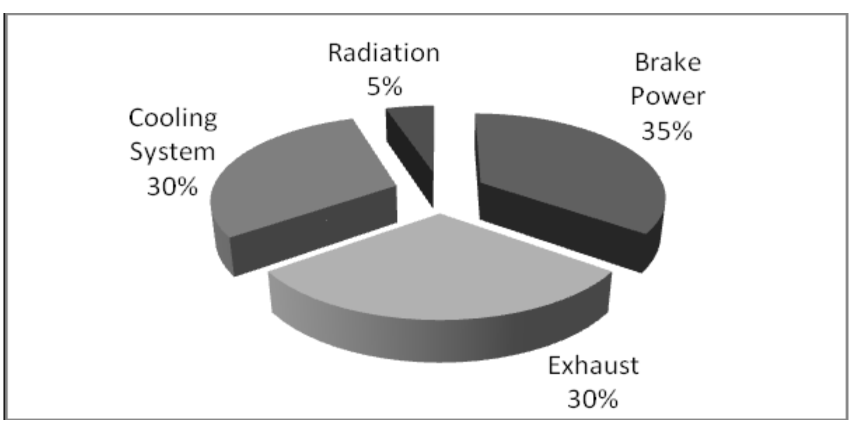


Figure 1.Total Fuel Energy Content in I. C. Engine [1]

Therefore, the exhaust energy recovery is a technology that contributes to substantial improvement of overall I.C. engine efficiency and can be used in parallel with all other engine techniques. In order to recover the waste exhaust energy various technologies, like Thermo-Electric Generator (TEG), Organic Rankine Cycle (ORC), Six-stroke internal combustion engine, Turbocharger etc., have been developed. But, turbochargers are being widely used, especially in automotive industry, because of their compactness, light-weight, portability, durability and low cost in comparison to other waste energy recovery technologies.

A turbocharger in its simplest definition is a type of supercharger that is driven by exhaust energy. To improve the power performance and fuel economy of internal combustion engines, boosting intake pressure is a useful way. Boosting intake pressure results in downsizing I.C. engine displacement, that is why most of modern advanced I.C. engines adopt various technologies of boosting pressure. I.C. engine energy balance indicates that, higher intake pressure results in higher brake mean effective pressure (BMEP), and finally it leads to higher thermal efficiency and power **[2].** As a boosting pressure technology exhaust driven supercharger i.e. turbocharger is being used most widely. Actually it is a kind of method for I.C. engine exhausts energy recovery (EER), because it uses I.C. engine exhaust pressure energy to drive the turbine **[3].** The increase in the air mass flow rate into the engine due to Turbocharging significantly reduces particulates for diesel engines, which are released into the atmosphere. Furthermore, **Saidur et al. [4],** in their review paper reported that turbocharged diesel engine can improve the fuel economy of passenger vehicles up to 30-50% and downsized turbocharged gasoline engine by 5-20%. BMW was the first to produce turbocharged passenger car, which they launched in 1973. The car was brilliantly packaged and paved the way for a simply magnificent Turbo Era in the automotive world **[5].** Turbochargers have been widely used for "engine downsizing" purpose because they can largely enhance the engines power and torque output without the need of increasing the swept volume of each cylinder. Even more power obtained by turbocharged downsized diesel engines, the slower response of the turbine at low engine speeds (turbo-lag), typically in a range of 1000-2000 RPM, appears to be a common problem. Turbo lag can poorly affect the drivability and performance of the engine. Superchargers have no lag time because they are driven directly by the crankshaft. But continuous compression of intake air requires some mechanical energy to accomplish, so the supercharger imposed a cost of reduced fuel efficiency when the engine is operating at low power range or when the engine simply unloaded and idling. Various solutions have been proposed and studied in order to reduce the turbo lag and improve the performance of turbocharger at low engine speed range. Variable Geometry Turbocharger (VGT), VGT with motor assistance, two stage turbocharger and three stage turbocharger, Turbocompounding, and twin charged engine etc. are the developments which have been made in order to reduce turbo lag and enhance the performance of turbocharged diesel engines. This report apart from experimental work summarizes and reviews the state of research in recovery of waste heat in internal combustion engine using Turbocharging principal. The goal of this project report is to inform the community about the advances in the developments of Turbocharging technology for recovery of waste energy from I.C. engines. Furthermore the focus is to review the drawbacks or deficiencies related to Turbocharging approach and latest developments to remove the drawbacks and improve the performance of turbocharged internal combustion engine.

1.1 SIGNIFICANT ENERGY RECOVERY TECHNOLOGIES

There are several methodologies to recover exhaust gases from automotive vehicles, whereas the dominating ones are;

1.1.1 THERMOELECTRIC GENERATOR (TEG)

It is a device in which thermoelectric modules (solid state device) are used to convert thermal energy from a temperature gradient to electrical energy. Modules are sandwiched between a heat source and a heat sink. This device works on the basic principle of **"Seebeck effect"**, ***"when a temperature difference is established between the hot and cold junctions of two dissimilar materials (metals or semiconductors) a voltage is generated", i.e. Seebeck voltage***

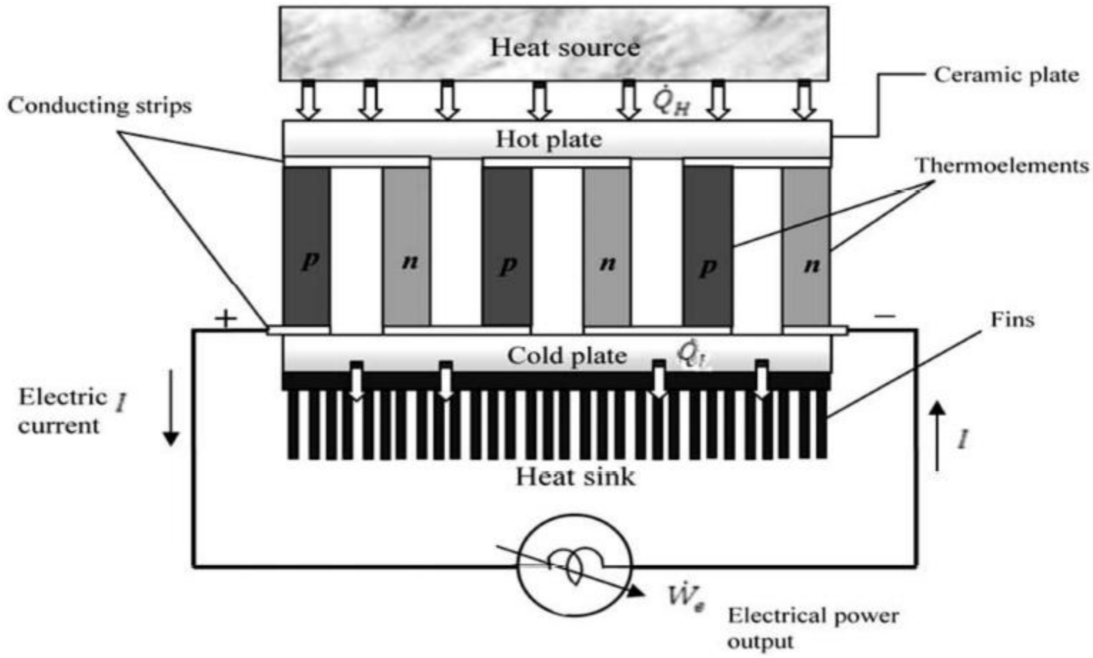


Figure 2 Schematic diagram showing components and arrangement of a typical single-stage thermoelectric power generator [6].

1.1.2 ORGANIC RANKINE CYCLE (ORC)

A special case of low temperature energy generation system is the use of certain organic fluids instead of water is so-called Organic Rankine Cycle (ORC).In this system the steam generation takes place in a secondary circuit due to evaporation of isentropic fluid having low latent heat of vaporization, using the exhaust gas thermal energy to produce additional power by means of a steam expander.

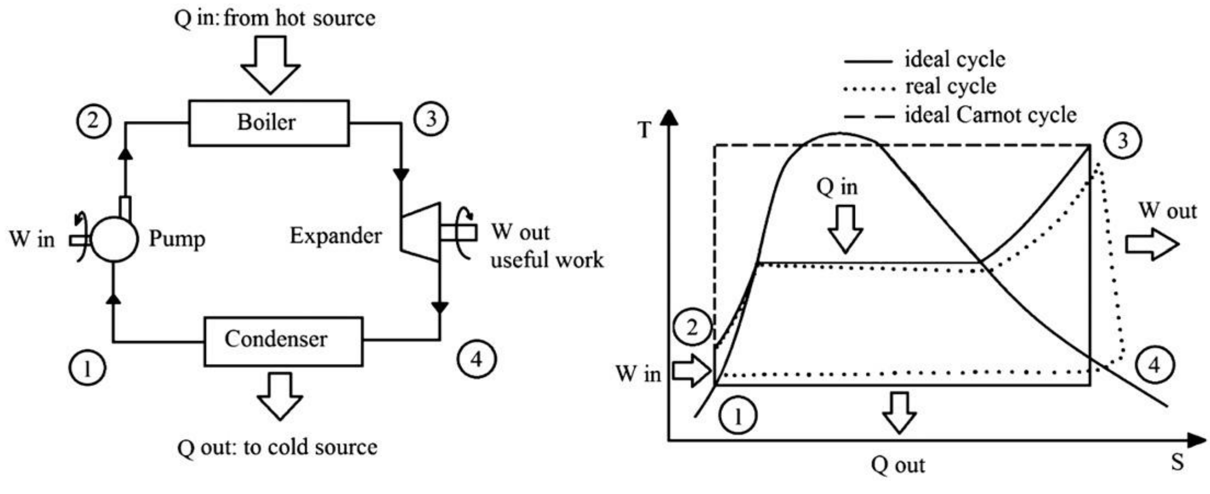


Figure 3 Rankine cycle system [4]

1.1.3 TURBOCHARGING

The energy available in the engine's exhaust is used to drive the turbocharger turbine which drives turbocharger compressor and it raises the inlet fluid density prior to entry to each engine cylinder and in turn provides boost to the inlet air (or mixture).

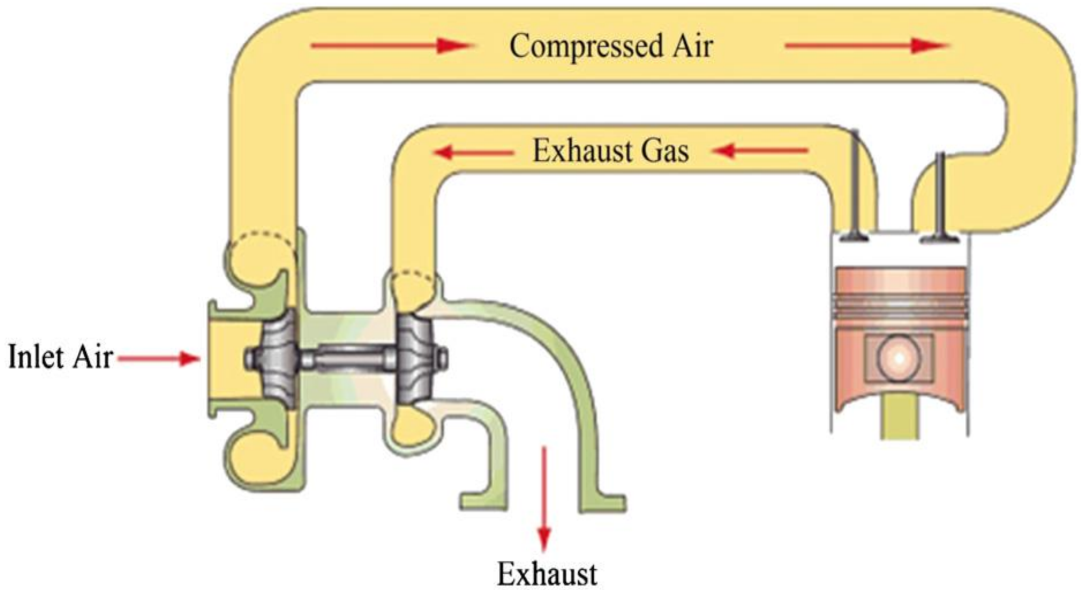


Figure 4 Typical turbocharger with compressor wheel and turbine [4]

1.1.4 TURBOCOMPOUNDING

Turbocompounding Of an engine refers to the use of a turbine to recover energy from the exhaust system of an internal combustion engine and reintroduce that energy back into the engine. It increases thermal efficiency and reduces toxic emissions of an engine. Turbocompounding are classified as;

1. **MECHANICAL TURBOCOMPOUNDING**

   In mechanical Turbocompounding, the energy recovered from exhaust gases is converted into kinetic energy and then added to the engine torque through system of shafts, gears and fluid couplings. Research studies reveals that mechanical Turbocompounding has negative effect at low loads and idling speeds and system work as an energy consumer, thus pulling down the average bsfc. Various researchers reported that the improvement in bsfc of a heavy duty diesel can be obtained in the range of 4-6% at full load and ~2-3% at part load condition

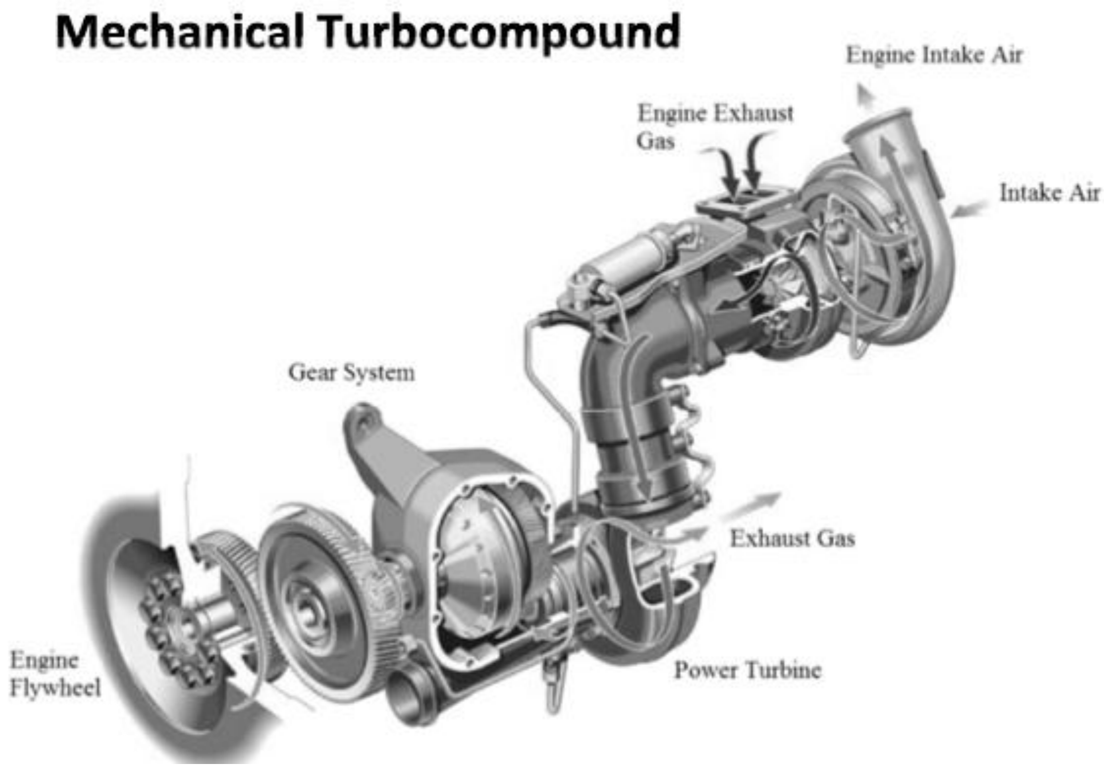


Figure 5 A typical layout of Mechanical Turbocompounding [7]

2. ELECTRICAL TURBOCOMPOUNDING

In electrical Turbocompounding the energy recovered from exhaust gases is converted into electrical power and then transmitted to the engine by a power electronics module. The important advantage offered by electrical Turbocompounding over mechanical Turbocompounding is that the former allows precise control of turbogenerator performance. This means that heat recovery can be optimized over a wide range of engine loadings, leading to greater overall efficiency of the engine

.

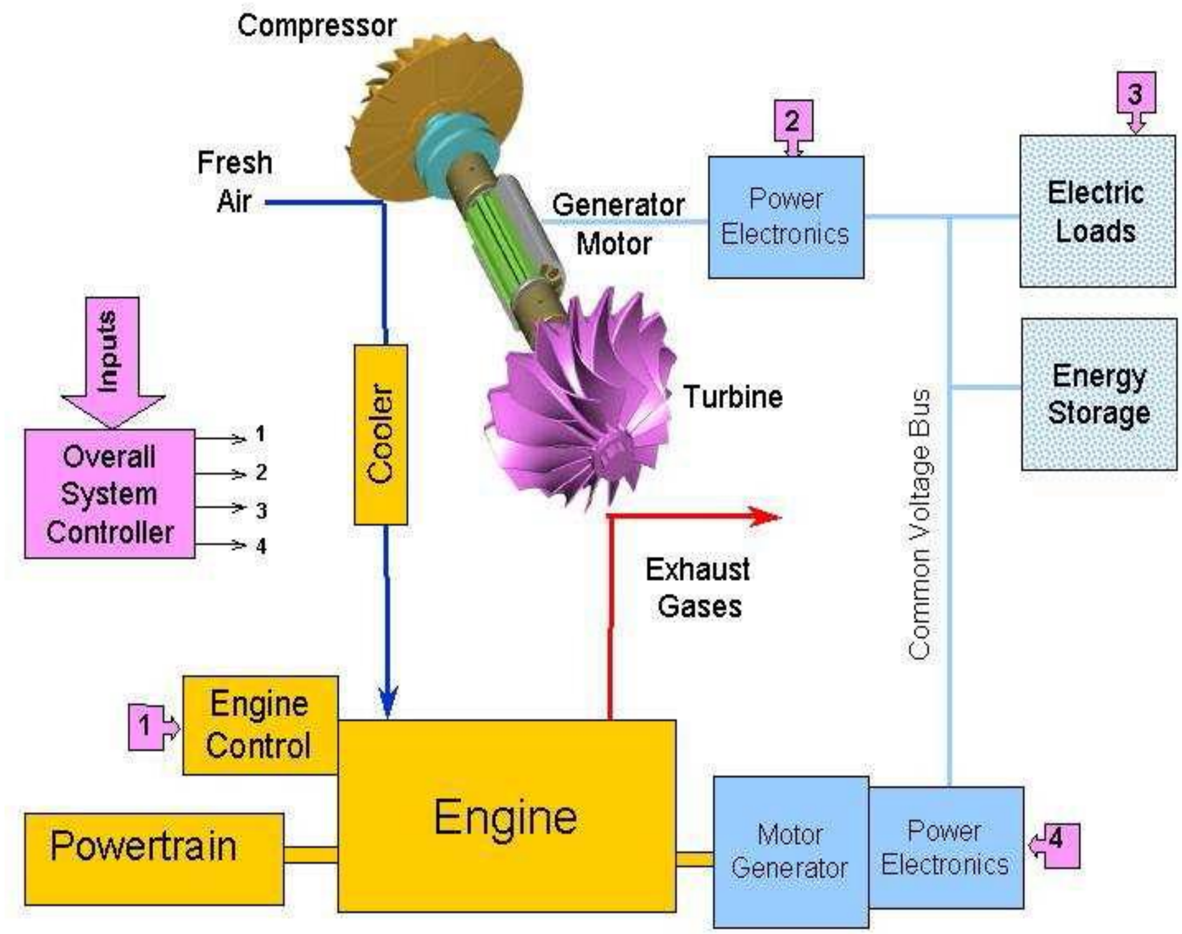


Figure 6 Electric Turbocompounding System [8]

1.1.5 SIX-STROKE INTERNAL COMBUSTION ENGINE CYCLE

The basic concept is similar to four stroke engine cycle but with two added strokes to produce higher efficiency and reduced emissions. In the proposed six-stroke cycle there are two methods of operation:

1. In the first method water is injected after complete finish of exhaust stroke. But when the water is injected into the cylinder it produces impingement on the combustion chamber surfaces, because hot combustion chamber surfaces are utilized as primary heat source.

2. The second method is by trapping and recompressing some of the exhaust gases from the fourth piston stroke, followed by a water injection and expansion of the resulting steam/exhaust mixture

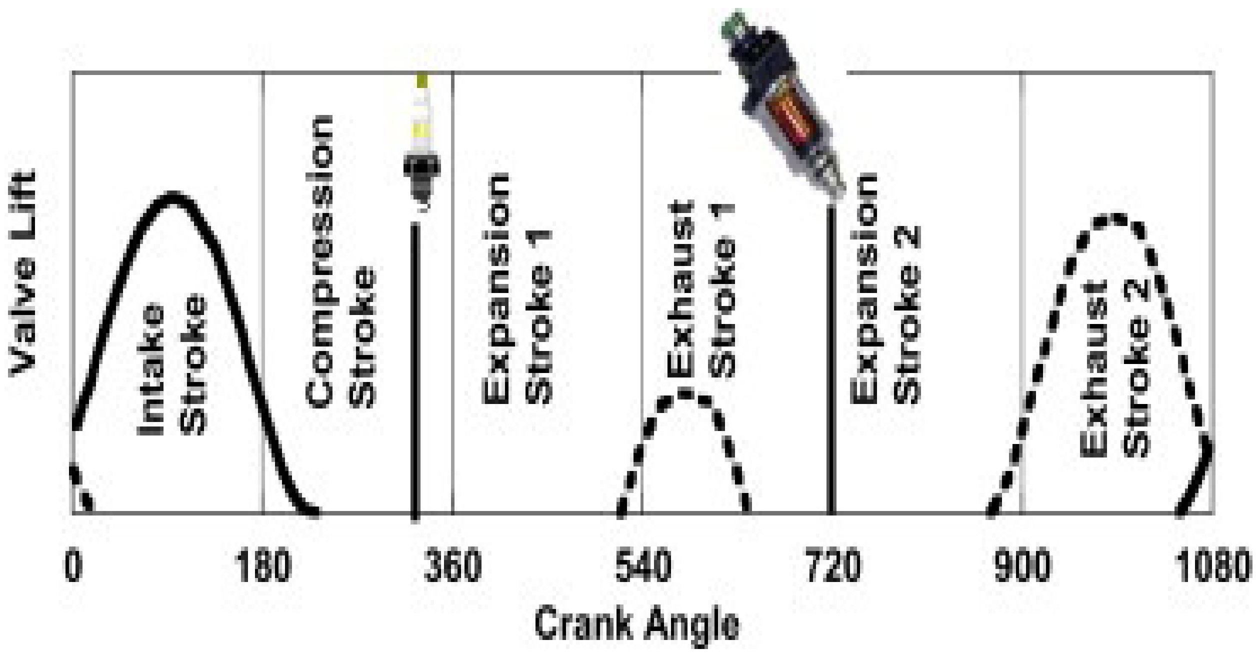


Figure 7 Six-stroke internal combustion engine cycle [4]

1.2 TRENDS IN TURBOCHARGER DEVELOPMENT

1.2.1 VARIABLE NOZZLE TURBINE (VNT)

In this type of turbine exhaust flow is controlled by the use of variable vanes. It is also known as Variable Nozzle Turbine (VNT). In VGT turbocharger the small movable vanes direct the exhaust flow throw the turbine blades. At different ranges of speed the angle of the vanes would vary to optimize the flow of the exhaust gas.

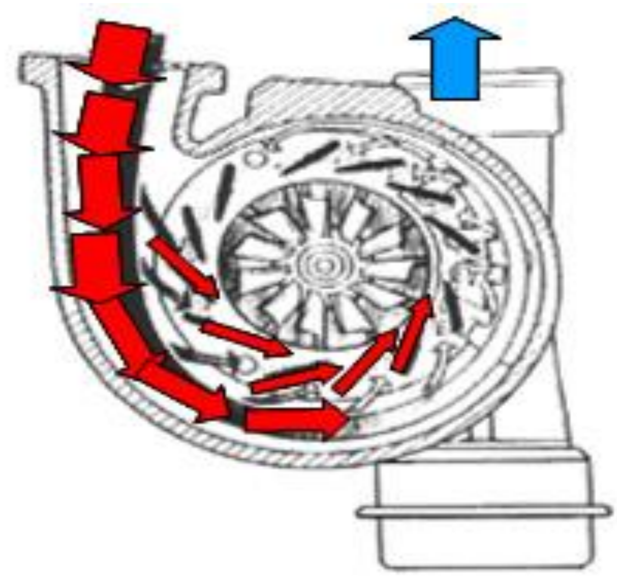
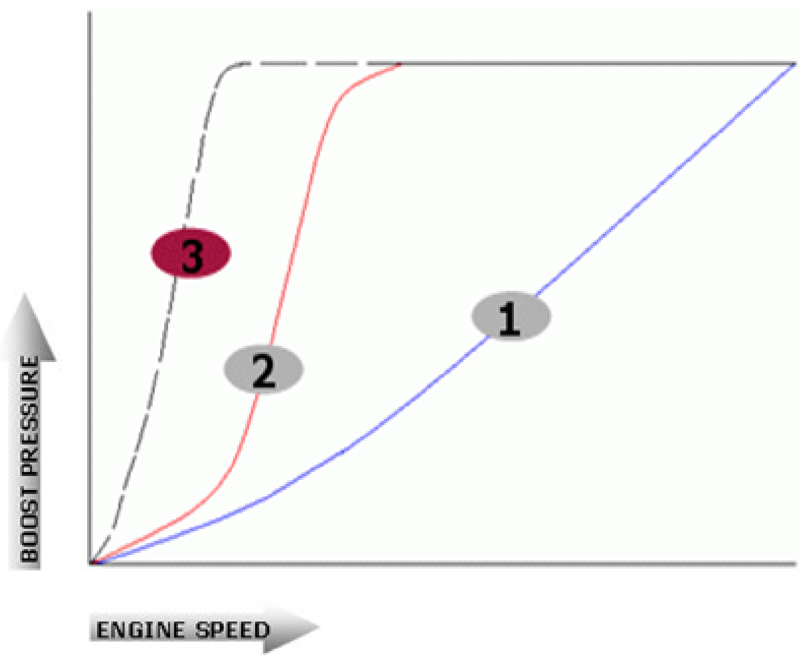


Figure 8 Variable Geometry turbocharger-air flow



1 Turbocharger without waste gate 2 turbocharger with wastegate 3 VGT (Variable Geometry Turbocharger)

Figure 9 Graph between boost pressure vs engine speed.

1.2.2 MULTI-STAGE TURBOCHARGING

Multistage Turbocharging is being used in order to improve the BSFC when the engine is working on steady conditions as well as to optimize the transient response of the engine. A two-stage turbocharger has two different sized turbochargers assembled in serial configuration. The smaller sized turbocharger responds at lower speed by producing higher torque and the larger sized turbocharger provides boost at higher engine speeds that will result in to reduction of fuel consumption and improvement in the efficiency of the engine. The introduction of two- stage Turbo charging can significantly boost for higher charging pressure over the entire engine speed map. In a study on BSFC of class 8 trucks by Bowman Company it is found that; Mechanical turbo compounding produces an average improvement of around 3%, but have a negative effect depending upon engine load whereas improvements offered by electrical turbo compounding are closer to 7% , with no negative impact at any point.

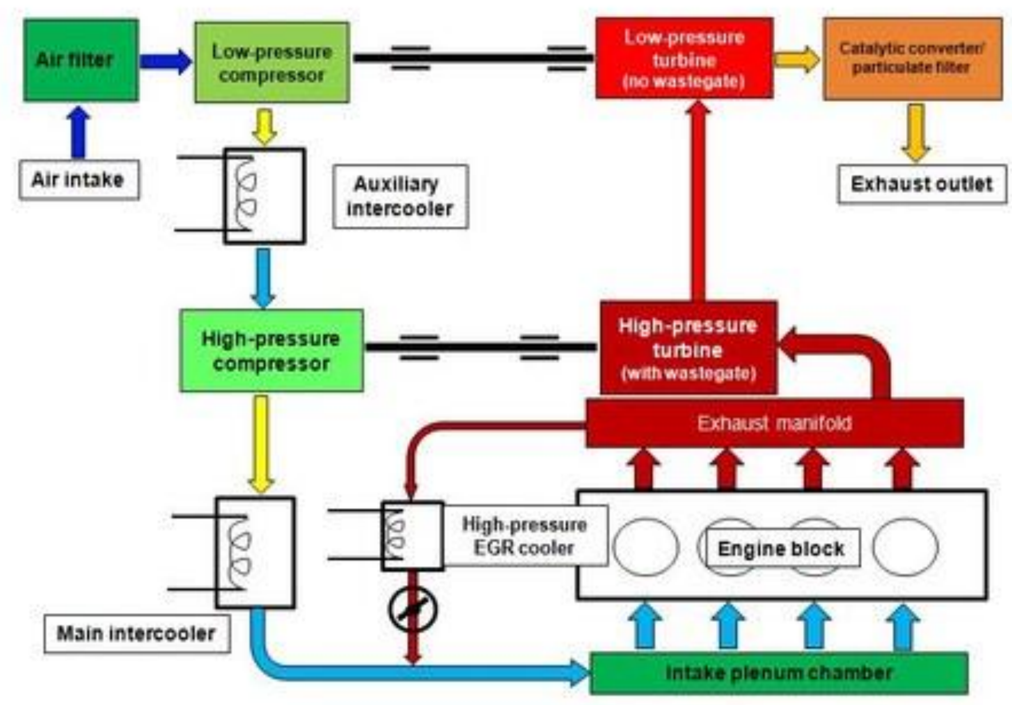


Figure 10 Schematic diagram of two-stage turbocharging – commercial vehicle large-scale applications variant: with intercooling [9]

CHAPTER-2

LITERATURE REVIEW

Turbocharger in its simplest definition is a supercharger driven by a gas turbine which uses the energy in the exhaust gases. The major parts of a turbocharger are turbine wheel, turbine housing, compressor wheel compressor housing, and turbo shaft connecting turbine wheel and compressor wheel and bearing housing. During the engine operation hot exhaust gases coming out through exhaust manifold of the engine are fed through the turbine housing of the turbocharger. As the gases pass through the turbine housing they strike the fins or blades on the turbine wheel and rotate the turbine wheel rapidly. Turbine wheel rotates the compressor wheel as the turbine wheel is connected to the compressor wheel through turbo shaft. As the compressor wheel rotates it sucks the air into the compressor housing. Centrifugal force throws the air outward, and the compressed air at higher pressure is discharged to the engine cylinder. Therefore the mass flow rate of air is increased and in turn engine breathing ability i.e. volumetric efficiency is also increased. So, the turbocharged engine of the same capacity will produce more power than the naturally aspirated engine.

Various research studies, theoretical as well as experimental, have been performed on the Turbocharging of internal combustion engines with different approaches of Turbocharging. This chapter deals with the detailed review of the literature (theoretical and review papers with some experimental studies) on waste energy recovery from internal combustion engines using Turbocharging.

2.1 DIFFERENT APPROACHES OF TURBOCHARGING

Jianqin et al [2] proposed a new concept of steam Turbocharging. They reported that the form of exhaust energy is thermal energy since it takes the largest portion all along and justified their stand with steam Turbocharging approach, which shows that the power of 1.5 L four cylinder diesel engine can be improved by 7.2% at most, and thermal efficiency can be improved by 2% points or more except at 1000 RPM, and it does not suffer the problem of exhaust back pressure. But the steam Turbocharging system based on open steam power cycle limits its application to stationary engines e.g. genset, steam plants etc. However, closed steam power cycle can find wide application. But size and cost of the system restricts the application on small automotive vehicles.

Jianqin et al [12] in another study explored an approach of steam assisted Turbocharging and compared the performance of exhaust Turbocharging, steam Turbocharging and steam assisted Turbocharging on a passenger car gasoline engine. The results show that at engine speed lower than 2000 RPM, steam Turbocharging has higher exhaust energy recovery efficiency (maximum 6.5%) than the other two kinds of boosting pressure approaches. The exhaust gas energy recovery efficiency of steam assisted Turbocharging (maximum 5.5%) is greater than exhaust Turbocharging but lower than steam Turbocharging. Steam-assisted Turbocharging can improve I.C. engine intake pressure at low engine speed, while the steam Turbocharging can achieve the target intake pressure in the entire speed range. As the engine speed increases, the exhaust energy recovery efficiency of steam Turbocharging decreases, while the recovery efficiency of steam-assisted Turbocharging and exhaust Turbocharging first increases and then decreases.

Vijay [13], in his review paper entitled “research in mechanical engineering on recovery of waste heat in internal combustion engine”, suggested that the use of Variable geometry turbine is a possible solution to minimize the turbo lag effect at low load and idling speed conditions. Variable geometry turbine (VGT) also known as the Variable Nozzle Turbine (VNT) is a type of turbine where the turbo controls the exhaust flow through the movable turbine blades. Further he says that in India usually in hill areas the power output of engine which is reduced during operation of air conditioner, can be mitigated by using exhaust heat to develop steam and then run the mini turbine to operate the compressor and hence the air conditioning of the automobile.

Ma et al. [14] mentioned in their work that, conversion of exhaust heat energy of the engine into electrical energy would directly reduce system fuel consumption, increase available electrical power and improve overall system efficiency, if the electrical power is added to the engine.

Koyess [15] in his experimental investigation on an F1 cosworth vehicle, found a power increase of about 7% over the useful range of the engine, from 8000 to 10000 rpm, and the peak power increase of 8% results into a 2% increase in efficiency.

Zhuge et al. [16] performed a simulation study on a gasoline electric turbocompoundig system using fixed geometry turbine against variable nozzle turbine (VNT) through engine performance GT DRIVE software, under United States Environmental Protection Agency (US EPA) standard US06 and FTP75 driving cycles. Results show that the electrical turbocompounding system with VNT has no considerable improvement on engine fuel economy under practical driving cycles; it can improve the engine efficiency only at high load operating conditions and increase the electrical power generation.

C. Bipin et al. [17], in their research paper "Study on waste heat recovery in an internal combustion engine", presented an experimental investigation. Throughout the task they made an attempt to look at the various possible methods of waste heat recovery in conventional commercial two wheeler and four wheelers. The study presents the concept of electro turbocharged hybrid engine with cogeneration, in which heat contained in exhaust gases is recovered in an auxiliary combustion chamber and at that temperature (in a petrol engine lies between 200 to 230^{o}C) fuel is injected at comparatively low injection pressures and burnt, and then high temperature gas is introduced to turbine stage where it is expanded. Their experimental model consist a turbocharger attached to the exhaust manifold of a HERO HONDA CD Deluxe bike, and the turbocharger shaft is coupled to a dc generator of voltage rating of 6 V. From the experiment, the power obtained by connecting the alternator to the turbocharger is 5.184 Watts which is 0.035% of the total power supplied by the fuel; hence out of 39% exhaust losses 0.06% can be recovered by electro turbo charging in that engine, which shows that the useful work obtained has been increased from 24% to 24.035%.

Bowman Company **(bowman power.com)** demonstrated the impact of mechanical and electrical Turbocompounding techniques on bsfc improvement in class 8 trucks, and found that mechanical Turbocompounding proved an average bsfc improvement of ~3%, but can have negative effect depending on engine load, whereas, improvement offered by electrical Turbocompoundig are closer to 7% with no possibility of a negative impact at any point.

Variable Geometry Turbochargers are turbochargers whose geometry and thus effective A/R can be changed as needed while in use. The most common design includes several adjustable vanes around a central turbine. As the angle of the vanes change, the angle of air flow onto the turbine blades changes, which changes the effective area of the turbine, and thus the aspect ratio (A/R) changes. The variable geometry (VG) turbocharger is an indispensable application that not only improves drivability and fuel consumption, but also satisfies stringent emission regulations **[18]**. Due to the requirement of accurate engineering mechanisms and high-strength materials to ensure reliability of variable geometry turbocharger the price is relatively high compared with that of waste gate turbochargers.

In a review paper on Trend of Turbocharging Technologies by **Hiroshi Uchida [19],** it has been suggested that the turbocharged engine with the VNT can raise the boost by controlling the VN at low engine speeds. However, the amount of pressure increase at low engine speeds is limited due to the low exhaust energy. He also depicted that, by adding motor assistance (MAT), we should be able to attain torque characteristics on a par with a large-displacement engine. A MAT (motor assisted turbine) by acting as a dynamo at high engine speeds can also recover exhaust energy. As a result of this we can expect a considerable improvement in fuel consumption.

Jaehoon et al.[20] in their experimental investigation shown that When a 2.5 liter DI Diesel engine equipped with VGT is compared with mechanically controlled waste gated turbocharger under the same limitation of a maximum cylinder pressure and exhaust smoke level, the low speed torque could be enhanced by about 44% at maximum. At high engine speed, with the same fuel delivery, the rated power can be enhanced by 3.5%, mainly caused by the reduction of pumping loss.

Turbochargers suffer turbo lag problem at low engine speed and high back pressure on the exhaust manifold at higher operating conditions. A supercharger which is directly driven by engine crankshaft does not suffer turbo lag but it suffers cost of reduced fuel efficiency when the engine is simply unloaded and idling. Thus the combination of supercharger and turbocharger can be a possible arrangement to overcome the limitations of each other. The experimental investigation conducted by **Rajesh et al. [5]** on a single cylinder four-stroke diesel engine, further modified by supercharged, turbocharged and combination of super-turbocharged engine, shows that use of super-Turbocharging results in an increase of brake power by 48.9%, volumetric efficiency increases by 39.17%, but thermal efficiency can be increased up to 3.15% only, because of 50% loading condition.

Bo et al. [21] presented a turbo expansion concept in a twin-charged engine and investigated the net fuel efficiency benefit from this concept using a super turbo twin charger 1-D simulation model. In this Turbocharging system after the intercooler the turbine like expander (supercharger) expands, instead of compression, the over compressed intake charge to the required plenum pressure and reduces its temperature, whilst recovering some energy through the connection to the crankshaft. The result obtained shows that energy recovered and pumping loss greatly depends upon isentropic efficiency of the supercharger as an expander, which is very low because it was not designed to operate efficiently under expansion. Even, when the isentropic efficiency of supercharger was increased manually from 45% to 90% the combustion phasing was not improved significantly due to the higher boost pressure and larger residual gas fraction (RGF), and the final BSFC was improved by approximately 1%. And as per their result discussion the new turbo expansion concept was not suitable for that specific simulated engine model because the Eaton supercharger, they were using was not designed to operate efficiently under expansion, but this gas exchange model could be used in some other engine for its own interest.

BYEONGIL et al. [22] introduced a variable two stage turbocharger, which will be used mainly for passenger car diesel engines. They developed each component of turbocharger (applicable to 2 liter 4-cylinder diesel engine) to reduce size and improve the controllability over a range of variable two stage turbocharger, which are the main issues for the variable two stage turbocharger. The performance test was conducted at engine speeds between 1000 and 4000 RPM at 250-rpm intervals. The test result shows that at full load torque was increased by 56% and 45% at 1000 and 1250 RPM respectively, as compared to VG turbocharger. Above engine speeds of 200 RPM the difference was small, partly due to the limitations of the engine control unit. However, it might be possible to improve the torque in the high speed range with high speed focused matching.

An experimental study on two stage Turbocharging system named as three phase sequential Turbocharging system with two unequal size turbochargers is presented by **Zhe Zhang et al.[23]**. In this three phase sequential Turbocharging system the two unequal size turbochargers are in parallel. When the engine works at low speeds, only the small turbocharger works and the other two valves related to big turbocharger are closed. When the engine works at medium speed range the two valves related to small turbocharger are closed and only the big turbocharger cuts in. Finally, in the higher engine speed range, all four switch valves related to the small and the big turbocharger are opened and both turbochargers work synchronously. Initially, with the new system the brake specific fuel consumption and smoke emission are measured in the complete engine operating range and compared with the based Turbocharger. The experimental result with the three-phase sequential Turbocharging shows that the engine performances can be improved in the complete engine speed range, especially in the low speed operation. The maximum reduction in the BSFC is about 30g/ (kW-h) and the extreme reduced value of smoke emission is 22%. Finally, the transient performances of three phase sequential Turbocharging system was analyzed by experiments and it is observed that the in the transient acceleration operation, the smoke emission with three phase sequential Turbocharging system is reduced when compared to smoke emission with conventional Turbocharging system.

The use of two or more turbochargers, for improving the performance of an engine over a wide flow range, can be either in parallel or series arrangement. **Q. Zhang et al [24]** investigated the benefits and drawbacks of series and parallel Turbocharging arrangements. According to them these both arrangements have their own advantages and disadvantages. A series Turbocharging arrangement can achieve higher boost pressure than its counterpart parallel arrangement and thus becomes more appropriate for systems requiring more extreme level of boost. A series Turbocharging arrangement also enables the manageable pressure ratios across the individual turbo machinery units, therefore maintains high efficiencies. But the unregulated form of two-stage Turbocharging is not suitable for modern high speed diesel engines because the transient response, fuel consumption and emissions production are also the key factors together with high boost pressure requirement. Another challenge with series arrangement is to make adaptable two stage turbocharger structure for automotive vehicles. On the other hand, in a parallel arrangement the smaller turbocharger improves the transient response, and wider engine flow range is also possible because at higher speeds the big turbocharger phases in and utilizes the high mass flow rate of exhaust gases to improve the performance of the engine. But transition between the different phases of operation is one of the most crucial challenges of parallel Turbocharging.

Now it’s clear that the Two-stage turbo charging along with boost control option is a better option and would be preferred over single stage on account of better fuel economy, better transient performance and reduced emissions. However matching of this two stage turbo charger is very important and is quite complex process, requires lot of skill. And also the problem associated with the trial and error nature of the matching process is also considerable. Further research study is required to develop more trustworthy and easier matching procedure to exploit the benefits of two stage turbocharger to the maximum extent.

To meet the target of the New Energy and Industrial Technology Development Organization (NEDO) project **Junichiro et al. [25]** proposed a sequential three stage Turbocharging system with the combined use of the exhaust after treatment system after the evaluation of the simulation result for the Turbocharging system which makes possible the high boost and high exhaust gas recirculation rate. They reported that the sequential three-stage turbocharging system reduced the fuel consumption by 11.5% compared to base engine for NOx emissions of 1.82g/kWh (engine out) in the final analysis and it resulted into the reduction of NOx emission to 0.14 g/kWh at tailpipe by the combined use of the exhaust after treatment system.

BorgWarner has introduced a new Turbocharging system in **2012 [26]** making its first appearance in the BMW M550d drive Sedan and Touring. The system uses two smaller high-pressure BV45 variable turbine geometry turbochargers integrated with one large B2 low pressure water-cooled turbocharger. Each turbocharger spool up at different engine speed, and therefore more effectively maintain a high volume of air flowing into the engine at all times. Compressed air allowed for a better burn from the fuel, generating more power out of the same amount of fuel. They claim that the system provides almost 25 percent more power and eight percent better fuel economy when compared to the older two-stage system used in cars like the BMW 740d (125.5 horsepower per liter of displacement versus 100.5 horsepower per liter of displacement for the old system).

In a detailed investigation conducted by **Hountalas et al. [27],** an engine simulation model was used to estimate the potential of energy recovery from exhaust of a heavy duty diesel engine by using both mechanical and electrical Turbocompounding. The results obtained shows that Mechanical Turbocompounding can offer a max. BSFC reduction of 0.5%-4.5% and NOx reduction in the range of 12-17% as load increases from 25% to 100% for a power turbine pressure ratio in the range of 1.7 to 1.9. In electrical Turbocompounding maximum BSFC reduction, of 0.2- 2.0% with conventional turbocharger and 3.3-6.5% using highly efficient turbocharger from 25-100% load and 1 bar exhaust pressure increase was experienced. The electrical Turbocompounding can offer the maximum 7% NOx reduction which is independent of load. The main problem related to mechanical and electrical Turbocompounding is the relatively low improvement at low and part engine load.

Ivan et al [28] analyzed the benefits achievable by the electric turbo compound using a comprehensive powertrain model. The results show that, by doing the suitable management of electric turbo compound (ETC) operation, significant improvement of fuel economy and CO2 reduction can be achieved. The reduction of fuel consumption and CO2 emissions has been estimated, reaching a fuel saving up to 4.6 % for the new European driving cycle (NEDC).

Marcelo Algrain from caterpillar Inc. [8] described the control system developments for an electric turbocompound system on heavy-duty diesel engines. The complete engine turbocompound system simulation consists of a turbocharger with an electric generator integrated into the turbocharger shaft, and an electric motor integrated into the engine crankshaft. In this system generator converts the excess mechanical power produced by the turbine of the turbocharger into electricity and this electricity is used to assist the engine by powering the electric motor mounted on the engine crankshaft. In the case when power produced by turbine of the turbocharger is not sufficient to meet the power requirement of the compressor then the generator mounted on the turbocharger shaft can be used as a electrical motor to assist the turbine and power required to operate it as a motor can be generated by electrical machine mounted on the crankshaft. Simulation results indicate that at the rated power, the fuel consumption of a Class-8 on-highway truck engine would be reduced by almost 10% and, at 1500 rpm, loads fluctuation between 25% and 50%, overall reduction in fuel consumption is around 5%.

An analytical study of two-stage Turbocharging was performed by **J.Galindo et al. [29].** They developed an analytical model giving the relationship between global compression ratio and global expansion ratio as a function of basic parameters related to engine and turbocharger. Basically, it was a parametric study by developing an analytical model. The impact of different variables namely, expansion ratio between high pressure and low pressure turbine, intercooler efficiency, turbochargers efficiency, cooling fluid temperature and exhaust temperature were studied. At the end they performed a simulation study on the derived analytical model to analyze the impact of different variables on brake thermal efficiency and pumping mean effective pressure. The results show that the influence of cooling fluid temperature is stronger than the intercooler efficiency on pumping mean effective pressure. When the cooling fluid temperature varied from 298 K to 358 K, the pmep (pumping mean effective pressure) varies from -0.6 bar to 0.3 bar, while the decrease of intercooler efficiency from 90% to 40% at 298 K results into small variation of pmep from -0.6 bar to -0.3 bar. The influence of turbocharger efficiencies and exhaust temperature on brake thermal efficiency is considerable and it is noted that the brake thermal efficiency decreases from 41% to 38% corresponding to varying compressor efficiencies from 70% to 50% or exhaust temperature from 1000K to 700K.

Lu et al. [30] performed a simulation study to evaluate the effectiveness of Turbocompounding in a 2.0 litres Gasoline engine. In their paper they discussed that the turbocharger being used for passenger cars have very high efficiency, therefore a small portion of exhaust energy is needed for compression of intake air in order to get require performance. So, a turbo compound arrangement may be advantageous for further utilization of exhaust energy if the extra produced power is sufficient to offset the pumping work. They built the system model in GT-power, a one dimensional simulation code. The results show that the variable driven supercharger assistance improved the output torque of engine system by up to 24% at lower engine speeds and the fuel efficiency also increased by up to 8%.

Yi et al. [31] presented a theoretical study on Miller-cycle regulatable, two-stage Turbocharging system design for marine diesel engines. The study was focused mainly to reduce the NOx emissions in order to comply with the stringent emission regulations of the International Marine Organization. They developed a multizone combustion model for the optimization of valve timings. HP and LP turbochargers were selected by an iterative process at 75% load point. The results for a highly boosted high speed marine diesel engine show that NOx emission can be reduced by 30% and the brake specific fuel consumption decreased by 6.7% using the Miller-cycle regulatable, two-stage Turbocharging system.

2.1 RESEARCH GAP

On the basis of detailed review of the literatures we can say that, in Turbocharging technology, introduction of Variable Geometry Turbine (VGT), two-stage turbochargers, three stage turbochargers, Turbocompounding and controlled Turbocharging are the successive stages of improvement in order to recover maximum possible amount of waste exhaust energy. But still there are some issues which require special attention in further research studies.

1. At low load and idling speed mechanical Turbocompounding system are an energy consumer. So, the mechanical Turbocompounding even though considered as standard technology deserves further research to examine the potential of exhaust energy recovery. And the focus of the study should be to improve the performance of mechanical Turbocompounding system at low loads and idling speed

2. The variable geometry (VG) turbocharger is a crucial application that minimizes the turbo lag problem at low load and idling conditions. It improves drivability and fuel consumption together with cope up the stringent emission regulations. But the complexity and then cost incurred by the solution is very high and may not be justified for the small and medium passenger cars. So, this is a big challenge for researchers and automotive manufacturers to bring down the cost and complexity of the turbocharged engines.

3. The Turbocompounding technology still requires some means to reduce engine back pressure, because increase in engine backpressure reduces net engine power

4. Since, a lot of research work has been performed for Turbocharging of automotive vehicles but for stationary diesel engines it is rarely found. Due to easy and steady operating conditions for stationary engines it seems that the Turbocharging of stationary engines have a great potential of energy saving. So, performance and emission study of Turbocharging on small stationary diesel engines require further research work.

5. Turbo-electric generator system is also a technology available for energy recovery, among many others, from I.C Engines. So, in order to find the appropriate one a comparative study of performance and emission characteristics should also be carried out for Turbocharging system and Turbo-electric generator system on small capacity engines.

2.2 OBJECTIVE

Following were the major objectives of the current research work:

- To select an appropriate turbocharger for the specified engine existing in the lab.
- To modify the existing experimental set-up for installing the turbocharger on the existing engine without major modifications in order to make it economically viable.
- To study the performance and emissions generated by the modified engine and compare them with the unmodified one.

CHAPTER-3

EXPERIMENTAL SET-UP & PROCEEDURES

3.1 DIESEL ENGINE USED FOR TURBOCHARGING

Kirloskar make single cylinder water cooled engine (AV-1) as shown in figure 11 is used for the present experimental investigation. The technical specification of the engine are given in Table 3.1

Table 1 Specification of Kirloskar AV1 Diesel Engine.

No. of cylinders	One
Bore x Stroke	80mm x 110mm
Cubic Capacity	0.553 L
Compression Ratio	16.5 : 1
Rated output as per BS5514/ISO 3046/IS 10001	3.7kW(5.0 HP) at 1500 RPM
SFC at rated output	245g/kWh
Lubricating oil consumption	1.0% of SFC maximum
Lubricating oil sump capacity	3.3 litre
Fuel tank capacity	6.5 litre
Fuel tank re-filling time period	Every 6 hours engine running at rated output
Engine weight (dry) w/o flywheel	114kg
Weight of flywheel	33 kg (Standard)
Rotation while looking at the flywheel	Clockwise, Optional- Anticlockwise
Power take -off	Fly wheel end. Optional-gear end half or full speed
Starting	Hand start with cranking handle

Figure 11 Kirloskar diesel engine coupled with electrical dynamometer

3.2 DYNAMOMETER

Power-star make electric dynamometer is coupled to the engine for measuring torque. It consist an alternator to which electric bulbs are connected to load the engine. The specifications of the alternator used are given in table 2.

Table 2 Specifications of electric dynamometer.

Technical specification	**Description**
Capacity	3.5 kVA @ 230V, 14 A
Frequency	50C/s
Rating	Continuous
RPM	1500
Phase/ Poles	Single/ 4

3.3 TURBOCHARGER SELECTION

Selection of an appropriate turbocharger for the given specification of the engine is a primary requirement. The matching process among turbine, compressor and I.C.Engine is a very difficult process since I.C.Engine working condition often changes. In order to decide an appropriate turbocharger for a diesel engine, the very first thing that needs to be established is the power target and then calculate the amount of air flow required to produce that power, since the air flow is proportional to the engine power and the turbocharger compressors are sized by how much air they can deliver. After calculating the required air flow rate for the target power we can calculate the compressor discharge pressure and then pressure ratio (ratio of compressor discharge pressure to the compressor inlet pressure i.e. atmospheric pressure). And if we have the pressure ratio and air flow rate then by using compressor map we can select an appropriate turbocharger for the specified engine.

In this experiment we used a very small capacity engine (displacement volume 553 cc, power rating 5 HP) whereas, the turbochargers available in the market are made for the automotive vehicles having engine displacement volume more than 900 cc. So, the turbochargers available in the market were not matching to our specified engine. Finally we decided to use the smallest turbocharger available which is being used for Chevrolet Beat (936 cc) passenger car. This turbocharger was purchased from Turbo Energy Limited (TEL). Item specification is TEL **Turbocharger** 1.0 Lit Xsde, Part No: 3133 902 002 and Serial No: 143388016.

Figure 12 Turbocharger with Waste gate valve

3.4 DEVELOPMENT OF EXPERIMENTAL SETUP

Since the turbocharger accessories available in the market were useful only for the installation of the turbocharger on automotive vehicles. We were connecting the turbocharger to the exhaust line of the stationary diesel engine so; we required few parts to be fabricated separately and then prepare a complete experimental set-up. The items which are fabricated and procured separately except turbocharger are:

1. A Conical shape connector to connect the exhaust outlet of the turbocharger to the piping arrangement throwing exhaust to the environment.
2. V- Band clamp for clamping of conical shape connector to the turbocharger exhaust outlet.
3. A flange coupling, to connect the exhaust outlet of the engine to the turbine inlet of the turbocharger.
4. Clips, pipes and hoses etc. for making flexible connections.

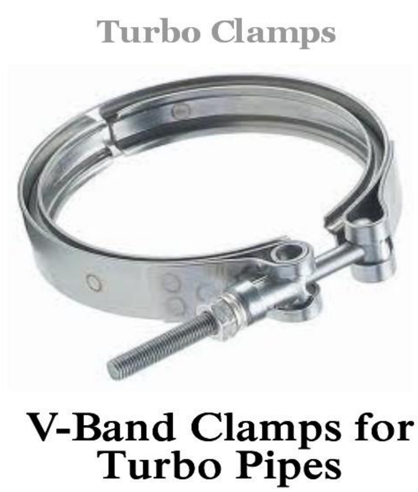


Figure 13 Flange coupling (left) and V- Band clamp(Right)

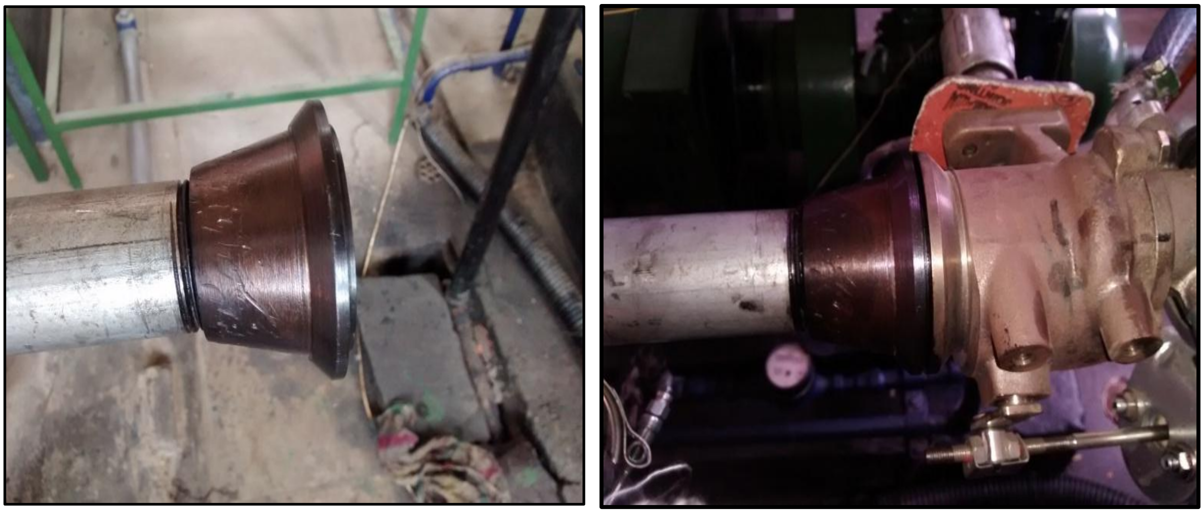


Figure 14 Conical shape connector separate (left) and connected with turbocharger exhaust outlet (right).

3.5 TURBOCHARGER LUBRICATION

A properly installed turbocharger will include a properly installed oil lubrication System. In most conventional turbochargers the bearings are located between the hot and cold sides of the turbo. These bearings are subjected to much greater heat transfer from the exhaust gases, so a constant flow of oil is needed. This flow lubricates the bearing and contributes to cooling. In Chevrolet Beat car turbocharger lubrication is provided by connecting the cooling engine oil line to the turbocharger lubrication system, and oil pressure around 3 to 4 bar is maintained throughout. But we were using the turbocharger for stationary diesel engine and in first attempt we tried to provide separate oil lubrication by using an oil pump. The oil pump arrangement could not work more than 1 hour because of excess heating of lubricating oil and in turn oil pump. The arrangement is shown in figure 15.

Then we made an arrangement of turbo lubrication by connecting the cooling engine oil line to the turbocharger lubrication system. During initial runs it was found that the turbo leaked a lot of oil past its bearings into the turbine housing, resulting in a lot of smoke. This problem was diagnosed as being due to too much back pressure causing oil to leak past the hydrostatic sleeve bearings. So we changed the dimensions of pipes used and then the problem was completely eliminated. Because the small dimension pipes used before were not able to withstand the oil pressure and flow rate and resulting into backpressure in oil supply line. The arrangement is shown in figure 16. Another probable solution was to add a scavenging pump to the oil return line.

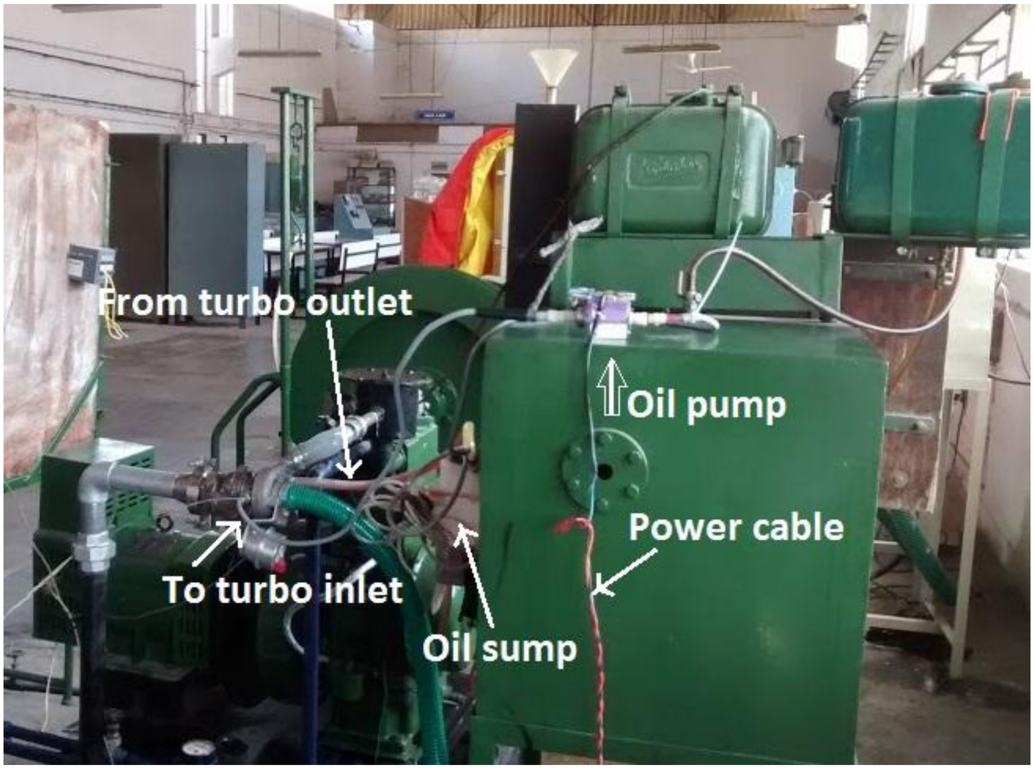


Figure 15 Turbocharger lubrication by using separate oil pump.

Figure 16 Experimental setup with turbocharger.

3.6 FUEL FLOW MEASUREMENT

To measure the volumetric flow rate of fuel (diesel) burette method was used. In this method a glass burette was connected to the fuel tank and engine with help of a Tee valve as shown in Figure 17. The time taken by engine to consume a fix volume (15 ml) of the fuel was measured with the help of a stop watch. There are three position of Tee valve.

1) Horizontal- In this valve position the fuel flows directly from the fuel tank to the engine.
2) Vertically downward- In this the fuel flows from the fuel tank to the burette.
3) Vertically upward- In this the fuel flows from the burette to the engine.

The fuel return line is connected with another burette to make the air out of the fuel line. So that time taken by engine to consume a fixed volume of the fuel was measured with the help of stopwatch. The burette also helps to let us know the fuel level in the fuel tank. The Pictorial view of fuel measurement with burette method is shown in figure 17.

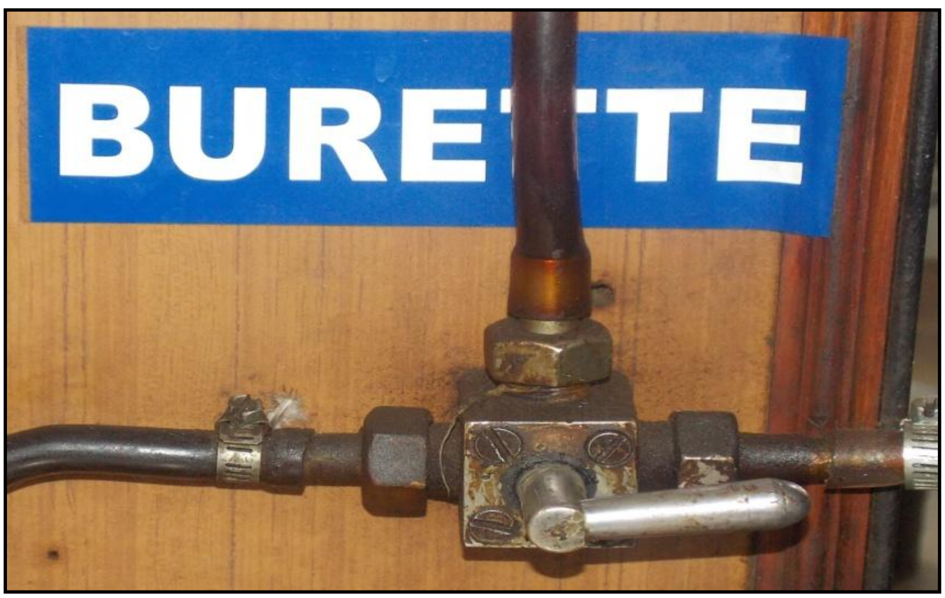


Figure 17 Fuel measuring system (Burette method)

Formula for measuring the mass flow rate of fuel is

Mass flow rate of fuel = Volume flow rate * Density of fuel

3.7 AIR FLOW MEASUREMENT

An air box (16x60x120 cm^3) was used to measure the air flow to engine (figure 18). An orifice of diameter 20 mm and coefficient of discharge (C_d) 0.6 was fitted on one of the side walls. The outlet was connected to the engine through an air filter. Pressure inside the air box remained less than the atmospheric pressure during operation and was measured with the help of manometer mounted on one of the side.

Since the engine was constant rpm engine hence the air flow rate was constant. The amount of air induced per second or volume flow rate of air (Va) was obtained with the help of the following relation

$$V_a = C_d A_{orifice}\sqrt{2g\ h_w \rho_w/\rho_a}\ \text{m}^3/\text{s}\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots 1$$

Where,

C_d= coefficient of discharge

$A_{orifice}$ = area of orifice

h_w= manometer reading of water column (meters)

ρ_w= density of water (1000 kg/cubic meter)

ρ_a= density of air (1.17 kg/cubic meter)

Mass flow rate of air m_a= (volumetric flow rate of air)*(density of air)……………………….2

Figure 18 Air flow measuring system

3.8 TEMPERATURE MEASUREMENT

Various K type thermocouples are placed at different places to measure the temperature of exhaust gases, cooling water inlet and outlet to run the engine under control condition. The K type thermocouple is inexpensive, and a wide variety of probes are available in its −200 °C to +1350 °C range.

3.9 EXHAUST EMISSIONS MEASUREMENT

The emissions were measured at different load values with and without turbocharger. Table 3 gives the emission constituent and their measuring unit.

Table 3 Component in the exhaust emission.

S. No.	Constituent	Unit
1.	Carbon mono-oxide (CO)	% volume
2.	Carbon dioxide (CO_2)	% volume
3.	Oxygen (O_2)	% volume
4.	Unburned hydro-carbon (HC)	PPM
5.	Nitrogen oxides (NOx)	PPM
6.	Smoke	% opacity

The first five gas were measured by the AVL DIGAS 4000 LIGHT. The AVL 437 SMOKE METER was used to measure the smoke opacity in the exhaust gases. The AVL DIGAS 4000 LIGHT and AVL smoke meter is shown in the figure 19.

Figure 19 AVL DIGAS 4000 LIGHT (left) & AVL 437 SMOKEMETER (Right)

Table 4 Technical specifications of AVL smoke meter.

Measuring value output	Opacity N [%] or absorption k [m^{-1}]
Measuring range	N=0…100% or k=0….10m^{-1}
Resolution of displayed values	0.01% opacity or 0.0025 m^{-1}
Limit of detection	0.1% opacity
Zero stability	{0.1% or 0.0025m^{-1}}/30 min (drift with zero gas)
Rise time	0,1s
Data transmission speed	50Hz with analogue output 2….10Hz with RS 232 serial interfaces
Ambient temperature	+5^{0}C…..+50^{0}C
Exhaust gas temperature	0….600^{0}C (800^{0}C with high pressure option)
Exhaust gas pressure	-100mbar……+400mbar (Including pulsation peaks) 0 mbar….. +3000 mbar with high pressure option (including pulsation peaks)
Dimensions of basic appliance	680mm x 440mm x 460mm (WxHxD)
Weight of basic appliance	Approx. 47 Kg

3.10 EXPERIMENTAL TECHNIQUE

The performance and emission study of a stationary diesel engine with and without turbocharger have been carried out. The observations have been taken for different load conditions ranging from 0.5 kW to 3.7 kW. During the experiment different parameters such as cooling water inlet and outlet temperature, exhaust temperature, fuel consumption and emission constituents were recorded for both, with and without turbocharger arrangement.

3.11 VARIATION OF LOAD

The Kirloskar AV1 engine has a rated power output (full load) of 3.7 kW. The different loads applied to the engine for both with and without turbocharger condition are: 0.5kW, 1kW, 1.5kW, 2kW, 2.5kW, 3kW, 3.5kW, and 3.7kW. Switching on bulbs of appropriate wattage increases the load.

3.12 FORMULA USED FOR PERFORMANCE PARAMETERS CALCULATION

3.12.1 BRAKE THERMAL EFFICIENCY

$$\boldsymbol{Brake\ thermal\ efficiency(\eta) = \frac{Brake\ power(BP)}{Indicated\ power(IP)}}$$

IP = m_f x C.V.

BP= in Kw available at the engine output shaft

m_f = mass of fuel (kg/h)

C.V. = Calorific Value of diesel fuel = 42.5 MJ/kg

Density of diesel fuel = 850 kg/ m^3

3.12.2 BRAKE SPECIFIC FUEL CONSUMPTION

$$BSFC = \frac{mf}{BP}$$

3.12.3 VOLUMETRIC EFFICIENCY

$$\boldsymbol{\eta_v = \frac{V_a}{\frac{N}{2} \times V_s} \times 100}$$

V_a = volume flow rate of air m^3/ min

N = Engine RPM

V_s = displacement volume in m^3

CHAPTER-4

RESULTS & DISCUSSION

As we have discussed earlier exhaust gas emissions and engine performance with and without turbocharger have been tested at varying loads. Graphical representation of different performance parameters and emissions characteristics with respect to load have been shown in this section. Engine performance parameters which have been discussed are; air flow rate, volumetric efficiency, brake specific fuel consumption (BSFC), and brake thermal efficiency (BTE). Gaseous emissions which have been observed and discussed are carbon monoxide (CO), carbon dioxide (CO_2), Oxygen (O_2), hydro carbon (HC), oxides of nitrogen (NOx), and smoke.

4.1 ENGINE PERFORMANCE PARAMETERS

4.1.1 VOLUMETRIC EFFICIENCY

Since the diesel engine used for experiment is constant RPM engine, so the air supplied (i.e. air flow rate) to the engine and in turn volumetric efficiency must be constant irrespective of load. But in case of turbocharged engine the mass flow rate of air increases with the speed of turbine. So, during the observation of the engine with turbocharger, as the engine load increases the mass flow rate of air supplied to the engine increases and volumetric efficiency also increases. Variation in the mass flow rate of air is measured with the help of an air box and then volumetric efficiency calculated at different loads. Observation shows that the volumetric efficiency of the engine increased by up to 32%. A graphical representation of volumetric efficiency and air flow rate with respect to varying engine load is shown in figures 20 and 21.

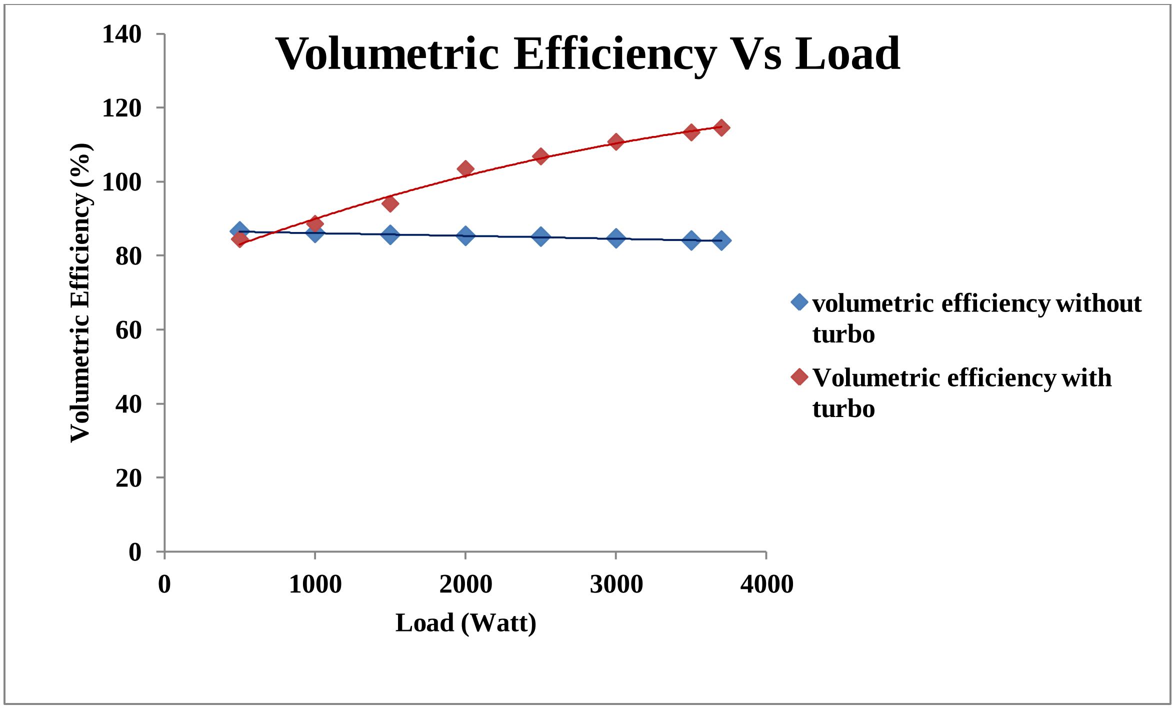


Figure 20 Effect of load on volumetric efficiency of the engine

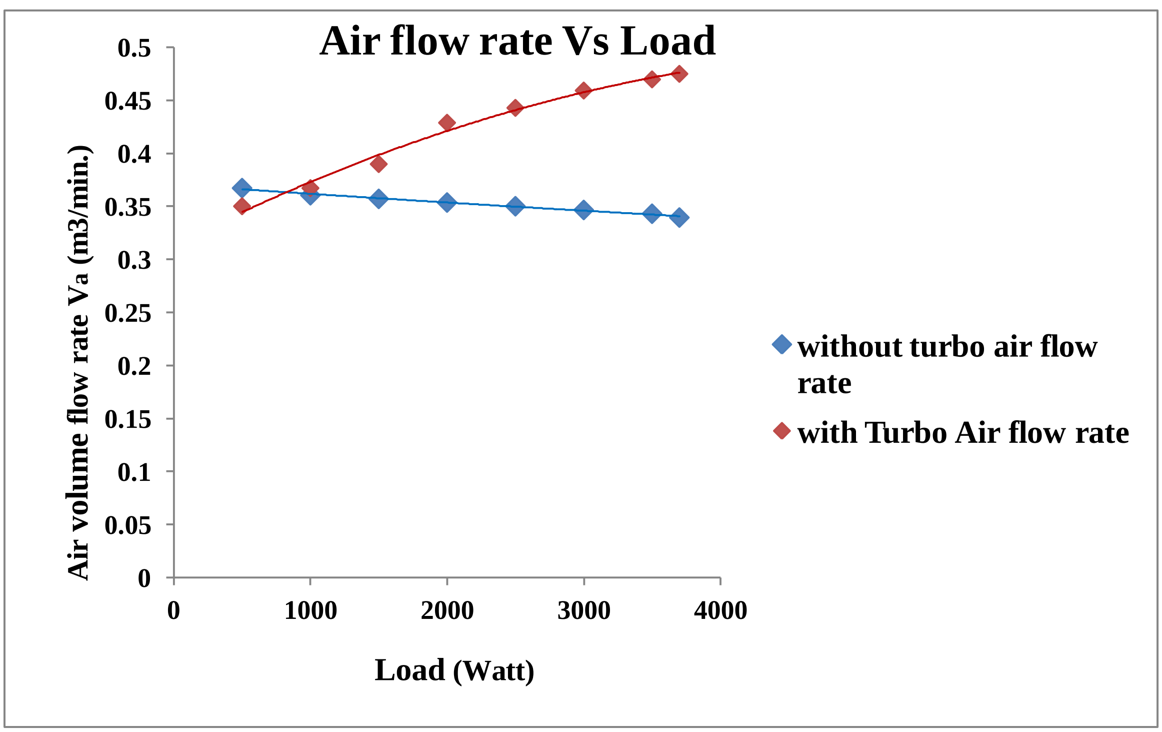


Figure 21 Effect of load on air flow rate.

4.1.2 BRAKE SPECIFIC FUEL CONSUMPTION

BSFC of the engine during Turbocharging have been measured. And result shows that a considerable decrease in BSFC of turbocharged engine compared to that of naturally aspirated is observed after 2kW loading and reduced by up to 8% at 3kW. A graphical representation of brake specific fuel consumption with respect to varying engine load is shown in figure 22.

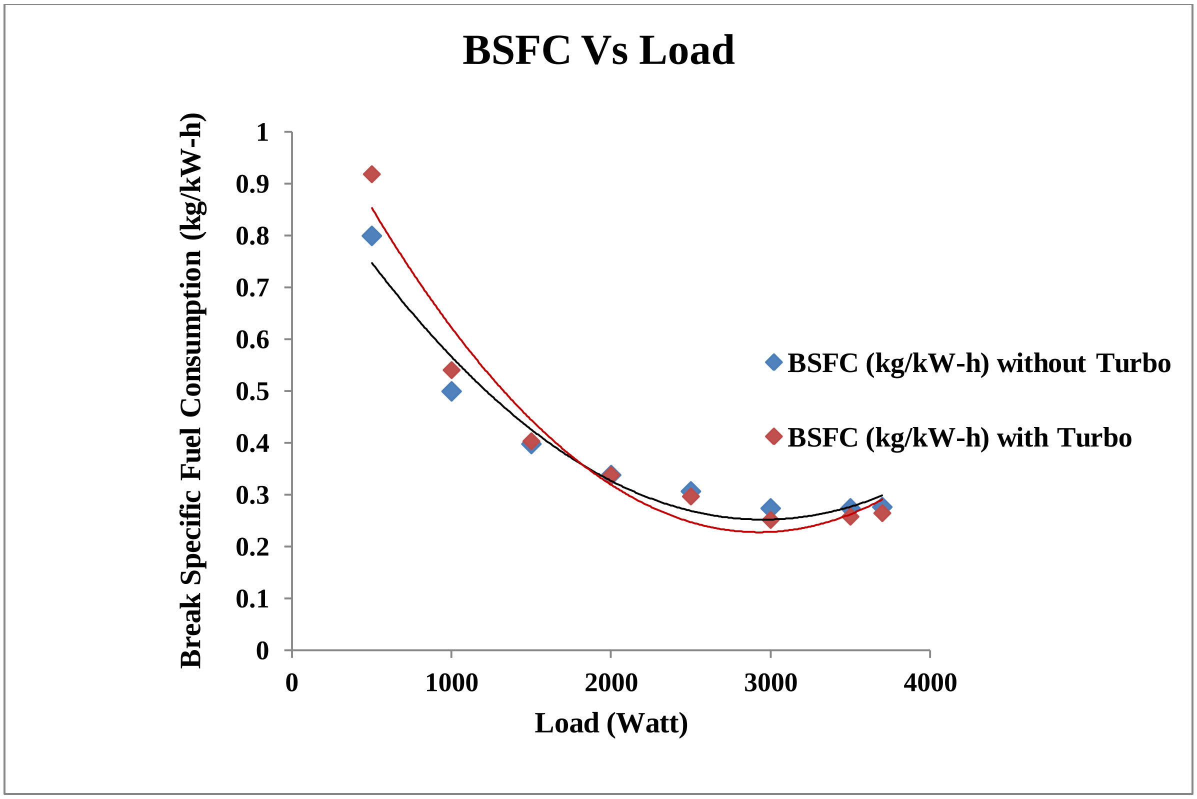


Figure 22 Effect of load on BSFC of the engine.

4.1.3 BRAKE THERMAL EFFICIENCY

Brake Thermal Efficiency (BTE) is a measure of net power developed by the engine which is readily available for use at the engine output shaft. It is observed that the BTE of the turbocharged engine compared to the naturally aspirated engine is increased by up to 8.5% at 3kW loading condition. The increase in BTE is significant when the load applied on the engine is above 2kW. Figure 23, shows the variation of thermal efficiency as a function of load for both with and without turbocharger condition.

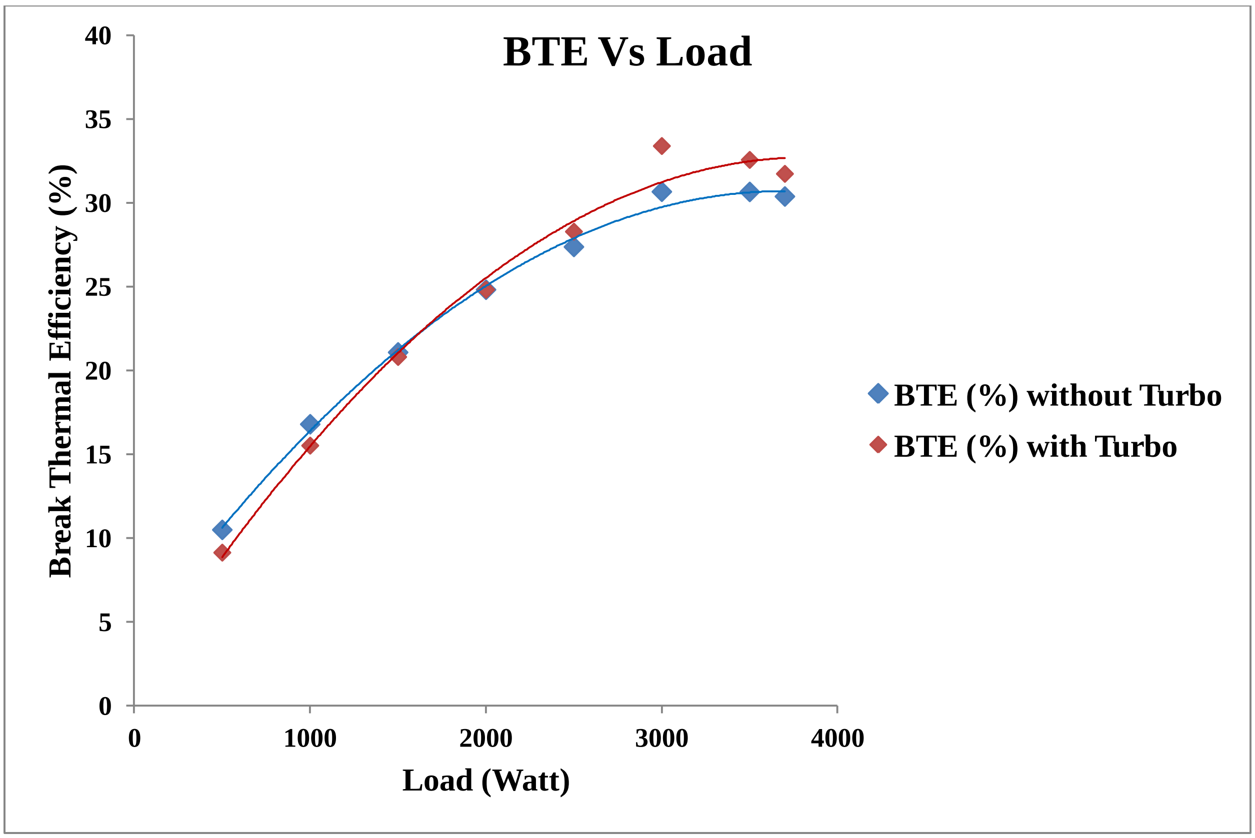


Figure 23 Effect of load on Brake Thermal Efficiency.

4.2 ENGINE EXHAUST EMISSIONS

4.2.1 OXIDES OF NITROGEN

Nitrogen and Oxygen reacts at relatively high temperature. NOx formation in an engine is a function of reaction temperature, available Oxygen and duration of availability. Actually, diatomic Nitrogen molecule at high temperature decomposes into highly reactive mono atomic Nitrogen molecule. NOx formation increases with higher combustion temperature, longer high-temperature combustion periods and greater availability of Oxygen. Variation of exhaust gas temperature and NOx emission with respect to load is shown in Figures 24 and 25.

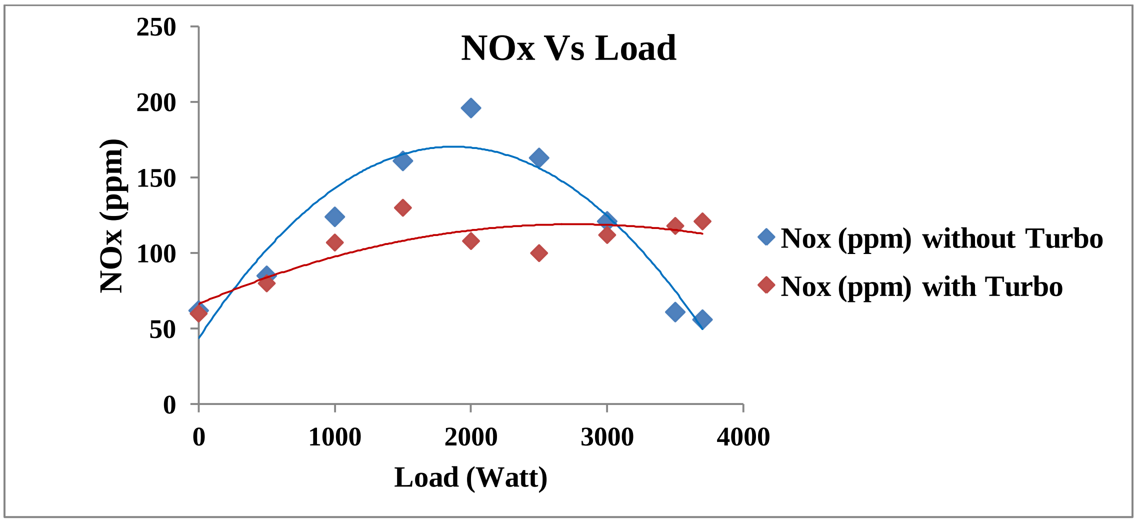


Figure 24 Effect of load on NOx emission

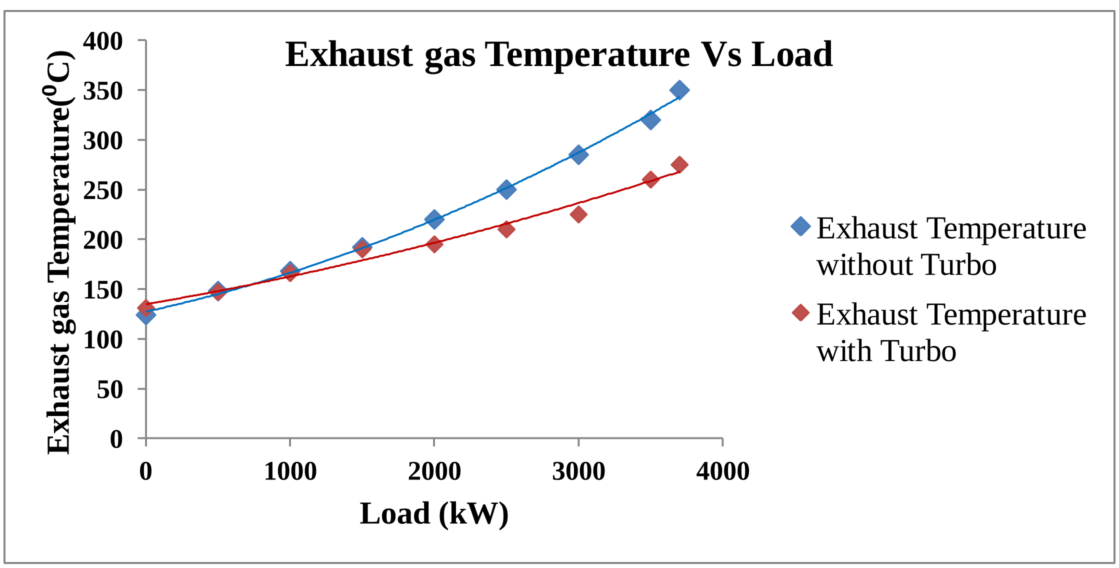


Figure 25 Effect of Load on Exhaust gas temperature.

4.2.2 CARBON MONOXIDE (CO) AND CARBON DIOXIDE (CO2) EMISSIONS

Carbon content of fuel oxidized with Oxygen available in the air to CO and then to CO_2. Carbon which is not converted to CO_2 will come out as CO in exhaust. Under low load and idling conditions due to low operating temperatures and improper mixing of fuel, incomplete combustion of fuel takes place giving high CO emissions. With the increase in the load both the operating temperature and mixing of fuel increases, resulting into the reduction of CO emissions. Again at higher loads due to low air-fuel ratios, the available Oxygen for combustion becomes less, results in increased CO emissions despite the favorable conditions of temperature and air fuel mixing. In case of turbocharged engine Oxygen available is always greater than that of naturally aspirated engine due to excess air temperature inside combustion chamber, which brings down the combustion chamber temperature. Therefore, CO emission with Turbocharging is greater compared to the naturally aspirated engine. The amount of CO_2 is also greater for turbocharged engine. For good combustion, all the carbon molecules in the fuel must be converted into the carbon dioxide but in real situation all the carbon atoms in the fuel cannot be converted to CO_2. In case of turbocharged engine the amount of air available is in highly excess amount. So, the favorable conditions are available for combustion of the fuel and therefore CO_2 percentage increases in case of Turbocharging. The graphical representation of CO and CO_2 Vs Load is shown in figure 26 and figure 27 respectively.

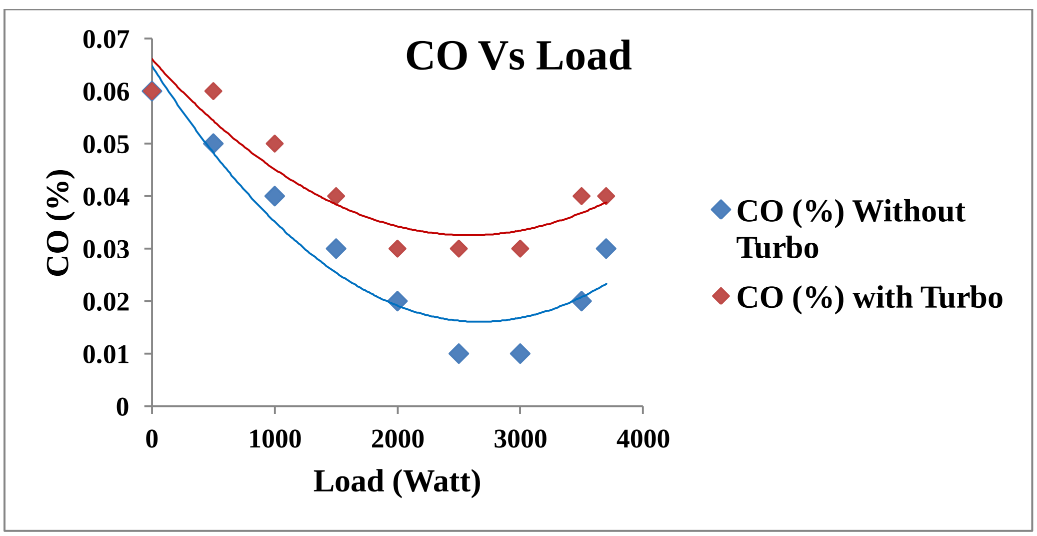


Figure 26 Effect of load on carbon monoxide emission.

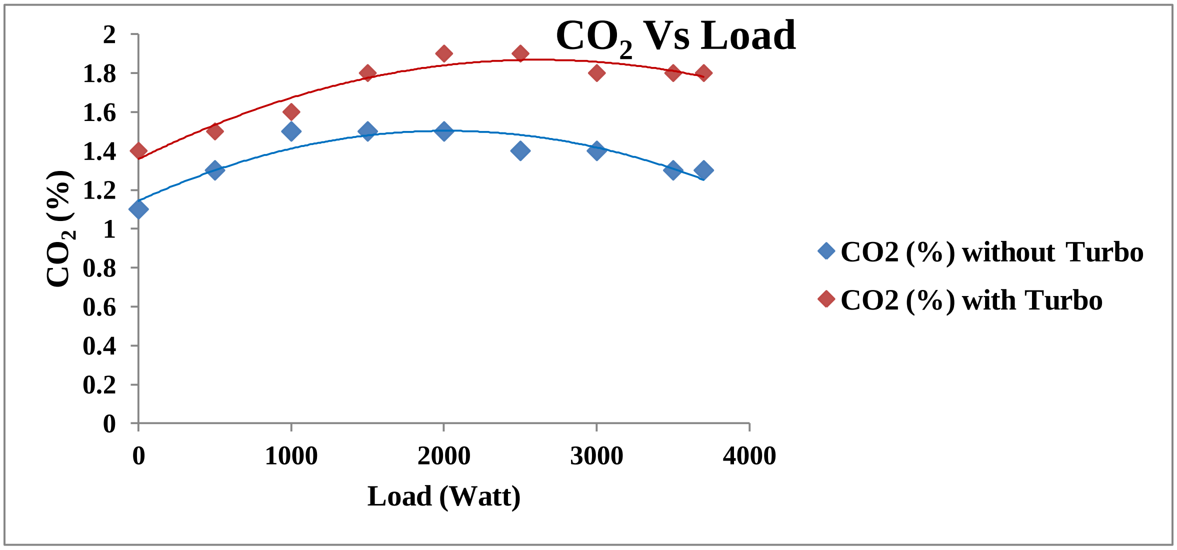


Figure 27 Effect of Load on Carbon dioxide emission.

4.2.3 OXYGEN (O2) EMISSION

Figure 28, shows the Oxygen percentage variation with load. During the Turbocharging it is always less than the Oxygen percentage in naturally aspirated engine. This might be due to favorable conditions available for the reaction of oxygen with the fuel and therefore conversion into CO and CO_2.

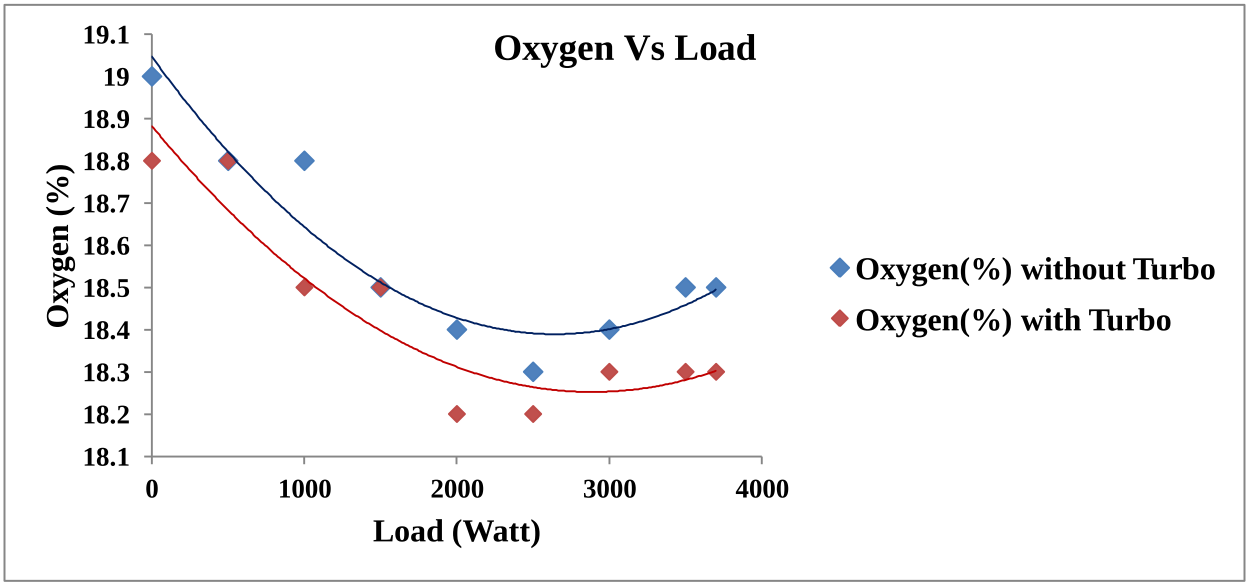


Figure 28 Effect of load on Oxygen emission.

4.2.4 HYDROCARBON (HC) EMISSION

Unburned hydrocarbon emissions are the direct result of incomplete combustion. The experimental observation shows that hydrocarbon emission increases at low load conditions and then at the moderate load condition it decreases slightly and again rises at higher load conditions. This might be due to the improper mixing of fuel which resulted in incomplete combustion of fuel giving high hydrocarbon emissions. In case of Turbocharging HC is higher than that during naturally aspirated at low load condition but it decreases as the load increases and it shows that due to excess availability of air at higher load end proper combustion occurs inside the engine cylinder. Figure 29, shows the effect of loading conditions on hydrocarbon generation.

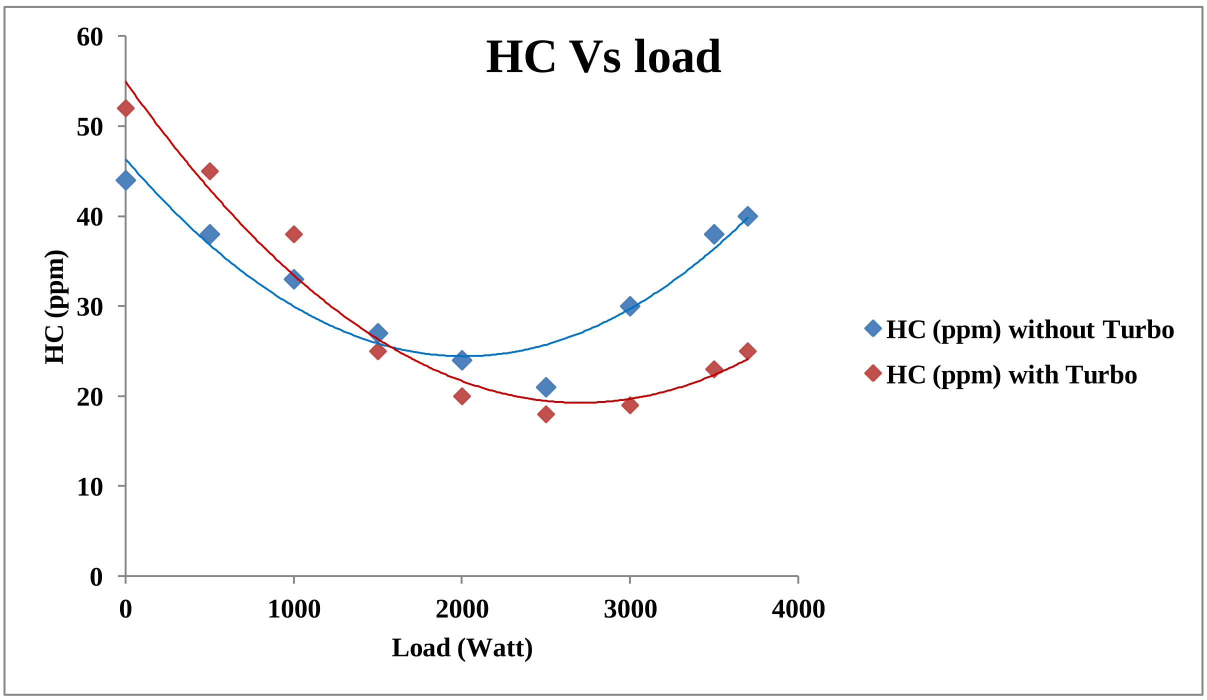


Figure 29 Effect of load on Hydrocarbon emission.

4.2.5 SMOKE

Smoke emission occurs mostly during acceleration, overloading or even during full load operation of the engine. Under these conditions more fuel is burned and the prevailing temperatures inside the combustion chamber become very high. Because of this high temperature thermal cracking of molecules rather than normal oxidation takes place. Due to this thermal cracking the carbon/soot appears in the form graphite structure of jet black color and is called smoke. During the observation it was found that the smoke percentage of turbocharged engine compared to naturally aspirated engine increased shown in Figure 30. And the reason behind this may be due to leakage of lubricating oil in the turbocharger's turbine housing.

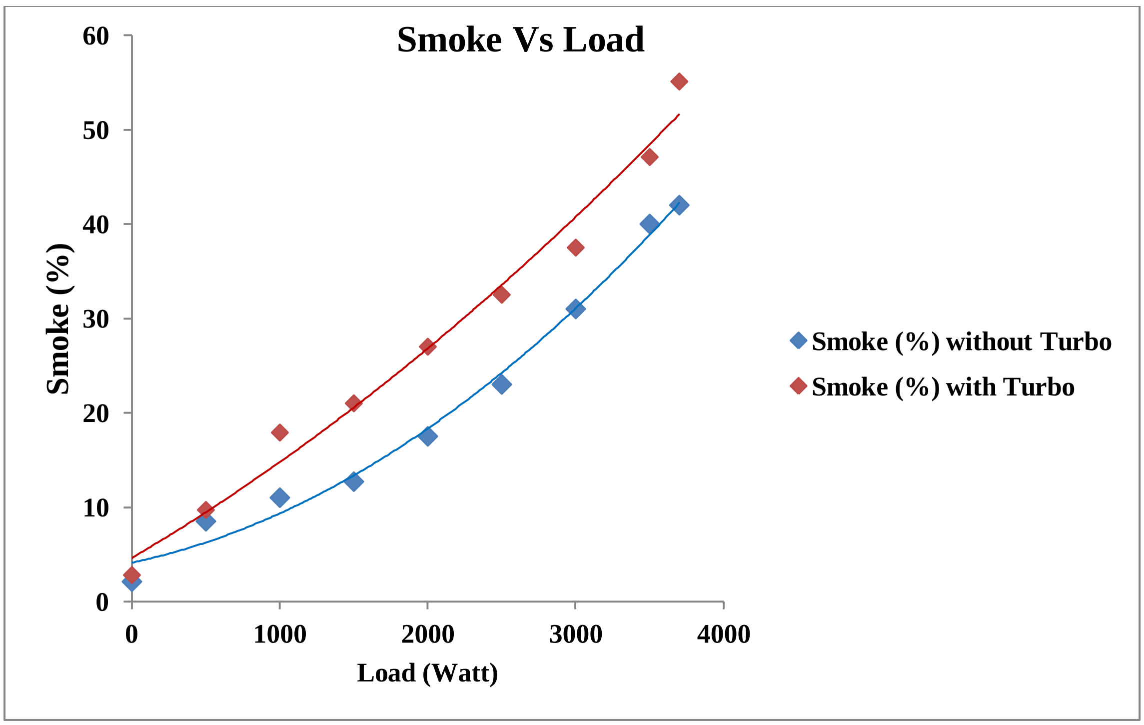


Figure 30 Effect of load on smoke emission.

CHAPTER-5

CONCLUSION

From Comparative study of performance and emission analysis of a stationary C.I engine with turbocharger and without turbocharger following conclusions have been derived.

- Volumetric efficiency of the turbocharged engine increased by up to 32% compared to naturally aspirated engine. Above 0.5 kW loading, turbocharger has shown positive effect and improved the breathing ability of the engine by increasing the mass flow rate of air supplied to the engine compared to naturally aspirated engine at the same RPM (1500).
- Brake thermal efficiency of the engine after Turbocharging also increased by up to 8.5% compared to the naturally aspirated case. After 2 kW loading the engine's BTE increased significantly due to higher exhaust energy available to rotate the turbocharger turbine and facilitating the compressor to discharge more air to the engine inlet manifold and increase the brake mean effective pressure on the piston. At low load conditions Turbocharging of the engine worked as energy consumer and shows negative effect on BTE up to 2 kW load.
- A considerable decrease in BSFC also observed above 2 kW loading of the engine and reached maximum value of around 8% at 3kW. After 1.5 kW load condition air mass flow rate of the engine increased significantly due to higher exhaust energy available to the turbine blades, which resulted into reduction in fuel consumption compared to that naturally aspirated engine.
- Exhaust gas emissions observation shows that temperature of exhaust gas in turbocharged condition is lower than the exhaust gas temperature of without turbocharged engine throughout the operation which resulted into the reduction of NOx emissions significantly. Reduction in the exhaust temperature of the turbocharged engine may be due to excess air supply and backpressure generated in the exhaust manifold which prevents the proper scavenging of the exhaust from combustion chamber. Since the turbocharger used is not properly matched with the engine so the chance of backpressure generation exists. Although the turbocharger used is waste gated turbocharger but during the experiment waste gate valve was completely closed and exhaust gases are not allowed to bypass the turbine blades of the turbocharger

- Hydrocarbon (HC) emission is also reduced significantly for turbocharged engine during moderate and higher load operating conditions. At low load condition HC is higher and it may be due to improper mixing of the air fuel mixture due to unavailability of sufficient amount of air, which resulted into unburned hydrocarbon.
- The emission percentage of both Carbon monoxide (CO) and carbon dioxide (CO_2) increased throughout the engine operation but the Oxygen percentage is reduced, in case of turbocharged engine. Since during the Turbocharging the mass of air available inside the combustion chamber is significantly higher than the naturally aspirated condition, so the oxygen available for reaction with the carbon atom of fuel is in excess which generated more CO_2 but the temperature inside the combustion chamber is also lower and the chances of incomplete combustion also exists which results into higher CO percentage. Now a significant portion of Oxygen available in the air reacted and converted into CO and CO_2 so, the oxygen available in the exhaust of a turbocharged engine is reduced compared to the naturally aspirated engine.
- From this experimental study, it has been identified that there is a large potential of energy saving through the use of Turbocharging as waste energy recovery technology. It would also help to recognize the improvement in performance and emissions of the engine if this technology is adopted by the stationary diesel engine manufacturers. The effort should be made in the design and development of the turbochargers matching with the stationary CI Engines.
- Since the variable geometry turbochargers (VGT) are being popular because of its ability to improve the performance (low speed torque, transient response and lower exhaust manifold pressure at high speed) of engine over a wide range of operation conditions. But due to the requirement of accurate engineering mechanism and high strength materials to make sure the reliability of variable geometry turbocharger the price is relatively high compared to the waste gated turbochargers. However, heavy duty vehicles on which variable geometry turbochargers are frequently being used can absorb the cost impact. Cost reduction is essential for the application of VGTs to the small and medium size passenger cars; therefore it becomes a big challenge for turbocharger manufacturers to bring down the cost and complexity of the turbocharged engines.

REFERENCES

1. J. S. Jadhao, D. G. Thombare, "Review on Exhaust Gas Heat Recovery for I.C. Engine" International Journal of Engineering and Innovative Technology (IJEIT) Volume 2, Issue 12, June 2013.
2. Jianqin Fu, Jingping Liu, Yanping Yang, Chengqin Ren, Guohui Zhu "A new approach for exhaust energy recovery of internal combustion engine: Steam turbo charging" Applied Thermal Engineering 52 (2013) 150e159.
3. M.G. He, X.X. Zhang, K. Zeng, K. Gao, A combined thermodynamic cycle used for waste heat recovery of internal combustion engine, Energy 36 (12) (2011) 6821e6829
4. R. Saidur, M.Rezaei, W.K.Muzammil, M.H.Hassan, S.Paria, M.Hasanuzzaman "Technologies to recover exhaust heat from internal combustion engines" Renewable and Sustainable Energy Reviews 16 (2012) 5649–5659.
5. Rajesh Bisane, Dhananjay katpatal "EXPERIMENTAL INVESTIGATION & ANALYSIS OF AN SINGLE CYLINDER FOUR STROKE TWIN-CHARGE C.I. ENGINE" IJRET: International Journal of Research in Engineering and Technology eISSN: 2319-1163 | pISSN: 2321-7308
6. Dipak Patil, Dr. R. R. Arakerimath "A Review of Thermoelectric Generator for Waste Heat Recovery from Engine Exhaust" INTERNATIONAL JOURNAL OF RESEARCH IN AERONAUTICAL AND MECHANICAL ENGINEERING Vol.1 Issue.8, December 2013. Pgs: 1-9
7. Alias Mohd Noor, Rosnizam Che Puteh and Srithar Rajoo, 'Waste Heat Recovery Technologies in Turbocharged Automotive Engine – A Review', *Journal of Modern Science and Technology Vol.2 No.1 March 2014. Pp.108-119.*
8. Marcelo Algrain, Caterpillar Inc. "Controlling an Electric Turbo Compound System for Exhaust Gas Energy Recovery in a Diesel Engine"
9. Bosch Mahle Turbosystems http://www.bmturbosystems.com/en/produkte/abgasturbolader-fuer-nkw.html
10. V Ganesan IC engines Third Edition, book.
11. Park S, Matsumoto T, Oda N. Numerical analysis of turbocharger response delay mechanism. SAE Paper no. 2010-01-1226, Presented at SAE 2010 world congress & exhibition, April 2010, Detroit, MI, USA, Session: Engine Boosting Systems (Part 1 of 2).

12. Jianqin Fu, Jingping Liu, Yong Wang, Banglin Deng, Yanping Yang, Renhua Feng, Jing Yang, "a comparative study on various turbo charging approaches based on I C engine exhaust energy recovery" Applied Energy 113 (2014) 248-257.
13. Vijay Chauhan "A REVIEW OF RESEARCH IN MECHANICAL ENGINEERING ON RECOVERY OF WASTE HEAT IN INTERNAL COMBUSTION ENGINE" Volume 2, Issue 12 (December 2012).
14. Ma, Z., Yang, D., and Guo, Q. (2012), 'Conceptual Design and Performance Analysis of an Exhaust Gas Waste Heat Recovery System for a 10000TEU Container Ship', *Polish Maritime Research* 2 (73), Vol.19, Pp.31-38.
15. Koyess, R. (2011), *Investigating F1's New Dawn. Engine Technology Turbocompounding*, www.racetechmag.com, April 2011, Pp 54-58.
16. http://www.cranfield.ac.uk/sas/pdf/enginetechturbocompoundingpost.pdf
17. Zhuge, W., Huang, L., Wei, W., Zhang, Y., and He, Y. (2011), "Optimization of an Electric Turbo-Compounding System for Gasoline Engine Exhaust Energy Recovery", *SAE International Paper No:* 2011-01-0377. SAE 2011 World Congress & Exhibition, April 12, 2011, Detroit, Michigan, United States.
18. C. Bibin, P. Seenikannan, N. Kanthavelkumaran "Study On Waste Heat Recovery In An Internal Combustion Engine" International Journal of Engineering Research & Technology (IJERT), Vol. 1 Issue 10, December- 2012.
19. YASUAKI JINNAI, HIROYUKI ARIMIZU, NAOTO TASHIRO. MASAKI TOJO, TAKAO YOKOYAMA, NORIYUKI HAYASHI, "A Variable Geometry (VG) Turbocharger for Passenger Cars to Meet European Union Emission Regulations", *Mitsubishi Heavy Industries Technical Review Vol. 49 No. 2 (June 2012)* 26
20. Hiroshi Uchida, "Trend of Turbocharging Technologies" Special Issue Turbocharging Technologies.
21. Jaehoon Cheong, Sunghwan Cho, Changho Kim, "Effect of Variable Geometry Turbocharger on HSDI Diesel Engine" Seoul 2000 FISITA World Automotive Congress June 12-15, 2000, Seoul, Korea.
22. Hu, B., Copeland, C., Brace, C., Akehurst, S. et al., "A New Turboexpansion Concept in a Twin-Charged Engine System," SAE Technical Paper 2014-01-2596, 2014, doi:10.4271/2014-01-2596.

23. BYEONGIL AN, TAKASHI SHIRAISHI, **“**Development of Variable Two-stage Turbocharger for Passenger Car Diesel Engines” *Mitsubishi Heavy Industries Technical Review Vol. 47 No. 4 (December 2010)*
24. Zhe Zhang and Kangyao Deng, Zhenbiao Wang and Xiangguo Zhu, “Experimental Study on the Threephase Sequential Turbocharging System with Two Unequal Size Turbochargers” 2008-01-1698
25. Q. Zhang, A. Pennycott and C.J. Brace, “A Review of Parallel and Series Turbocharging for the Diesel Engine”
26. Junichiro Nitta, Akihiko Minato and Naoki Shimazaki, “Performance Evaluation of Three Stage Turbocharging System for Heavy-Duty Diesel Engine” 2011-01-0374 Published 04/12/2011
27. http://www.digitaltrends.com/cars/the-power-of-three-borgwarner-builds-worlds-first-three stage-turbosystem-for-diesels/
28. D.T. Hountalas, C.O. Katsanos and V.T. Lamaris, “Recovering Energy from the Diesel Engine Exhaust Using Mechanical and Electrical Turbocompounding” *SAE TECHNICAL PAPER SERIES* 2007-01-1563
29. Arsie, I., Cricchio, A., Pianese, C., De Cesare, M. et al., "A Comprehensive Powertrain Model to Evaluate the Benefits of Electric Turbo Compound (ETC) in Reducing CO2 Emissions from Small Diesel Passenger Cars," SAE Technical Paper 2014-01-1650, 2014, doi:10.4271/2014-01-1650.
30. J. Galindo, J.R. Serrano, H. Climent, O. Varnier “Impact of two-stage Turbocharging architectures on pumping losses of automotive engines based on an analytical model” Energy Conversion and Management 51 (2010) 1958–1969
31. Lu, P., Brace, C., and Hu, B., "Explore and Extend the Effectiveness of Turbocompounding in a 2.0 litres Gasoline Engine," SAE Technical Paper 2015011279, 2015, doi: 10.4271/2015011279.
32. Yi Cui, Zhilong Hu, Kangyao Deng, Qifu Wang “Miller-Cycle Regulatable,Two-Stage Turbocharging System Design for Marine Diesel Engines” Key Laboratory for Power Machinery and Engineering of Ministry of Education, School of Mechanical Engineering, Shanghai Jiaotong University, Shanghai 200240, China.