

PERFORMANCE MAP MEASUREMENT, ZERO-DIMENSIONAL MODELLING & VIBRATION ANALYSIS OF A SINGLE CYLINDER DIESEL ENGINE

D. Khan^{1,*}, Z. Gül²

ABSTRACT

Single Cylinder Diesel Engines are simple and very economical in manufacturing. Their multipurpose usability and the capability to deliver the maximum power possible within a given envelope makes them very demanding engines in the market. Simulation tools are widely used nowadays to minimize the energy and time needed for a real engine design and development. Zero-dimensional models are very suitable and reliable to observe the engine operation under different conditions. Contrary to the previous studies, this paper presents a comparison between the practical and simulation model data of a single cylinder Diesel Engine. The purpose of this research was to investigate the fundamental variations between the simulation and experimental results with the help of characteristic engine performance maps. Experiments were conducted on a practical 1.16 L Diesel Engine under variable conditions which were then repeated on the simulation model to analyze and evaluate the differences between the obtained results. Zero-dimensional modelling was performed using GT-Power, a powerful commercial engine simulation software. This study also involved the prediction of optimum speed (RPM) of the engine by performing a vibration analysis using a wireless accelerometer. The maximum torque of the 1.16 L Diesel Engine is given to be 80 Nm @ 1,800 RPM, while the simulation model indicated it to be 78 Nm at the same RPM value. Likewise, maximum power output was indicated to be 18 kW @ 2,400 RPM, while the experimental results showed it to be 15 kW @ 2,400 RPM. These results laid down a liable basis for the prediction of several operating parameters of the engine which could act as a solid rung for further studies on this subject.

Keywords: *Single-cylinder, Diesel Engines, Zero-dimensional Model, Combustion Analysis, Engine Maps, Vibration Analysis*

INTRODUCTION

With an increase in the fuel prices and emission standards, researchers have been forced to look for a perfect combination of efficient, low fuel-combustion and low emission engine.

Diesel engines have a very high expansion ratio and possess the ability of lean burning which makes them one of the most highly efficient engine in the market.

The emission standard and regulations all over the world are becoming increasingly stern and region due to the risk these emission possess on human health [1]. This poses a serious challenge to the researchers and Automobile industries to obtain an optimal value of efficiency-emissions-MPRR trade-offs. This task is not only intricate but requires a lot of statistical and experimental research [2].

In the current scenario, practically examining an engine is becoming less popular because of the high cost and energy needed for its setup. Simulation techniques such as: Zero Dimensional Simulation Models [3-7] are being practiced more often to nearly estimate the behavior of the engine under different circumstance. This means that the engine output parameters can be predicted based on the engine input parameters with a very low deviation from practical results [8].

The performance of an Internal Combustion Engine under variable loads and conditions can fully represented by a performance map [9-10]. GT-Power (GT-Suite) is designed for steady state and transient simulations suitable for engine/power train control analysis and can be used to simulate all kinds of Internal Combustion (IC) engines [11]. The software uses one dimensional gas dynamics to represent the flow and heat transfer in the components of the engine model. GT-Power has the capability to build models of engines that are very close to reality. To make the model work as the real engine, it is most crucial to emulate the real engine

This paper was recommended for publication in revised form by Regional Editor Omid Mahian

¹Department of Mechanical Engineering, Marmara University, Istanbul, TURKEY

²Department of Mechanical Engineering, Marmara University, Istanbul, TURKEY

*E-mail address: daniyal179@hotmail.com

Manuscript Received 28 October 2016, Accepted 1 January 2017

down to the smallest pipe and angle. To model the engine in GT-Power there are objects like cylinders, crankcases, pipes, turbochargers and so on, that are easy to modify by desire and requirement.

As by its name Zero Dimensional Models do not consider any dimensions of flow field but only involve the in-cylinder envelope of heat release rates. The first law of thermodynamics is widely involved in the working fluid of this system. Many of these parameters directly affect the performance of the engine [12-13]. Thus, these tools do not only help in the pre-optimization of the actual thermodynamic cycle of the engine, but also give a close overview of their real working under various environments and conditions [14-17].

Repeating a practical experiment on a simulated model gives drastically similar results provided that the input parameters are given precisely and accurately [18]. This comparison of experimental and simulation data can not only help in better understanding of the contrast between the realistic and simulated model but will also give estimations of various parameters which can be utilized in further studies.

In-cylinder pressure and engine block vibration signals have a direct connection [19]. Vibration measurement on the engine block or cylinder heads has been used successfully for the detection of faults in diesel engines [20-23]. This phenomenon can be applied in further studies for engine parameters optimization.

Vibration and noise analysis of the engine are becoming more practical to determine the performance of the engine under different operating conditions. These analyses can setup a core basis of non-intrusive measurements of diesel engine and can play an important role in engine optimization [24-26].

EXPERIMENTAL SETUP AND METHODOLOGY

The engine chosen to carry out experimentation was a single cylinder, four stroke, vertical, water cooled, direct injection computerized Engine from Erin Motors as shown in Figure 1. Table 1 gives a detailed account of the technical specifications of the engine, while its technical dimensions can be seen from Figure 2. It is a 1.16 Liters four-stroke engine with single cylinder. The engine is naturally aspirated with a compression ratio of 14.6. The fuel injection system is directly mounted in the cylinder head and the fuel is injected directly into the combustion chamber. Because this injector sprays directly into the combustion chamber, therefore, it operates at very high pressures.



Figure 1. 1.16 L Single Cylinder Diesel Engine by Erin Motors

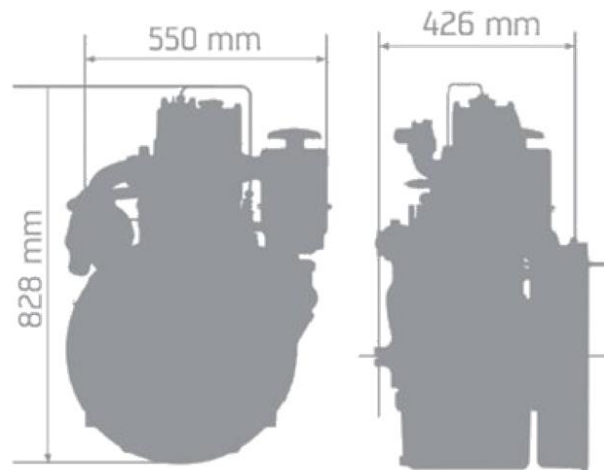


Figure 2. Dimensions of the Engine

The setup of the Engine in the Engine Testing Laboratory is shown in Figure 3. The engine was connected with an AC Dynamometer to examine the performance of the engine under various load conditions. The whole operational support was provided by Erin Motor and Automotive Technologies Research & Development

Table 1. Engine Specifications

Number of Cylinders	1
Displacement (cm³)	1,160 or 1.16 L
Valves Per Cylinder	4
Bore x Stroke (mm)	108 x 127
Compression Ratio	14.6 : 1
Power Output (kW) @ Speed (rpm)	18 @ 2,400
Max. Torque (Nm) @ Speed (rpm)	80 @ 1,800
Specific Fuel Consumption (g/kWh)	255
Cooling	Water Cooled
Min. Idling Speed (rpm)	600
Aspiration	Naturally Aspirated
Net Weight	169 kg
Max Oil Capacity	4.5 Litres
Combustion System	Direct Injection
Rotation	Clockwise
Starter	12V – 2 kW
Alternator	14 V – 125 A
Intake Valve Opening Crank Angle	54°
Intake Valve Closing Crank Angle	74°
Exhaust Valve Opening Crank Angle	74°
Exhaust Valve Closing Crank Angle	54°
Maximum Valve Lift	11 mm
Crank Angle at start of Injection	19° BTDC

Company Turkey (OTAM). All the practical experimentation was conducted under controlled environments. The Engine was operated from the Control/Operating Room by a special software, iLAB, which connected the whole Engine system with the computer.

**Figure 3.** Experimental Setup in Engine Testing Laboratory

The parameters of piston and cylinder of the Engine were measured and recorded as shown in Table 2. The engine's performance was investigated under many conditions and environments and the data was recorded digitally by various sensors. This data was then processed to get the results in the form of graphics to make them more comparable. The next stage involved the creation of the model of the same engine on GT-Power.

The first step for GT-Power modelling was to measure each technical parameter of the diesel engine.

Table 2. Cylinder, Piston and Crank Dimensions

B	Bore	108 mm
S	Stroke	127 mm
r	Connecting rod length	198 mm
a	Crank offset	63.5 mm
s	Piston position	Varies
θ	Crank angle	Varies
S_c	Clearance Height	9.3 mm
V_c	Clearance Volume	85 cm ³
V_d	Displacement Volume	1165 cm ³

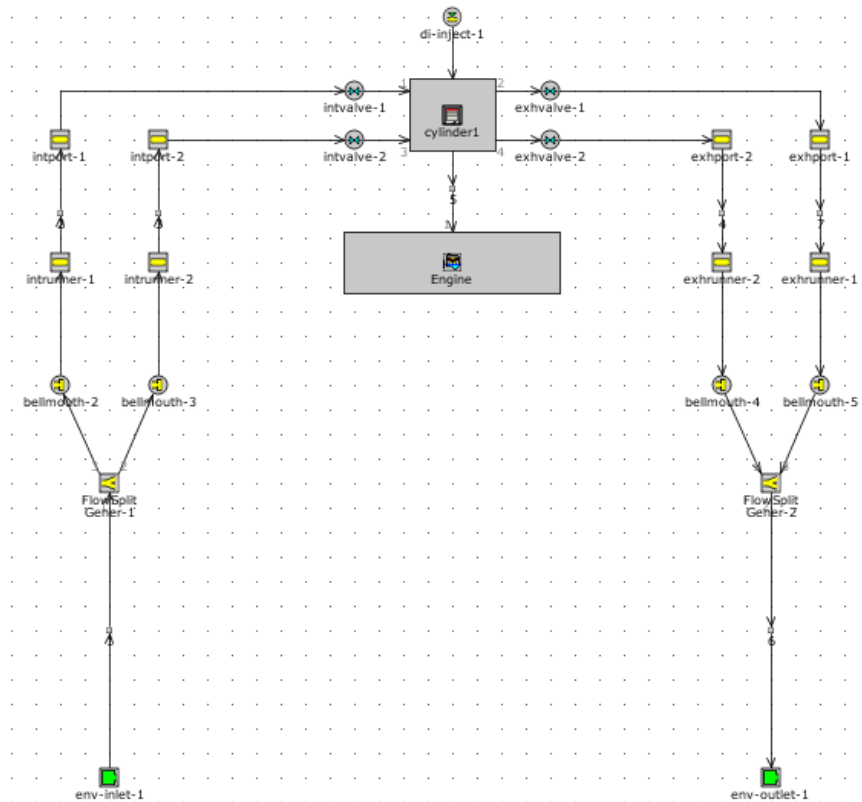


Figure 4. Engine Simulation Model in GT-Power

Numerous parameters of the engine were taken into account and the measurements of even small portions were recorded using Laboratory Measurement Tools. This data of measurement was performed by practically measuring the components and also by taking their readings from the technical drawings. This is a very critical step for GT-Power data input library.

A simulation model of the engine was setup in GT-Power also in order to get the data from zero dimensional simulation. Figure 4 gives a basic over view of the simulation map setup in this commercial engine software. The model breaks down to a number of branches containing intake system, single cylinder and exhaust system. Every single component of the engine was defined with its particular reading to obtain accurate and precise results.

The system consisted of one Environment Inlet and Exit, Two Inlet and Exhaust Runners, Two Inlet and Exhaust Ports, Two Inlet and Exhaust Valves, One Cylinder, One Direct Injection Injector and One Crank-Shaft Engine System Once the model was fully setup, information such as case specific input, type of simulation, and desired output were described. Most of this was accomplished through case setup in GT-Power.

RESULTS AND DISCUSSION

Average piston speed of an engine can be described as the mean speed of the piston and is a function of stroke and RPM as shown in equation 1 [9].

$$Up = 2.S.N \quad (1)$$

where;

- Up : Average Piston Speed in m/s
- S : Stroke length in m
- N : rpm/60 revolutions per second

According to the above mentioned equation, the maximum mean piston speed for our engine can be calculated as:

$$Up_{max} = 2(127 \times 10^{-3}) \left(\frac{2400}{60} \right) = 10.16 \text{ m/s}$$

The term of maximum piston speed (Up_{max}) is widely used for the classification of the engines. The maximum mean piston speed of this engine at 2400 RPM is calculated to be 10.16 m/s which classifies this engine in the category of 'low speed diesel and medium speed diesel engines'.

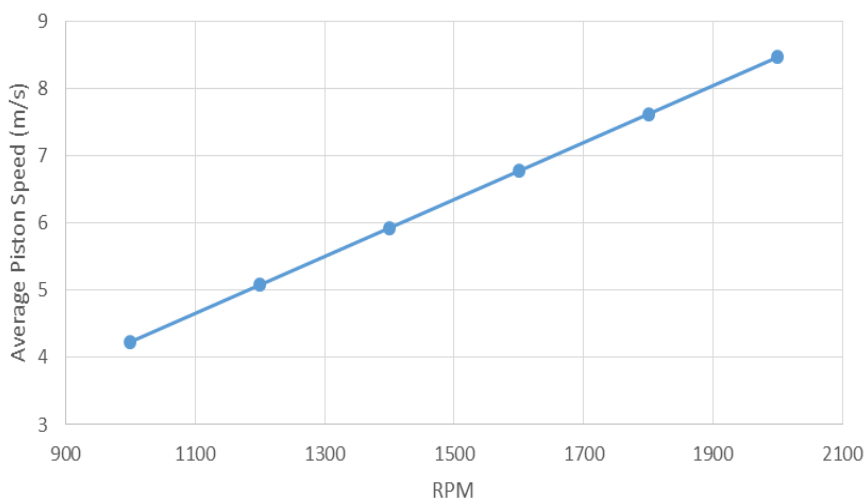


Figure 5. Average Piston Speed with RPM of the Engine

The trend of average piston speed against RPM is shown in the Figure 5. This clearly goes along the logic of increase in the average piston speed with an increase in RPM.

The conchoids of the practical and simulated model engine maps, shown in Figure 6 and Figure 7, appeared to be similar apart from some little contraries shown at lower RPMs, which indicate practical operational performance difference of the engine for low RPMs. For the range of high load, the minimum Specific Fuel Consumption (SFC) is seen to be at low engine speed. The conchoids start to become flat at low load range. The increase in throttle and frictional losses in relation to the useful torque output is the main reason behind this trend. A visible increase in fuel consumption is described by a sharply increasing gradient of fuel consumption at a constant load with respect to increasing RPMs because of the effect caused by these two factors. Moving from the region of lowest BSFC on a line of constant RPM, towards lower values of BMEP, mechanical efficiency falls, because the IMEP decreases. The specific fuel consumption is shown to be the lowest (highest fuel efficiency) at high BMEP with a relatively low piston speed.

The comparison of these maps clearly indicates a similarity between the obtained results from both cases. Although some fluctuations and anomalies were seen to exist but generally the trend of the lines was seen to be the same. Comparing Figure 8 and Figure 9, the torque values for particular BSFC values in the experimental

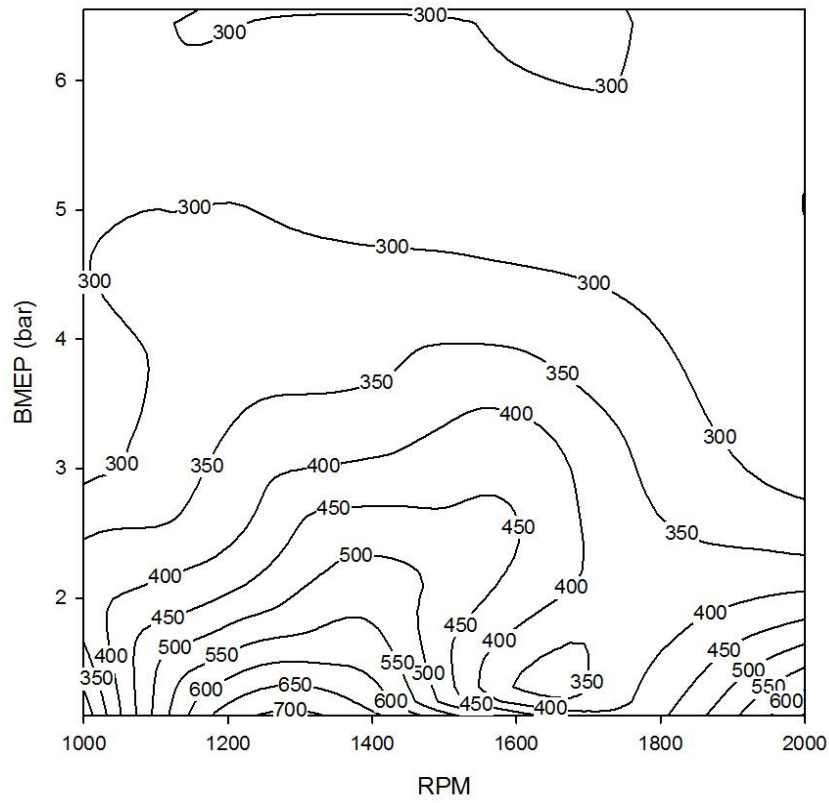


Figure 6. Engine Performance Map from Experimental Results (BMEP-RPM-BSFC)

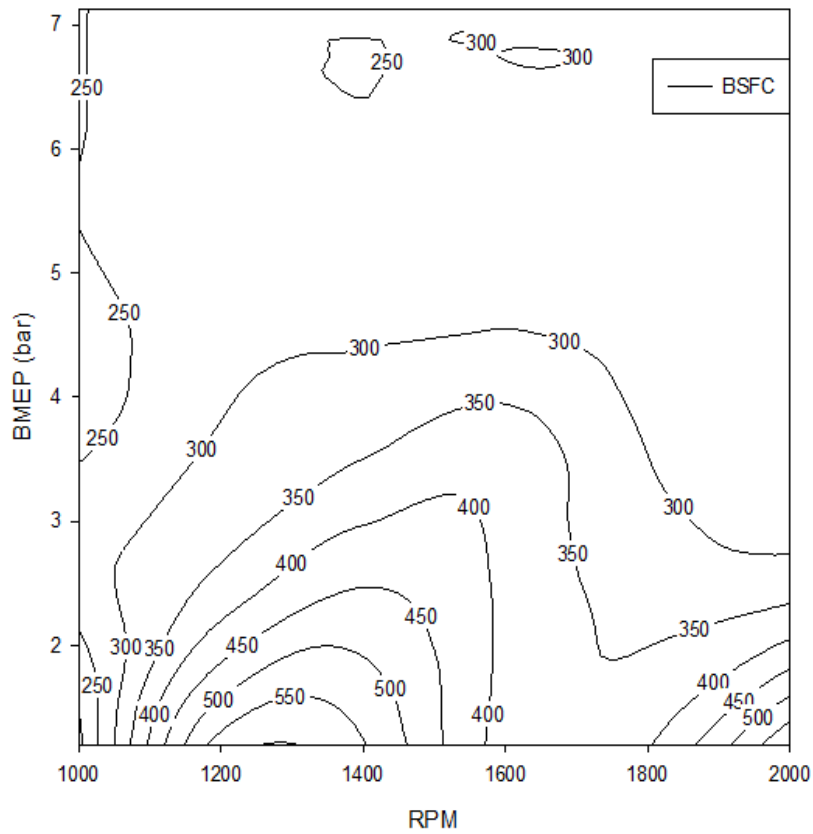


Figure 7. Engine Performance Map from GT-Power Model (BMEP-RPM-BSFC)

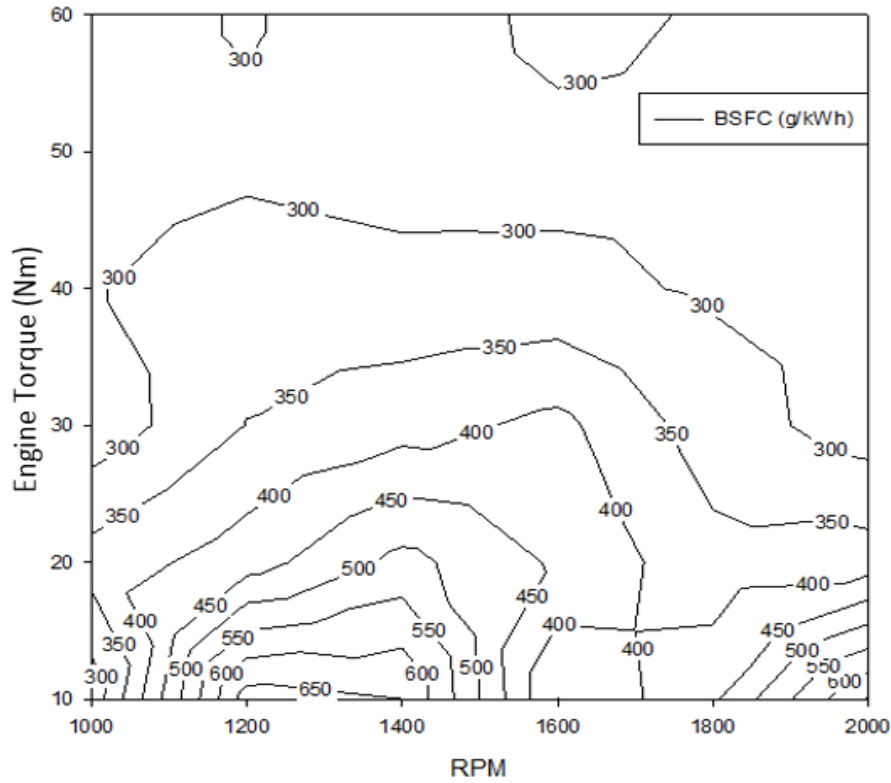


Figure 8. Engine Performance Map from Experimental Results (Engine Torque-RPM-BSFC)

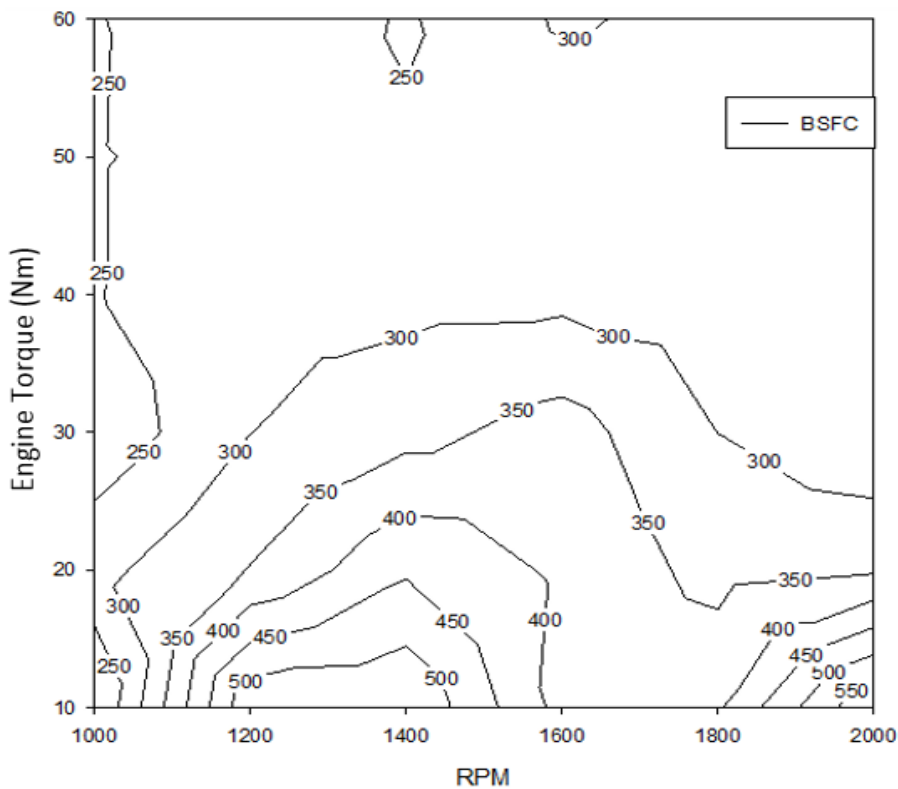


Figure 9. Engine Performance Map from GT-Power Model (Engine Torque-RPM-BSFC)

Note the region of lowest Brake Specific Fuel Consumption (BSFC) (highest efficiency) at a relatively low piston speed and a relatively high Brake Mean Effective Pressure (BMEP). This is where the product of

indicated, mechanical and combustion efficiencies is highest. A broad comparison between the performance maps obtained from practical and simulation results gives a perspective on working differences for further conjectures.

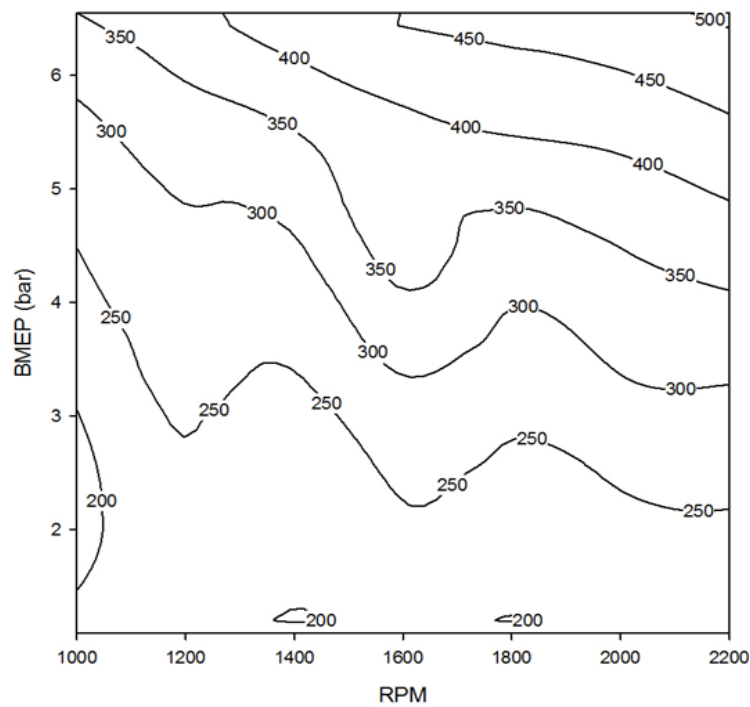


Figure 10. Engine Performance Map from Experimental Data (BMEP-RPM-Exhaust Temperature)

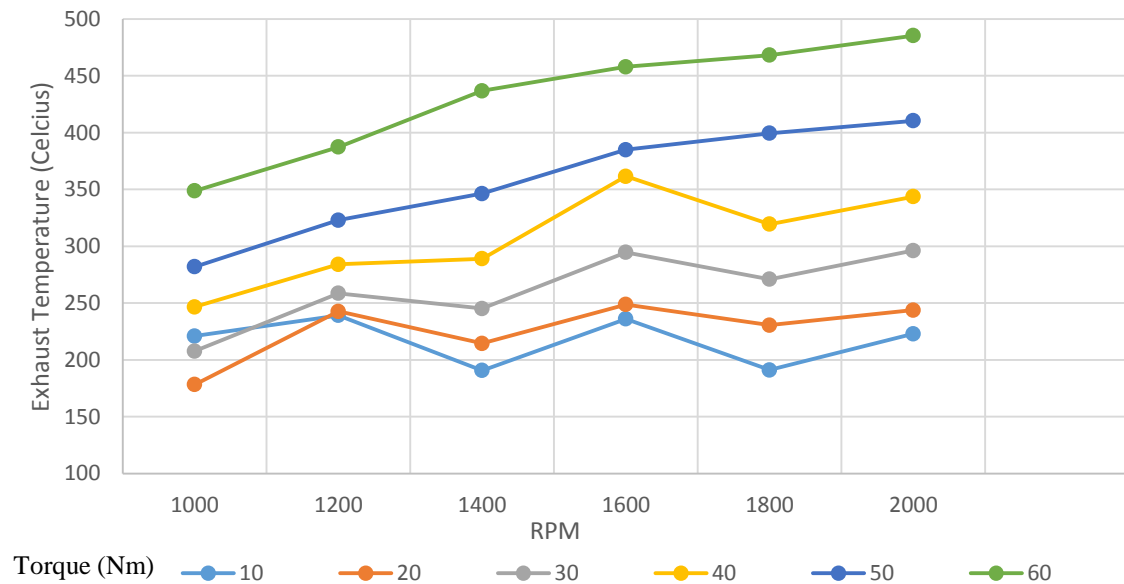


Figure 11. Exhaust Temperature with RPM and Torque (Experimental Results)

The sharp increase in exhaust gas temperature to high loads necessitates specific measures to protect the exhaust gas. This map is very important in calibrating the ignition angle and Exhaust Gas Recirculation (EGR) rates which can lead to an effective operation of the catalytic converters, protecting them from thermal aging at high temperatures and cooling-down at low temperatures. It is very important for the catalytic converters to quickly reach the light-off temperature in order to convert the raw emission to harmless components. The map shown in Figure 10 clearly indicates the relation between Exhaust Temperature, BMEP and RPM of the engine. As the RPM increases, the BMEP is ought to increase. The same trend is also shown in Figure 11, where the

exhaust temperature is seen to increase with an increase in torque and RPM. This means that more fuel combustion will ultimately result in more heat being produced in the engine. Therefore, the indication of higher exhaust temperature with an increase in RPM, BMEP and torque is a true validation of more combustion.

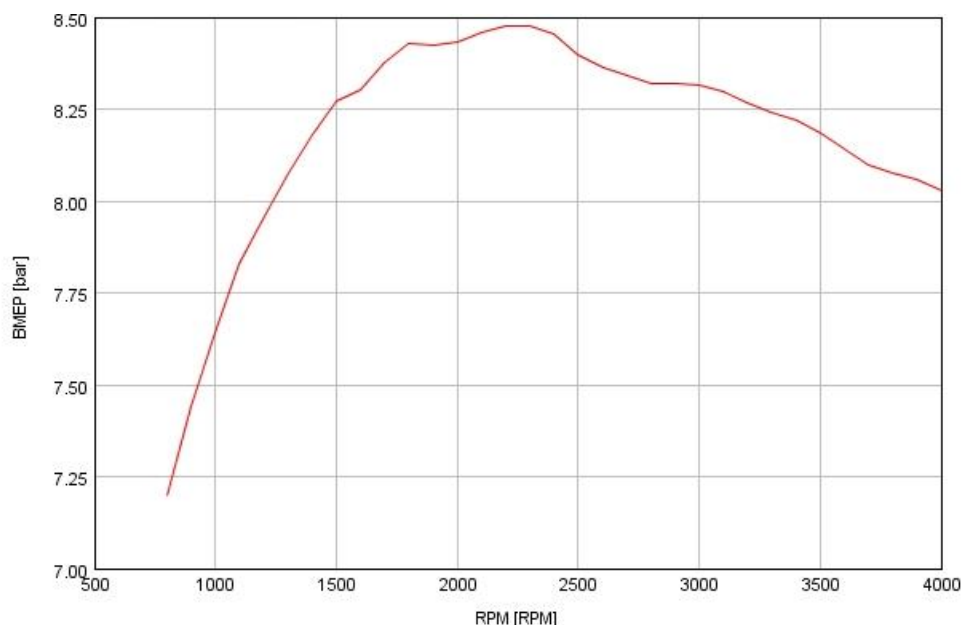


Figure 12. BMEP with RPM obtained from GT-Power Model

The hypothetical constant pressure inside the cylinder acting on the piston during the expansion stroke is the mean effective pressure and produces the same amount of work as of an actual cycle. It is one of the major parameters to determine the engine performance. Top Dead Center Firing (TDCF) is the point that indicates the highest pressure inside the cylinder because this step needs a high pressure to initiate combustion. The highest BMEP is around 8.48 bars at 2250 rpm engine speed. The graph shown in Figure 12 indicates that BMEP increases with an increase in the value of RPM and then starts to decrease after reaching its maximum value of 8.47 bars at 2250 RPM.

The P-v diagram of the simulated engine model was obtained from GT-Power with respect to different engine speeds. P-v diagram for RPMs from 800 till 1200 is shown in the Figure 13 with their particular IVO, IVC, EVO, EVC, SOI and SOC for every 200 rpm interval. Figure 14 diversifies the experimental and simulation results of applied torque in characteristic with output power and RPM. The solid lines in the Figure represent the practical results while the dotted lines represent the data obtained from the simulation model. For low RPMs, the difference between the two cases seemed to be within an average of 25 % range, while with an increase in engine speed (RPM) the lines tend to get closer to each other. At RPM value of 2,000 the difference between them remained to be within 5%. The results shown for applied torques of 30 Nm (grey lines) and 40 Nm (yellow lines) seemed to show the greatest difference between practical and simulated results.

The trend of power with rpm of the Engine is described by the graph shown in Figure 15. The red line indicates the data from GT-Power while the blue line indicates the results obtained from experimental data. The dashed line shows predicted trend for higher values of RPM for experimental data.

This prediction is made because the practical experimentation was done until the practically maximum achievable RPM value of 2,400. Power increases with engine speed and starts to decrease at a very high engine speed values, usually one and a half times the speed of maximum torque. Normally the power of diesel engines tend to show a linearly increasing trend until 5,000-6,000 RPM, after which a decrease is expected.

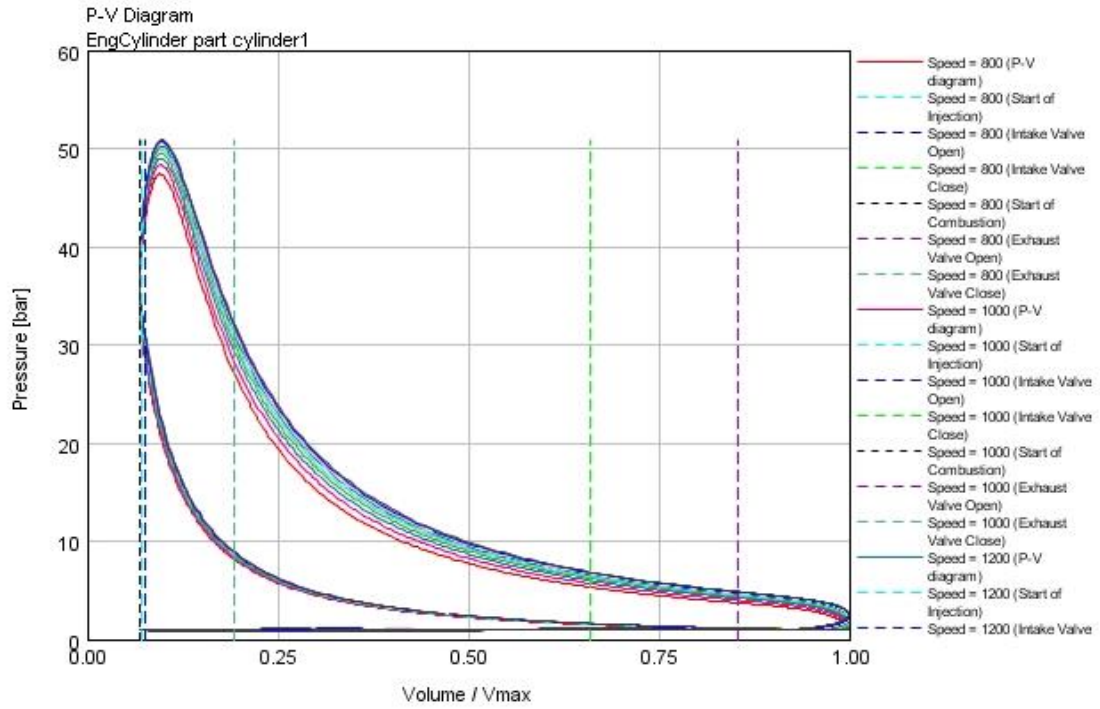


Figure 13. P-v Diagram of the model from GT-Power

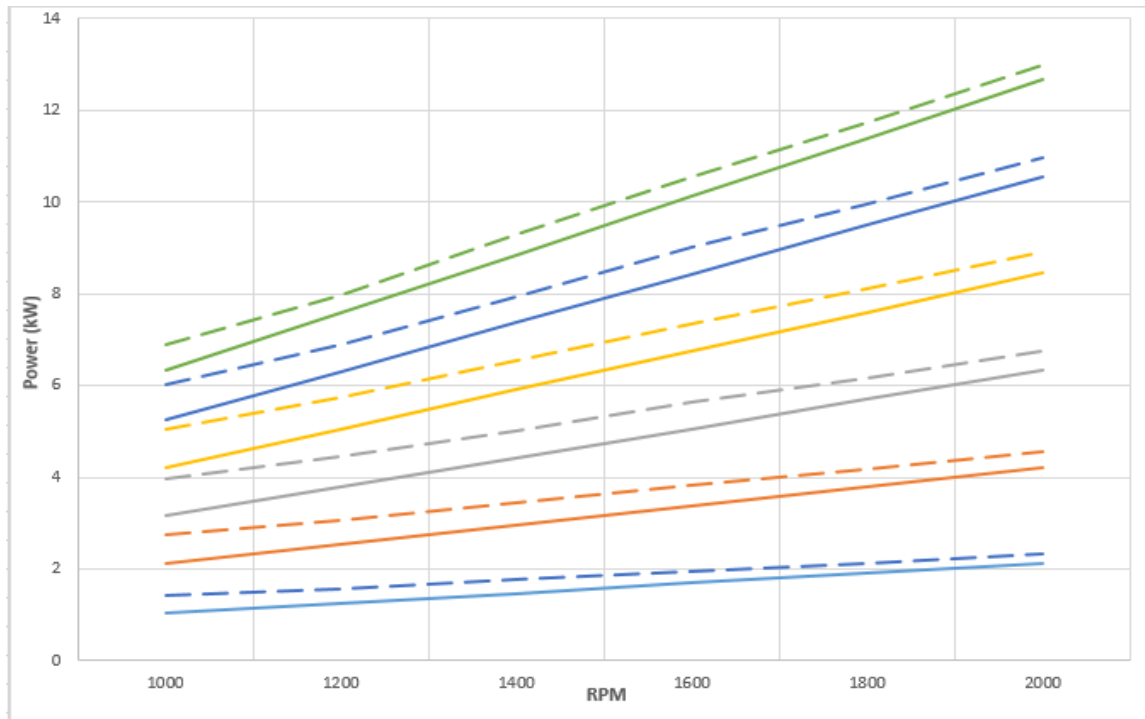


Figure 14. Relation of Power, RPM and Applied Torque



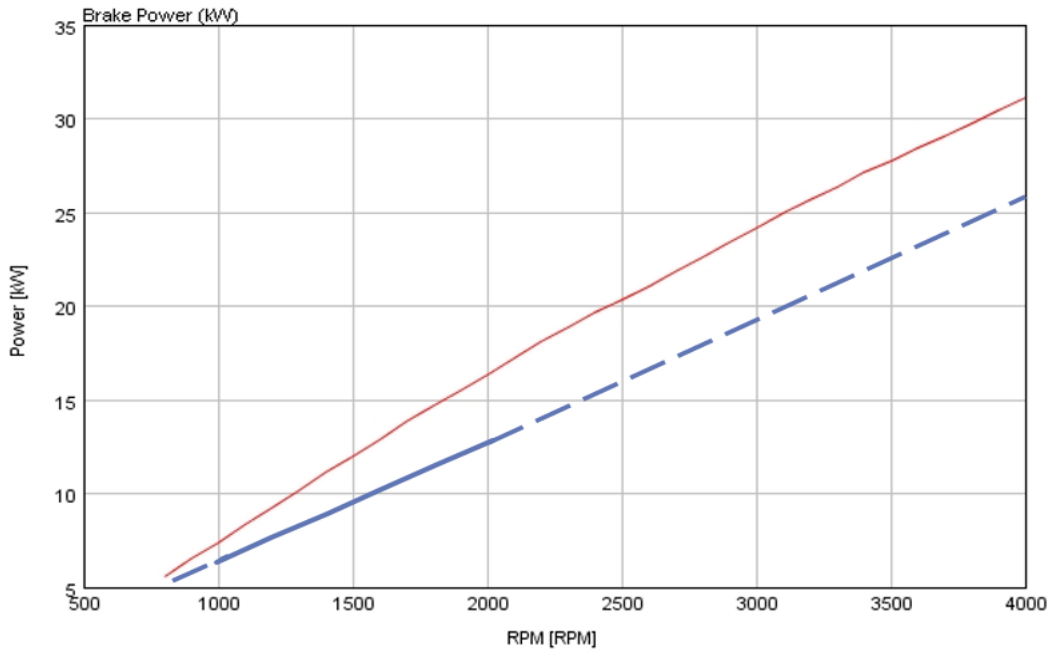


Figure 15. Brake Power with RPM from Simulation and Experimental Results

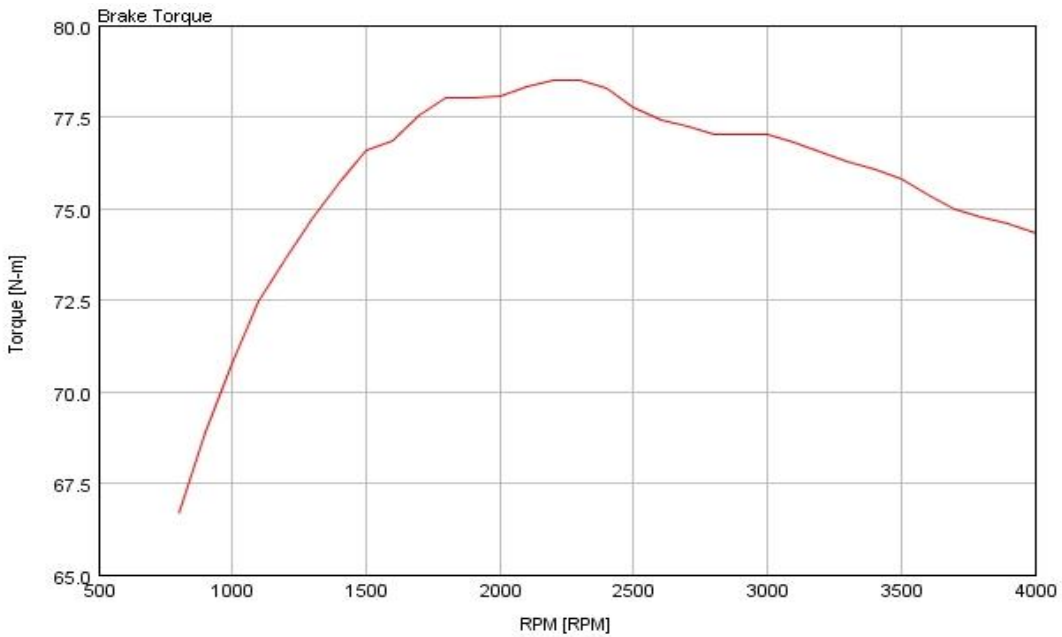


Figure 16. Brake Torque with RPM from GT-Power Model

By definition, brake torque is the amount of torque delivered by the engine to the shaft. Over a full engine speed, Maximum Brake Torque (MBT) helps a designer by indicating the ability of obtaining a higher air flow over maximum rpm for an effective use of air. The torque of the engine is directly measured from the dynamometer. This reading is the measure of the Engine’s ability to perform work. The brake torque of the engine from our simulation model is shown in Figure 16 as brake torque.

The MBT was found to be at 2,250 rpm of engine speed. Before reaching the maximum value the brake torque increases with an increase in RPM. It then starts to decrease for values higher than 2250 RPM. It is demonstrated in figure 17. This happens because at higher speeds the engine is unable to gulp a complete charge of air. Another factor contributing to this trend is frictional losses. Frictional forces increase with an increase in the speed and become more prominent at higher RPM values.

Brake specific fuel consumption decreases as engine speed increases, reaches a minimum, and then increases for high RPM values. At low speeds, a longer time per cycle allows more heat loss. Therefore, the fuel consumption goes up. Fuel consumption increases at high speeds because of greater friction losses. As our engine has compression ratio of 14.6, therefore it can be categorized in terms of having a medium Thermal Efficiency. This value of compression ratio (CR) is important as it has a significant effect on BSFC values. A higher CR means more compression, which in turn means better ignition/combustion inside the cylinder. Thus, BSFC decreases with an increase in CR. It also decreases with higher CR due to higher thermal efficiency.

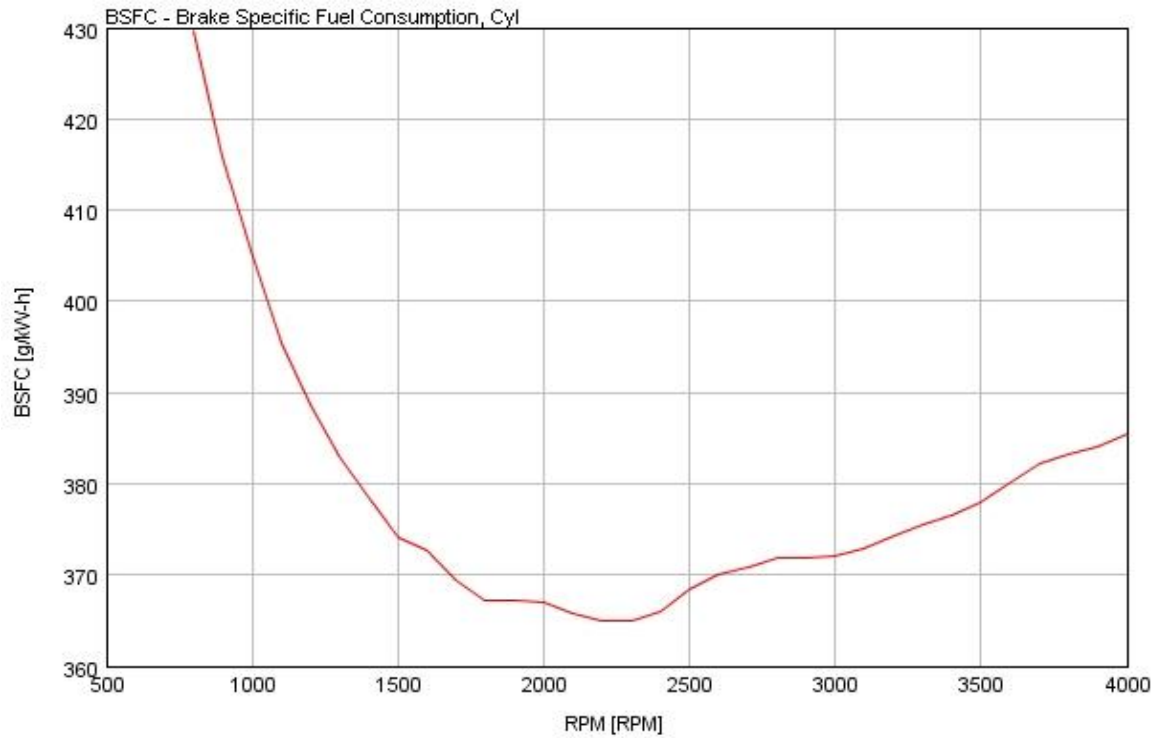


Figure 17. BSFC with RPM from GT-Power Model

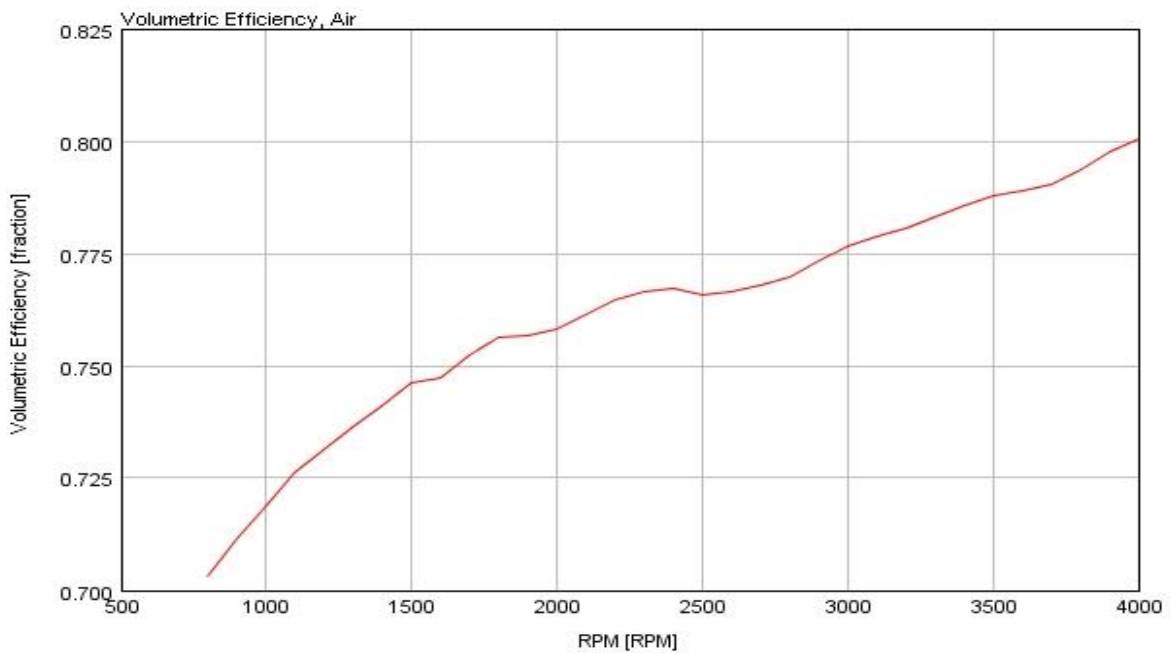


Figure 18. Volumetric Efficiency with RPM from GT-Power

Volumetric efficiency (VE) is actually a measure of the productivity of the engines induction process. This can be directly defined as the ratio of air volume flow rate into the intake system to the rate of volume displacement by the piston. In Figure 18, the volumetric efficiency is seen to increase with engine speed, apart from showing some fluctuations. This is because as the engine speed increases, the rate of air suction in the cylinders also increases exponentially with respect to the valve opening time which results in the charging of the cylinder with a greater volume of air. As shown the volumetric efficiency at 800 RPM is shown to be around 70.50% (0.705) and at 4,000 RPM it is about 80.00 % (0.800). As VE depends greatly on the throttle opening, induction and exhaust system layout, port size, valve opening duration and valve timing, therefore, these fluctuations are a result of a combination of all these factors at different engine speeds. At very high RPM values (around 5,000–6,000) the VE is expected to decrease with further increase in RPM due to air flow resistance caused by:

- Air filter resistance
- Throttle damper resistance
- Aerodynamic resistance of manifolds
- Intake valve resistance

Table 3 shows the details of the fuel used in our engine during the experiments. The same values of fuel was entered in GT-Power for our simulation model. As seen from the table the Heat of Vaporization of the fuel was 270 kJ/kg which was taken into consideration for the calculation of Power Input given to the engine.

Table 3. Fuel Properties

Euro Light Diesel	
Molecular Form	C _{12,3} H _{22,2}
Molecular Weight	170
Heating Value – LHV (kJ/kg)	42500
Stoichiometric (AF)_s	14.5
Stoichiometric (FA)_s	0.069
Heat of Vaporization (kJ/kg)	270
Cetane Number	40-55

The Power input to the engine at different RPM and applied torque is shown in Figure 20. With some minute fluctuations, the graph clearly represents an increase in input power with an increase in applied torque and RPM. This phenomena matches with the theoretical logic as increase in RPM and in applied torque demands more fuel intake in the cylinder for higher combustion as represented by Figure 19. Thus, a higher fuel intake represents a higher power input.

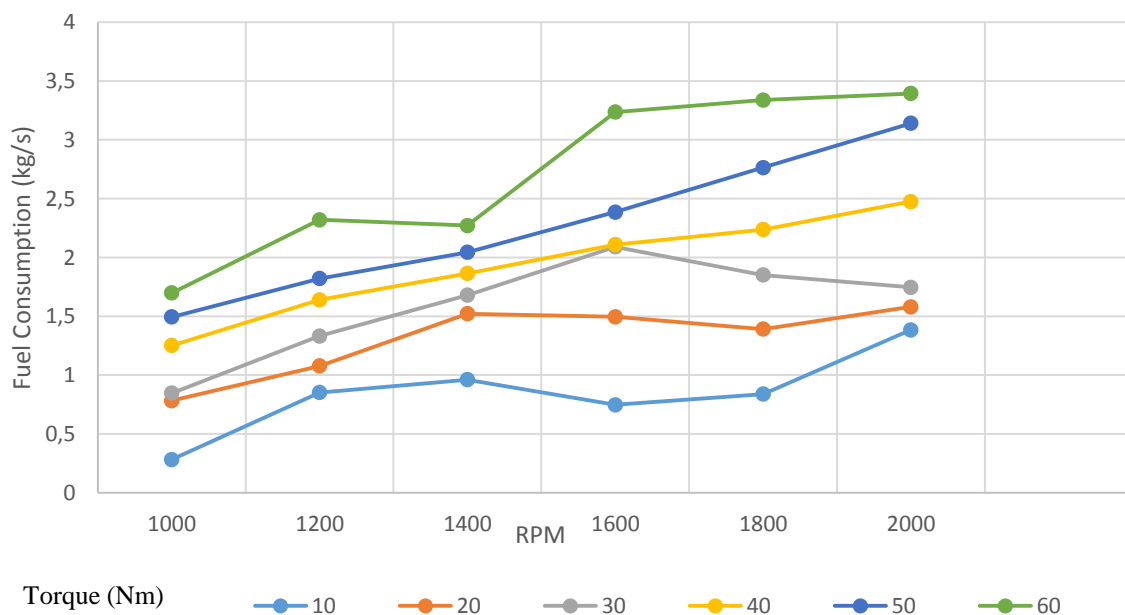


Figure 19. Relation of Fuel Consumption, RPM and Torque from Experimental Results

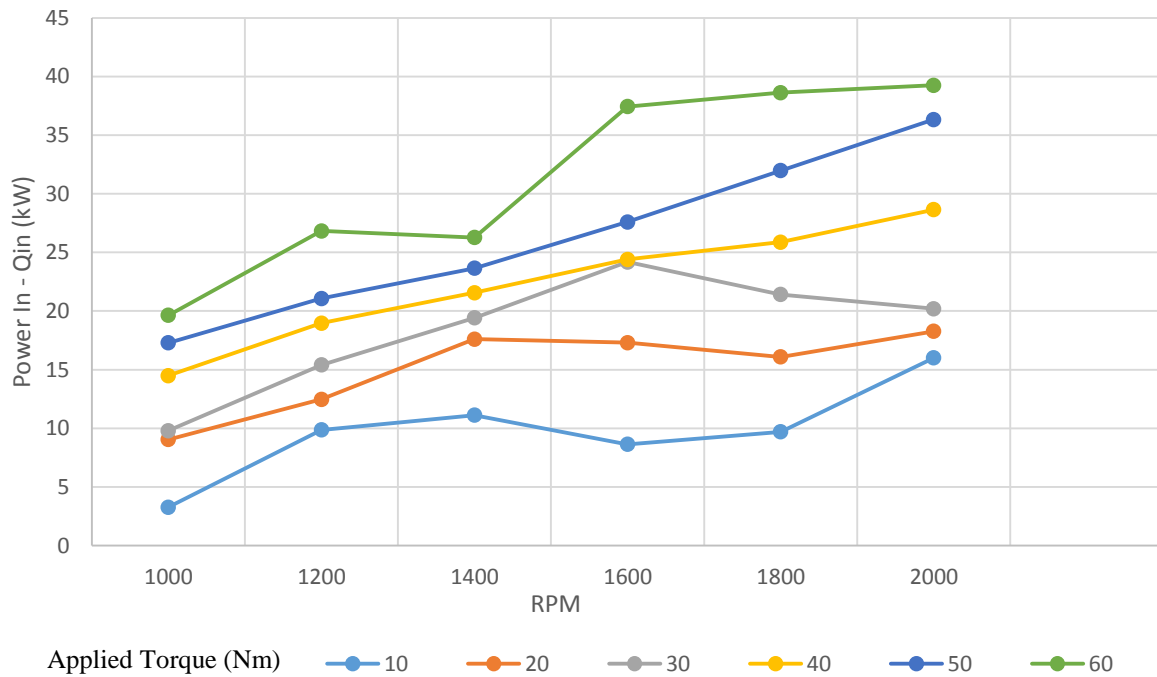


Figure 20. Relation of Power In, RPM and Applied Torque from Experimental Results

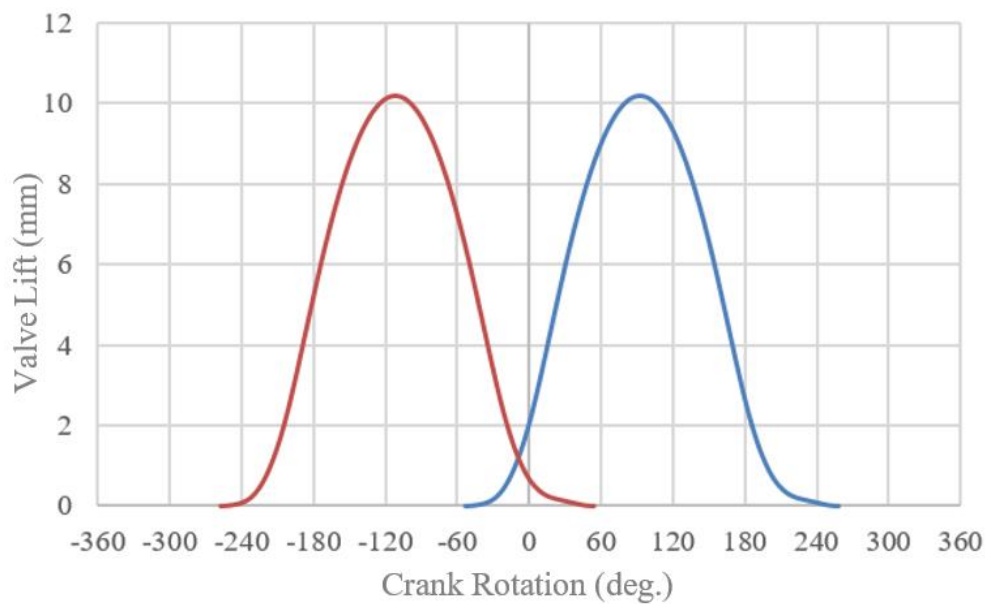


Figure 21. Valve Lift Profile of the Engine

The valve lift profile was measured experimentally in the laboratory from real intake and exhaust cams on the cam shafts. The data was then plotted and the two cam profiles (intake and exhaust) were linked with each other on a single graph. The red represents the exhaust valve profile while the blue represents the intake valve profile and it can be seen in figure 21.

The maximum valve lift for Erin engine was given to be 11 mm while practically it was recorded to be 10.40 mm. The intake valve opening crank angle was measured to be 54° while the intake valve closing crank angle was shown to be 74°. The exhaust valve opening crank angle was 74° and the exhaust valve closing crank angle was 54°. The crank angle at the start of injection appeared to be 19° Before Top Dead Centre (BTDC).

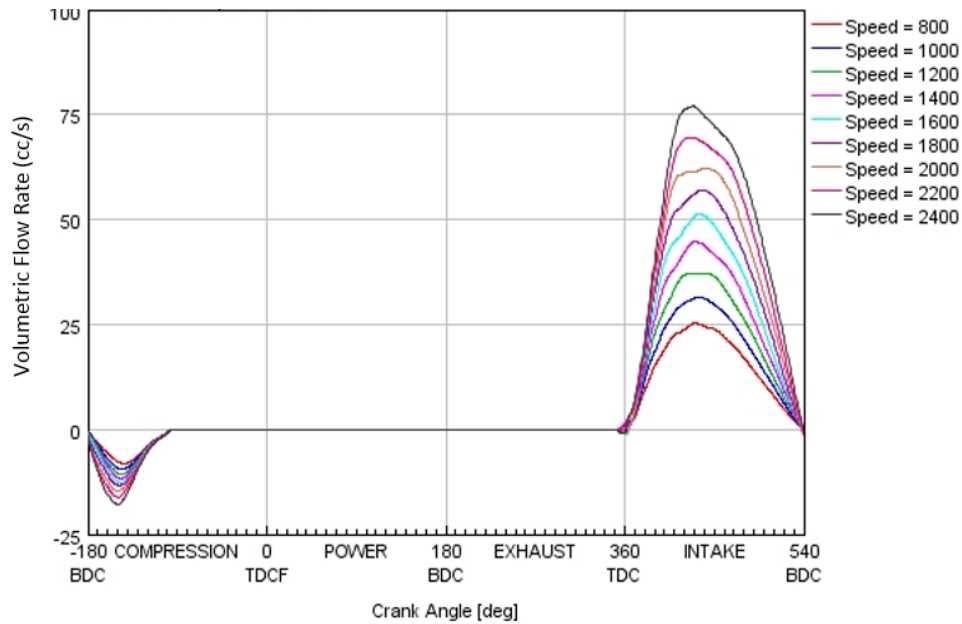


Figure 22. Volumetric Flow with Crank Angle from GT-Power Model

The mass flow rate of the intake valve plotted against crank angle obtained from our GT-Power model of the engine is shown in Figure 22. This figure helps us to understand the valve actions during the intake, compression, power and exhaust stroke. The figure clearly indicates that the volumetric flow rate starts to increase from TDC, reaches a maximum and then starts to decrease until it reaches zero at BDC. This happens because the valve starts to open at TDC, then starts to close after reaching the maximum valve lift and closes fully at BDC, where the cylinder has the maximum volume. It shows negative values of volumetric flow during the compression stroke which can be explained by the flow of the mixture out of the intake stroke when the compression has already begun and the intake valve is still closing.

A higher engine speed means more combustion and this ultimately means more air flow in the cylinder. The same trend is shown by the graph in Figure 22.

VIBRATION ANALYSIS

Diesel engine noise and vibration can create harmful effects on hearing and user's body. This is specially observed in the engines with high compression ratios and fast rising combustion pressures.

This has a wide scope in future as it can also be used to detect the injector fault detection. Early detection of diesel engine faults is essential in order to take early corrective measures and avoid expensive repairs. Vibration measurement on the engine block or cylinder heads a non-intrusive method and has been used successfully for fault detection of diesel engines.

A wireless accelerometer was fixed at the head of the engine to record the readings of z-axis (Vertical) Acceleration of the Engine. The Accelerometer had the capacity to record 3200 values of acceleration in one second. The data from the accelerometer was sent wirelessly to the computer using wireless adapters. The device was powered using a portable battery and operated on low-power operation. The vibrational data was recorded and transferred efficiently without any errors in the readings or tussle of wires.

The vibration data was recorded between the RPM values of 1,000 and 2,000 at every 100 RPM interval. Vibration and Fast Fourier Transform (FFT) Graph was made using MATLAB.

According to Figure 23, the optimum operational RPM of the engine is shown to be 1800 RPM. The engine was seen to operate very smoothly at this particular RPM. Figure 24 also clearly indicates that the most smooth transition between Thermal Efficiency and Brake Power is shown at 1800 RPM which was indicated previously to be the optimum RPM of the engine. Table 4 shows the acceleration and Fast Fourier Transform (FFT) plots.

Optimum Operational RPM (No Load)

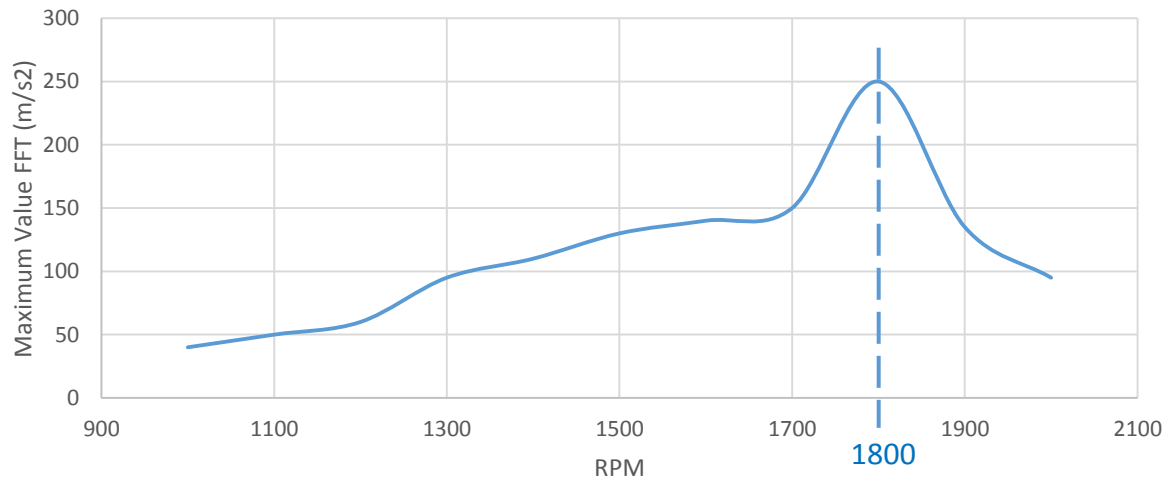


Figure 23. Maximum FFT values with RPM

Table 4. Demonstration of Acceleration and FFT plots

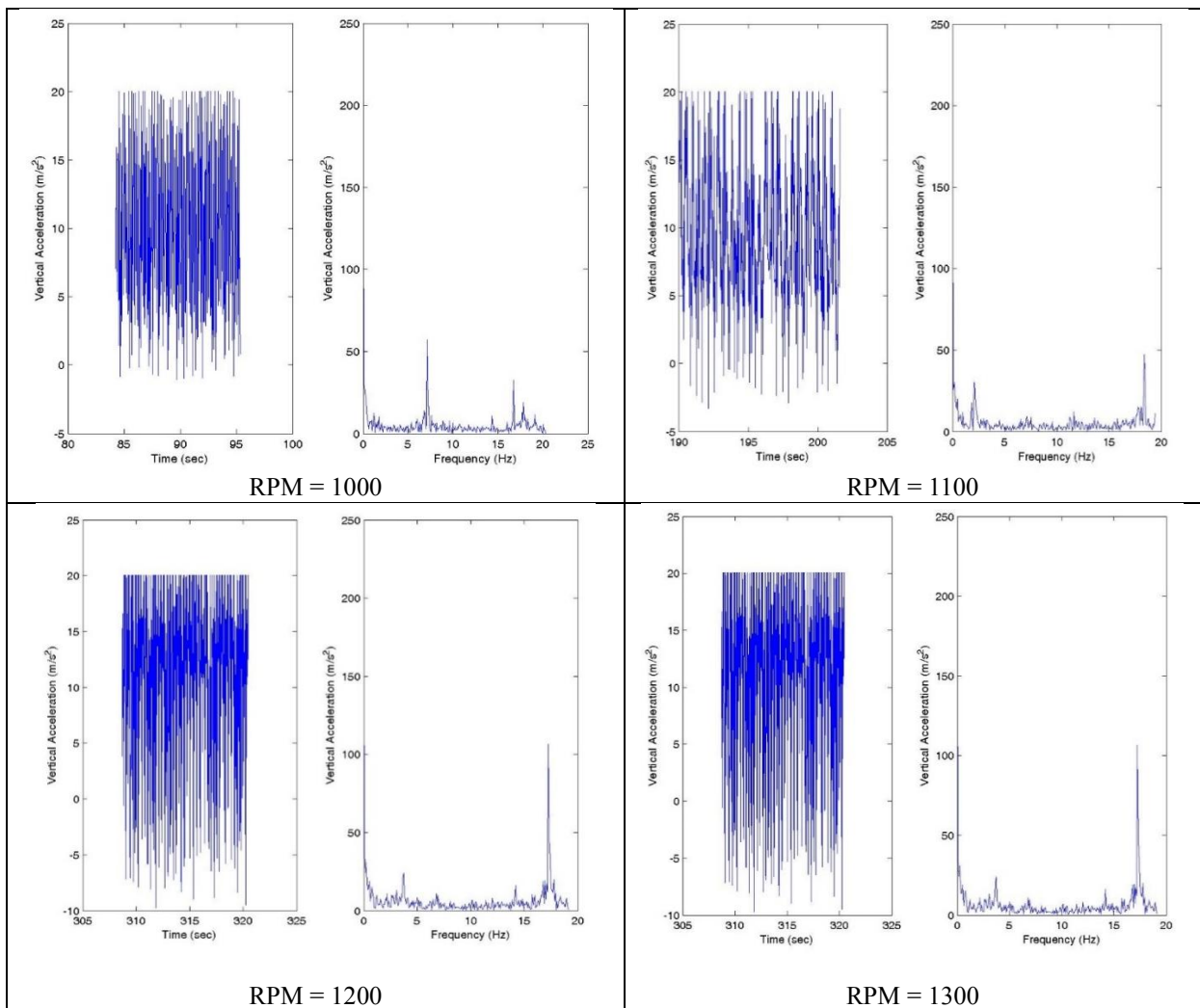


Table 5. Demonstration of Acceleration and FFT plots (Continued)

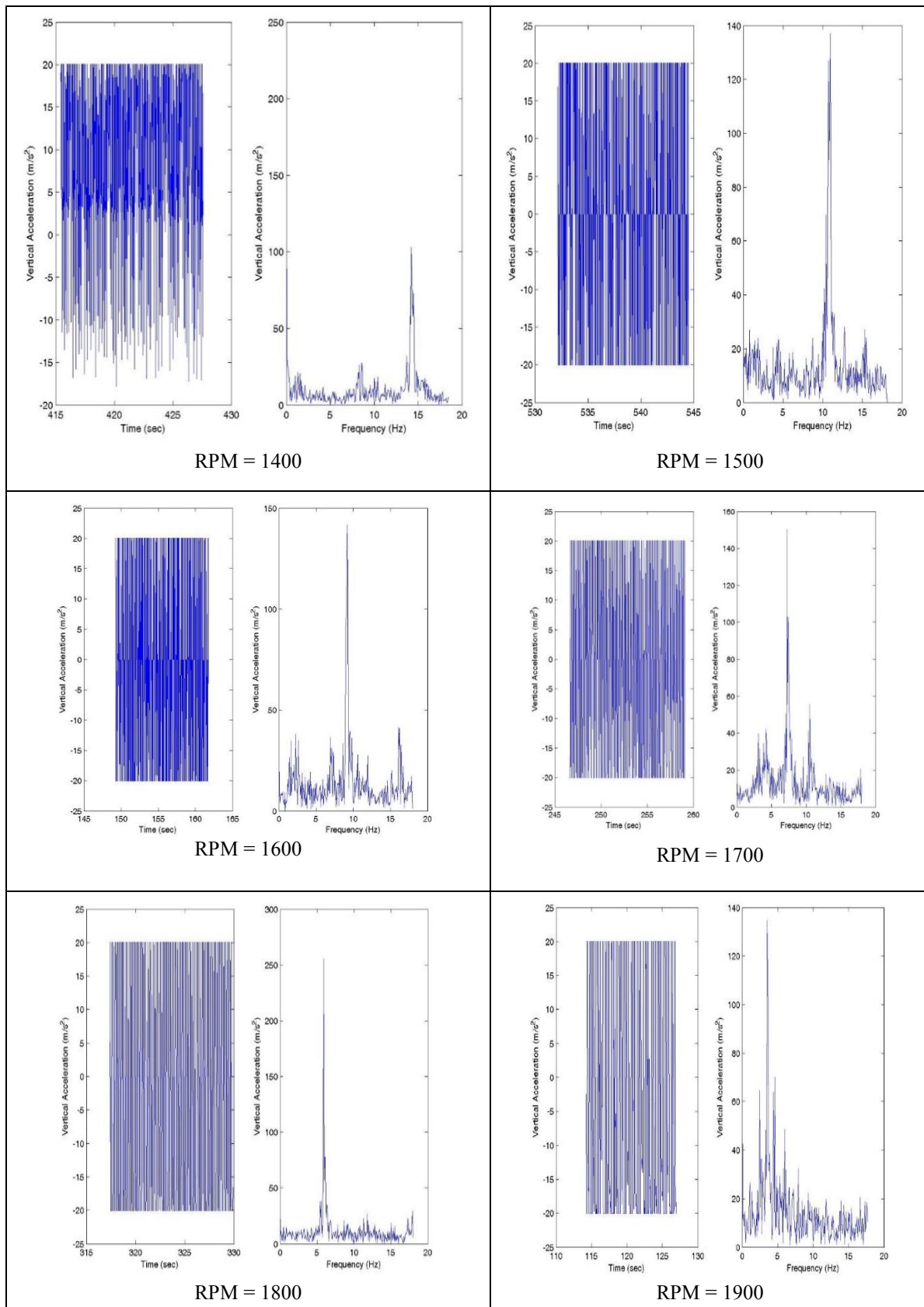


Table 6. Demonstration of Acceleration and FFT plots (Continued)

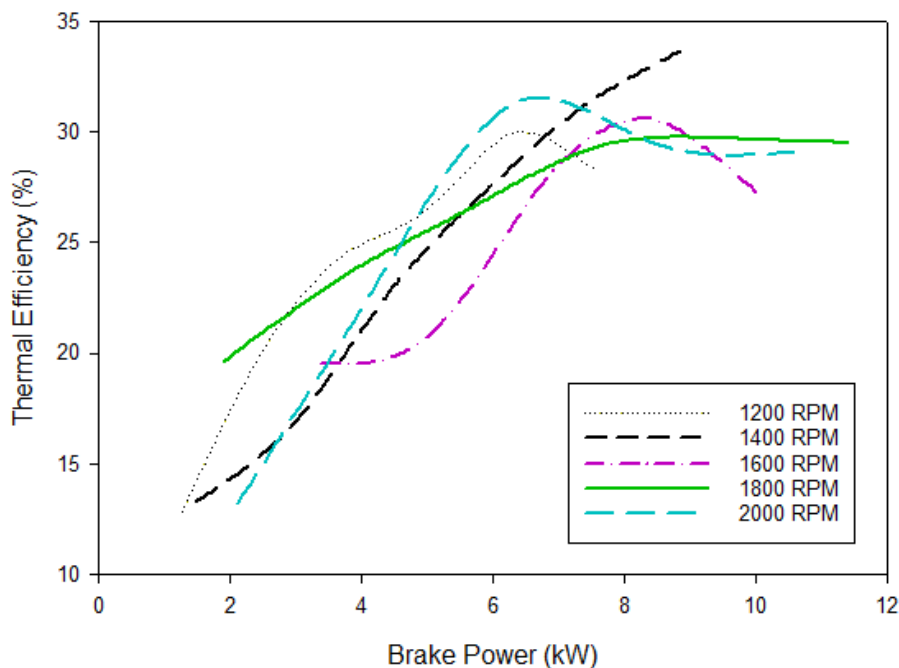
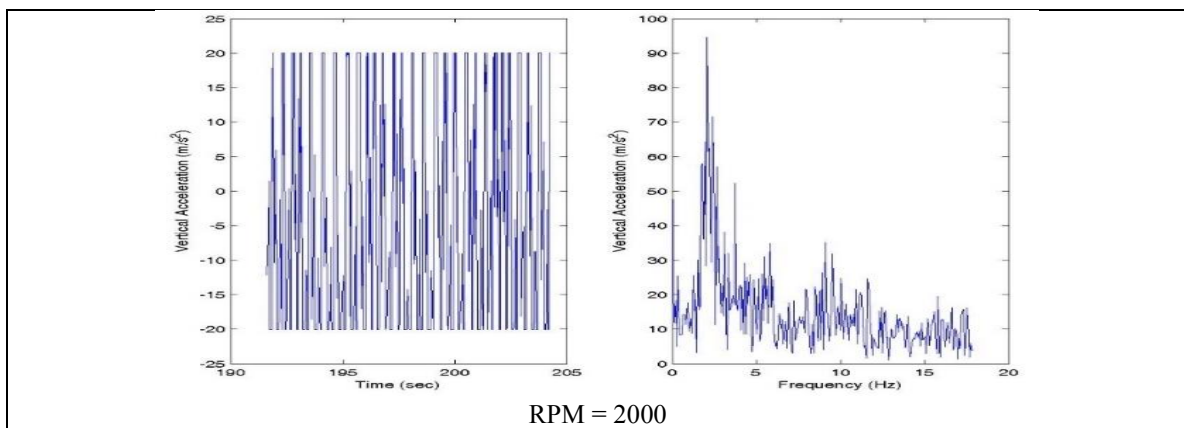


Figure 24. Relation of Thermal Efficiency with Braking Power and RPM

CONCLUSION

In this investigation, the major differences between the trend lines of the graphical results of experimental and simulation results were studied.

Simulated results appear to be near the experimental results but not necessarily follow the trend shown by a practical engine. This happens because the condition of the practical engine changes under extreme low or extreme high conditions. The results appear to be closest to each other near the optimum operating conditions. The results of the simulation widely depend on every small detail entered in the simulated model. Brake power is the feature that can be compared in both the situations and appeared to be 11.3 kW for experimental case while 13.2 kW for the simulated case at the optimum RPM value of 1,800.

The trend of engine performance maps for both cases appeared to be in correlation with each other apart from some anomalies shown in the practical results for low BSFC values.

The maximum torque of the 1.16 L Erin Engine is given to be 80 Nm @ 1,800 RPM, while the simulation model indicated it to be 78 Nm at the same RPM value. Likewise, maximum power output was indicated to be 18 kW @ 2,400 RPM, while the experimental results showed it to be 15 kW @ 2,400 RPM.

The vibration analysis of the engine proved to be very accurate in its purpose of determining the optimum RPM. The optimum engine speed at which the engine performs very smoothly and efficiently was given to be

1,800 RPM. This value was cross-matched with the previous results of engine performance and it was clarified that the engine performed the best at this value of RPM. Vibration analysis along with noise analysis can further be used to predict the faults and changes in the engine behavior, in future studies.

ACKNOWLEDGMENTS

I would like to express sincere gratitude to Prof. Dr. Mehmet Zafer GÜL for the continuous support. I am extremely thankful and indebted to him for sharing his expertise, sincere and invaluable guidance and encouragement extended to me.

Also I would like to thank my family for their encouragement with support of all kinds and their unconditional affection.

NOMENCLATURE

AFR	Air-Fuel Ratio
BDC	Bottom Dead Center
TDC	Top Dead Center
ABDC	After Bottom Dead Center
BBDC	Before Bottom Dead Center
ATDC	After Top Dead Center
BTDC	Before Top Dead Center
CN	Fuel Cetane Number
EGR	Exhaust Gas Recycle
IVO	Intake Valve Opening
IVC	Intake Valve Closing
EVO	Exhaust Valve Opening
EVC	Exhaust Valve Closing
MEP	Mean Effective Pressure
SFC	Specific Fuel Consumption
BSFC	Brake Specific Fuel Consumption
ISFC	Indicated Specific Fuel Consumption
BMEP	Brake Mean Effective Pressure
TC	Top-center crank position
ATC	After TC
BTC	Before TC
RPM	Revolutions per minute

REFERENCES

- [1] Kagawa J. Health effects of diesel exhaust emissions—a mixture of air pollutants of worldwide concern. *Toxicology* 2002; 181:349–53.
- [2] Du, Jiakun, Wanchen Sun, Liang Guo, Senlin Xiao, Manzhi Tan, Guoliang Li, and Luyan Fan. "Experimental study on fuel economies and emissions of direct-injection premixed combustion engine fueled with gasoline/diesel blends." *Energy Conversion and Management* 100 (2015): 300-309.
- [3] G. P. Merker, B. Hohlbaum, M. Rauscher, Two-zone model for calculation of Nitrogen-Oxide formation in direct-injection diesel engines, SAE paper 932454, 1993.
- [4] R. J. Asay, K. I. Svensson, D. R. Tree, An Empirical, Mixing-Limited, Zero- Dimensional Model for Diesel Combustion, SAE paper 2004-01-0924, 2004.
- [5] D.T. Hountalas, Prediction of marine diesel engine performance under fault conditions, *Applied Thermal Engineering* 20 (2000) 1753e1783.
- [6] D. Descieux, M. Feidt, One zone thermodynamic model simulation of an ignition compression engine, *Applied Thermal Engineering* 27 (2007) 1457e1466.
- [7] X. Tauzia, A. Maiboom, P. Chesse, N. Thouvenel, A new phenomenological heat release model for thermodynamical simulation of modern turbocharged heavy duty, *Diesel Engines* 26 (2006) 1851e1857.

- [8] S. Fabio, S. H. Stefansson, F. Haglind, A. A, and U. Larsen. "Validation of a zero-dimensional model for prediction of NO_x and engine performance for electronically controlled marine two-stroke diesel engines." *Applied Thermal Engineering* 37 (2012): 344-352.
- [9] V. Ganesan, *Internal combustion engines*. McGraw Hill Education (India) Pvt Ltd, 2012.
- [10] V. Çelik, and E. Arcaklioğlu. "Performance maps of a diesel engine." *Applied Energy* 81, no. 3 (2005): 247-259.
- [11] Gamma Technologies, 2004. *GT-Power User's Manual Version 6.1*, Gamma Technologies Inc.
- [12] İc, ingü r Y, Altıparmak D. Effect of fuel cetane number and injection pressure on a diesel-engine_s performance and emissions. *Energy Convers Manage* 2003;44:389–97.
- [13] Y. Zhu, R. Stobart, and J.Deng. Analysis of the impact on diesel engine fuel economy and emissions by variable compression ratio using GT-Power simulation. No. 2010-01-1113. SAE Technical Paper, 2010.
- [14] Z Şahin, and O. Durgun. "Multi-zone combustion modeling for the prediction of diesel engine cycles and engine performance parameters." *Applied Thermal Engineering* 28, no. 17 (2008): 2245-2256.
- [15] Scappin, Fabio, Sigurður H. Stefansson, Fredrik Haglind, Anders Andreasen, and Ulrik Larsen. "Validation of a zero-dimensional model for prediction of NO_x and engine performance for electronically controlled marine two-stroke diesel engines." *Applied Thermal Engineering* 37 (2012): 344-352.
- [16] Arrègle, J., J. J. López, J. M. García, and C. Fenollosa. "Development of a zero-dimensional diesel combustion model. Part 1: analysis of the quasi-steady diffusion combustion phase." *Applied Thermal Engineering* 23, no. 11 (2003): 1301-1317.
- [17] Catania, Andrea Emilio, Roberto Finesso, and Ezio Spessa. "Predictive zero-dimensional combustion model for DI diesel engine feed-forward control." *Energy Conversion and Management* 52, no. 10 (2011): 3159-3175.
- [18] T.Sasa, P. Tunestål, and B.Johansson. Simulation of a pneumatic hybrid powertrain with VVT in GT-power and comparison with experimental data. No. 2009-01-1323. SAE Technical Paper, 2009.
- [19] C. Ornella, G. Chiatti, L.Arnone, and S. M. Combustion characterization in diesel engine via block vibration analysis. No. 2010-01-0168. SAE Technical Paper, 2010.
- [20] G. Chandroth, *Acoustic emission, cylinder, pressure and vibration: A multisensory approach to robust fault diagnosis*, Ph.D. Thesis, 1999, UK
- [21] J. Antoni, J. Danière, F. Guillet, Effective vibration analysis of IC engines using cyclestationary. Part I: A methodology for condition monitoring, *Journal of Sound and Vibration* (2002) 257 (5), 815-837.
- [22] Z. Geng, J. Chen, J.B. Hull, Analysis of engine vibration and design of applicable diagnosing approach, *International Journal of Mechanical Sciences* 45 (2003) 1391-1410.
- [23] A. P. Carlucci, F. F. Chiara, D. Laforgia, Analysis of the relation between injection parameter vibration and block vibration of an internal combustion Diesel engine, *Journal of Sound and Vibration* 295 (2006) 141-164.
- [24] G., Zunmin, J. Chen, and J. B. Hull. "Analysis of engine vibration and design of an applicable diagnosing approach." *International Journal of Mechanical Sciences* 45, no. 8 (2003): 1391-1410.
- [25] Ajovalasit, M., and J. Giacomini. "Analysis of variations in diesel engine idle vibration." Part D: *Journal of Automobile Engineering* 217, no. 10 (2003): 921-933.
- [26] T.A. Ahmad, B.Ghobadian, T. Tavakoli-Hashjin, and S. S. Mohtasebi. "Vibration analysis of a diesel engine using biodiesel and petrodiesel fuel blends." *fuel* 102 (2012): 414-422.