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SIMULATION STUDY ON EFFECT OF GAS CHARGING AND EGR IN A DUAL-FUEL ENGINE

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SIMULATION STUDY ON EFFECT OF GAS CHARGING AND EGR IN A DUAL-FUEL ENGINE

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This report has been approved in partial fulfillment of the requirements for the Degree of MASTER OF SCIENCE in Mechanical Engineering.

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List of Abbreviations

Acronyms

aTDC	After Top Dead Center
BMEP	Brake Mean Effective Pressure
BDC	Bottom Dead Center
BSFC	Brake Specific Fuel Consumption
bTDC	Before Top Dead Center
BTE	Brake Thermal Efficiency
CAC	Charge Air Cooler (Intercooler)
CAD	Crank Angle Degree
CNG	Compressed Natural Gas
Cyl	Cylinder
DPF	Diesel Particulate Filter
EGR	Exhaust Gas Recirculation
EID	Electronic Injection Duration
EPA	Environmental Protection Agency
ESC	European Standard Cycle
EVC	Exhaust Valve Closing
EVO	Exhaust Valve Opening
FMEP	Friction Mean Effective Pressure
GHG	Greenhouse Gases
HCCI	Homogeneous Charge Compression Ignition
HRR	Heat Release Rate
IMEP	Indicated Mean Effective Pressure

MAP	Manifold Absolute Pressure
MJ	Mega Joules
mg	milligrams
ms	milliseconds
NG	Natural Gas
PCCI	Premixed Charged Compression Ignition
PCP	Peak Cylinder Pressure
PF	Peak Cylinder Pressure Factor
PM	Particulate Matter
PPM	Parts Per Million
RPM	Rotations Per Minute
SCR	Selective Catalytic Reduction
SF	Mean Piston Speed Factor
SOI	Start Of Injection
SOC	Start Of Combustion
SSF	Mean Piston Speed Squared Factor
TDC	Top Dead Center
TDCF	Top Dead Center for Firing Stroke
TWC	Three Way Catalyst
Vs	Versus
WHSC	World Harmonized Stationary Cycle

Symbols

±	Plus or Minus
%	Percentage
$ma\dot{s}s_{air}$	Mass flow rate of air
$m_{d\dot{i}esel}$	Mass flow rate of diesel
$ma\dot{s}s_{EGR}$	Mass flow rate of EGR
$\dot{m}_{NG}$	Mass flow rate of Natural Gas
P_{max}	Maximum Cylinder Pressure

Abstract

Natural gas combined with diesel as micro pilot has the capabilities of achieving lower NO_x and soot emissions.[1] Optimization of the combustion process in engines with natural gas and diesel micro-pilot is essential to achieve higher efficiencies and loads.[2] Gas charging (intake air boosting) and EGR are two technologies which when implemented in the natural gas-diesel engines, provide the opportunity to achieve higher efficiencies and loads and low emissions.[3] Simulation study is one of the approaches to investigate the extent and effects of gas charging and EGR on the performance of the engine. With the rapid improvements over the past decade in the field of engine simulation and modeling, it has become an efficient, economical and reliable approach [4]. GT-Suite, one of the widely-used Vehicle and Engine Simulation tools in the industries, provides the capabilities to calculate the combustion rate in Internal combustion engines for conventional as well as dual-fuel engines.

Current research work uses GT-Suite software to study the effect of gas charging and EGR on a Cummins 2010 ISB 6.7 L engine in dual-fuel mode. One-dimensional simulation model for a Cummins 2010 ISB 6.7 L engine is developed by acquisition of dimensions from the engine. The simulation model is calibrated with the experimental data available from the diesel engine. The calibrated model is then developed into a dual-fuel model which is used to study the effect of EGR for diesel energy contribution percentages of 1,

3, 5 and 10 and injection pressures of 300,600 and 1000 bar at diesel injection timings of 0° and 10° bTDC and a boost pressure of 2.5 bar. EGR levels were varied from 0-18%. Based on the simulation results for the test conditions, cases for lowest BMEP at 0° bTDC and 10° bTDC were selected and a boost pressure sweep was performed from 2.5 bar to 3 bar to study the effect of gas charging.

The simulation results proved that the target BMEP of 25 bar and fuel conversion efficiency of up to 41% could be achieved in dual fuel mode for the Cummins 6.7L engine.

1 Introduction

Internal combustion engines have been subjected to stringent emission regulations over the last few decades. Diesel engines, being efficient prime movers, are used for economic power generation in most of the world's equipment and vehicles and are one of the largest contributors of increasing greenhouse gases in the environment. [5] US Environmental Protection Agency (EPA) and National Highway Traffic Safety Administration (NHTSA) have set the goal to regulate greenhouse gases to 6 billion metric tons for the vehicle lifetime for vehicle models from 2012-2025. [6]

To meet the emission standards in diesel engines, combined technologies of In-cylinder combustion and after-treatment systems must be implemented.[7] In the after-treatment, novel technologies like combination of after-treatment modules such as SCRF (Combination of SCR and DPF) are being tested. Some of the present challenges for the after-treatment technologies include cost reduction, cold phase emission reduction and optimization of control strategy for Urea-SCR technology [7-9]. In-cylinder combustion strategies provide the opportunity to improve thermal efficiencies along with reducing emissions. Many new concepts for Compression Ignition (CI) engines have been proposed over the last two decades. Homogeneous charge compression ignition (HCCI) has been researched extensively because of the scope to achieve higher thermal efficiencies and near zero NO_x and soot emissions. Lower exhaust temperatures in HCCI when compared to

other combustion strategies makes it difficult to achieve higher boost pressures.[2] Combustion control is another aspect of challenge in HCCI combustion.[10] To reduce NO_x and soot emissions along with greenhouse gases and to achieve higher thermal efficiencies, dual-fuel engine combustion is extensively being researched. High octane number, availability in nature and wide use make natural gas an excellent fuel to be used in Dual-fuel engines [1]. Natural Gas-Diesel combination must be extensively studied to optimize the process of combustion to achieve higher loads and higher efficiencies [11]. One area of interest would be to achieve high loads and high efficiencies with minimum possible diesel injection, thus increasing the natural gas energy contribution in the engine, which would reduce NO_x and soot emissions as proved by previous studies. [10]

In medium and heavy-duty diesel engines, to reduce the NOx and PM emissions and to achieve high loads and higher efficiencies, EGR and turbocharging methods are used. [2] To achieve high loads and efficiencies, the extent of gas charging and the amount of EGR required should be tested.

To study the optimization of combustion process and engine performance, numerical simulation procedure is an active area of research. 3-D computational models have been used to successfully demonstrate the combustion process in a diesel engine. [12] Considering the high computational cost demands for 3D modeling, efficient and reliable simulation models need to be developed and calibrated. [4] Gamma Technologies has been developing simulation tools for engine and vehicle simulation. GT-Suite software,

developed by Gamma Technologies is one of the leading simulation tools used for prediction of engine performance, emissions and acoustic characteristics. The current project uses GT-Suite software to develop one dimensional simulation model of dual fuel engine with the objective of identifying the need for gas charging and predicting the effect of gas charging and EGR on the performance characteristics of the dual-fuel engine.

1.1 Objective of Research

The current work is the part of the research project sponsored by Department of Energy (DOE) under the Vehicle Technologies Program (VTP) titled "High BMEP and High Efficiency Micro-Pilot Ignition Natural Gas Engine". The objective of the DOE project is to achieve BMEP of up to 25 bar and brake thermal efficiency of up to 44% under the constraint of diesel pilot contribution of 1-5% on a Cummins 6.7L engine. The engine is to be operated at stoichiometric conditions with charge dilution in dual-fuel mode to enable emissions control and use of a simple, low cost TWC system for aftertreatment.

The main objective of this work is to develop and validate a One-dimensional simulation model of Cummins 2010 ISB 6.7 L engine to study the effect of gas charging and EGR in dual fuel mode to achieve high BMEP and high efficiencies using GT-Suite software. Chapter **2** discusses the experimental setup, data collection required for one dimensional simulation model development. Chapter **3** discusses the Simulation model setup, combustion models in GT-Power for conventional diesel engine and dual-fuel engine. Chapter **4** describes the calibration procedure followed for the simulation model of diesel

engine. Chapter **5** summarizes the validation of the simulation models and the critical assumptions made for the engine simulation. Chapter **6** summarizes the calibration and test procedure for the dual-fuel engine simulation model. Chapter **7** discusses the results of the dual-fuel engine simulation model and the effect of boost pressure on the engine performance. Chapter **8** discusses the conclusions of the simulation for the dual-fuel model and the future scope of the work.

2 Experimental Setup, Data Acquisition and Testing

This chapter describes the experimental setup for the engine testing, data acquired for the one-dimensional simulation model and the procedure followed for experimental testing.

2.1 Experimental Setup- Engine and Instrumentation

This section discusses the engine setup in the APS labs. The heavy-duty diesel emissions and aftertreatment dynamometer test cell in APS labs has a Cummins 2010 ISB 6.7 L engine which meets the 2010 U.S. Environmental Protection Agency (EPA) regulations. The simulation model is designed for the engine whose specifications are specified in ***Table 1***.

Table 1 *Specifications of Cummins 2010 ISB 6.7L Engine*

Model	Cummins 2010 ISB6.7 224 kW (300 hp)
Bore and Stroke	107x124 mm
Engine Displacement	409 in^3 (6.7 L)
Configuration	Inline 6 Cylinder with Variable Geometry Turbocharger
Aspiration	Turbocharged
Rated Power	224 kW @ 2600 RPM
Peak Torque	896 N-m @ 1600 RPM
Turbine Model	Cummins Holset HE351VE
Compressor Type	Radial
Valve train	Cam operated, 4 valves per cylinder

The engine is equipped with an electronically controlled EGR valve and a High Pressure Common Rail (HPCR) fuel injection system with fuel injectors from Cummins (P/N 5256034). A Dynamatic Eddy current dynamometer (Model: DM8121HS) is used for load and speed control on the engine.

For data acquisition from the engine, the following instrumentation is used in the test cell: For internal combustion engine analysis and data acquisition, a DSP Technologies ACAP hardware system is used. The hardware crate has the following modules: A system controller module, a real-time processor module, ADC modules and a spincoder module. The in-cylinder pressure is obtained from Engine cylinders 4-6 with AVL Pressure transducers (Part # GH15D) which have a range of 0-250 bar.

For smoke measurements from the exhaust, an AVL 415SE Smoke meter is used. For emission gas analysis, Pierburg AMA4000 is used.

Design and simulation of the engine are carried out with GT-Suite software from Gamma Technologies. GT-Suite is a powerful tool widely used in industries to carry out engine and vehicle powertrain simulations. GT-Suite has the capability to fuse 1D and 3D simulations as one tool, which can be used to perform detailed sub-system analysis along with design optimization and performance analysis for various components of the engine and the vehicle.[13]

2.2 Data Acquisition for Simulation Model

The data required for designing the One-dimensional simulation model was characterized based on the key components of the engine: Intake system, Piston, Cylinder Head, Valves and Ports, Engine, Exhaust system, Turbocharger. The measurements obtained from the engine components for Intake System, Piston and EGR crossover connection are as summarized in ***Table 2***.

Table 2 *Data Acquired from Cummins 2010 ISB 6.7L Engine, for Design of Simulation Model*

Component	Part	Unit	Dimensions[1]
Compressor - Intercooler Pipe	No. of Pipe sections	#	6
	Length	mm	1524
Intercooler-Inlet Pipe	No. of Pipe sections	#	6
	Length	mm	1651
	Diameter	mm	76.7±1.2
Intake Manifold	Length	mm	900
	Diameter	mm	78.5±0.1
EGR-Intake Crossover Pipe	No. of Pipe sections	#	4
	Length	mm	762
	Diameter	mm	33.3±0.1
Piston	Piston Cup Diameter (max)	mm	72.8±0.1
	Piston cup depth at maximum diameter	mm	19.1±0.1
	Piston cup diameter (edge)	mm	60.9±0.1
	Piston cup center depth	mm	7.3±0.1
	Piston height	mm	103.2±0.1
	Skirt thickness	mm	8.9±0.7
	Ring thickness	mm	2.4±0.1
	Piston Top(deck) thickness	mm	27.5±0.3

[1] Measurements collected with Vernier calipers with a dial gauge. Least count of the instrument is 0.001".

The lengths of the pipes for compressor inlet, compressor to intercooler, intercooler to intake manifold, EGR crossover, exhaust and intake manifolds are measured approximately with a measuring tape due to the non-availability of engine design drawings. For cylinder head, valves and ports, engine geometry and the exhaust manifold, the data acquired from engine is summarized in ***Table 3***.

Table 3 *Cylinder head, Valve, Engine & Exhaust Manifold geometry acquired from Engine*

Component	Part	Unit	Dimensions[2]
Compressor Inlet	Length	mm	1600
Cylinder Head	Head gasket thickness	mm	1.6±0.1
	Cylinder Wall thickness	mm	6.9±0.1
	Cylinder Length	mm	230
Nozzle Geometry	Nozzle Hole Diameter	mm	0.2[3]
	Number of Holes	-	8
Valves and Ports	Intake Valve Face diameter	mm	32.8±0.2
	Exhaust Valve Face diameter	mm	32.3±0.2
	Intake Valve Lash	mm	0.2
	Exhaust Valve Lash	mm	0.7
	Port thickness	mm	2.6±0.1
Engine Geometry	TDC Clearance Height	mm	0.6
	Bore	mm	106.9
	Stroke	mm	123.9
	Connecting Rod Length	mm	192
	Compression Ratio	-	17.3:1
	Crank Angle at IVC (before TDCF)	deg	162
	Firing Order	-	1-5-3-6-2-4
Exhaust Manifold	No. of Pipe sections	mm	9
	Diameter	mm	36.5±0.2

[2] Measurements Collected with Vernier calipers with a dial gauge. Least count of the instrument is 0.001".

[3] Hole geometry obtained using Stereo microscopic imaging at 50x magnification

Valve lift profile for both the intake and exhaust valves for the engine was measured using Dial indicator, degree wheel. Before measuring the valve lift, valve lash was set to zero on both the intake and the exhaust valves.

Three sets of measurements were collected each, for intake and exhaust valves on cylinder 1 of the Cummins 2010 ISB6.7L engine. The mean valve lift for the data collected for the Exhaust and Intake valve with reference to TDC of power stroke (CAD=0°) is as shown in ***Figure 1***.

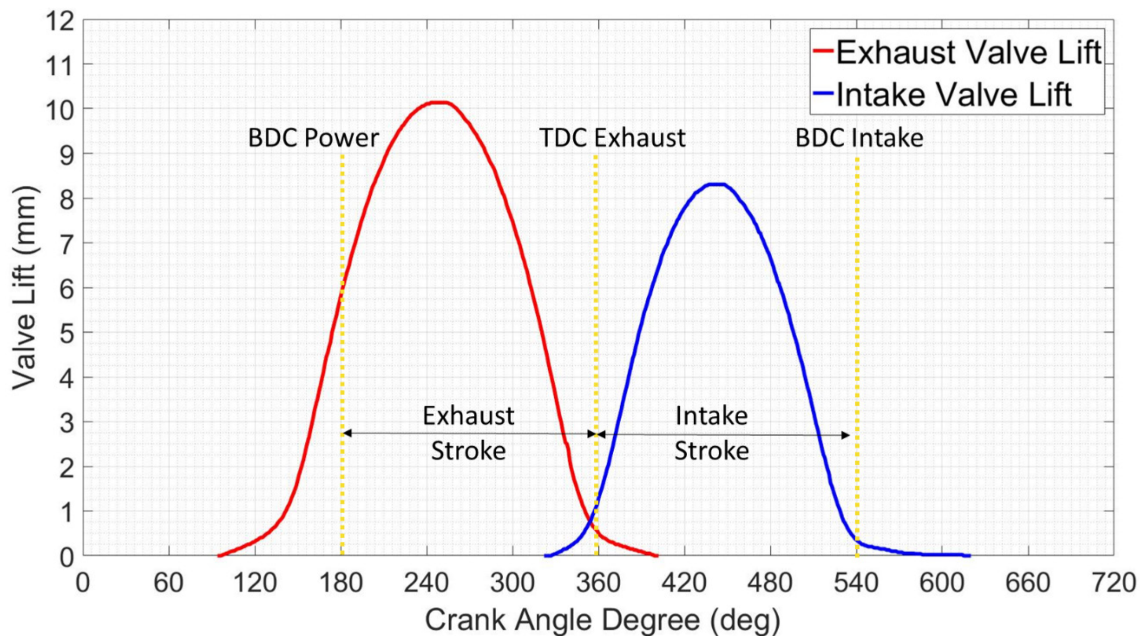

Figure 1 *Valve Lift for Cummins 2010 ISB6.7L Engine*

3 Simulation Model Setup

This chapter describes the simulation model setup for conventional diesel engine as well as the dual fuel engine to simulate the Cummins 2010 ISB 6.7 L engine.

3.1 One Dimensional Model for Diesel Engine

The layout of the engine, which is simulated using GT-Power, is as shown in ***Figure 2***. A turbine and a compressor constitute the Variable Geometry Turbocharger (VGT) in the engine.

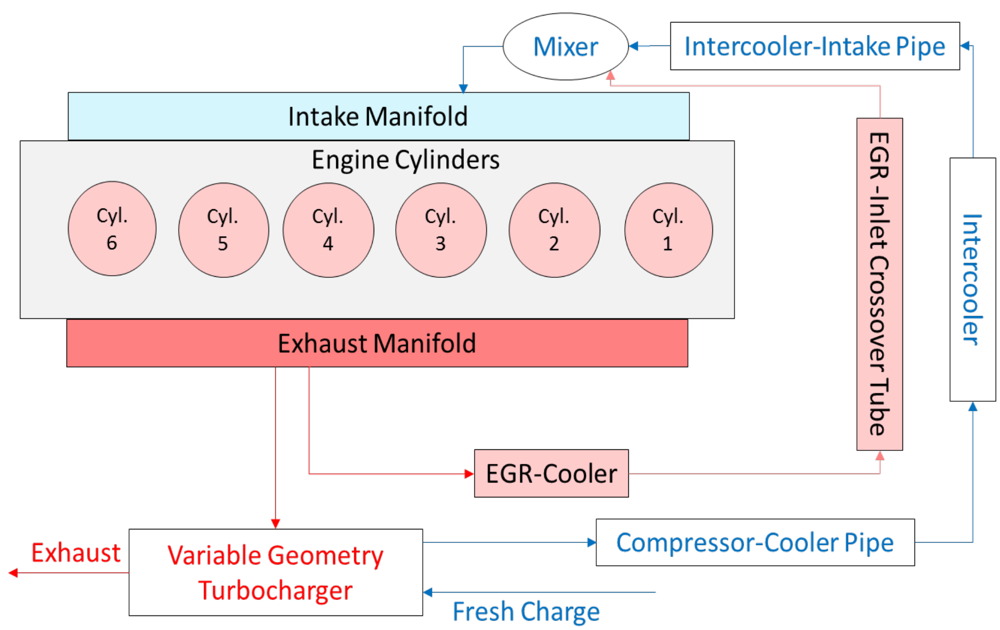

Figure 2 *Layout of Cummins 2010 ISB6.7L Engine for diesel mode*

The engine setup in the test cell and the flow directions for the air and the exhaust is as shown in ***Figure 3*** and ***Figure 4***.

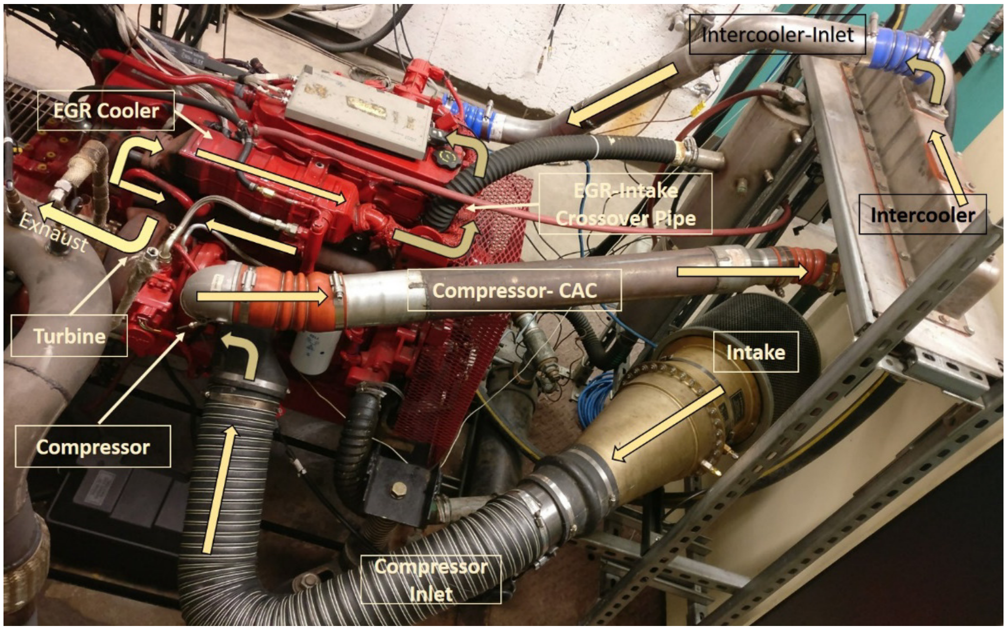

Figure 3 *Flow path for air and Exhaust in Engine - View from Turbocharger End*

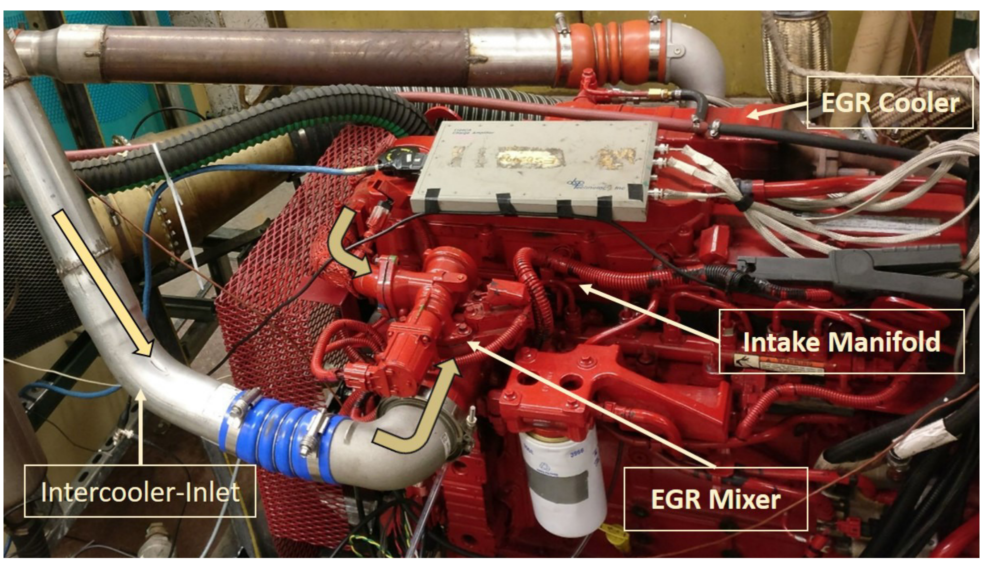

Figure 4 *Flow path for air and EGR in Engine - View from Intake End*

For each of the pipe section in GT-Power three input categories must be defined for simulation calculations: Main conditions, Thermal Conditions and Pressure Drop conditions.

- Main conditions consist the information regarding the initial state properties (Temperature, Pressure); discretization length for simulation calculations and surface finish characteristics (if necessary).
- Thermal conditions consist of the information regarding the wall temperature calculation methods along with additional thermal options regarding heat transfer.
- Pressure drop conditions consist of the information regarding the friction losses and the factor to change the pressure losses across the component.

Between any two flow components, an orifice connection is placed by default in GT-Power, which can be used to reduce pressure losses and provide smooth transition for components with large dimension changes in the flow.

The initial conditions provided as inputs to the simulation model serve as initial estimates for the simulation. The actual values are calculated according to the conditions pertaining to each of the engine operating condition.

Suitable assumptions are made, according to the design procedure as prescribed in the engine performance manual of GT-power and based on an example model-Diesel_VGT_EGR.gtm from GT-Power resources.

The model development is divided into 18 sections. The procedure of the design and the assumptions for each of the component is explained in the following sub-sections:

The overview of the simulation model designed in GT-Power is as shown in ***Figure 5***.

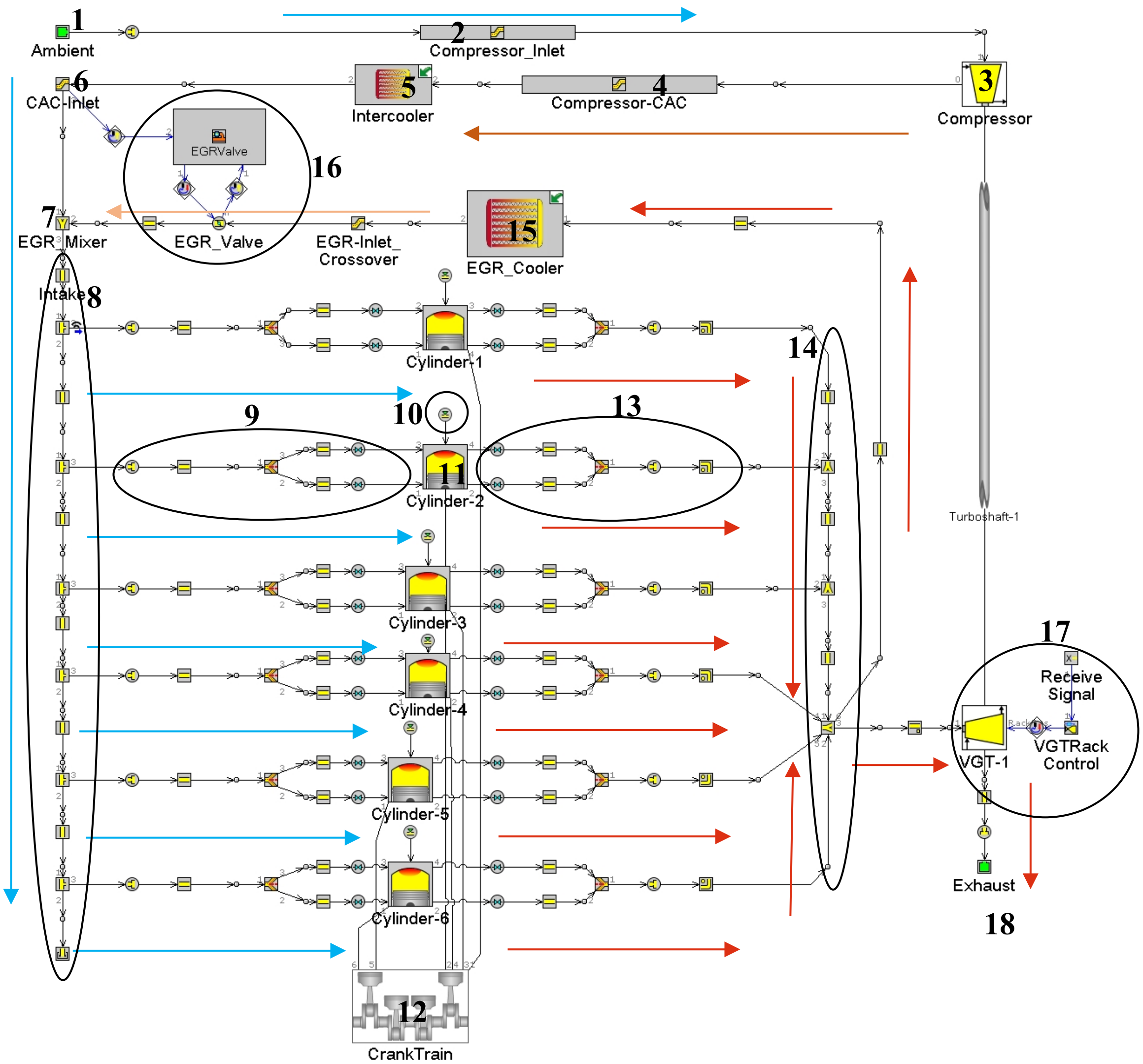

Figure 5 *Layout of Cummins 2010 ISB6.7L Engine designed using GT-Power for diesel mode*

3.1.1 Inlet Conditions

The inlet conditions for the fresh air entering into the engine is defined using the end environment template in GT-Power. The conditions for the ambient temperature and pressure are specified in the template.

3.1.2 Compressor Inlet

The length of the compressor inlet pipe is measured approximately to be 1600mm and the bends in the pipe are modeled with PipeTable template in GT-Power.

3.1.3 Compressor

In GT-Power, the inputs for the compressor model are the compressor maps, reference values for pressure temperature and specific heat ratio.

The reference values are the standard atmospheric conditions. Default values- Reference Temperature = 298K, Reference pressure =1 bar and Reference Specific heat ratio =1.4, are used.

The compressor map data is obtained by interpolation of plots for flow rate versus the pressure ratio and efficiency versus flow rate as shown in ***Figure 6*** and ***Figure 7*** obtained from reference.[14]

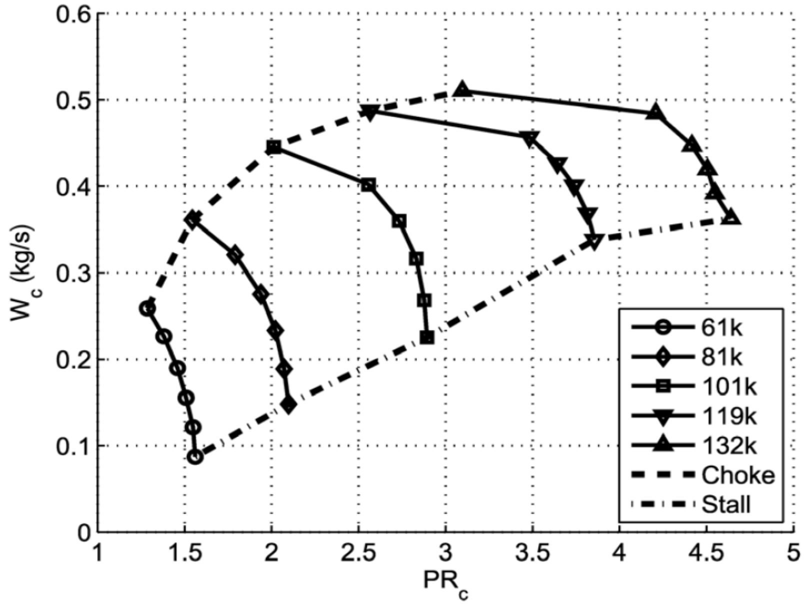

Figure 6 *Mass flow rate Versus Pressure Ratio for Compressor used in simulation*[4]

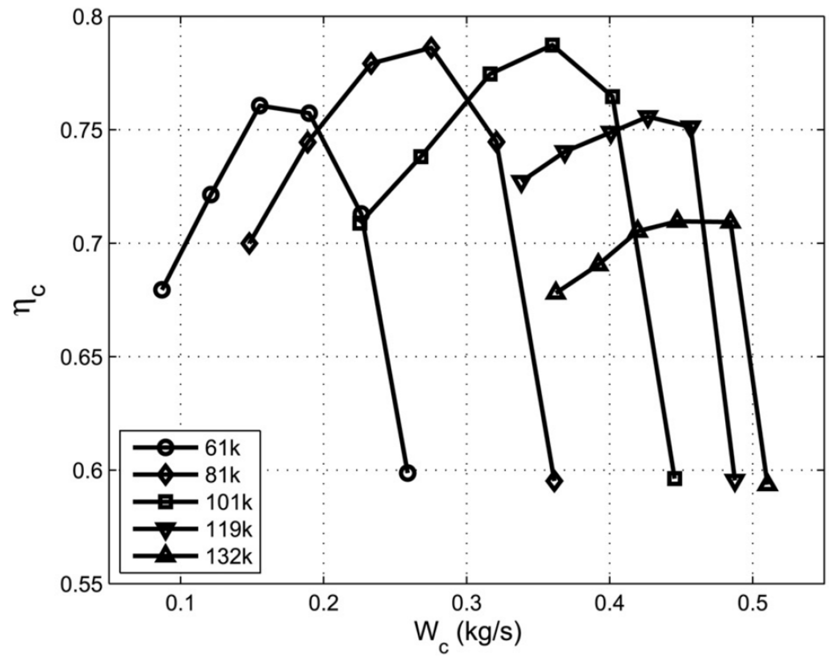

Figure *7 Efficiency versus Mass flow rate for Compressor used in simulation*[4]

[4] Reproduced with permission of AMERICAN SOCIETY OF MECHANICAL ENGINEERS

The data obtained from interpolation of plots as shown in ***Figure 6*** and ***Figure 7*** is entered into the data column of GT-Power compressor map object. The preprocessing plot obtained for the compressor maps for GT-Power is as shown in ***Figure 8***.

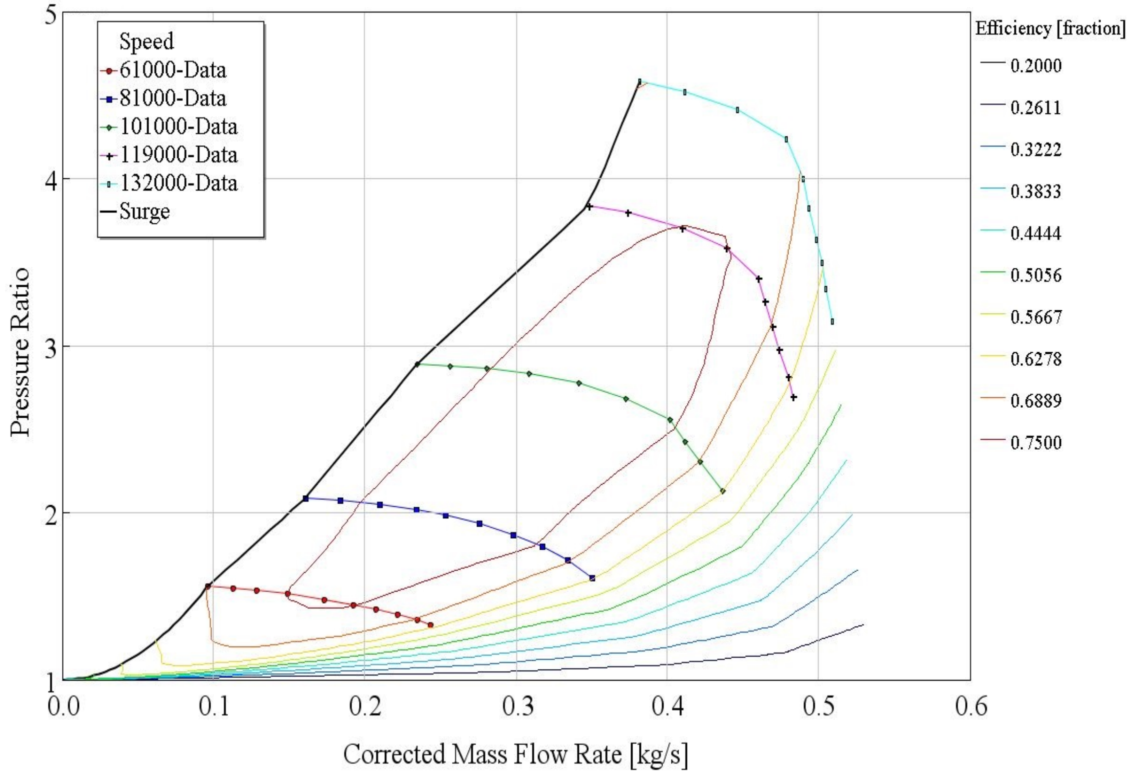

Figure 8 *Contour map for the Compressor generated for the data*

3.1.4 Compressor-Charge Air Cooler (Intercooler) Pipe

The length of the pipe was measured to be 1524mm. The pipe is modeled using the PipeTable template in GT-Power and is designed to have two 90° bends at either ends of the pipe. For thermal condition, the temperature of the charge is initialized to be the temperature of the compressed air.

3.1.5 Charge-Air Cooler (Intercooler)

The intercooler is modeled in GT-Power using a non-predictive methodology. The layout of the Intercooler as modeled using GT-Power is as shown in ***Figure 9***.

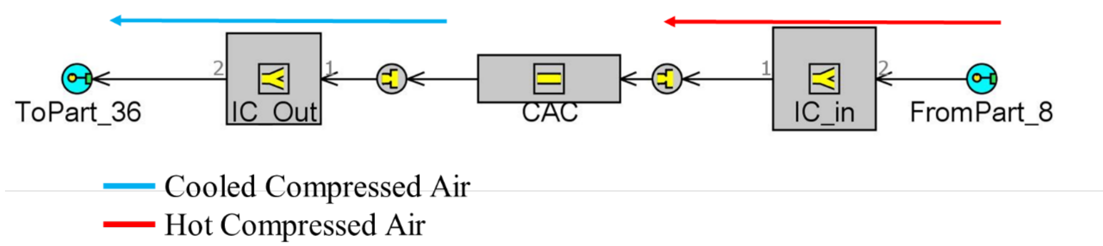

Figure 9 *Non-Predictive intercooler model in GT-Power*

CAC pipe emulates the function of the heat exchanger. The temperature required for the compressed air at the outlet of the intercooler is given as an input. The CAC pipe is designed as several pipes placed in series, which is specified in the additional geometry options of the pipe template in GT-Power. The heat transfer is modeled in CAC pipe until the temperature of the compressed air reaches the specified value in thermal condition of the pipe template.

3.1.6 Intercooler– Inlet Pipe

An initial state to represent the state properties of the fluid entering the intercooler-inlet pipe is created. Initial pressure of the fluid is assumed equal to the boost pressure and the initial temperature is assumed to be same as the manifold temperature. Boost pressure for the simulation refers to the pressure in the intake manifold of the engine. The dimensions

of the pipe are as indicated in **Table 2**, which is modeled using the pipe-table template of the GT-Power.

3.1.7 EGR Mixer

The cooled compressed air enters from the intercooler to the EGR mixer through the intercooler-inlet pipe and mixes with the exhaust gas coming through the EGR-Intake crossover pipe. The EGR mixer is modeled using a flow-split template in GT-Power. The required inputs for the EGR mixer are assumed based on the dimensions of the diameter of the pipe at inlet (78.5 ±0.1 mm) and an assumed length of 100 mm for mixing.

3.1.8 Intake Manifold

The intake manifold has an approximate length of 900 mm. The intake manifold is modeled to consist of flow-split pipes and piperound templates placed alternatively over the entire length of the intake manifold. 6 flow-splits and 6 piperound templates are used to model the intake manifold for the 6 cylinders of the engine. The length of the flow-split is fixed as 100 mm and the length of piperound is assumed as 50 mm. An end-flow cap template is also designed at the end of the sixth flow-split to indicate the end of the pipe.

3.1.9 Intake Runner and Intake Valve

The engine has two intake valves. The intake runner and the valve system is modeled to consist of a runner, followed by a flow-split and two intake ports leading to the valves. The intake runner and the intake ports are modeled using piperound template and the flow-split

is modeled using FlowSplitY template in GT-Power. A cut-section of the intake runner arrangement for the engine was used to take the measurements and the intake runner diameter was found to be approximately 35mm. The intake runner and the FlowSplitY are assumed to have a length of 50 mm whereas the intake ports are assumed to have a length of 95 mm. The assumptions are made from the intake runner cut-sections for the engine.

For intake valves, the valve lash of 0.254 mm prescribed by Cummins engine maintenance manual for the engine is set in the model. The cam timing anchor reference is set with respect to TDC firing and the cam timing lift array reference is set to maximum lift. The flow coefficients provided from the example model- Diesel_VGT_EGR.gtm are used.

3.1.10 Diesel Injector

The injector nozzle geometry was obtained using microscopic imaging technique. A Wild Heerbrugg stereo microscope is used to take images of the nozzle geometry at 50x magnification and the nozzle hole diameter is measured as 0.169 ± 0.001 mm. There are 8 nozzle holes for the injector and the nozzle discharge coefficient is assumed to be 0.82 (obtained from example model: Diesel_VGT_EGR.gtm). Diesel2-combust is used as fuel which is obtained from the GT-Suite library. The lower heating value of the fuel as specified in the template is 43 MJ/kg. The injector profile information is discussed in detail in chapter **4**.

3.1.11 Cylinder

The cylinder is one of the important aspects in the design of the model. The sub-sections of the cylinder design include:

- Wall temperature object
- Heat Transfer object
- Flow object
- Combustion object

3.1.11.1 Wall temperature object

The wall temperature object template in GT-Power follows a finite element representation of cylinder liner, piston and head to predict the in-cylinder heat transfer.

The finite element cylinder structure template has dedicated input windows for each cylinder head, piston, cylinder, valves and ports.

The initial temperatures for the head, piston and cylinder are assigned to 450 K based on the range of 450-650 K specified in GT-Power performance manual. The values are assumed for initialization of simulation and are overridden based on the calculations during the simulation.

3.1.11.2 Heat Transfer Object

The heat transfer method WoschniGT which is based on the heat transfer correlation without swirl, based on the heat transfer model proposed by Woschni [15], is used to predict the in-cylinder heat transfer. This heat transfer model assumes average gas velocity being proportional to mean piston speed, without the swirl number for the calculation of the heat transfer in the engine and is more suitable for the simulation model, due to unavailability of swirl calculations from the engine cylinder.

3.1.11.3 Combustion Object

For diesel engines, GT-Power has developed the following combustion models:

1. Imposed Combustion Profile
2. Diesel Wiebe Model
3. Multi Wiebe Model
4. Direct-Injection Diesel Multi-Pulse Model
5. Direct-Injection Diesel Jet Model
6. Homogeneous Charge Compression Ignition Model

The combustion models 1-3 are non-predictive combustion models, which impose the burn rate as a function of crank angle irrespective of the operating conditions of the engine.

These models can be used to study the effect of parameters, which have no significant effect on the burn rate in the cylinder.

To predict the effect of parameters on the engine performance, which have a significant effect on the burn rate, models 4-6 must be used.

Direct-Injection Diesel Multi-Pulse Model (DI-Pulse model) is a predictive model, which calculates the combustion rate for diesel engines with single or multiple injections. As suggested in the Engine performance manual by GT-Power, DI-Pulse model is more suitable for predicting the diesel injection when compared with Direct-Injection Diesel Jet Model (DI-Jet model). Due to the higher run-time and a better accuracy of combustion calculation[5], DI-Pulse model is selected as combustion model for the diesel simulation model.

3.1.11.3.1 DI-Pulse Model

The combustion model, DI-Pulse discretizes the cylinder contents into three zones: Main unburned zone, spray unburned zone and spray burned zone. The main unburned zone consists of the charge mass at IVC; the spray unburned zone consists of the injected fuel and the gas charge; and the spray burned zone consists of the combusted products[5].

[5] As described in Engine Performance Manual, GT-Power- Gamma Technologies, 2016

The process of combustion is further divided into many sub-models from the injection of fuel to the complete combustion, based on the physical conditions in the cylinder which is represented as the block diagram as shown in ***Figure 10***.

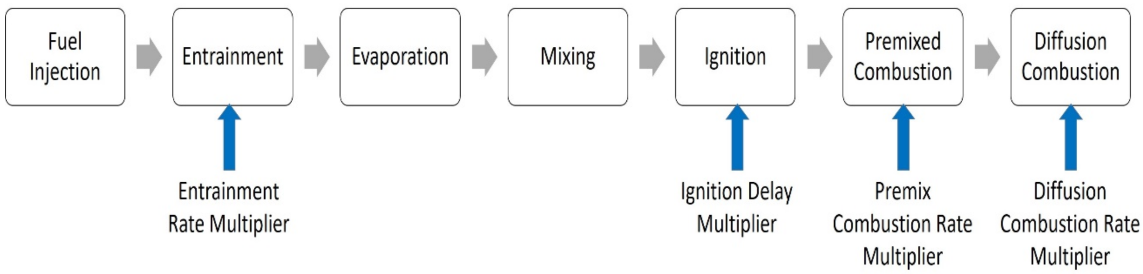

Figure 10 *DI Pulse combustion process and the control parameters*

The control parameters, which can be tuned based on the engine operating conditions to match the combustion process as in the real engine, are given as:

1. Entrainment Rate Multiplier
2. Ignition Delay Multiplier
3. Premix Combustion Rate Multiplier
4. Diffusion Combustion Rate Multiplier

The correct set of the multipliers is obtained by the calibration of the model, which is described in chapter **4**.

3.1.12 Crank Train

The crank train design consists of the engine layout information along with the mode of simulation. The following attributes are designed in GT-Power Crank Train:

- Engine type (4-stroke or a 2-stroke): Cummins 2010 ISB 6.7 L engine is a 4 –stroke engine
- Engine operating mode (Speed/ Load Mode): Load mode is used when the engine is coupled to a vehicle model in GT-Suite Vehicle wizard software. Speed mode is used for the engine performance evaluation and optimization. Speed mode is selected to model the Cummins 2010 ISB 6.7 L engine.
- Engine friction model: Mechanical friction in the engine is modeled in GT-Power based on the Chen Flynn model[16] as calculated using equation **(1)**. The recommended values in GT-Power are as specified for each of the term

$$FMEP = C + (\mathrm{PF} * P_{max}) + \left(\mathrm{SF} * Speed_p\right) + \left(\mathrm{SSF} * Speed_p^2\right) \qquad (1)$$

Where FMEP - Friction Mean effective pressure,

C- Constant part of the FMEP (recommended value: 0.3-0.5 bar)

PF- Peak cylinder pressure factor (recommended value: 0.004-0.006)

P_{max} – Maximum cylinder pressure

$Speed_p$ – Mean piston speed

SF- Mean piston speed factor (recommended value: 0.08 – 0.10 bar/(m/s))

SSF- Mean piston speed squared factor

Minimum values are assumed from the recommended range for each of the parameter, based on the example model- Diesel_VGT_EGR.gtm.

- The cylinder geometry obtained from the Cummins specification manual as summarized in ***Table 3*** is specified in the cylinder geometry of the Crank train template in GT-Power
- The firing order of the engine is specified as 1-5-3-6-2-4 for 6 cylinders with a firing interval of 120°.
- The initial states for calculation of the volumetric efficiency is specified as the ambient air temperature and pressure and the part for manifold volumetric efficiency reference is specified as the intake manifold flowsplit pipe 1 in the Real-Time Norms window of the Crank Train.

3.1.13 Exhaust Valve and Exhaust Runner

For exhaust valve, the valve lash is specified as 0.66mm, which is the nominal valve lash prescribed in the Cummins maintenance manual for the engine.

The exhaust runner is modeled in a similar manner as the intake runner. Cummins 2010 ISB 6.7 L engine for which the simulation model is designed has two exhaust valves. The assumptions for the exhaust runner dimensions are made from the cut-section of the

exhaust runner of the Cummins 6.7L Engine for which the simulation model is designed. The length of the exhaust ports is assumed 85 mm. The two exhaust ports merge into the exhaust FlowSplitY which is assumed to have a length of 55 mm. The initial thermal conditions for the exhaust FlowSplitY and the exhaust manifold are assumed to have the temperature same as the temperature measured before turbine. Bellmouth connections are placed for smooth transition and to estimate the orifice losses between the flow components.

3.1.14 Exhaust Manifold

The exhaust manifold layout designed in GT-Power is as shown in ***Figure 11***. The exhaust gas coming out of the cylinders is divided into two paths. The exhaust coming from the cylinders 1-5 goes into the turbine and then exits to the building exhaust. Another flow path for the exhaust gas is to the EGR Cooler from the exhaust manifold, placed after cylinder 6.

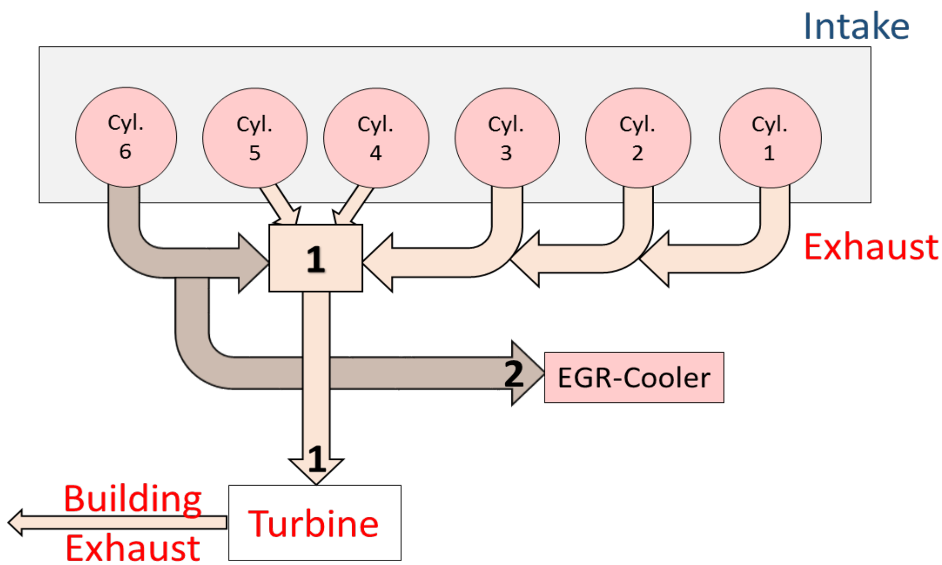

Figure 11 *Exhaust flow diagram - Simulation model design*

The pipes with bends are modeled using the additional geometry options in the pipe round template of GT-Power.

The exhaust manifold connecting the exhaust valves has a diameter of 36.5±0.2 mm. The exhaust gas from cylinder 1 comes through the exhaust port and runner into the round pipe, which is 210 mm long approximately, having a 90° bend in the middle. The exhaust gas goes into another round pipe having an approximate length of 90 mm. The exhaust from cylinder 2 passes through a round pipe having a 90° bend in the middle, which has a length of 210 mm, which mixes with the exhaust from cylinder 1 in a Flowsplit pipe. The exhaust gas from cylinder 3 is modeled similarly, which mixes with exhaust from cylinders 1 and 2, into another flowsplit pipe.

The combined exhaust from cylinders 1-3 and the exhaust from 4-6 cylinders mix into a single flowsplit pipe. The exhaust then branches into two paths, one going into the turbine through a rectangular pipe having dimensions of 100 x 44 x 75 mm and the other going into EGR cooler through round pipe of diameter 41.7 ± 1 mm.

3.1.15 EGR Cooler

The EGR cooler is modeled using the non-predictive technique like **Charge-Air Cooler (Intercooler)** model. The initial wall temperature for the pipe template is provided as input and calculations are based on the flow measurements during simulation of the model. The EGR cooler is modeled as a pipe template consisting of 100 identical pipes in series to increase the heat transfer area, thus reducing the temperature of the EGR entering the EGR cooler.

3.1.16 EGR Valve

EGR Valve is modeled as a throttle valve in GT-Power. A controller is designed to vary the angle of opening of the valve to allow exhaust gas flow into the EGR mixer. Mass flow rate of the intake air is measured in the Intercooler- Inlet Pipe with a sensor connection template of the GT-Power. Mass flow rate of the exhaust flowing into the throttle is also measured. The values of mass flow rates of air and EGR are used to calculate the EGR fraction as shown in equation (2). The EGR fraction obtained is then compared to the target EGR demanded by the user. Based on the error between the actual and the target value, the EGR valve is controlled such that the error is reduced to minimum.

$$EGR\ Fraction\ =\ \frac{ma\dot{s}s_{EGR}}{(ma\dot{s}s_{EGR} + ma\dot{s}s_{air})} \tag{2}$$

3.1.17 Variable Geometry Turbocharger (VGT)

Variable geometry turbocharger (VGT) optimizes the boost pressure requirement over a wide range of engine operation by varying the area of contact of exhaust gas onto the turbine wheel. Holset VGT turbocharger uses a sliding wall technology to vary the nozzle area to control the flow of exhaust gas through the turbine wheel. [17]

The data for turbine in GT-Power consists of many rack arrays based on the percentage of nozzle opening. Each array consists of pressure ratios, reduced mass flow rates and the efficiency points for various reduced turbocharger speeds. The process of obtaining the turbocharger maps from engine requires many tests to be carried out in controlled environment, which is not practical, considering the limitations of the engine operating conditions. The turbocharger map data is sensitive to exhaust temperatures before and after turbine, the exhaust mass flow rate and the turbocharger speed. To achieve better control the turbocharger, default maps from the example model Diesel_VGT_EGR.gtm are used. The data for the turbine maps consists of rack positions for 10, 20, 40, 60, 80 and 100% nozzle area openings.

In GT-Power variable geometry, turbocharging is achieved with a turbocharger rack controller. Boost pressure (bar) control is selected as the controller type. At every time step, the rack position is calculated based on the engine and turbocharger system conditions. The

controller receives a signal for pressure from the intake manifold pipe which is specified in the receive signal template of GT-Power. The rack position is then varied, until the pressure in the intake manifold pipe matches the target specified by the user.

3.1.18 Exhaust Conditions

The exhaust gas from the turbine is assumed to pass through a round pipe having a diameter same as the diameter of turbine exhaust which is approximately measured as 70 mm. The length of the pipe is assumed 400 mm. An end environment for exhaust gas is specified with the pressure being equal to atmospheric conditions.

3.2 One Dimensional model for Dual-Fuel Engine

The diesel engine model is used as the base model for the design of dual-fuel model. Additional features are added to the diesel engine model, to convert the model to run on dual-fuel. The layout for the dual-fuel model designed in GT-Power is as shown in ***Figure 12***.

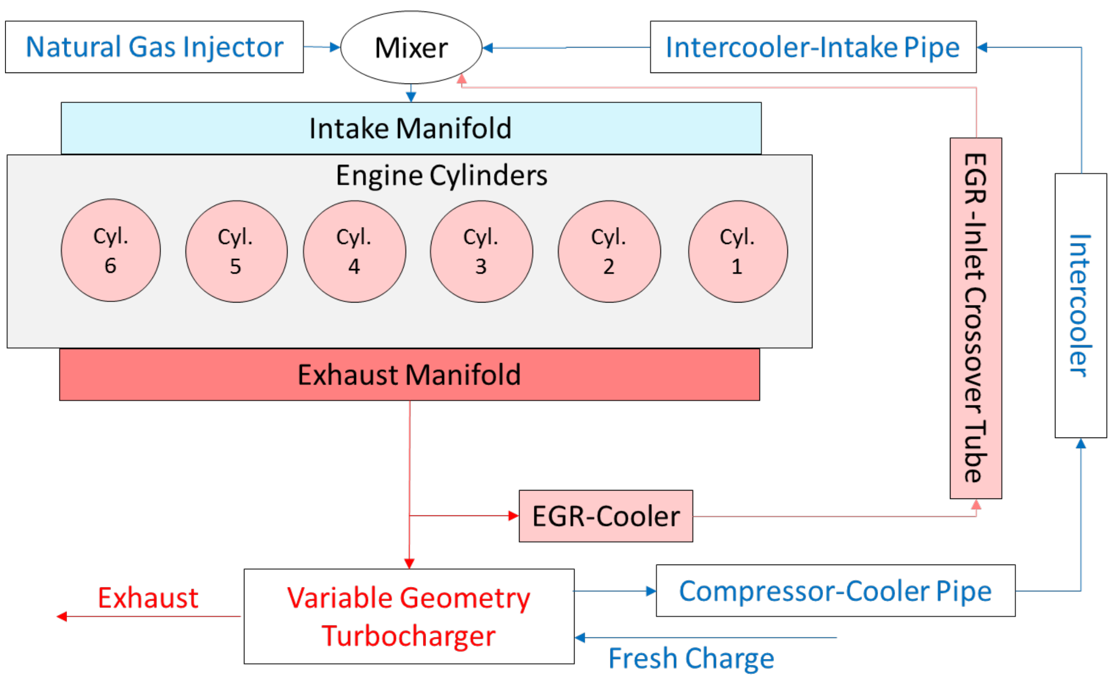

Figure 12 *Layout of Cummins 2010 ISB 6.7L Engine designed using GT-Power for dual-fuel mode*

The additions are explained in the following sub-sections:

3.2.1 Natural gas injector

To facilitate the natural gas injection in the system, injector based on the air fuel ratio is used. Air-Fuel Ratio, location of injector, fluid temperature, fluid object and vaporized fluid fraction are the inputs to be defined for the injector template.

- The injector is connected to a round pipe with a length of 150 mm, which is connected between the Intercooler and the EGR mixer. The location of injector in the pipe is assumed to be at the center of the pipe.
- The natural gas is assumed to be injected at 300 K.
- Composition of natural gas assumed in the simulation is as shown in ***Table 4***[18]

Table 4 *Natural gas composition*

Component	Molar %
Methane	96.36
Ethane	1.41
Propane	0.32
Nitrogen	1.47
CO_2	0.44

- The vaporized fluid fraction is assumed to be 1.

3.2.2 Combustion Model

To calculate combustion rate in dual-fuel engines with diesel pilot injection, Dual-Fuel Combustion model is developed by GT-Power. Two sub models for combustion, exist in the dual-fuel model: DI-Pulse model and SI turbulent flame combustion model. The DI-

Pulse model calculates the combustion of the diesel pilot injection and the SI turbulent flame combustion model predicts the burn rate for the air-natural gas mixture.

3.2.2.1 SI Turbulent flame combustion model

This model is used to predict the combustion rate in spark-ignition engines. As given in the GT-Power Engine Performance manual[6], the calculation of burn rates in the model is based on the governing equations (3),(4) and (5) from references [19-21].

$$\frac{dM_e}{dt} = \rho_u * A_e * (S_T + S_L) \tag{3}$$

$$\frac{dM_b}{dt} = (M_e - M_b)/\tau \tag{4}$$

$$\tau = \frac{\lambda}{S_L} \tag{5}$$

Where, M_e is the entrained mass of unburned mixture,

S_T, S_L turbulent and laminar flame speeds,

M_b is the burned mass,

A_e is the entrainment surface area at the flame front edge,

ρ_u is the density of unburned gas,

λ is the characteristic eddy radius (Microscale length).

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[6] v2016 GT-POWER Engine Performance Manual, 2016, p. 58

As per the equations, the entrained mass is dependent on the flame speeds and the flame area. The burn rate is dependent on the microscale length and the laminar flame speed. The model has control parameters to scale the effect of each of the components of combustion[7]:

- To modify the effect of residuals and EGR on the laminar flame speed, a dilution effect multiplier is used.
- To control the ignition delay, flame kernel growth multiplier is used
- To change the overall duration of combustion, a turbulent flame speed multiplier is used
- The effect of tail part of the combustion could be varied with the help of a Taylor length scale multiplier.

3.2.2.2 Dual Fuel Combustion Model

The combustion in dual-fuel engines with pilot injection is divided into two sub-models: Ignition delay model and the transition model of flame from diesel to air-natural gas mixture.

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[7] As per the Help manual specified for combustion model in GT-Power

The ignition model assumes the diesel injection spray being divided into discrete parcels based on the entrainment and evaporation process. Ignition delay is calculated at each time step based on the temperature and composition of each of the parcel [22].

For the transition of combustion from diesel to air-natural gas mixture, a linear transition model based on the penetration depth of the flame is developed. More is the penetration of the diesel spray, more would be the time taken for the linear transition from diesel jets to spherical flame front for air-natural gas combustion. The flame shape for the transition is assumed to have a truncated cone and a hemisphere as specified in reference [22].

The calibration procedure for the model and the process of optimization of the combustion parameters is discussed in detail in chapter **4**.

4 Calibration Procedure for Model

This chapter describes the calibration procedure followed for the diesel simulation model.

The combustion model DI-Pulse, selected for calculating the combustion rate in the diesel engine simulation and the Dual-fuel combustion model for dual fuel engine simulation require calibration to achieve results for engine simulation with better accuracy.

The steps involved in the calibration of the simulation model as suggested in engine performance manual are as follows:

1. Collection of engine experimental data
2. Setup of a combustion calibration model
3. Optimization of combustion model constants

4.1 Collection of Engine Experimental Data

Baseline diesel engine data was collected for two standard engine cycles: World Harmonized Stationary Cycle (WHSC) and European Standard Cycle (ESC). The WHSC and ESC test cycles, each consist of 13 operating points across the engine operating range.[23, 24]

Two control points were selected for the experimental testing to serve the purpose of engine consistency check. The controls points are selected in the WHSC cycle for test Modes WHSC12 and WHSC9. The summary of the control points is as shown in ***Table 5***.

Table 5 *Control points for engine experimental testing*

Parameter	WHSC 12	WHSC 9
Engine Speed (RPM)	1398	1768
Engine Load (N-m)	224	448

The following data was collected from the engine experimental testing:

- In-cylinder pressure from cylinders 4-6
- Diesel injection current, injection quantity, injection timing
- Temperature measurements from exhaust manifold, intake manifold
- Emissions measurements for CO_2, unburnt hydro-carbons, NO_x, O_2 and CO
- Air, fuel and exhaust flow rates
- EGR rate.

For calibration of simulation model, the control point WHSC 9 is selected. As it is a medium speed, medium load condition of the engine operating range, it is expected that the calibration results would be consistent with the entire range of operation.

4.2 Setup of a combustion calibration model

The calibration of the engine simulation model requires the setup of a single cylinder model, which only performs a closed volume pressure analysis, excluding the gas exchange process.

A single cylinder model with no valves and ports is modeled in GT-Power for calibration. The model setup is as shown in ***Figure 13***.

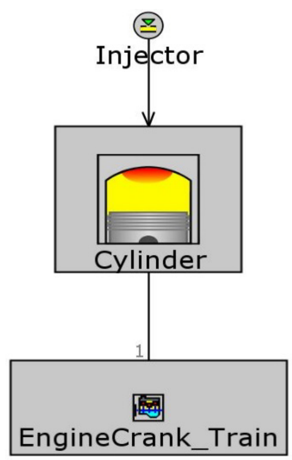

Figure 13 *Calibration model setup in GT-Power*

4.2.1 Injector Profile

The predictive combustion models in GT-Power require an accurate injection profile (injected fuel flow rate as a function of time) and injection timing. The injector profile could not be measured from the engine experimental testing. A GT-power model is hence used to obtain the injection profile for the calibration as well as the simulation models.

A pre-designed GT-Power detailed injector model (DetailedInjector-InjRateMap.gtm) was used to obtain an injection profile map [25]. The model, through the process of design of experiments generates a detailed injector map for various injection pressures and pulse widths. A case setup is created within the model, consisting of data of the injector, which is obtained from spray tests carried out on the injector in Advanced Energy Research Building of Michigan Technological University. The test data for electronic injection duration (EID) of 1.2 milliseconds is used for the case setup. The injection data required for the model is as summarized in ***Table 6***.

Table 6 *Injector spray test data in case setup of injector model in GT-Power*

Back Pressure (bar)	Injection Pressure (bar)	EID (ms)	Injection Delay (ms)	Injection Quantity (mg)
4.0	100	1.2	0.57	20
	300		0.43	49
	600		0.35	87
	1000		0.33	105
17.5	100		0.53	20
	300		0.52	50
	600		0.40	83
	1000		0.33	104
35.9	100		0.49	19
	300		0.39	51
	600		0.35	86
	1000		0.32	104

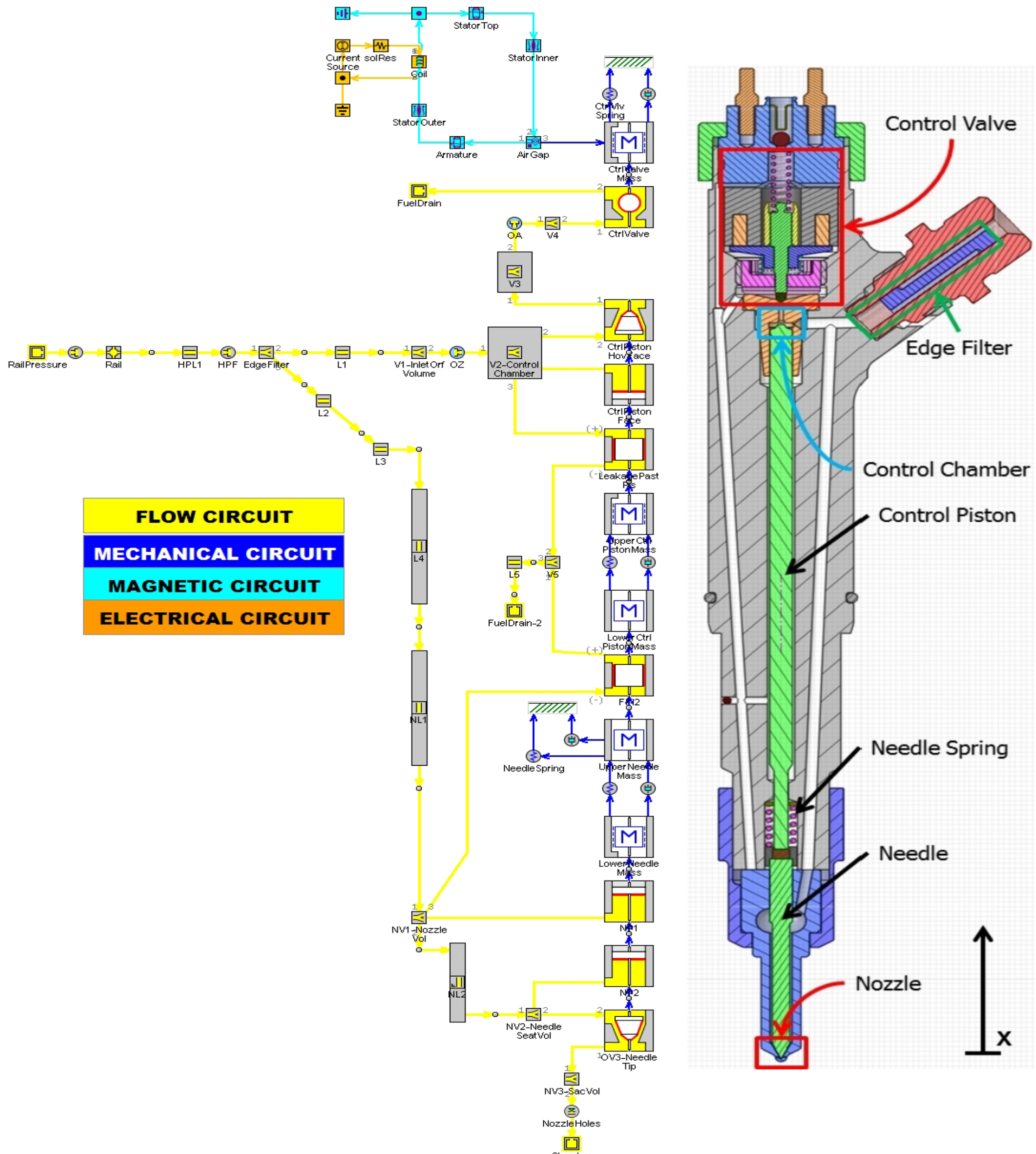

***Figure 14** GT-Power layout of the detailed injector model used for generation of injector map*[8]

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[8] v2016 GT-POWER Example Model-DetailedInjector-InjRateMap.gtm, GT-SUITE Software, 2016

To include injection profiles for injection pressures above 1000 bar, additional cases were included in the setup, with data obtained from baseline testing. The additional case data included in the simulation model is as summarized in ***Table** 7*

***Table** 7 Additional injector data, used in case setup of injector model in GT-Power*

Test Mode	Injection Pressure (bar)	Back Pressure (bar)	EID (ms)	Injection Delay (ms)	Injection Quantity (mg)
ESC6	1327	92	1.2	0.32	81.3
WHSC4	1521	92		0.32	75.0
ESC10	1746	109		0.32	105.4

From **Table 6**, it is observed that the injection delay reduces with the increase in the injection pressure for a constant electronic injection duration (EID). Based on the data obtained from spray tests, the higher injection pressures are assumed to have an injection delay of 0.32 milliseconds. The injection quantities are obtained from the Engine data log for test modes ESC 6, WHSC 4 and ESC10. These cases are selected to include the injection pressures over a uniform range above 1000 bar.

The output of the injector model is a detailed injector rate map of main injection events, which consists of 112 injection profiles for injection timings from 0.2 to 2.5 milliseconds for the seven injection pressures indicated in ***Table 6*** and ***Table** 7*.

GT-power interpolates the injection profiles for various injection profiles based on the injector rate map obtained from the injector model. This injector rate maps is used for calibration model, diesel and the dual-fuel models.

4.2.2 Cylinder and Crank Train

For the calibration model, the setup of the model is followed as per the instructions given in the GT-Power Engine performance manual. A closed volume analysis is carried out for calibration of the diesel model for test mode WHSC 9. The setup procedure is summarized as follows:

- An initial state object consisting the details for volumetric efficiency, trapping ratio, fuel at IVC and residual fraction is used. The volumetric efficiency is obtained from engine data logged from engine software CalTerm III. As suggested in the engine performance manual of GT-Power, air-trapping ratio (ratio of air trapped in the cylinder to the air delivered to the cylinder) is assumed as one, the residual fraction includes the EGR obtained from CalTerm III engine log for test mode WHSC 9.
- Heat transfer model is selected as Woschni GT and the default values are used for the initialization of remaining values in the template
- Measured data for cylinder pressure, averaged for 300 cycles obtained from post-processing of the combustion analyzer is used in the Measured Cylinder Pressure analysis object along with the emission data collected from the Pierburg analyzer AMA 4000.
- The combustion object consists of the combustion parameters, which are used as independent variables for design of experiment setup.

Table 8 *Input data for test mode WHSC 9 in calibration model*

Parameter	Unit	Value
Volumetric Efficiency	%	90
Air-trapping ratio	-	1
EGR Fraction	%	17
Fuel Mass trapped at IVC	mg	0
NO concentration	PPM	219
CO concentration	PPM	46

The crank train consists of the engine geometry as described in section **3.1.12**. For calibration, the firing order is specified only for cylinder 1.

4.3 Optimization of Combustion Model constants

The model constants for the DI-Pulse and SI turbulent flame combustion models are described in sections **3.1.11** and **3.2.2** respectively. The acceptable range of the multipliers for DI-Pulse combustion model is as summarized in ***Table** 9*.

Table 9 *DI-Pulse combustion multipliers recommended range*[9]

Parameter	Minimum	Maximum
Entrainment Rate Multiplier	0.95	2.8
Ignition Delay Multiplier	0.3	1.7
Premixed Combustion Rate Multiplier	0.05	2.5
Diffusion Combustion Rate Multiplier	0.4	1.4

[9] As recommended in GT-Power Engine Performance Manual

The acceptable range of the multipliers for SI-Turbulent flame combustion model is as summarized in ***Table 10***.

Table 10 *SI-Turbulent flame model - combustion multipliers recommended range9*

Parameter	Minimum	Maximum
Dilution Exponent Multiplier	0.5	3
Flame Kernel Growth Multiplier	0.5	3
Turbulent Flame Speed Multiplier	0.5	3
Taylor Length Scale Multiplier	0.5	3

For optimization of model constants for DI-Pulse combustion model for the diesel engine, advanced direct optimizer template is used. The default settings for the optimization algorithm as summarized in the engine performance manual of GT-Power is as specified in ***Table 11***.

Table 11 *Recommended settings for direct optimizer algorithm*

Parameter	Value
Population Size	30
Number of Generations	34
Mutation Rate	0.5
Mutation Rate Distribution Index	15

The optimization target is the dependent variable- Improved Burn Rate RMS error (Measured Vs Predicted), selected from the list of elements corresponding to the cylinder. The objective of the optimization is to minimize the error between the measured and the predicted values of this burn rate.

The independent variables controlling the dependent variable are specified as the combustion model constants. The direct optimizer template has an independent variable

setup window, where each of the combustion constant given in ***Table 9*** is added and the corresponding range is specified for sweep setup.

4.4 Results for Calibration Model

Advanced direct optimizer in GT-Power performed 1021 iterations to reduce the Improved Burn rate RMS error function to a minimum value of 0.0037 (Unit less number) and the results obtained for the combustion constants of the model are summarized in ***Table 12***.

Table 12 *Optimized combustion model constants, obtained from calibration*

Parameter	Optimized Value
Entrainment Rate Multiplier	1.54
Ignition Delay Multiplier	0.30
Premixed Combustion Rate Multiplier	2.47
Diffusion Combustion Rate Multiplier	1.08

The optimized values for the multipliers are used in the validation of the simulation models for diesel engine and further used in the dual-fuel model.

5 Validation of Simulation Models

The simulation model setup as described in chapter **3** is validated with the experimental results by using the calibration results for the combustion constants in the model. Key assumptions and the validation results for the model are described in this chapter.

5.1 Assumption for injection- Single injection event

For engine experimental testing, the start of injection (SOI) which is measured in degrees before TDC is obtained from the CalTerm engine log files. It has been observed that the pilot SOI was very close to the main SOI. In simulation model, successive injections that have small difference in crank angle degrees leads to failure in the injection. Since a very small amount of injection occurs in pilot injection, it is assumed for the simulation models that only a main injection event exists.

5.2 Assumption for Multipliers of Turbine Maps

Due to non-availability of standard turbine map data, default turbine maps from GT-Power example model Diesel_VGT_EGR.gtm are used. The model consists of turbine maps for turbocharger for a turbine having a maximum pressure ratio limit of 4.5 and a maximum reduced speed limit of 8000 RPM/K^0.5. The maximum pressure ratio observed from the test cases for the Cummins 2010 ISB 6.7 L engine in the lab is observed to be 3.3 for test case WHSC 10. To match the boost pressure output from the turbocharger, suitable multipliers, which are present in GT-power, are used to shift the range of the default maps.

The multipliers, which are tuned in GT-power, are for turbine efficiency, mass flow and turbocharger speed. The value specified for each of the multiplier is multiplied with each element specified in the array. The effect of multiplier is as shown in ***Figure 15*** and ***Figure 16*** for a rack position of 80%. ***Figure 15*** shows the default rack data plot for reduced mass flow rate as a function of pressure ratio.

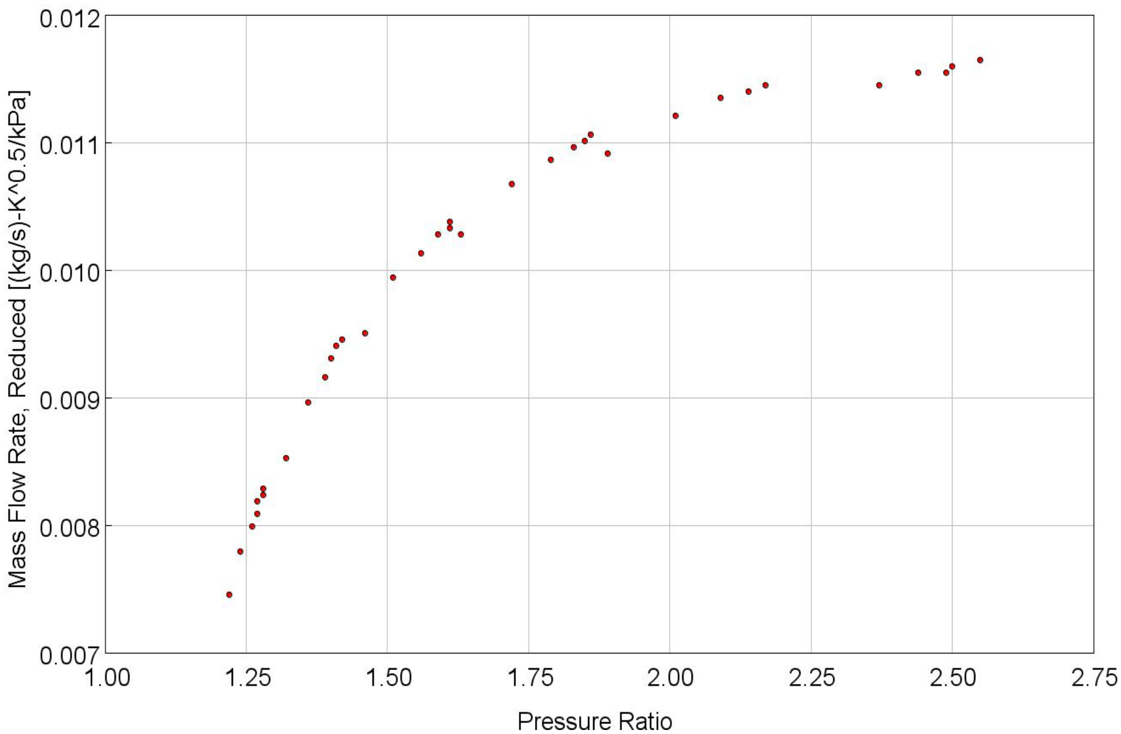

Figure 15 *Mass flow rate versus Pressure Ratio for Rack 80% in turbine maps with default multipliers*

For a mass flow rate multiplier of 1.5, the data for the mass flow rate is multiplied by 1.5, which is as shown in ***Figure 16***.

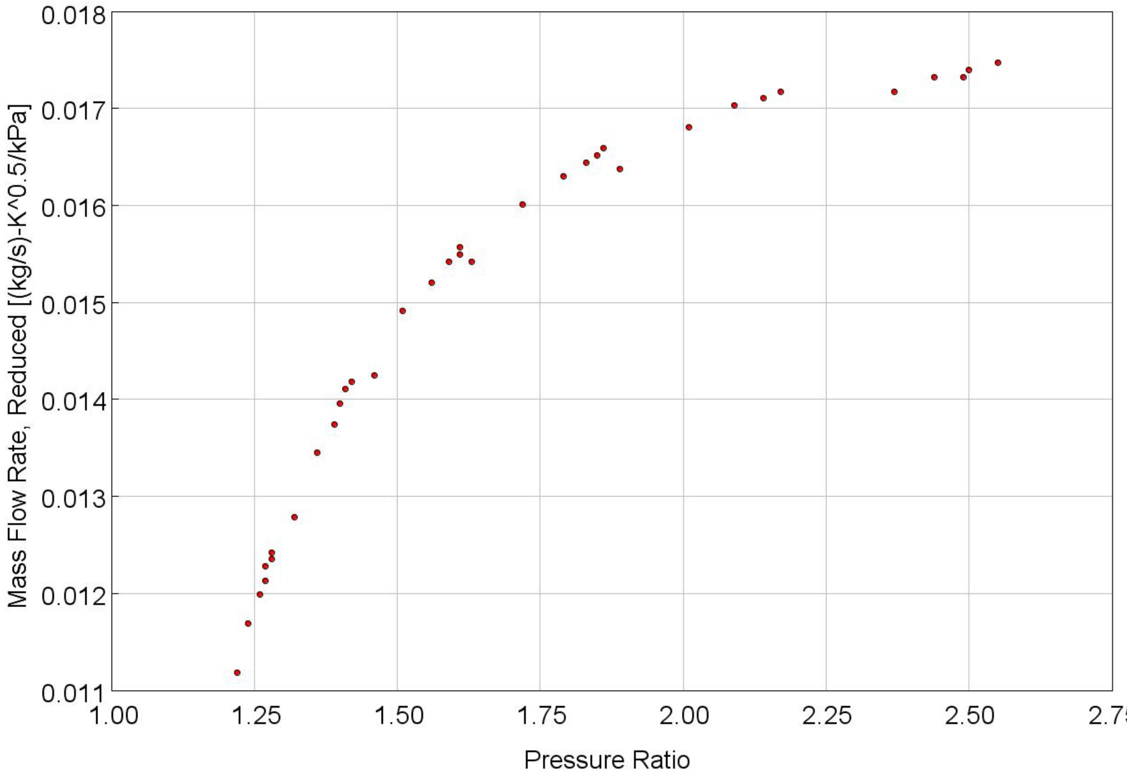

Figure 16 *Mass flow rate versus Pressure Ratio for Rack 80% in turbine maps for mass flow rate multiplier = 1.5*

The multipliers do not affect the combustion model as the combustion model is controlled by the independent parameters, which are calibrated as described in chapter **4.**

5.3 Validation of Diesel Model

Validation of the diesel simulation model is performed for test cases WHSC 5, 9, 10 and ESC 7, 12, 13.

5.3.1 Validation of WHSC 5

The diesel engine simulation model is simulated for WHSC case 5, which is a low-speed high-load case. The input data for the model is as summarized in ***Table 13***.

Table 13 *Operating conditions input data for WHSC 5 test case for diesel simulation model*

Parameter	Unit	Value
Engine Speed	RPM	1397
EGR Target	%	0
Diesel mass injected	mg/stroke	107.9
Main SOI	deg bTDC	-0.33
Fuel Rail Pressure	bar	1139
Boost Pressure	bar	1.83

The initialization parameters used for the simulation are as summarized in ***Table 14***.

Table 14 *Initialization data for WHSC 9 test case for diesel simulation model*

Parameter	Unit	Value
Ambient Temperature	°C	16.8
Coolant Temperature	°C	91.2
Cooled Compressed Air temperature	°C	30.9
Fuel Temperature	°C	25.9
Intake Manifold Temperature	°C	39.3
Exhaust Temperature before Turbine	°C	675.7
Exhaust Temperature after Turbine	°C	537.3
Ambient Pressure	bar	1.01
Exhaust Pressure before Turbine	bar	2.21
Exhaust Pressure after Turbine	bar	1.01
Turbocharger Initial Speed	RPM	81258

The comparison of results for WHSC 5 for simulation and experimental tests is as summarized in ***Table 15***.

Table 15 *Comparison of simulation and experimental results for WHSC 5*

Parameter	Units	Experimental Result	Simulation Result	Difference
Brake Torque	N-m	894	865	29
Gross IMEP[10]	bar	18.1	18.0	0.1
A/F Ratio	-	19.7	19.0	0.7
Maximum Cylinder Pressure (P_{max})	bar	107.7	108.2	-0.5
CAD for P_{max}	deg bTDC	17	16	-1

The in-cylinder pressure trace obtained from the pressure transducer installed on cylinder-5 of the engine is compared with the simulation results, which is as shown in ***Figure 17***.

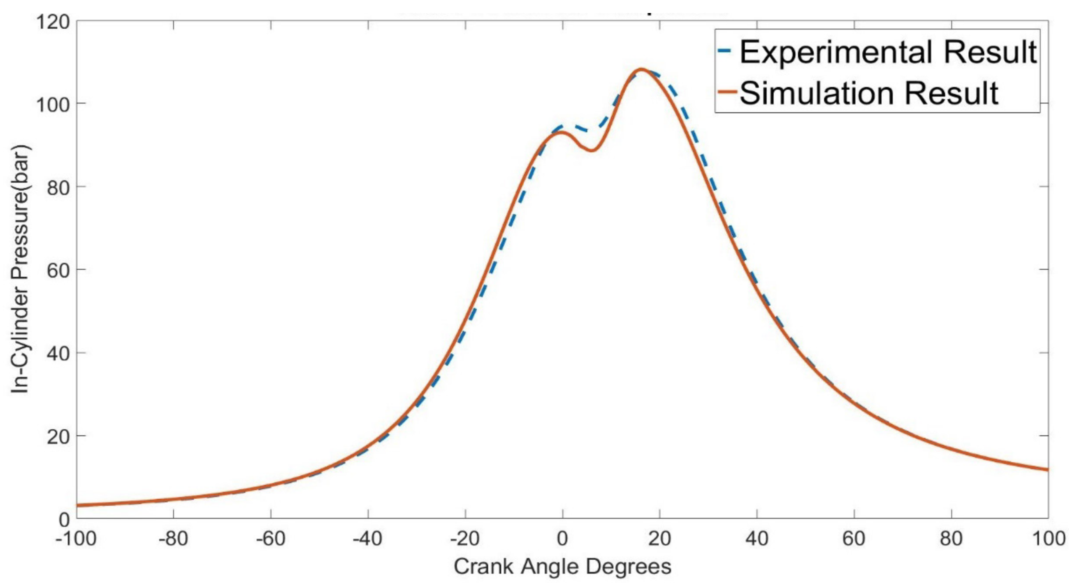

Figure 17 *Comparison of In-cylinder pressure results for simulation and experimental tests for WHSC 5*

[10] Gross IMEP for Cylinder 5 from engine and simulation

The log p-v diagram comparison for the simulation and experimental tests is as shown in ***Figure 18***.

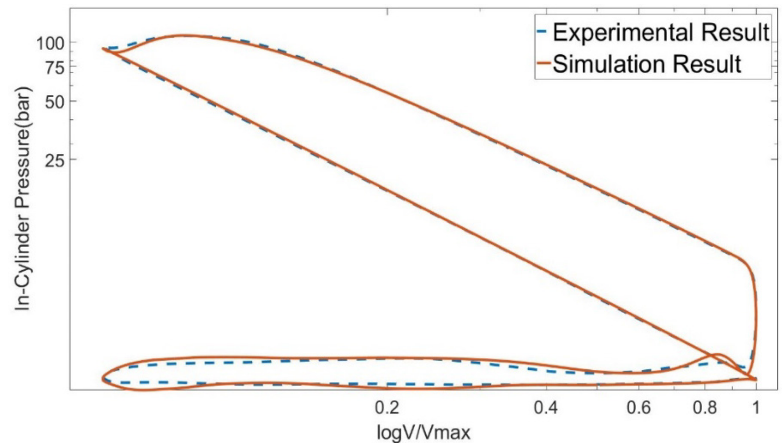

Figure 18 *Comparison of pressure vs volume results for simulation and experimental tests for WHSC 5*

Apparent heat release rate calculated from the pressure and the volume data for both experimental tests and simulation for WHSC 5 is as shown in ***Figure 19***.

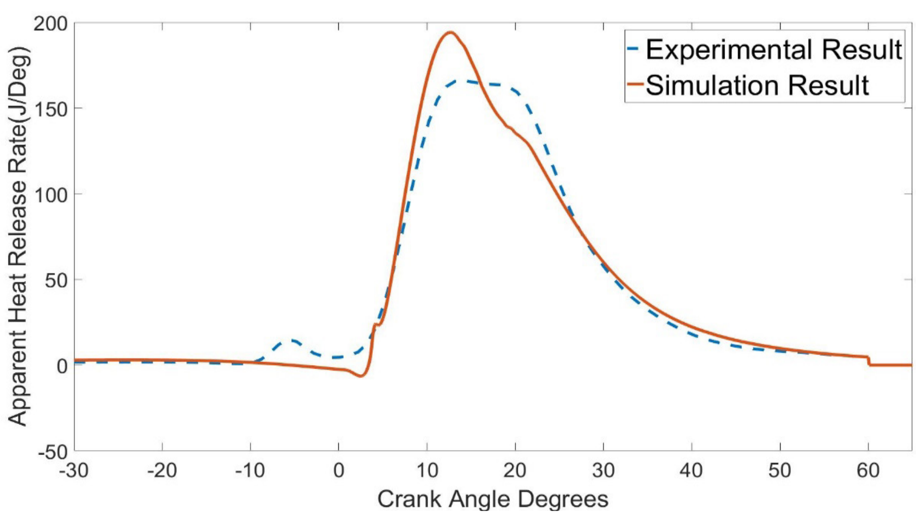

Figure 19 *Comparison of apparent heat release rate results for simulation and experimental tests for WHSC 5*

The experimental pressure data is collected over 300 engine cycles and the mean cylinder pressure is obtained from the post-processing of the data using MATLAB.

Apparent heat release rate is calculated from the pressure and the volume data for both experimental tests and simulation using the heat release correlation given by equation (6) from reference [15]. The equation is based on the first law of thermodynamics and ideal gas assumptions, neglecting the crevice effects.

$$Apparent\ Heat\ Release\ rate, \frac{dQ}{d\theta} = \frac{\gamma}{\gamma - 1} p \frac{dV}{d\theta} + \frac{1}{\gamma - 1} V \frac{dp}{d\theta} \qquad (6)$$

The variation with respect to crank angle for in-cylinder pressure (dP/dθ) and volume (dV/dθ) is used to obtain the heat release rate as a function of crank angle (dQ/dθ).

5.3.2 Validation of WHSC 9

The simulation model designed for diesel engine is simulated for WHSC case 9 with input data as summarized in ***Table 16***

Table 16 *Operating conditions input data for WHSC 9 test case for diesel simulation model*

Parameter	Unit	Value
Engine Speed	RPM	1768
EGR Target	%	17
Diesel mass injected	mg/stroke	59.2
Main SOI	deg bTDC	0.60
Fuel Rail Pressure	bar	1530
Boost Pressure	bar	1.52

For simulation, the initial states for various components need the state properties (pressure, temperature) for initialization. The initialization parameters used for the simulation are as summarized in ***Table 17***.

Table 17 *Initialization data for WHSC 9 test case for diesel simulation model*

Parameter	Unit	Value
Ambient Temperature	°C	16.0
Coolant Temperature	°C	89.8
Cooled Compressed Air temperature	°C	24.0
Fuel Temperature	°C	27.1
Intake Manifold Temperature	°C	41.6
Exhaust Temperature before Turbine	°C	508.6
Exhaust Temperature after Turbine	°C	404.6
Ambient Pressure	bar	1.01
Exhaust Pressure before Turbine	bar	2.19
Exhaust Pressure after Turbine	bar	1.01
Turbocharger Initial Speed	RPM	69535

The comparison of results for WHSC 9 for simulation and experimental tests is as summarized in ***Table 18***.

Table 18 *Comparison of simulation and experimental results for WHSC 9*

Parameter	Units	Experimental Result	Simulation Result	Difference
Brake Torque	N-m	450	439	11
Gross IMEP[11]	bar	10.3	10.5	-0.2
A/F Ratio	-	24.9	24.4	0.5
Maximum Cylinder Pressure (P_{max})	bar	84.5	83.7	0.8
CAD for P_{max}	deg bTDC	14	15	-1

[11] IMEP for Cylinder 5 from engine and simulation

The in-cylinder pressure trace obtained from the pressure transducer installed on cylinder-5 of the engine is compared with the simulation results, which is as shown in ***Figure 20***.

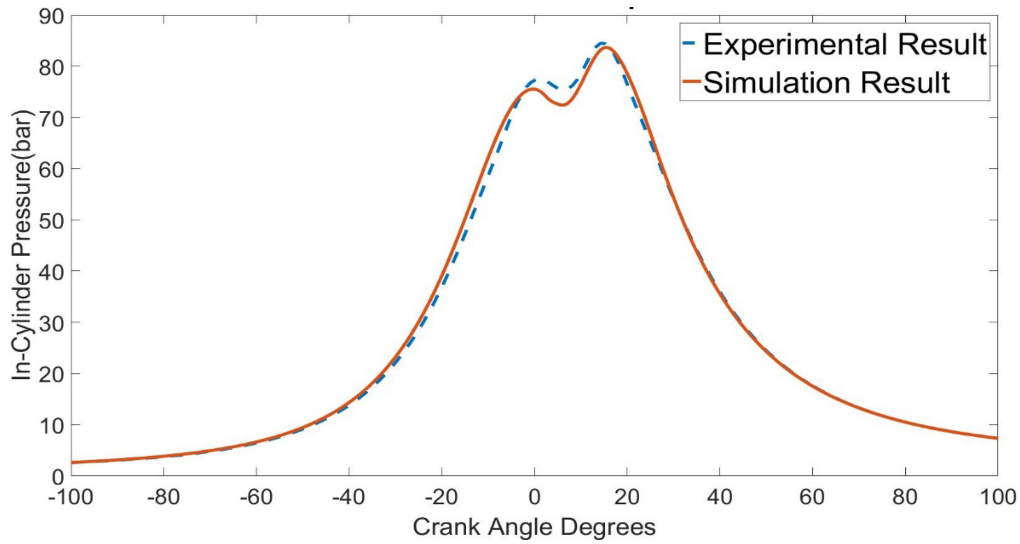

Figure 20 *Comparison of In-cylinder pressure results for simulation and experimental tests for WHSC 9*

The log p-v diagram comparison for the simulation and experimental tests is as shown in ***Figure 21***.

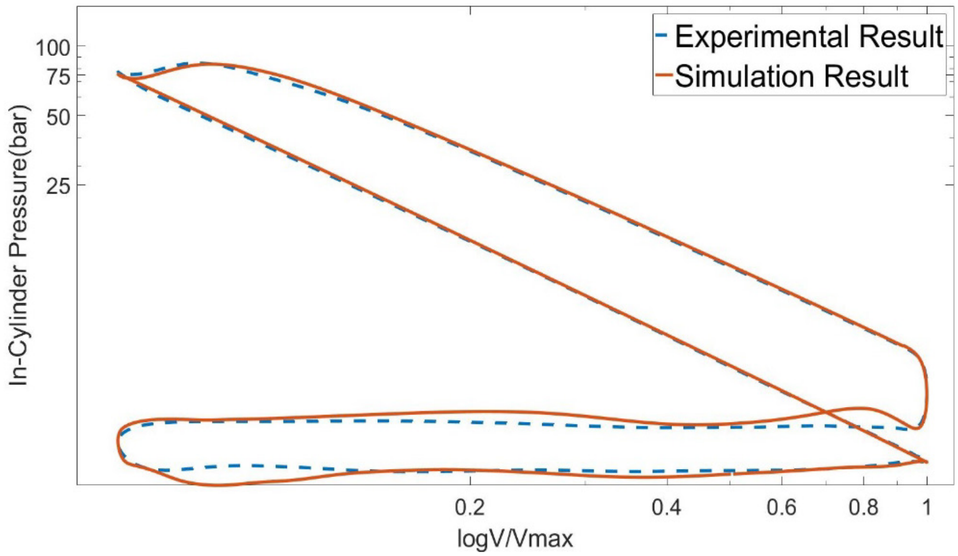

***Figure 21** Comparison of pressure vs volume results for simulation and experimental tests for WHSC 9*

Apparent heat release rate calculated from the pressure and the volume data for both experimental tests and simulation for WHSC 9 is as shown in ***Figure 22***.

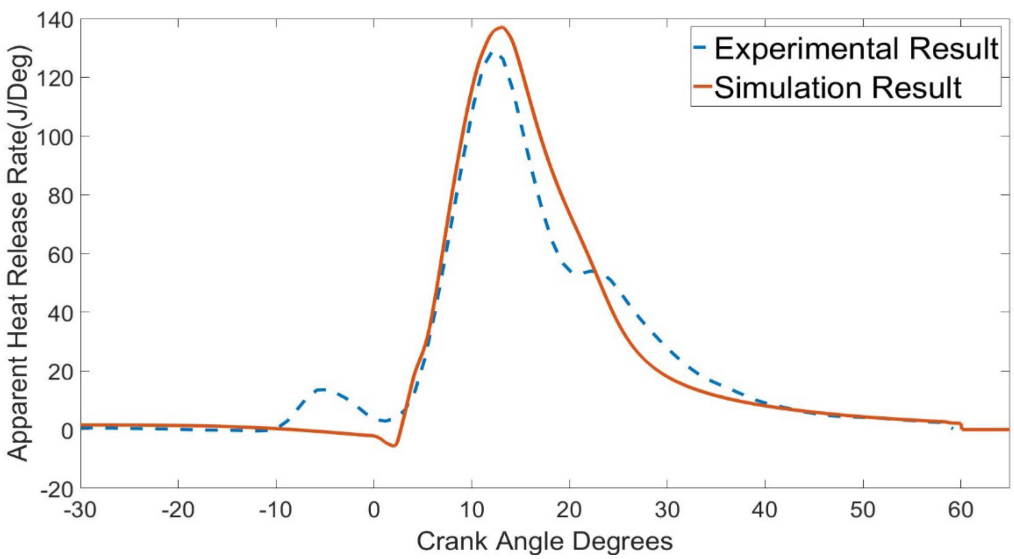

***Figure 22** Comparison of apparent heat release rate results for simulation and experimental tests for WHSC 9*

5.3.3 Validation of WHSC 10

The diesel engine simulation model is simulated for WHSC case 10, which is a high-speed high-load case. The input data for the model is as summarized in ***Table 19***

Table 19 *Operating conditions input data for WHSC 10 test case for diesel simulation model*

Parameter	Unit	Value
Engine Speed	RPM	2138
EGR Target	%	6
Diesel mass injected	mg/stroke	104.2
Main SOI	deg bTDC	5.94
Fuel Rail Pressure	bar	1707.4
Boost Pressure	bar	2.52

The initialization parameters used for the simulation are as summarized in ***Table 20***.

Table 20 *Initialization data for WHSC 10 test case for diesel simulation model*

Parameter	Unit	Value
Ambient Temperature	°C	17.5
Coolant Temperature	°C	93.4
Cooled Compressed Air temperature	°C	61.0
Fuel Temperature	°C	26.0
Intake Manifold Temperature	°C	62.4
Exhaust Temperature before Turbine	°C	636.4
Exhaust Temperature after Turbine	°C	459.8
Ambient Pressure	bar	1.01
Exhaust Pressure before Turbine	bar	3.13
Exhaust Pressure after Turbine	bar	1.01
Turbocharger Initial Speed	RPM	109064

The comparison of results for WHSC 10 for simulation and experimental tests is as summarized in ***Table 21***.

Table 21 Comparison of simulation and experimental results for WHSC 10

Parameter	Units	Experimental Result	Simulation Result	Difference
Brake Torque	N-m	899	832	67
Gross IMEP[12]	bar	19.2	18.9	0.3
A/F Ratio	-	23.7	24.7	1
Maximum Cylinder Pressure (P_{max})	bar	144	157	-13
CAD for P_{max}	deg bTDC	12	11	1

The in-cylinder pressure trace obtained from the pressure transducer installed on cylinder-5 of the engine is compared with the simulation results, which is as shown in ***Figure 23***.

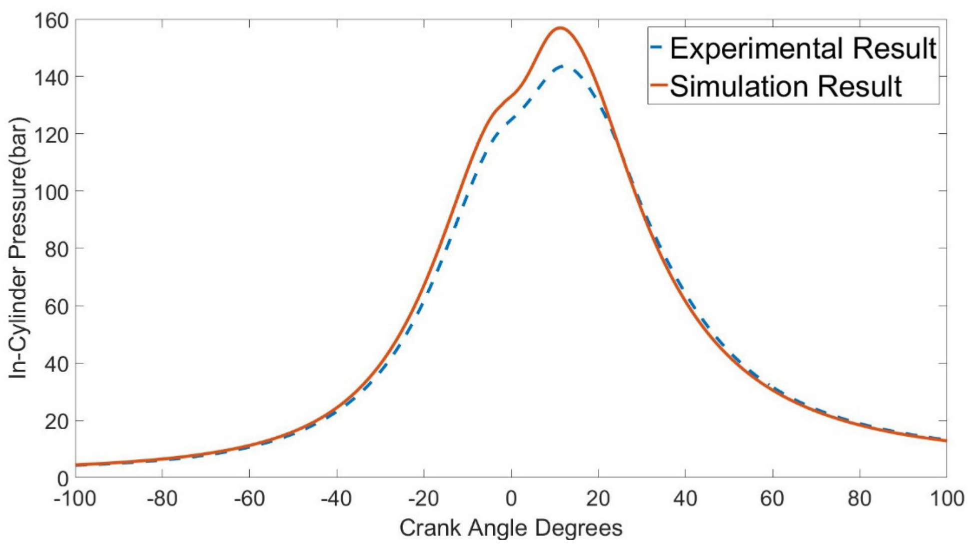

Figure 23 *Comparison of In-cylinder pressure results for simulation and experimental tests for WHSC 10*

The log p-v diagram comparison for the simulation and experimental tests is as shown in ***Figure 24***.

[12] Gross IMEP for Cylinder 5 from engine and simulation

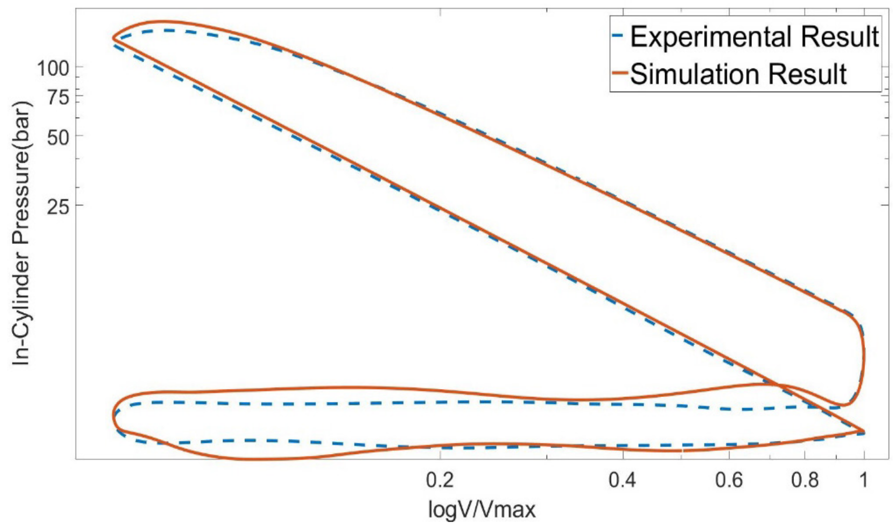

Figure 24 *Comparison of pressure vs volume results for simulation and experimental tests for WHSC 10*

Apparent heat release rate calculated from the pressure and the volume data for both experimental tests and simulation for WHSC 10 is as shown in ***Figure 25***.

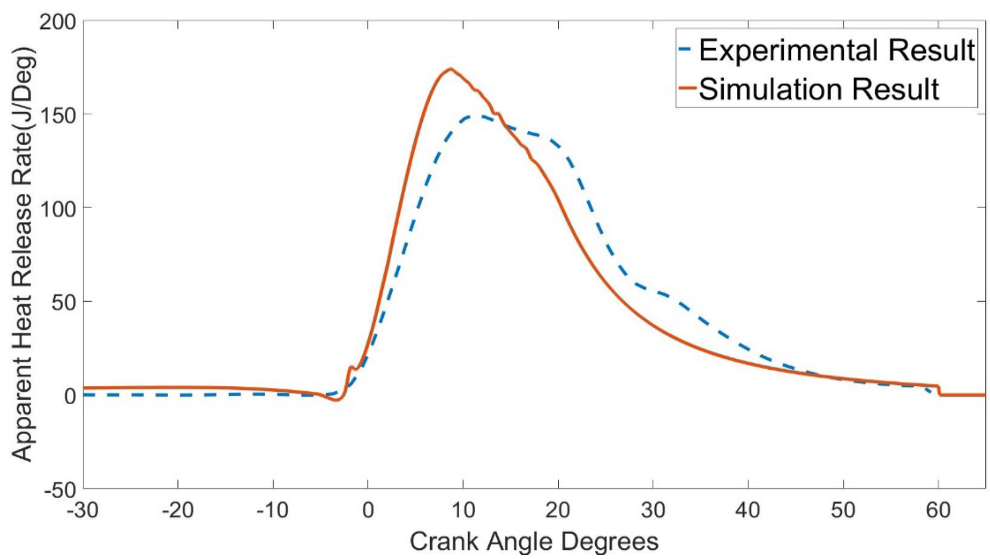

Figure 25 *Comparison of apparent heat release rate results for simulation and experimental tests for WHSC 10*

5.3.4 Validation of ESC 7

The diesel engine simulation model is simulated for ESC case 7, which is a medium-speed low-load case. The input data for the model is as summarized in ***Table 22***

Table 22 *Operating conditions input data for ESC 7 test case for diesel simulation model*

Parameter	Unit	Value
Engine Speed	RPM	1618
EGR Target	%	13
Diesel mass injected	mg/stroke	35.6
Main SOI	deg bTDC	-1.91
Fuel Rail Pressure	bar	1353.4
Boost Pressure	bar	1.27

The initialization parameters used for the simulation are as summarized in ***Table 23***.

Table 23 *Initialization data for ESC 7 test case for diesel simulation model*

Parameter	Unit	Value
Ambient Temperature	°C	17.7
Coolant Temperature	°C	82.1
Cooled Compressed Air temperature	°C	19.1
Fuel Temperature	°C	28
Intake Manifold Temperature	°C	38.1
Exhaust Temperature before Turbine	°C	374.1
Exhaust Temperature after Turbine	°C	300
Ambient Pressure	bar	1.01
Exhaust Pressure before Turbine	bar	2.11
Exhaust Pressure after Turbine	bar	1.01
Turbocharger Initial Speed	RPM	56184

The comparison of results for ESC 7 for simulation and experimental tests is as summarized in ***Table 24***.

Table 24 *Comparison of simulation and experimental results for ESC 7*

Parameter	Units	Experimental Result	Simulation Result	Difference
Brake Torque	N-m	220	227	-7
Gross IMEP[13]	bar	6.1	6.4	-0.3
A/F Ratio	-	37.7	36.0	1.7
Maximum Cylinder Pressure (P_{max})	bar	65	64	1
CAD for P_{max}	deg bTDC	1	15	-14

The in-cylinder pressure trace obtained from the pressure transducer installed on cylinder-5 of the engine is compared with the simulation results, which is as shown in ***Figure 26***.

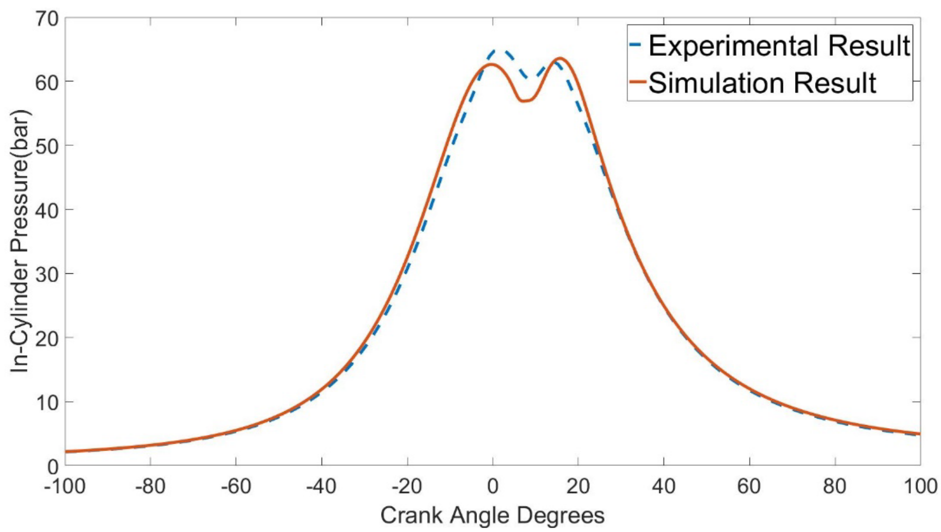

Figure 26 *Comparison of In-cylinder pressure results for simulation and experimental tests for ESC 7*

The log p-v diagram comparison for the simulation and experimental tests is as shown in ***Figure 27***.

[13] Gross IMEP for Cylinder 5 from engine and simulation

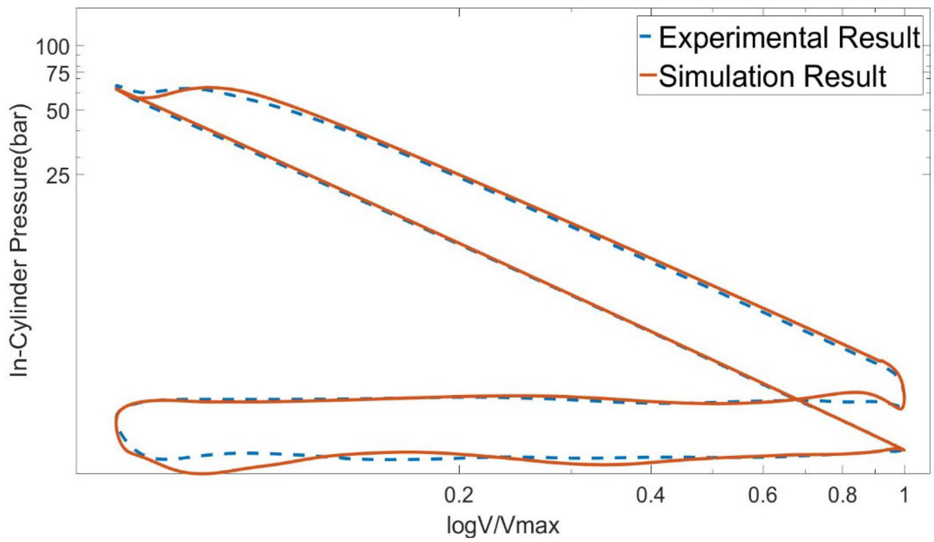

Figure 27 *Comparison of pressure vs volume results for simulation and experimental tests for ESC 7*

Apparent heat release rate calculated from the pressure and the volume data for both experimental tests and simulation for ESC 7 is as shown in ***Figure 28***.

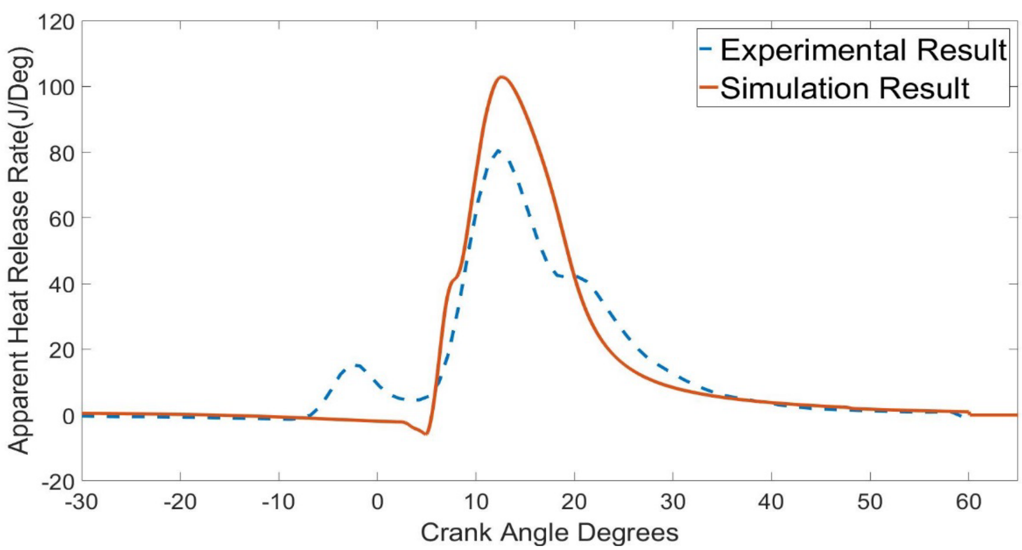

Figure 28 *Comparison of apparent heat release rate results for simulation and experimental tests for ESC 7*

5.3.5 Validation of ESC 8

The diesel engine simulation model is simulated for ESC case 8, which is a high-speed high-load case. The input data for the model is as summarized in ***Table 25***

Table 25 *Operating conditions input data for ESC 8 test case for diesel simulation model*

Parameter	Unit	Value
Engine Speed	RPM	1980
EGR Target	%	8
Diesel mass injected	mg/stroke	102.8
Main SOI	deg bTDC	5.3
Fuel Rail Pressure	bar	1677.6
Boost Pressure	bar	2.39

The initialization parameters used for the simulation are as summarized in ***Table 26***.

Table 26 *Initialization data for ESC 8 test case for diesel simulation model*

Parameter	Unit	Value
Ambient Temperature	°C	18.7
Coolant Temperature	°C	92.8
Cooled Compressed Air temperature	°C	53.2
Fuel Temperature	°C	26.8
Intake Manifold Temperature	°C	57.4
Exhaust Temperature before Turbine	°C	630.5
Exhaust Temperature after Turbine	°C	464.3
Ambient Pressure	bar	1.01
Exhaust Pressure before Turbine	bar	3.12
Exhaust Pressure after Turbine	bar	1.01
Turbocharger Initial Speed	RPM	103515

The comparison of results for ESC 8 for simulation and experimental tests is as summarized in ***Table 27***.

Table 27 *Comparison of simulation and experimental results for ESC 8*

Parameter	Units	Experimental Result	Simulation Result	Difference
Brake Torque	N-m	896	860	36
Gross IMEP[14]	bar	18.9	18.5	0.4
A/F Ratio	-	22.8	23.5	-0.7
Maximum Cylinder Pressure (P_{max})	bar	139.6	151.7	-12.1
CAD for P_{max}	deg bTDC	12	12	0

The in-cylinder pressure trace obtained from the pressure transducer installed on cylinder-5 of the engine is compared with the simulation results, which is as shown in ***Figure 29***.

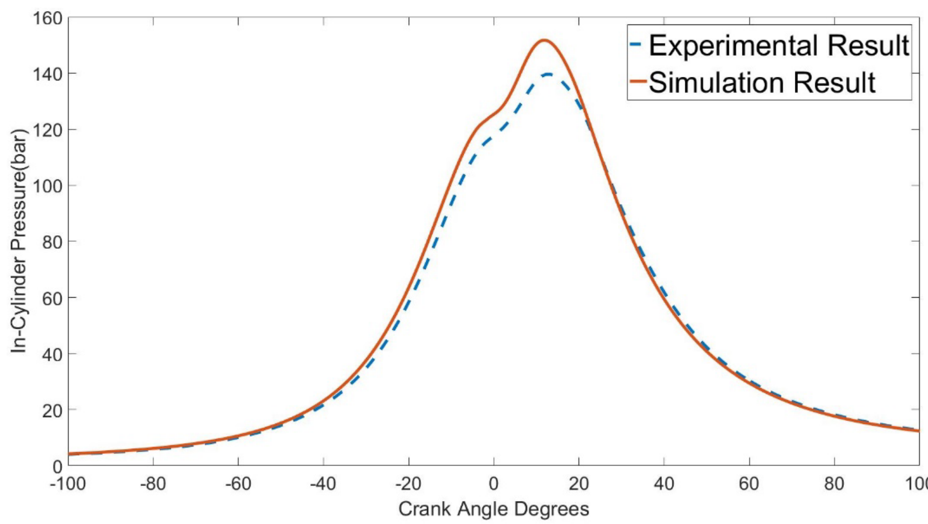

***Figure 29** Comparison of In-cylinder pressure results for simulation and experimental tests for ESC 8*

The log p-v diagram comparison for the simulation and experimental tests is as shown in ***Figure 30***.

[14] Gross IMEP for Cylinder 5 from engine and simulation

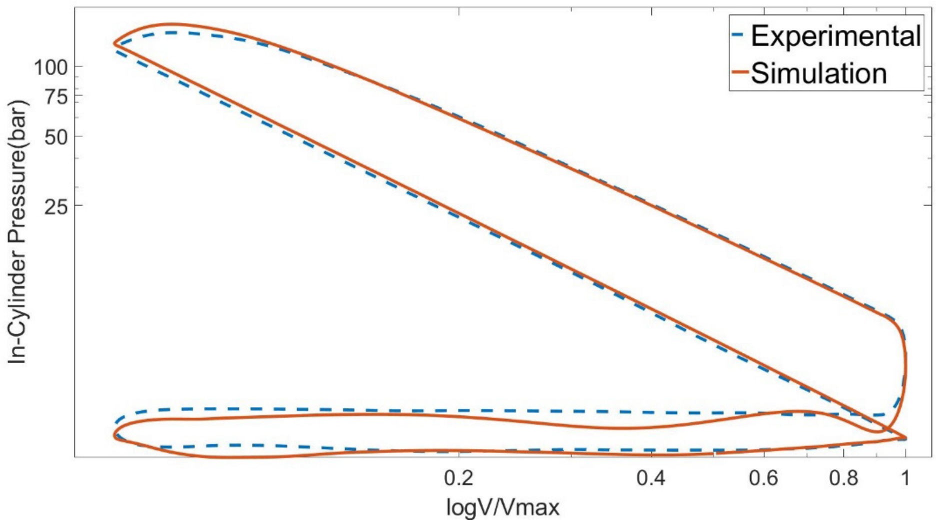

Figure 30 *Comparison of pressure vs volume results for simulation and experimental tests for ESC 8*

Apparent heat release rate calculated from the pressure and the volume data for both experimental tests and simulation for *ESC 8* is as shown in ***Figure 31***.

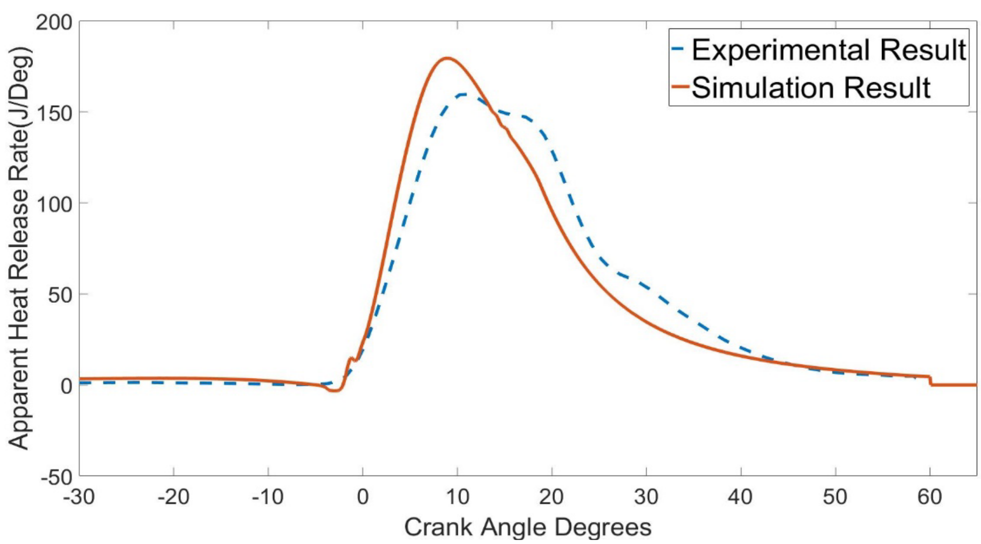

Figure 31 *Comparison of apparent heat release rate results for simulation and experimental tests for ESC 8*

5.3.6 Validation of ESC 12

The diesel engine simulation model is simulated for ESC case 12, which is a high-speed medium-load case. The input data for the model is as summarized in ***Table 28***.

Table 28 *Operating conditions input data for ESC 12 test case for diesel simulation model*

Parameter	Unit	Value
Engine Speed	RPM	2340
EGR Target	%	18
Diesel mass injected	mg/stroke	81.9
Main SOI	deg bTDC	4.5
Fuel Rail Pressure	bar	1714.5
Boost Pressure	bar	2.28

The initialization parameters used for the simulation are as summarized in **Table 29**.

Table 29 *Initialization data for ESC 12 test case for diesel simulation model*

Parameter	Unit	Value
Ambient Temperature	°C	19.7
Coolant Temperature	°C	93.9
Cooled Compressed Air temperature	°C	51.7
Fuel Temperature	°C	27.1
Intake Manifold Temperature	°C	62.7
Exhaust Temperature before Turbine	°C	596
Exhaust Temperature after Turbine	°C	436.2
Ambient Pressure	bar	1.01
Exhaust Pressure before Turbine	bar	3.15
Exhaust Pressure after Turbine	bar	1.01
Turbocharger Initial Speed	RPM	101679

The comparison of results for ESC 12 for simulation and experimental tests is as summarized in ***Table 30***.

__Table 30__ Comparison of simulation and experimental results for ESC 12

Parameter	Units	Experimental Result	Simulation Result	Difference
Brake Torque	N-m	673	609	64
Gross IMEP[15]	bar	15.3	14.7	0.6
A/F Ratio	-	23.3	24.6	-1.3
Maximum Cylinder Pressure (P_{max})	bar	117.5	122.9	-5.4
CAD for P_{max}	deg bTDC	12	13	-1

The in-cylinder pressure trace obtained from the pressure transducer installed on cylinder-5 of the engine is compared with the simulation results, which is as shown in ***Figure 32***.

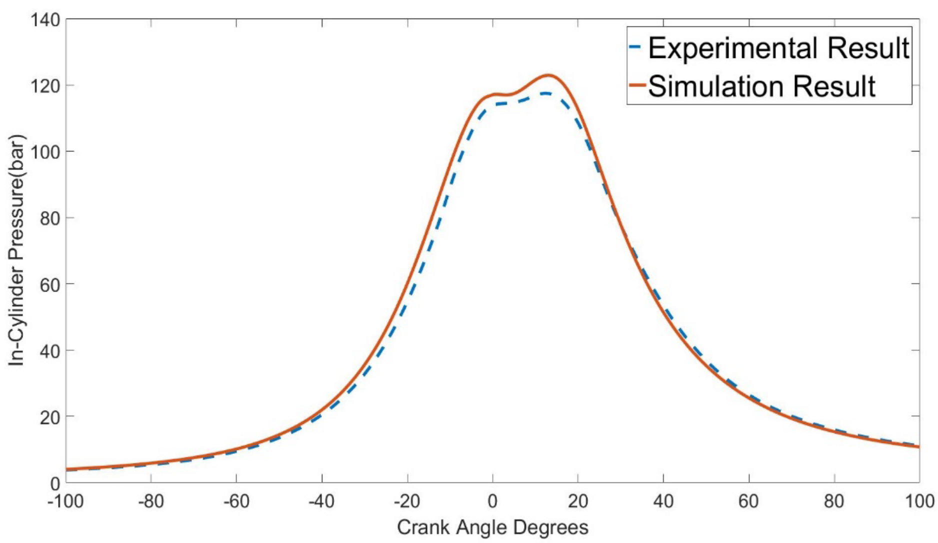

__Figure 32__ Comparison of In-cylinder pressure results for simulation and experimental tests for ESC 12

The log p-v diagram comparison for the simulation and experimental tests is as shown in ***Figure 33***.

[15] Gross IMEP for Cylinder 5 from engine and simulation

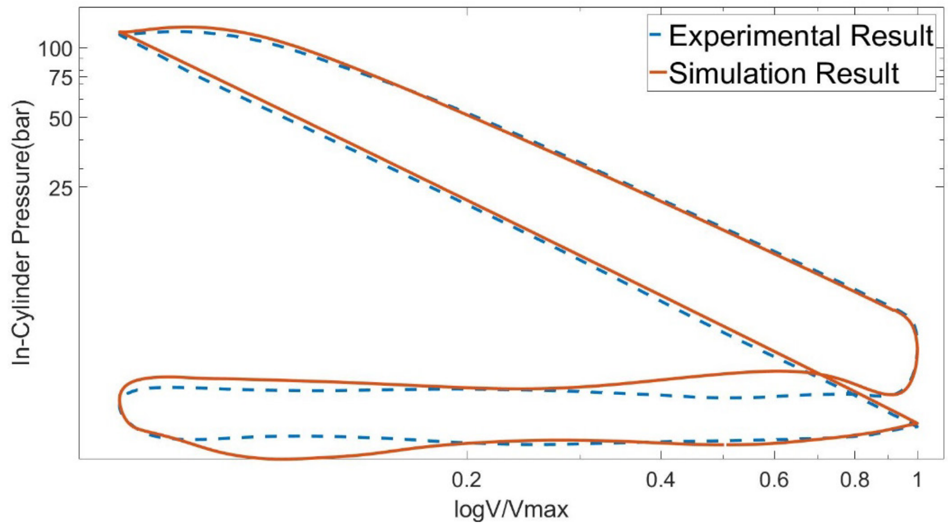

***Figure 33** Comparison of pressure vs volume results for simulation and experimental tests for ESC 12*

Apparent heat release rate calculated from the pressure and the volume data for both experimental tests and simulation for *ESC 12* is as shown in ***Figure 34***.

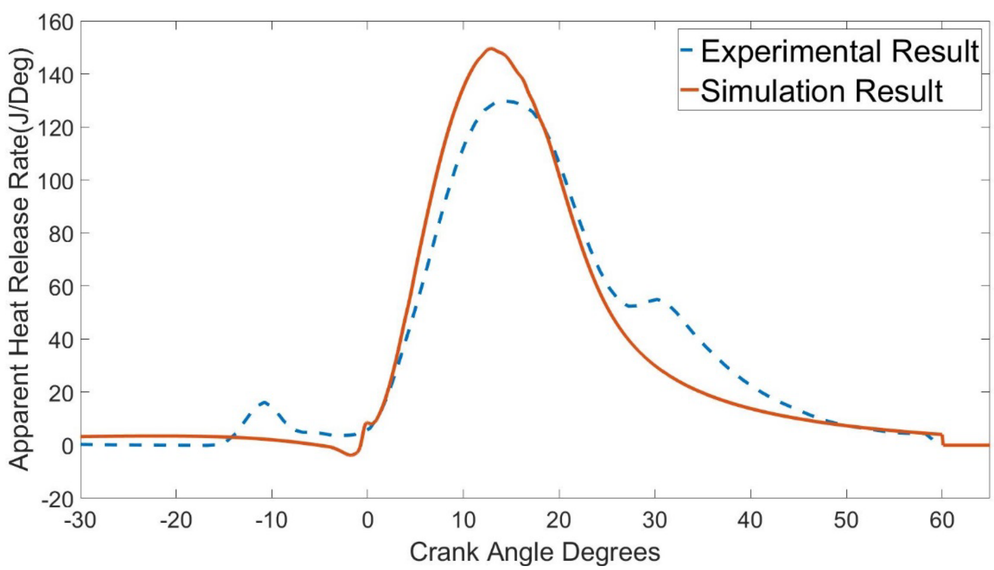

***Figure 34** Comparison of apparent heat release rate results for simulation and experimental tests for ESC 12*

For validation of the simulation for the engine operating conditions, the calibrated values for combustion constants as described in ***Table 12*** have been used. For the validated cases, the difference between the engine experimental results and the simulation results was found to be less than 5% for all the engine performance parameters. It can be concluded that the diesel combustion model DI-Pulse is calibrated and the combustion constants could be used for the simulation of the dual-fuel engine.

6 Calibration and Test Procedure for Dual-Fuel Simulation Model

This chapter describes the calibration and test procedure followed for the dual-fuel simulation model.

6.1 Calibration of Dual Fuel Model

To perform calibration of the dual fuel simulation model, experimental data is not available for the dual-fuel mode of the Cummins 2010 ISB 6.7L engine. As described in section **3.2.2,** the dual fuel combustion model used to calculate the burn rate of the dual-fuel model consists of two sub-models: DI Pulse and SI Turbulent flame models. The transition from one model to the other is modeled as a linear transition from diesel jet to spherical flame. The combustion parameters for DI-Pulse model are used from the calibration results obtained from ***Table 12*.** The combustion constants for the SI turbulent flame and the combined effect of the combustion parameters on the combustion and power output had to be studied.

To analyze the effects of the combustion model, a case study was performed in which the combustion constants for the SI turbulent flame model were varied over the range specified as in ***Table 10*** to study the effect on the BMEP and combustion. As the experimental result provided by Westport Fuel Systems Inc., was from an engine with different specifications,

including a different displacement volume from the ISB 6.7L engine modelled here, the simulation results could not be matched closely to the experimental results.

The trend for the in-cylinder pressure trace and the location of the peak pressure are used for determination of the values of combustion constants for the dual fuel combustion model. The engine operating condition for which calibration study is performed is as given in ***Table 31***.

Table 31 *Experimental data for calibration of Dual-Fuel model*[16]

Parameter	Unit	Value
Engine Speed	RPM	1200.0
Air-Fuel Ratio for Natural Gas	kg/kg	16.7
Diesel Fuel Mass per Injection	mg/shot	5.6
CNG Fuel Mass per Injection	mg/shot	125.9
Intake Air Temperature (Compressor Inlet)	°C	25.3
Intake Air Temperature (Compressor Outlet)	°C	151.4
Intake Air Temperature (Charge Air Cooler Outlet)	°C	45.7
Intake Air Temperature (Inlet Manifold)	°C	40.8
Exhaust Temperature (Turbine Inlet)	°C	721.6
Exhaust Temperature (Turbine Outlet)	°C	487.8
Intake Air Pressure (Inlet Manifold)	kPa	229.5
Diesel SOI [aTDC]	Degree	0.0
Fuel Injection Pressure	bar	1000
EGR Rate	%	9.75

[16] Data provided by Westport Fuel Systems Inc.,

For the specified engine operating condition, dual-fuel model was setup and test cases were simulated for various values of multipliers specified in ***Table 10***. Trial and error method was used to study the effect of each of the combustion constant on the pressure trace. It was observed that to obtain a pressure trace similar to the experimental result, the value of the combustion constants- Dilution Exponent multiplier, Flame Kernel Growth Multiplier, Turbulent Flame Speed multiplier must be maintained as 3 (maximum value specified in Engine Performance Manual of GT-Power). The pressure trace was dependent on Taylor length scale multiplier. For various values of Taylor length scale multiplier, the effect on the pressure trace is observed and the results are as summarized in ***Table 32***.

Table 32 *Simulation results for variation in Taylor length scale multiplier (T), comparison with experimental results*

Parameter	Units	Experimental Result	Simulation Result			
			T=0.5	**T=0.75**	**T=1**	**T=1.5**
Brake Torque	N-m	1075.8	1124.5	1123.4	1119.3	1103.1
Brake Power	kW	135.2	141.3	141.2	140.7	138.6
Thermal Efficiency	%	35.2	38.3	38.1	37.8	37.1
Peak Cylinder Pressure (P_{max})	bar	116.1	147.1	136.3	126.5	110.0
Location of P_{max}	Deg aTDC	21.2	19.8	21.5	23.0	25.5
Ratio of Peak Pressures	-	1.22	1.37	1.27	1.17	1.02

From ***Table 32***, it is concluded that, when the Taylor length scale multiplier is varied over the range of 0.5-1.5, the engine performance parameters- Brake Torque, Brake Power and thermal efficiency do not vary significantly.

The in-cylinder pressure comparison for the simulation results for the variation of Taylor length scale multiplier with the experimental result is as shown in ***Figure 35***.

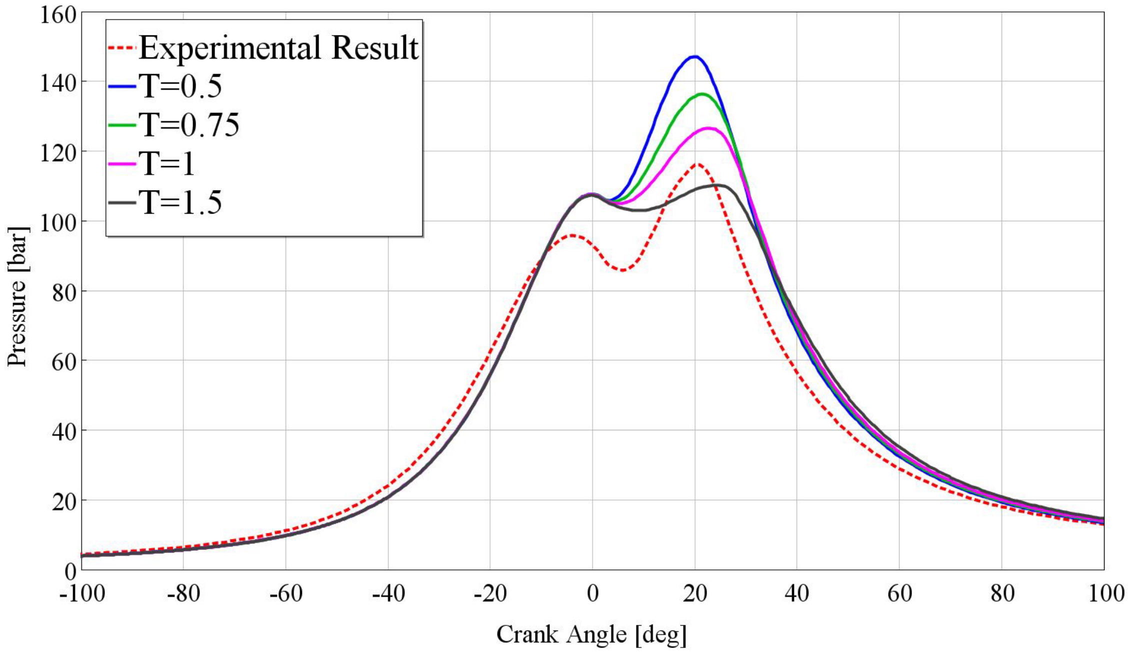

Figure 35 *Comparison of simulation results for variation of Taylor length scale multiplier(T) with experimental results*

The ratio of peak pressures is defined as the ratio of combustion peak pressure to the motoring peak pressure and the values obtained for experimental and the simulation are as summarized in ***Table 32***. Based on the ratio of peak pressures and the location of peak cylinder pressures, simulation result for Taylor length scale multiplier 0.75 is found to

match the experimental result. The Taylor length scale multiplier is hence selected as 0.75 for the dual-fuel model.

The summary of the calibration results for the dual-fuel engine model is as given in ***Table 33***.

Table 33 *Dual-Fuel model calibration results*

Parameter	Value
Entrainment Rate Multiplier	1.54
Ignition Delay Multiplier	0.30
Premixed Combustion Rate Multiplier	2.47
Diffusion Combustion Rate Multiplier	1.08
Dilution Exponent Multiplier	3
Flame Kernel Growth Multiplier	3
Turbulent Flame Speed Multiplier	3
Taylor Length Scale Multiplier	0.75

6.2 Simulation Procedure for Dual-Fuel Engine

The objective of the tests is to achieve high brake mean effective pressure (BMEP) through the effect of gas charging, EGR and injection pressures on dual-fuel engine for operating at stoichiometric air-fuel ratio and minimum diesel energy contribution. Stoichiometric conditions and charge dilution with EGR are selected for dual-fuel mode simulation, to meet the targets of the DOE project to use a simple TWC aftertreatment system. A target BMEP of 25 bar is set and a test matrix is designed to study effect of each of the parameter- EGR, Boost Pressure and Injection pressure on BMEP.

From the experimental engine data of test cycles WHSC and ESC, it is observed that the engine delivered a maximum gross indicated mean effective pressure ($IMEP_{gross}$) of 19.2 bar for test mode WHSC 10, which is a high-speed high load case. In the low speed range, maximum $IMEP_{gross}$ of 18.2 bar was observed for test mode WHSC 5, which is a low-speed, high load case. The maximum boost pressure (Intake Manifold Pressure, measured in bar) for the test cases is observed to be 2.57 bar. The test conditions are summarized in ***Table 34***.

Table 34 *Data for engine operating conditions WHSC 5 and WHSC 10*

Parameter	Unit	WHSC5	WHSC10
Engine Speed	RPM	1397	2138
EGR Target	%	0	6
Diesel mass injected	mg/stroke	107.9	104.2
Main SOI	deg bTDC	-0.33	5.94
Fuel Rail Pressure	bar	1139	1707.4
Boost Pressure	Bar	1.83	2.52

6.2.1 Simulation Matrix and assumptions

To achieve 25 bar BMEP in dual-fuel simulation model, the following assumptions are made based on data for the operating conditions for WHSC 5 & WHSC 10.

- The boost pressure for the simulation of dual-fuel model is assumed to be 2.5 bar
- The injection pressures for diesel are assumed to be 300, 600, 1000 bar
- The EGR conditions for the dual-fuel simulation are selected to be 0,6,12 and 18%
- The stoichiometric Air-Fuel Ratio for Natural gas is 16.77[17]
- Lower Heating Value (LHV) for Natural gas is 48.99 MJ/kg[17]
- LHV for diesel is 43 MJ/kg[17]

Based on the assumptions a test matrix is created for high-speed (2138 RPM) and low-speed (1397 RPM) which is as shown in ***Figure 36***.

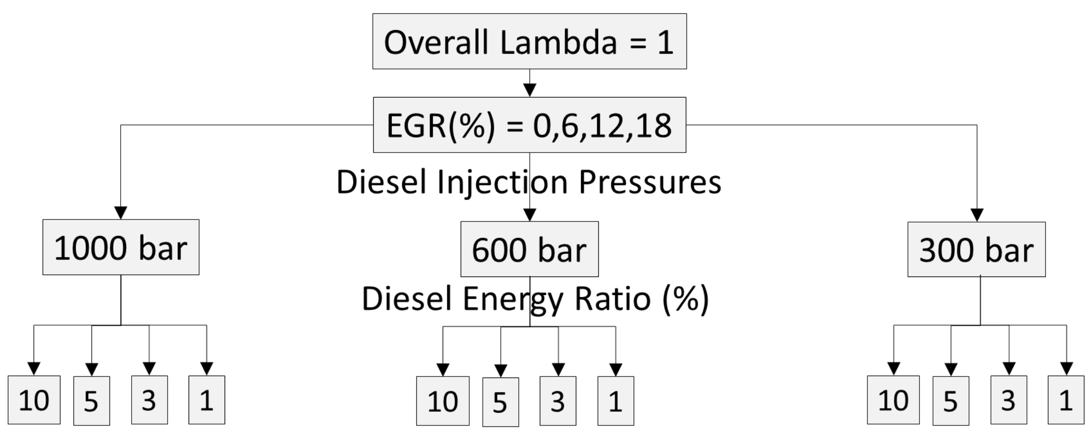

Figure 36 *Test matrix for Dual-Fuel simulation*

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[17] Obtained from post-processing of simulation results using GT-Post software

6.2.2 Calculation of Air-Fuel Ratio for Natural Gas

The objective of the study is to simulate the dual-fuel engine at stoichiometric conditions. For various diesel energy proportions, the air-fuel ratio for natural gas differs to maintain the overall relative air/fuel ratio ($\lambda_{overall}$) to be equal to one. To calculate the air-fuel ratio for natural gas for different diesel energy proportions, the procedure followed is described in this section.

6.2.2.1 Diesel Energy Contribution

Based on the lower heating values of the diesel fuel and the natural gas (NG), the relation between the diesel mass flow rate and the natural gas flow rate is calculated as the first step. The total fuel energy is defined as the sum of energy contribution of diesel and the energy contribution of natural gas, given by equation **(7)**. The diesel energy contribution percentage is hence defined as the energy content of diesel to the total energy content of the fuel, given by equation **(8).**

$$Total\ Fuel\ Energy = \dot{m}_{diesel} * LHV_{diesel} + \dot{m}_{NG} * LHV_{NG} \tag{7}$$

$$Diesel\ Energy\ (\%) = \frac{(\dot{m}_{diesel} * LHV_{diesel})}{Total\ Fuel\ Energy} \tag{8}$$

As defined in the assumptions, LHV_{diesel} = 43 MJ/kg and LHV_{NG} = 48.99 MJ/kg.

For 10% energy contribution, relation between the fuel flow rates of diesel and natural gas (NG) is obtained by modifying the equation **(7)** is as given by equation **(9)**

$$\dot{m}_{diesel,10\%} = 0.1266 * \dot{m}_{NG} \tag{9}$$

Similarly the relation for fuel flow rates for 5, 3 and 1 % diesel energy contribution is calculated and is given by equations **(10)**, **(11)** and **(12)** respectively.

$$\dot{m}_{diesel,5\%} = 0.0599 * \dot{m}_{NG} \tag{10}$$

$$\dot{m}_{diesel,3\%} = 0.0352 * \dot{m}_{NG} \tag{11}$$

$$\dot{m}_{diesel,1\%} = 0.0115 * \dot{m}_{NG} \tag{12}$$

6.2.2.2 Relative Air-fuel Ratio Calculation

The relative air-fuel ratio for the dual-fuel engine is defined as the ratio of actual air-fuel ratio (AFR_{actual}) to the stoichiometric air-fuel ratio (AFR_{stoich}), given by equation **(13)**

$$\lambda_{overall} = \frac{AFR_{actual}}{AFR_{stoich}} \tag{13}$$

Stoichiometric air-fuel ratio for the combined fuel (Natural gas and the diesel), as obtained from the post processing software GT-Post is found to be 16.77. The actual air-fuel ratio is given by equation **(14)**

$$AFR_{actual} = \frac{\dot{m}_{air}}{(\dot{m}_{diesel} + \dot{m}_{NG})} \tag{14}$$

Based on the values for the stoichiometric Air-fuel ratio for combined fuel and equation **(14)**, equation **(13)** simplifies to **(15).**

$$\lambda_{overall} = \frac{\dot{m}_{air}/(\dot{m}_{diesel} + \dot{m}_{NG})}{16.77} \tag{15}$$

For stoichiometric mixtures, $\lambda_{overall} = 1$. Based on this, the equation the relation between the mass flow rates of air and fuel is as given by equation

$$\frac{\dot{m}_{air}}{(\dot{m}_{diesel} + \dot{m}_{NG})} = 16.77 \tag{16}$$

From equations **(9)**, **(10)**, **(11)** and **(12)**, and by the definition of air fuel ratio of natural gas, the equation **(16)** is simplified to obtain air-fuel ratio of natural gas for each of the energy contributions as summarized in ***Table 35***

Table 35 *Natural gas Air-fuel ratio for diesel energy contributions to keep overall relative air-fuel ratio equal to one*

Diesel Energy Contribution (%)	Air-Fuel Ratio for Natural Gas
1	18.89
3	17.77
5	17.36
10	16.96

The relation between the energy contribution and the air-fuel ratio of natural gas required to keep the overall relative air-fuel ratio ($\lambda_{overall}$) as 1 is graphically represented as shown in ***Figure 37***.

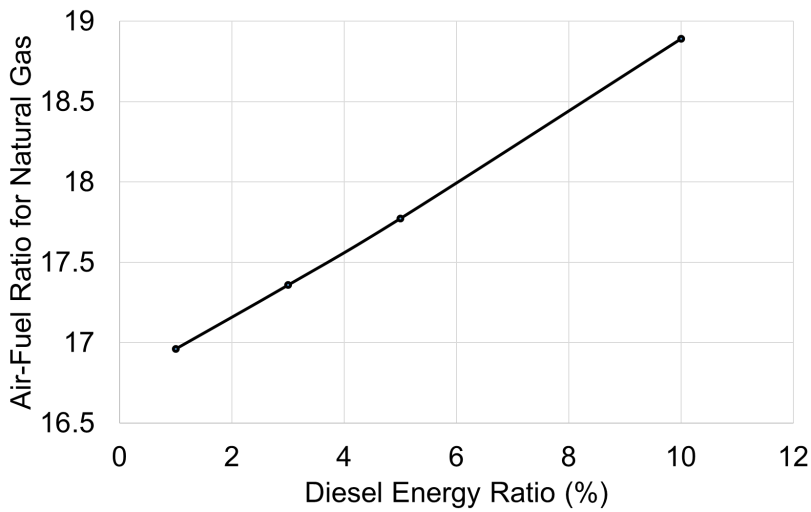

Figure 37 *Relation between diesel energy ratio and natural gas Air-fuel ratio*

6.2.3 Simulation Conditions for the Dual-Fuel Engine

The engine operating conditions for the dual-fuel simulation are summarized in ***Table 36***.

Table 36 *Test conditions for dual fuel engine simulation*

Parameter	Unit	High Speed	Low Speed
Engine Speed	RPM	2138	1397
Boost Pressure	bar	2.5	
Overall Lambda	-	1	
Diesel Injection Pressure	bar	300, 600, 1000	
Diesel Energy Contribution	%	1, 3, 5, 10	
EGR	%	0, 6, 12, 18	

At each engine speed and diesel injection timing (SOI), a sweep is performed over the values of EGR from 0 to 18%, diesel injection pressures of 300, 600 and 1000 bar and the diesel energy contributions of 1, 3, 5 and 10%. The diesel energy contribution for each of the simulation case is calculated based on the natural gas flow obtained from the simulations as per equations (9)-(12).

7 Results

Dual-fuel model is simulated at engine speed conditions 1397 RPM and 2138 RPM. The simulation results and the effects of injection timing of diesel, diesel energy contributions, and EGR levels on the engine performance are discussed in this chapter.

The air-fuel ratio obtained for natural gas as described in section **6.2.2** is used in the simulation for the corresponding diesel energy contributions 1, 3, 5 and 10% and the dual-fuel engine model is simulated.

7.1 Effect of Diesel Injection Timing on Engine Performance

To study the effect of Diesel injection timing on engine performance, engine-operating speed of 2138 RPM is simulated for diesel injection timings from -10° aTDC to 40° bTDC. The simulation conditions for the injection timing sweep are summarized in ***Table 37***.

Table 37 *Test conditions for injection timing sweep for dual fuel engine*

Parameter	Unit	Value
Engine Speed	RPM	2138
Boost Pressure	bar	2.5
Overall Lambda	-	1
AFR for natural gas	-	18.89
Diesel Injection Pressure	bar	1000
Diesel Energy Contribution	%	10
EGR	%	0

The results for the injection-timing sweep are as summarized in ***Table 38***.

Table 38 *Results for injection timing sweep for dual fuel engine*

Parameter	Unit	Diesel Injection Timing (degrees bTDC)					
		40	**30**	**20**	**10**	**0**	**-10**
Net IMEP	bar	16.3	20.1	24.0	26.7	26.0	21.9
BMEP	bar	14.0	17.7	21.6	24.7	24.4	20.4
Torque	N-m	742.3	939.8	1150.5	1310.6	1296.7	1082.5
NO_x emissions	ppm	3840.5	3888.3	4043.6	4309.8	4008.1	2761.8
BSFC	g/kW-h	302.7	243.8	204.8	185.9	192.5	232.0
Brake Efficiency	%	24.7	30.6	36.4	40.1	38.7	32.1
PCP	bar	340	342	338	251	147	123
Location of PCP	deg aTDC	-1	-1	4	12	24	0
Burn Duration (0-50%)	CA	10	11	12	14	24	28

As the diesel injection timing is advanced before top dead center (bTDC), the brake efficiency of the engine decreases and the in-cylinder pressure rises to a maximum of 342 bar for 40° bTDC. In addition, the BMEP decreases as the injection is advanced. If the diesel injection timing is delayed after top dead center (aTDC), the burn duration increases and the BMEP reduces. The BMEP of the engine is found to be maximum for diesel injection timing of 10° bTDC. The BMEP at top dead center is close to the value at 10° bTDC and the cylinder pressure is lower at TDC than the value at 10° bTDC.

The in-cylinder pressure comparison for the diesel injection timing variation is as shown in ***Figure 38***.

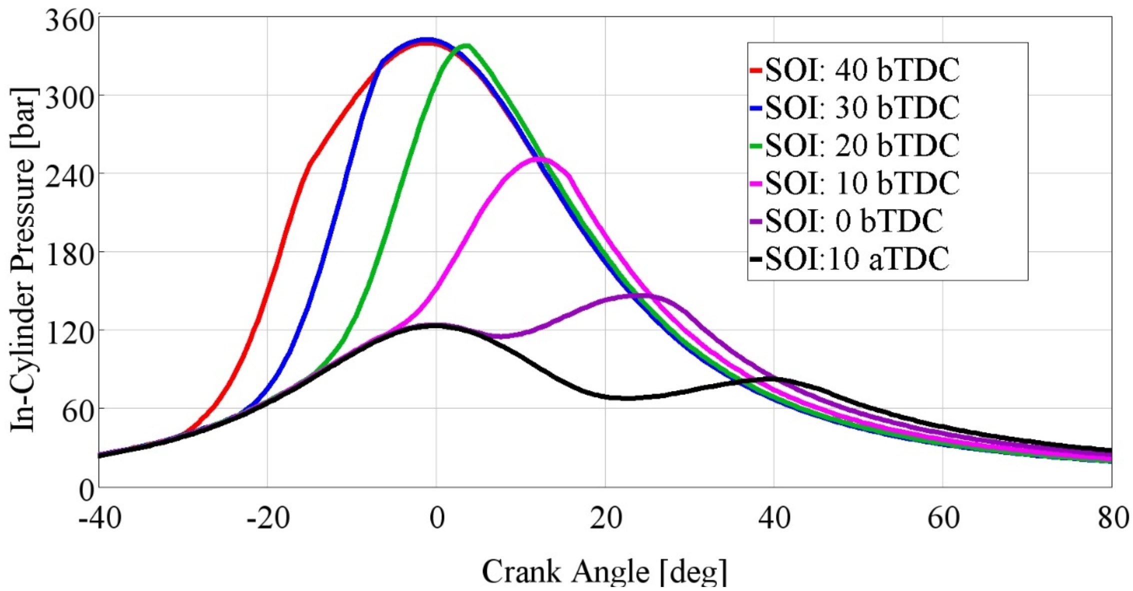

Figure 38 *In-cylinder pressure (Cylinder-1) comparison for variation of diesel injection timing (SOI) for dual-fuel engine simulation*

Considering the high BMEP and high efficiency values for the case of diesel injection timings 10° bTDC and 0° bTDC, the effects of EGR and diesel energy ratio are analyzed at these two diesel injection timings and the results are presented in the following sections.

7.2 Simulation results for Injection Timing 0° bTDC, Engine Speed = 2138 RPM

For diesel injection timing 0° bTDC, the simulation model for dual-fuel engine is simulated for two different engine speeds: 2138 RPM (High-Speed) and 1397 RPM (Low-Speed).

Simulation results obtained for the dual-fuel engine model at 0° bTDC for high-speed case (Engine Speed = 2138 RPM) are discussed in this section.

7.2.1 Effect of Diesel Energy Contribution on Engine Performance

For EGR percentages of 0,6,12 and 18%, the diesel energy ratio is varied for 1, 3, 5 & 10% of the total fuel energy and tested for diesel injection pressures of 300, 600 and 1000 bar.

7.2.1.1 Diesel Energy Contribution Vs BMEP

The BMEP results for the variation of diesel energy ratio for each fraction of EGR are as summarized from ***Figure 39*** to ***Figure 42***. It is observed that as the diesel energy ratio increases, the BMEP of the engine increases. Higher injection pressures have higher BMEP values across the different energy ratios. The trend is similar for the EGR levels from 0 to 18%. The BMEP increase between the diesel energy ratios 1-5% is found to be more than that of the increase between 5-10% diesel energy ratio.

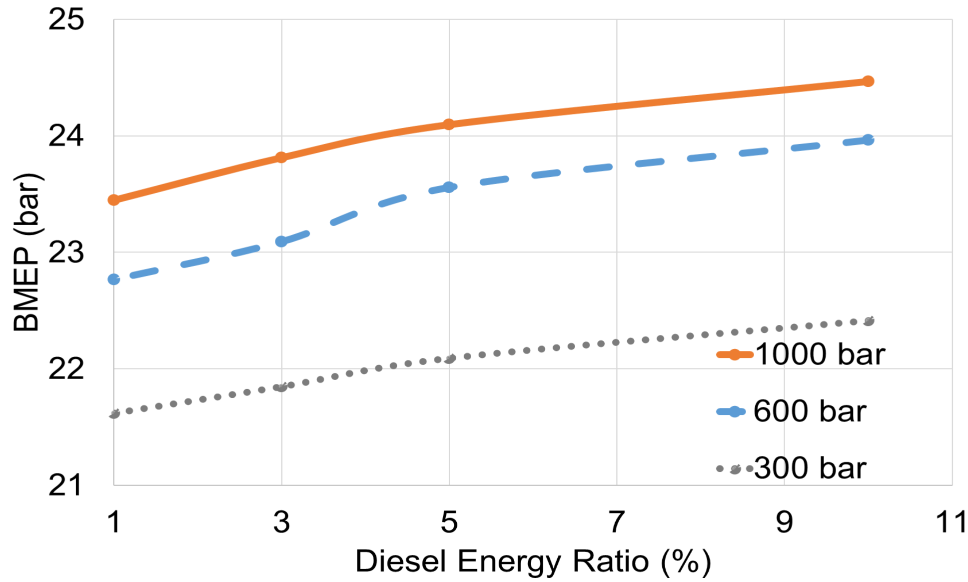

***Figure 39** Engine Speed = 2138 RPM, SOI=0° BTDC: Effect of Diesel Energy Ratio (%) on BMEP (bar) for **EGR=0** % for diesel injection pressures of 300,600 and 1000 bar*

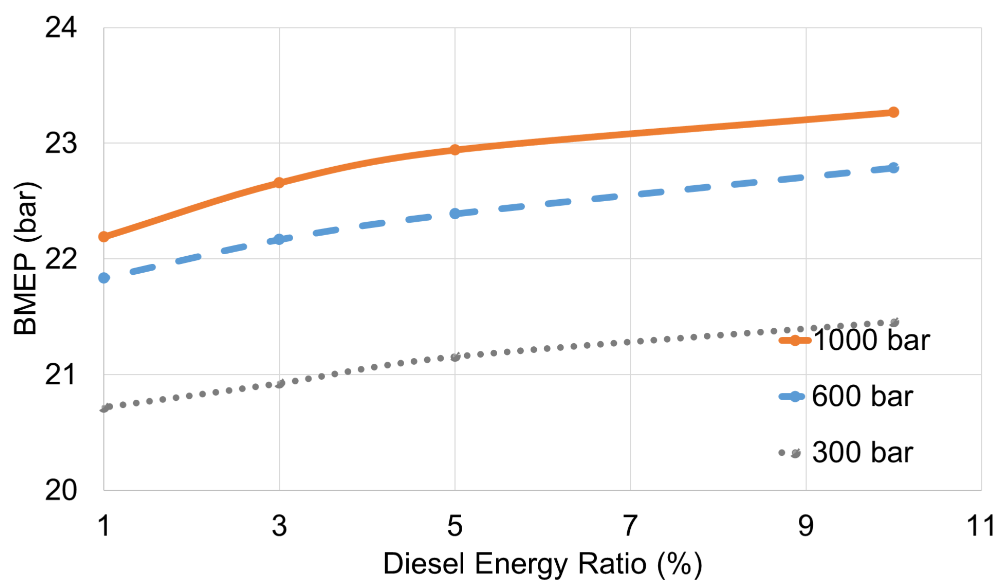

***Figure 40** Engine Speed = 2138 RPM, SOI=0° BTDC: Effect of Diesel Energy Ratio (%) on BMEP (bar) for **EGR=6** % for diesel injection pressures of 300,600 and 1000 bar*

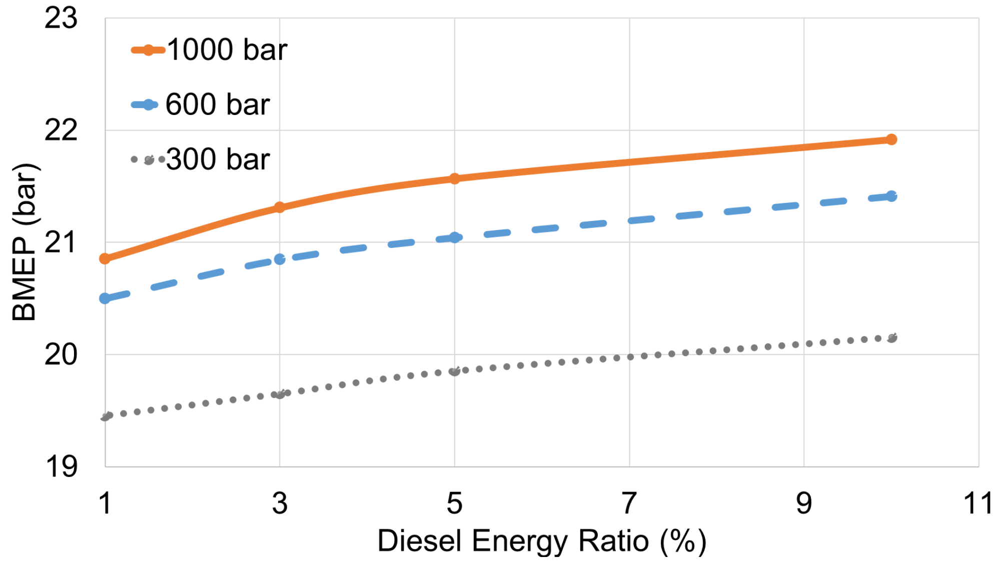

Figure 41 *Engine Speed = 2138 RPM, SOI=0° BTDC: Effect of Diesel Energy Ratio (%) on BMEP (bar) for* ***EGR=12*** *% for diesel injection pressures of 300,600 and 1000 bar*

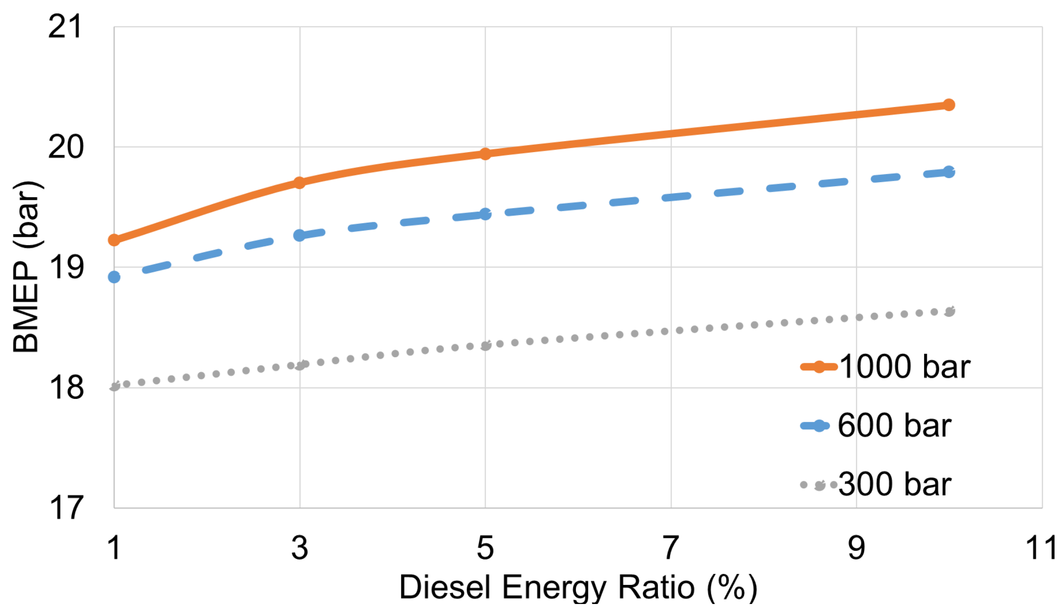

Figure 42 *Engine Speed = 2138 RPM, SOI=0° BTDC: Effect of Diesel Energy Ratio (%) on BMEP (bar) for* ***EGR=18*** *% for diesel injection pressures of 300,600 and 1000 bar*

7.2.1.2 Diesel Energy Contribution Vs NO_x emissions

For the variation of diesel energy contribution for 1,3,5 and 10%, for the diesel injection pressures of 300, 600 and 1000 bar, the NO_x emissions (in PPM) obtained from the simulation for each case of EGR are as shown from ***Figure 43*** to ***Figure 46***. It is observed that as the diesel energy ratio increases, the NO_x emissions increases. Lower diesel injection pressures are observed to have lower NO_x emissions. The trend remains same across all the EGR levels (EGR = 0% to EGR = 18%).

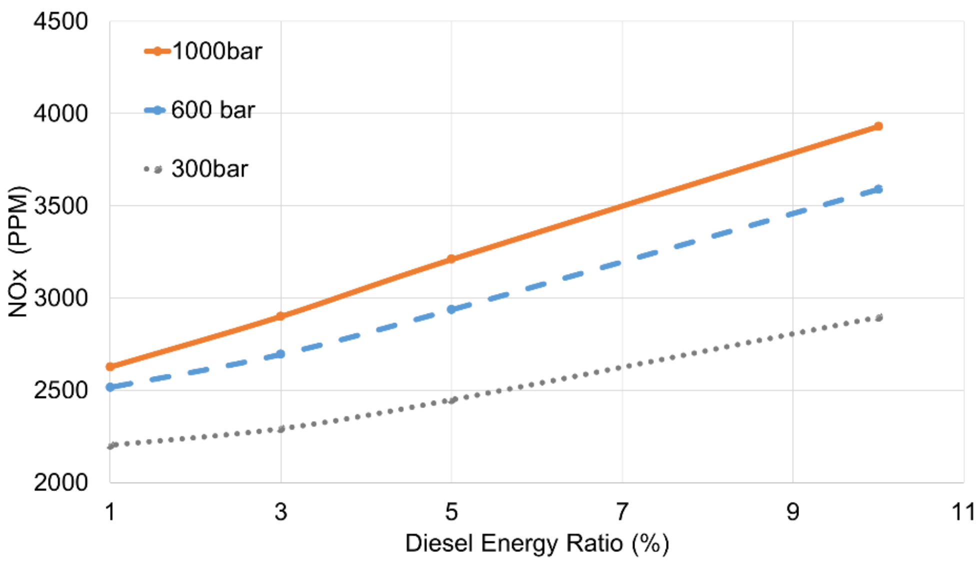

Figure 43 *Engine Speed=2138 RPM, SOI=0° BTDC: Effect of Diesel Energy Ratio (%) on NO_x (PPM) for* ***EGR=0%*** *for diesel injection pressures of 300,600 and 1000 bar*

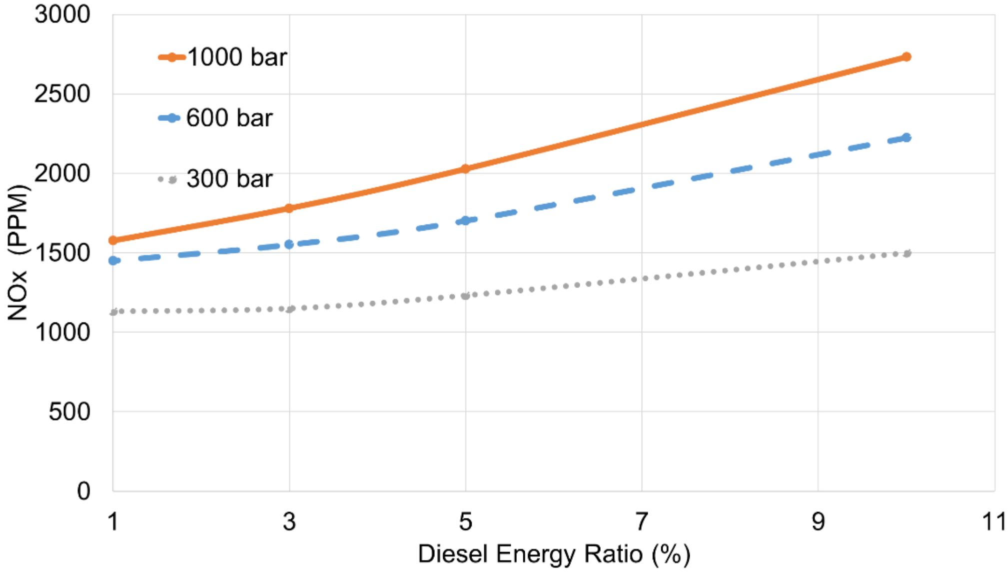

Figure 44 *Engine Speed=2138 RPM, SOI=0° BTDC: Effect of Diesel Energy Ratio (%) on NO_x (PPM) for* ***EGR=6%*** *for diesel injection pressures of 300,600 and 1000 bar*

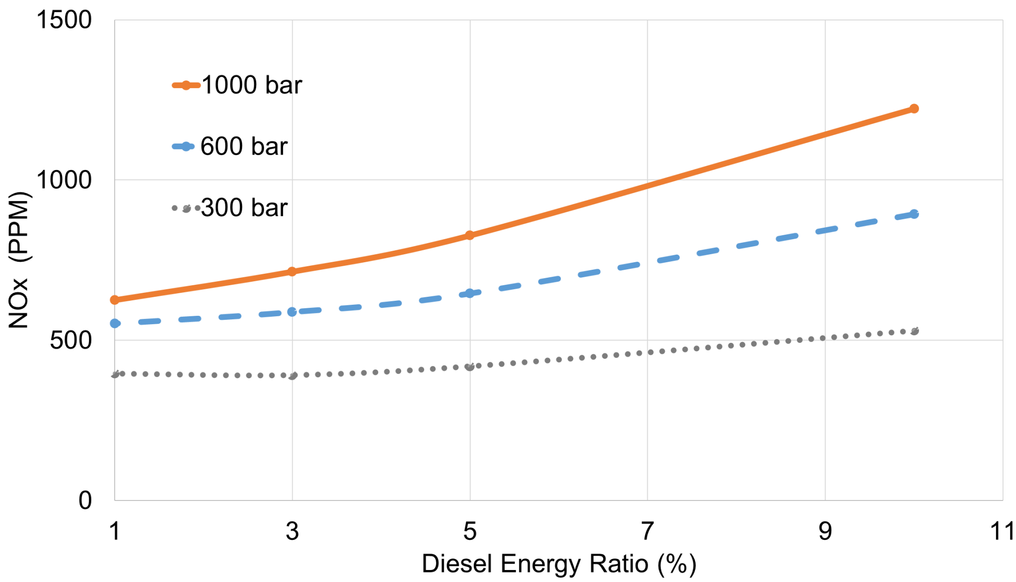

Figure 45 *Engine Speed=2138 RPM, SOI=0° BTDC: Effect of Diesel Energy Ratio (%) on NO_x (PPM) for* ***EGR=12%*** *for diesel injection pressures of 300,600 and 1000 bar*

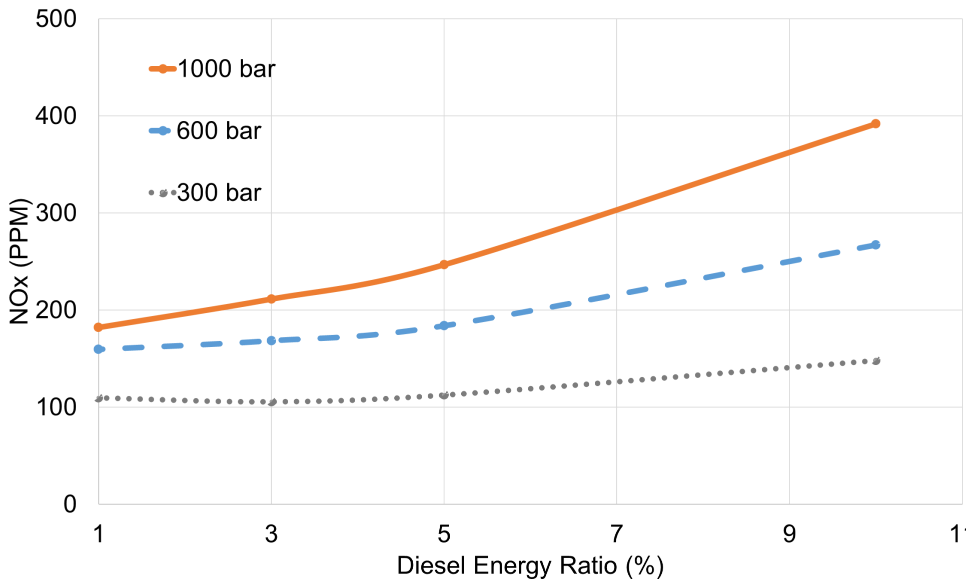

Figure 46 *Engine Speed=2138 RPM, SOI=0° BTDC: Effect of Diesel Energy Ratio (%) on NO_x (PPM) for* ***EGR=18%*** *for diesel injection pressures of 300,600 and 1000 bar*

7.2.1.3 Diesel Energy Contribution Vs Brake Thermal Efficiency

The brake thermal efficiency for the diesel energy ratios 1, 3, 5 and 10% are plotted for the diesel injection pressures of 300, 600 and 1000 bars and the results for different EGR levels are as shown from ***Figure 47*** to ***Figure 50***.The brake thermal efficiency follows a similar trend as that of the BMEP. The efficiency increases as the diesel energy contribution increases and the increase in efficiency is more between 1-5% diesel energy ratio when compared to 5-10% diesel energy ratio. The efficiency is higher for higher injection pressures. The trend is observed to be same for all the levels of EGR (0-18%).

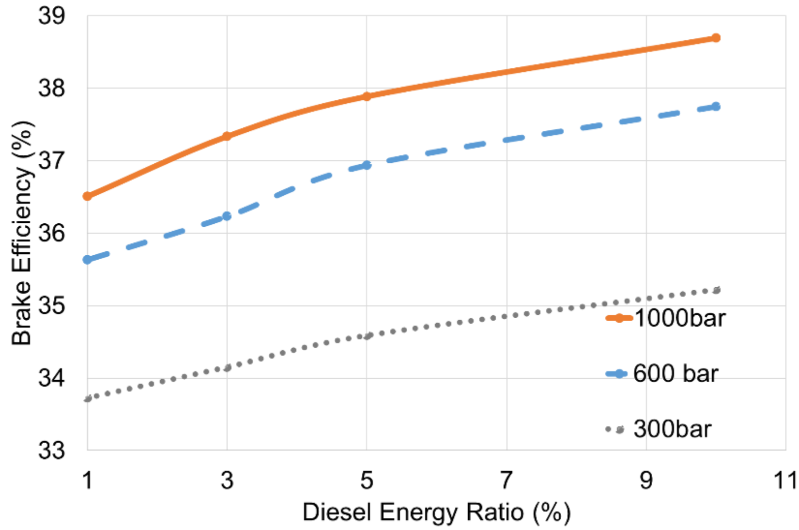

Figure 47 *Engine Speed=2138 RPM, SOI=0° BTDC: Effect of Diesel Energy Ratio (%) on Brake Thermal Efficiency (%) for* ***EGR=0%*** *for diesel injection pressures of 300,600 and 1000 bar*

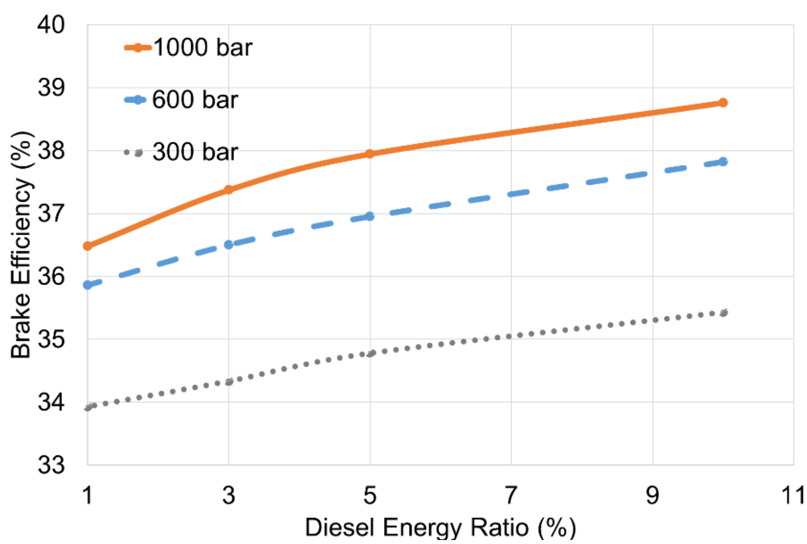

Figure 48 *Engine Speed=2138 RPM, SOI=0° BTDC: Effect of Diesel Energy Ratio (%) on Brake Thermal Efficiency (%) for* ***EGR=6%*** *for diesel injection pressures of 300,600 and 1000 bar*

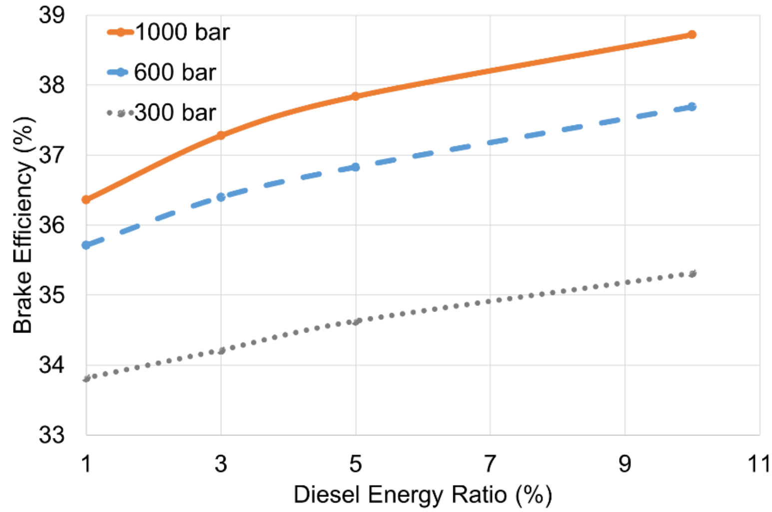

Figure 49 *Engine Speed=2138 RPM, SOI=0° BTDC: Effect of Diesel Energy Ratio (%) on Brake Thermal Efficiency (%) for* ***EGR=12%*** *for diesel injection pressures of 300,600 and 1000 bar*

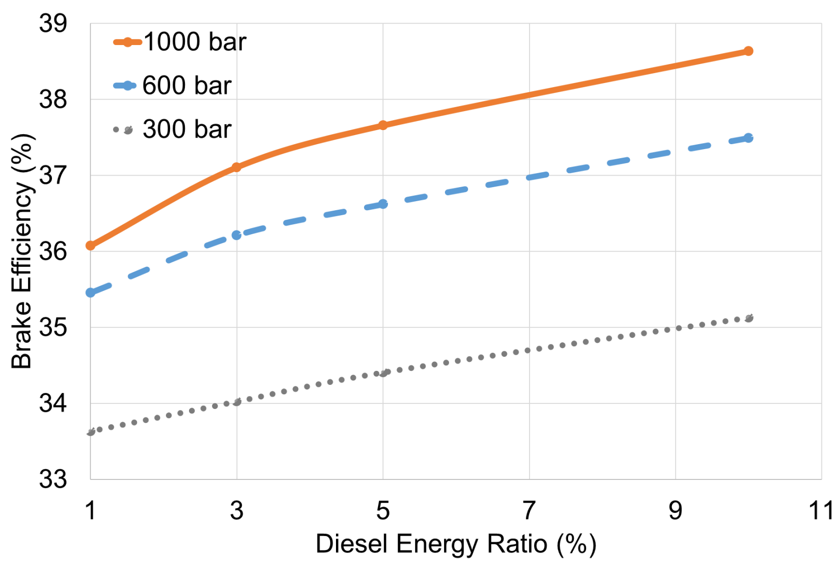

Figure 50 *Engine Speed=2138 RPM, SOI=0° BTDC: Effect of Diesel Energy Ratio (%) on Brake Thermal Efficiency (%) for* ***EGR=18%*** *for diesel injection pressures of 300,600 and 1000 bar*

7.2.1.4 Diesel Energy Contribution Vs Peak Cylinder Pressure

For diesel injection timing of 0° bTDC, the results for the peak cylinder pressures for cylinder-1 are as shown from ***Figure 51*** to ***Figure 54***.

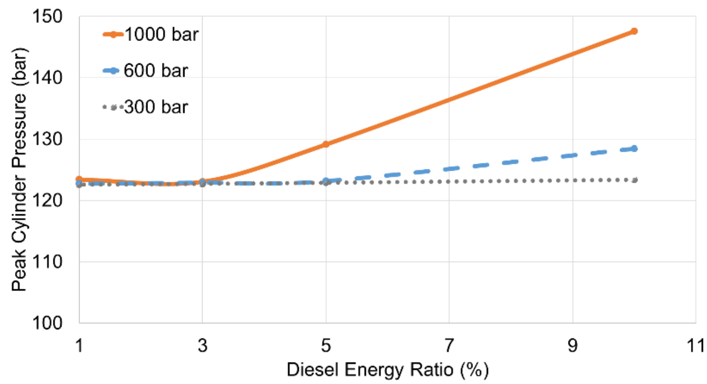

Figure 51 *Engine Speed=2138 RPM, SOI=0° BTDC: Effect of Diesel Energy Ratio (%) on Peak-Cylinder Pressure (bar) for* ***EGR=0%*** *for diesel injection pressures of 300,600 and 1000 bar*

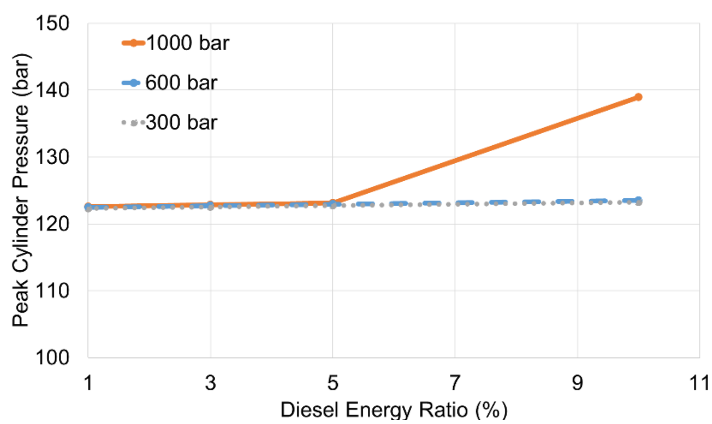

Figure 52 *Engine Speed=2138 RPM, SOI=0° BTDC: Effect of Diesel Energy Ratio (%) on Peak-Cylinder Pressure (bar) for* ***EGR=6%*** *for diesel injection pressures of 300,600 and 1000 bar*

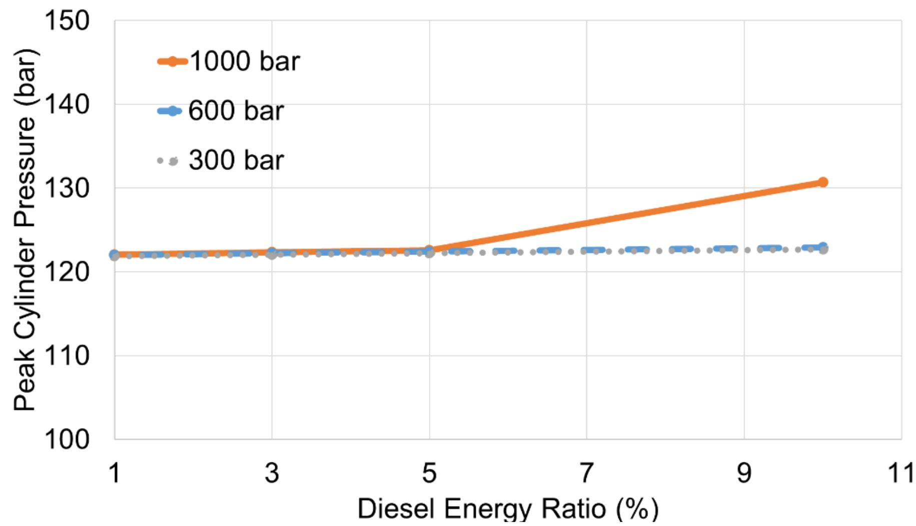

Figure 53 *Engine Speed=2138 RPM, SOI=0° BTDC: Effect of Diesel Energy Ratio (%) on Peak-Cylinder Pressure (bar) for* ***EGR=12%*** *for diesel injection pressures of 300,600 and 1000 bar*

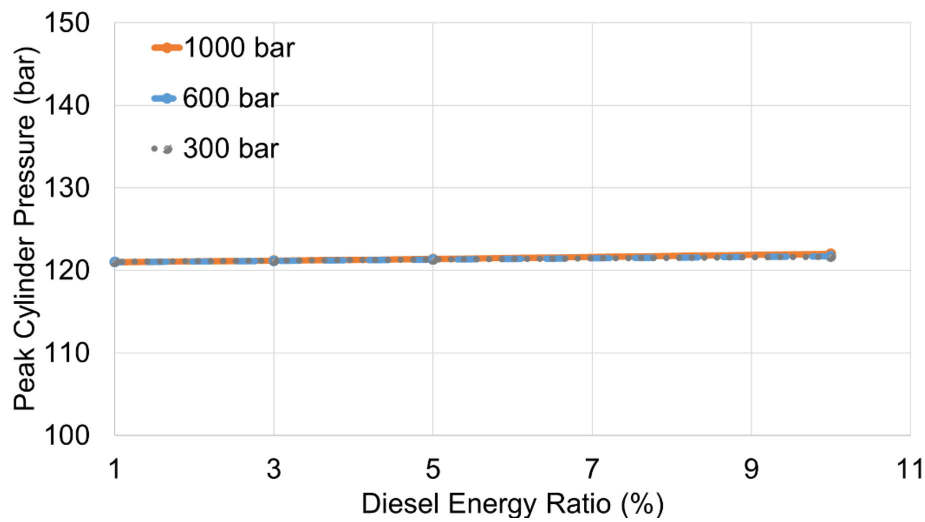

Figure 54 *Engine Speed=2138 RPM, SOI=0° BTDC: Effect of Diesel Energy Ratio (%) on Peak-Cylinder Pressure (bar) for* ***EGR=18%*** *for diesel injection pressures of 300,600 and 1000 bar*

For EGR = 0%, the rise in peak-cylinder pressure for 10% diesel energy ratio at injection pressures 1000 and 600 bars is explained by the pressure trace as shown in ***Figure 55***.

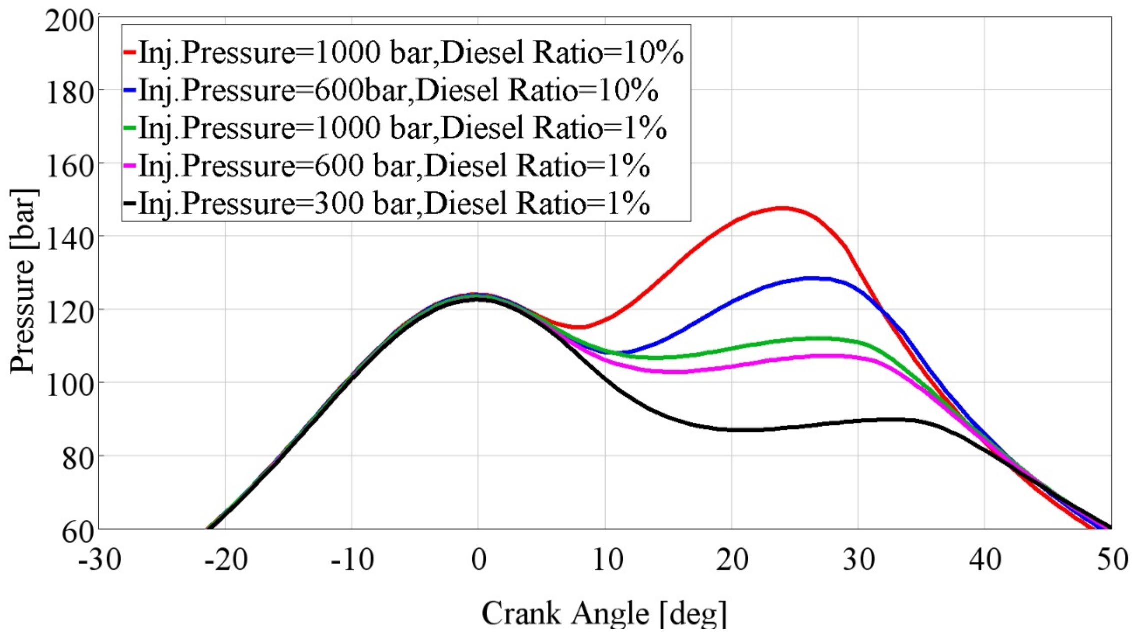

Figure 55 *Engine Speed=2138 RPM, SOI=0° BTDC: In-Cylinder Pressure trace for* ***EGR=0%*** *for cases of higher peak-cylinder pressures*

Rise in peak-cylinder pressures for some of the cases is because of the combustion peak-pressure and the constant peak-cylinder pressures observed in the remaining cases is the motoring cylinder pressure.

The burn duration for the operating points is as shown in ***Figure 56***. For higher injection pressures, the burn duration is quicker. Higher diesel energy ratio is found to have smaller burn duration.

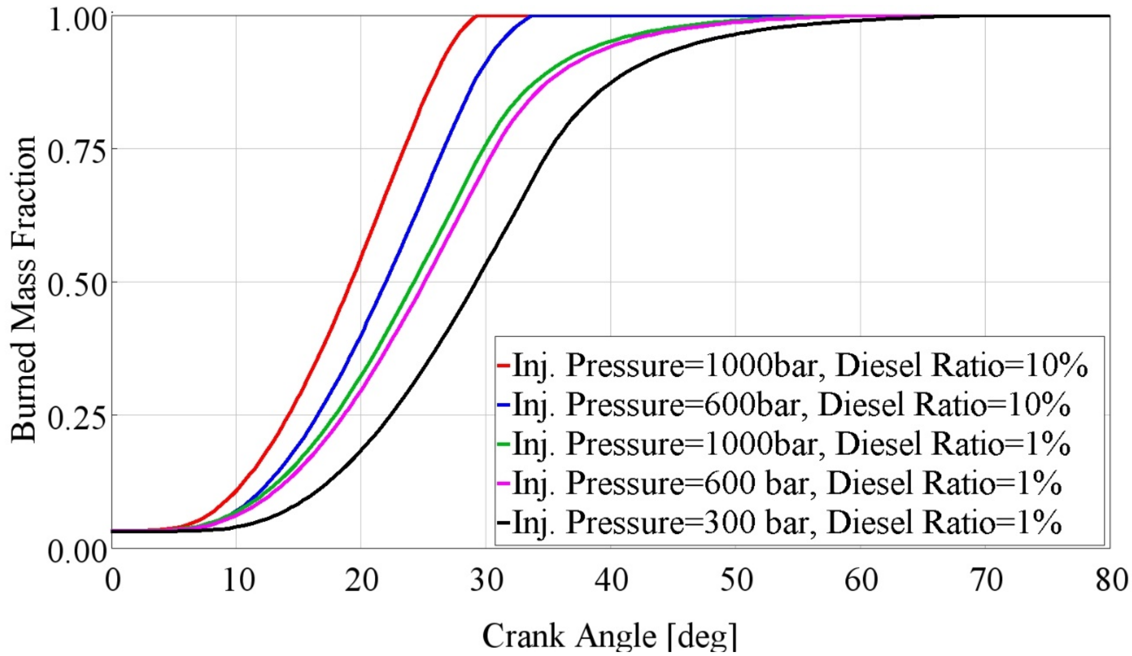

***Figure 56** Engine Speed=2138 RPM, SOI=0° BTDC: Burned Mass fraction for **EGR=0%** for cases of higher peak-cylinder pressures*

Higher peak-cylinder pressures can be attributed to higher injection pressures at higher diesel energy ratios.

The rise in peak cylinder pressures for the EGR levels 6 and 12% for injection pressure of 1000 bar can be attributed to the same reason as for EGR = 0%.

7.2.2 Effect of EGR on Engine Performance

The results for the high-speed (Engine Speed = 2138 RPM) engine operating condition at 0° bTDC are analyzed to study the effect of EGR on the engine performance.

7.2.2.1 EGR Vs BMEP

To study the effect to EGR on BMEP, the results for diesel energy ratios 0 and 10% for the diesel injection pressures of 300,600 and 1000 bar are compared and the results are as shown in ***Figure 57*** and ***Figure 58***. It is observed that as the EGR level increases, the BMEP decreases. BMEP values for diesel injection pressure of 300 bar are lower than that of the values for diesel injection pressure of 1000 bar. The trend remains same for diesel energy ratios of 1 and 10%.

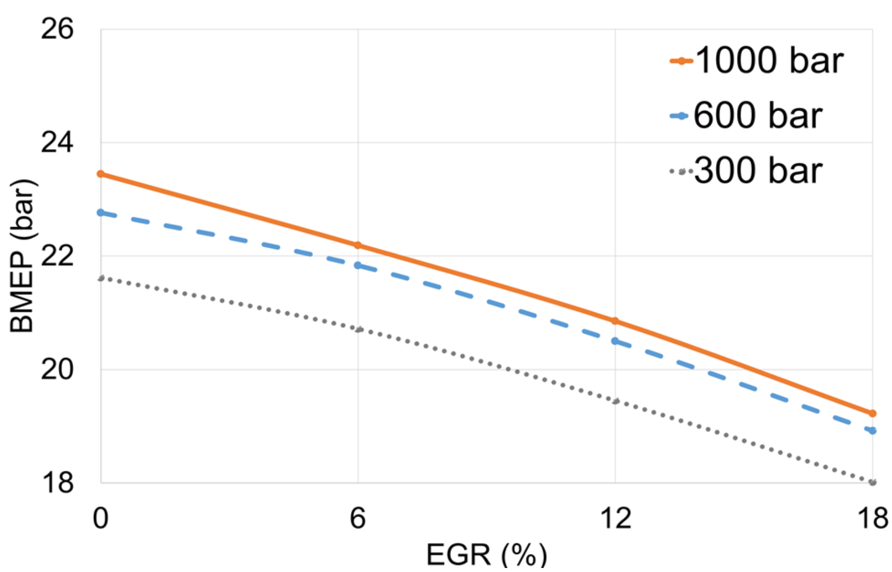

Figure 57 *Engine Speed = 2138 RPM, SOI=0° BTDC: Effect of EGR (%) on BMEP (bar) for* ***Diesel Energy Ratio = 1%*** *for diesel injection pressures of 300,600 and 1000 bar*

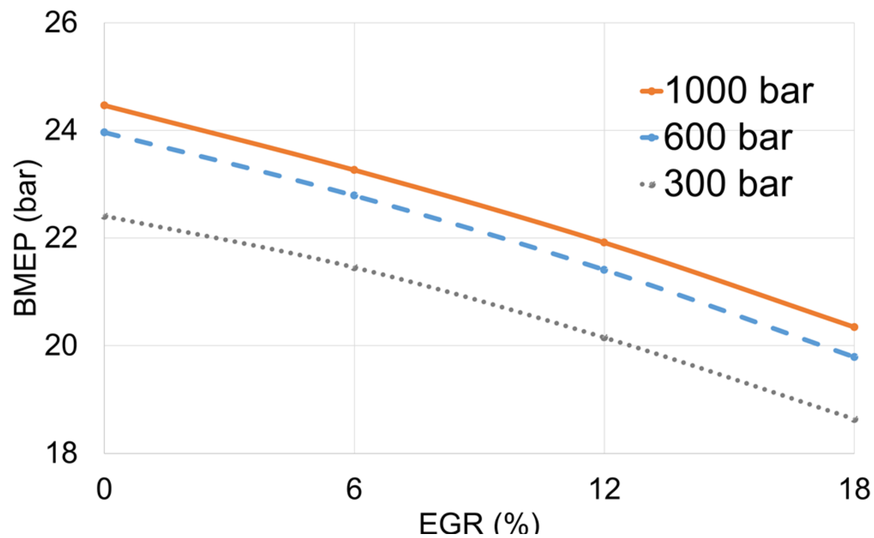

Figure 58 *Engine Speed = 2138 RPM, SOI=0° BTDC: Effect of EGR (%) on BMEP (bar) for* ***Diesel Energy Ratio = 10%*** *for diesel injection pressures of 300,600 and 1000 bar*

7.2.2.2 EGR Vs NO_x emissions

NO_x emissions for the variation of EGR levels for diesel energy ratios 1 and 10% are as shown in ***Figure 59*** and ***Figure 60***. As the EGR level is increases, the NO_x emissions are increased. As the diesel injection pressure is reduced, the NO_x emissions are low. The effect of diesel injection pressure reduces as the EGR level increases.

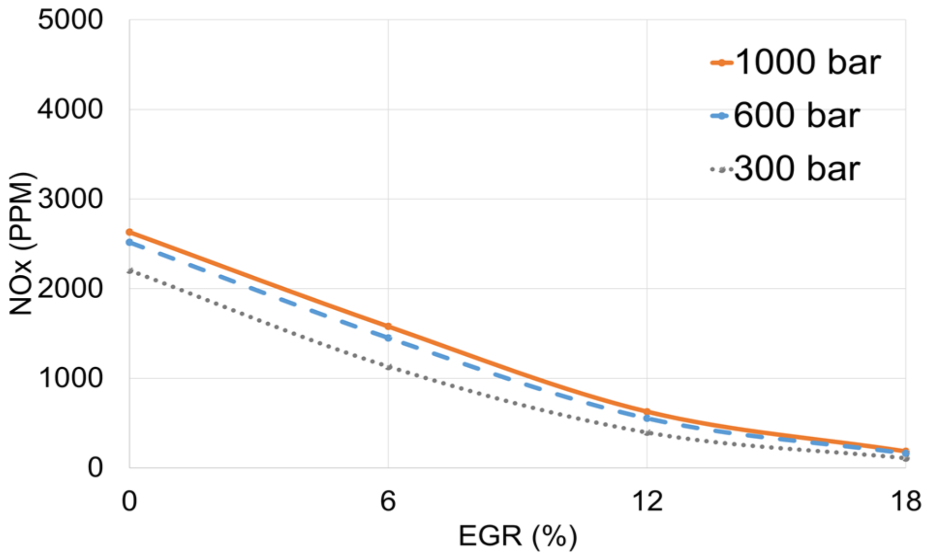

Figure 59 *Engine Speed = 2138 RPM, SOI=0° BTDC: Effect of EGR (%) on NO_x emissions (PPM) for* ***Diesel Energy Ratio = 1****% for diesel injection pressures of 300,600 and 1000 bar*

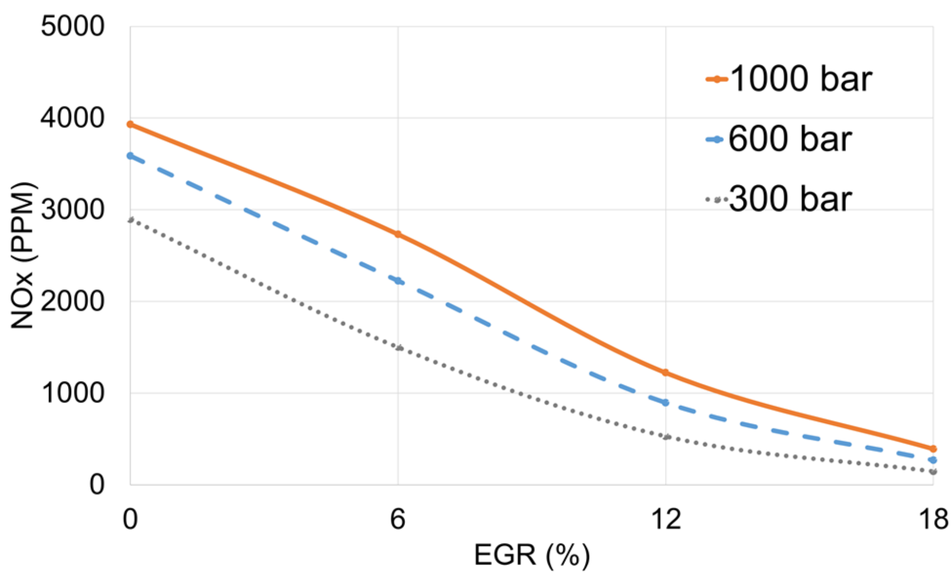

Figure 60 *Engine Speed = 2138 RPM, SOI=0° BTDC: Effect of EGR (%) on NO_x emissions (PPM) for* ***Diesel Energy Ratio = 10****% for diesel injection pressures of 300,600 and 1000 bar*

7.3 Simulation results for Injection Timing 0° bTDC, Engine Speed = 1397 RPM

Simulation results obtained for the dual-fuel engine model at 0° bTDC for low-speed case (Engine Speed = 1397 RPM) are discussed in this section.

7.3.1 Effect of Diesel Energy Contribution on Engine Performance

For EGR percentages of 0,6,12 and 18%, the diesel energy ratio is varied for 1, 3, 5 & 10% of the total fuel energy and tested for diesel injection pressures of 300, 600 and 1000 bar.

7.3.1.1 Diesel Energy Contribution Vs BMEP

The BMEP results for the variation of diesel energy ratio for each fraction of EGR are as summarized from ***Figure 61*** to ***Figure 64***. It is observed that as the diesel energy ratio increases, the BMEP of the engine increases. Higher injection pressures have higher BMEP values across the different energy ratios. The trend is similar for the EGR levels from 0 to 18%. The BMEP increase between the diesel energy ratios 1-5% is found to be more than that of the increase between 5-10% diesel energy ratio. The results for the BMEP for variation of diesel energy ratio are similar to the results for high-speed operating condition (2138 RPM).

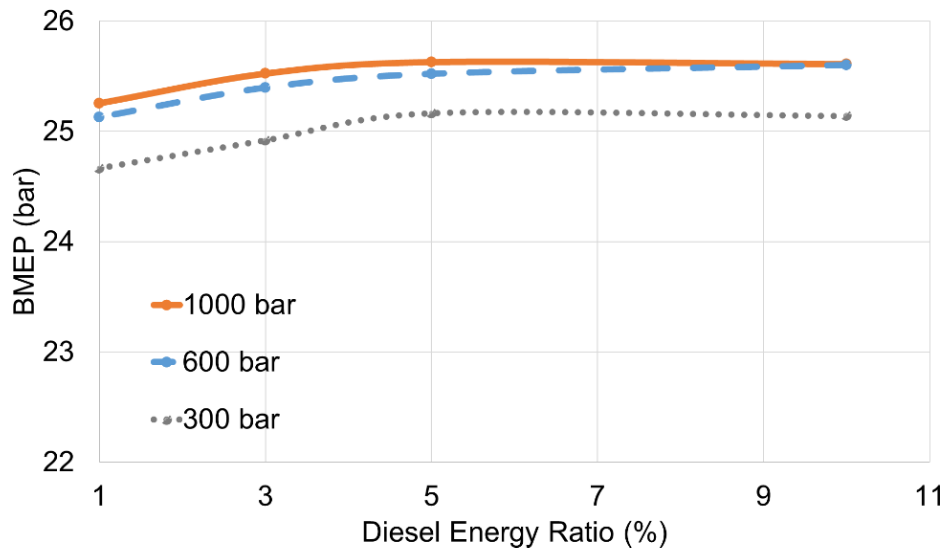

Figure 61 *Engine Speed = 1397 RPM, SOI=0° BTDC: Effect of Diesel Energy Ratio (%) on BMEP (bar) for* ***EGR=0*** *% for diesel injection pressures of 300,600 and 1000 bar*

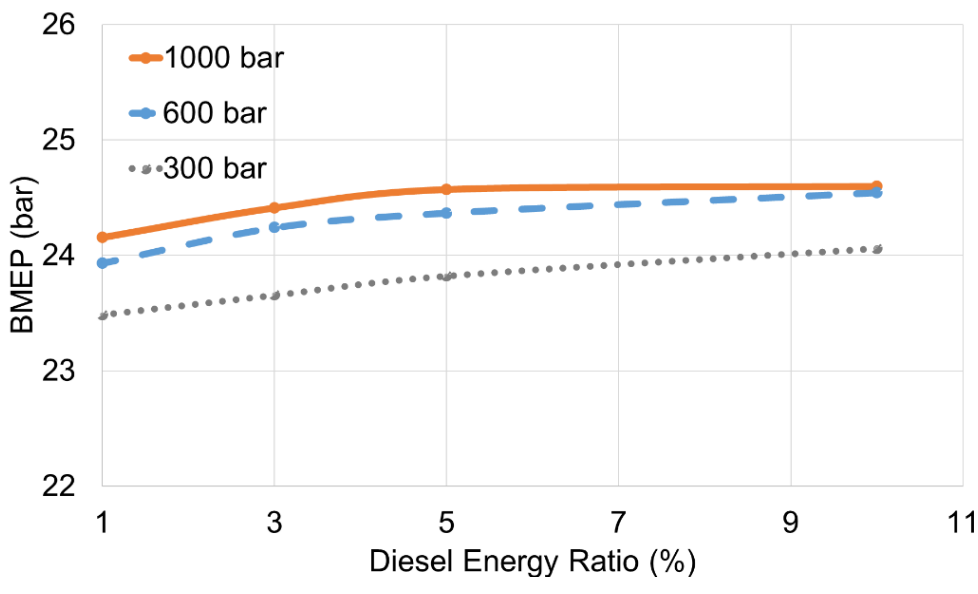

Figure 62 *Engine Speed = 1397 RPM, SOI=0° BTDC: Effect of Diesel Energy Ratio (%) on BMEP (bar) for* ***EGR=6*** *% for diesel injection pressures of 300,600 and 1000 bar*

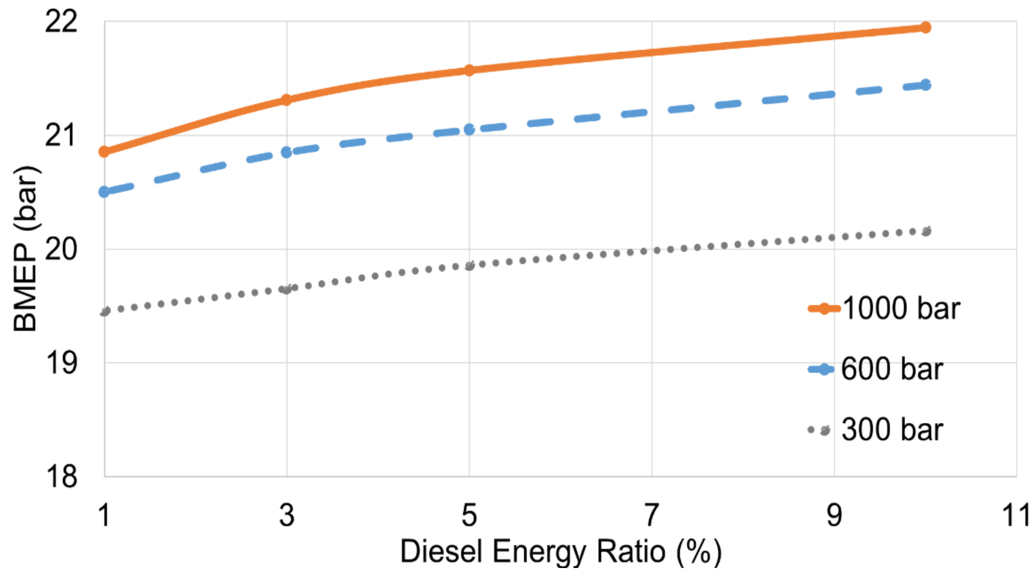

Figure 63 *Engine Speed = 1397 RPM, SOI=0° BTDC: Effect of Diesel Energy Ratio (%) on BMEP (bar) for* ***EGR=12*** *% for diesel injection pressures of 300,600 and 1000 bar*

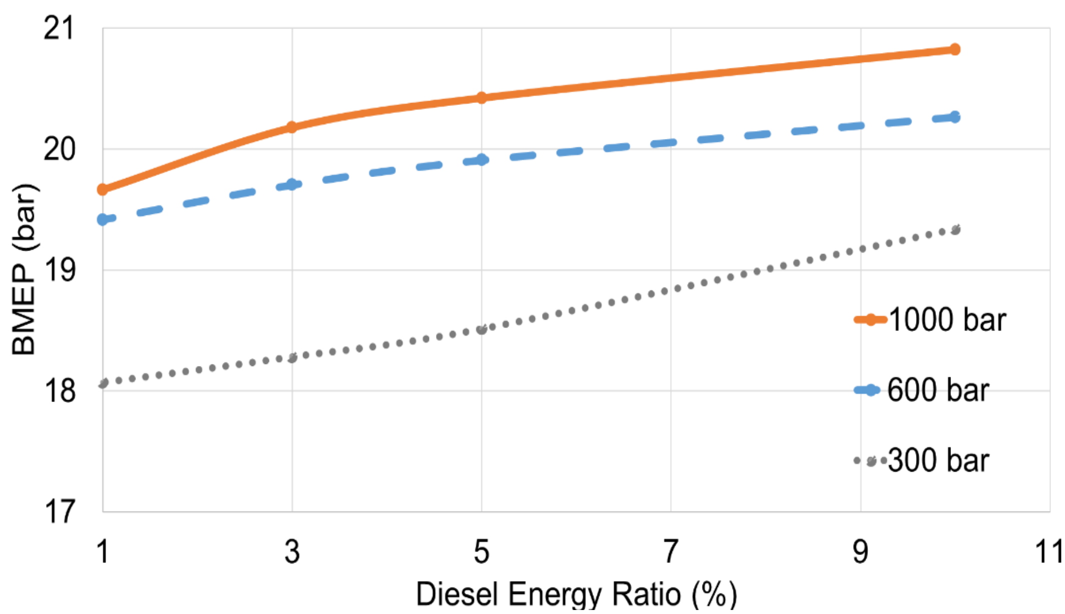

Figure 64 *Engine Speed = 1397 RPM, SOI=0° BTDC: Effect of Diesel Energy Ratio (%) on BMEP (bar) for* ***EGR=18*** *% for diesel injection pressures of 300,600 and 1000 bar*

7.3.1.2 Diesel Energy Contribution Vs NO_x emissions

For the variation of diesel energy contribution for 1,3,5 and 10%, for the diesel injection pressures of 300, 600 and 1000 bar, the NO_x emissions (in PPM) obtained from the simulation for each case of EGR are as shown from ***Figure 65*** to ***Figure 68***. It is observed that as the diesel energy ratio increases, the NO_x emissions increases. Lower diesel injection pressures are observed to have lower NO_x emissions. The trend remains same across all the EGR levels (EGR = 0% to EGR = 18%). For diesel injection pressure of 300 bar for EGR = 18%, the increase in NO_x emissions is not significant compared to other levels of EGR.

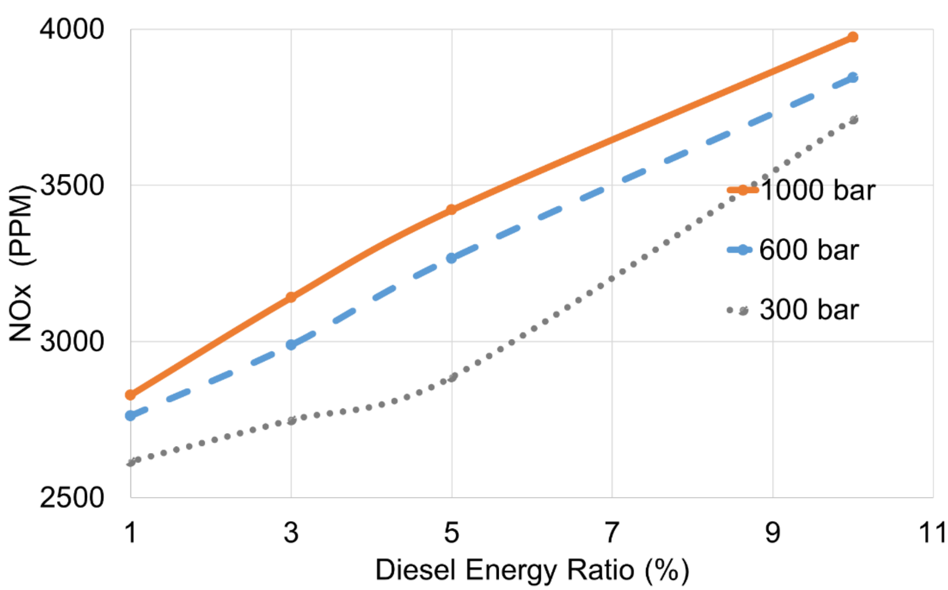

Figure 65 *Engine Speed=1397 RPM, SOI=0° BTDC: Effect of Diesel Energy Ratio (%) on NO_x emissions (PPM) for* ***EGR=0%*** *for diesel injection pressures of 300,600 and 1000 bar*

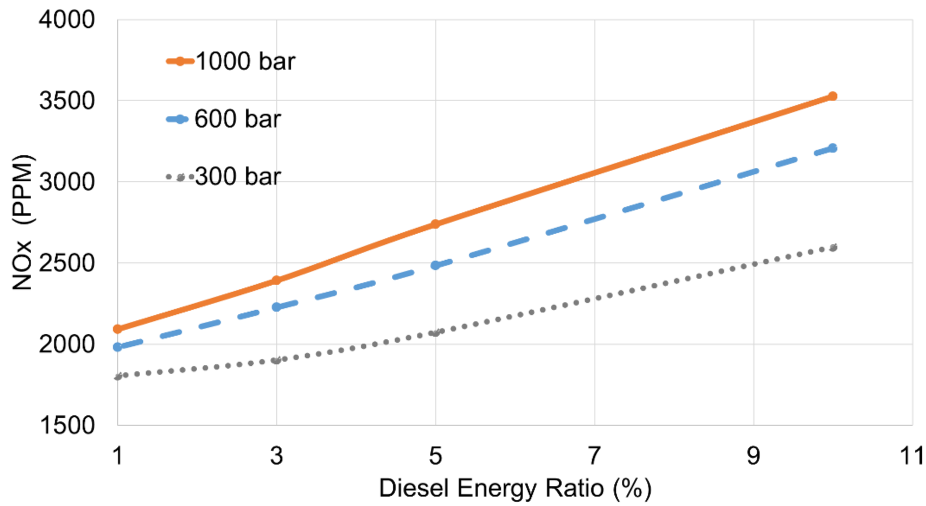

Figure 66 *Engine Speed=1397 RPM, SOI=0° BTDC: Effect of Diesel Energy Ratio (%) on NO_x emissions (PPM) for* ***EGR=6%*** *for diesel injection pressures of 300,600 and 1000 bar*

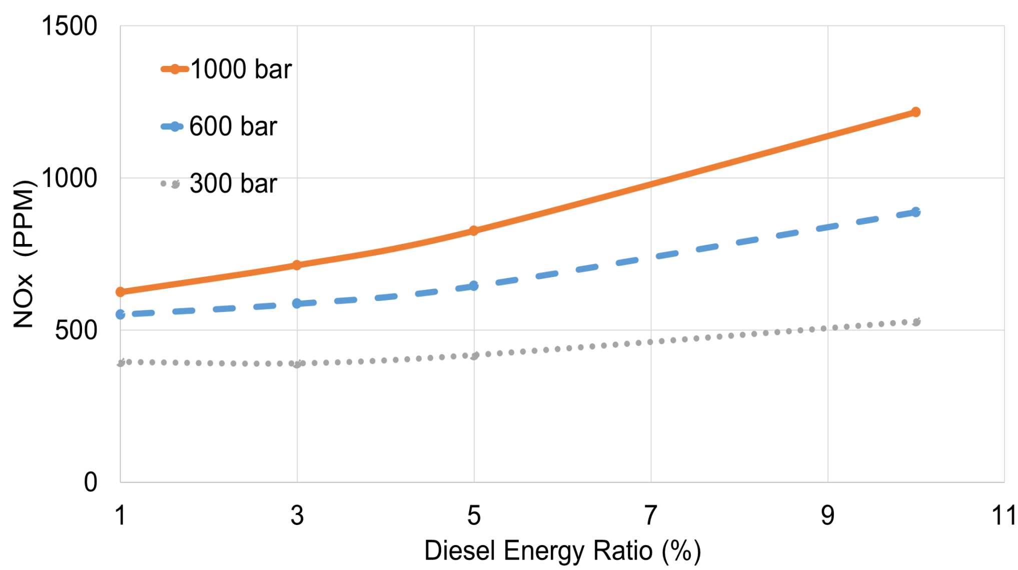

Figure 67 *Engine Speed=1397 RPM, SOI=0° BTDC: Effect of Diesel Energy Ratio (%) on NO_x emissions (PPM) for* ***EGR=12%*** *for diesel injection pressures of 300,600 and 1000 bar*

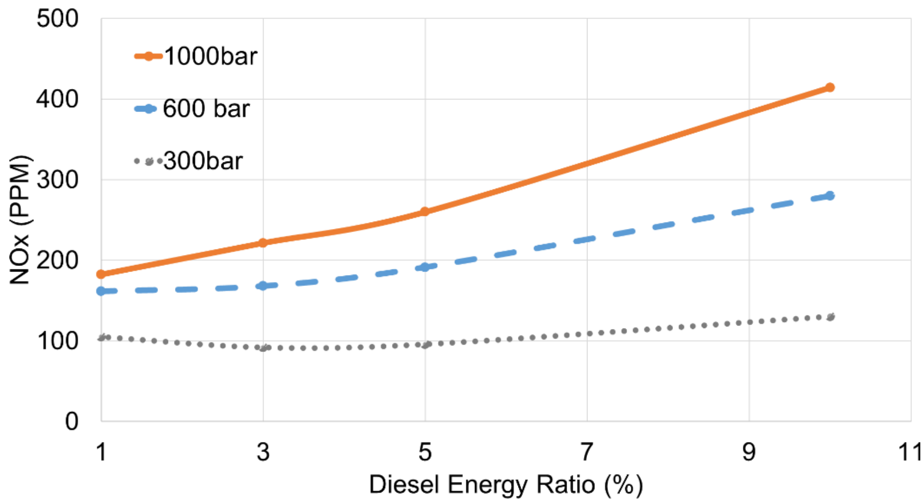

Figure 68 *Engine Speed=1397 RPM, SOI=0° BTDC: Effect of Diesel Energy Ratio (%) on NO_x emissions (PPM) for* ***EGR=18%*** *for diesel injection pressures of 300,600 and 1000 bar*

7.3.1.3 Diesel Energy Contribution Vs Brake Thermal Efficiency

The brake thermal efficiency for the diesel energy ratios 1, 3, 5 and 10% are plotted for the diesel injection pressures of 300, 600 and 1000 bars and the results for different EGR levels are as shown from ***Figure 69*** to ***Figure 72***. The brake thermal efficiency follows a similar trend as that of the BMEP. The efficiency increases as the diesel energy contribution increases and the increase in efficiency is more between 1-5% diesel energy ratios when compared to 5-10% diesel energy ratios. The efficiency is higher for higher injection pressures. The trend is observed to be same for all the levels of EGR (0-18%).

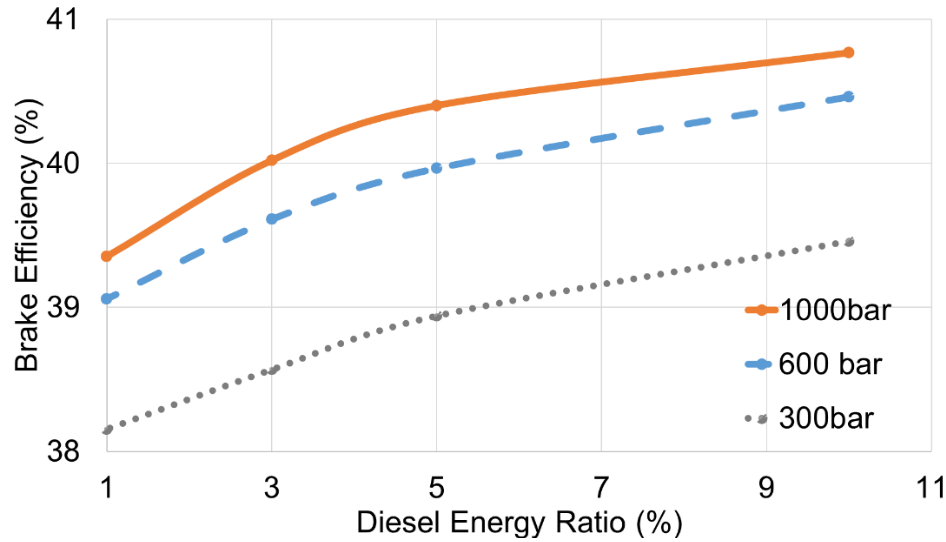

Figure 69 *Engine Speed=1397 RPM, SOI=0° BTDC: Effect of Diesel Energy Ratio (%) on Brake Thermal Efficiency (%) for* ***EGR=0%*** *for diesel injection pressures of 300,600 and 1000 bar*

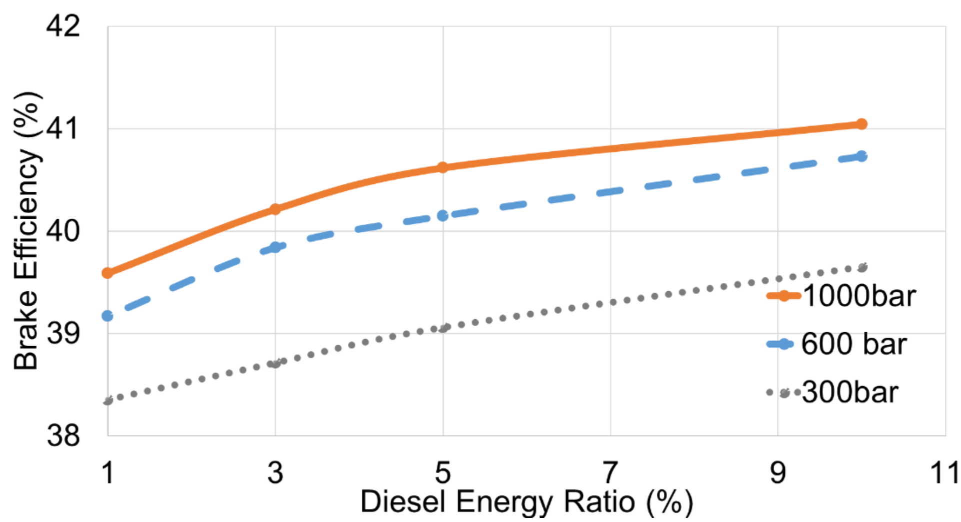

Figure 70 *Engine Speed=1397 RPM, SOI=0° BTDC: Effect of Diesel Energy Ratio (%) on Brake Thermal Efficiency (%) for* ***EGR=6%*** *for diesel injection pressures of 300,600 and 1000 bar*

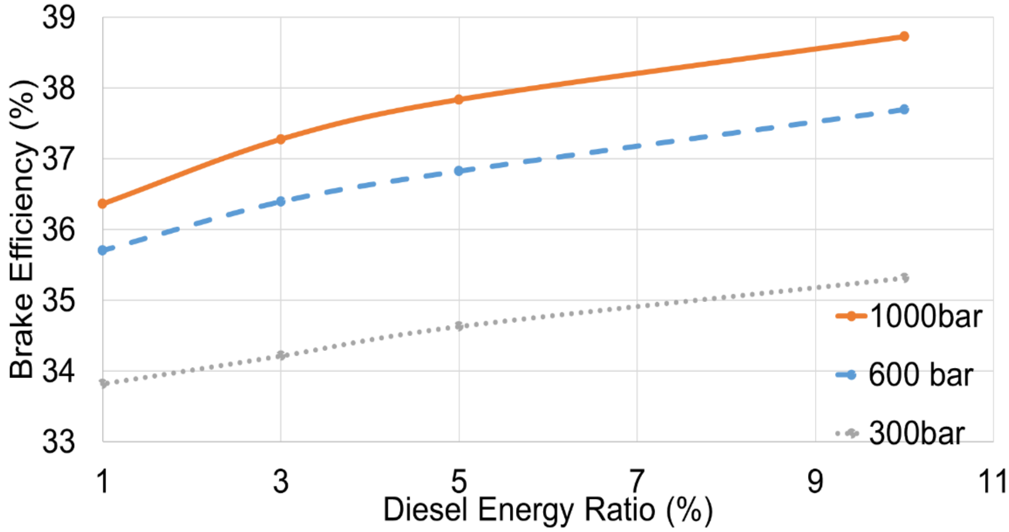

Figure 71 *Engine Speed=1397 RPM, SOI=0° BTDC: Effect of Diesel Energy Ratio (%) on Brake Thermal Efficiency (%) for* ***EGR=12%*** *for diesel injection pressures of 300,600 and 1000 bar*

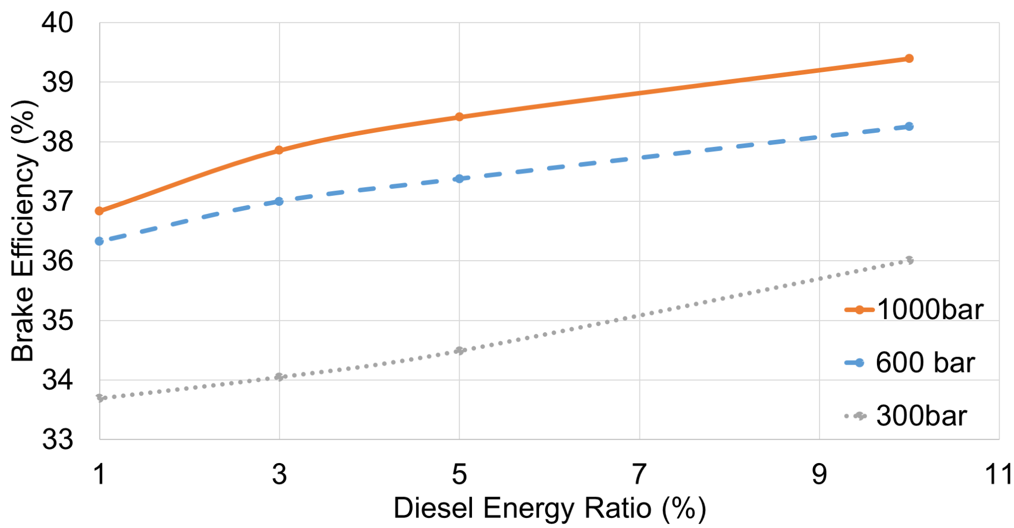

Figure 72 *Engine Speed=1397 RPM, SOI=0° BTDC: Effect of Diesel Energy Ratio (%) on Brake Thermal Efficiency (%) for* ***EGR=18%*** *for diesel injection pressures of 300,600 and 1000 bar*

7.3.1.4 Diesel Energy Contribution Vs Peak Cylinder Pressure

For diesel injection timing of 0° bTDC, the results for the peak cylinder pressures for cylinder-1 are as shown from ***Figure 73*** to ***Figure 76***.

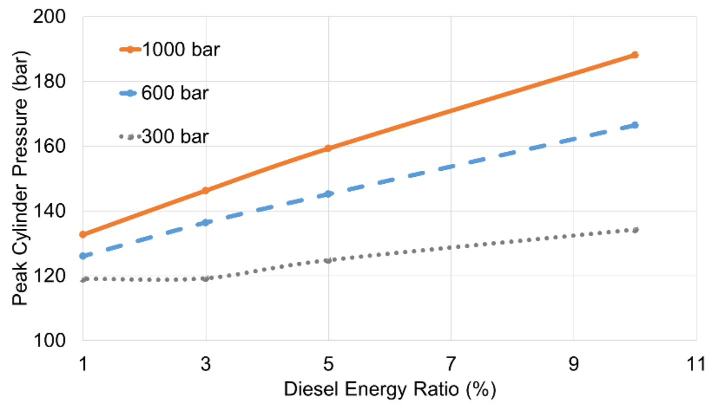

Figure 73 *Engine Speed=1397 RPM, SOI=0° BTDC: Effect of Diesel Energy Ratio (%) on Peak-Cylinder Pressure (bar) for* ***EGR=0%*** *for diesel injection pressures of 300,600 and 1000 bar*

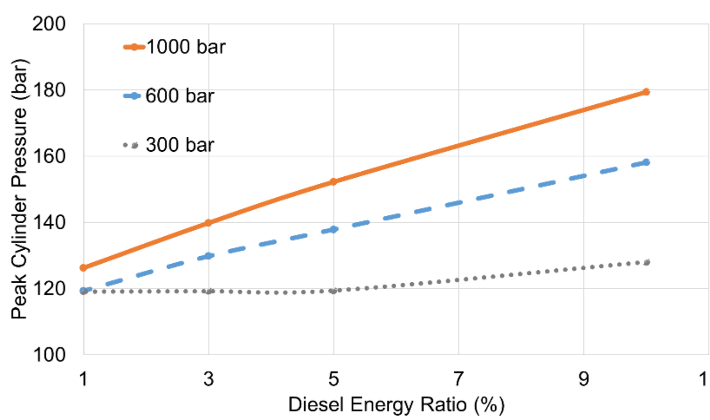

Figure 74 *Engine Speed=1397 RPM, SOI=0° BTDC: Effect of Diesel Energy Ratio (%) on Peak-Cylinder Pressure (bar) for* ***EGR=6%*** *for diesel injection pressures of 300,600 and 1000 bar*

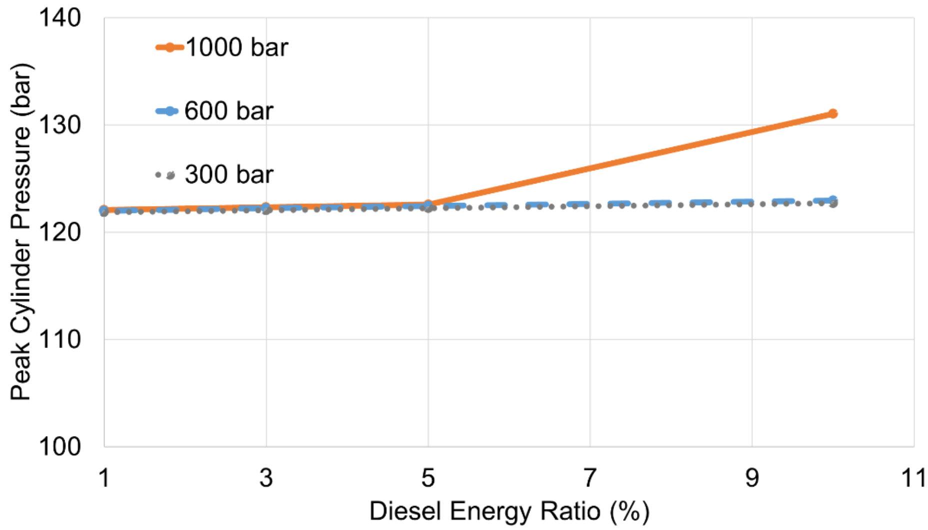

Figure 75 *Engine Speed=1397 RPM, SOI=0° BTDC: Effect of Diesel Energy Ratio (%) on Peak-Cylinder Pressure (bar) for* ***EGR=12%*** *for diesel injection pressures of 300,600 and 1000 bar*

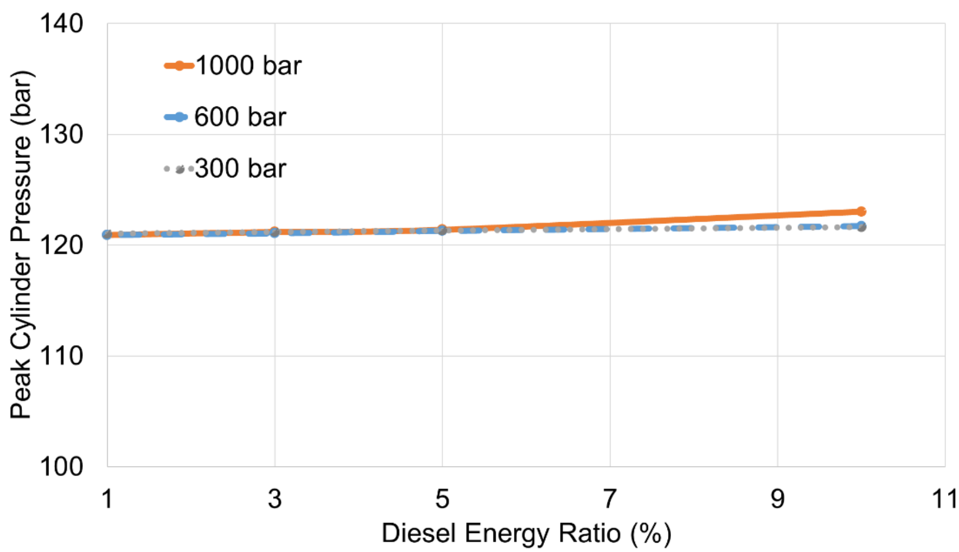

Figure 76 *Engine Speed=1397 RPM, SOI=0° BTDC: Effect of Diesel Energy Ratio (%) on Peak-Cylinder Pressure (bar) for* ***EGR=18%*** *for diesel injection pressures of 300,600 and 1000 bar*

It is observed that as the diesel energy ratio increases, the peak cylinder pressure increases for injection pressures of 600 and 1000 bar for EGR =6%. The constant line in the results indicates that the peak-cylinder pressure is due to the motoring peak cylinder pressure as explained in section **7.2.1.4.**

7.3.2 Effect of EGR on Engine Performance

The results for the low-speed (Engine Speed = 1397 RPM) engine operating condition at 0° bTDC are analyzed to study the effect of EGR on the engine performance.

7.3.2.1 EGR Vs BMEP

The trend for BMEP for variation of EGR at diesel energy contribution of 1 and 10% is as shown in ***Figure*** *77* and ***Figure*** ***78***. It is observed that as the EGR level increases, the BMEP of the engine decreases. The reduction in BMEP is more between the EGR levels of 6-12% when compared with reduction between 0-6% and 12-18%. The higher diesel injection pressures result in higher BMEP and the effect of lower diesel injection pressure leads to more reduction in BMEP at higher EGR levels. The trend for the BMEP is same for different diesel energy contribution percentages.

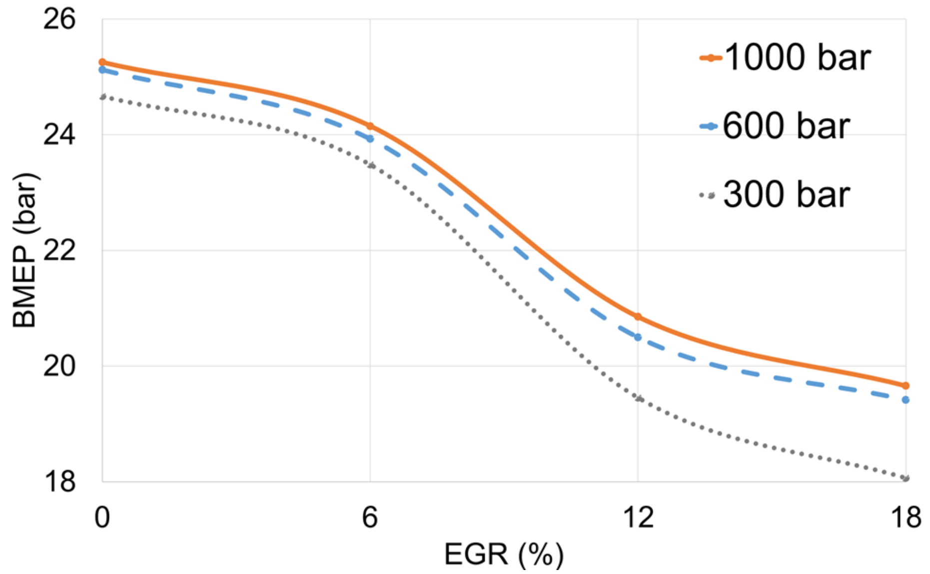

***Figure** 77 Engine Speed = 1397 RPM, SOI=0° BTDC: Effect of EGR (%) on BMEP (bar) for **Diesel Energy Ratio = 1**% for diesel injection pressures of 300,600 and 1000 bar*

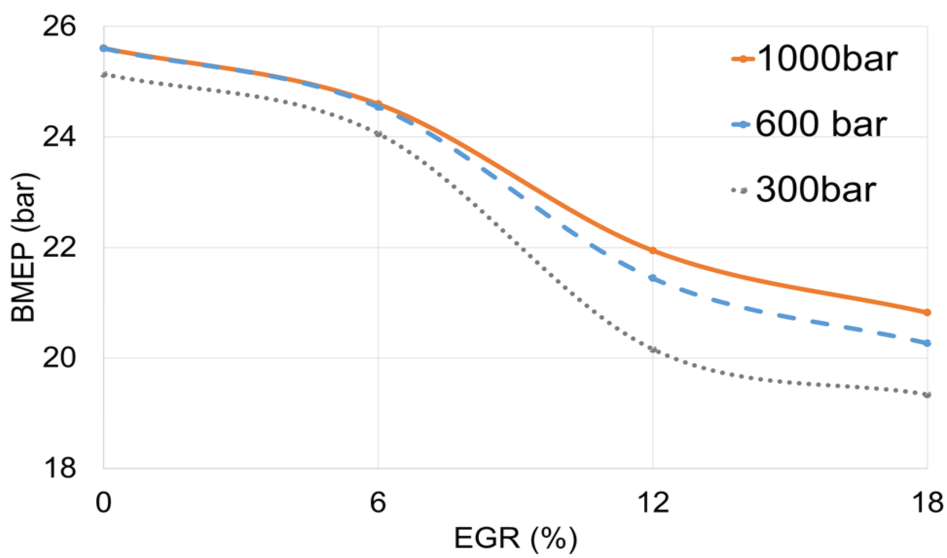

***Figure 78** Engine Speed = 1397 RPM, SOI=0° BTDC: Effect of EGR (%) on BMEP (bar) for **Diesel Energy Ratio = 10**% for diesel injection pressures of 300,600 and 1000 bar*

7.3.2.2 EGR Vs NO_x emissions

NO_x emissions for the variation of EGR levels for diesel energy ratios 1 and 10% are as shown in ***Figure 79*** and ***Figure 80***. As the EGR level is increases, the NO_x emissions are increased. As the diesel injection pressure is reduced, the NO_x emissions are low.

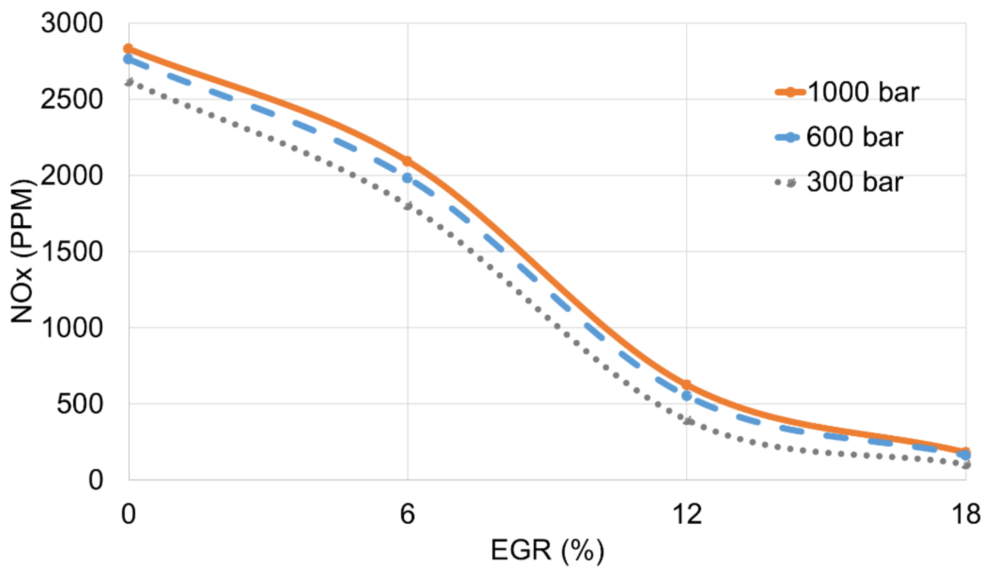

Figure 79 *Engine Speed = 1397 RPM, SOI=0° BTDC: Effect of EGR (%) on NO_x emissions (PPM) for* ***Diesel Energy Ratio = 1%*** *for diesel injection pressures of 300,600 and 1000 bar*

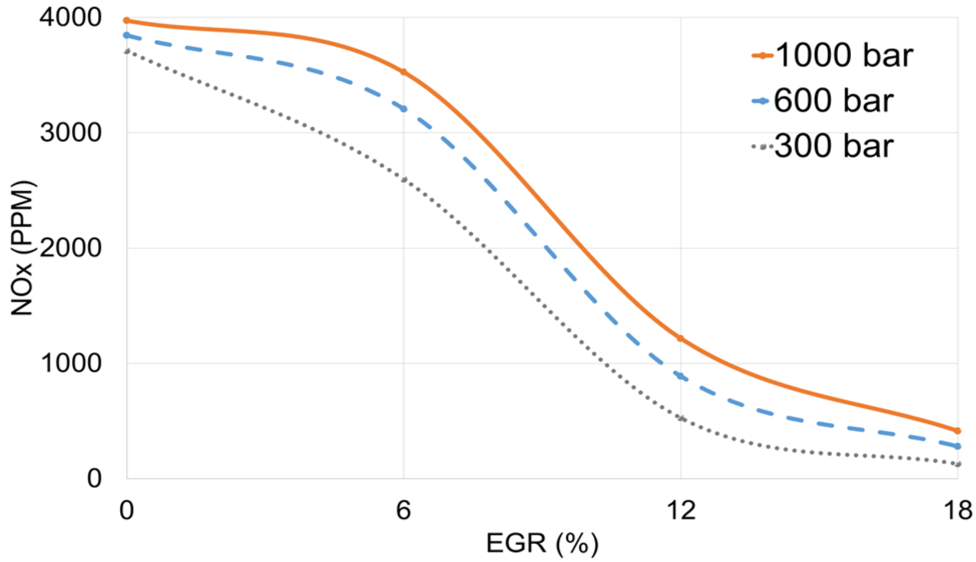

Figure 80 *Engine Speed = 1397 RPM, SOI=0° BTDC: Effect of EGR (%) on NO_x emissions (PPM) for* ***Diesel Energy Ratio = 10%*** *for diesel injection pressures of 300,600 and 1000 bar*

7.4 Simulation results for Injection Timing 10° bTDC, Engine Speed = 2138 RPM

For diesel injection timing 10° bTDC, the simulation model for dual-fuel engine is simulated for two different engine speeds: 2138 RPM (High-Speed) and 1397 RPM (Low-Speed)

Simulation results obtained for the dual-fuel engine model at 0° bTDC for high-speed case (Engine Speed = 2138 RPM) are discussed in this sub-section.

7.4.1 Effect of Diesel Energy Contribution on Engine Performance

For EGR percentages of 0,6,12 and 18%, the diesel energy ratio is varied for 1, 3, 5 & 10% of the total fuel energy and tested for diesel injection pressures of 300, 600 and 1000 bar.

7.4.1.1 Diesel Energy Contribution Vs BMEP

The BMEP results for the variation of diesel energy ratio for each fraction of EGR are as summarized from ***Figure 81*** to ***Figure 84***. For each case of EGR percentage, the BMEP values are observed to be nearly close. It is observed that as the diesel energy contribution increases, the values of BMEP remain similar for the cases where injection pressures are 300 bar and 600 bar. For the case of higher diesel injection pressure=1000 bar, as the diesel energy contribution increases, the BMEP is found to reduce.

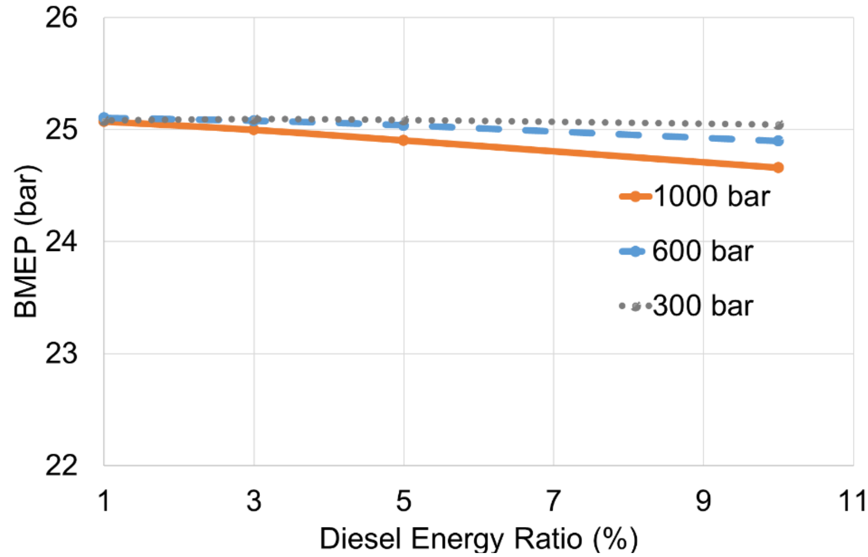

Figure 81 *Engine Speed = 2138 RPM, SOI=10° BTDC: Effect of Diesel Energy Ratio (%) on BMEP (bar) for* ***EGR=0*** *% for diesel injection pressures of 300,600 and 1000 bar*

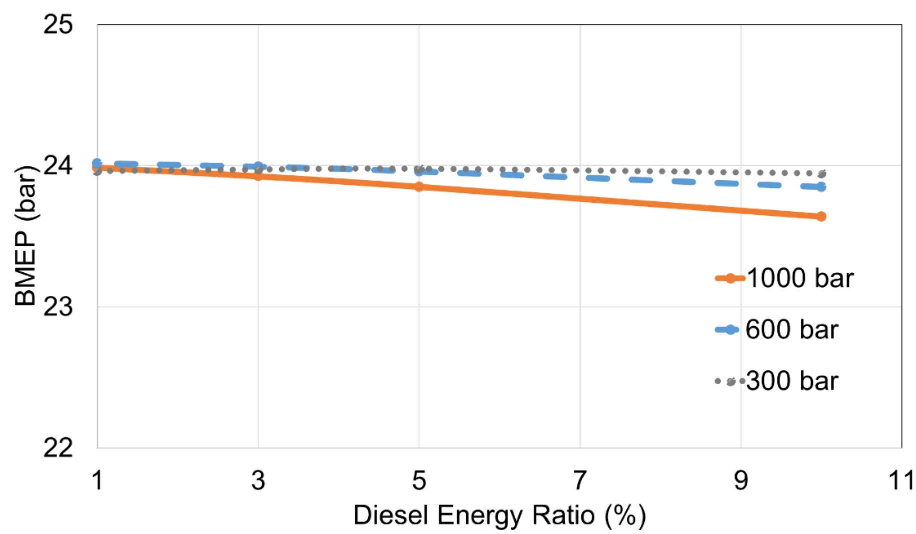

Figure 82 *Engine Speed = 2138 RPM, SOI=10° BTDC: Effect of Diesel Energy Ratio (%) on BMEP (bar) for* ***EGR=6*** *% for diesel injection pressures of 300,600 and 1000 bar*

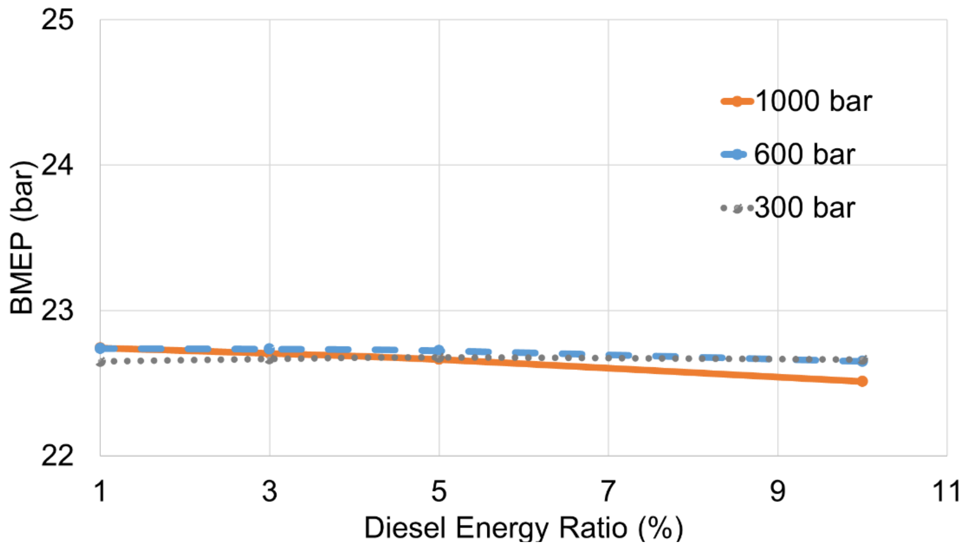

Figure 83 *Engine Speed = 2138 RPM, SOI=10° BTDC: Effect of Diesel Energy Ratio (%) on BMEP (bar) for* ***EGR=12*** *% for diesel injection pressures of 300,600 and 1000 bar*

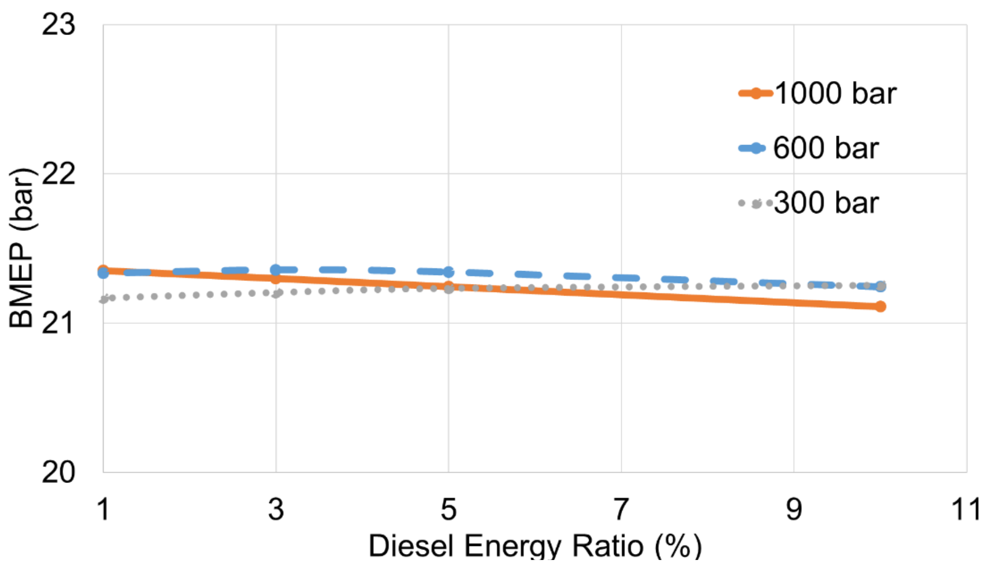

Figure 84 *Engine Speed = 2138 RPM, SOI=10° BTDC: Effect of Diesel Energy Ratio (%) on BMEP (bar) for* ***EGR=18*** *% for diesel injection pressures of 300,600 and 1000 bar*

7.4.1.2 Diesel Energy Contribution Vs NO_x emissions

For the variation of diesel energy contribution for 1,3,5 and 10%, for the diesel injection pressures of 300, 600 and 1000 bar, the NO_x emissions (in PPM) obtained from the simulation for each case of EGR are as shown from ***Figure 85*** to ***Figure 88***. The results indicate that as the diesel energy contribution percentage increases, NO_x emissions increase for all diesel injection pressure cases. As the diesel injection pressure increases, NO_x emissions increase. Similar trend of increase is observed for all the values of EGR.

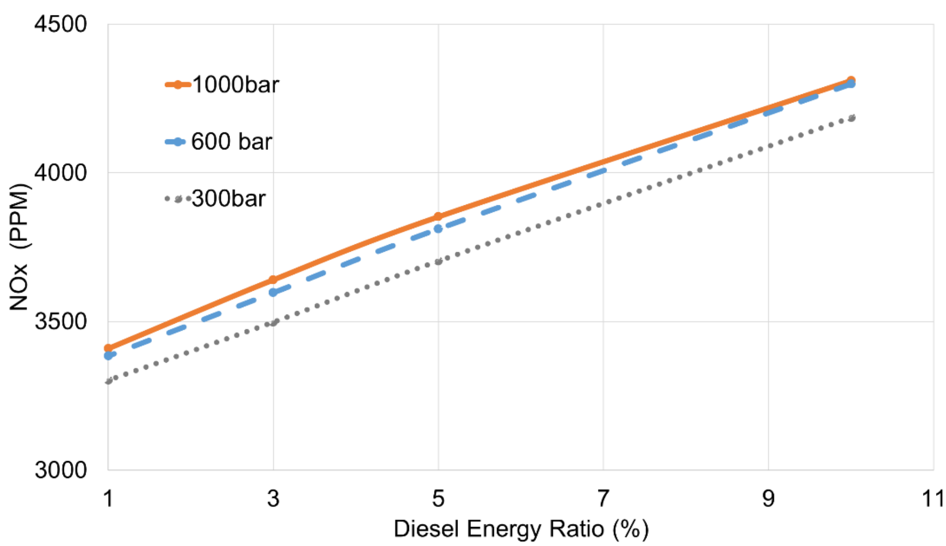

Figure 85 *Engine Speed = 2138 RPM, SOI=10° BTDC: Effect of Diesel Energy Ratio (%) on NO_x (PPM) for **EGR=0** % for diesel injection pressures of 300,600 and 1000 bar*

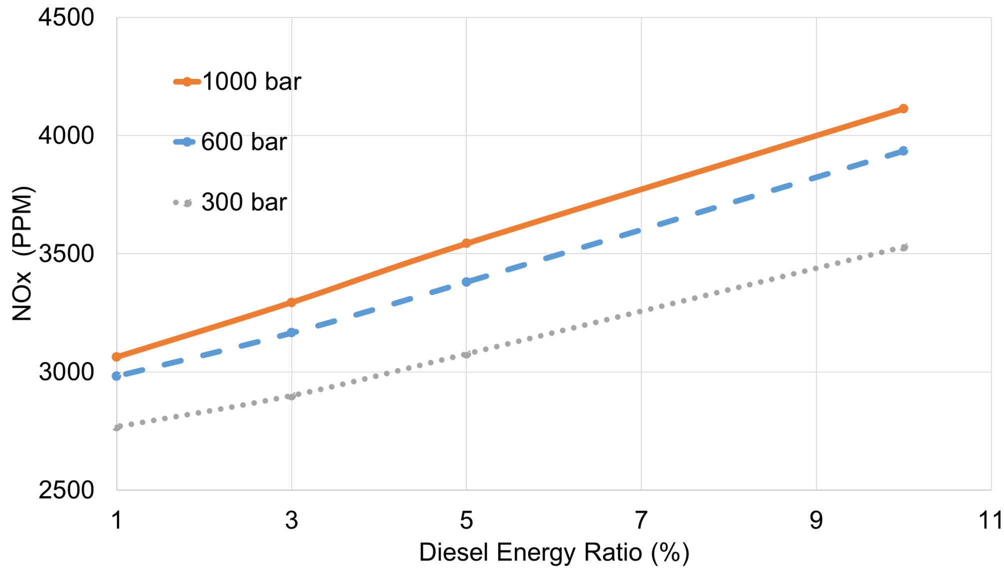

Figure 86 *Engine Speed = 2138 RPM, SOI=10° BTDC: Effect of Diesel Energy Ratio (%) on NO_x (PPM) for **EGR=6** % for diesel injection pressures of 300,600 and 1000 bar*

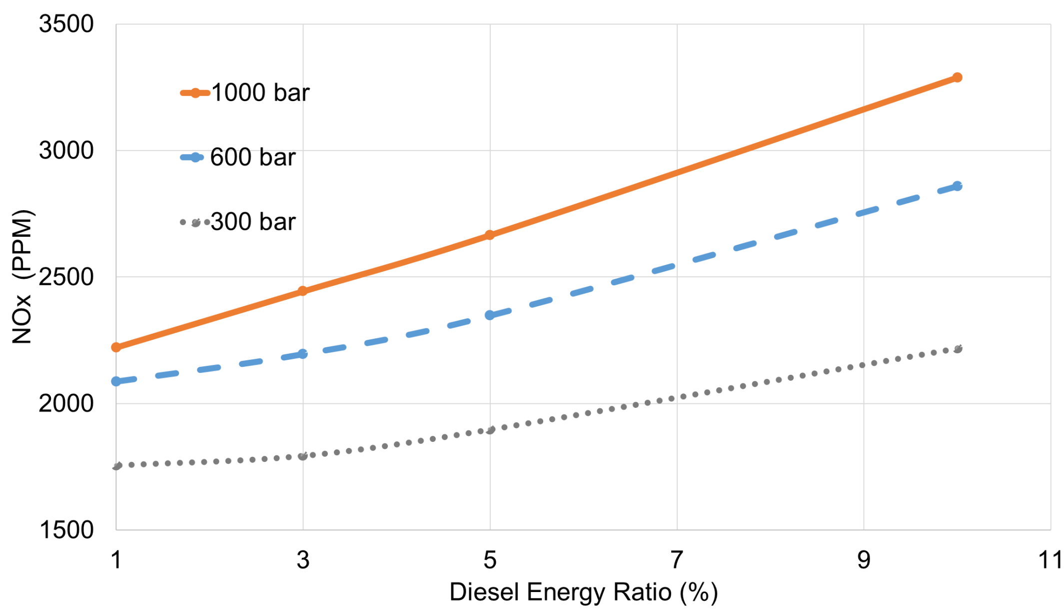

Figure 87 *Engine Speed = 2138 RPM, SOI=10° BTDC: Effect of Diesel Energy Ratio (%) on NO_x (PPM) for **EGR=12** % for diesel injection pressures of 300,600 and 1000 bar*

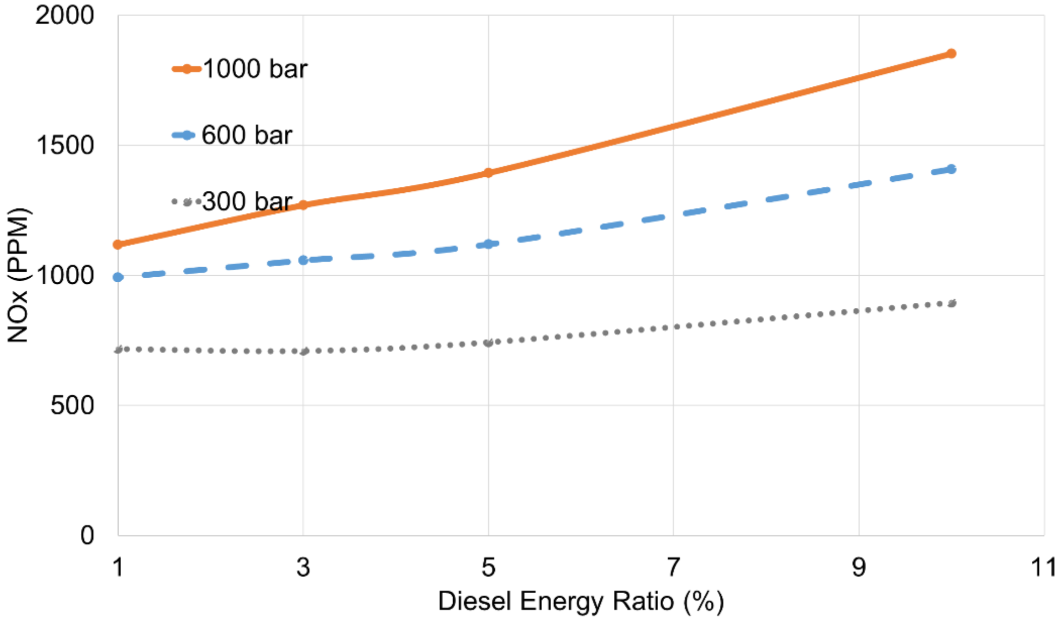

Figure 88 *Engine Speed = 2138 RPM, SOI=10° BTDC: Effect of Diesel Energy Ratio (%) on NO_x (PPM) for **EGR=18** % for diesel injection pressures of 300,600 and 1000 bar*

7.4.1.3 Diesel Energy Contribution Vs Brake Thermal Efficiency

The brake thermal efficiency for the diesel energy ratios 1, 3, 5 and 10% are plotted for the diesel injection pressures of 300, 600 and 1000 bars and the results for different EGR levels are as shown from ***Figure 89*** to ***Figure 92***.It is observed that the brake thermal efficiency remains the same as the diesel energy contribution increases. The efficiency is lower for lower injection pressures and the trend remains same for different levels of EGR.

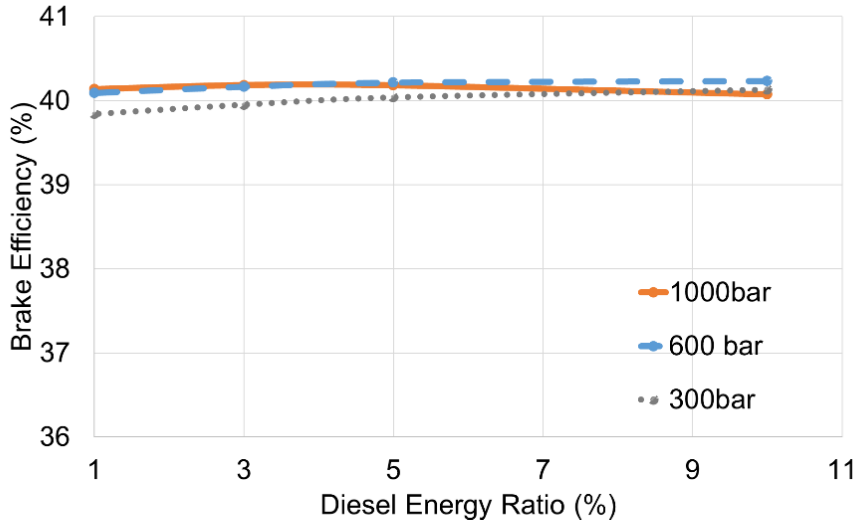

Figure 89 *Engine Speed=2138 RPM, SOI=10° BTDC: Effect of Diesel Energy Ratio (%) on Brake Thermal Efficiency (%) for* ***EGR=0%*** *for diesel injection pressures of 300,600 and 1000 bar*

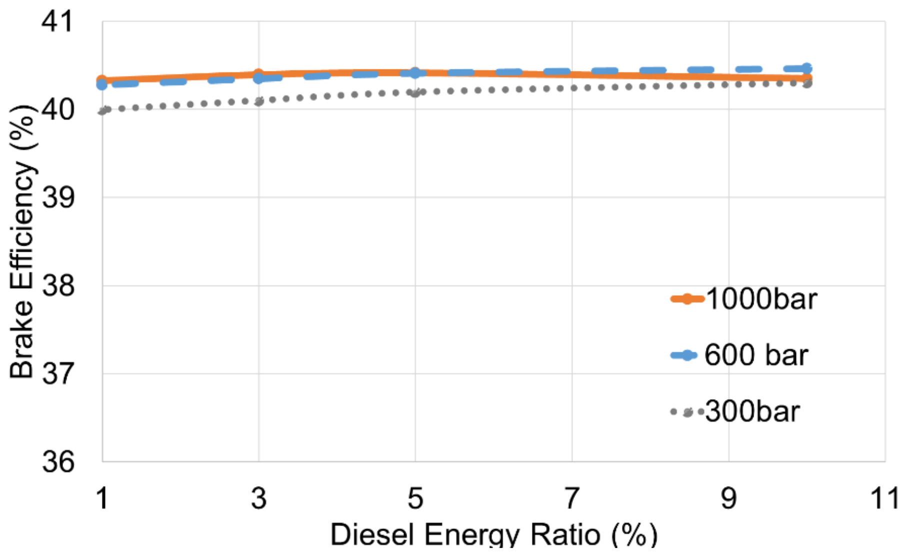

Figure 90 *Engine Speed=2138 RPM, SOI=10° BTDC: Effect of Diesel Energy Ratio (%) on Brake Thermal Efficiency (%) for* ***EGR=6%*** *for diesel injection pressures of 300,600 and 1000 bar*

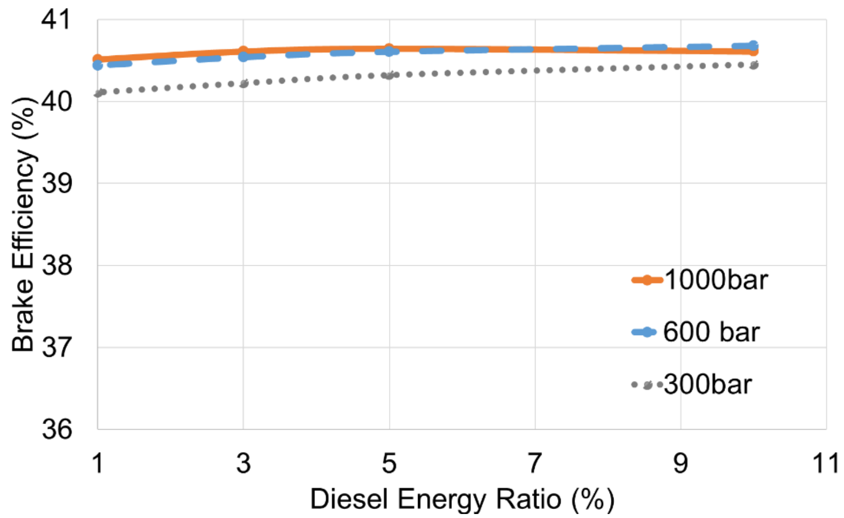

Figure 91 *Engine Speed=2138 RPM, SOI=10° BTDC: Effect of Diesel Energy Ratio (%) on Brake Thermal Efficiency (%) for* ***EGR=12%*** *for diesel injection pressures of 300,600 and 1000 bar*

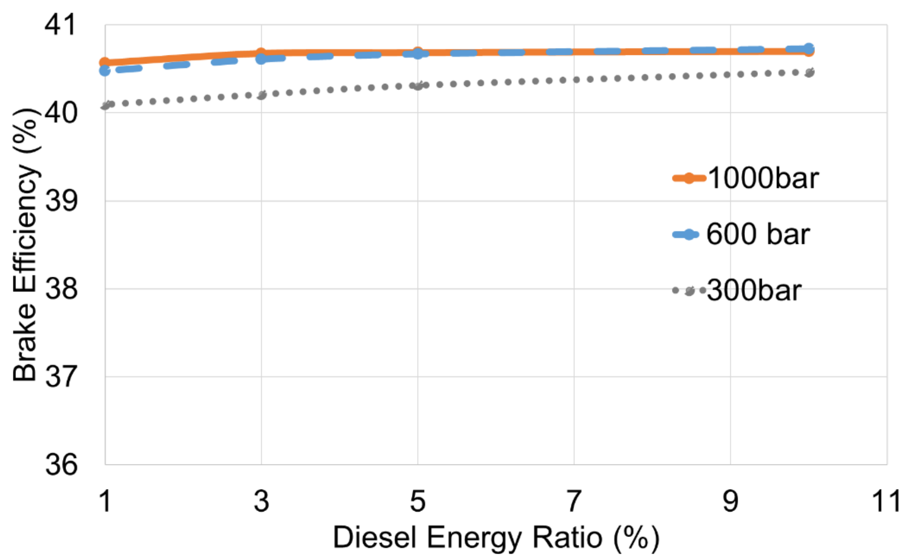

Figure 92 *Engine Speed=2138 RPM, SOI=10° BTDC: Effect of Diesel Energy Ratio (%) on Brake Thermal Efficiency (%) for* ***EGR=18%*** *for diesel injection pressures of 300,600 and 1000 bar*

7.4.1.4 Diesel Energy Contribution Vs Peak Cylinder Pressure

The peak cylinder pressure data for engine cylinder-1 is collected from simulation for the diesel energy contribution of 1, 3, 5 and 10% operating with injection pressures of 300,600 and 1000 bars at various EGR levels (0, 6, 12 and 18%). The results compared with diesel energy distribution percentage are as shown from ***Figure 93*** to ***Figure 96***. It is observed that as the peak cylinder pressure increases as the diesel energy contribution increases from 1-10 %.

The peak cylinder pressure is higher for higher diesel injection pressures. The trend is similar for all the cases of EGR percentage.

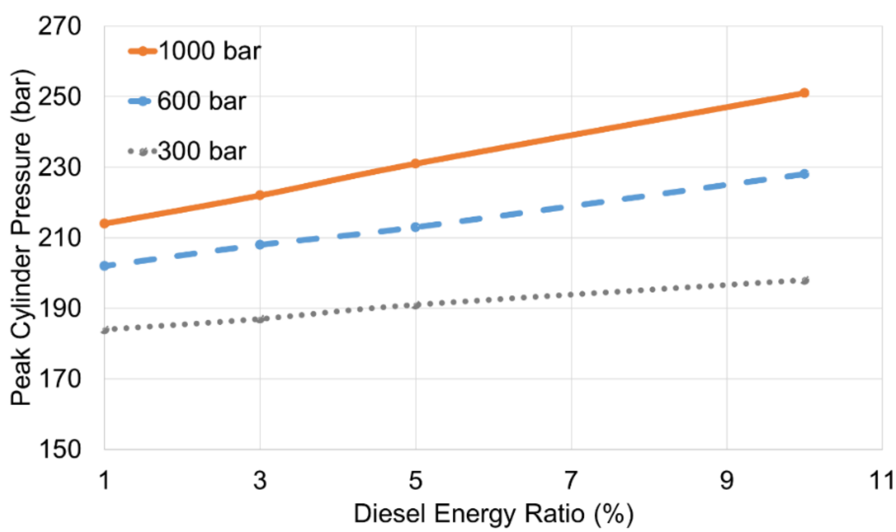

Figure 93 *Engine Speed = 2138 RPM, SOI=10° BTDC: Effect of Diesel Energy Ratio (%) on Peak Cylinder Pressure (bar) for* ***EGR=0%*** *for diesel injection pressures of 300,600 and 1000 bar*

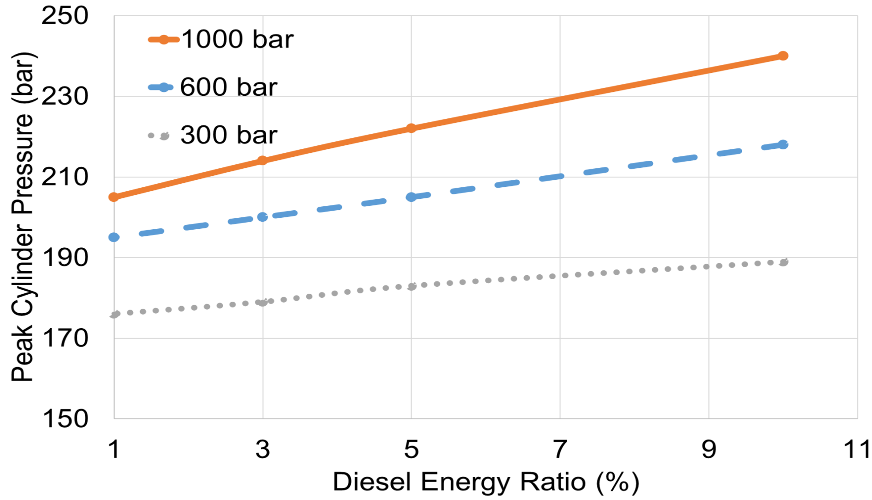

Figure 94 *Engine Speed = 2138 RPM, SOI=10° BTDC: Effect of Diesel Energy Ratio (%) on Peak Cylinder Pressure (bar) for* ***EGR=6****% for diesel injection pressures of 300,600 and 1000 bar*

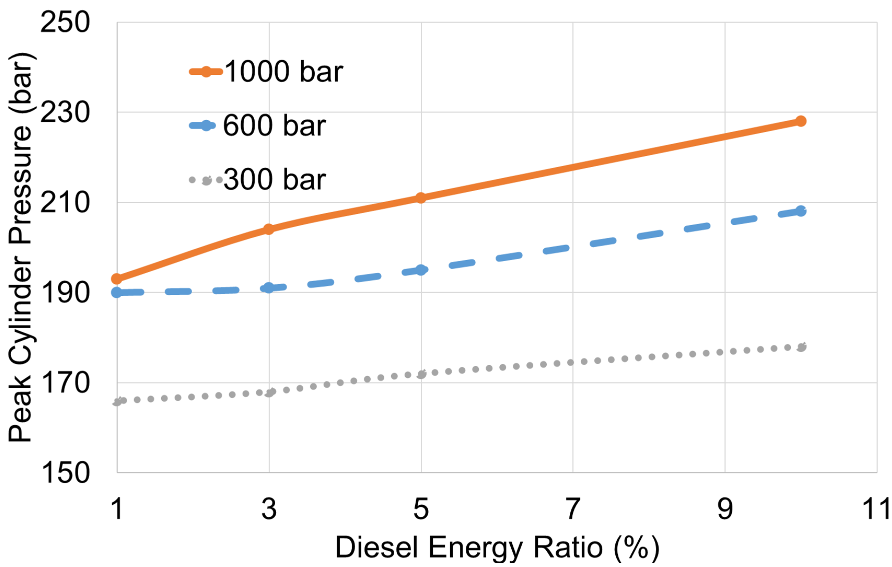

Figure 95 *Engine Speed = 2138 RPM, SOI=10° BTDC: Effect of Diesel Energy Ratio (%) on Peak Cylinder Pressure (bar) for* ***EGR=12****% for diesel injection pressures of 300,600 and 1000 bar*

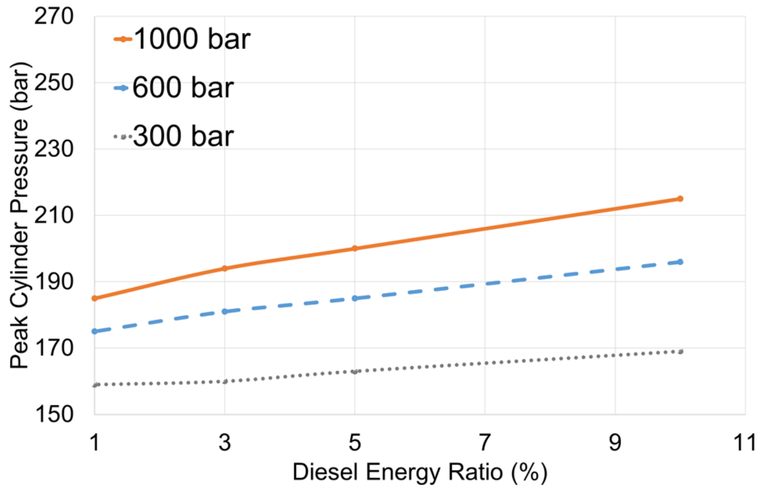

Figure 96 *Engine Speed = 2138 RPM, SOI=10° BTDC: Effect of Diesel Energy Ratio (%) on Peak Cylinder Pressure (bar) for* ***EGR=18%*** *for diesel injection pressures of 300,600 and 1000 bar*

The in-cylinder pressure trace and the burned mass fraction plots for cases of EGR=0% for diesel energy ratios of 1% and 10% at different injection pressures are as shown in ***Figure 97*** and ***Figure 98***.

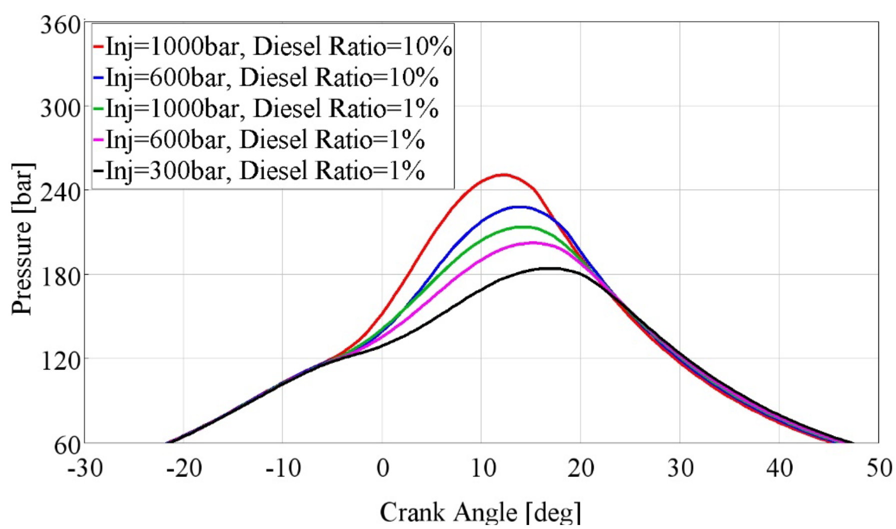

Figure 97 *Engine Speed=2138 RPM, SOI=10° BTDC: In-Cylinder Pressure trace for* ***EGR=0%*** *for different diesel energy ratios and injection pressures (Inj)*

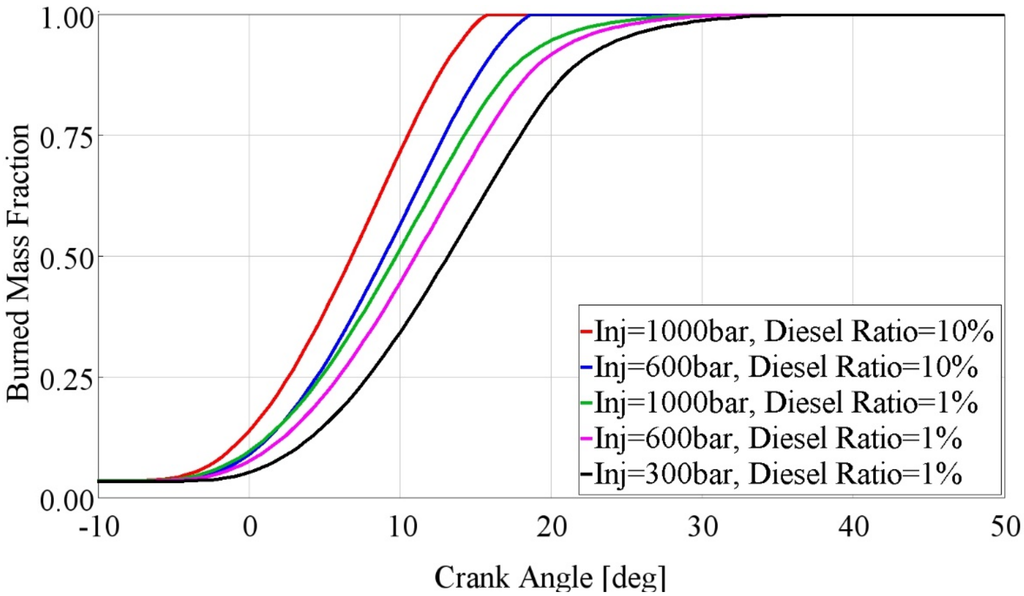

Figure 98 *Engine Speed=2138 RPM, SOI=10° BTDC: Burned mass fractions for* ***EGR=0%*** *for different diesel energy ratios and injection pressures (Inj)*

The burned mass fractions are quicker for higher diesel energy ratios and higher injection pressures, causing high in-cylinder pressures.

7.4.2 Effect of EGR on Engine Performance

The results obtained for high-speed cases (Engine Speed= 2138 RPM) at diesel injection timing of 10° bTDC are further analyzed to study the effect of EGR on the performance of the engine.

7.4.2.1 EGR Vs BMEP

The results for BMEP versus EGR for diesel energy contributions 1 & 10 percentage are summarized in ***Figure 99*** and ***Figure 100***. It is observed that as the EGR level increases, the value of BMEP reduces. The effect of diesel injection pressure is less for diesel energy ratio contribution of 1% and for 10% diesel energy contribution, lower diesel injection pressures lead to higher BMEP.

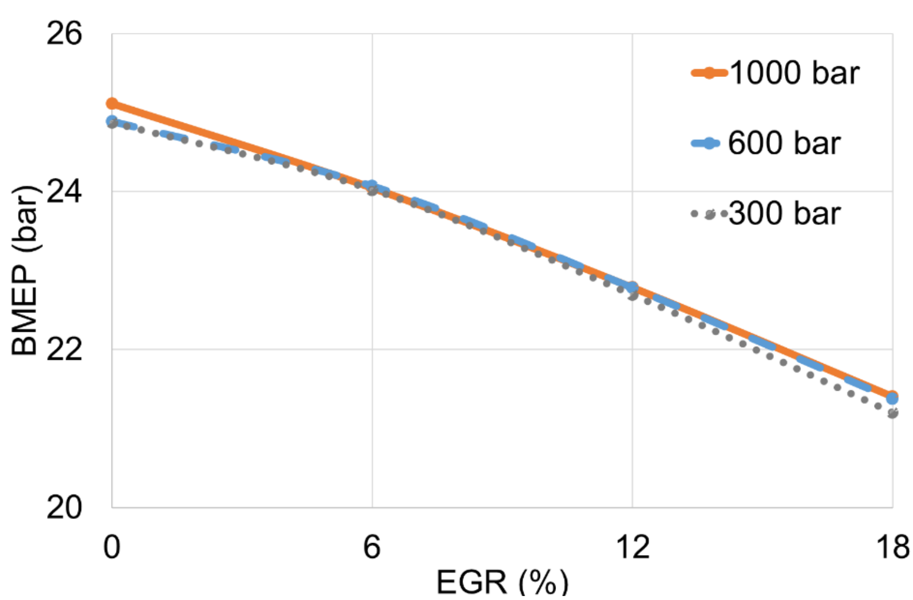

Figure 99 *Engine Speed = 2138 RPM, SOI=10° BTDC: Effect of EGR (%) on BMEP (bar) for* ***Diesel Energy Ratio = 1%*** *for diesel injection pressures of 300,600 and 1000 bar*

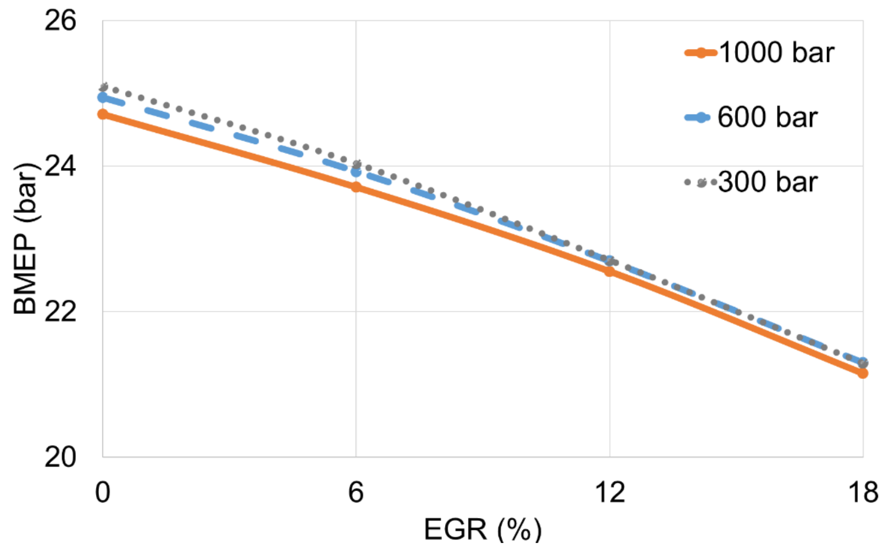

***Figure 100** Engine Speed = 2138 RPM, SOI=10° BTDC: Effect of EGR (%) on BMEP (bar) for **Diesel Energy Ratio = 10%** for diesel injection pressures of 300,600 and 1000 bar*

7.4.2.2 EGR Vs NO_x emissions

The results for NO_x emissions versus the EGR variation for diesel energy distributions 1 & 10% are as shown in ***Figure 101*** and ***Figure 102***. As the EGR level is increased, the NO_x emissions decrease. It is observed that as the diesel energy contribution increases, the NO_x emissions increase due to higher amount of diesel in the charge mixture. The level of NO_x is also high for higher injection pressures. From the figures, it is observed that the value of NO_x emissions is the lowest for higher EGR and lower injection pressure.

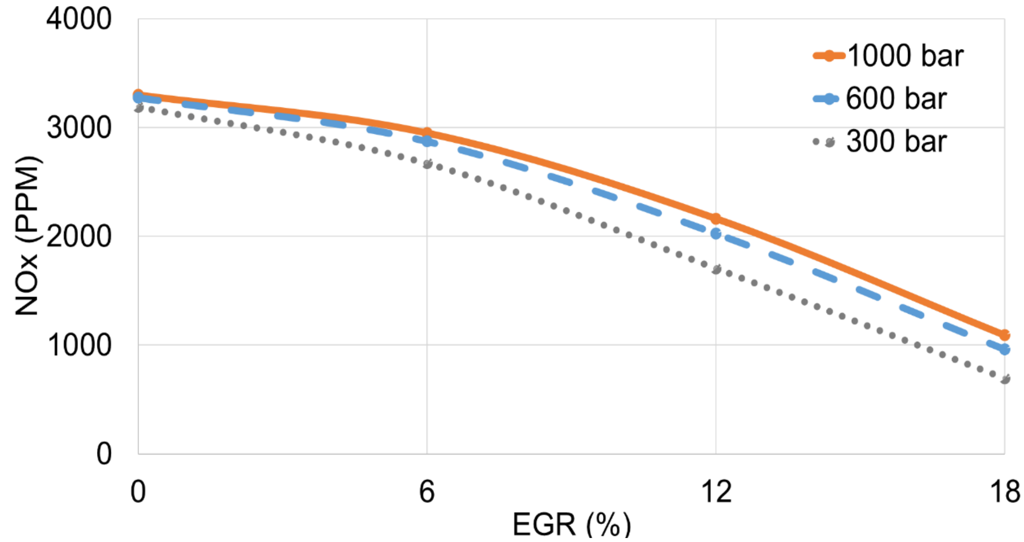

Figure 101 *Engine Speed = 2138 RPM, SOI=10° BTDC: Effect of EGR (%) on NO_x emissions (PPM) for* ***Diesel Energy Ratio = 1%*** *for diesel injection pressures of 300,600 and 1000 bar*

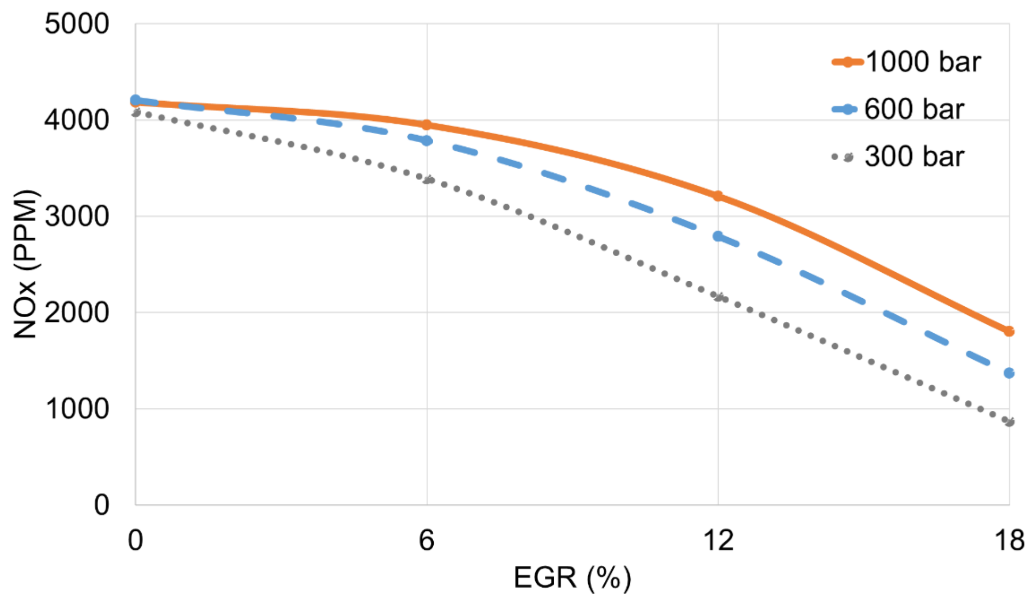

Figure 102 *Engine Speed = 2138 RPM, SOI=10° BTDC: Effect of EGR (%) on NO_x emissions (PPM) for* ***Diesel Energy Ratio = 10%*** *for diesel injection pressures of 300,600 and 1000 bar*

7.4.2.3 EGR Vs Peak Cylinder Pressure

The results for peak cylinder pressure versus EGR for diesel energy contributions 1 &10 percentages are summarized in ***Figure 103*** and ***Figure 104***. Peak cylinder pressures follow BMEP and NO_x emissions for EGR variation. The value of peak cylinder pressure reduces for higher EGR percentages. At each value of EGR, lower diesel injection pressures lead to lower values of peak cylinder pressures.

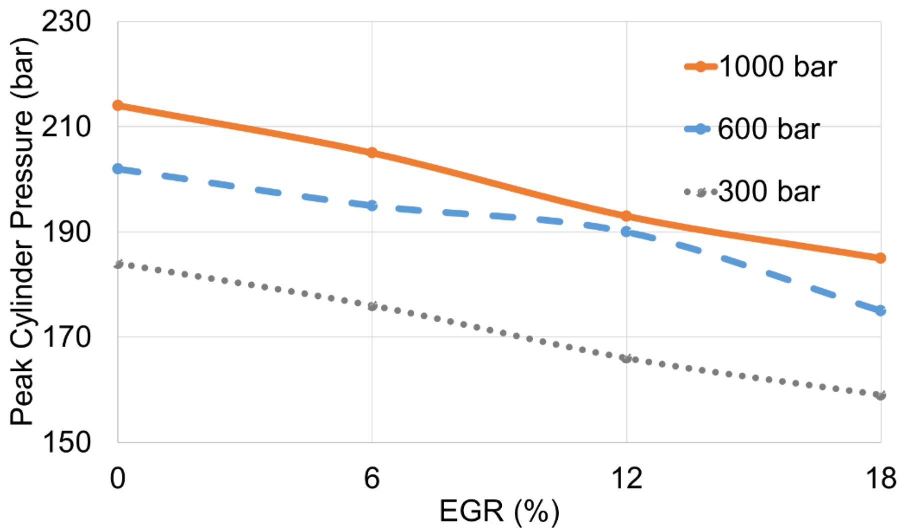

Figure 103 *Engine Speed = 2138 RPM, SOI=10° BTDC: Effect of EGR (%) on Peak Cylinder Pressure (bar) for **Diesel Energy Ratio = 1%** for diesel injection pressures of 300,600 and 1000 bar*

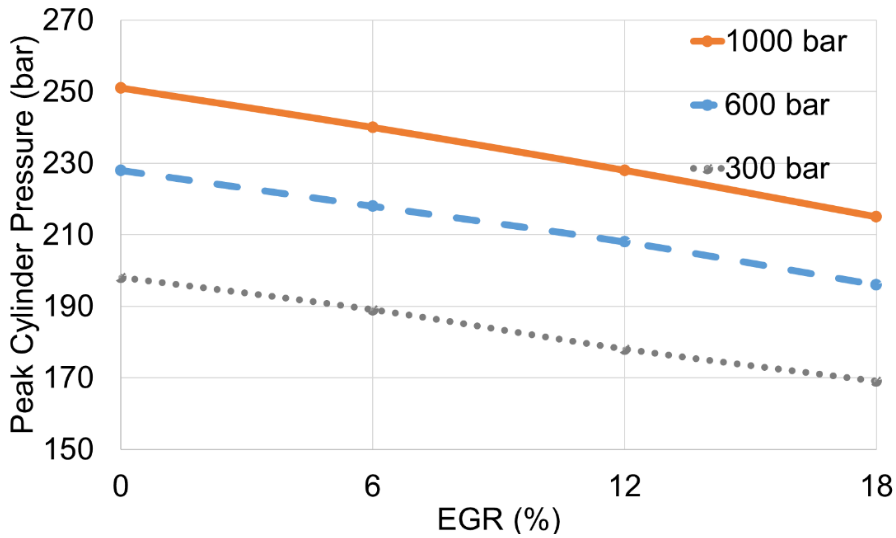

Figure 104 *Engine Speed = 2138 RPM, SOI=10° BTDC: Effect of EGR (%) on Peak Cylinder Pressure (bar) for* ***Diesel Energy Ratio = 10****% for diesel injection pressures of 300,600 and 1000 bar*

7.5 Simulation results for Injection Timing 10° bTDC, Engine Speed = 1397 RPM

Simulation results obtained for the dual-fuel engine model at 10° bTDC for low-speed case (Engine Speed = 1397 RPM) are discussed in this section.

7.5.1 Effect of Diesel Energy Contribution on Engine Performance

For EGR percentages of 0,6,12 and 18%, the diesel energy ratio is varied for 1, 3, 5 & 10% of the total fuel energy and tested for diesel injection pressures of 300, 600 and 1000 bar.

7.5.1.1 Diesel Energy Contribution Vs BMEP

The BMEP results for the variation of diesel energy ratio for each fraction of EGR are as summarized from ***Figure 105*** to ***Figure 108***. For cases of EGR 0% and 6%, the BMEP values decreases as the diesel energy contribution percentage increases. For lower diesel injection pressure (bar) of 300 bar, the reduction in value of BMEP for an increase in diesel energy contribution is very low.

For cases where EGR is 12 and 18 %, no significant reduction in BMEP is observed. Definite conclusions could not be made on the effect of injection pressures on the BMEP for the cases of higher EGR (12 and 18%).

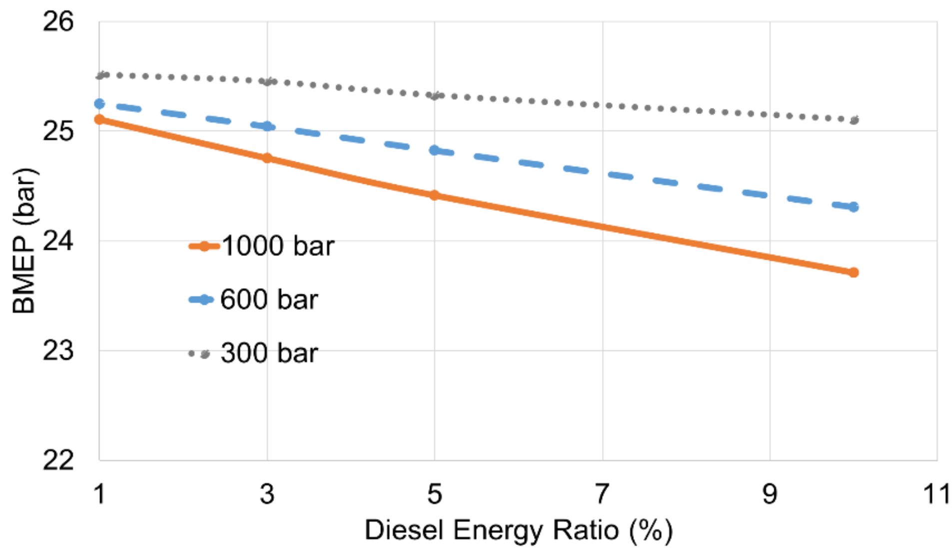

Figure 105 *Engine Speed = 1397 RPM, SOI=10° BTDC: Effect of Diesel Energy Ratio (%) on BMEP (bar) for* ***EGR=0*** *% for diesel injection pressures of 300,600 and 1000 bar*

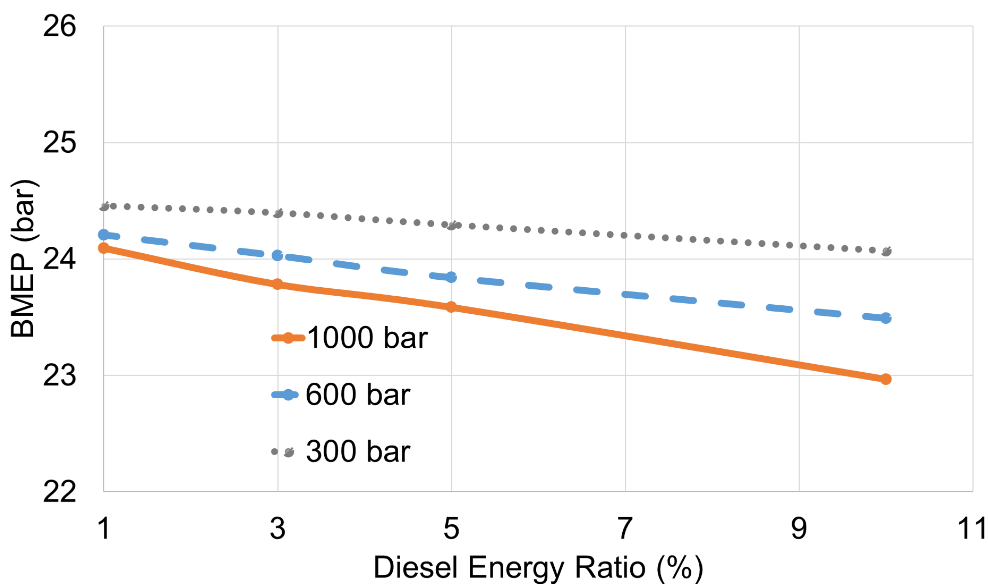

Figure 106 *Engine Speed = 1397 RPM, SOI=10° BTDC: Effect of Diesel Energy Ratio (%) on BMEP (bar) for* ***EGR=6*** *% for diesel injection pressures of 300,600 and 1000 bar*

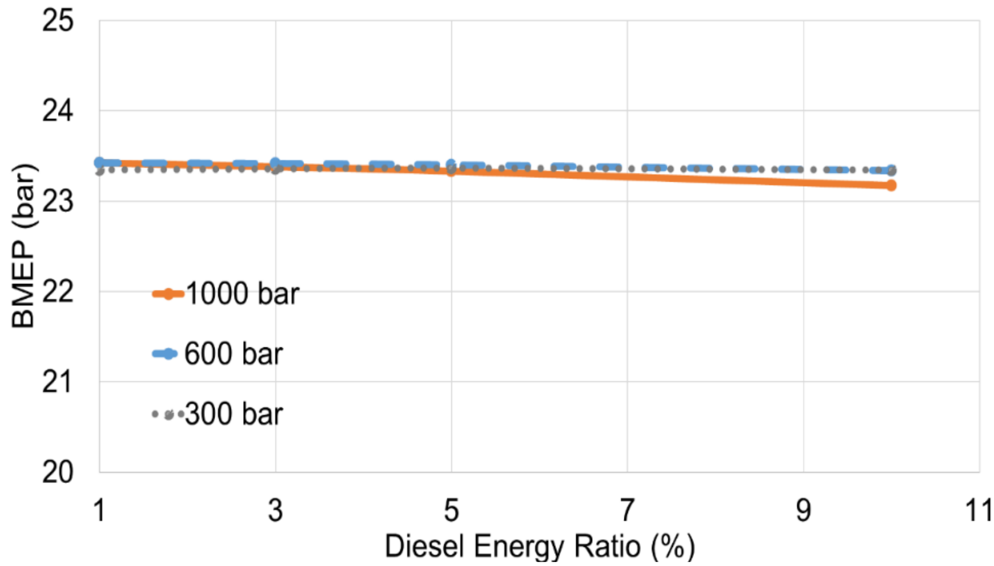

Figure 107 *Engine Speed = 1397 RPM, SOI=10° BTDC: Effect of Diesel Energy Ratio (%) on BMEP (bar) for* ***EGR=12*** *% for diesel injection pressures of 300,600 and 1000 bar*

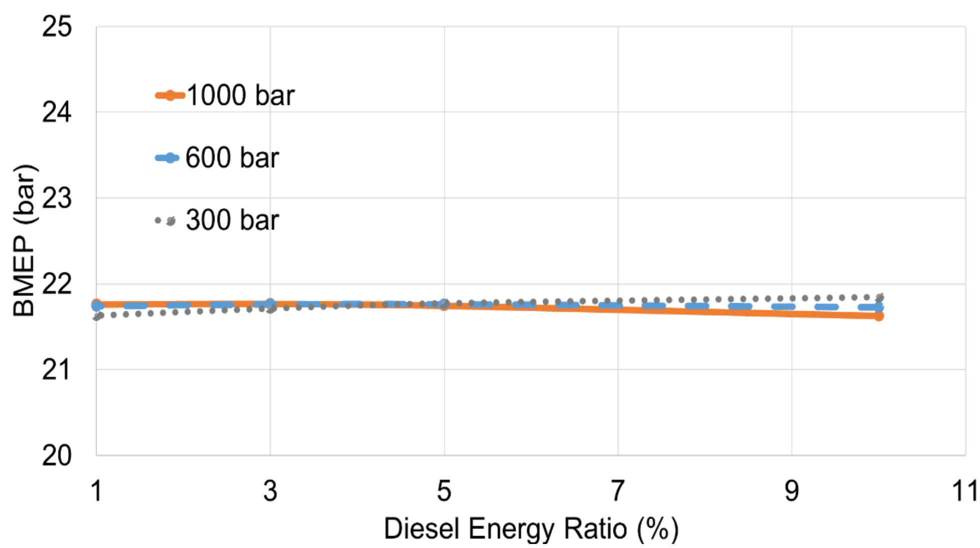

Figure 108 *Engine Speed = 1397 RPM, SOI=10° BTDC: Effect of Diesel Energy Ratio (%) on BMEP (bar) for* ***EGR=18*** *% for diesel injection pressures of 300,600 and 1000 bar*

7.5.1.2 Diesel Energy Contribution Vs NO_x emissions

For the variation of diesel energy contribution for 1,3,5 and 10%, for the diesel injection pressures of 300, 600 and 1000 bar, the NO_x emissions (in PPM) obtained from the simulation for each case of EGR are as shown from ***Figure 109*** to ***Figure 112***. The results for the low-speed case are similar as the high-speed case. The results indicate that as the diesel energy contribution percentage increases, NO_x emissions increase for all diesel injection pressure cases. Similar trend of increase is observed for all the values of EGR.

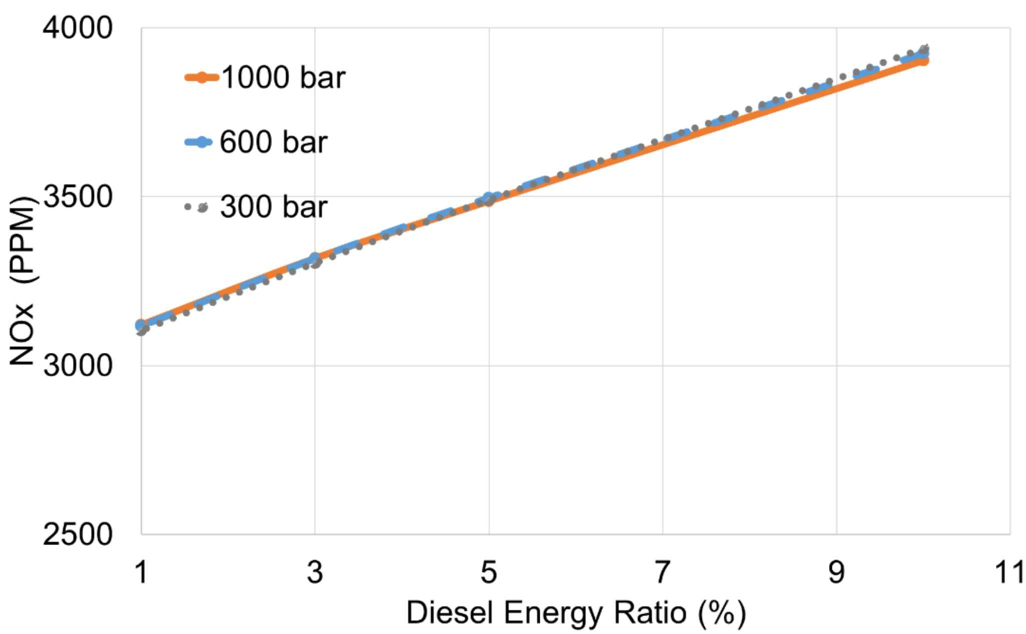

Figure 109 *Engine Speed = 1397 RPM, SOI=10° BTDC: Effect of Diesel Energy Ratio (%) on NO_x (PPM) for* ***EGR=0*** *% for diesel injection pressures of 300,600 and 1000 bar*

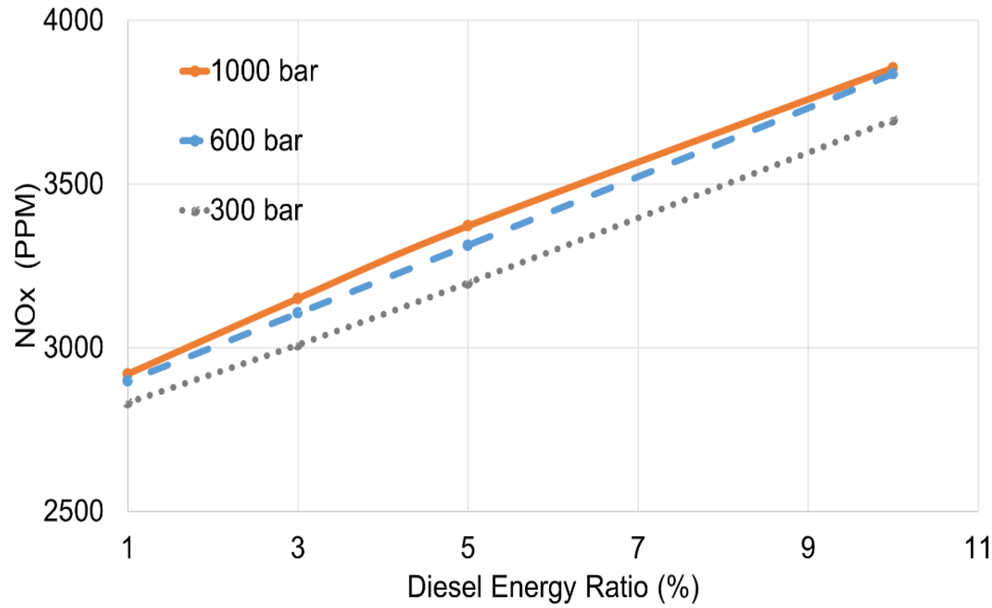

Figure 110 *Engine Speed = 1397 RPM, SOI=10° BTDC: Effect of Diesel Energy Ratio (%) on NO_x (PPM) for* ***EGR=6*** *% for diesel injection pressures of 300,600 and 1000 bar*

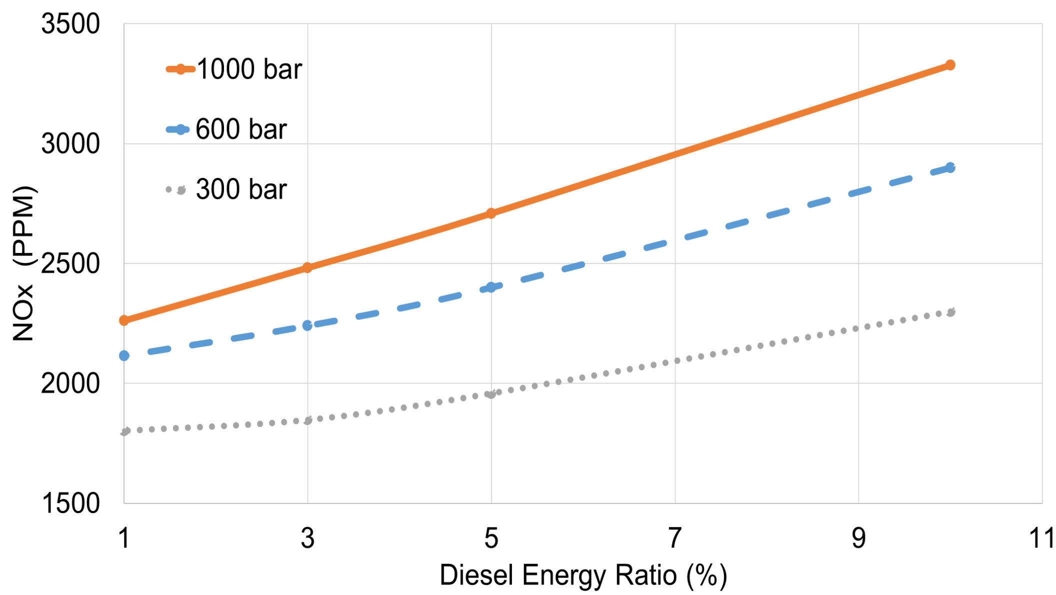

Figure 111 *Engine Speed = 1397 RPM, SOI=10° BTDC: Effect of Diesel Energy Ratio (%) on NO_x (PPM) for* ***EGR=12*** *% for diesel injection pressures of 300,600 and 1000 bar*

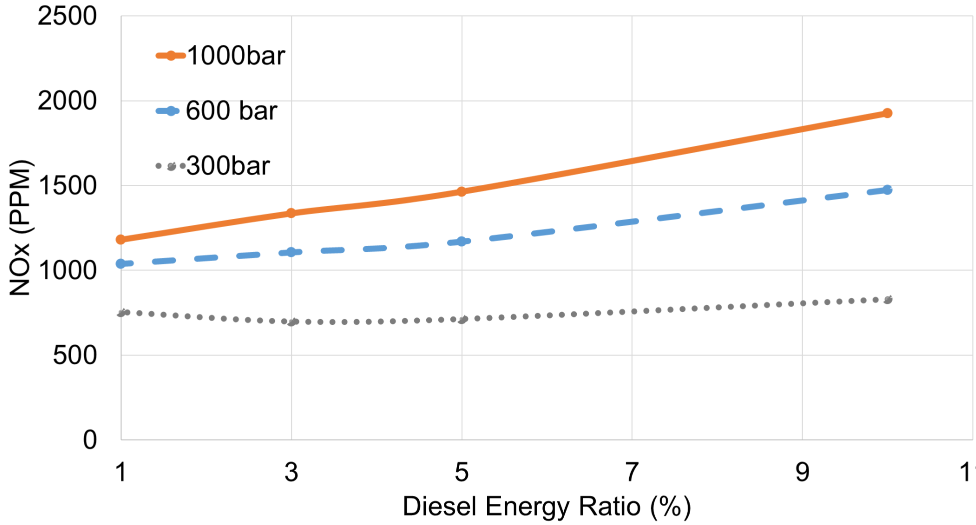

Figure 112 *Engine Speed = 1397 RPM, SOI=10° BTDC: Effect of Diesel Energy Ratio (%) on NO_x (PPM) for* ***EGR=18*** *% for diesel injection pressures of 300,600 and 1000 bar*

7.5.1.3 Diesel Energy Contribution Vs Brake Thermal Efficiency

The brake thermal efficiency for the diesel energy ratios 1, 3, 5 and 10% are plotted for the diesel injection pressures of 300, 600 and 1000 bars and the results for different EGR levels are as shown from ***Figure 113*** to ***Figure 116***.

It is observed that the brake thermal efficiency follows a similar trend as BMEP. For cases of EGR 0% and 6%, brake thermal efficiency values decreases as the diesel energy contribution percentage increases. For lower diesel injection pressure (bar) of 300 bar, the reduction in value of efficiency for an increase in diesel energy contribution is very low. For higher diesel injection pressures, lower values of thermal efficiency are observed and

as the diesel energy contribution increases, the reduction in efficiency for higher diesel injection pressures is more than that of the efficiency values for lower injection pressures. For cases where EGR is 12 and 18 %, significant reduction in efficiency is not observed. In addition, the effect of injection pressures on the efficiency for the cases of higher EGR (12 and 18%) is not significant. The EGR level does not affect the values of thermal efficiency as the thermal efficiency across different EGR levels remains almost the same.

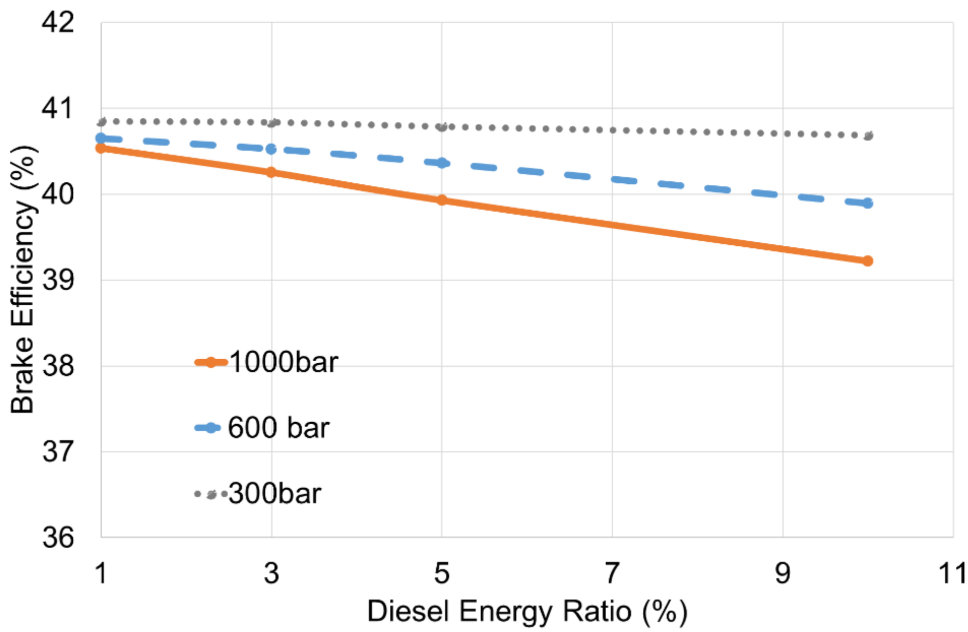

Figure 113 *Engine Speed=1397 RPM, SOI=10° BTDC: Effect of Diesel Energy Ratio (%) on Brake Thermal Efficiency (%) for* ***EGR=0%*** *for diesel injection pressures of 300,600 and 1000 bar*

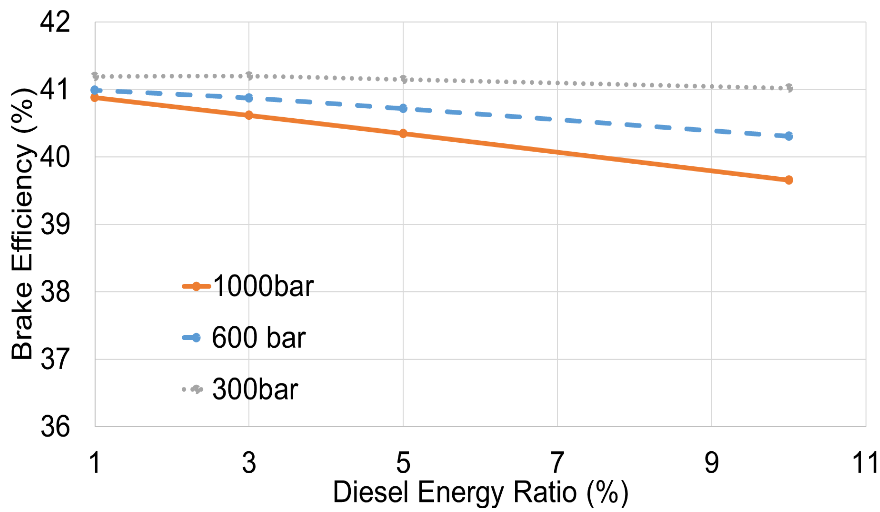

Figure 114 *Engine Speed=1397 RPM, SOI=10° BTDC: Effect of Diesel Energy Ratio (%) on Brake Thermal Efficiency (%) for* ***EGR=6%*** *for diesel injection pressures of 300,600 and 1000 bar*

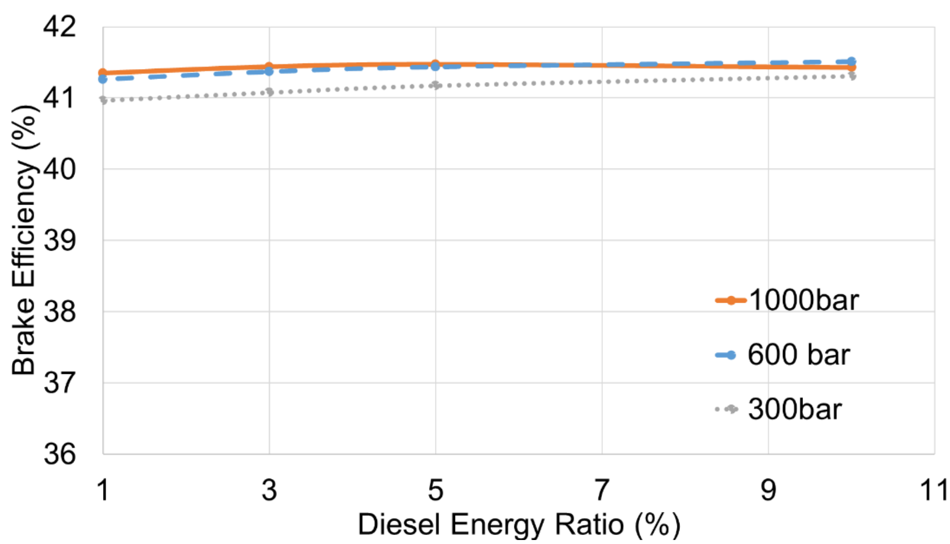

Figure 115 *Engine Speed=1397 RPM, SOI=10° BTDC: Effect of Diesel Energy Ratio (%) on Brake Thermal Efficiency (%) for* ***EGR=12%*** *for diesel injection pressures of 300,600 and 1000 bar*

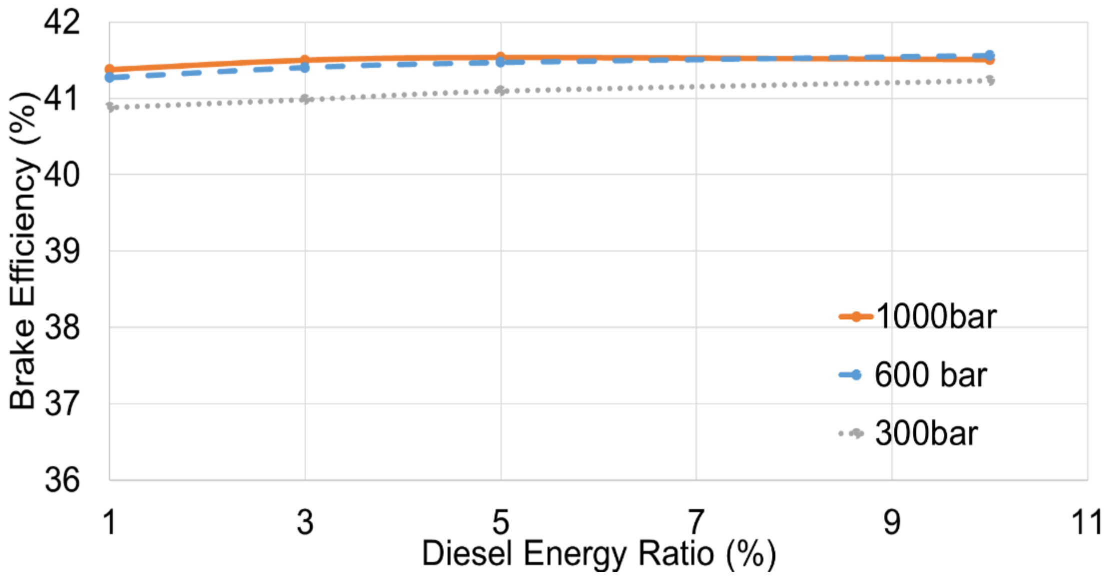

Figure 116 *Engine Speed=1397 RPM, SOI=10° BTDC: Effect of Diesel Energy Ratio (%) on Brake Thermal Efficiency (%) for* ***EGR=18%*** *for diesel injection pressures of 300,600 and 1000 bar*

7.5.1.4 Diesel Energy Contribution Vs Peak Cylinder Pressure

The peak cylinder pressure data for engine cylinder-1 is collected from simulation results for the diesel energy contribution of 1, 3, 5 and 10% operating with injection pressures of 300,600 and 1000 bars at EGR levels 0,6,12 and 18%. The results compared for variation of diesel energy distribution percentage are as shown from ***Figure 117*** to ***Figure 120***. It is observed that as the peak cylinder pressure increases as the diesel energy contribution increases from 1-10 percentages. The peak cylinder pressure is higher for higher diesel injection pressures. The trend is similar for all the cases of EGR percentage.

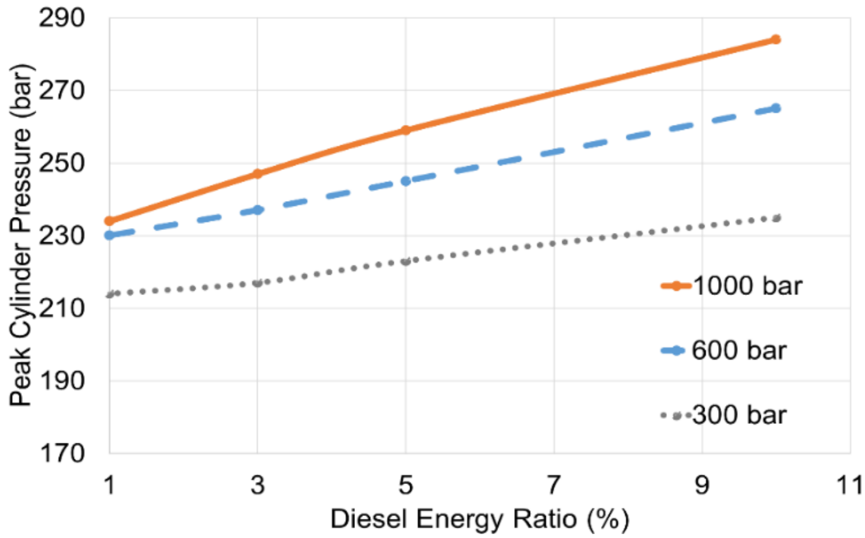

Figure 117 *Engine Speed = 1397 RPM, SOI=10° BTDC: Effect of Diesel Energy Ratio (%) on Peak Cylinder Pressure (bar) for **EGR=0** % for diesel injection pressures of 300,600 and 1000 bar*

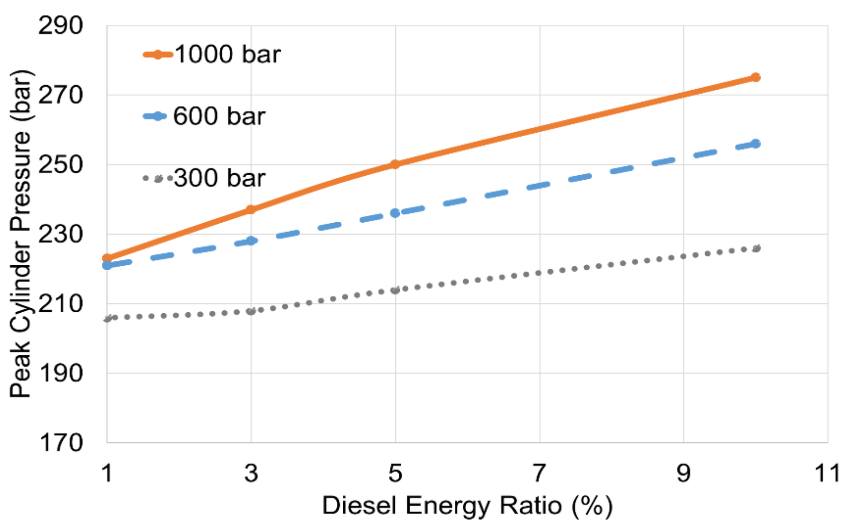

Figure 118 *Engine Speed = 1397 RPM, SOI=10° BTDC: Effect of Diesel Energy Ratio (%) on Peak Cylinder Pressure (bar) for **EGR=6** % for diesel injection pressures of 300,600 and 1000 bar*

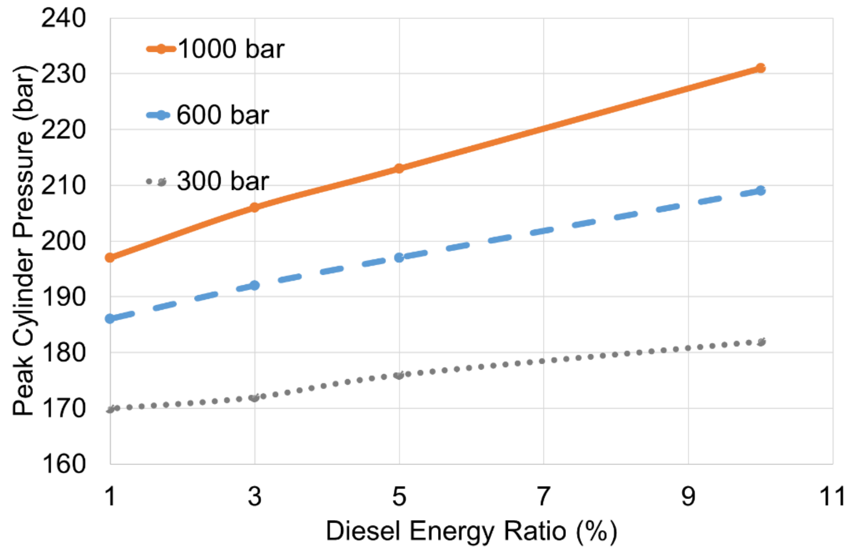

Figure 119 *Engine Speed = 1397 RPM, SOI=10° BTDC: Effect of Diesel Energy Ratio (%) on Peak Cylinder Pressure (bar) for* ***EGR=12*** *% for diesel injection pressures of 300,600 and 1000 bar*

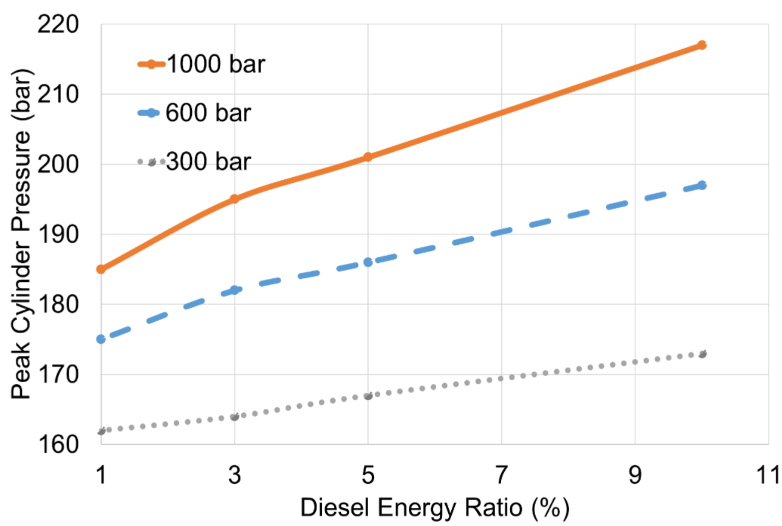

Figure 120 *Engine Speed = 1397 RPM, SOI=10° BTDC: Effect of Diesel Energy Ratio (%) on Peak Cylinder Pressure (bar) for* ***EGR=18*** *% for diesel injection pressures of 300,600 and 1000 bar*

7.5.2 Effect of EGR on Engine Performance

The results for the low-speed case obtained from simulation are further analyzed to study the effect of EGR on the performance of the engine.

7.5.2.1 EGR Vs BMEP

The results for Brake mean effective pressures versus EGR for diesel energy contributions 1 and 10 percentages are summarized in ***Figure 121*** and ***Figure 122***. It is observed that as the EGR level increases, the value of BMEP reduces. For diesel energy contribution of 1%, at EGR levels 0 and 6%, the BMEP is higher at lower injection pressures. The BMEP is not affected by the injection pressures at higher EGR levels (12,18%).

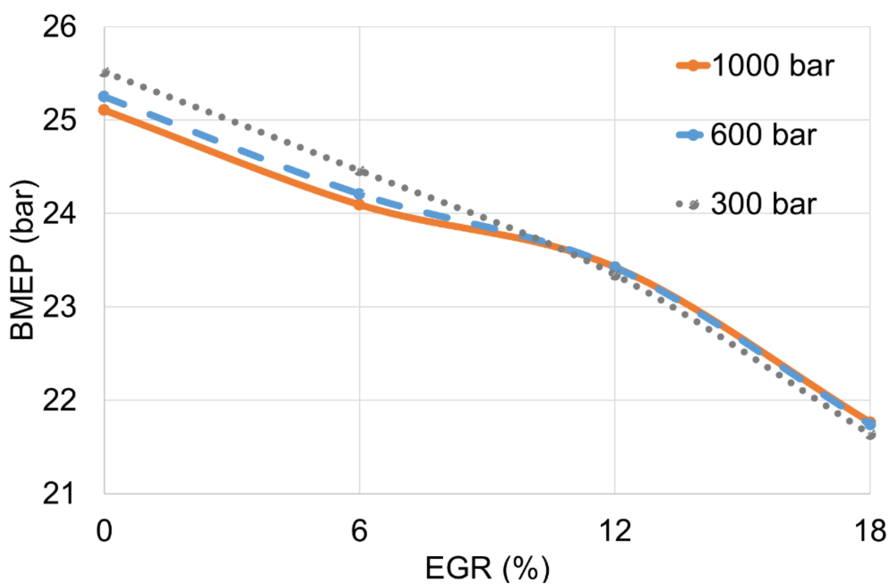

Figure 121 *Engine Speed = 1397 RPM, SOI=10° BTDC: Effect of EGR (%) on BMEP (bar) for* ***Diesel Energy Ratio = 1%*** *for diesel injection pressures of 300,600 and 1000 bar*

For 10% diesel energy contribution, it is observed that at lower EGR levels, the BMEP is lower for higher diesel injection pressures and as the EGR level increases, the diesel injection pressure has negligible effect on BMEP.

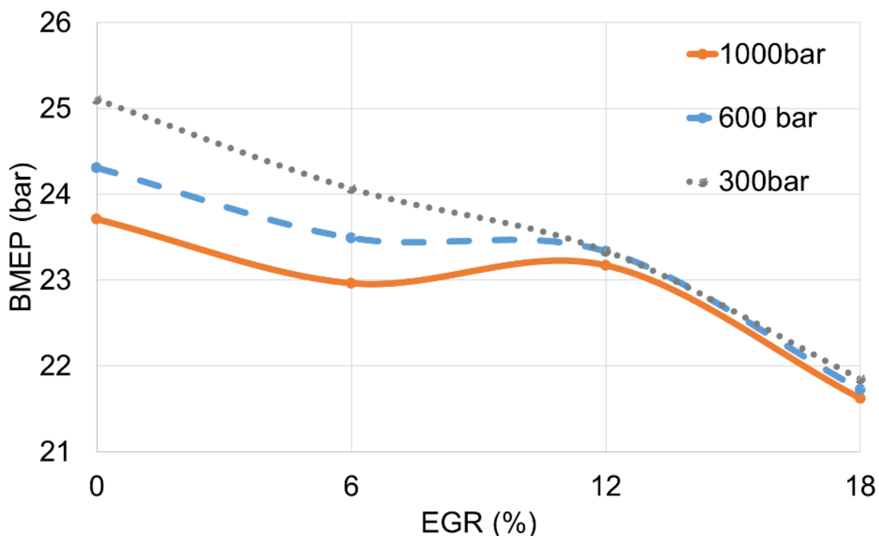

Figure 122 *Engine Speed = 1397 RPM, SOI=10° BTDC: Effect of EGR (%) on BMEP (bar) for* ***Diesel Energy Ratio = 10%*** *for diesel injection pressures of 300,600 and 1000 bar*

7.5.2.2 EGR Vs NO_x emissions

The results for NO_x emissions versus the EGR variation for diesel energy distributions 1 & 10% are as shown in ***Figure 123*** and ***Figure 124***. For EGR variation from 0-6%, the NO_x emissions do not vary significantly. As the EGR level increases from 6 to 18%, the reduction in NO_x emissions is almost linear. The level of NO_x is high for higher injection pressures. From ***Figure 123*** and ***Figure 124***, it is observed that the value of NO_x emissions

is the lowest for higher EGR and lower injection pressure and the trend remains same over the diesel energy contribution percentages.

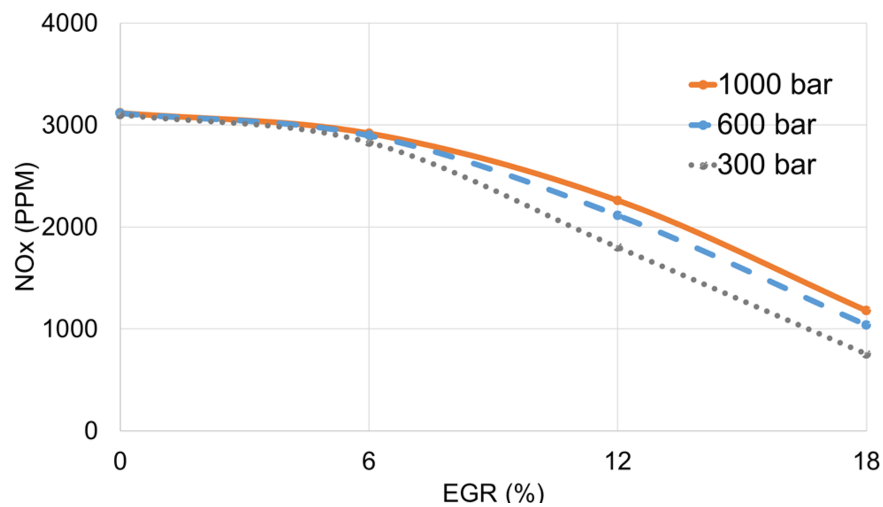

Figure 123 *Engine Speed = 1397 RPM, SOI=10° BTDC: Effect of EGR (%) on NO_x emissions (PPM) for **Diesel Energy Ratio = 1**% for diesel injection pressures of 300,600 and 1000 bar*

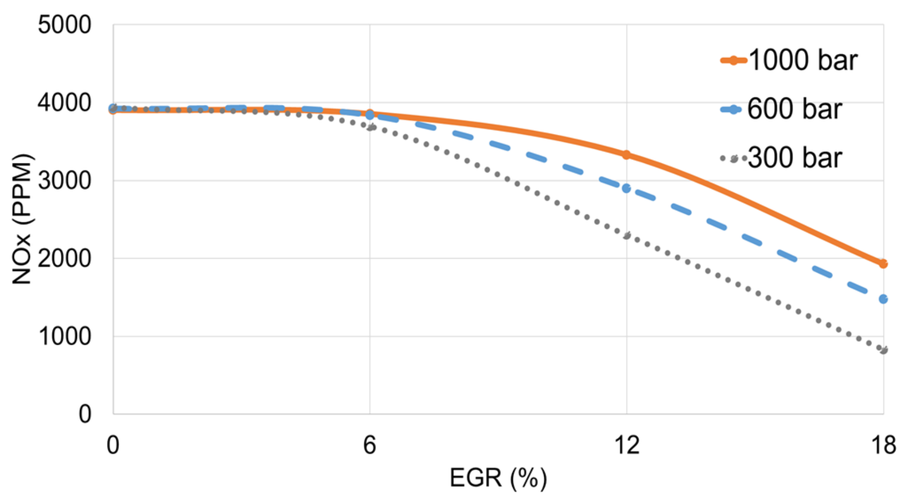

Figure 124 *Engine Speed = 1397 RPM, SOI=10° BTDC: Effect of EGR (%) on NO_x emissions (PPM) for **Diesel Energy Ratio = 10**% for diesel injection pressures of 300,600 and 1000 bar*

7.5.2.3 EGR Vs Peak Cylinder Pressure

The results for peak cylinder pressure versus EGR for diesel energy contributions 1 & 10 percentages are summarized in ***Figure 125*** and ***Figure 126***. The value of peak cylinder pressure reduces for higher EGR percentages. The trend for decrease in the peak cylinder pressure is not linear. The reduction in peak cylinder values is less than 10 bar from EGR level 0 to 6% and the reduction is higher between EGR levels 6 and 12%. From 12 to 18% EGR level variation, the peak cylinder pressure does not decrease as rapid as that from 6-12% EGR levels. At each value of EGR, lower diesel injection pressures lead to lower values of peak cylinder pressures.

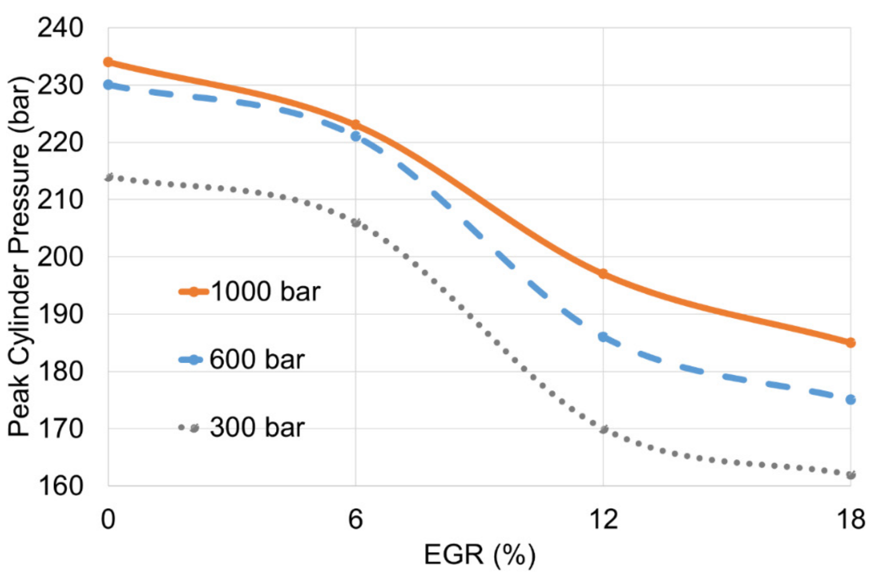

Figure 125 *Engine Speed = 1397 RPM, SOI=10° BTDC: Effect of EGR (%) on Peak Cylinder Pressure (bar) for* ***Diesel Energy Ratio = 1%*** *for diesel injection pressures of 300,600 and 1000 bar*

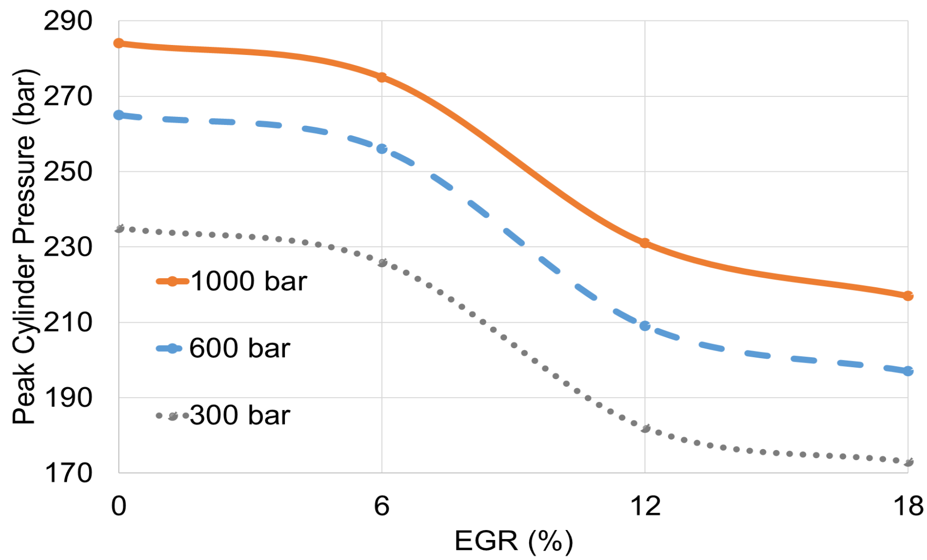

***Figure 126** Engine Speed = 1397 RPM, SOI=10° BTDC: Effect of EGR (%) on Peak Cylinder Pressure (bar) for **Diesel Energy Ratio = 10**% for diesel injection pressures of 300,600 and 1000 bar*

7.6 Effect of boost pressure

To study the effect of boost pressure on the engine performance, the cases for which lowest BMEP is achieved for dual-fuel engine simulation at diesel injection timings 0° and 10° bTDC are selected and a sweep is performed for boost pressure. The results at each diesel injection timing are discussed as follows:

7.6.1 Diesel Injection Timing 0° bTDC

The lowest BMEP for the simulation at 0° bTDC is 18.1 bar, obtained for EGR=18%, diesel energy contribution of 1% at diesel injection pressure of 300 bar. The NO_x emissions is found to be 109.6 PPM and the peak cylinder pressure is 121 bar, at this engine operating condition.

To study the effect of boost pressure on the BMEP, the boost pressure sweep is performed for the dual-fuel simulation model from 2.5 bar to 3 bar and the engine operating conditions are as summarized in ***Table 39***.

Table 39 *Engine Test Conditions to study the effect of boost pressure at diesel injection timing of 0° bTDC in dual-fuel engine*

Parameter	Unit	Value
Engine Speed	RPM	2138
Main SOI	deg bTDC	0
Diesel Energy Contribution	%	1
Overall Lambda	-	1
Air-fuel Ratio for Natural gas	-	16.96
EGR Rate	%	18
Diesel Injection Pressure	bar	300

The simulation results for the tests performed by variation of boost pressure for the optimum case are summarized in ***Table 42***

Table 40 *Effect of Boost Pressure at diesel injection timing of 0° bTDC on the engine performance*

Parameter	Unit	Boost Pressure (bar)			
		2.5	**2.7**	**2.8**	**3**
Net IMEP	bar	19.6	21.1	21.9	23.1
BMEP	bar	18.1	19.6	20.3	21.5
Torque	N-m	960.9	1039.7	1079.9	1144.0
NO_x emissions	ppm	103.8	109.4	112.7	111.1
BSFC	g/kW-h	218.4	217.4	216.7	218.4
Brake Efficiency	%	33.7	33.9	34.0	33.7
PCP	bar	121	131	136	147
Location of PCP	deg aTDC	0			

The results of BMEP for the boost pressure sweep is represented as shown in ***Figure 127***.

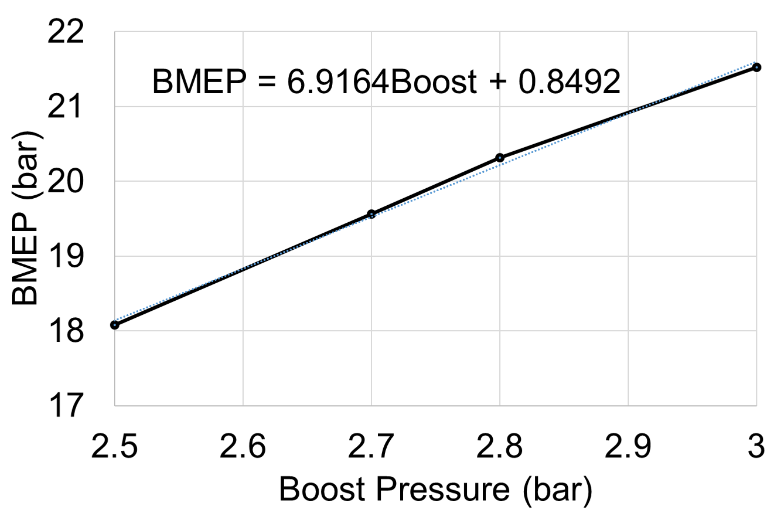

Figure 127 *SOI=0°bTDC: Effect of Boost Pressure on BMEP of Dual-Fuel Engine*

The trend line equation established for the BMEP as a function of boost pressure is as shown in ***Figure 127***. Based on the trend line equation, for a BMEP of 25 bar, the boost pressure required is estimated to be **3.49 bar**.

For the boost pressure sweep, the operating points as plotted on the compressor performance map are as shown in ***Figure 128***.

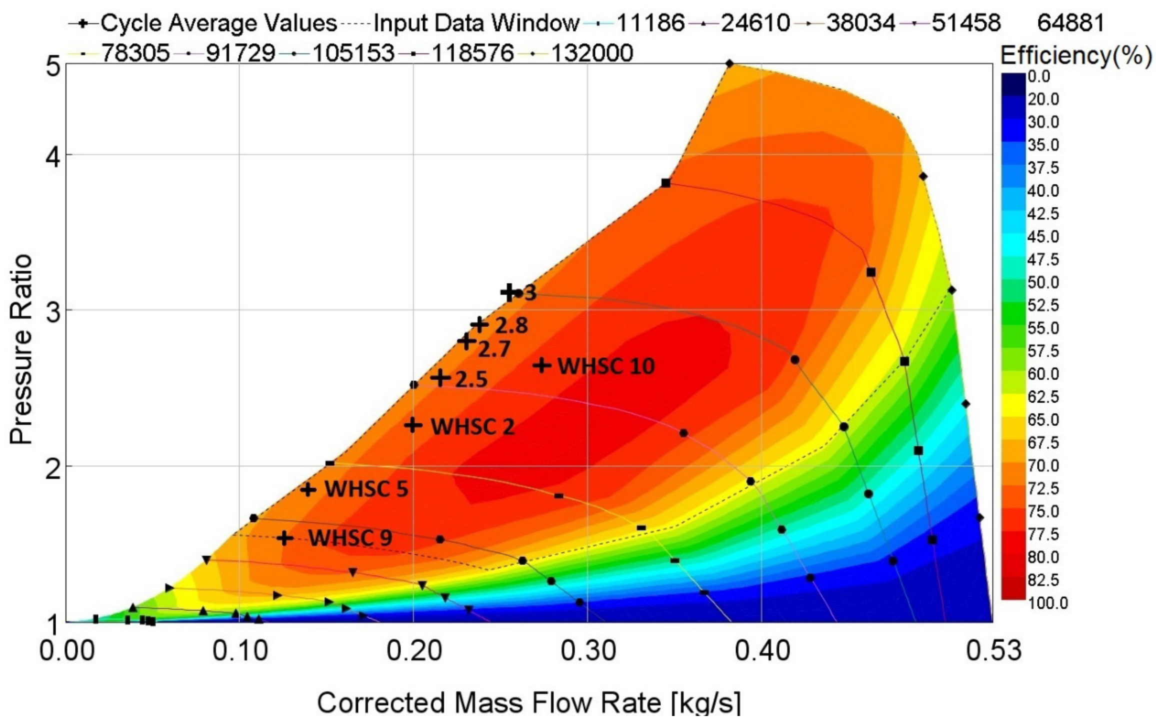

Figure 128 *SOI=0°bTDC: Compressor Performance Map for Boost pressure sweep and diesel mode test points for WHSC 2,5,9 and 10*

The operating points for validated cases WHSC 2, 5, 9 and 10 for diesel mode are plotted for comparison. The operating points for boost pressure sweep (represented as 2.5, 2.7, 2.8 and 3 in ***Figure 128***) are found to be operating close to the surge limit of the compressor.

7.6.2 Diesel Injection Timing 10° bTDC

For high speed operating condition in dual-fuel model, at a boost pressure of 2.5 bar, for diesel injection pressure of 300 bar, diesel energy contribution of 1% at EGR level of 18%, a BMEP of 21.2 bar is achieved from simulation model. This operating condition is found to have the lowest NO_x emissions of 718 PPM and the lowest peak cylinder pressure of 159 bar.

To achieve the target BMEP of 25 bar, the boost pressure is increased keeping the diesel energy contribution as 1% as per equation **(12) .** The test conditions are specified in ***Table 41***.

Table 41 *Engine test conditions to study the effect of boost pressure at diesel injection timing of 10° bTDC in dual-fuel engine*

Parameter	Unit	Value
Engine Speed	RPM	2138
Main SOI	deg bTDC	10
Diesel Energy Contribution	%	1
Overall Lambda	-	1
Air-fuel Ratio for Natural gas	-	16.963
EGR Rate	%	18
Diesel Injection Pressure	bar	300

The simulation results for the tests performed by variation of boost pressure for the optimum case are summarized in ***Table 42***.

Table 42 *Effect of boost pressure on the engine performance*

Parameter	Unit	Boost Pressure (bar)			
		2.5	**2.7**	**2.8**	**3**
Net IMEP	bar	22.8	24.6	25.6	27.4
BMEP	bar	21.2	22.9	23.8	25.6
Torque	N-m	1125.0	1218.9	1266.9	1361.5
NO_x emissions	ppm	718.0	753.3	787.3	846.6
BSFC	g/kW-h	183.5	182.5	181.9	180.9
Brake Efficiency	%	40.1	40.3	40.5	40.7
PCP	bar	159	169	178	188
Location of PCP	deg aTDC	16			

The effect of boost pressure on BMEP, obtained from simulation results as given in ***Table 42***, is represented by ***Figure 129***.

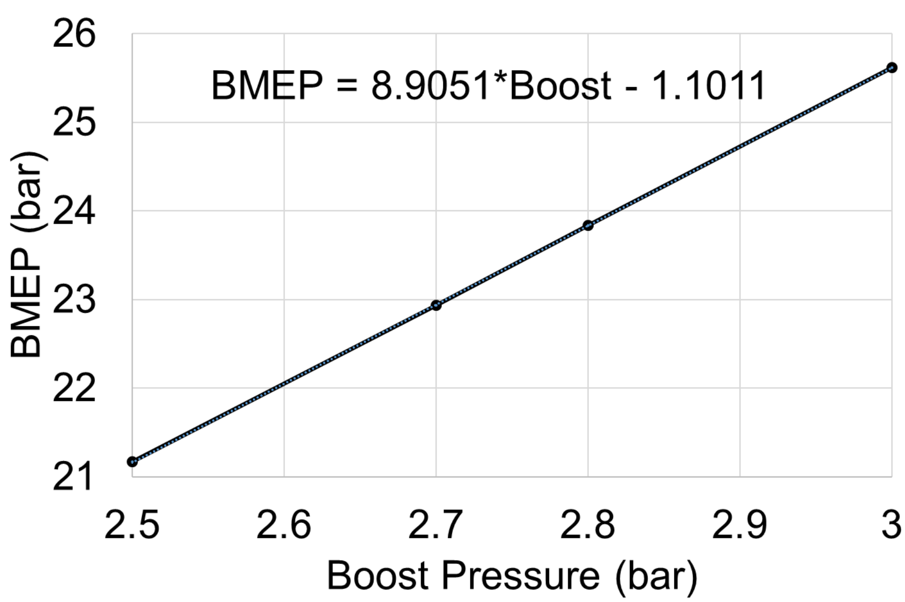

Figure 129 *SOI=10°bTDC: Effect of boost pressure on BMEP of Dual-Fuel engine*

The trend line equation established for the BMEP as a function of boost pressure is as shown in ***Figure 129***. Based on the trend line equation, for a BMEP of 25 bar, the boost pressure required is found to be **2.93 bar**.

For the boost pressure sweep, the operating points as plotted on the compressor performance map are as shown in ***Figure 130***. Similar to the cases at SOI=0° bTDC, the operating points at SOI=10° bTDC are found to be operating close to the surge line and for boost pressure of 3 bar, the operating point is beyond the surge line, which would cause compressor surge when operated in the engine.

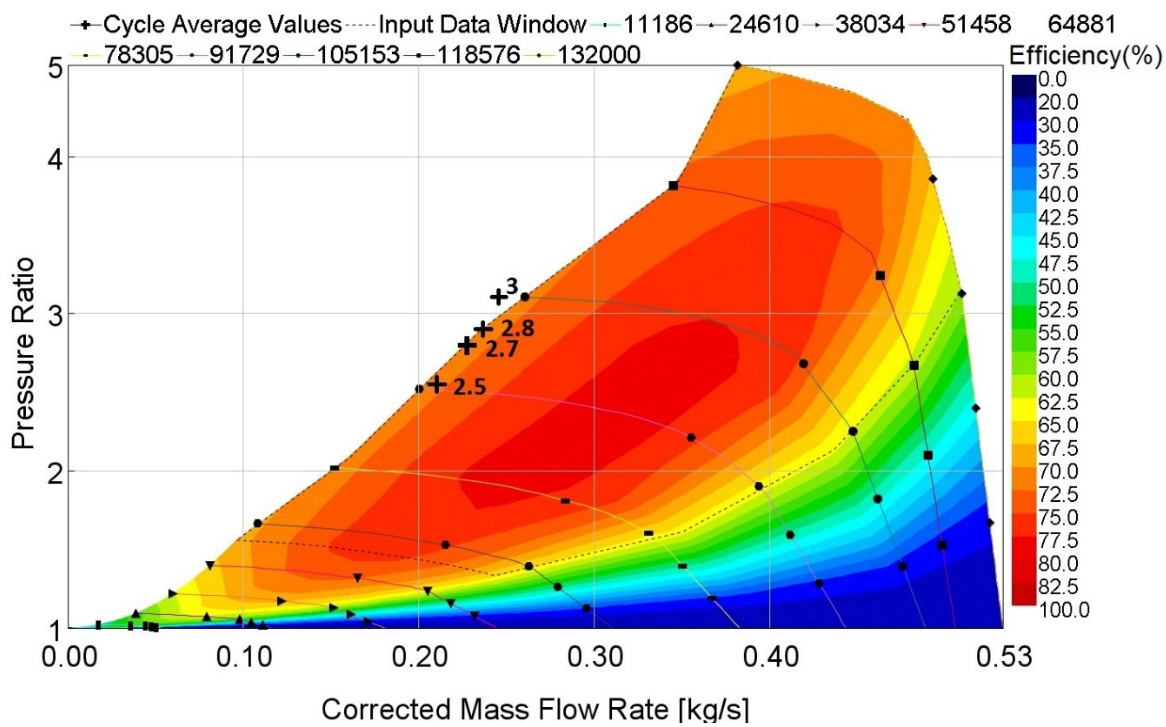

Figure 130 *SOI=10°bTDC: Compressor performance map for boost pressure sweep*

8 Conclusions and Future Work

A dual-fuel engine simulation model was developed and calibrated using the diesel engine experimental data and the calibrated dual-fuel model was used to simulate a low-speed and a high-speed case with different configurations of EGR, diesel injection pressures and diesel energy contribution percentages for diesel injection timings 0° and 10° bTDC.

The simulation results are analyzed and the conclusions from the engine simulations are summarized in this chapter.

- At 300 bar injection pressure, diesel energy ratio of 1% and EGR= 18%, Injection timing 10° bTDC requires lower Boost Pressure and has higher brake fuel conversion efficiency than injection at TDC.
- Thermal efficiency was observed to be higher for injection timing at 10° bTDC than at TDC for Engine speed 2138 RPM and most of the cases of Engine speed 1397 RPM.
- For EGR=18% and stoichiometric conditions in dual fuel mode, increase in boost pressure above 2.5 bar causes the engine to operate near the surge limit of the compressor.

8.1 Injection Timing 0° bTDC

- BMEP of 25.6 bar could be reached for 5% diesel energy contribution with brake fuel conversion efficiency of 41% at EGR = 0%
- Peak cylinder pressures were less than 200 bar for all operating conditions.
- Lower diesel injection pressures lead to low BMEP, thermal efficiency and NO_x emissions
- As the EGR level is increased, the variation of BMEP and NO_x emissions is not linear for lower engine speeds when compared to higher engine speeds, where the BMEP and NO_x emissions decrease in a linear fashion. For lower engine speeds, the reduction in BMEP and NO_x emissions between EGR levels 6 and 12% is more than the reduction between EGR levels 0-6% and 12-18%.

8.2 Injection Timing 10° bTDC

- 25 bar BMEP and brake fuel conversion efficiencies up to 41% was achieved for boost pressure of 2.9 bar at 300 bar injection pressure and EGR=18%.
- As the injection timing is advanced to 10° bTDC, the heat release rate occurs near TDC, which leads to high In-Cylinder pressures.
- High BMEP and thermal efficiencies occur at lower injection pressures.
- Effect of diesel injection pressures on BMEP and thermal efficiency is observed to be more significant at low EGR levels and lower engine speeds than at higher engine speeds.

Scope of Future Work

- Study on engine performance and stability of combustion at low loads under stoichiometric and ultra-lean conditions for charge dilution, different injection pressures and diesel energy ratios could be performed.
- Using Kinetics-Fit-Natural-Gas knock model introduced in GT-Power v2017, study on occurrence of knock and pre-ignition for high loads could be performed.
- Exhaust gas dynamics of the simulation models could be improved with Turbocharger map data over a wide range of variable rack geometries and engine operating speeds for the Holset turbocharger HE351VE.
- Dual-fuel experimental data from Cummins 2010 ISB 6.7 L engine, could be used to calibrate the duel fuel combustion parameters and validate the simulation results for dual-fuel engine model.
- Further analysis and testing on charging systems and limitation of the current compressor

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APPENDIX

The engine simulation results elaborated in chapter **7,** performed for the simulation matrix as shown in ***Figure 36*** in section **6.2** are summarized from ***Table 43*** to ***Table 58***.

Table 43 *Engine simulation data: Diesel injection timing 0° bTDC, Engine Speed 2138 RPM, EGR=0%*

Injection Pressure	Diesel Energy Ratio	Diesel Quantity	$IMEP_{net}$	BMEP	Torque	NOx		BSFC	PCP	Location of PCP	Brake Thermal Efficiency
bar	%	mg/stroke	bar	bar	N-m	ppm	g/kW-h	g/kW-h	bar	deg aTDC	%
1000	10	16.4	26.1	24.5	1300.4	3931.3	22.2	192.5	148	24	38.7
1000	5	8.2	25.6	24.1	1280.7	3210.7	18.5	195.3	129	25	37.9
1000	3	5.0	25.3	23.8	1265.7	2901.7	16.9	197.6	123	0	37.3
1000	1	1.7	24.9	23.4	1246.3	2627.9	15.7	201.7	123	0	36.5
600	10	16.4	25.5	24.0	1273.7	3588.4	20.8	197.3	128	26	37.7
600	5	8.3	25.1	23.6	1252.1	2937.6	17.3	200.3	123	0	36.9
600	3	5.0	24.6	23.1	1227.4	2695.5	16.2	203.6	123	0	36.2
600	1	1.7	24.3	22.8	1210.1	2517.3	15.4	206.5	123	0	35.6
300	10	16.5	23.9	22.4	1191.4	2895.2	18.0	211.5	123	0	35.2
300	5	8.3	23.6	22.1	1174.2	2448.7	15.4	213.9	123	0	34.6
300	3	5.0	23.3	21.8	1161.3	2293.0	14.6	216.0	123	0	34.2
300	1	1.7	23.1	21.6	1149.2	2203.6	14.2	218.1	123	0	33.7

Table 44 *Engine simulation data: Diesel injection timing 0° bTDC, Engine Speed 2138 RPM, EGR=6%*

Injection Pressure	Diesel Energy Ratio	Diesel Quantity	$IMEP_{net}$	BMEP	Torque	NOx		BSFC	PCP	Location of PCP	Brake Thermal Efficiency
bar	%	mg/stroke	bar	bar	N-m	ppm	g/kW-h	g/kW-h	bar	deg aTDC	%
1000	10	15.2	24.8	23.3	1236.6	2732.1	15.5	192.1	139	24	38.8
1000	5	7.6	24.4	22.9	1219.3	2027.5	11.7	194.9	123	0	37.9
1000	3	4.6	24.2	22.7	1204.2	1779.4	10.4	197.4	123	0	37.4
1000	1	1.5	23.7	22.2	1179.4	1575.8	9.4	201.7	123	0	36.5
600	10	15.3	24.3	22.8	1211.2	2225.0	12.9	196.9	124	0	37.8
600	5	7.7	23.9	22.4	1190.1	1702.3	10.1	200.2	123	0	37.0
600	3	4.6	23.7	22.2	1178.3	1551.5	9.3	202.1	123	0	36.5
600	1	1.5	23.3	21.8	1160.6	1450.2	8.8	205.2	123	0	35.9
300	10	15.4	23.0	21.5	1140.5	1500.6	9.3	210.2	123	0	35.4
300	5	7.7	22.7	21.2	1124.5	1231.9	7.7	212.7	123	0	34.8
300	3	4.6	22.4	20.9	1112.3	1149.2	7.3	214.9	123	0	34.3
300	1	1.6	22.2	20.7	1101.3	1131.2	7.2	216.9	122	0	33.9

Table 45 *Engine simulation data: Diesel injection timing 0° bTDC, Engine Speed 2138 RPM, EGR=12%*

Injection Pressure	**Diesel Energy Ratio**	**Diesel Quantity**	**IMEP$_{net}$**	**BMEP**	**Torque**	**NOx**		**BSFC**	**PCP**	**Location of PCP**	**Brake Thermal Efficiency**
bar	**%**	**mg/stroke**	**bar**	**bar**	**N-m**	**ppm**	**g/kW-h**	**g/kW-h**	**bar**	**deg aTDC**	**%**
1000	10	14.4	23.5	21.9	1164.9	1222.3	6.9	192.3	131	24	38.7
1000	5	7.2	23.1	21.6	1146.4	827.3	4.8	195.5	123	0	37.8
1000	3	4.3	22.8	21.3	1132.6	714.4	4.2	197.9	122	0	37.3
1000	1	1.5	22.3	20.9	1108.4	625.7	3.7	202.4	122	0	36.4
600	10	14.4	22.9	21.4	1138.0	894.1	5.2	197.6	123	0	37.7
600	5	7.2	22.5	21.0	1118.5	646.1	3.8	200.9	122	0	36.8
600	3	4.4	22.3	20.8	1108.0	587.9	3.5	202.7	122	0	36.4
600	1	1.5	22.0	20.5	1089.6	551.8	3.4	206.0	122	0	35.7
300	10	14.5	21.7	20.2	1071.3	530.0	3.3	210.9	123	0	35.3
300	5	7.3	21.3	19.9	1055.2	418.7	2.6	213.6	122	0	34.6
300	3	4.4	21.1	19.7	1044.4	391.2	2.5	215.6	122	0	34.2
300	1	1.5	20.9	19.5	1034.0	396.0	2.5	217.6	122	0	33.8

Table 46 *Engine simulation data: Diesel injection timing 0° bTDC, Engine Speed 2138 RPM, EGR=18%*

Injection Pressure	Diesel Energy Ratio	Diesel Quantity	$IMEP_{net}$	BMEP	Torque	NOx		BSFC	PCP	Location of PCP	Brake Thermal Efficiency
bar	**%**	**mg/stroke**	**bar**	**bar**	**N-m**	**ppm**	**g/kW-h**	**g/kW-h**	**bar**	**deg aTDC**	**%**
1000	10	13.4	21.8	20.3	1081.5	391.7	2.22	192.8	122	23	38.6
1000	5	6.8	21.4	19.9	1059.9	246.6	1.43	196.5	121	0	37.7
1000	3	4.1	21.2	19.7	1047.2	211.3	1.24	198.8	121	0	37.1
1000	1	1.4	20.7	19.2	1021.7	182.1	1.10	204.0	121	0	36.1
600	10	13.5	21.3	19.8	1051.9	267.2	1.56	198.7	122	0	37.5
600	5	6.8	20.9	19.4	1033.2	183.8	1.09	202.0	121	0	36.6
600	3	4.1	20.8	19.3	1023.9	168.3	1.01	203.7	121	0	36.2
600	1	1.4	20.4	18.9	1005.6	159.4	0.98	207.5	121	0	35.5
300	10	13.6	20.1	18.6	990.8	148.1	0.92	212.1	122	0	35.1
300	5	6.8	19.8	18.4	975.7	112.4	0.71	215.0	121	0	34.4
300	3	4.1	19.7	18.2	967.0	105.6	0.68	216.8	121	0	34.0
300	1	1.4	19.5	18.0	957.8	109.6	0.71	218.7	121	0	33.6

Table 47 *Engine simulation data: Diesel injection timing 0° bTDC, Engine Speed 1397 RPM, EGR=0%*

Injection Pressure	Diesel Energy Ratio	Diesel Quantity	$IMEP_{net}$	BMEP	Torque	NOx		BSFC	PCP	Location of PCP	Brake Thermal Efficiency
bar	**%**	**mg/stroke**	**bar**	**bar**	**N-m**	**ppm**	**g/kW-h**	**g/kW-h**	**bar**	**deg aTDC**	**%**
1000	10	16.3	27.1	25.6	1361.3	3974.6	21.4	182.8	188	19	40.8
1000	5	8.2	27.0	25.6	1362.2	3421.1	18.5	183.1	159	22	40.4
1000	3	5.0	26.9	25.5	1356.6	3141.0	17.1	184.3	146	23	40.0
1000	1	1.7	26.5	25.3	1342.2	2829.1	15.6	186.9	133	24	39.4
600	10	16.6	27.0	25.6	1360.8	3844.9	20.8	184.2	166	22	40.5
600	5	8.5	26.9	25.5	1356.6	3266.4	17.8	185.1	145	23	40.0
600	3	5.3	26.7	25.4	1349.9	2988.3	16.4	186.3	136	24	39.6
600	1	1.8	26.4	25.1	1335.5	2761.4	15.4	188.4	126	25	39.1
300	10	16.3	26.4	25.1	1336.3	3712.0	20.6	188.8	134	26	39.5
300	5	9.5	26.4	25.2	1337.6	2886.4	16.1	190.1	125	27	38.9
300	3	5.7	26.2	24.9	1324.5	2749.6	15.5	191.4	119	0	38.6
300	1	2.0	25.9	24.7	1311.1	2616.6	14.9	192.9	119	0	38.2

Table 48 *Engine simulation data: Diesel injection timing 0° bTDC, Engine Speed 1397 RPM, EGR=6%*

Injection Pressure	Diesel Energy Ratio	Diesel Quantity	$IMEP_{net}$	BMEP	Torque	NOx		BSFC	PCP	Location of PCP	Brake Thermal Efficiency
bar	**%**	**mg/stroke**	**bar**	**bar**	**N-m**	**ppm**	**g/kW-h**	**g/kW-h**	**bar**	**deg aTDC**	**%**
1000	10	15.5	26.1	24.6	1307.4	3526.6	18.8	181.4	179	20	41.0
1000	5	7.8	25.9	24.6	1306.0	2738.3	14.7	182.0	152	22	40.6
1000	3	4.7	25.7	24.4	1297.4	2392.3	12.9	183.3	140	23	40.2
1000	1	1.6	25.4	24.2	1283.7	2092.6	11.5	185.9	126	24	39.6
600	10	15.6	25.9	24.5	1304.5	3207.4	17.2	182.9	158	22	40.7
600	5	7.9	25.7	24.4	1295.1	2484.9	13.5	184.1	138	23	40.1
600	3	4.7	25.5	24.2	1288.3	2227.6	12.2	185.2	130	24	39.8
600	1	1.6	25.2	23.9	1272.0	1983.0	11.0	187.8	119	25	39.2
300	10	15.7	25.3	24.1	1278.8	2598.0	14.3	187.8	128	26	39.6
300	5	7.9	25.1	23.8	1266.2	2073.1	11.6	189.4	119	0	39.1
300	3	4.8	24.9	23.7	1257.3	1902.7	10.7	190.5	119	0	38.7
300	1	1.6	24.7	23.5	1248.3	1805.2	10.2	191.8	119	0	38.3

__Table 49__ Engine simulation data: Diesel injection timing 0° bTDC, Engine Speed 1397 RPM, EGR=12%

Injection Pressure	Diesel Energy Ratio	Diesel Quantity	$IMEP_{net}$	BMEP	Torque	NOx		BSFC	PCP	Location of PCP	Brake Thermal Efficiency
bar	**%**	**mg/stroke**	**bar**	**bar**	**N-m**	**ppm**	**g/kW-h**	**g/kW-h**	**bar**	**deg aTDC**	**%**
1000	10	14.5	23.5	21.9	1166.4	1217.1	6.89	192.3	131	24	38.7
1000	5	7.2	23.1	21.6	1146.5	826.8	4.77	195.5	123	0	37.8
1000	3	4.4	22.8	21.3	1132.7	713.7	4.17	197.9	122	0	37.3
1000	1	1.5	22.4	20.9	1108.5	625.2	3.74	202.3	122	0	36.4
600	10	14.6	22.9	21.4	1139.7	888.7	5.17	197.6	123	0	37.7
600	5	7.3	22.5	21.0	1118.7	645.2	3.82	200.9	122	0	36.8
600	3	4.4	22.3	20.8	1108.1	587.3	3.51	202.7	122	0	36.4
600	1	1.5	22.0	20.5	1089.6	551.8	3.36	206.0	122	0	35.7
300	10	14.5	21.7	20.2	1071.6	528.8	3.29	210.9	123	0	35.3
300	5	7.3	21.4	19.9	1055.4	417.9	2.63	213.6	122	0	34.6
300	3	4.4	21.1	19.7	1044.5	390.7	2.49	215.6	122	0	34.2
300	1	1.5	21.0	19.5	1034.2	395.5	2.54	217.6	122	0	33.8

Table 50 *Engine simulation data: Diesel injection timing 0° bTDC, Engine Speed 1397 RPM, EGR=18%*

Injection Pressure	Diesel Energy Ratio	Diesel Quantity	$IMEP_{net}$	BMEP	Torque	NOx		BSFC	PCP	Location of PCP	Brake Thermal Efficiency
bar	**%**	**mg/stroke**	**bar**	**bar**	**N-m**	**ppm**	**g/kW-h**	**g/kW-h**	**bar**	**deg aTDC**	**%**
1000	10	13.5	22.3	20.8	1106.7	414.3	2.30	188.9	123	23	39.4
1000	5	6.8	21.9	20.4	1085.4	260.0	1.48	192.4	121	0	38.4
1000	3	4.1	21.7	20.2	1072.4	221.6	1.27	194.7	121	0	37.9
1000	1	1.4	21.2	19.7	1045.1	182.6	1.08	199.6	121	0	36.8
600	10	13.6	21.8	20.3	1077.2	280.0	1.60	194.5	122	0	38.3
600	5	6.8	21.4	19.9	1058.2	191.5	1.12	197.7	121	0	37.4
600	3	4.1	21.2	19.7	1047.3	168.1	0.99	199.3	121	0	37.0
600	1	1.4	20.9	19.4	1032.0	161.6	0.97	202.7	121	0	36.3
300	10	15.0	20.8	19.3	1028.0	130.5	0.79	207.2	122	0	36.0
300	5	7.7	20.0	18.5	984.0	96.0	0.60	214.7	121	0	34.5
300	3	4.7	19.8	18.3	971.7	92.1	0.59	216.8	121	0	34.0
300	1	1.6	19.6	18.1	960.4	105.2	0.68	218.4	121	0	33.7

Table 51 *Engine simulation data: Diesel injection timing 10° bTDC, Engine Speed 2138 RPM, EGR=0%*

Injection Pressure	**Diesel Energy Ratio**	**Diesel Quantity**	**IMEP$_{net}$**	**BMEP**	**Torque**	**NOx**		**BSFC**	**PCP**	**Location of PCP**	**Brake Thermal Efficiency**
bar	**%**	**mg/stroke**	**bar**	**bar**	**N-m**	**ppm**	**g/kW-h**	**g/kW-h**	**bar**	**deg aTDC**	**%**
1000	10	15.9	26.7	24.7	1310.6	4309.9	23.6	185.9	251	12	40.1
1000	5	8.0	26.8	24.9	1323.6	3852.4	20.9	184.1	231	13	40.2
1000	3	4.8	26.9	25.0	1328.6	3640.4	19.7	183.6	222	14	40.2
1000	1	1.6	26.9	25.1	1332.6	3409.5	18.5	183.3	214	14	40.1
600	10	16.0	26.8	24.9	1323.3	4299.2	23.4	185.2	228	14	40.2
600	5	8.1	26.9	25.0	1330.7	3811.9	20.7	184.0	213	15	40.2
600	3	4.9	26.9	25.1	1332.9	3597.5	19.5	183.7	208	15	40.2
600	1	1.6	26.9	25.1	1334.2	3384.7	18.4	183.5	202	15	40.1
300	10	16.2	26.8	25.0	1331.1	4185.7	22.8	185.6	198	17	40.1
300	5	8.1	26.9	25.1	1333.3	3703.4	20.2	184.8	191	16	40.0
300	3	4.9	26.9	25.1	1333.8	3497.8	19.1	184.7	187	17	40.0
300	1	1.6	26.8	25.1	1333.1	3302.3	18.0	184.7	184	17	39.8

__Table 52__ Engine simulation data: Diesel injection timing 10° bTDC, Engine Speed 2138 RPM, EGR=6%

Injection Pressure	**Diesel Energy Ratio**	**Diesel Quantity**	**$IMEP_{net}$**	**BMEP**	**Torque**	**NOx**		**BSFC**	**PCP**	**Location of PCP**	**Brake Thermal Efficiency**
bar	**%**	**mg/stroke**	**bar**	**bar**	**N-m**	**ppm**	**g/kW-h**	**g/kW-h**	**bar**	**deg aTDC**	**%**
1000	10	15.2	25.6	23.6	1256.4	4113.5	22.3	184.6	240	13	40.4
1000	5	7.6	25.7	23.8	1267.7	3543.1	19.1	183.0	222	13	40.4
1000	3	4.6	25.8	23.9	1271.6	3294.0	17.8	182.6	214	14	40.4
1000	1	1.5	25.8	24.0	1274.9	3062.8	16.5	182.4	205	14	40.3
600	10	15.3	25.7	23.9	1267.7	3934.9	21.3	184.1	218	14	40.5
600	5	7.7	25.8	24.0	1273.5	3380.9	18.2	183.1	205	15	40.4
600	3	4.6	25.8	24.0	1275.3	3165.7	17.1	182.8	200	15	40.4
600	1	1.5	25.8	24.0	1276.5	2982.7	16.1	182.7	195	15	40.3
300	10	15.4	25.7	23.9	1272.8	3530.1	19.2	184.8	189	16	40.3
300	5	7.7	25.7	24.0	1274.6	3076.7	16.7	184.1	183	17	40.2
300	3	4.6	25.7	24.0	1274.3	2900.1	15.7	184.0	179	17	40.1
300	1	1.6	25.7	24.0	1273.7	2768.2	15.1	183.9	176	17	40.0

__Table 53__ Engine simulation data: Diesel injection timing 10° bTDC, Engine Speed 2138 RPM, EGR=12%

Injection Pressure	Diesel Energy Ratio	Diesel Quantity	$IMEP_{net}$	BMEP	Torque	NOx		BSFC	PCP	Location of PCP	Brake Thermal Efficiency
bar	**%**	**mg/stroke**	**bar**	**bar**	**N-m**	**ppm**	**g/kW-h**	**g/kW-h**	**bar**	**deg aTDC**	**%**
1000	10	14.4	24.4	22.5	1196.6	3289.4	17.7	183.5	228	13	40.6
1000	5	7.2	24.5	22.7	1204.6	2666.3	14.3	182.0	211	13	40.6
1000	3	4.3	24.5	22.7	1206.8	2444.1	13.1	181.7	204	14	40.6
1000	1	1.5	24.5	22.7	1208.8	2222.4	11.9	181.6	193	15	40.5
600	10	14.4	24.5	22.6	1203.8	2858.5	15.4	183.1	208	14	40.7
600	5	7.2	24.5	22.7	1207.8	2348.1	12.6	182.2	195	14	40.6
600	3	4.4	24.5	22.7	1208.4	2196.1	11.8	182.0	191	15	40.5
600	1	1.5	24.5	22.7	1208.6	2087.9	11.2	181.9	190	15	40.4
300	10	14.5	24.4	22.7	1204.5	2217.8	12.0	184.2	178	16	40.5
300	5	7.3	24.4	22.7	1205.4	1897.8	10.3	183.5	172	17	40.3
300	3	4.4	24.4	22.7	1204.8	1794.1	9.7	183.4	168	17	40.2
300	1	1.5	24.3	22.6	1203.8	1756.3	9.5	183.4	166	17	40.1

__Table 54__ Engine simulation data: Diesel injection timing 10° bTDC, Engine Speed 2138 RPM, EGR=18%

Injection Pressure	Diesel Energy Ratio	Diesel Quantity	$IMEP_{net}$	BMEP	Torque	NOx		BSFC	PCP	Location of PCP	Brake Thermal Efficiency
bar	%	mg/stroke	bar	bar	N-m	ppm	g/kW-h	g/kW-h	bar	deg aTDC	%
1000	10	13.4	23.0	21.1	1122.1	1852.3	10.0	183.0	215	13	40.7
1000	5	6.8	23.1	21.2	1129.1	1393.5	7.5	181.7	200	13	40.7
1000	3	4.1	23.1	21.3	1132.0	1269.4	6.8	181.3	194	14	40.7
1000	1	1.4	23.1	21.3	1134.7	1117.5	6.0	181.3	185	14	40.6
600	10	13.5	23.0	21.2	1129.1	1408.7	7.6	182.8	196	14	40.7
600	5	6.8	23.1	21.3	1134.3	1119.0	6.0	181.8	185	14	40.7
600	3	4.1	23.1	21.4	1135.1	1056.8	5.7	181.6	181	14	40.6
600	1	1.4	23.1	21.3	1133.8	991.3	5.3	181.7	175	15	40.5
300	10	13.6	22.9	21.3	1129.6	893.9	4.8	184.1	169	16	40.5
300	5	6.8	22.9	21.2	1128.5	743.1	4.0	183.6	163	16	40.3
300	3	4.1	22.9	21.2	1127.0	710.3	3.8	183.5	160	16	40.2
300	1	1.4	22.8	21.2	1125.0	718.0	3.9	183.5	159	16	40.1

Table 55 *Engine simulation data: Diesel injection timing 10° bTDC, Engine Speed 1397 RPM, EGR=0%*

Injection Pressure	**Diesel Energy Ratio**	**Diesel Quantity**	**$IMEP_{net}$**	**BMEP**	**Torque**	**NOx**		**BSFC**	**PCP**	**Location of PCP**	**Brake Thermal Efficiency**
bar	**%**	**mg/stroke**	**bar**	**bar**	**N-m**	**ppm**	**g/kW-h**	**g/kW-h**	**bar**	**deg aTDC**	**%**
1000	10	15.7	25.6	23.7	1260.3	3903.2	21.8	190.0	284	8	39.2
1000	5	7.9	26.2	24.4	1297.7	3487.1	19.0	185.3	259	10	39.9
1000	3	4.8	26.5	24.8	1315.7	3317.5	18.0	183.4	247	11	40.3
1000	1	1.6	26.8	25.1	1334.4	3120.1	16.8	181.6	234	12	40.5
600	10	15.8	26.1	24.3	1292.0	3923.7	21.5	186.8	265	10	39.9
600	5	8.0	26.6	24.8	1319.4	3496.2	18.9	183.4	245	12	40.4
600	3	4.8	26.7	25.0	1331.0	3316.6	17.8	182.1	237	12	40.5
600	1	1.6	26.9	25.2	1342.0	3117.0	16.7	181.0	230	13	40.6
300	10	16.0	26.8	25.1	1334.5	3938.4	21.2	183.2	235	13	40.7
300	5	8.1	27.0	25.3	1346.3	3490.1	18.7	181.5	223	14	40.8
300	3	4.9	27.1	25.5	1353.1	3303.1	17.6	180.7	217	14	40.8
300	1	1.6	27.1	25.5	1356.3	3103.5	16.5	180.2	214	14	40.8

Table 56 *Engine simulation data: Diesel injection timing 10° bTDC, Engine Speed 1397 RPM, EGR=6%*

Injection Pressure	**Diesel Energy Ratio**	**Diesel Quantity**	**$IMEP_{net}$**	**BMEP**	**Torque**	**NOx**		**BSFC**	**PCP**	**Location of PCP**	**Brake Thermal Efficiency**
bar	**%**	**mg/stroke**	**bar**	**bar**	**N-m**	**ppm**	**g/kW-h**	**g/kW-h**	**bar**	**deg aTDC**	**%**
1000	10	15.0	24.8	23.0	1220.6	3853.8	21.3	187.8	275	9	39.7
1000	5	7.6	25.3	23.6	1253.6	3371.4	18.2	183.4	250	11	40.3
1000	3	4.6	25.5	23.8	1264.0	3150.1	16.9	181.7	237	12	40.6
1000	1	1.5	25.8	24.1	1280.6	2919.4	15.5	180.0	223	12	40.9
600	10	15.1	25.3	23.5	1248.6	3836.4	20.8	184.8	256	11	40.3
600	5	7.6	25.5	23.8	1267.2	3312.8	17.7	181.7	236	12	40.7
600	3	4.6	25.7	24.0	1277.2	3105.1	16.6	180.5	228	12	40.9
600	1	1.5	25.8	24.2	1286.6	2898.7	15.4	179.5	221	13	41.0
300	10	15.2	25.7	24.1	1279.3	3695.3	19.7	181.6	226	13	41.0
300	5	7.7	25.9	24.3	1291.3	3197.5	16.9	179.8	214	14	41.1
300	3	4.6	26.0	24.4	1296.8	3008.3	15.9	179.2	208	14	41.2
300	1	1.5	26.0	24.5	1300.1	2831.3	15.0	178.7	206	14	41.2

Table 57 *Engine simulation data: Diesel injection timing 10° bTDC, Engine Speed 1397 RPM, EGR=12%*

Injection Pressure	Diesel Energy Ratio	Diesel Quantity	$IMEP_{net}$	BMEP	Torque	NOx		BSFC	PCP	Location of PCP	Brake Thermal Efficiency
bar	%	mg/stroke	bar	bar	N-m	ppm	g/kW-h	g/kW-h	bar	deg aTDC	%
1000	10	14.5	25.1	23.2	1231.7	3328.7	17.6	179.8	231	13	41.4
1000	5	7.2	25.2	23.3	1240.2	2709.8	14.3	178.4	213	13	41.5
1000	3	4.4	25.2	23.4	1242.8	2482.6	13.0	178.0	206	14	41.4
1000	1	1.5	25.2	23.4	1245.1	2261.9	11.9	177.9	197	14	41.3
600	10	14.6	25.2	23.3	1240.3	2899.7	15.3	179.5	209	14	41.5
600	5	7.3	25.2	23.4	1243.7	2400.6	12.6	178.5	197	14	41.4
600	3	4.4	25.2	23.4	1244.7	2240.7	11.8	178.3	192	15	41.4
600	1	1.5	25.2	23.4	1245.0	2115.2	11.1	178.3	186	15	41.3
300	10	14.5	25.1	23.3	1240.7	2299.2	12.2	180.3	182	16	41.3
300	5	7.3	25.1	23.4	1241.9	1958.5	10.4	179.7	176	16	41.2
300	3	4.4	25.1	23.4	1241.4	1847.7	9.8	179.6	172	16	41.1
300	1	1.5	25.0	23.3	1240.8	1802.2	9.6	179.6	170	17	41.0

Table 58 *Engine simulation data: Diesel injection timing 10° bTDC, Engine Speed 1397 RPM, EGR=18%*

Injection Pressure	**Diesel Energy Ratio**	**Diesel Quantity**	**$IMEP_{net}$**	**BMEP**	**Torque**	**NOx**		**BSFC**	**PCP**	**Location of PCP**	**Brake Thermal Efficiency**
bar	**%**	**mg/stroke**	**bar**	**bar**	**N-m**	**ppm**	**g/kW-h**	**g/kW-h**	**bar**	**deg aTDC**	**%**
1000	10	13.5	23.5	21.6	1149.4	1926.5	10.2	179.5	217	13	41.5
1000	5	6.8	23.6	21.7	1155.8	1464.4	7.7	178.2	201	13	41.5
1000	3	4.1	23.6	21.8	1156.9	1336.3	7.0	177.8	195	14	41.5
1000	1	1.4	23.5	21.8	1156.6	1181.1	6.2	177.9	185	14	41.4
600	10	13.6	23.5	21.7	1154.9	1475.4	7.8	179.3	197	14	41.6
600	5	6.8	23.5	21.8	1156.8	1169.8	6.2	178.4	186	14	41.5
600	3	4.1	23.5	21.8	1156.7	1106.0	5.8	178.2	182	14	41.4
600	1	1.4	23.5	21.7	1155.6	1037.8	5.5	178.3	175	15	41.3
300	10	15.0	23.5	21.8	1161.3	829.6	4.4	180.9	173	16	41.2
300	5	7.7	23.4	21.8	1157.5	713.5	3.8	180.2	167	16	41.1
300	3	4.7	23.4	21.7	1154.3	698.8	3.7	180.1	164	16	41.0
300	1	1.6	23.3	21.6	1149.9	754.5	4.0	180.0	162	16	40.9