Downspeeding the Diesel Engine – A Performance Analysis

Master's Thesis in Automotive Engineering

ARUNACHALAM PRAKASH NARAYANAN

Department of Applied Mechanics
Division of Combustion
CHALMERS UNIVERSITY OF TECHNOLOGY
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Abstract

Owing to the high overall efficiency, truck manufacturers stick to diesel engines particularly in the case of long haul heavy duty trucks. In the ever increasing demand to increase the efficiency and to decrease the emission levels of the internal combustion engines an attempt is made in this thesis work to increase the fuel efficiency.

There are many ways to increase the fuel efficiency like varying compression ratio, downsizing, down speeding, advanced combustion concepts (PCCI, HCCI) etc. but each of the mentioned method comes with challenges. In this thesis work an effort is made to study downspeeding by means of extensive simulation using commercial software GT-Power.

By down speeding, the engine is made to run at low speeds and with high torques. For the same power the engine is operated at higher specific load (BMEP) which results in higher efficiency and reduced fuel consumption (BSFC). The reasons for increased fuel efficiency are reduced engine friction due to low piston speeds, reduced relative heat transfer and increased thermodynamic efficiency.

Since this is a conceptual study and experiments may require a new engine design altogether, scope of the study is limited to simulation results and no validation is done.

Contents

Preface

This report is a result of master thesis work carried out during spring and summer 2011 as a part of Master studies at the Department of Applied Mechanics. This is a collaborative work between Chalmers Combustion Divison and Volvo Powertrain. The examiner at Chalmers University has been Proffessor Ingemar Denbratt and the supervisor at Volvo Powertrain has been Udd Sören.

First and foremost I would like to thank my Professor Ingemar Denbratt who believed in me and gave me the thesis work in the first place; also I would like to extend my thanks to him for the support and advice he rendered during the work and thesis writing. I would also like to thank my guide at Volvo Powertrain Dr. Jakob Fredriksson who tutored throughout my thesis with his patience, knowledge and encouragement, without him the thesis would not be completed. Also thanks to Dr. Johan Wallesten for providing all infrastructure needs.

I would also like to thank Thomas Klang and Soren Udd of Volvo Powertrain for creating such a wonderful opportunity and for all the help they provided during the work. Finally I would like to thank all the people at Analysis fluids and combustion department – 91517 for the nice time.

Nomenclature

BMEP	Brake mean effective pressure
BSFC	Brake specific fuel consumption
BS	Brake specific
CAD	Crank angle degree
CFD	Computational Fluid Dynamic
DoE	Design of experiments
EGR	Exhaust gas recirculation
EATS	Exhaust after treatment systems
FMEP	Friction mean effective pressure
HCCI	Homogeneous charge compression ignition
HPC	High pressure compressor
HPT	High pressure turbine
LPC	Low pressure compressor
LPT	Low pressure turbine
MBT	Maximum brake-torque
N	Speed
PID	Proportional, Integral and Derivative
PCCI	Premixed charge compression ignition
T	Torque
VGT	Variable geometry position

1D/3D	one dimension / Three dimensional
rpm	Revolution per minute
D_{scale}	Scaled diameter
$D_{original}$	Original diameter
η_v	Volumetric efficiency
η_f	Fuel conversion efficiency
Q_{HV}	Heating value
$\rho_{a,i}$	Inlet air density

V_d	Displacement volume
F/A	Fuel to air ratio
A/F	Air to fuel ratio
A/R	Aspect ratio
$\dot{m}$	Mass flow
NO_x	Nitric oxide + Nitrogen dioxide

1. Introduction

Diesel engines are the primary source of power for trucks, bus, trains, ships and various power generators. In the past few decades the diesel engine has undergone many technological transformations, from fuel injector to exhaust catalysts. Still there is ample number of methods present to increase the efficiency of the engine. Recent focus in the diesel technology is downsizing, downspeeding, advanced combustion concepts. In this thesis work, downspeeding concept is extensively studied in the simulation environment. By downspeeding, the engine is operated at low speeds but with higher torque (same power produced). The speed change can be achieved by changes to the gear ratio.

1.1 GT-Suite

GT-Suite is overall virtual vehicle simulation software[1]. It has GT-ISE for model building, GEM3D for converting CAD models to GT-Suite models, GT-Cool for modelling under hood heat exchangers, fans...etc and it also has VT-Design for valve train modelling[1].

1.1.1 GT-Power

GT-Power is an industry standard simulation tool used for gas exchange, combustion analysis, control system analysis, exhaust gas analysis, coupled 1-D/3-D simulations, EGR system design etc[1]. This software is used to carry out the simulation work throughout the thesis. It uses 0-D simulations for in-cylinder thermodynamical analysis and 1-D simulations for flow analysis. It is used to perform both steady state (torque, power, efficiency, cylinder pressure analysis, turbomatching...etc) and transient response analysis (Turbocharger response, A/F ratio response…etc). It has an extensive library for engine models (includes all systems and subsystems), it is also possible to build our own models from simple engine to highly sophisticated concepts. For exhaust gas aftertreatment it has the built-in kinetics and aftertreatment device library which is used to analyse the different combinations of catalysts or kinetics involved in the catalytic process.

1.1.1.1 Engine model

The engine model used is a thirteen litre six cylinder diesel engine (D-13) with a rated power of 520 hp. Volvo uses both CFD models and 0D/1D models, due to the calculation time the engine calculations are done with 0D/1D models. The combustion model, VCATO, used in the simulations is developed by Volvo. VCATO (Volvo combustion analysis tool) is a combustion model based on same approach as chmela et al[2]. The input for VCATO is injection rate, trapped mass in the cylinder, composition of trapped mass (air or air and EGR), and pressure estimate during compression stroke[3]. The VCATO model calculates the rate of heat release (rate of fuel combustion), ignition delay, pressure, temperature etc[3]. Similarly EUISIM is a tool developed to calculate the fuel injection rate; the input parameters fed to model are fuel pressure in the injector and physical parameters of the injector. VNO_x is a NO_x model to calculate the NO_x produced during the combustion, the input parameters are cylinder pressure, rate of heat release, turbulence intensity, trapped gas mass and gas composition[3].

1.1.1.2 Friction model

The engine friction is low at low engine speeds; however operating at high BMEP means higher cylinder pressure. Care should be taken when choosing the engine operating speed, since at very low speeds the engine friction can be high with increasing loads because of the higher lateral forces on the piston.

In GT-Power the friction model used is a Chen-Flynn friction model, the engine friction is calculated using the following equation

$$FMEP = constant\ FMEP + (\max cylinder\ pr * peak\ cylinder\ pr\ factor) + (Mean\ piston\ speed * mean\ piston\ speed\ factor) + (Mean\ piston\ speed^2 * mean\ piston\ speed\ squared\ factor) \tag{1}$$

Constant FMEP = Accounted for accessories friction, recommended values are 0.3-0.5 bar

Peak cylinder pressure = Cylinder pressure dependency term in FMEP, recommended values are 0.004-0.006

Mean piston speed factor = Accounted for hydrodynamic friction in power cylinder (pressure/velocity), recommended values are 0.08-0.1 bar / (m/sec)

Mean piston speed squared factor = Accounted for windage losses (pressure/velocity^2), recommended values are 0.0006-0.0012 bar / (m/sec)^2

1.1.2 GT-Post

It is a data analysis tool used to plot, view and analyse data resulted from either GT-Suite simulations or external source[1].

1.1.3 DoE and DoE Post

With DoE (Design of Experiments) it is possible to create a large design of experiment simulation with ease. The DoE Post is a post processing tool used to analyse the DoE runs simulated in GT-Power, using DoE post optimiser it is also possible to optimise the factors used in DoE run.

2. Down speeding approach

The present max torque is reached between 1050-1450 rpm; the proposed torque curve of the downspeeded engine has maximum torque in between 800-1200 rpm. By producing the same power for lower speeds, i.e. the engine is operated with higher BMEP with lower speeds. By doing so the engine is made to operate in the efficient fuel consumption zone as shown in Figure 1. The required speed to wheels can be modified by changing either the transmission ratio or final drive ratio.

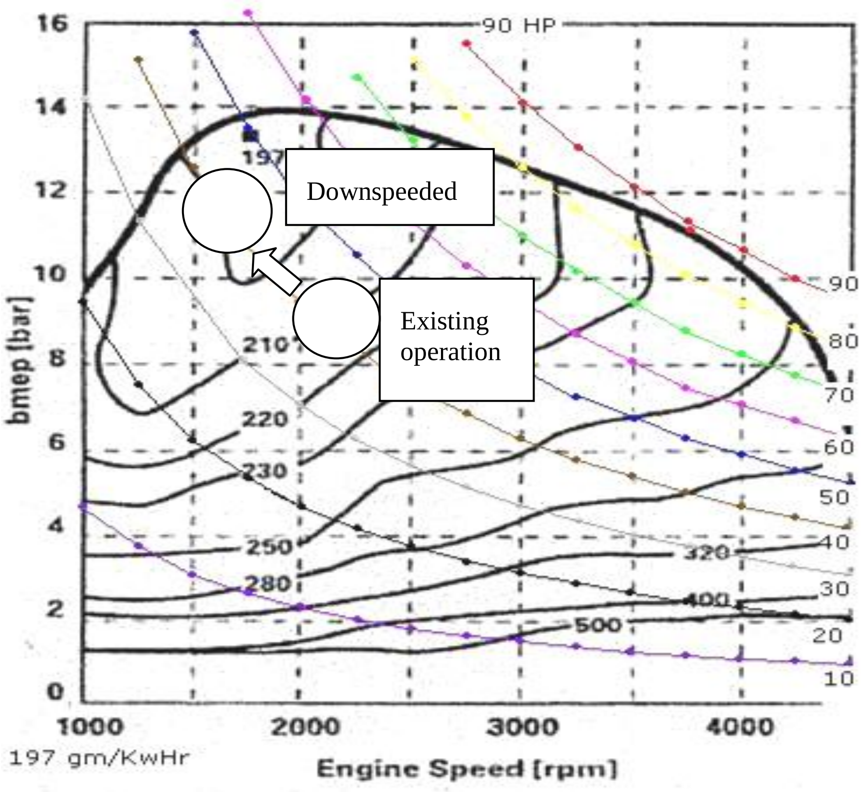

Figure 1. BSFC curve model for a diesel engine (courtesy TDI forum)

3. Hypothesis

3.1 Friction reduction

The engine speed and load has the major influence over the engine friction[4]. Engine friction contributes nearly 20% of the losses and reducing the frictional loss could increase the fuel economy[5]. Lowering the speed could result in reduced engine friction; however, the load factor cannot be neglected. For the BMEP of 15-20bars and beyond, the load plays an important role[6], since higher loads increase the lateral forces on the piston and can result in increased friction. When it comes to the task of reducing engine friction, the engine speed and the load applied plays a vital role.

3.2 Heat transfer

Reduced heat transfer in downspeeded engines when compared with existing engines could reduce the heat losses. The heat transfer is given by the Eqn.2.

$$Q = h_g * A * (T_g - T_w) \qquad (2)$$

With area being unchanged for both the engines and also the variation in wall temperatures and gas temperatures are not significant, the heat transfer coefficient is the factor that changes. According to Woschni's correlation[7] the heat transfer coefficient is directly proportional to the average gas velocities which in turn related to mean piston speed. Since the engines are operated with low mean piston speeds the relative heat transfer can be lower.

3.3 Faster combustion in CAD

Since the downspeeded engine is operated at low speeds the combustion takes places relatively faster in CAD (Crank angle degrees) when compared to the original engine speed. By having faster combustion in CAD the expansion work increases, thus a better fuel conversion efficiency. This increases the indicated work per cycle for the same fuel input, hence, increased combustion efficiency.

4. Engine Parameters considered in this thesis work

4.1 Injection timing

It plays an important role in combustion process, if the injection is too early majority of the combustion takes place in the compression stroke causing high compression work and heat losses losing much of useful energy, if the injection is retarded then majority of the combustion takes place in the expansion stroke causing a loss of expansion, hence a correct injection timing is required to achieve MBT timing.

4.2 Injection pressure

The injector's task is to inject fuel and mix with air. If the injection pressure is low, the fuel droplets will be large and proper mixing is not feasible which results in improper combustion resulting in high emissions, especially particulates.

4.3 Air-fuel ratio

The problem of air utilization arises when we try to increase the fuel quantity per cycle, this air utilization problem results in excessive soot which cannot be burned before exhaust. This black smoke or soot in the exhaust limits the air-fuel ratio. Therefore a minimum of lambda 1.25 is maintained.

4.4 NO_x emissions

As the emission regulations become more stringent, the need to reduce the NO_x emissions in an engine is inevitable. NO_x is primarily formed because of the high temperatures and presence of abundance of oxygen to oxidise the nitrogen during combustion[7]. The allowed engine-out NO_x level for this particular engine model is 5 g/kWh. Furthermore the NO_x emissions are reduced using a SCR (Selective catalytic reduction, sec 4.4.1).

4.4.1 SCR – Selective Catalytic Reduction

It is an after treatment system which uses reducing agent such as ammonia or urea to convert NO_x into diatomic nitrogen N_2 and water H_2O. The urea cost is also a considerable factor while calculating BSFC.

4.5 Cylinder peak pressure

Since the engine is operated at higher BMEP the pressure in the cylinder increases and for mechanical reasons the cylinder pressures are limited to 250 bars.

4.6 Exhaust temperature and pressure

The exhaust temperature and pressure are limited because of the design limitations. The max permissible exhaust temperature and pressure are 953K and 5.5 bars respectively.

4.7 Turbine and compressor speeds

The turbine wheel and compressor wheel are not entitled to run faster than certain speeds due to mechanical limitations. The speed limit is calculated by the following equation

$$D_{scale} = \sqrt{Mass\ flow\ factor} * D_{original} \tag{3}$$

$$D_{orig\,inal} - \ obtained\ from\ turbine\ and\ compressor\ map\ in\ the\ model$$

And from manufacturers data it is noted that a Titanium 85mm compressor wheel can run up to 124,000 rpm and using this relation, the speed limit is chosen for the scaled diameter.

4.8 EGR

Exhaust gas recirculation is a technique to reduce the NO_x formation in the engine. NO_x is mainly formed by the oxidation of nitrogen. This can be reduced by reducing the oxygen content and the temperature in the cylinder, this could be done recirculating already burned exhaust gas, which has a low oxygen content to the inlet, it replaces volume which could be filled by the air (more oxygen content that EGR), by doing so, the temperature and oxygen concentration during combustion is reduced and hence the production of the NO_x is decreased.

5. Methodology flowchart

Methodology followed in this thesis work is explained using a flowchart as shown in Fig.2. The work commenced with two torque curves obtained from Volvo Powertrain, one for downspeeded engine model and another for existing engine model. To find appropriate turbine and compressor sizes, turbomatching was done for both torque curves. Since only one turbine and compressor map is available in GT-Power, the turbine and compressor maps are multiplied with three massflow factors to create three sizes of turbines and compressors. The combination of the turbines and compressors are simulated for 8 operating points defined by Volvo Powertrain. Using the GT-Power optimizer the results were optimized for better BSFC. From the optimized results, comparison for the better configuration was done. (More details in section 6, 7, and 8)

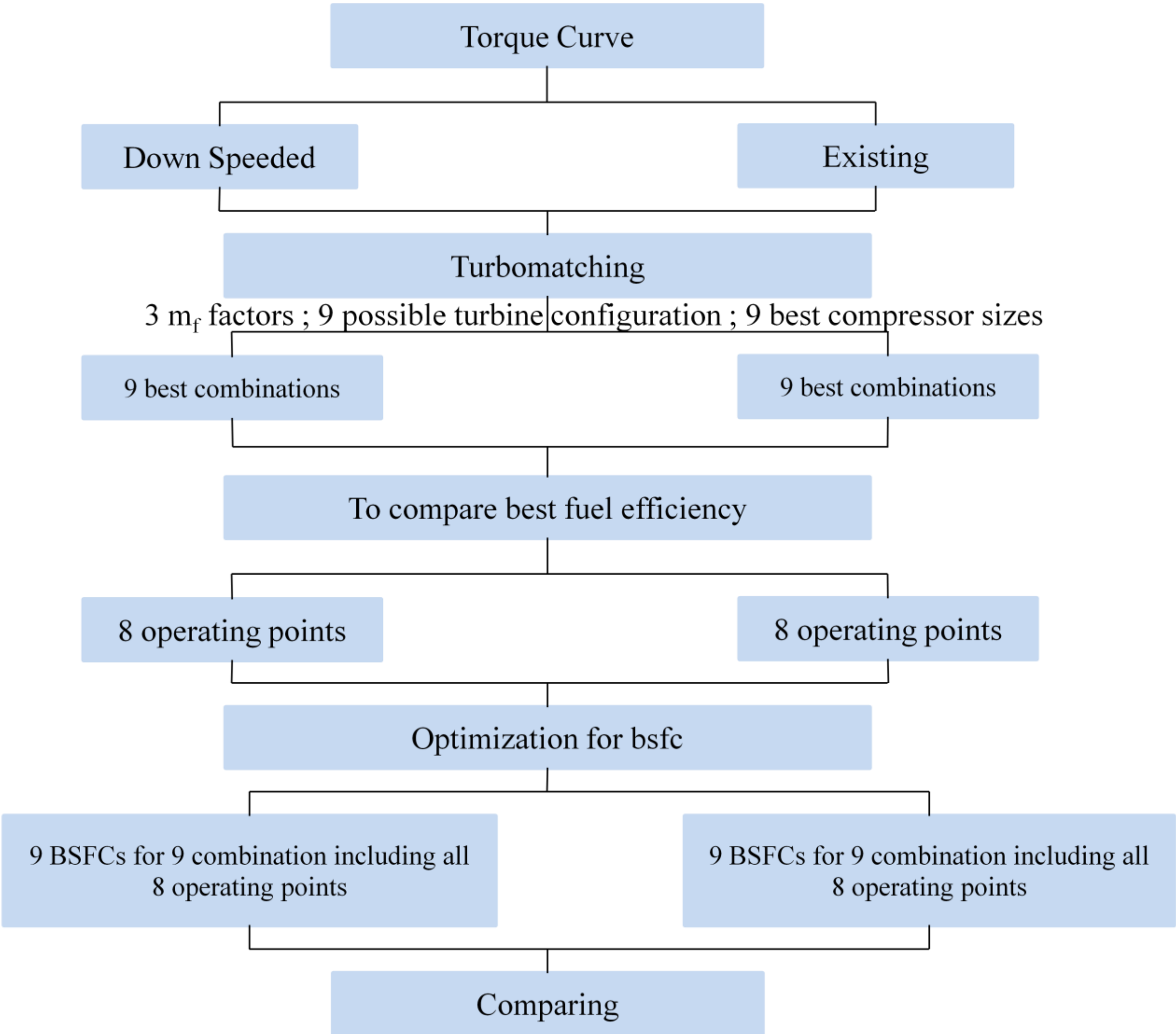

Figure 2. Methodology flow chart

6. Charge pressure

6.1 Introduction

Increasing the mean effective pressure of an engine operated with same fuel at lower engine speeds can only be achieved by either increasing volumetric efficiency, density or changing the A/F ratio substantially. The air-fuel ratio can be modified to operate the engine in rich conditions but the smoke arises, hence there is a smoke limit for increasing the F/A ratio for an engine. On the other hand increase in volumetric efficiency or inlet air density will not hamper any other factor.

$$Power = \frac{\eta_v * \eta_f * N * V_d * Q_{HV} * \rho_{a,i} * \frac{F}{A}}{2} \qquad (4)$$

Increasing volumetric efficiency and density is possible by forced induction. It is done to increase the charge density during the intake and it can be done by either supercharging or turbocharging. The advantage of supercharger over turbocharger is the absence of turbolag.

6.1.1 Supercharger

A supercharger uses a compressor driven by the crankshaft to compress the inlet air prior to the entry into the cylinder. It is commonly used in race cars or vehicles which require better throttle response. In many ways it is less efficient than a turbocharger.

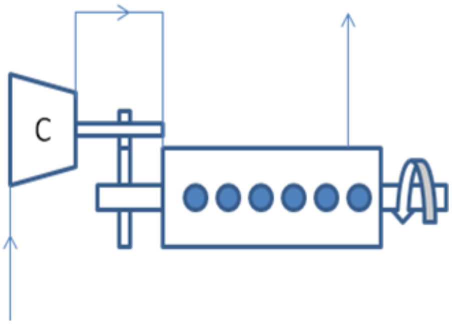

Figure 3. Supercharger

6.1.2 Turbocharger

In a turbocharger a turbine and compressor is connected in a single shaft and the turbine is driven by energy from the exhaust gas which in turn drives the compressor which increase the density of the inlet air. The advantage of using a turbocharger over supercharger is, it does not use the power from the engine crankshaft

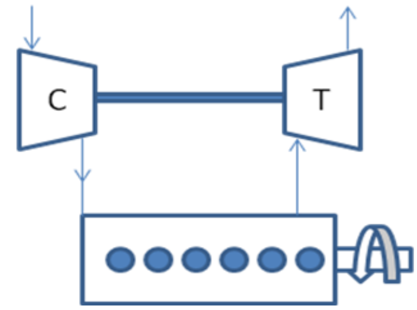

Figure 4. Turbocharger

6.1.3 Engine driven turbocharger (Turbocompound) and compressor

In this type an additional turbine is present in the exhaust downstream and it is connected to crankshaft by means of a set of gear wheels. It gives a better transient response than the conventional turbochargers and high boost pressure at low speeds. Similarly in engine driven compressor, an additional compressor is used in the system which is driven by engine. These types are usually used in marine engines.

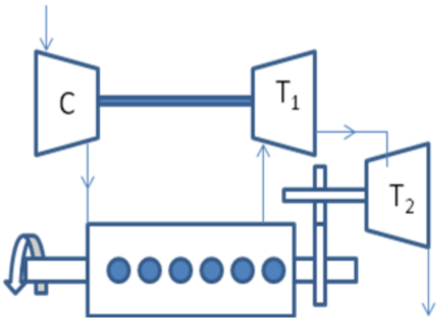

Figure 5. Turbocompound

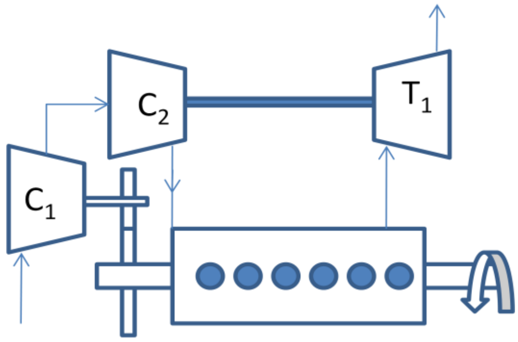

Figure 6. Engine driven compressor and turbocharger, part of engine power is given to the additional compressor C_1

6.1.4 Dual staged turbocharger

These types of turbochargers are used for creating very high boost pressures to get higher BMEP.

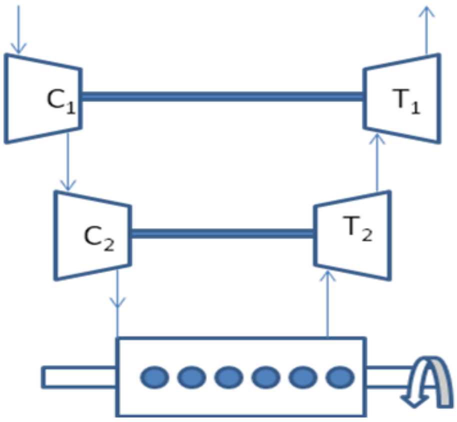

Figure 7. Dual staged turbocharger

6.1.5 Turbocharger with intercooler

This is the most common type found in the trucks. The temperature raise caused by the compression of air is extracted by using an intercooler before the inlet.

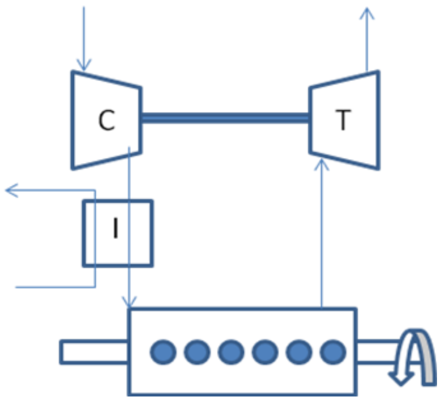

Figure 8. Turbocharger with intercooler.

6.1.6 Variable Geometry Turbocharger

With conventional turbine vanes the boost pressure cannot be varied for different speeds because of constant A/R (Aspect) ratio. The VGT's have movable vanes in the housing, which is used to control the boost pressure, when the exhaust pressure is low the vanes are partially closed so that the increasing exhaust pressure is used to rotate the turbine at higher speeds resulting in high boost. There are two types of VGT one is moving vanes type and another is sliding vane type (axially). Usually for large diesel engines the sliding vane type is used, in this type the vanes are slided axially (shown in Fig.9). The sliding ring varies the gap and thus the geometry varies, which is used to control the boost.

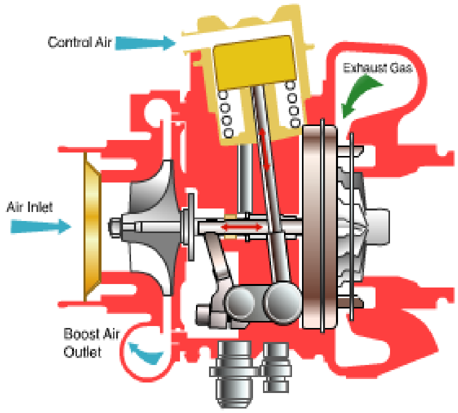

Figure 9. Sliding vane type VGT (Courtesy: Cummins Holset turbotechnologies)

6.2 Turbocharger chosen for this concept

Dual stage turbocharger with variable geometry HP turbine with intercooler and aftercooler was selected in this thesis work owing to its ability to produce higher pressures at low speeds[8].

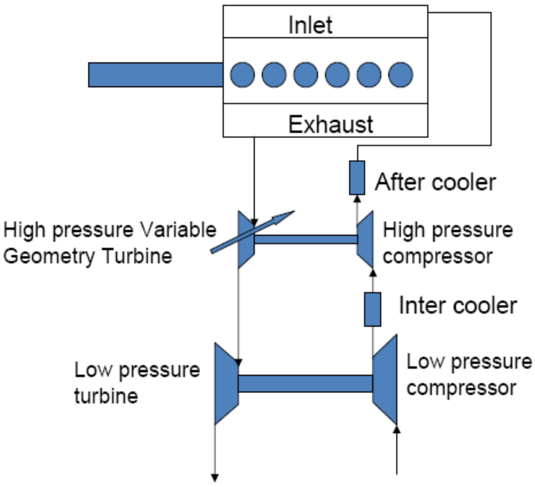

Figure 10. Dual stage turbocharger with intercooler and aftercooler also with Variable geomtry high pressure turbine

To find the appropriate size of the turbines and compressors, turbomatching was done.

6.3 Turbomatching

Turbomatching involves choosing the best size for compressors and turbines depending upon the engine mass airflow requirement. It is better to look for the high torque and low speed operation region since this is where the charging becomes difficult because of low compressor speeds and higher mass flow requirement (operation close to surge line as shown in Fig.13). Four different torque curves were presented by Volvo as shown in Fig.11. Option 1 shows the existing torque curve for the engine model and the remaining three options are proposed torque configurations, but due to the computational time only the curves, option1 and option 4 are studied for the work. Herein after proposed torque curve/proposed torque configuration refers to option4 curve in Fig.11 and existing torque curve/existing torque configuration refers to option 1 in the same figure.

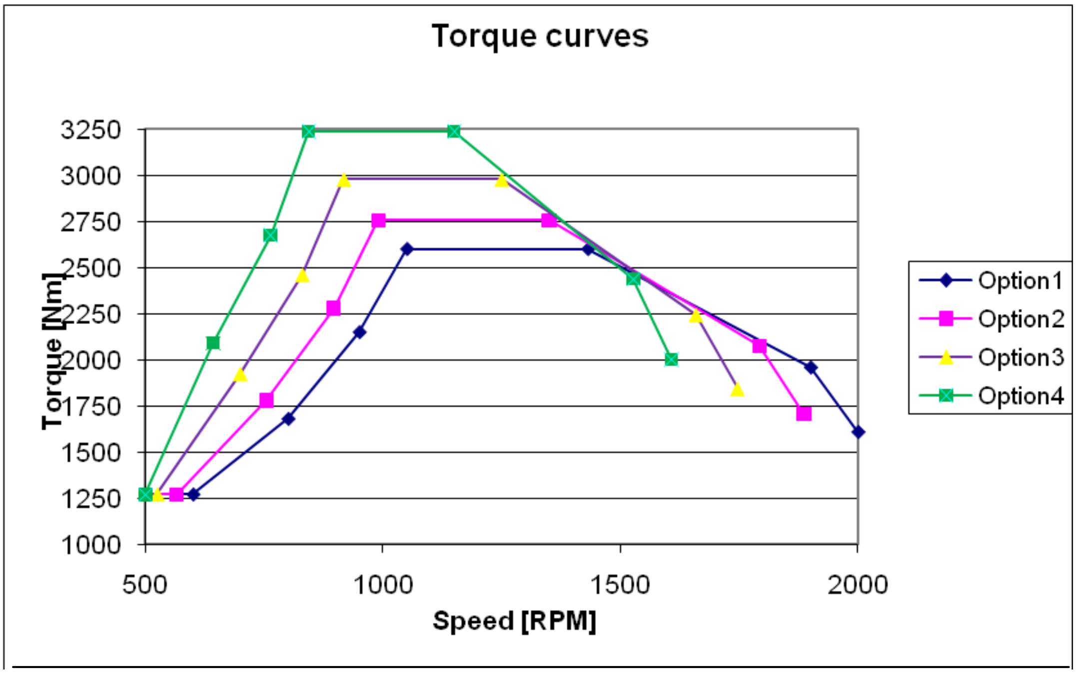

Figure 11. Torque curves presented by Volvo

For the low speed and high load operating point from the torque curve Fig.11 the turbine and compressor mass flow factors were selected. It was done using the design of experiments setup in GT-Power (full factorial design). Since we simulate using the same compressor and turbine maps the scaling of size can be done by changing the mass flow factor in the model. For the existing torque curve operating point, three variants of mass flow factors were selected 1, 1.15, 1.3 (3 levels, 9 combinations) for both high pressure and low pressure compressors and high pressure and low pressure turbines, with variable geometry turbine (VGT) position varying between 0.25 and 0.4 (4 levels), a total of 324 simulations were carried out to select the 9 best combination of mass flow factors of compressors.

Table 1. Full Factorial DOE setting for existing torque curve

Factors	Operating Range
VGT position	0.25,0.3,0.35,0.4
High pressure turbine	1,1.15,1.3
Low pressure turbine	1,1.15,1.3
High pressure compressor	1,1.15,1.3
Low pressure compressor	1,1.15,1.3

GT-Post and Matlab were used to analyze compressor efficiency plots and high pressure compressor data respectively. Firstly the reduced mass flow, pressure ratio, mass flow factors from simulated data were exported to Matlab and data which has a pressure ratio greater than 2.5 were eliminated. The efficient combination was selected by choosing a surge margin between 10% to 25% and the combination which had surge margins between these limits. If two or more combinations fell within this surge limits, the combination with lower pressure ratio was chosen as the best combination. Compressor maps from GT-Power was used to find the surge margin percentage, the surge points from the compressor map was chosen and then the surge points were multiplied by the chosen mass flow factors to form the scaled surge line (corrected mass flow). From the simulated data from GT post the reduced mass flow factor (actual mass flow) was chosen, and the corresponding value in scaled surge line was interpolated and found (mass flow surge). By comparing the actual mass flow and surge mass flow the surge margin could be found.

$$surge\ percentage = \frac{m_{actual}}{m_{surge}} \tag{5}$$

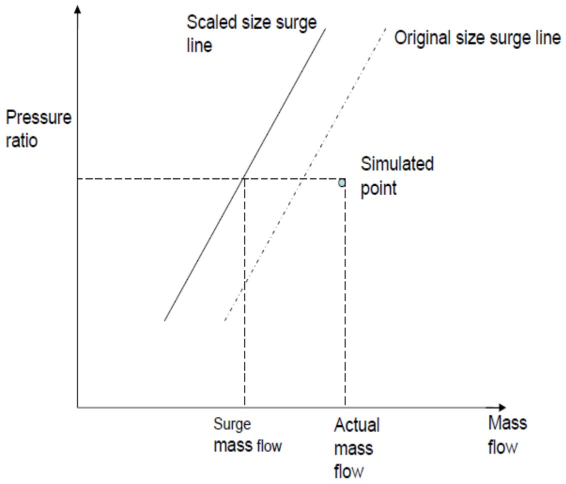

Figure 12. Calculating Surge margin by scaling SAE compressor map

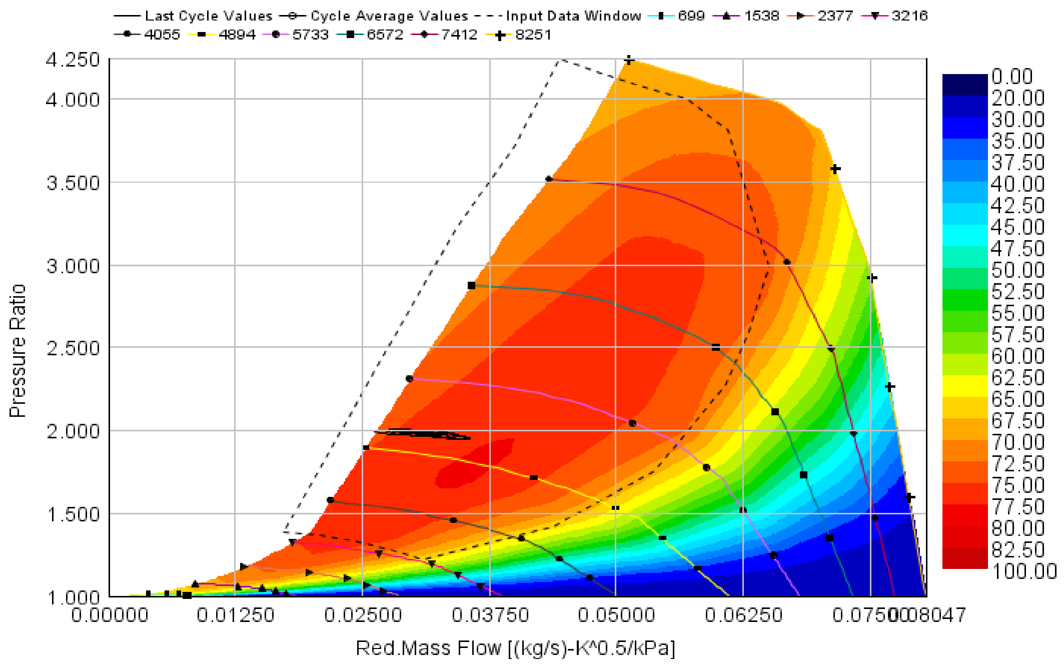

Figure 13. Compressor efficiency map model

Among the nine combinations some combination fell out of the surge margin limits and these were additionally simulated with extended variable geometry position from 0.4 to 0.55 and the same calculation method was carried out to fill the 9 combinations. The low pressure compressor efficiency map was also analysed in GT-Post and it was selected based on the efficiency with which it is operated. The combinations chosen were those which had the high pressure compressor surge margin between the prescribed limits and low pressure compressor in the efficient zone. The nine best combinations for Low torque curve are shown in Table.2.

Table 2. Nine best compressor configurations for existing torque curve

Combination	VGT position	Mass flow factor of HP turbine	Mass flow factor of LP turbine	Mass flow factor of HP Compressor	Mass flow factorof LP Compressor	Surge margin
1	0.4	1	1	1.15	1.15	1.19
2	0.4	1.15	1	1.15	1.15	1.24
3	0.4	1.3	1	1.3	1.15	1.13
4	0.4	1	1.15	1.15	1.15	1.20
5	0.4	1.15	1.15	1.3	1.15	1.10
6	0.4	1.3	1.15	1.3	1.15	1.12
7	0.55	1	1.3	1.3	1.15	1.11
8	0.4	1.15	1.3	1.3	1.15	1.10
9	0.4	1.3	1.3	1.3	1.15	1.11

For the proposed torque curve configuration similar methodology as that of the existing torque curve configurations is adapted and the combinations were found. The changes made for the proposed torque curve are the decreased mass flow factors for all the compressors and turbines 0.7, 0.85, 1 (3 levels, 9 combinations). The mass flow is reduced (small turbochargers) in order to get the required boost pressure at such low speeds.

Table 3. Full factorial DOE settings for existing torque curve

Factors	Operating Range
VGT position	0.25,0.3,0.35,0.4
High pressure turbine	1,1.15,1.3
Low pressure turbine	1,1.15,1.3
High pressure compressor	1,1.15,1.3
Low pressure compressor	1,1.15,1.3

The nine best combinations for the proposed torque curve are shown in Table.4.

Table 4. Nine best compressor combinations for proposed torque curve

Combination	VGT position	Mass flow factor of HP turbine	Mass flow factor of LP turbine	Mass flow factor of HP Compressor	Mass flow factorof LP Compressor	Surge margin
1	0.45	0.7	0.7	1	0.85	1.10764
2	0.45	0.7	0.85	1	0.85	1.11933
3	0.55	0.7	1	1	1	1.20344
4	0.4	0.85	0.7	1	0.85	1.15276
5	0.45	0.85	0.85	1	1	1.20902
6	0.45	0.85	1	1	0.85	1.22161
7	0.4	1	0.7	1	0.85	1.20476
8	0.4	1	0.85	1	0.85	1.23840
9	0.4	1	1	1	1	1.24600

7. Operating points for BSFC calculation

Out of the nine possible combinations, to find the best combination for fuel efficiency i.e. to find the combination which has the least brake specific fuel consumption, another set of design of experiments were simulated. Eight operating points with time weights and energy weights given by Volvo were considered for this work. The operating points are shown in Table.5.

Table 5. Operating points for existing torque curve configuration

Speed(rpm)	Torque(Nm)	Time Weight (%)	Energy Weight (%)
1032	2511	3.1	5.4
1200	650	35.8	18.8
1250	1821	10.9	16.7
1279	2601	10.3	23.0
1279	1301	13.1	14.6
1350	650	23.1	13.6
1466	1908	2.2	4.1
1526	2444	1.5	3.8

Proposed torque curve configuration

Table 6. Operating points for proposed torque curve configurations

Speed(rpm)	Torque(Nm)	Time Weight (%)	Energy Weight (%)
829	3126	3.1	5.4
964	809	35.8	18.8
1005	2265	10.9	16.7
1028	3236	10.3	23.0
1028	1619	13.1	14.6
1085	809	23.1	13.6
1178	2374	2.2	4.1
1226	3042	1.5	3.8

7.1 Design of experiments for BSFC optimization

Latin hypercube DoE (200 experiments) were carried out with four operating factors namely VGT position, injection pressure, start of injection, and exhaust gas recirculation level. For each operating point, all the nine combinations were simulated for both the torque curves.

Table 7. Total Simulations

Torque curve configuration	No. of operating points	No. of combinations	Experiments in latin hypercube	Total simulations
Proposed	8	9	200	14400
Existing	8	9	200	14400

Table 8. Factor settings in DOE setup

Factors	Min	Max
VGT position	0.25	0.95
Injection pressure (bar)	500	2300
Start of injection (CAD)	-12	5
EGR level	0.06	0.2

The compressor speed limit was calculated using the Eqn. 2 and the speed limit for high pressure and low pressure for both the torque curves (existing and proposed) are shown in Table.9.

Table 9. Compressor speed limit

Compressor	Mass flow factor	Diameter of the wheel	Sqrt of m_f*Dia of wheel	Max rpm limit
HP, proposed torque curve	0.7	77	64.4182	163618
	0.8		70.994	148463
	1		77	136883
LP, proposed torque curve	0.7	98	81.986	128557
	0.8		90.356	116649
	1		98	107551
HP, existing torque curve	1	77	77	136883
	1.15		82.544	127689
	1.3		87.78	120072
LP, existing torque curve	1	98	98	107551
	1.15		105.056	100327
	1.3		111.72	94343

Using DOE-Post the post process of the DoE runs were carried out. Simple response surface were created with quadratic model resolution setting. By using fit statistics plot which uses R-Square fitness, the fitness of the DoE runs was made sure to be above 90%. Using the GT-Power optimizer tool, the DoE runs were optimised for better fuel consumption. Constraints were set for the responses as shown in the following table.

Table 10. Responses and constraints for optimising

Responses	*Constraints*
HP compressor speed	from the compressor speed limit table
HP compressor temperature	Lower than 523K
LP compressor speed	from the compressor speed limit table
Exhaust temperature	Lower than 950 K
Exhaust pressure	Lower than 5.5 bar
Combustion pressure	Lower than 250 bar
NO_x	Lower than 5 g/kWh
EGR level	0.06 – 0.22 volume fraction
Lambda	Greater than 1.25

For the runs which have lower fitness than 90% the constraints were simulated with another 100 Latin Hypercube DoE, for the factors, the range of the values obtained in the previous runs were extended on both sides as presented in Table.11.

Table 11. Resimulated for better fitness

Factors	Range extending for factor values obtained from previous run
VGT position	+/- 0.20
Injection Pressure	+/- 400
EGR level	+/- 0.05
Start of Injection	+/- 3

After the DoE runs are optimized the best combination was calculated using the following equations

$$Energy, kWh = \frac{speed}{60} * 2 * \pi * torque * weight\ factor * \frac{3600}{1000} \tag{6}$$

$$BSFC\ incl. of\ urea = BSFC + 2.2 * urea\ cost\ to\ diesel * \frac{0.835}{1.091} * (BSNO_x - 0.32) \tag{7}$$

$$fuel\ consumption\ in\ gms = BSFC * kWh \tag{8}$$

$$NO_x\ in\ gms = BSNO_x * kWh \tag{9}$$

7.2 Turbine Shaft inertia calculation

For the newly sized turbocharger the inertia is calculated from the typical rotor inertia chart from Cummins Turbo technologies. The inertia calculated from the chart includes the turbine inertia and impeller inertia. The calculated values are as shown in Table.12.

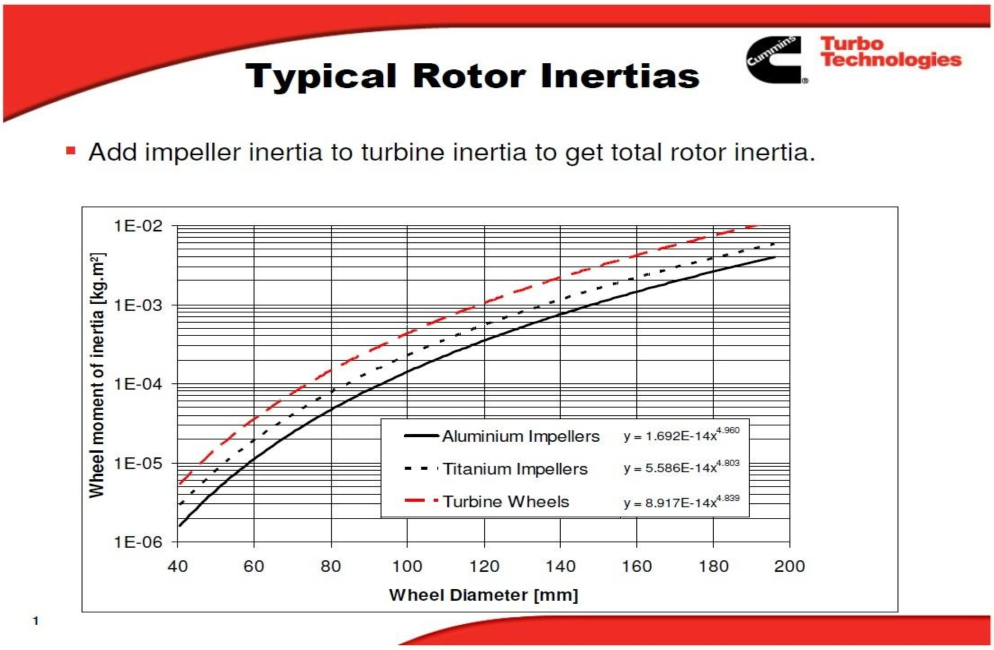

Figure 14. Typical rotor inertia chart [courtesy: Cummins turbo technologies]

Table 12. Moment of inertia calculated from the Cummins chart

	High pressure turbine shaft kg-m^2	Low pressure turbine shaft kg-m^2
Moment of inertia	9E-5	2.8E-4

8. Transient simulation

Transient simulation of any system is done to find how responsive the system is. In real time it is done to find the responsiveness and also to calculate pollutant formation during transient conditions as per the regulations. In this project work with two staged turbochargers, a transient response simulation was carried out to find out how responsive the turbocharger is. The VNO_x model in our engine is not designed to calculate the NO_x formation in transient conditions and hence the emission formation in the transient simulation is disregarded.

The transient simulation was done by simulating the engine at idling condition and then the engine is operated with full throttle to attain a specific torque (10% - 90%) according to Fig15.

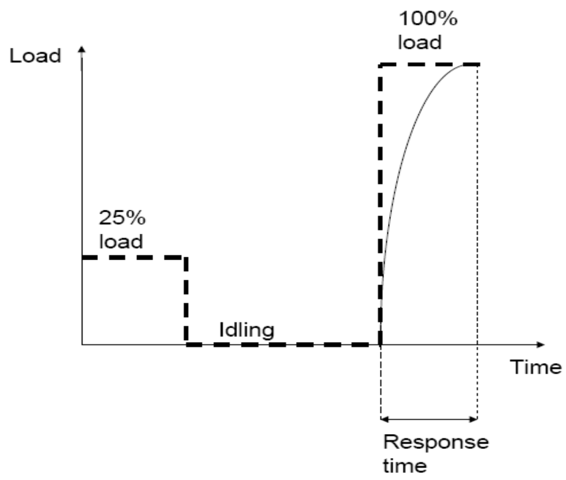

Figure 15. Transient simulation

To get a better response the exhaust gas rate is limited to 10% EGR (this is achieved by almost completely closing the EGR valve), thereby the exhaust gas is completely utilized by turbines and hence a better response will be achieved, care has to be taken since this may increase the NO_x production. Also to get a fast response it is necessary to control the VGT position, a PID controller was built for this task.

8.1 PID controller for VGT position

Proportional Integral Derivative controller is a closed loop control; it is used to maintain a target value of some quantity by controlling some input value to the controlled device[1]. The sensed value is fed as input to PID and the output from the controller is used to control some actuated device. Usually for industrial applications the derivative control is omitted in the PID controller.

Steps in building a PID controller

- Simulating the step function and plant response
- Calculation of the PID values

8.1.1 Simulating the step response and plant response

The equation of a PI controller is similar to first order linear response, a first order linear response is defined by the following reaction to the step input.

Table 13.Step input for system response

Time	VGT position
0	0.25
1	0.25
1.0001	0.95

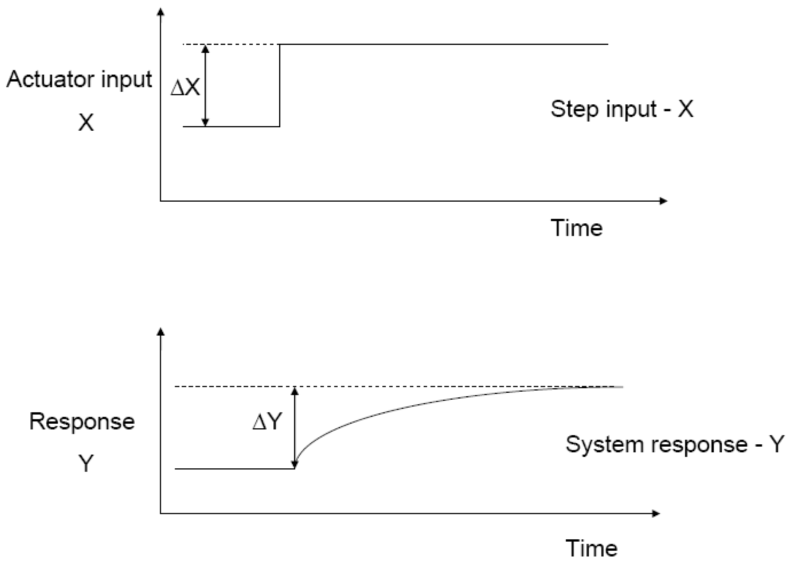

Figure 16.Step input and System response [Courtesy: Gamma technologies]

$K=\Delta Y/\Delta X$; Output ratio

The step value is chosen based on the plant responsiveness. The VGT position ranges from 0.25 to 0.95 (from maps in the model) hence the step is 0.25 to 0.95. For the Proposed torque the speed chosen was 1178 rpm with a maximum torque of 2374 Nm and for the existing torque curve the chosen speed was 1466 rpm and the maximum torque 1908 Nm.

Using the GT-Post the data was analysed and the necessary parameters were fed into the GT-SUITE PID template to find the PID values.

Table 14.PID values for both the existing and proposed configurations

	Proposed torque curve	Existing torque curve
Proportional gain	-0.0011877	-0.0011102
Integral gain	-0.0634475	-0.0694033
Derivative gain	0	0

Using the above PID values a PID controller was built. The load fraction signal generator, which is used to give the load input to the control system, is modified in the acceleration control for the transient load. The transient load is given as illustrated in Fig.15 using a time step of 0.05 sec. The 25% load point was simulated for a time period of 2 seconds and for another 15 seconds the engine was operated at idling condition and then the engine was operated at 100% load for the remaining time. During the simulation the engine was made to run at 25% load initially to initialize the simulations and then the engine was operated in idling condition for 15 seconds to bring down the speed effects of the turbine, according to Fig15.

9. Results and Discussions

9.1 Brake power

The requested power for both the proposed torque curve and existing torque are the same as shown in the graph below but the differences in the losses incurred by both the configurations vary the fuel efficiency.

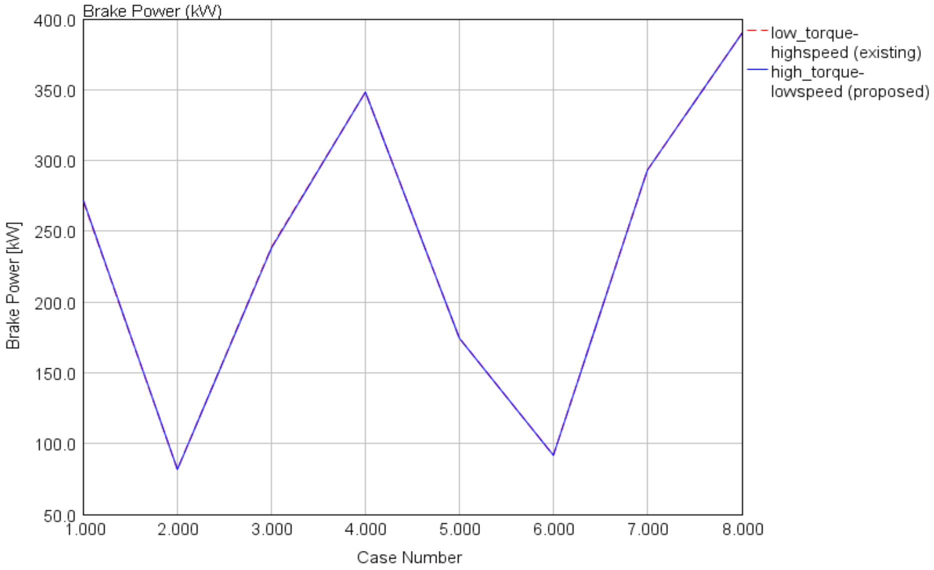

Figure 17. Brake power for all the operating points

9.2 Friction power

Not all the power produced by fuel converts into useful energy i.e. not all indicated power is converted into brake power some power is lost to friction. Friction power is a loss of useful energy. From the graph below it is seen that for the existing torque curve (high speeds) the friction power is higher than that of proposed torque curve. It is evident from the Fig.18, as the speeds become lower the friction power is also reduced as it is directly proportional to the mean piston speed.

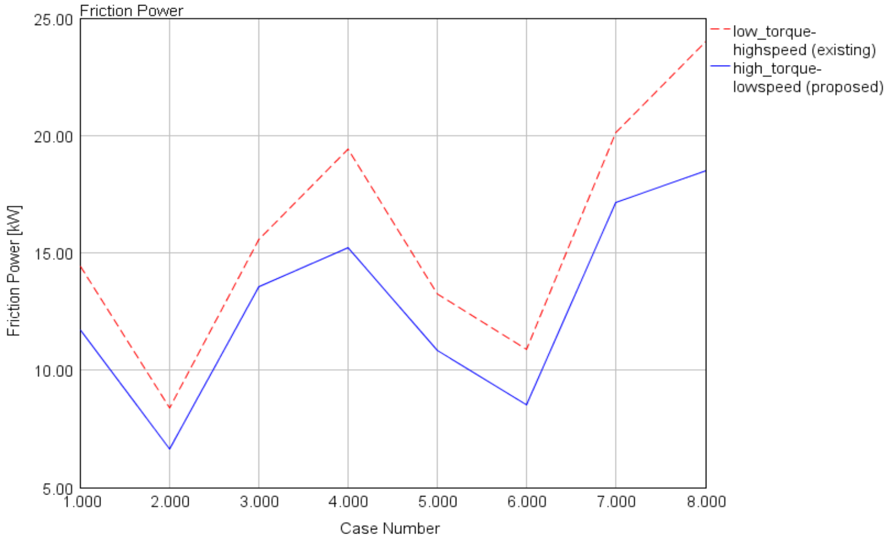

Figure 18. Friction power for all operating points

9.3 Attachment Power

Attachment power is consumed by the fuel system; there is a slight improvement in attachment power requirement for the proposed torque model over existing torque model.

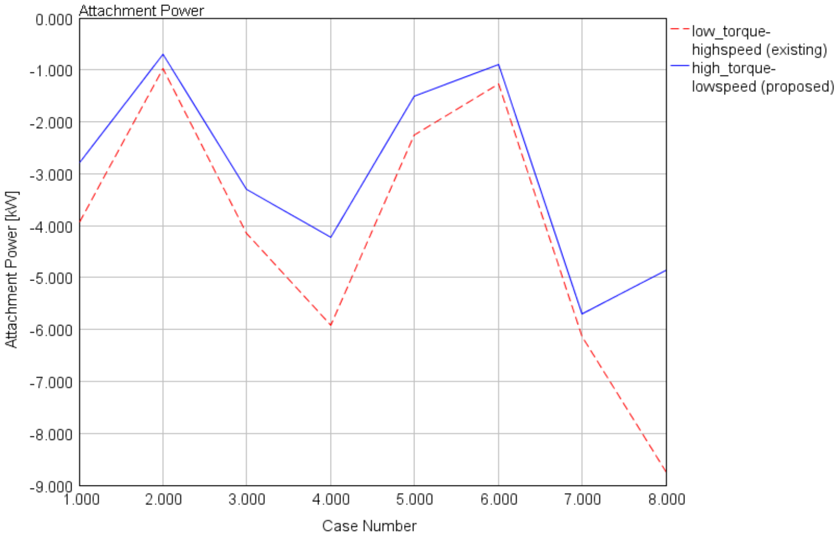

Figure 19. Attachment power for all operating points

9.4 Heat transfer

The heat transfer is a loss of useful energy. Heat transfer affects engine performance, efficiency and emissions. Higher heat transfer to the combustion chamber walls will reduce the average cylinder gas temperature and pressure and hence reduce the work transferred to piston[6]. It can be seen from the graph that there is a significant reduction in heat losses for the proposed torque curve over existing torque curve.

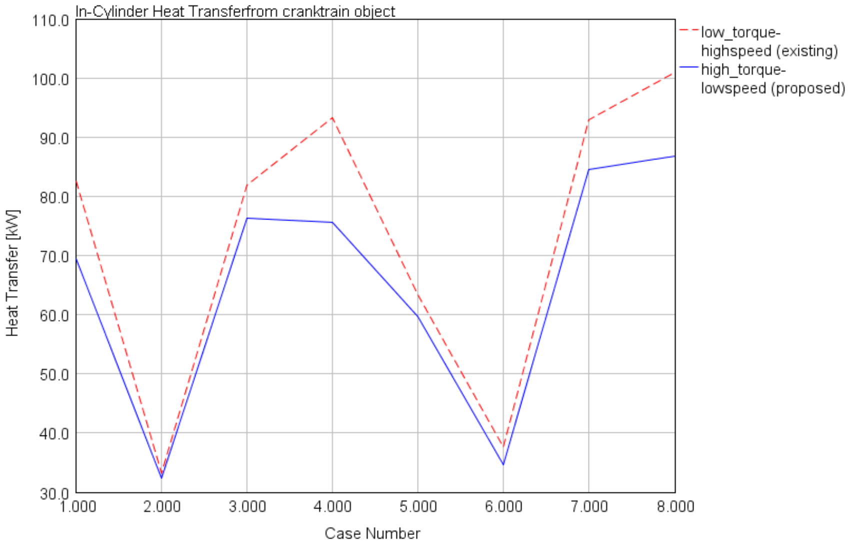

Figure 20. In cylinder Heat transfer for all the operating points

9.5 Brake mean effective pressure

BMEP can be used as the index for useful utilization of engine's volume. For the proposed torque curve there is a substantial increase in BMEP for all the operating points which is evident from the BMEP plot in Fig.21. Operating the engine with higher BMEP and lower speeds than that of the existing model proves the engine runs in the efficient zone in the BSFC map.

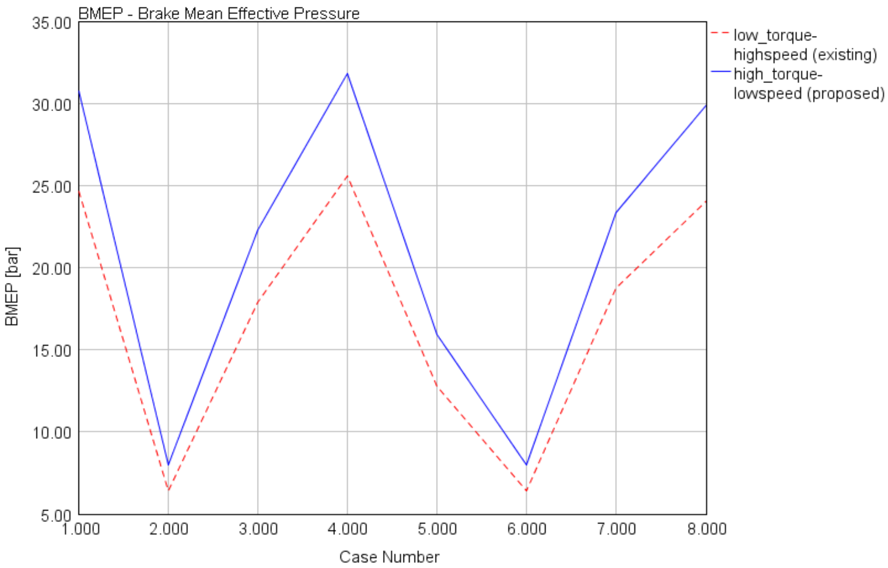

Figure 21. BMEP curves

9.6 Brake specific fuel consumption calculation

Using the formulae in Eqn (4, 5, 6, and 7) the BSFC was calculated for 25%, 50% and 100% urea cost to diesel cost. This is done in order to examine the changes in the total BSFC if the fuel cost is reduced or increased in the future. From the results presented below it is noted that only for 25% urea cost to diesel cost, which is likely to happen in future with rising diesel cost, the total BSFC for the existing torque curve is changed by 0.1 g/kWh. This is not a significant improvement and also this might be due to simulation error. For all other combinations irrespective of urea cost, the trend looks same.

Table 15. BSFC including cost of urea (50% to diesel cost)

Urea cost 50% to diesel cost		
Combination of turbine and compressor (table 2 and 4)	Existing	Proposed
1	Reference	-3.39%
2	+0.52%	-3.36%
3	+1.13%	-2.71%
4	+0.65%	-3.27%
5	+1.27%	-3.01%
6	+1.62%	-2.52%
7	+1.32%	-3.06%
8	+1.76%	-2.63%
9	+2.16%	-2.19%

Table 16. BSFC including cost of urea (25% to diesel cost)

Urea cost 25% to diesel cost		
Combination of turbine and compressor (table 2 and 4)	Existing	Proposed
1	+0.29%	-3.64%
2	+0.26%	-3.61%
3	+0.75%	-2.98%
4	+0.40%	-3.57%
5	+0.95%	-3.36%
6	+1.30%	-2.84%
7	+0.95%	-3.41%
8	+1.40%	-3.00%
9	+1.71%	-2.61%

Table 17. BSFC including cost of urea (equal to diesel cost)

Urea cost 100% (equal) to diesel cost		
Combination of turbine and compressor (table 2 and 4)	Existing	Proposed
1	+1.94%	-1.65%
2	+2.23%	-1.62%
3	+2.61%	-1.02%
4	+2.38%	-1.65%
5	+2.87%	-1.48%
6	+3.24%	-0.92%
7	+2.82%	-1.54%
8	+3.28%	-1.13%
9	+3.49%	-0.81%

9.6.1 BSFC comparison

The optimizations during the simulations were carried out with 50% urea cost, so the 50% urea cost calculated BSFC comparison is done below. From Fig.22, it is clear that the proposed torque curve which is the downspeeded engine has better BSFC for all operating points. The combination with the best BSFC is combination 1 in both the cases. By comparing both the torque configurations for the combination 1, there is a BSFC improvement of 3.38%.

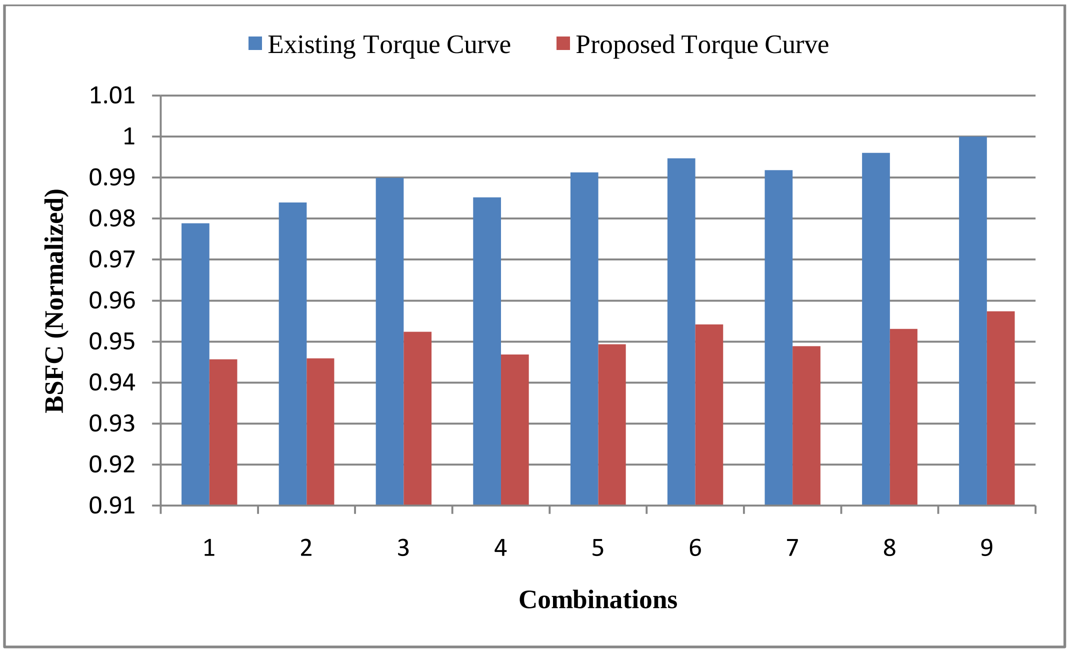

Figure 22. BSFC comparison

9.7 Trend Analysis

Since the optimization is done for 50% urea cost, the trend analysis is done only for BSFC calculated with 50% urea cost to diesel cost. The trend analysis is done to find whether reducing the size of the turbine further influence the BSFC. The 3-D plot for different combinations of turbine and the corresponding BSFC's were plotted.

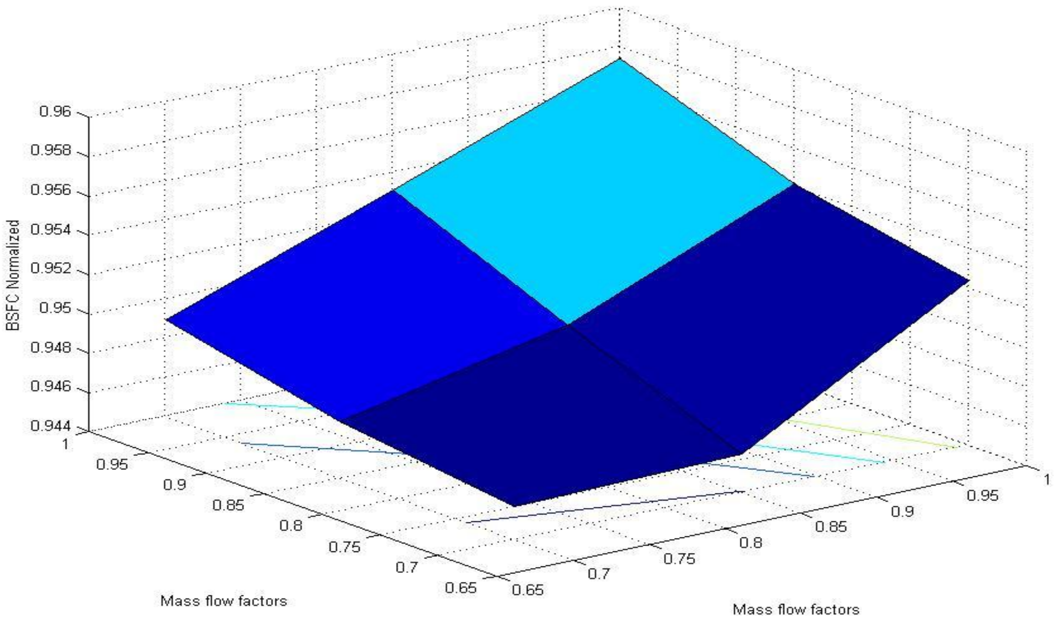

Figure 23. Surface plot for trend analysis on turbine size for existing torque configuration

The 3-D plot for the proposed torque configuration shows, reducing the size of the turbine further will not affect the BSFC substantially.

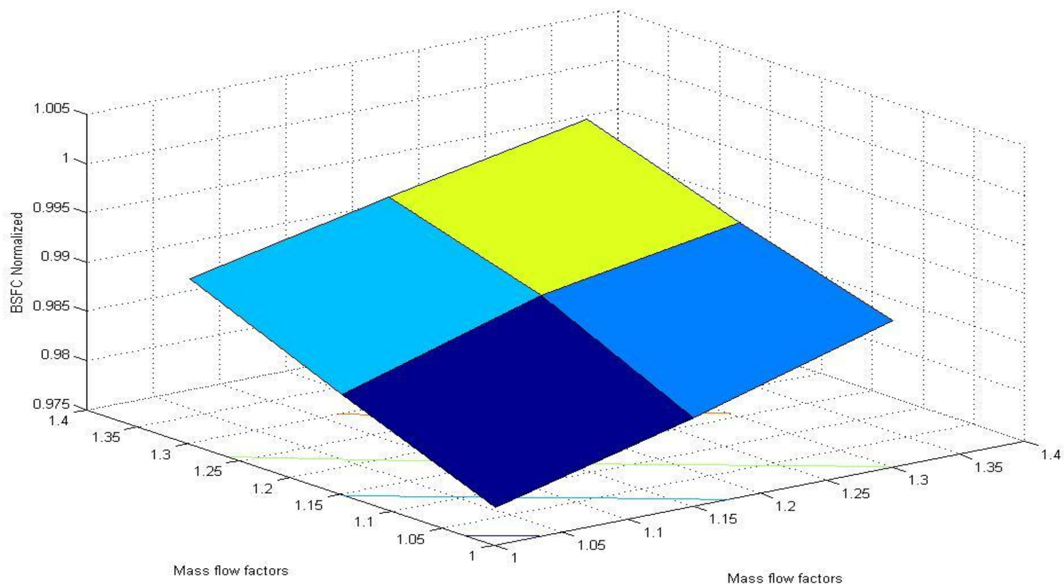

Figure 24. Surface plot for trend analysis on turbine size for proposed torque configuration

The plot in Fig.24 (surf plot for the existing torque curve) shows that reducing the size of the turbine further down might reduce BSFC to an extent.

9.8 Exhaust power

Exhaust temperature is an important parameter for EATS (Exhaust After Treatment System) systems, both the studied torque curves were analysed for exhaust power mass averaged specific heat and average mass flow rate using the exhaust temperature just before the EATS. The downspeeded torque curve, because of the efficient use of fuel energy, has lower exhaust power than the existing torque configuration; it can be seen from the table 18 and 19. But both torque configurations has the exhaust temperature required for the efficient operation of EATS.

Table 18. Exhaust power calculation for existing torque configuration

Existing torque configuration								
	1	2	3	4	5	6	7	8
Mass Averaged Temperature (Outlet)	725.4	607.7	661.6	715.0	614.0	570.6	685.5	697.7
mass averaged specific heat	1177.1	1124.6	1152.3	1172.6	1133.6	1113.2	1159.5	1166.9
average mass flow rate	279.4	130.8	283.9	368.6	235.0	155.6	346.8	434.4
exhaust power in kW	238.6	89.4	216.4	309.1	163.6	98.8	275.7	353.7

Table 19. Exhaust power calculation for proposed torque configuration

proposed torque configuration								
	1	2	3	4	5	6	7	8
Mass Averaged Temperature (Outlet)	667.9	600.8	638.6	677.2	589.95	552.2	618.3	693.7
mass averaged specific heat	1153.9	1124.2	1149.3	1160.8	1127.2	1111.9	1139.3	1163.9
average mass flow rate	301.0	121.7	259.7	373.7	226.7	136.8	347.7	456.2
exhaust power in kW	231.9	82.2	190.6	293.7	150.7	84.1	245.0	368.3

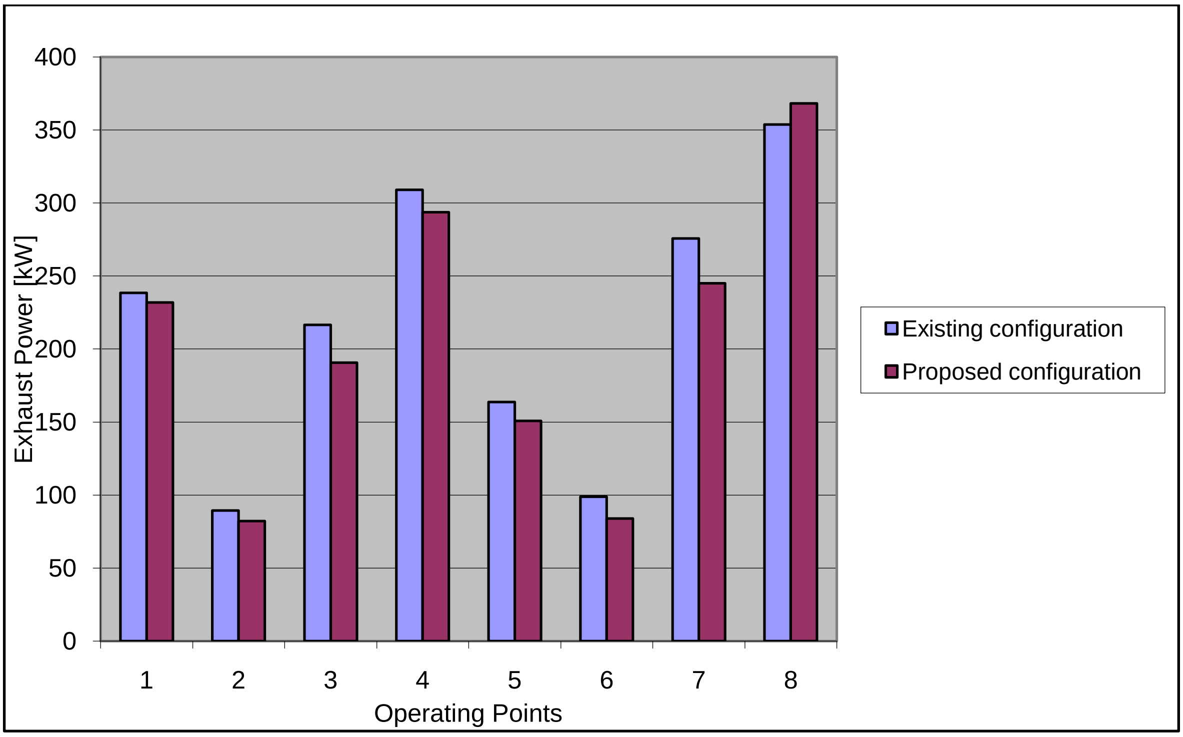

Figure 25. Comparison between exhaust power for both configurations

9.9 Transient response results

From the transient response results it is clearly seen that it is advantageous to run the engine at higher speeds (existing torque configuration) rather than the downspeeded configuration from the perspective of turbocharger response time. At higher engine speed it is easier to rev up the turbocharger and it is evident from the following plots. Response analysis is done for two torque points in each torque configurations and the results are presented in the table 20. Response time is calculated from 10% to 90% full torque as shown in Fig.26

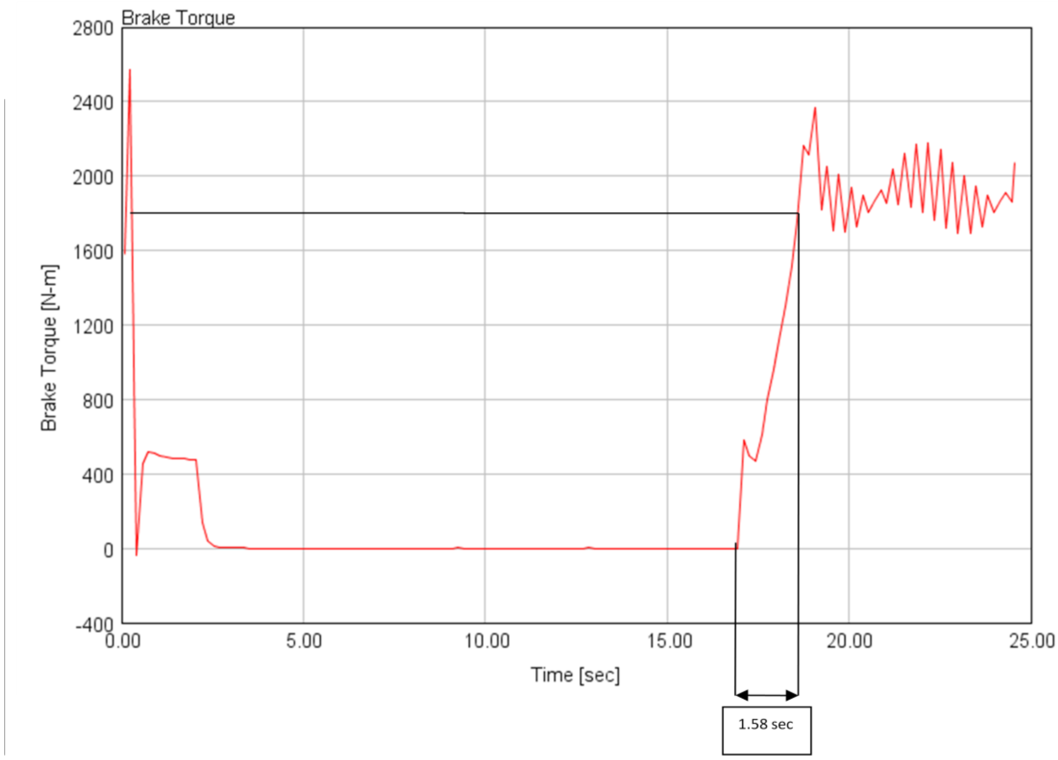

Figure 26. Transient response with 1908 Nm torque for existing torque configuration

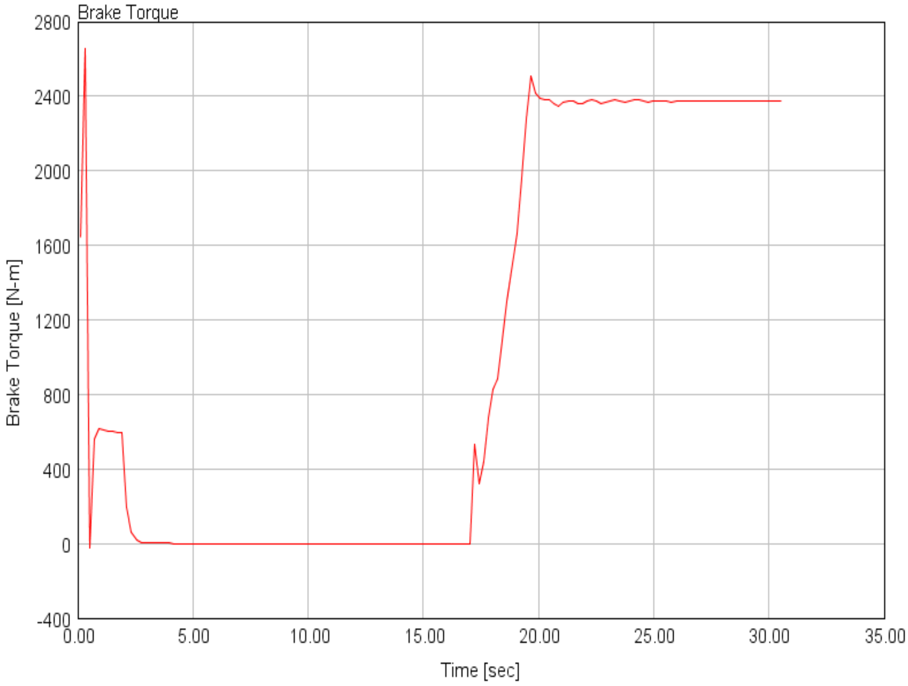

Figure 27. Transient response with 2374 Nm for proposed torque configuration

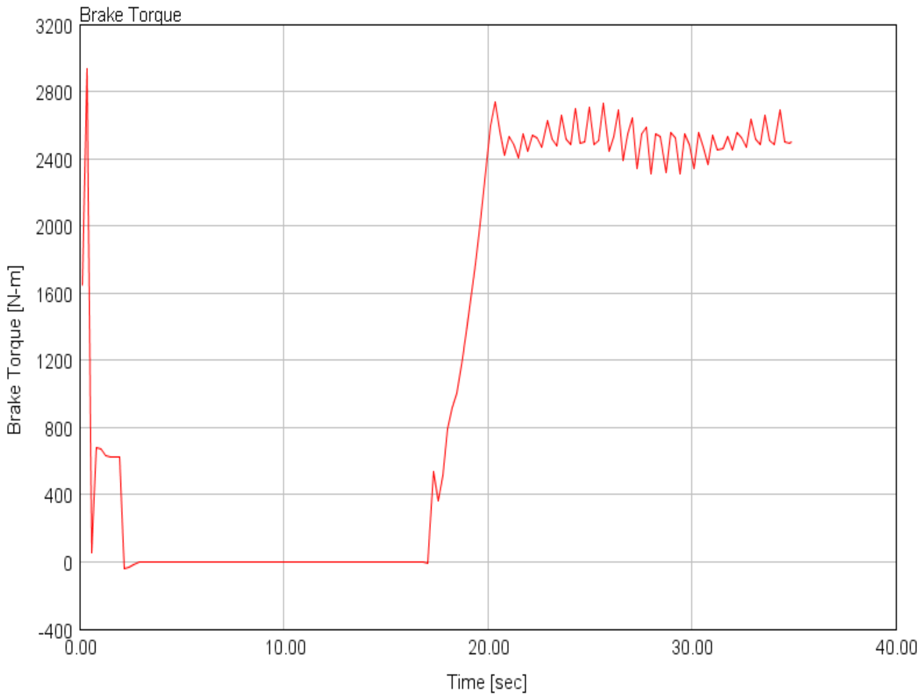

Figure 28. Transient response with 2511 Nm for existing torque configuration

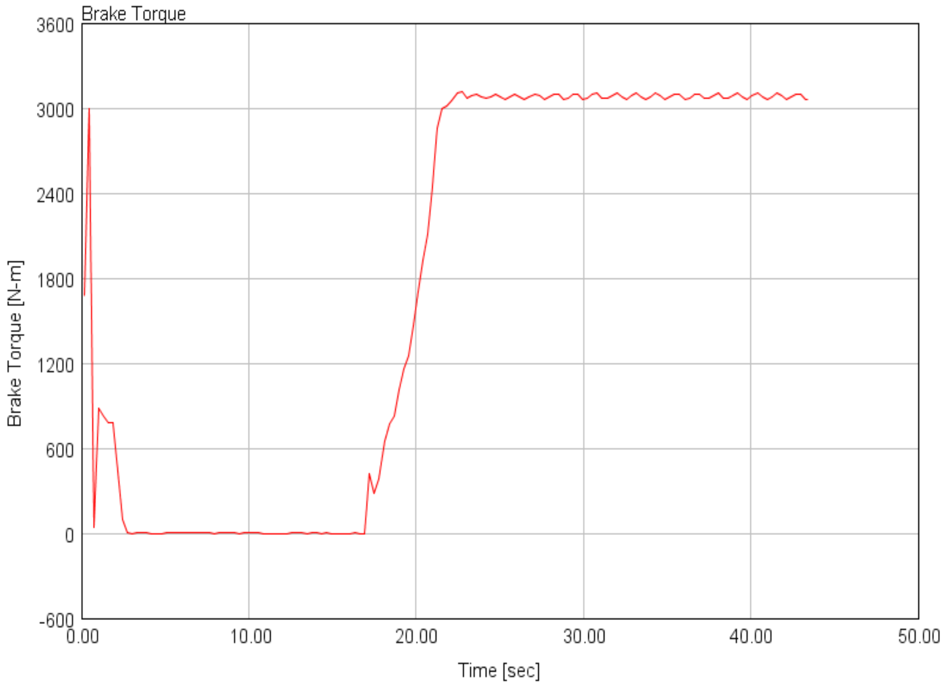

Figure 29. Transient response with 3126 Nm for proposed torque configuration

Table 20. Transient response results

Max torque in Nm	Engine speed in rpm	Power in kW	Existing torque curve configuration-response time in sec	Proposed torque curve configuration-response time in sec
1908	1466	293	1.58	
2374	1178	293		2.35
2511	1032	272	2.88	
3126	829	272		4.2

10. Conclusions

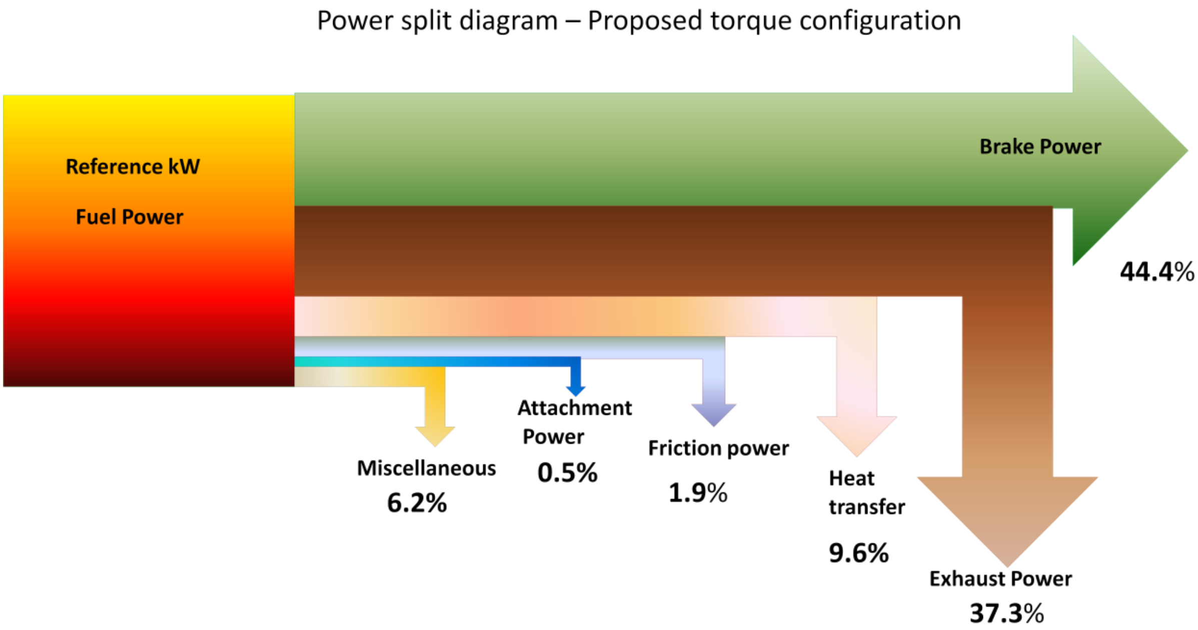

Figure 30. Power split for proposed torque configuration

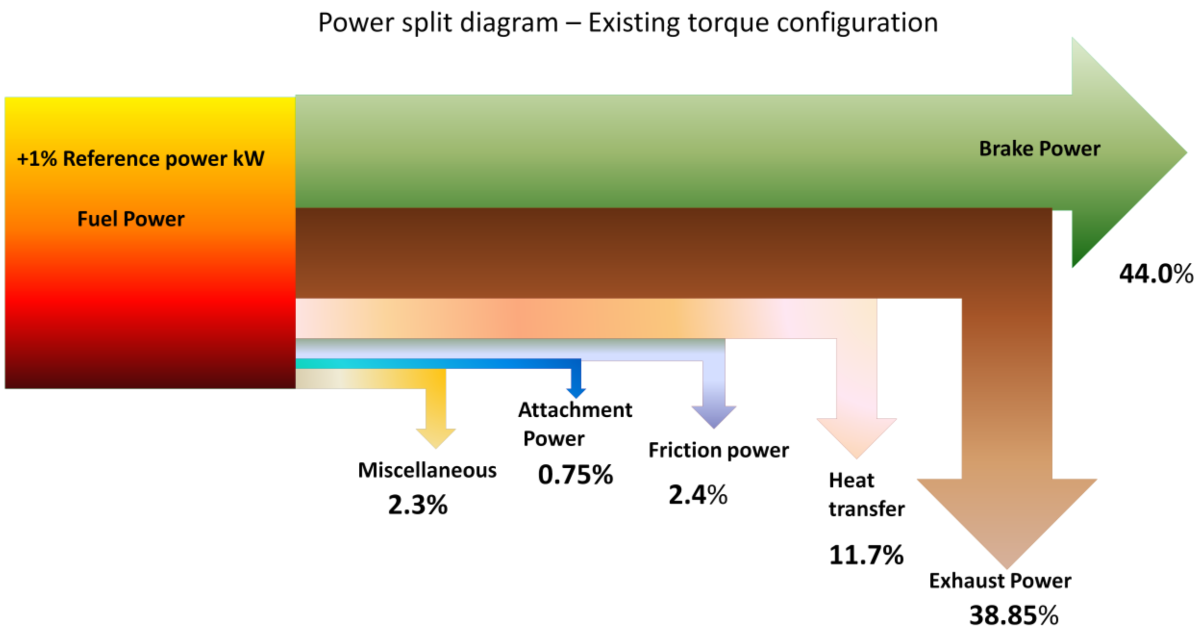

Figure 31. Power split for existing torque configuration

The results show that downspeeding proves to be efficient. There is a significant fuel economy improvement of 3.38% by switching from the existing engine operation to downspeeded engine operation. The fuel economy improvement is a result of three main factors suggested in the hypothesis, friction reduction, reduced relative heat transfer and increased fuel conversion efficiency (faster combustion in CAD). It can be seen from the results there is a substantial decrease in frictional power loss and heat transfer power loss.

From the transient analysis it is seen that the dual stage turbocharger for the existing configuration has shorter response time than the downspeeded configuration engine. This is because the massflow across the turbine is low for the downspeeded engine.

As already mentioned the validation of the concept has not been able to carry out in this thesis work since it requires a new design of engine/engine auxiliaries altogether.

11. Future work

The turbine and compressor sizes are scaled sizes, and to have a better pragmatic comparison the maps for different turbine and compressor sizes from manufacturer has to be employed in the simulation work. Depending upon the efficiency of turbines and compressors the whole efficiency could be changed.

The engine speed can further be lowered down for part load operations and could be analysed for how the engine behaves at such speeds.

Advanced combustion concepts like HCCI and PCCI could be simulated for increasing the combustion efficiency of the engine.

12. References

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