Analytical and Experimental Turbocharger Matching to an off-Road Engine

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***Abstract*— The demand for construction equipment vehicles in India is rising with the GDP growth one among the highest in the world. The diesel engines that power these off road vehicles have different operating cycle and conditions which are always on the demanding side, hence the engines have to be robust in construction primarily and meet all the load cycles. On the other hand the rise in fuel prices have led to increase in operating cost hence now the market is looking for more fuel efficient engines for counter the fuel price rise. This work deals with selection and matching of Turbocharger to a direct injection diesel engine which was adopted for automotive application to suite CEV vehicles. First, selection of turbine and compressor is done by some assumptions and analytical method is used to match proper turbocharger and results were validated with experimental results.**

Keywords*— Construction Equipment Vehicles, Off Road, Operating Cycle, Direct Injection, Turbochrger.

INTRODUCTION

The fuel consumption rate across the world is increasing rapidly due to globalization and incerase in living standards. The reduction in fuel consumption rate is the major challenge faced by the combustion engineers is the world. Along with lower fuel consumption rate strigent emission norms to be attained. The off-road engines emit around 65% PM amoung all transport vehicles, hence to meet target power and torque curves and emissions norms the diesel engines must be fitted with turbochargers. An Off-Road vehicle having featues like better Power and backup torque , fuel efficent is obtained by selection and matchnig a proper turbocharger to the engine. The varing the configaration like inertia of the wheel, Wheel trim, A/R value the optimum turbocharger can be matched. In present work A/R ratio is selected for matching turbocharger.The brief discription about A/R ratio and its effect on engine performance is as follows.

A/R ratio is a dimensional parameter, used to define turbine and compressor housings. It is defined as ratio inlet cross-sectional area of the housing to the distance from center of turbine or compressor and center of cross sectional area.

Compressor A/R –The Compressor and Engine performance is not affected by any change in the A/R value of Compressor inlet housing, but normally in low boost application engines to attain Optimum performance larger A/R housings are used and in high boost application Engines compressor housings of smaller A/R value are used.

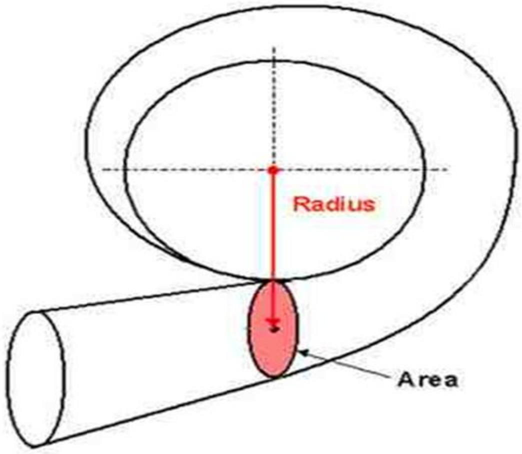

Figure.1 Compressor housing showing A/R characteristic

Turbine A/R – By changing the A/R value of turbine housing the turbine and engine performance is greatly affected. The Exhaust gas flow rate through the turbine is adjusted by changing A/R value of the turbine. The smaller A/R housing increases the exhaust gas velocity inside the turbine housing; hence the turbine wheel spins faster at lower speeds thus it delivers the quicker boost. The smaller A/R housing engines require higher exhaust backpressure at exit hence less power is obtained at higher speeds.

Conversely, the larger A/R value decreases the rotational speed of the turbine and hence resulting in lower boost value, but better power is obtained at higher speeds due to lower exhaust back pressure at exit. When selecting turbocharger with different A/R options, the vehicle application type and performance targets are taken into consideration for deciding A/R ratio.

The Off Road engines and lower speed vehicles require higher boost pressure and higher torque level at lower speeds; the smaller A/R can be used. Conversely, for higher speed applications i.e. race cars and where higher speed with lower peak power and torque are prime requirements, hence larger A/R ratio can be used.

Engine Specification and Experimental Setup

Selected Engine for this work is a direct injection, inline diesel engine, with specifications as listed in Table 1.

Table.1 Engine Specifications

Type	4- Stroke, Water Cooled
Bore/Stroke ratio	0.84
No. of Cylinders	4 Cylinder
Aspiration	Turbocharged Intercooler
Injection System	Inline pump
Compression Ratio	16.5:1

Experimental Setup

The test was performed at PTE lab of ARAI, Pune. The engine was coupled with Zollerner B300AC eddy current dynamometer connected by a propeller shaft. The engine was instrumented with ABB, sensiflow, SFC-05 air flow meter to measure mass flow rate of intake air. Smoke and PM emissions are measured using AVL 439 smoke meter and AVL, SPC 472-04 equipment respectively. The engine was facilitated with AVL Indiset setup for cylinder pressure measurement. The fuel flow measurement was done by KS, FC-150, Dynamic FC meter and BS-IV diesel fuel was used for experimental work. The weight balance is done by Sartorius, CP2P-F.Testing conditions such as intake; ambient pressure and temperature were maintained as per standards. The test method used for experimental work was FTP test.

Figure.2 Engine test bed Setup

Figure.3 Engine Setup outside test cell

Dynamometer Engine Testing

In this investigation, environmental conditions in which the engine was tested were controlled at specific levels so that the influence of these factors on engine emissions was minimized. To isolate the influence the of environment conditions like ambient temperature, pressure, humidity etc. on exhaust emission performance, ambient air that the engine aspirates was conditioned and controlled to specific temperature, pressure and humidity as required by the emission regulations, using a series of complex instruments called Sea Level Altitude Simulation System (SLASS). Other background factors like fuel temperature, exhaust backpressure, sulphur content in fuel, intercooler temperature, etc. were maintained within the specified limits simulating engine operation on the vehicle.

Table .2 Conditions for Dynamometer Engine Testing

Boundary Conditions Maintained During Testing	
Diesel Fuel Type	BS-IV Reference Fuel
Fuel Inlet Temperature	40 ± 2°C
Air Intake Depression	200 mm of H_2O @ 2500 rpm
Exhaust Back Pressure	90 mm of Hg @ 2500 rpm
IC outlet temperature	48 ± 2°C
Water Outlet Temperature	85 ± 5°C
Lubricating Oil Temperature	Max. 120°C
Relative Humidity	40 ± 5°C
Inlet Air Temperature	25 ± 2°C
Specific Gravity of Fuel	0.84

TURBOCHARGER SELECTION

Turbocharging a 4-stroke diesel engine is complicated process because of different mass flow rate and operating conditions (Speed, load) of both the turbocharger and engine. Especially compressor mass flow rate should be matched to engine breathing requirements.

Compressor selection

The objective of turbocharger matching is to have the engine system operate within the heart of the map at all times. Operation in the other three region choke, limiting motor speed and surge produce unacceptable engine operation and must be avoided. For initial estimation of flow parameter, actual air flow rate is calculated at rated power and rated torque speed of the engine, based on following steps

1- Calculation of required actual air flow rate (m_A)

$$m_A = \frac{(\eta\, \boldsymbol{vol * D * N * p})}{(\boldsymbol{2 * R * T})} \tag{1}$$

Where, ρ- Density of air in intake manifold

D – Displacement per cycle

N – Engine speed (rev/sec)

P – Intake manifold pressure

R – Gas constant of air

T – Intake manifold temperature

Above equation shows that actual mass flow requirements of air through the engine can be determined once approximations for **η vol, D, N, P, R** and **T** are established. We are working with air; valve for gas constant, **R** for air is 287.05 J/Kg.K

By definition a breathing line is the characteristic aspiration of the engine at a given speed. Breathing requirements are found for N = 1300 rpm (max. torque speed) & N = 2500 rpm (Rated Speed)

By considering volumetric efficiency of other engines of nearby rating, pier experience and consultation with turbocharger supplier, volumetric efficiency is assumed as 90% at 1300 rpm and & at rated rpm (75% for worst case scenario). And target to achieve for backup torque is 34%.

$m_A = 0.017 \times (\frac{P}{T})$ @ Rated Speed

$m_A = 8.828 \times 10^{-3} \times (\frac{P}{T})$ @ Rated Torque Speed

In order to solve the above equations for m_A , P values can be written in terms of compressor pressure ratio.

$$PR_c = \frac{[P_{manifold} + P_{Intercooler\ loss}]}{[P_{ambient} - P_{air\ intake\ depression}]} \quad (2)$$

$P_{manifold} = PR_c \times [P_{manifold} - P_{Intercooler\ loss}] - P_{Intercooler\ loss}$

By knowledge and experience of declared values of P_(air intake depression) & P_(Intercooler loss) are assumed as

Table.3 Assumed values of P_{AID} ,$P_{int.loss}$

	P_{AID}	$P_{Inter.loss}$
@ 1300 rpm	7 mbar	120 mbar
@ 2500 rpm	20mbar	140 mbar

T in above equation is that of intake manifold. For intercooled engine 48±2°C intake manifold temperature at rated speed is declared by Engine manufacture therefore it is assumed as 48°C and at rated Torque speed it is considered as 36°C.

These are approximate air flow through engine; in order to place this relationship in compressor map coordinates the flow parameter must be used.

$$FP = m_A \times \sqrt{(T_{ref}/T_{in.t})} \quad (3)$$

Where, T_{ref} = reference temperature declared by manufacturer.

T_{ref} = 293K (declared)

$T_{in.t}$ = air intake temperature = 300K (declared)

From above all assumptions for 2.2 pressure ratio we have

Table.4 Results at 2.2 Pressure ratio

	At 2500 rpm	At 1300 rpm
m_A (m3/sec)	0.175	0.098
FP (m3/sec)	0.173	0.0968

Similarly the Flow Parameter is calculated for different pressure ratio to find working zone in compressor map. Breathing lines are plotted on a graph between pressure ratio and flow rate as show in Figure 4. They are used to determine the compatibility of compressor flow range with that of reciprocator. Initial check is accomplished by superimposing these breathing lines of rated and peak torque on compressor map and seeing if the flow requirements fit within proposed compressor map. Generally breathing lines of two speeds rated and peak torque is used because this covers whole useful working range. This plot of flow requirement is shared with turbocharger manufacture and asked for samples fulfilling these flow requirements.

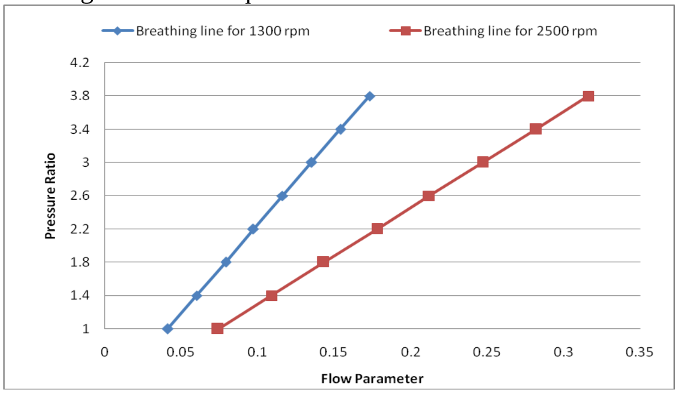

Figure.4 Breathing requirements for 1300 rpm & 2500 rpm

Turbine selection

The specific location on breathing line is established by the combination of reciprocator, compressor and turbine. The first step in seeing how the system will come to gather is to estimate the mass flow requirements of rated and peak torque operation. This can be accomplished by setting the desired power level and estimating BSFC and air fuel ratio. Given power and BSFC the fuel rate can be determined. From the combination of fuel rate and air fuel ratio the system mass flow requirements can be established.

Data shared with turbocharger supplier for rated speed i.e. 2500 rpm

Power – 121.2 KW from target data.

BSFC - 230 g/ KWh benchmarking.

A/F ratio – 23 from experience and emission requirements

This leads to

Fuel – approximately 27.88 Kg/h

Air flow – 641.15 Kg/h

Pressure ratio found by means of breathing line, horizontal line where air flow will cut the breathing line will give pressure ratio, as shown in Figure 5 below. Approximately 2.18 is the pressure ratio we get.

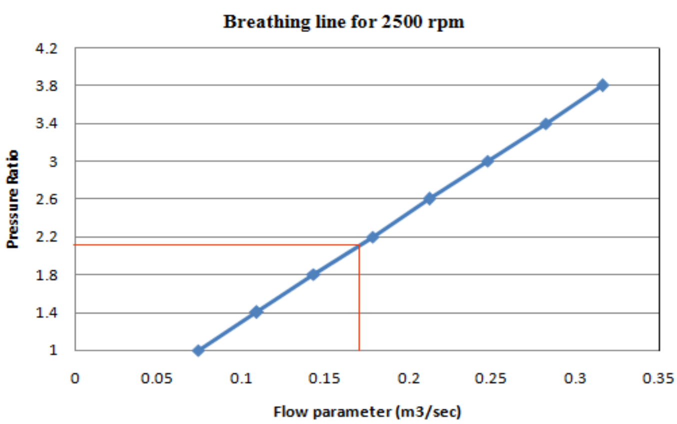

Figure.5 Breathing requirements for 2500 rpm

Analytical Turbocharger Matching

In this section the work conducted by P. F. Freeman et al is used for analytical matching of turbocharger. This method uses the minimum of detailed information and certain assumptions are made for various parameters like A/F ratio, BSFC, Volumetric Efficiency and heat lost to coolant about the running conditions of the engine. The analysis may be made with power for a known boost, or boost for a known power. In both cases the basic calculations are similar and minimum information required is bore, stroke, number of cylinder, speed and power or boost level.

Consider the case where an engine match is required at a given speed and power. The given in formation will be speed, power and swept volume, so estimate will be required of volumetric efficiency, air-fuel ratio, fuel consumption and fraction of heat lost to the coolant from similar engines, also of P_1, T_1 and P_4 (usually ambient conditions). If an intercooler is fitted intercooler effectiveness (ε), ratio of pressure drop across intercooler (PD) and temperature of intercooler coolant (T_c) will be needed. The analytical matching at rated speed is based on following steps:

(1) Calculation of Fuel mass flow rate (M_{fuel})

$$M_{fuel} = \frac{\text{EP}\times\text{BSFC}}{3600} \qquad (4)$$

(6.7)

Where,

M_{fuel}= Fuel mass flow rate (g/s),

EP = Engine Power,

BSFC = brake specific fuel consumption (g/Kw.h)

$$M_{fuel} = \frac{121.2 \times 230}{3600}$$

= 7.743 g/s

(2) Calculation of air mass flow rate (M_{air})

$$M_{air} = M_{fuel} \times \text{A/F} \qquad (5)$$

Where,

M_{air} = Air mass flow rate (g/s)

A/F = air fuel ratio

M_{air} = 7.743 × 23

= 178.1 g/s

(3) Calculation of actual volumetric air flow (V_a)

$$V_a = \frac{M_{air}\times T}{D_a\times P\times 1000} \qquad (6)$$

Where,

T = ratio of T_1/288

P = ratio of P_1/1.013

D_a = Standard air density at 288K & 1.013 bars

V_a = Actual volumetric air flow (m³/s)

$$V_a = \frac{178.1 \times (\frac{293}{288})}{\left(\frac{1.013\times10^5}{287\times288}\right)\times(\frac{0.981}{1.013})\times1000}$$

V_a = 0.153 m³/s

The density ratio across the compressor (and intercooler, if fitted) may be calculated, but it should be noted that this is based on total condition rather than static. The latter would be requiring an input of compressor geometry, which is not appropriate to this level of calculation. The error introduced is very small.

The required Engine air density is calculated as

$$\text{ED} = \frac{\text{V}_\text{a}\times120}{\eta_\text{vol}\times\text{D}\times\text{N}} = \frac{P_a}{R\times T} \qquad (7)$$

Where,

η_{vol} = Engine volumetric efficiency

D = Displacement of engine (liter)

N = engine Speed (rpm)

$$\text{ED} = \frac{1.45\times10^5}{287\times321}$$

= 1.57

The turbocharger is matched in such a way that the engine density ratio and the density across the compressor both value should match to get turbocharger and engine combination performance characteristics.

To calculate the density ratio across the compressor the inlet and outlet temperature across the compressor must be known, hence

$$T_{c2} = T_{c1} + \frac{T_{c1}}{\eta_c} \times (\{R_c\}^{(\frac{\gamma-1}{\gamma})} - 1) \qquad (8)$$

Where,

T_{c1} = inlet temperature of the compressor (K)

T_{c2} = outlet temperature of the compressor (K)

η_c = Compressor Efficiency

R_c = Compressor pressure ratio

γ = ratio of specific heat = 1.4 for air

The air mass flow rate is already known, so from the compressor map approximate value of pressure ratio & efficiency may be selected. And iterative method is followed to specify the correct operating point on the compressor map. The engine density value and compressor density both should match.

$$T_{c2} = 293 + \frac{293}{0.74} \times [(2.433)^{(\frac{1.4-1}{1.4})} - 1]$$

= 407.508K

Since the engine is facilitated with intercooler, there is some losses across the intercooler these losses are taken into account to calculate the compressor density ratio. By experience and knowledge and the pressure drop (PD) across the intercooler is assumed as 0.14 at rated speed and 0.12 at intermediate speed.

$$DR_c = \frac{T_{c1}}{T_{c2}} \times R_c \times PD \quad (9)$$

$$= \frac{293}{407.508} \times 2.433 \times 0.9091$$

$$= 1.59$$

Now, both density values are matching at pressure ratio = 2.433 & compressor efficiency = 74%. Now for this pressure ratio (2.433) and rated speed (2500 rpm) conditions, we have designed the turbine.

Table.5 Results from compressor map at various speeds

Engine Speed(N) rpm	Manifold Pressure (P_a) bar	Compressor Pressure ratio (P_2/P_1)	Compressor mass flow (m^3/sec)
2500	1.45	2.433	0.153
2400	1.41	2.36	0.148
2300	1.38	2.30	0.143
2200	1.35	2.26	0.139
2100	1.32	2.2	0.136
2000	1.31	2.18	0.133
1900	1.30	2.199	0.1315
1800	1.29	2.2	0.125
1700	1.28	2.12	0.115
1600	1.25	2.10	0.109
1500	1.23	2.05	0.103
1400	1.22	2.05	0.097
1300	1.21	2.0	0.09
1200	1.1	1.8	0.079
1100	1.05	1.7	0.066
1000	0.98	1.55	0.0615

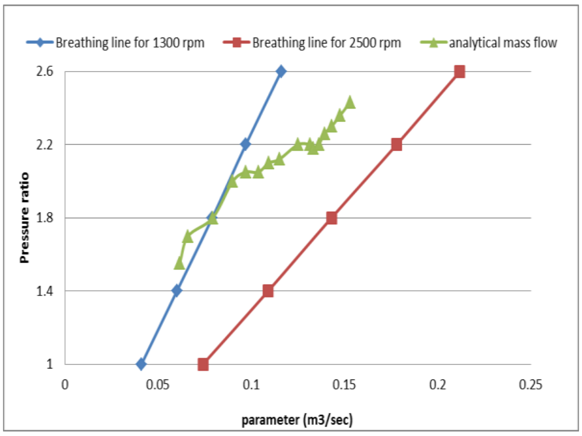

Figure.6 Comparison between initial assumptions and theoretical results.

Inference:

From initial assumptions and theoretical turbocharger matching procedure it can be inferred that the density values across the compressor and engine are matched by iteration method and the corresponding pressure ratios are lies in the 74% to 76% efficiency zone of the compressor map as shown in figure 6. Further experimental method is used to complete the matching procedure.

Experimental Turbocharger Selection

The lower A/R ratio produces smaller incident angle hence higher peripheral speed, which produces more turbine speed and quick boost. Quicker and higher boost makes trapping more air in the cylinder. The availability of more air produces good combustion of fuel and air mixture. Based on the initial assumptions and theoretical matching results, three Turbochargers having different A/R ratio configuration of turbine and compressor housing were used in experiments

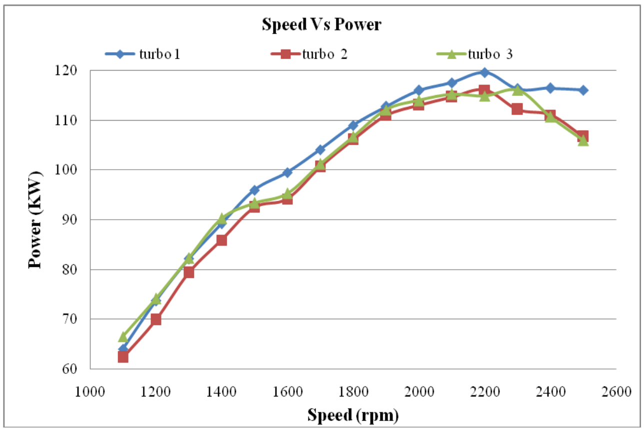

Figure.7 Speed Vs Power

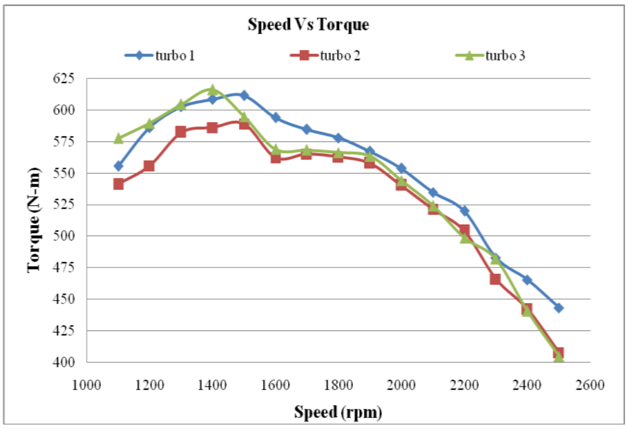

Figure.8 Speed Vs Torque

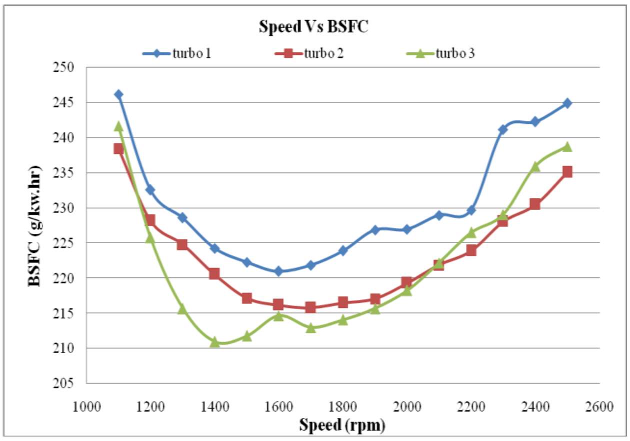

Figure. 9 Speed Vs BSFC

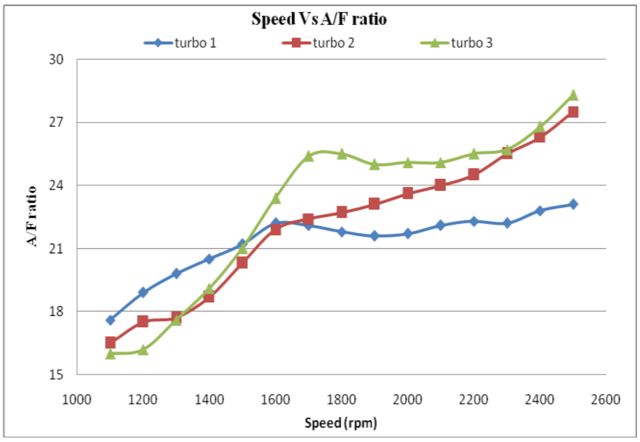

Figure.10 Speed Vs A/F Ratio

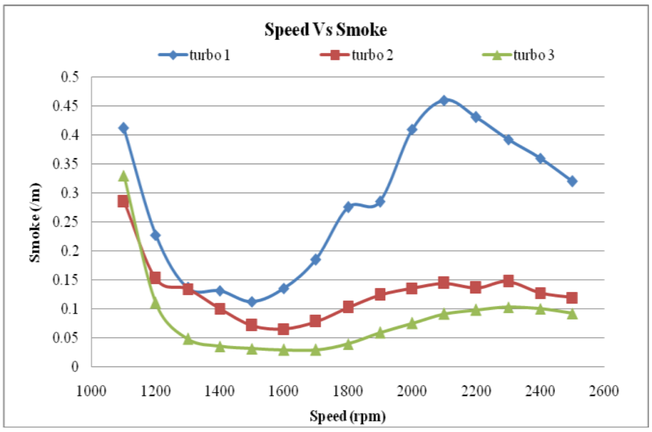

Figure.11 Speed Vs Smoke

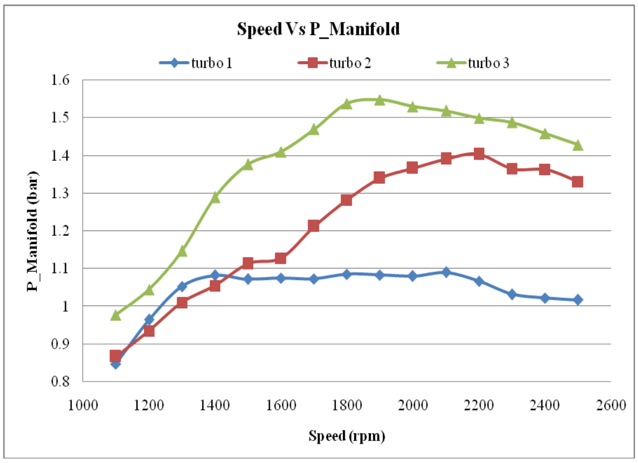

Figure.12 Speed Vs P_Manifold

The FTP test results are shown in fig. 7-12 .After conducting the FTP test of 3 turbochargers, Turbocharger 1 & turbocharger 3 giving almost same power and Torque at lower rpm, however Turbocharger 3 is giving slightly higher at max. Torque speed. After maximum Torque speed turbocharger 1 is giving more power and Torque than turbocharger 3 and at rated speed turbocharger 1 giving 9.5% more power than turbocharger 3Turbocharger 3 is giving 37.3% backup torque and turbocharger 1 and 2 giving 43.8% and 52.15 % respectively. Turbocharger 3 shows 6% improvement BSFC compared to turbocharger 1 at same power. The increased air density makes better combustion in diffusion phase and fuel required is also less in later combustion process. The interval between oxidation of soot and exhaust valve opening is short; hence fewer of smoke is formed in turbocharger 3.

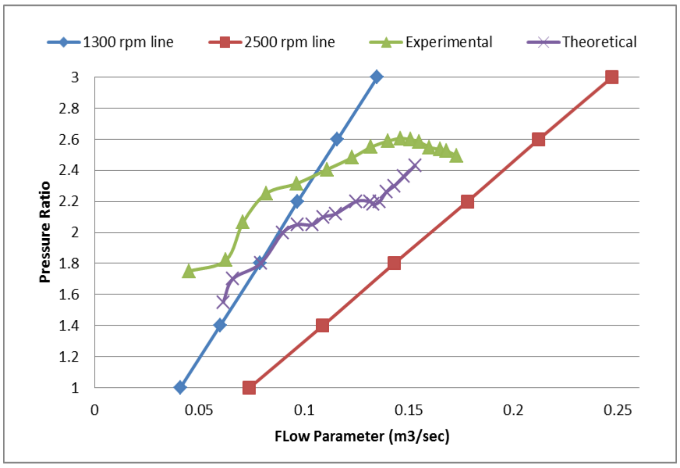

Figure.13 Comparison between analytical and experimental results

The turbocharger 3 gives optimum values in terms of BSFC, smoke, Power and Torque (up to rated Torque speed), fuel flow and it is working in the 70% to 74% of the compressor map as shown in figure 13.

CONCLUSION

Turbocharger matching was done of an Off-Road Engine suited for Excavator application with rated power of 121.2KW. The analytical turbocharger matching method gives the boost pressure ratio which in nearer to the heart region of the compressor map. And are lies in the 74% to 76% working zone of compressor map.

The experimental matching technique shows that the BSFC, smoke, Power and Torque (up to rated Torque speed), fuel flow are in favor of turbocharger 3

The experimental matching of turbocharger 3 on compressor map shows that it is working in the 70% to 74% of the compressor map and near to the heart region of the compressor map. The other two turbochargers are working in 60% to 65% zones. Finally turbocharger 3 was selected optimum engine combustion and performance.

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ABBREVIATIONS

CEV – Construction Equipment Vehicle
PM – Particulate Matter
PTE – Power Train Engineering
FTP – Full Throttle Performance
BSFC – Brake Specific Fuel Consumption
BS-VI – Bharat Stage VI