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EFFECT OF COMPRESSION RATIO AND EGR ON DIESEL ENGINE HEAT BALANCE AND PARASITIC LOSSES USING 1D SIMULATION TOOL

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ABSTRACT
This research focused on understanding effect of compression ratio and EGR on diesel engine heat balance and parasitic losses using 1D simulation tool. The first law of thermodynamics was applied to 3.0 Liter, 55 kW turbocharged after cooled diesel engine with high pressure EGR system. The engine was operated under steady state conditions and the data was collected for 8 mode cycle. Heat balance sheet was calculated for rated power and rated torque conditions. The heat distribution to different paths was found to be varying with engine speed and load. Brake power and heat lost to exhaust showed opposite trends at different speeds. Engine was modeled using 1D SIMULATION TOOL software. For validation various parameters like volumetric efficiency, torque, power, BSFC, mass flow rate of air, pressures, temperatures, peak firing pressure and air-fuel ratio were considered. The model was validated for less than 10% error at both the operating points. Further the heat distribution was estimated by simulation for various compression ratios and EGR at rated power and rated torque for full load conditions.

Keywords:
Diesel engine, Compression ratio, EGR, Heat balance, 1D simulation

INTRODUCTION

Diesel engines are compression ignition engines. A higher compression ratio is required so that charge is ignited. A very high C.R. leads to higher thermal stresses whereas lower value leads to non-ignition. However, varying the compression ratio in suitable range can help determine the optimum value which leads to better combustion and hence thermal efficiency. Heat balance sheet helps determine the heat distribution in the engine. Engine modelling will aid in determining the optimum value of compression ratio and also to study the response of compression ratio over the heat distribution. 1-D simulation tools help in predicting the proportion of various primary paths by varying the two parameters in faster way.

Energy Balance

Diesel engines convert fuel chemical energy into useful work. In so doing they also reject a significant amount of heat to the environment. The amount and proportions of heat rejected via the many paths will vary depending on engine speed and load and on the operating environment. The determination of these heat fluxes is generally referred to as either "Heat Balance Analysis" or "Heat Path Analysis". Application of the First Law to engine heat balance studies has been carried out for decades and is now an established tool in engine research and development. The basic concept of heat balance is simple. The First Law of Thermodynamics is applied to an imaginary control volume that surrounds the engine. The control volume is treated as an open system because there is transfer of mass as well as heat and work across its boundaries. The First Law is applied to the control volume for the special case where the engine is operating at steady speed and load and has achieved thermodynamic equilibrium with its environment.
The second law of thermodynamics states that it is impossible for a machine to convert all the heat energy supplied to it into work. This means energy must be rejected in order to obtain work. Heat balance in automotive engines is used to verify the thermal efficiency and the heat distribution of the engine.
The purpose of heat path analysis is to quantify the various contributions shown in the figure 1.

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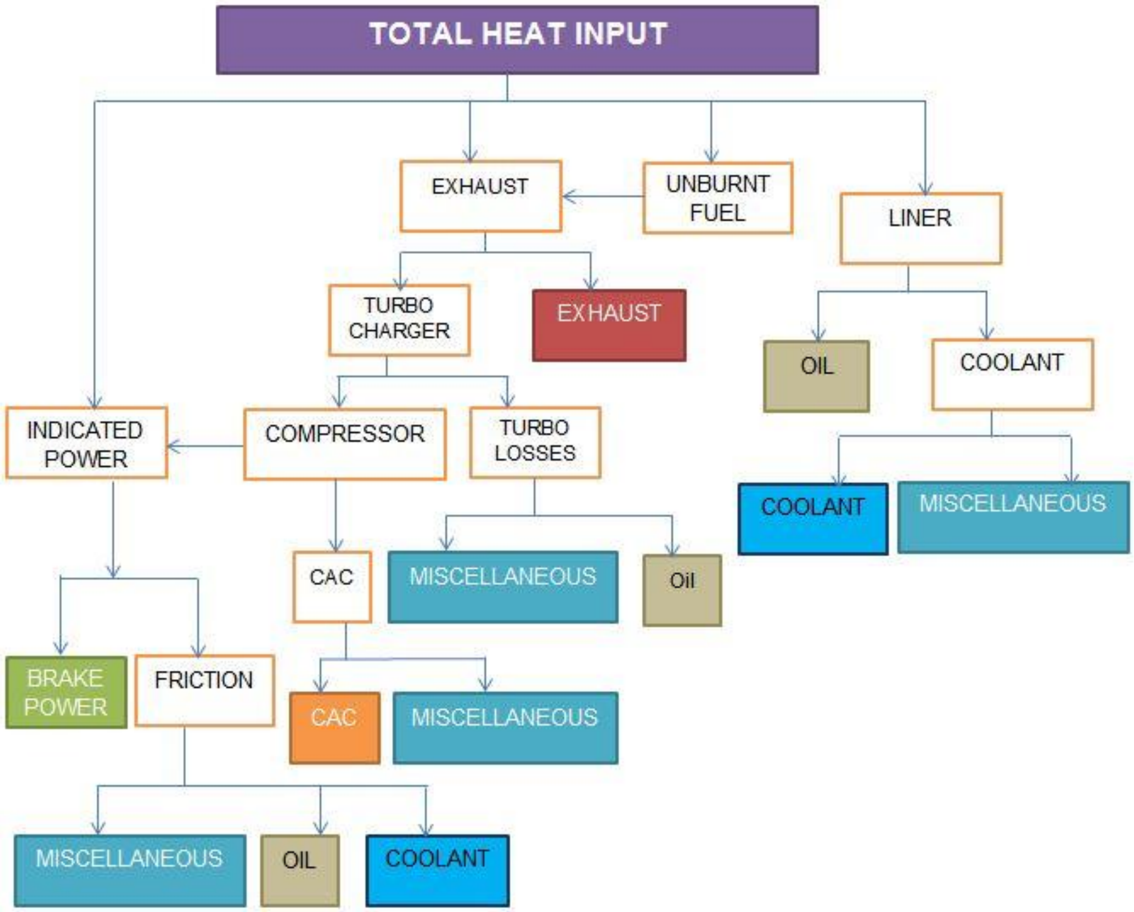

Fig. 1: Heat Distribution Schematic

Figure 1 shows a schematic of the heat balance concept for a turbocharged diesel engine. Note that the proportions displayed in this figure1 are NOT intended to be representative of actual values, and have been chosen for illustrative purposes only. The figure 1 shows on the left hand side the fuel chemical energy input, and the conversion from one form of energy to another is represented by the process arrows moving left to right. Initially, fuel heat liberated via combustion is converted directly into indicated work in the engine cylinders or rejected as heat to the oil and coolant circuits via conduction through the liner surfaces. Remaining direct terms are the exhaust gas enthalpy, and that fraction of the fuel energy that remains chemically bound in the fuel. For diesel engines this term is generally small and of the order of 1% of the total fuel energy. The turbocharger turbine extracts useful work from the exhaust gas by an expansion process. Some of this work is dissipated as mechanical losses in the bearings and then appears as heat to the oil. The turbine volute also rejects heat directly to the surrounding external environment. A fraction of the compressor power appears as charge heating, and some of the compressor work assists in reducing engine pumping work, and hence appears as indicated engine power. There is some small heat rejection from the connecting pipe surfaces but most of the compression heat is rejected via the charge cooler. A large fraction of the indicated work survives as brake shaft work whilst the remainder is dissipated by mechanical friction as heat to oil, coolant, or external surfaces, or is used to drive ancillaries such as the fuel injection pump, water pump, and oil pump.

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1-D THERMODYNAMIC SIMULATION TOOL

Overview

Engine cycle simulation codes are widely accepted tool today for the steady state performance prediction of engines at the design stage or for the analysis of the thermodynamic process of existing engines.
1D SIMULATION TOOL simulates a wide variety of engines, 4-stroke or 2-stroke, spark or auto-ignited. Applications range from small capacity engines for motorcycles or industrial purposes up to large engines for marine propulsion. 1D SIMULATION TOOL can also be used to simulate the characteristics of pneumatic systems.
The 1D SIMULATION TOOL program package consists of an interactive pre-processor which assists with the preparation of the input data for the main calculation program. Results analysis is supported by an interactive post-processor.
The pre-processing tool of the AVL Workspace Graphical User Interface features a model editor and a guided input of the required data. The calculation model of the engine is designed by software is MATLAB compatible. The software can give very accurate results, provided the input data fed is just as accurate and assumptions made are fair. Even if much of the input data is not known the software's help browser gives the standard input values for most of the parameters. A large number of combustion models have been included, to get fairly accurate results if the exact rates of heat release curves are not known.

Limitations

Many input parameters are generally asked for in every model. Most of these cannot be measured directly and involve lot of experimental work (e.g. port swirl ratios, injector discharge coefficients). To truly exploit the potential of this software, a thorough understanding of each and every parameter that the software requires, is essential which involves a lot of reading, particularly of the user's guide. The software has no feature to pin point the exact location, number of any errors which may arise when the model is simulated. This makes it very tedious to find where the model is malfunctioning. It also has no feature which suggests ways of avoiding those errors. The optimization tool has to be purchased separately. Manually optimizing the engine performance by changing the parameters for every case is very cumbersome.

Inputs (Overview)

As soon as the laying out the basic structure of the model is completed, every element has to be fed with some input data. Geometrical Inputs: Diameters, lengths, bending radii, volumes are some of the geometrical inputs required. Thermodynamic Properties: Pressures, temperatures, mass flow rates, target pressure drops and other thermodynamic properties are generally asked for to set a reference for the 1st cycle of calculation. Physical Inputs: Flow coefficients, correction factors, friction coefficients. Crank Angle Dependent Data: Valve lift curves, swirl ratio curves, Rate of heat release curves, rate of injection curves, engine friction curves.

- Fuel: Type of fuel used should be specified and the calorific values, stoichiometric ratio values are generated automatically.
- Firing Order: For the combustion data, we have to specify the order on which cylinder should fire.

Reference Conditions: The reference conditions are required in order to calculate specific engine performance data such as delivery ratio, volumetric data such as delivery ratio, volume efficiency, etc. related to ambient conditions.

LITERATURE SURVEY

L. A. Smith et al. [1] applied first law heat balance method to a turbocharged automotive diesel Engine. The difficulty of calculating heat lost to environment via convection and radiation through surfaces and ancillaries was overcome by an ingenious technique developed by authors wherein they encapsulated the engine within a wooden box which was hermetically sealed from the bottom and bright steel reflective inner walls. Air was forced through the encapsulation using large capacity constant speed fan and the difference in enthalpies at the inlet and outlet was measured which represented the heat lost to surroundings via external surfaces. The proportion of brake power and exhaust energy showed opposite trends at rated torque and rated speed conditions for all loads. Friction was primarily a function of speed with little dependencies on load.

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The portion of heat lost to external surfaces was found to be proportional to load and heat lost to charge cooler had higher proportion at rated power speed than at rated torque speed.

Qianfan Xin [2] performed theoretical analysis of internal combustion engine miscellaneous heat losses. They found that engine coolant heat rejection is dependent on many design parameters such as in-cylinder swirl ratio and turbulence level, cylinder heat transfer area, fuel injection timing, EGR rate and air-to –fuel ratio. The author has emphasized on the conducting gas side measurement as against the coolant side measurements which lead to large measurement errors. They had developed some empirical formulae for calculation of miscellaneous losses. Miscellaneous heat losses were found to be a function of characteristic gas temperature, sink temperature, engine speed and load.

Christian Donn et al. [3] carried out extensive analysis to study the influence of the operating point and operating parameters like EGR rate, injection strategy and coolant temperature on the engine energy balance. Brake thermal efficiency and charge air cooler heat flux were found to be proportional to load. Also the 1D Simulation Tool pressure had a significant influence on the charge air cooling power. EGR leads to a significant reduction in exhaust gas enthalpy flux and turbocharger load with it. The effect of EGR on exhaust gas enthalpy flux in combination with higher speed and load leads to a significant increase of exhaust gas enthalpy proportion for different operating points with and without EGR. The proportional coolant heat flux changes in the opposite direction and decreases from low load and speed with EGR to operating point with high speed, medium load and without EGR. Engine surface temperatures were found to be a function of oil and exhaust gas temperatures. It was also concluded that the heat rejection via coolant and surfaces was dominant at low speed and loads.

N. Ravi Kumar et al. [4] Studied effects of Compression Ratio and EGR on Performance, Combustion and Emissions of DI Diesel Engine. They found that with increase in compression ratio the brake thermal efficiency increases and specific fuel consumption decreases. They also observed that the brake thermal efficiency increased with amount of EGR. They noted a drop in combustion duration with increase in compression ratio due to less ignition delay. The amount of NOx produced reduced significantly with the use of EGR whereas the smoke opacity was observed to be reducing with increase in load at all compression ratios.

Avinash kumar et al. [5] studied the effect of EGR on exhaust gas temperature and exhaust opacity in CI engines. They found that that the exhaust gas temperatures reduce drastically by employing EGR. Thermal efficiency and brake specific fuel consumption are not affected significantly by EGR. However particulate matter emission in the exhaust increases, as evident from smoke opacity observations.

M. M. Abou Al-Sood et al. [6] conducted an optimization analysis on a thermodynamic simulation model to seek optimum compression ratio for maximum brake power and torque and minimum soot and NOx formation. They observed that varying compression ratio for achieving maximum torque showed significant increase in brake power and reduction in BSFC and soot. However, there was sharp rise in NOx, Pmax and Tmax. Similarly, for constant minimum NOx, the optimization results in drop of brake power and increase in BSFC and soot. Operation under constant maximum brake power or minimum soot emission was impractical as it required compression ratio up to 35 which in turn caused Pmax to rise as high as 19 MPa.

Alain Maiboom et al. [7] conducted experimental study on various effects of cooled EGR on combustion and emission. Apart from NOx reduction other effects such as increase of intake temperature, delay of rate of heat release (ROHR), decrease in O2 concentration and flame temperature, increase of fuel-air ratio at lift-off length, etc. were studied. They observed that at low load conditions, very low NOx and PM emissions can be obtained with high EGR rates, because the combustion is delayed due to the high dilution. It was also accompanied with an increase of BSFC. For some operating points, EGR at constant AFR seem to be a promising way to reduce both NOx and PM emissions without penalty on BSFC.

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Ashish Jashvantlal Modi [8] investigated the effect of insulated heat transfer surfaces on diesel engine energy balance system. He observed reduction in fuel consumption and heat loss to engine cooling system of the ceramic coated engine however due to increase in wall temperatures there was decrease in volumetric efficiency of the engine. The author also observed rise in exhaust gas temperature due less heat transfer to wall. This increase in exhaust gas temperature can be used to increase the 1D Simulation Tool pressure from the turbocharger.

OBJECTIVE

Calculate the energy balance of turbocharged diesel engine and validate the 1D Simulation Tool 1D Thermodynamic model heat balance with experiments. Study the effect of EGR and compression ratio on heat balance and parasitic losses using experiments & 1D simulation tool.

METHODOLOGY

- Testing of engine and collect necessary data.
- The engine was operated on 8-mode cycle at steady state conditions. The data required for calculation of heat balance sheet is collected.
- Estimation/Calculation of engine heat balance.
- 1D model development in commercially available 1D simulation tool
- Validate the model. The model was validated for full load points at rated torque and rated speed conditions.
- Compression ratio response study and validate with experiment
- The heat distribution was plotted for different compression ratios.
- EGR response study and validate with experiment The heat distribution was plotted for different EGR ratios.
- To Tune the model to predict compression ratio and EGR effect on heat balance & engine parasitic losses.

ENGINE SETUP

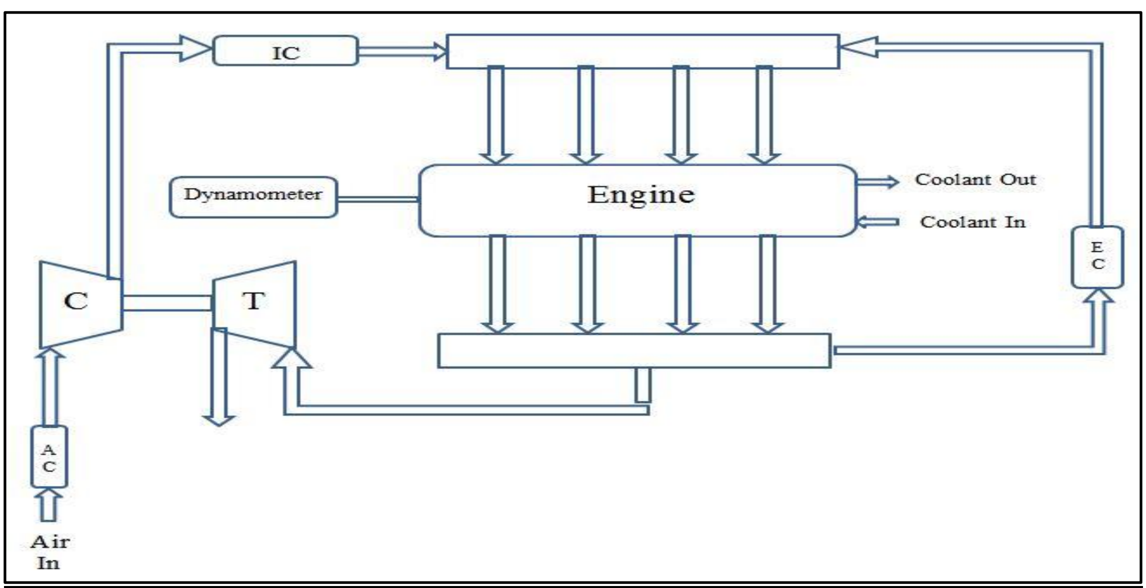

Fig. 1: Engine setup

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Figure 1 indicates a 3.0 L, 55kW, turbocharged, after cooled diesel engine with high pressure EGR loop is used for the present work. The engine is setup as shown in the above diagram. All the temperature and pressure sensors are calibrated before installing at respective location. The mass air flow, fuel flow and coolant flow sensors are installed to measure air, fuel and coolant flow respectively. The coolant flow sensor is positioned at the outlet path of the coolant system. The engine specifications are as follows:

Table 1: Engine Specification

Sr. No.	Specifications	Value
1	Bore	95
2	Stroke	110
3	Con-rod length	192
4	Compression ratio	17
5	No. of cylinders	4
6	Charging	Turbo-charged inter-cooled
7	EGR system	High pressure loop system

ENGINE MODELING AND VALIDATION

Engine Modelling

The model is created by selecting required elements from the elements tree on the left in the AWS workspace and making all the necessary physical connections. Later all the required data at each and every element is entered by double clicking the element. Some data is defined as parameters which can help in simulating multiple cases.

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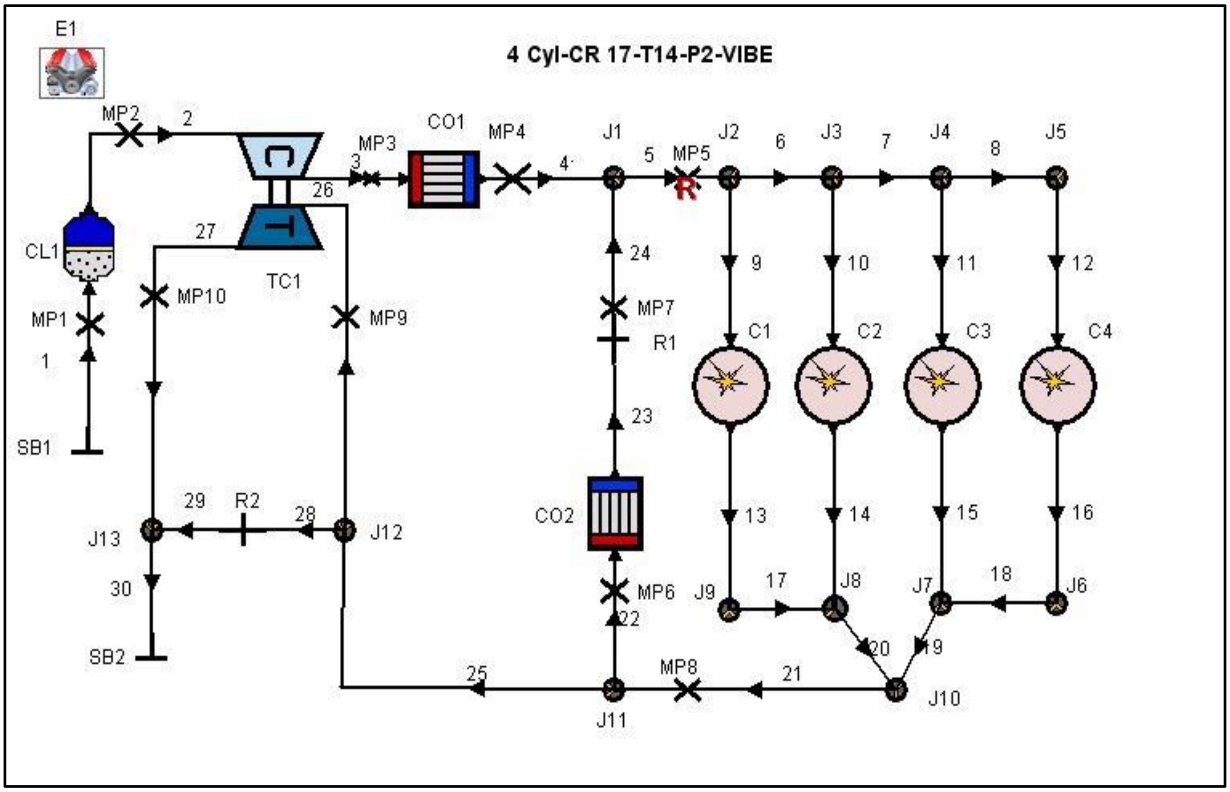

Fig. 3: Engine model in 1D SIMULATION TOOL

Pre-Requisites for Execution of the Model

In order for successful run of the model, certain simulation controls must be provided. Classic species transport is selected. The average cell size for spatial pipe discretization is chosen such that the computational time is optimum. Too large a value of cell will lead to inaccurate results whereas a lesser value may increase the computational time significantly. Also convergence control is applied so as to obtain the output within permissible tolerance limits. In present work a cell size of 20 mm and convergence criterion of 1e-3 bar for IMEP are considered. Further it is important to provide initial values in the initialization subsection. The software utilizes these values as an initial value in the iterative process.

Validation of The Model

The engine model is validated with the experimental data with an error of less than 10%. The validation is carried for two operating points viz. rated speed and rated torque. The parameters considered are torque, power, BSFC, volumetric efficiency, air fuel ratio, peak firing pressure, temperatures and pressures, IMEP and mass flow of air, fuel and EGR. Also the heat balance sheet calculated from the simulated data was found to be within permissible range.

The model is tuned by varying the flow coefficients. In order to achieve the peak firing pressure obtained from the in-cylinder pressure data the shape parameter ‘m’ is adjusted within permissible range.
Once the model was created and all the general input parameters were provided it essential to check for any errors occurring while execution of the model. It is important that there should not be any warnings occurring in the summary of results. Once this was done the tuning of the model was carried out until all of the above parameters were matched within the specified target. Following are some of the graphs showing the accuracy of the model.

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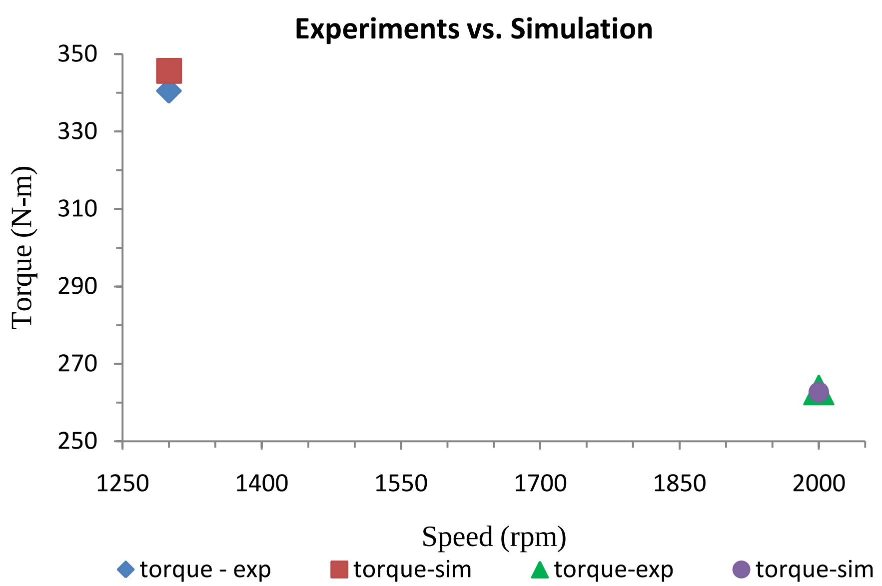

Fig. 4: Comparison of Experimental and Simulation Torque at Two Operating Speeds

For torque a deviation of 1.52% and 0.11% was found from the experimental values for intermediate and rated speed respectively.

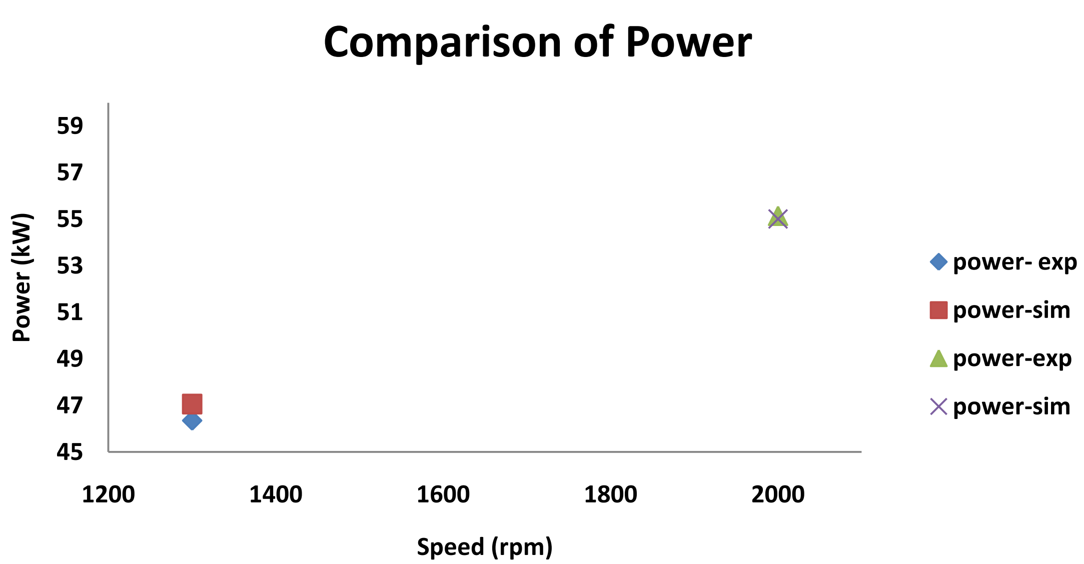

Fig. 5: Comparison of power

For power a deviation of 1.52% and 0.11% was found from the experimental values for intermediate and rated speed respectively.

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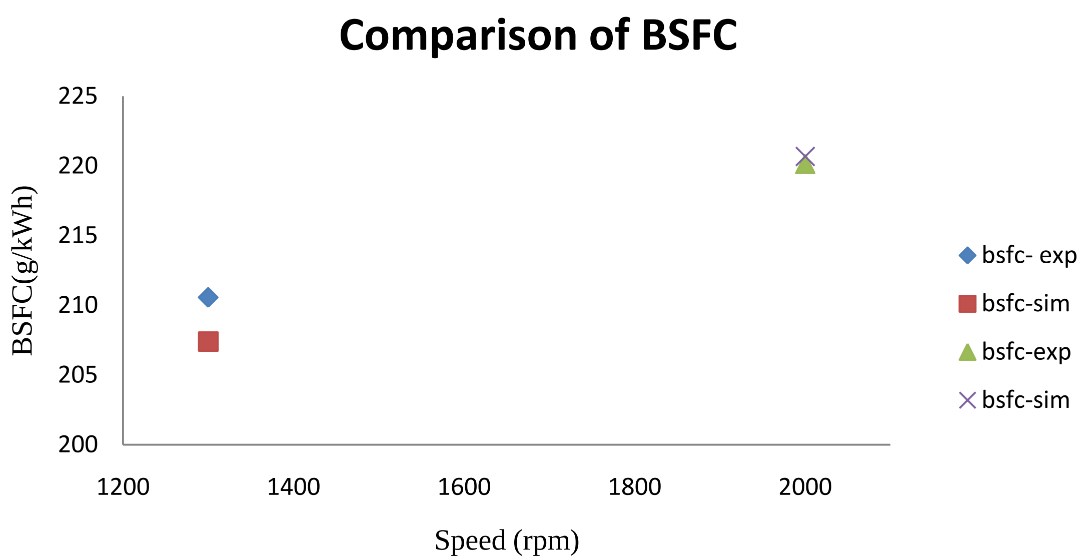

Fig. 6: Comparison of BSFC

For BSFC a deviation of 1.5% and 0.13% was found from the experimental values for intermediate and rated speed respectively.

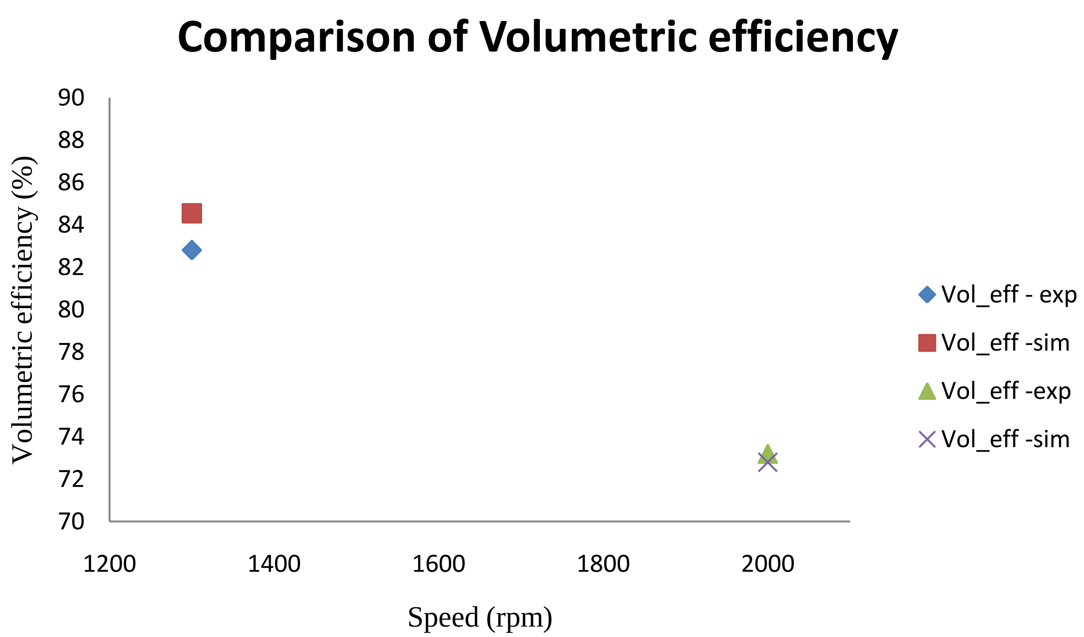

Fig. 7: Comparison of volumetric efficiency

For volumetric efficiency a deviation of 2.11% and 11.35% was found from the experimental values for intermediate and rated speed respectively.

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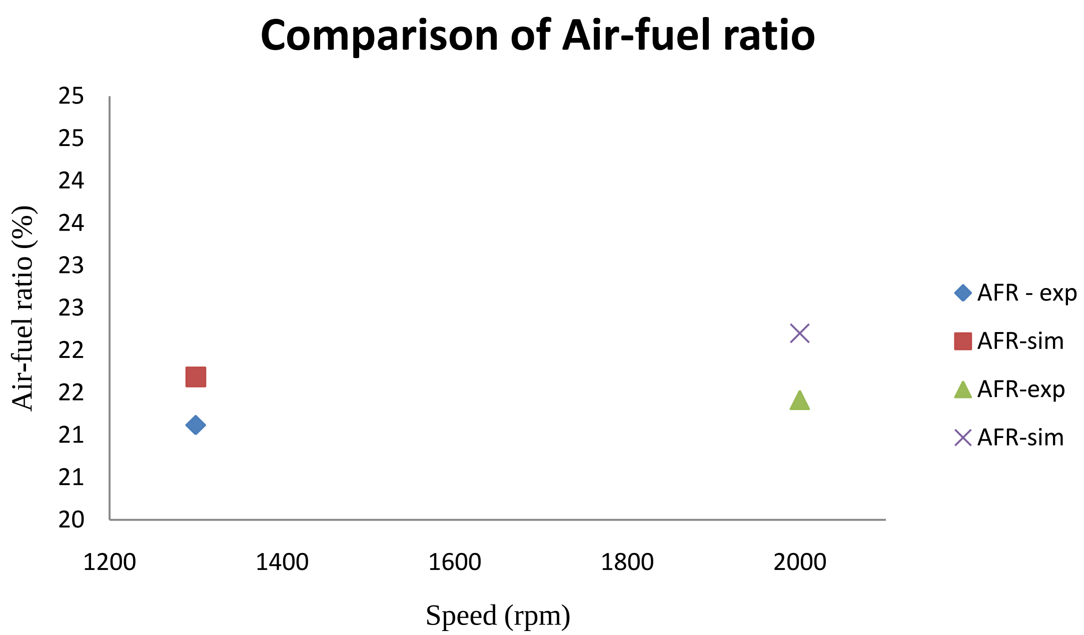

Fig. 8: Comparison of air-fuel ratio

For air-fuel ratio a deviation of 2.69% and 11.05% was found from the experimental values for intermediate and rated speed respectively.

RESULTS AND DISCUSSION

Experimental Analysis-Heat Balance Sheet

A 3.0 L, 55kW, turbocharged, after cooled diesel engine with high pressure EGR loop was operated on 8-mode cycle as in table 2 under steady state conditions. Because of the transient thermal behavior of the engine it is difficult for analyzing the thermal heat flux experimentally. Even if working under steady state conditions utmost care has to be taken to minimize any unsteady behavior of the engine. Therefore, the engine was allowed to run for 5-8 minutes in order to stabilize its outcome. The required temperatures, pressure and the mass flow rates were recorded. In-cylinder pressure data was also recorded.

Table 2: Operating points

Mode	Speed (rpm)	Load (%)
1	2000	100
2	2000	75
3	2000	50
4	2000	10
5	1300	100
6	1300	75
7	1300	50
8	750	0

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The heat balance sheet was calculated for rated power speed and the rated torque speed points. The energy distribution across the primary paths was found to be a function of speed and load. Following graphs show the proportion of heat distribution.

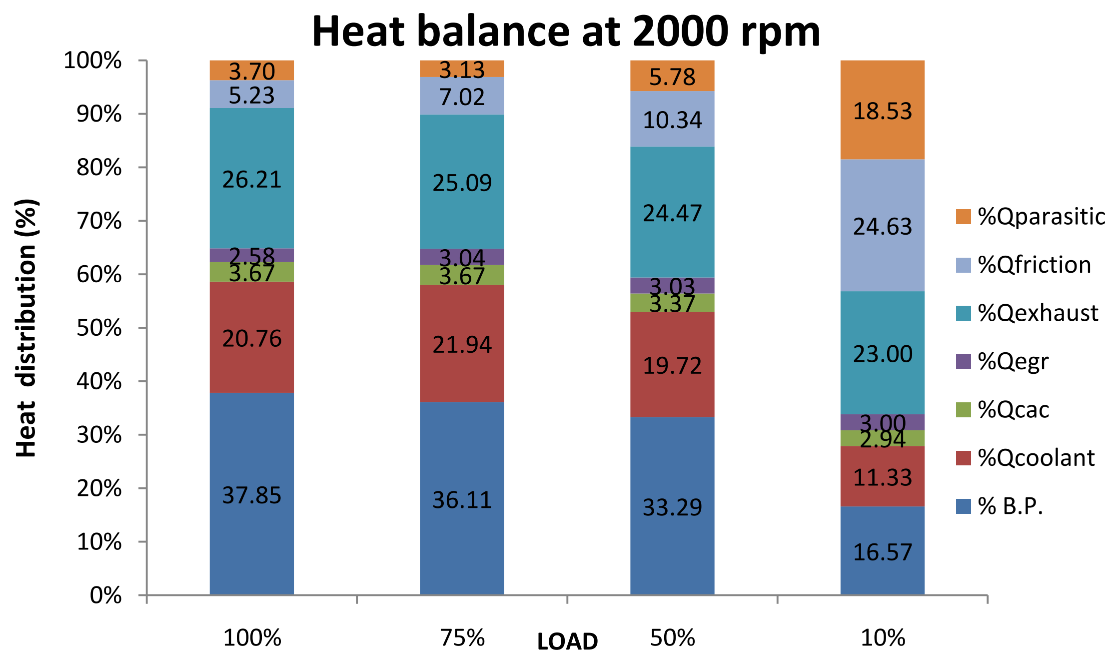

Fig. 9: Heat balance at rated speed

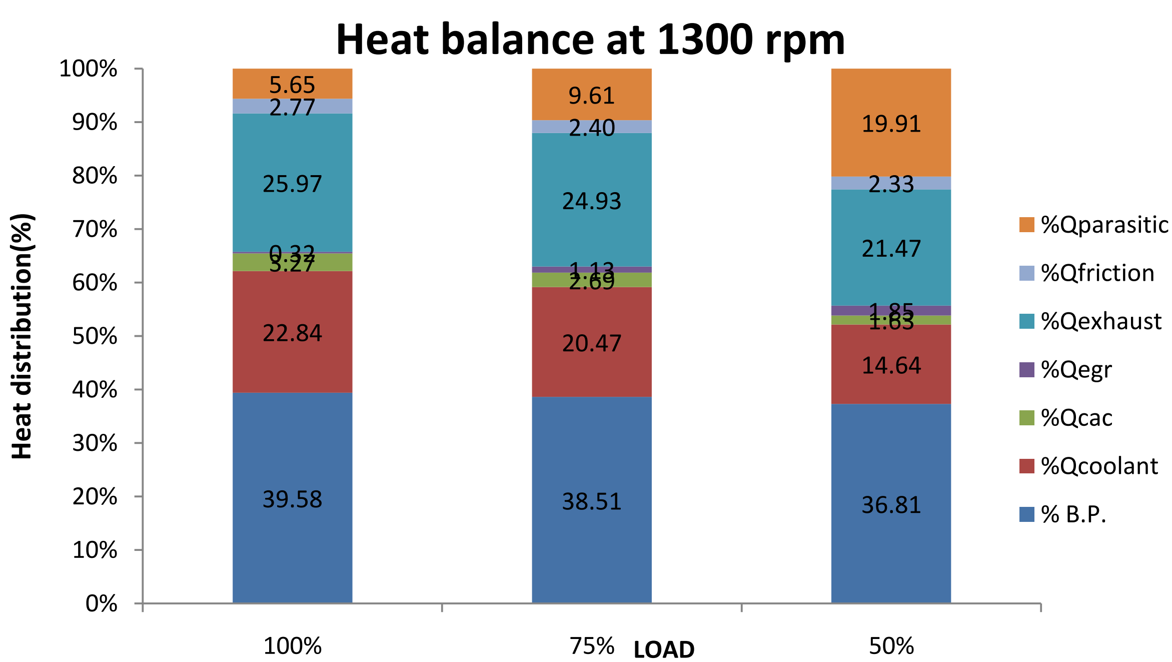

Fig. 10: Heat balance at intermediate speed

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The contribution of brake power to the energy balance was seen to be increasing with the amount of load. Also the percentage of brake power at rated torque conditions was found to be more than at rated speed for load points. This was due the fact that the frictional losses are a function of speed and the point of minimum BSFC lies closer to the rated torque condition implying that the rated torque conditions burns fuel more efficiently. The heat lost to the coolant also increased with the amount of load and higher at higher speed. The coolant flow is function of engine speed and hence the cooling improves with the speed. The heat lost to exhaust gas increases with increase in loading conditions. However, the portion of exhaust gas at rated speed condition is more than at rated torque condition. This trend is opposite to that of the brake power.

Simulated Results-Effect of Compression Ratio

A base model was established by experimental validation. The accuracy of the model was over 90 %. This model was further used to study the effect of compression ratio on the heat distribution of the engine. The brake thermal efficiency, the percentage of heat lost to exhaust gas and coolant are of prime importance.

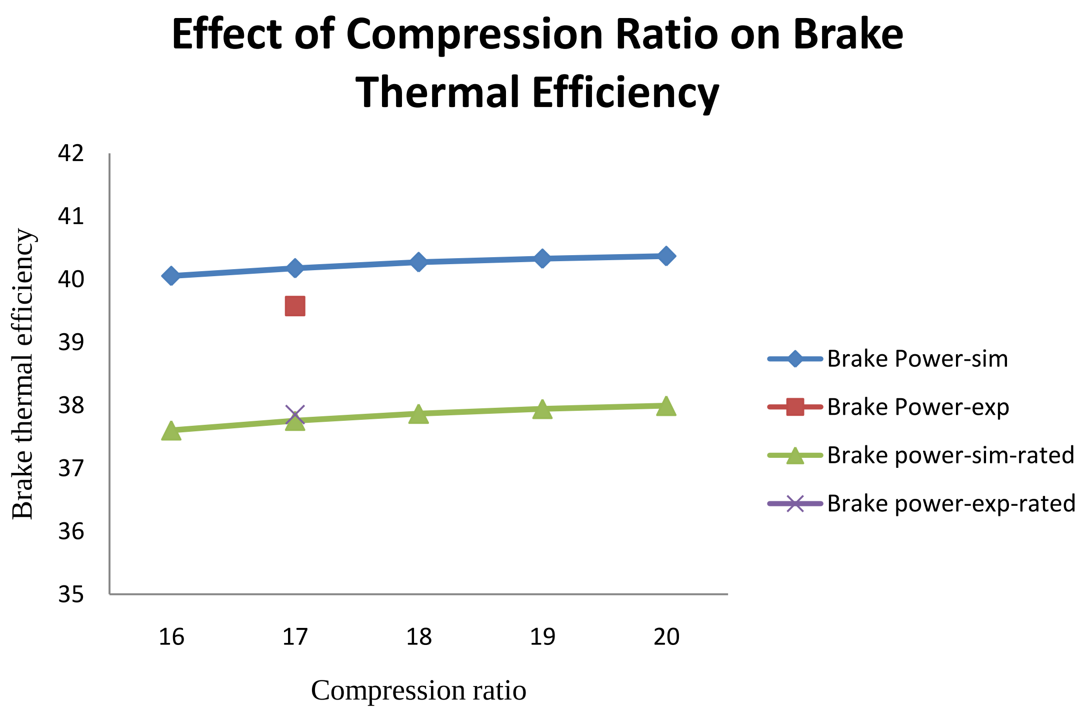

Fig. 11: Effect of compression ratio on brake thermal efficiency

The compression ratio plays a crucial role in the combustion of the fuel. Higher compression ratio leads to higher in-cylinder pressure and temperature favorable for better combustion. Since the fuel is more efficiently burnt, the brake specific fuel consumption is reduced leading to increase in brake thermal efficiency. Figure 11 shows the effect of compression ratio on the brake thermal efficiency. Over the range of compression ratios of 16-20 the brake thermal efficiency increased from 40% to 40.4% for rated torque condition whereas for rated speed it varied from 37.6% to 38%. As the combustion process is dependent on many parameters like the swirl ratio, injection timing and pressure, the shape of the bowl, etc. the increment seen in the simulated results was small. The different compression ratios need further optimization.

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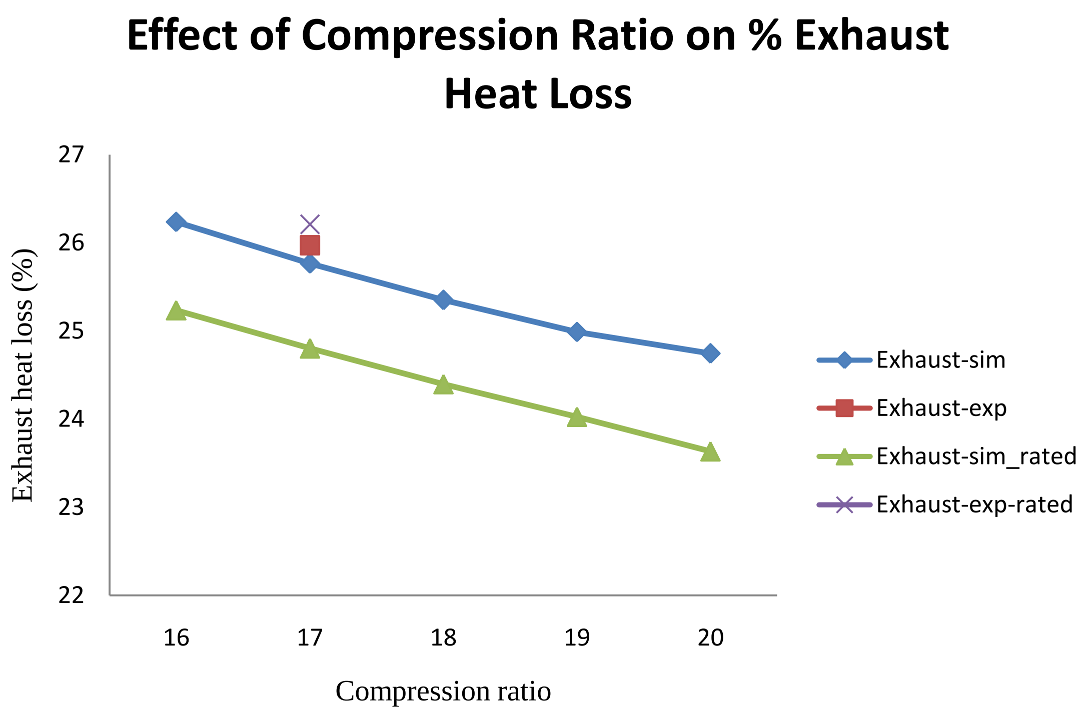

Fig. 12: Effect of compression ratio on % exhaust heat loss

As seen from figure 12 the exhaust heat loss proportion reduces with increase in compression ratio. With better combustion there is slight increase in exhaust manifold pressure. Due to this and same EGR valve lift there is an increase in the mass flow rate of EGR. Therefore, the proportion of exhaust gas heat loss reduces. For rated torque the contribution of heat loss to exhaust reduces from 26.4 % to 24.9 % whereas for rated speed it reduces from 25.2% to 23.6%.

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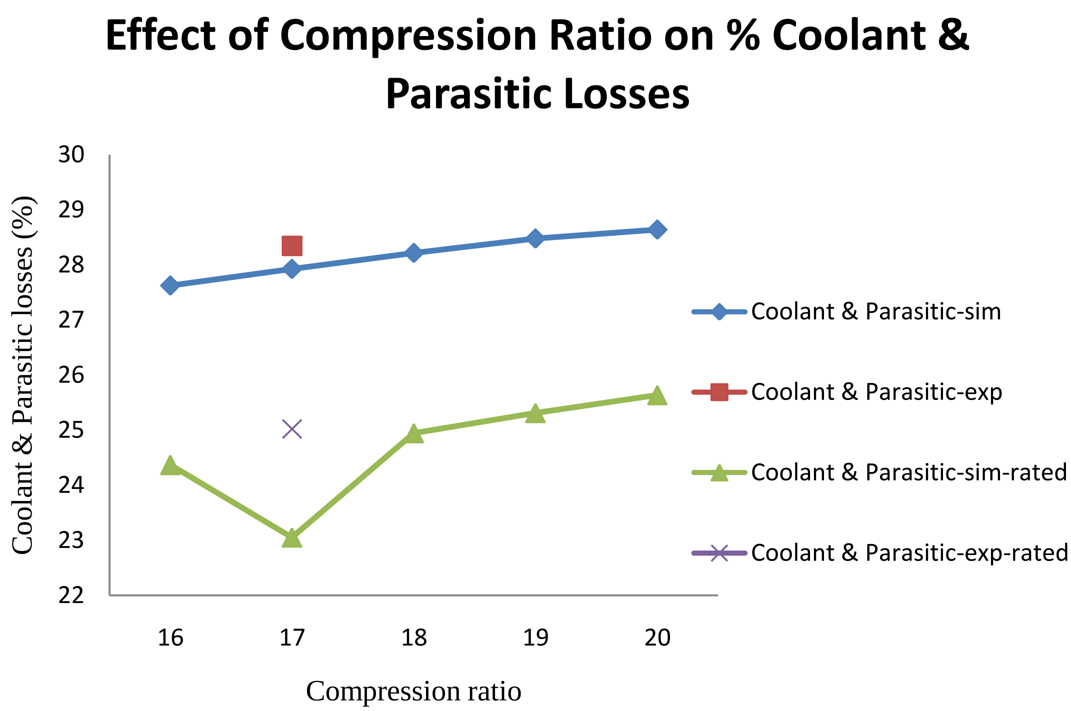

Fig. 13: Effect of compression ratio on coolant & parasitic losses

As the compression ratio is increased the in-cylinder temperatures are increased. This causes the wall temperatures to go up. Due to increased temperature the extra heat is partly lost to coolant and partly to environment through radiation or convection. The exact contribution can be known through experimental analysis only. Figure 13 depicts the combined variation of coolant and parasitic losses.

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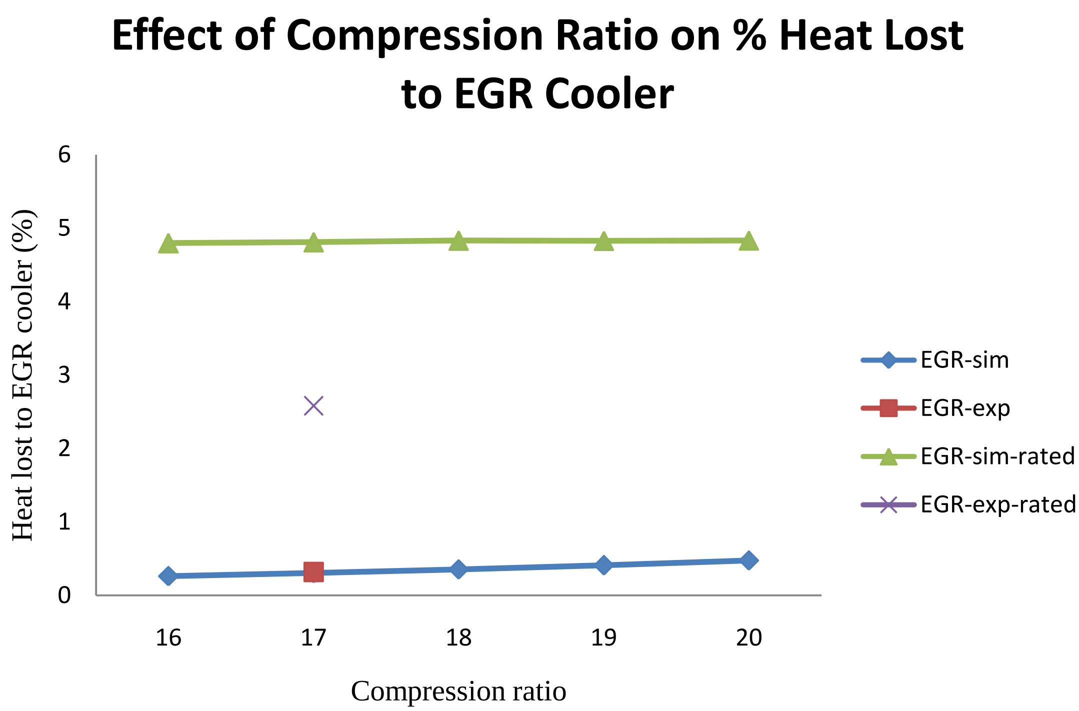

Fig. 14: Effect of Compression Ratio on Heat Lost to EGR Cooler

With increase in exhaust manifold pressure there is increase in EGR mass flow rate as mentioned above. This causes an increase in heat lost to EGR cooler. Figure 14 shows the effect heat lost to EGR cooler.

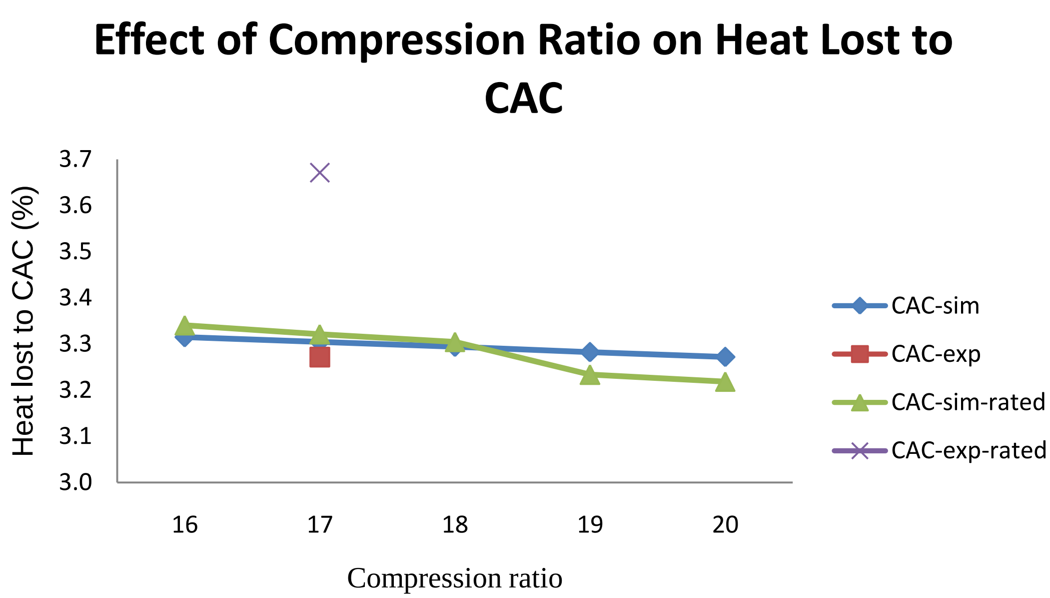

Fig. 15: Effect of compression ratio on heat lost to CAC

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The volumetric efficiency reduces slightly with increase in compression ratio as the EGR mass flow rate is increased. Thus the air flow is decreased. As a result, the proportion of heat lost in the charge air cooler reduces with increase in compression ratio. For rated speed the value reduces from 3.34% to 3.22 %. On the other hand, for rate torque condition it reduced from 3.31% to 3.27%.

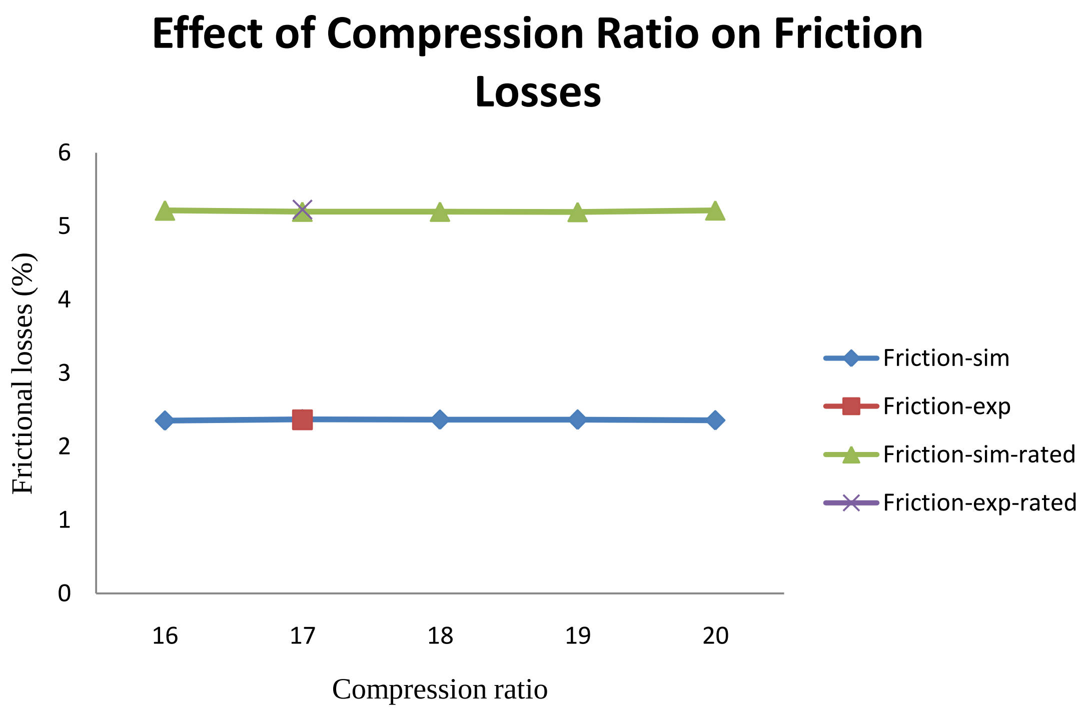

Fig. 16: Effect of compression ratio on friction losses

The variation in compression ratio has very little on the frictional losses at both rated torque and rated power speed.

Simulated Results-Effect of EGR Ratio

The validated model with compression ratio 17 was considered again as a base model for the study of variation of EGR.

Table 3: Cases considered for EGR variation

Case	1	2	3
Intermediate speed	0	2.32	3.8
Rated speed	0	18.5	33

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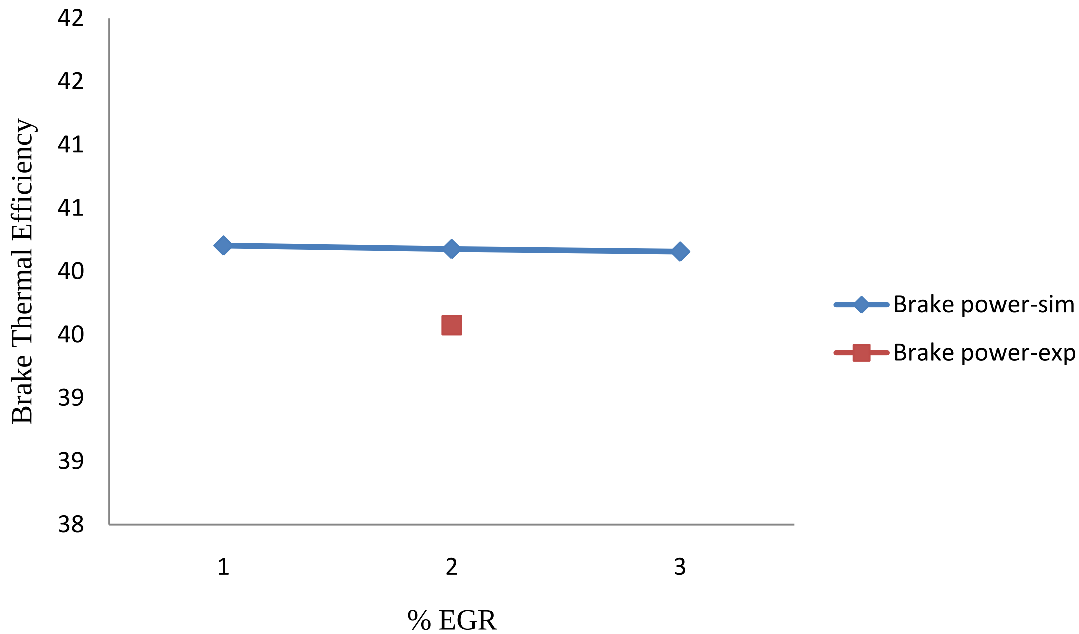

Fig. 17: Effect of EGR on brake thermal efficiency

Figure 17 shows the effect of EGR on brake thermal efficiency. It is seen that there is very little or no variation in brake thermal efficiency. With increase in EGR ratio there is slight increase in BSFC. However, the corresponding change in brake thermal efficiency is not differentiable.

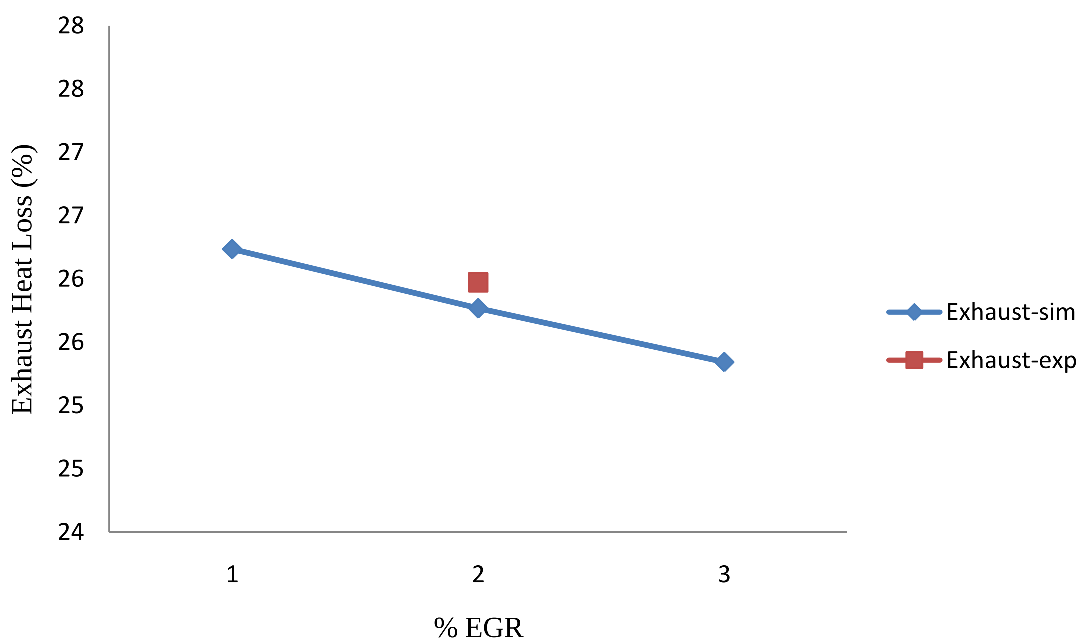

Fig. 18: Effect of EGR on % heat lost to exhaust

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EGR reduces the amount of exhaust gas bypassed for given speed and load. Turbocharger with a waste gate bypasses the extra amount of exhaust gas mass flow to the environment. This leads to increase in heat lost to exhaust. However, if certain amount of exhaust gas to be bypassed is re-circulated considering the permissible limits of emissions then heat lost to exhaust gas reduces. This is very much evident from the figure 18. As EGR rate is increased the percentage of heat lost to the exhaust is reduced. The value varies from 26.23% to 25.34% for intermediate speed.

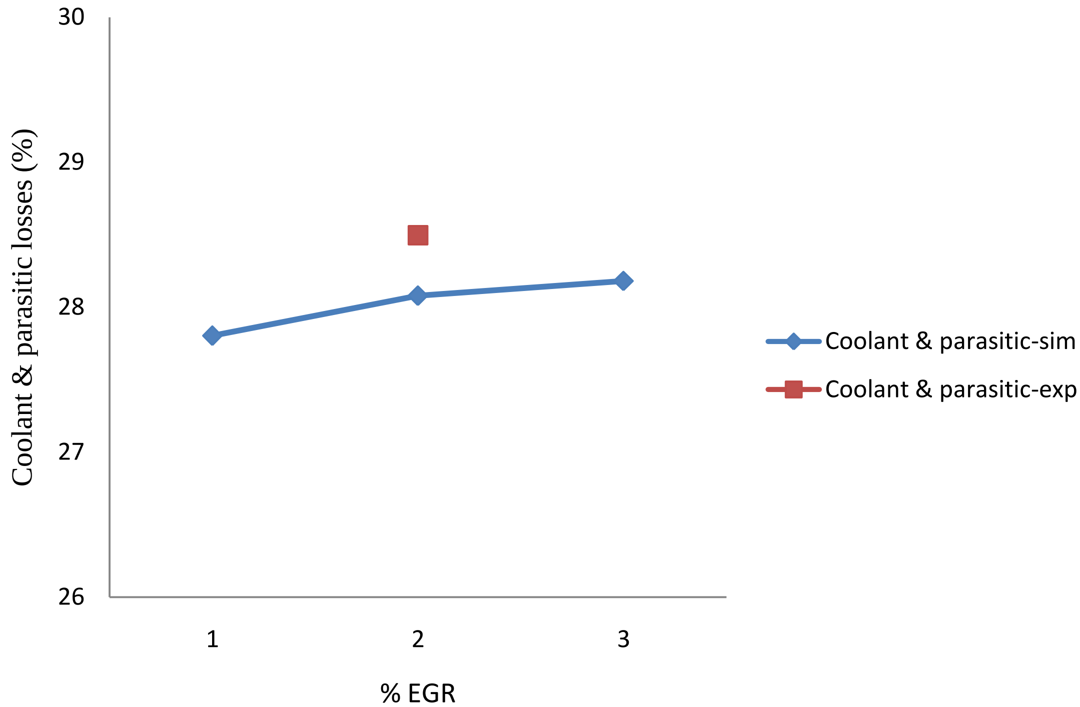

Fig. 19: Effect of EGR on % coolant & parasitic losses

A slight variation in the heat lost to coolant and heat loss to radiation and convection was observed. For intermediate speed the value varied from 27.8% to 28.18%

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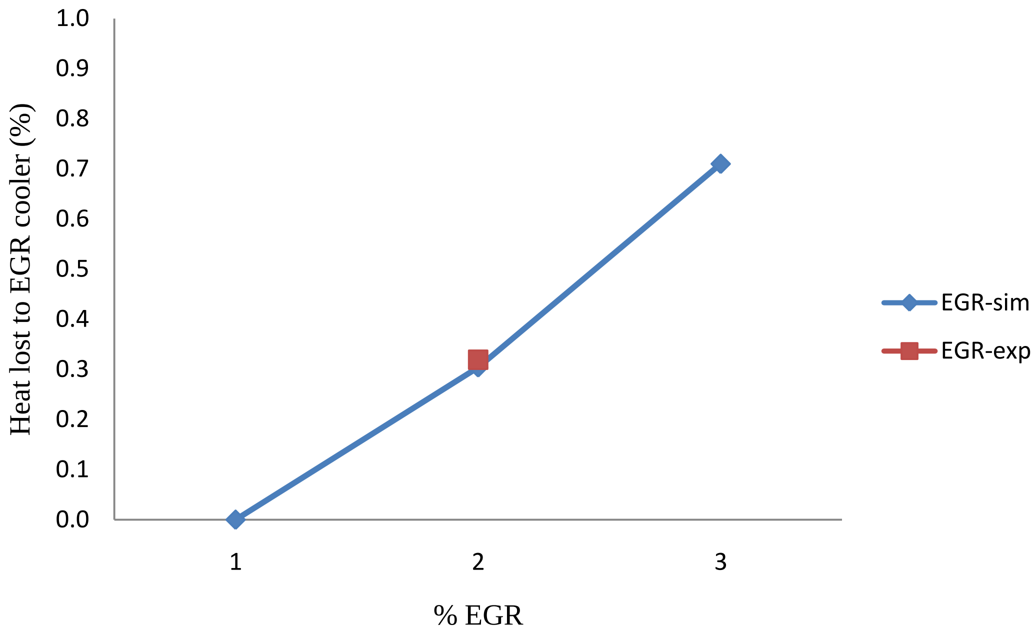

Fig. 20: Effect of EGR on % heat lost to EGR cooler

As EGR flow rate is increased the heat lost to EGR cooler is proportionally increased. The value varies from 0% to 0.71% as the EGR rate is increased.

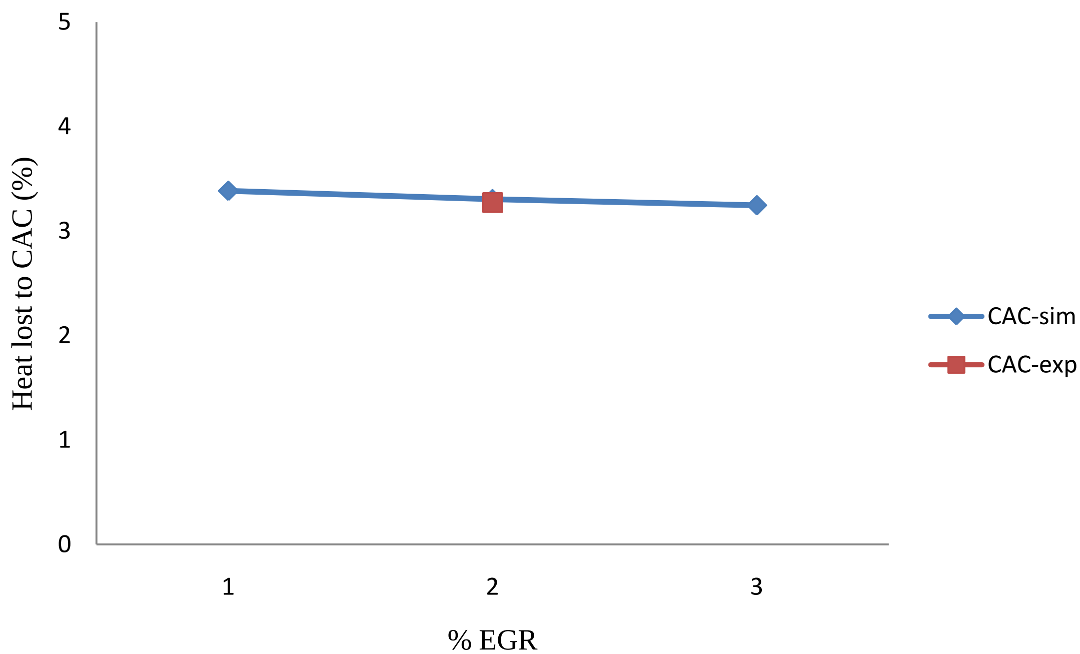

Fig. 21: Effect of EGR rate on Heat lost to CAC

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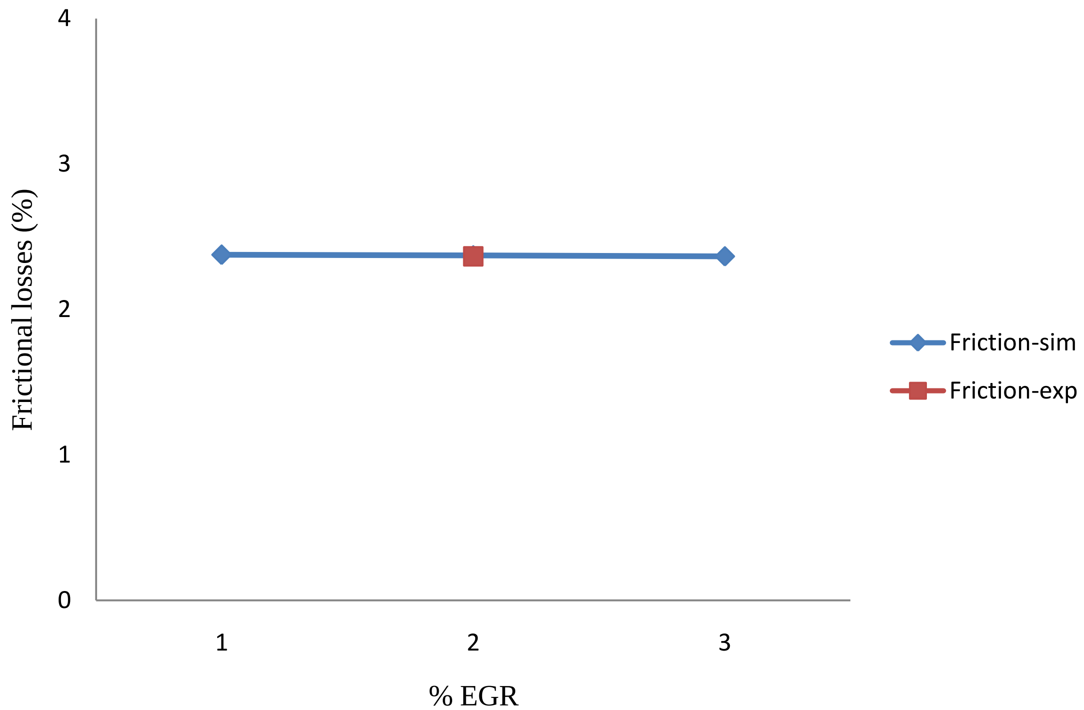

Fig. 22: Effect of EGR rate on (%) Heat lost due to friction

CONCLUSION

A 3.0 L, 55 kW, turbo charged, inter-cooled engine with high pressure EGR system is operated under steady state condition on 8-mode cycle in order to estimate the heat balance across the engine. Later a 1-D thermodynamic model of the same engine was created to simulate the actual experiment. The model was validated with the experimental data. Heat balance sheet was again estimated using the simulated data and was found to be in coherent with the one calculated from experimental data. The validated model was further used to simulate for different compression ratios and EGR rates. The effect of compression ratio and EGR rate on the heat distribution was studied. A summary of the effects on each of the primary paths of heat distribution is given below:

- Over the range of compression ratios of 16-20 the brake thermal efficiency increased from 40% to 40.4% for rated torque condition whereas for rated speed it varied from 37.6% to 38%. The brake thermal efficiency had very little or no variation with change in EGR rate. From the experimental analysis it was seen that the brake thermal efficiency increased with load and was higher at rated torque condition than at rated speed.
- The proportion of exhaust heat loss reduces with increase in compression ratio. For rated torque the contribution of heat loss to exhaust reduces from 26.4 % to 24.9 % whereas for rated speed it reduces from 25.2% to 23.6%. Similarly, trend is observed for increasing EGR rates. The value varies from 26.23% to 25.34% for intermediate speed. From experimental analysis the heat lost to exhaust gas increases with increase in loading conditions. However, the portion of exhaust gas at rated speed condition is more than at rated torque condition.
- The portion of total heat input lost to coolant and parasitic losses together increases with increase in compression ratio from 27.63% to 28.64% for rated torque speed and 24.37% to 25.64% for rated speed condition. The parasitic losses reduced with increase in load and were higher at lower speed. The proportion of heat lost to coolant increased with both speed and load.

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- With increase in compression ratio the percentage of heat lost to EGR cooler increases from 0.26% to 0.48% for intermediate speed and from 4.8% to 4.83% for rated speed. As EGR flow rate is increased the heat lost to EGR cooler is proportionally increased. The value varies from 0% to 0.71% for intermediate speed and % to % for rated speed as the EGR rate is increased. Also heat lost to EGR was found proportional to speed.
- The proportion of heat lost in the charge air cooler reduces with increase in compression ratio. For rated speed the value reduces from 3.34% to 3.22 %. On the other hand, for rate torque condition it reduced from 3.31% to 3.27%. The heat lost to charge air cooler has little variation with EGR rate.
- The variation of compression ratio and EGR rate had a little effect on the frictional losses.

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