# Thermodynamic Analysis of Two-Stage Turbocharging for Diesel Engine

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**Abstract - Turbochargers are used in automotive industry to enhance the power output of an internal combustion (IC) engines. The useful applications of such a turbocharger are in the engines for the reduction of engine size and fuel consumption by increasing existing power output. It is because of these reasons turbochargers are now becoming more popular in automobile applications. It is also used where compensation of altitude requires and for boosting of the engine power. Single stage turbocharger is not recovered all the heat of the exhaust therefore it is decided to use two stage turbocharging. Two stage turbocharging is used to recover all the heat of exhaust and to develop the more power from the engine. The present study is based on the improvement of the engine power of the existing diesel engine. This is then compared with naturally aspirated engine and engine with single stage turbocharging. In the present study the operating parameters are discussed and compared. It is observed that Brake mean effective pressure (BMEP) and Brake specific fuel consumption (BSFC) are increased irrespective of the speed of the engine. The rises in volumetric efficiencies are observed in the existing engine and thereby rise in brake thermal efficiencies.**

**Keywords - Engine, Turbocharger, Two stage turbocharging, Volumetric efficiency, Power output, Brake men effective pressure, Brake specific fuel consumption.**

## I. INTRODUCTION

In a naturally aspirated diesel engine, air is drawn in the cylinder during suction stroke into the engine cylinders at atmospheric pressure acting against a partial vacuum that occurs as the piston travels downwards toward bottom dead center during the suction stroke. During the suction stroke friction is produced between the piston and cylinder. This friction reduces the pressure in the cylinder. Also there is pressure drop due to friction in the components like intake manifold, air filter, and throttle etc. It is observed that due to pressure drop overall volumetric efficiency reduces by the amount of 10 to 20% [1]. The overall effect of the entire pressure drop is reduction in air density which reduces the combustion quality of that engine. The net effect is reduction of the power output of the engine. Also the net output is reduced at high altitude. This reduction of air density on the road and high altitude may be compensated by device known as Supercharger. A supercharger is an air compressor used for increasing the pressure and density of air supplied to an internal combustion engine. This Supercharger is connected to crankshaft of the engine [2]. Hence, the power of the engine reduces. Supercharger uses 15% of engine power output [3]. To eliminate the loss of power, later on the compressor was driven by a turbine utilizing the energy of exhaust gases of the engine by passing them through the turbine blades. Then this technology became popular by the name as turbocharging in early 1980s.

A turbocharger operates in much the same way as a centrifugal supercharger, except it is not driven by pulleys and belts attached to the engine's crank. Power required to derive the compressor is derived from engine exhaust rather than from engine as in mechanical supercharger [4]. The study compared the 5.7 liter, V-8, naturally aspirated engine with a downsized 3.8 liter, V-6, turbocharged engine but with similar maximum power. The study summarizes that same power turbocharged engines always have lower fuel consumption than naturally aspirated engines and a 3.8 liter turbocharged engine is more efficient than a 5.7 liter, naturally aspirated at the same power level as it has higher mechanical efficiency [5].The 2.3L turbocharged engine performed better than the 2.3L NA engine with a 36% increase in maximum torque and a 41% increase in maximum power. Although the unmodified 2.1L engine did not perform as well as the turbocharged 2.3L engine it did outperform the naturally aspirated 2.3L engine. This result supports the theory of downsizing engines while retaining power and torque output [6]. A fixed geometry turbine is not capable of supplying enough power to the compressor for the boost pressure required for low speed and during transient conditions. In addition, the flow range of a centrifugal compressor is a limiting factor. With the variable inlet guide vanes, the compressor can operate at higher boost level for lower mass flows without surge [7]. The primary motive for using variable geometry turbines in automotive applications is to reduce turbo-lag. The study shows that to reach full load torque the time required by Variable geometry turbine (VGT) turbocharger was 1.8 sec whereas 1.2 sec for two stage turbocharger because of better transient characteristics of two stage turbocharger [8].

In two stage turbocharging the exhaust gas mass flow coming out from the cylinders first flows into the exhaust gas manifold, from here, either the entire exhaust gas mass flow is expanded through the high pressure turbine (HP) or a part of the mass flow is conducted through the bypass. The intake air mass flow is first pre-compressed through the low pressure stage and, ideally, intercooled.Further compression and charge air cooling takes place in the high pressure stage. As a result of the pre-compression, the relatively small HP compressor operates at a higher pressure level, so that the required air mass flow throughout can be obtained [9].In order to increase the specific power output of engines, engineers are opting for downsizing. The higher operating brake mean effective pressures (BMEP) also means that turbocharged engines have lower pumping losses at low load [10]. Engine downsizing and subsequent downsizing of other related components are also advantageous in packaging, and frees up more space in the engine compartment for other features such as vehicle safety enhancement and allows better aerodynamic design of the vehicle [11].

## II. EXPERIMENTAL DETAILS

Study was performed on Bolero, Mahindra of 2.5D four cylinder and 4 stroke engine. Analysis was carried out at speed4000rpm. The intake manifold temperature and pressure are 298K and 100 KPa respectively.
The volume flow rate of air is measured using the air box method. Piezoelectric pressure sensor is used for the measurement of engine pressure inside the cylinder. For the measurement of pressure inside engine cylinder 6045A type piezostar @ pressure sensor M8x 0.75 is used. Its range is up to 250 bar. By measuring the cylinder pressure the indicated mean effective pressure is obtained. Various readings of temperatures were recorded at different RPMs of the engine across the intercooler, HP stage compressor, after-cooler, engine exhaust, high pressure stage turbine, low pressure stage turbine at inlet and outlet. To measure the temperature, two sensors are placed at the inlet and outlet. The two sensors are called thermistors – they are components that change in resistance with temperature. These two sensors are connected to an electronic temperature measuring device which records the temperature of air at the inlet and the outlet at different RPMs of the engine.

Table -1: Test Engine Specification

| | |
|---|---|
| Model | Mahindra Bolero 2.5D engine |
| Engine Type | Four stroke, four cylinder , 8 valves, water cooled, direct injection diesel engine |
| Rated speed | 4000 rpm |
| Maximum power @4000 rpm | 53 KW |
| Maximum torque @4000 rpm | 153 N.m |
| Engine capacity | 2.5litre / 2498 cc |
| Bore | 94 mm |
| Stroke | 90 mm |
| Compression ratio | 23:1 |

Table 1 shows the test engine specification of Mahindra Bolero engine.

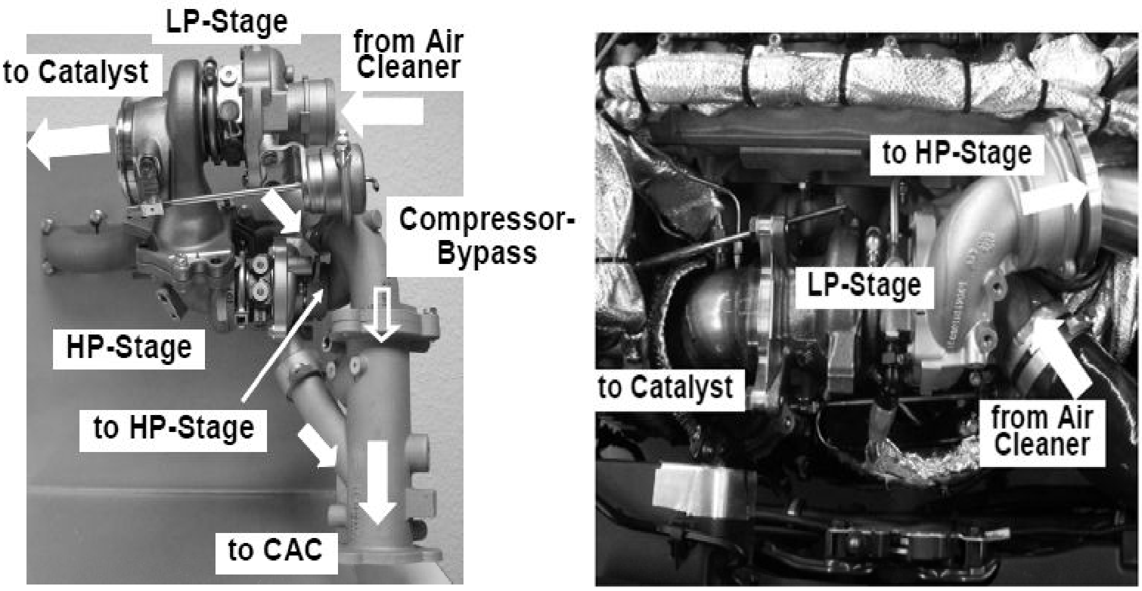


Figure 1: Two-stage turbocharger with Mahindra Bolero engine

Figure 1 shows the experimental setup of two stage turbocharger with Mahindra Bolero engine.

Table -2:Specifications of two stage turbocharger

| Parameters | LP stage turbocharger | HP stage turbocharger |
|---|---|---|
| Number of blades on Compressor | 8 | 6 |
| Compressor efficiency | 70% | 70% |
| Compressor inlet blade tip diameter (mm) | 57 | 50 |
| Compressor outlet blade tip diameter (mm) | 76 | 62 |
| Absolute air angle at inlet to impeller | 90° | 90° |
| Relative blade angle at inlet to impeller | 42° | 42° |
| Absolute air angle at outlet to impeller | 75° | 75° |
| Relative blade angle at outlet to impeller | 36° | 36° |
| Number of blades on turbine | 12 | 9 |
| Turbine efficiency | 60% | 60% |
| Turbine inlet blade tip diameter (mm) | 62 | 50 |
| Turbine outlet blade tip diameter (mm) | 70 | 55 |

Table 2 shows the specification of two stage turbocharger installed with Mahindra Bolero engine.

## III. RESULTS AND DISCUSSION

The results of volumetric efficiency, density, brake power, brake thermal efficiency brake specific fuel consumption, mechanical efficiency and intercooler has been obtained on single stage and two stage turbocharging on a Mahindra Bolero engine of 53 KW at rated speed 4000 rpm.

Table -3: Input parameters of Mahindra Bolero naturally aspirated engine at 4000 rpm

| | |
|---|---|
| Engine speed | 4000 rpm |
| Inlet Temperature of air | 298K |
| Inlet pressure of air | 100 KPa |
| Volumetric efficiency | 80% |

Table 3 shows the input parameters for naturally aspirated engine. Volumetric efficiency for naturally aspirated engine was 80%.

Table -4:Output parameters of Mahindra Bolero naturally aspirated engine at 4000 rpm

| PARAMETER | RESULT |
|---|---|
| Brake mean effective pressure (BMEP) | 636KPa |
| Brake power ($P_b$) | 53 KW |
| Brake thermal efficiency | 22.47% |
| Density | 1.169$Kg/m^3$ |
| Brake specific fuel consumption | 0.377$Kg/Kw.hr$ |
| Mechanical efficiency | 86.52% |

Table 4 shows the output parameters for naturally aspirated engine. The BMEP and the power output were 636 KW and 53 KW respectively.

Table -5:Input parameters of Mahindra Bolero engine with single stage turbocharger at 4000 rpm

| | |
|---|---|
| Engine speed | 4000 rpm |
| Inlet Temperature of air | 298K |
| Inlet pressure of air | 100 KPa |
| Volumetric efficiency | 133.62% |
| compressor pressure ratio | 1.71 |
| Pressure at inlet to engine cylinder | 167KPa |
| Effectiveness of intercooler | 0.9 |

Table 5 shows the input parameters for engine with single stage turbocharger. It has been seen that the volumetric efficiency is increased to 133.62% from 80%.

Table -6:Output parameters of Mahindra Bolero engine with single stage turbocharger at 4000 rpm

| PARAMETER | RESULT |
|---|---|
| Outlet temperature from compressor | 369 K |
| Outlet temperature of intercooler | 305 K |
| Brake mean effective pressure (Bmep) | 1103.78Kpa |
| Brake power ($P_b$) | 91.85 KW |
| Brake thermal efficiency | 23.29% |
| Density | 1.9$Kg/m^3$ |
| Brake specific fuel consumption | 0.36$Kg/Kw.hr$ |
| Mechanical efficiency | 91.75% |
| Work of compressor ($W_C$) | 6.65 KW |
| Work of Turbine ($W_T$) | 7.57 KW |
| Exhaust gas pressure | 241KPa |
| Exhaust gas temperature | 980 K |

Table 6 shows the output parameters for engine with single stage turbocharger. It has been seen that the brake power is increased to 91.85 KW from 53 KW.

Table -7: Input parameters of Mahindra Bolero engine with two stage turbocharger at 4000 rpm

| | |
|---|---|
| Engine speed | 4000 rpm |
| Inlet Temperature of air | 298K |
| Inlet pressure of air | 100 KPa |
| Volumetric efficiency | 180% |
| LP compressor pressure ratio | 1.71 |
| HP compressor pressure ratio | 1.37 |

| | |
|---|---|
| Pressure at inlet to engine cylinder | 224 KPa |
| Effectiveness of intercooler and aftercooler | 0.9 |

Table 7 shows the input parameters for two stage turbocharger. It has been seen that volumetric efficiency increased to 180% from 80%.

Table -8: Results for output parameter of Mahindra Bolero engine with two-stage turbocharger at 4000 rpm

| PARAMETER | RESULT |
|---|---|
| Outlet temperature from LP compressor ($T_2$) | 369 K |
| Outlet temperature of intercooler ($T_3$) | 305 K |
| Outlet temperature from HP compressor ($T_4$) | 346 K |
| Outlet temperature of aftercooler ($T_5$) | 303 K |
| Brake mean effective pressure (BMEP) | 1516Kpa |
| Brake power ($P_b$) | 126.25 KW |
| Work of HP compressor($W_{HPC}$) | 6.37 KW |
| Work of HP Turbine ($W_{HPT}$) | 7.33 KW |
| Work of LP compressor($W_{LPC}$) | 6.65 KW |
| Work of LP Turbine ($W_{LPT}$) | 7.57 KW |
| Exhaust gas pressure | 435 KPa |
| Exhaust gas temperature | 1000 K |
| Brake specific fuel consumption | 0.34 $Kg/Kw.hr$ |
| Density of air at inlet to engine cylinder | 2.58 $Kg/m^3$ |
| Mechanical efficiency | 93.86% |
| LP stage turbocharger Rotation speed | 80000 rpm |
| LP Compressor pressure ratio | 1.71 |
| LP Turbine pressure ratio | 2.45 |
| HP stage turbocharger Rotation speed | 91000 rpm |
| HP Compressor pressure ratio | 1.37 |
| HP Turbine pressure ratio | 1.75 |

**Table 8**: shows the output parameters for engine with two stage turbocharger. It has been seen that the BMEP and power output increased to 1516 KPa and 126.25 KW from 635 KPa and 53 KW respectively.

Table -9: Results for naturally aspirated, single-stage and two-stage turbocharged engine at 4000 rpm

| Type of engine | Brake power(KW) | Brake thermal efficiency (%) | Volumetric efficiency (%) | BSFC (Kg/KWh) |
|---|---|---|---|---|
| Naturally Aspirated engine | 53 | 22.47 | 80 | 0.377 |
| Single stage turbocharged engine | 91.85 | 23.29 | 133.62 | 0.36 |
| two stage turbocharged engine | 126.25 | 24.75 | 180 | 0.34 |

Table 9 shows the results of naturally aspirated, single-stage and two-stage turbocharged engine at 4000 rpm. It has been seen that two stage turbocharging increases brake power, brake thermal efficiency and volumetric efficiency and decreases BSFC.

Table -10: Results for naturally aspirated, single stage and two turbocharged engine at 2000 rpm

| Type of engine | Brake power(KW) | Brake thermal efficiency | Volumetric efficiency | BSFC (Kg/KW hr) |
|---|---|---|---|---|
| Naturally Aspirated engine | 32 | 33.04 | 80 | 0.258 |
| Single stage turbocharged engine | 42.2 | 33.20 | 105.76 | 0.254 |
| two stage turbocharged engine | 57 | 33.78 | 142.78 | 0.25 |

Table 10 shows the results of naturally aspirated, single-stage and two-stage turbocharged engine at 2000 rpm. It has been seen that two-stage turbocharging increases brake power, brake thermal efficiency and volumetric efficiency and decrease BSFC.

Since two-stage turbocharging increases the engine specific power output the engine is downsized. The length to bore diameter ratio$(L/D)$ is maintained same for both original engine and downsized engine is 0.957. The length and bore diameter of downsize engine are reduced from 90 mm and 94 mm to 81.4 mm and 85 mm respectively. The fuel consumption of downsized engine to produce the power output oforiginal engine of53 KW is reduced from 5.55 $X10^{-3}$ $Kg/s$ to 4.09 $X10^{-3}$ $Kg/s$ at 4000 rpm.

The graphs are drawn for different parameters of naturally aspirated engine, engine with single-stage turbocharger and engine with two-stage turbocharger as given below.

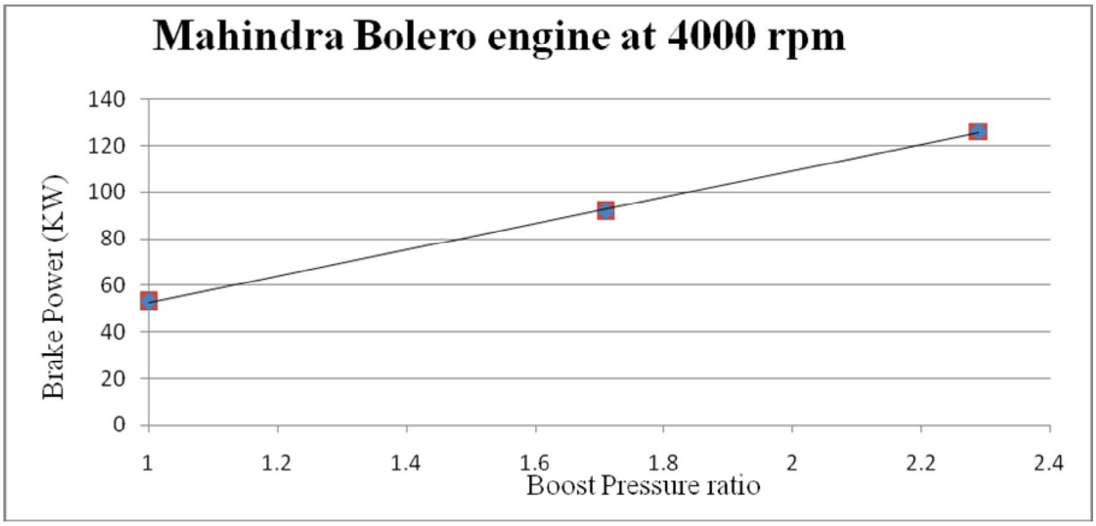


Figure 2: Brake Power v/s Compressor Pressure Ratio at 4000 rpm

Figure 2 shows the change of brake power with respect to pressure ratio of compressor. From the above graph it is seen that as the compressor pressure ratio increase. It increases because of increased in the speed of turbocharger because of high exhaust mass as the speed of engine increase. Increase in compressor boost pressure increases inlet pressure of air going inside the engine cylinder. As the intake pressure increases it increase the density which enhances the combustion because of more oxygen in the dense air as a result of which brake power increases. For naturally aspirated engine the compressor pressure ratio was 1 and corresponding power of 53 KW. For single stage turbocharging compressor pressure ratio increase to 1.71 which increase the intake pressure. And increase in intake pressure increase the brake power to 91.85 KW. For two stage turbocharging compressor pressure ratio further increase to 2.29 which increase the intake pressure. And increase in intake pressure increase the brake power to 126.25 KW.

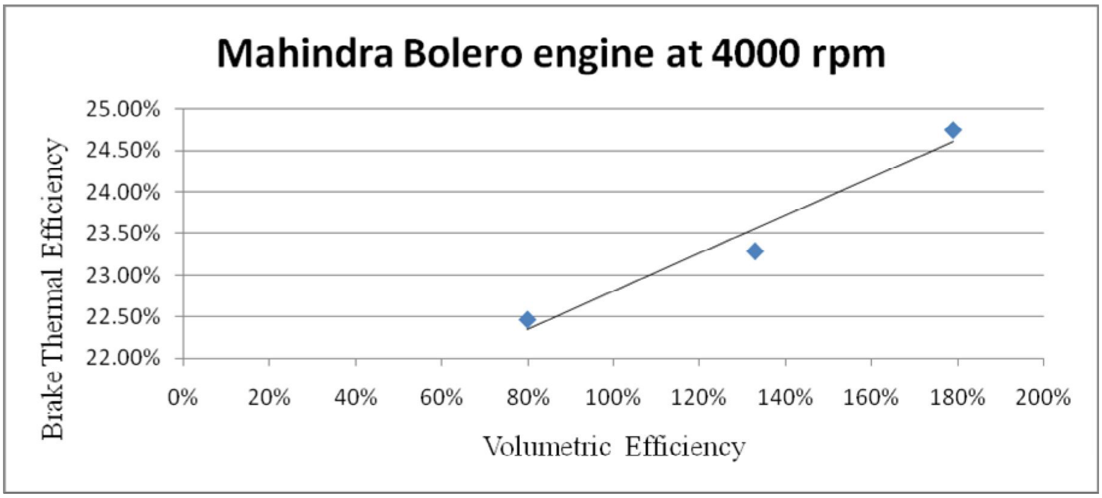


Figure 3: Volumetric efficiency v/s Brake Thermal efficiency at 4000 rpm

Figure 3shows the change of volumetric efficiency with respect to brake thermal efficiency. From the above graph it is seen that as a volumetric efficiency increases because of increase in volume of air sucked inside the engine cylinder.This is because of increase in pressure and density of air the volume of air inside the ylinder increased. The brake thermal efficiency increases because of increase in volumetric efficiency. The increase in density of air which enhances the combustion which directly increases the brake thermal efficiency. For naturally aspirated engine volumetric efficiency was 80% and corresponding brake thermal effieency was 22.47%. For single stage turbocharger volumetric efficiency increase to 133% and corresponding brake thermal efficiency was 23.29%. For two stage turbocharger the volumetric effciency increased to 179% and corresponding brake thermal efficiency is 24.9%. For turbocharging volumetric efficiency increases because of increasing the intake pressure of air and increase in density which enhances the combustion and increases the brake thermal efficiency.

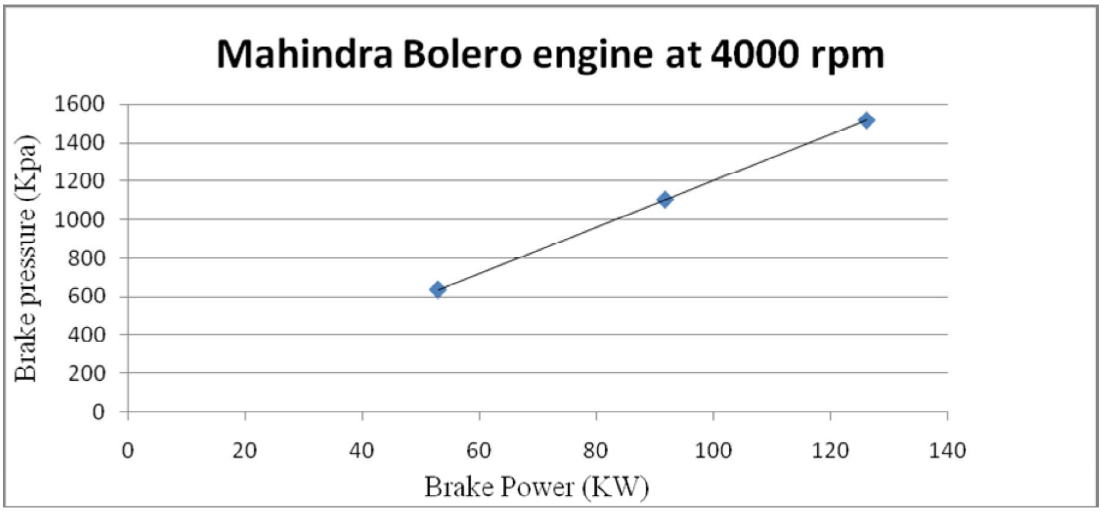


Figure 4: Brake mean effective pressure v/s Brake power at 4000 rpm

Figure 5 shows the change of brake power with respect to brake mean effective pressure. From the above graph it is seen that brake power of engine increases as the brake mean effective pressure of engine increases. Brake mean effective pressure is proportional to density of air going inside the engine cylinder. As the density of air increase it increases the brake mean effective pressure which increases the power. The brake mean pressure for naturally aspirated engine is 636KPa and corresponding brake power is 53 KW. For single-stage turbocharging brake mean pressure increases because of increasing the intake pressure of air inlet the engine cylinder. Brake mean pressure for single stage turbocharger is increasing to 1103.08Kpa and corresponding brake power is 91.85 KW. Turbocharging increase the pressure of air inlet to engine cylinder by compressing the air and it increases the density. For 2-stage turbocharging brake mean pressure is increase from 636 KPa to 1516.28KPa and corresponding brake power is 126.25 KW.

## IV. CONCLUSION

Turbocharging increases the density of air inlet to the engine as a result of which air intake capacity of engine increases hence more amount of air can be pumped inside a cylinder which enable an enhanced combustion of diesel. Two stage turbocharging increases the brake mean effective pressure by 138%. Power output by 138% as the more amount of fuel will be burnt within the same period as the mass taken per stroke is increased, brake thermal efficiency by 11% and volumetric efficiency increases by 125%. As a result of increased in intake manifold pressure by 124% the mean effective pressure, brake mean effective pressure increases which directly increases the braking torque by 77%. Also two stage turbocharged engine reduces brake specific fuel consumption by 10%. Since the two stage turbocharging increase the specific power output .The engine is downsized from 2.5L to 1.84L which decreases the fuel consumption by 26%.

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