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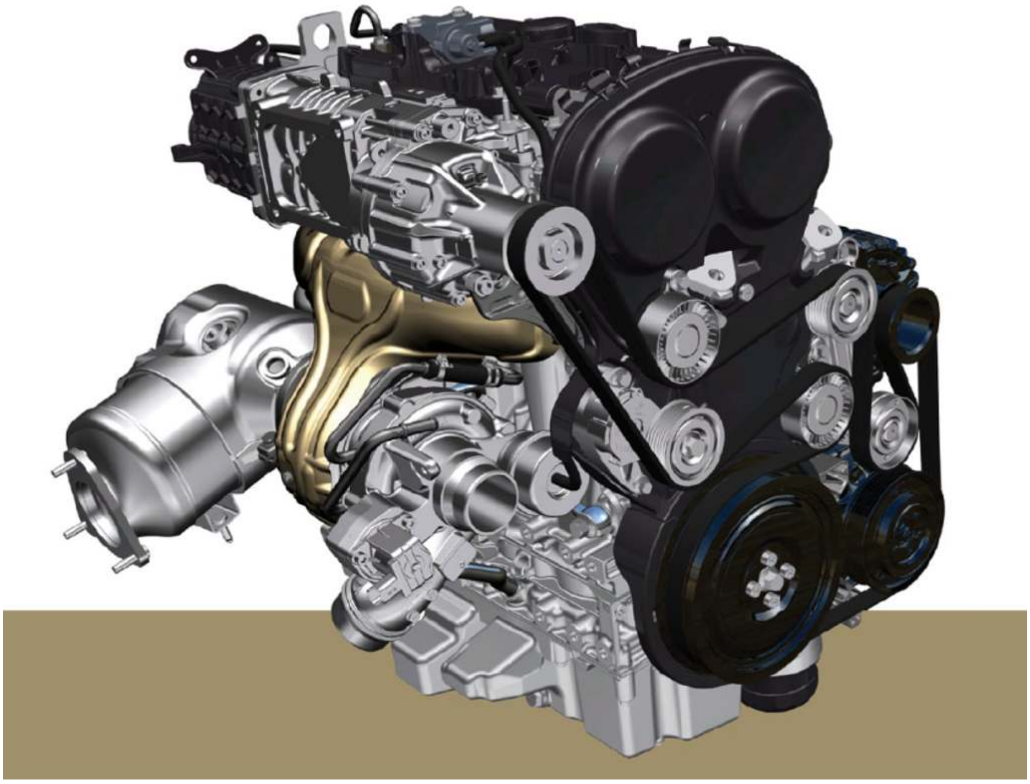

Comparison of Predictive Gasoline Combustion Models Using GT-Power

Master's thesis in Automotive Engineering

MANJUNATH NAGAPPA
ADITYA HEBBUR RAMESHBABU

Department of Applied Mechanics
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Gothenburg, Sweden 2016

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Abstract

With rising fuel prices and stringent emission regulations, OEMs are constantly working on improving the performance, fuel efficiency and reducing emissions of the internal combustion engines. Gas exchange simulation tools plays a very important role in the development of new concepts. Simulation tools helps to reduce the lead time significantly in developing new concepts.

The engine performance is dependent on the combustion efficiency which is analysed through burn rate. But measuring burn rate during the combustion is a difficult task. One of the major tool developers Gamma Technologies have developed a predictive combustion model which predicts the burn rate using the cylinder pressure. Calibration of the models is important for obtaining reliable results. Volvo Car Corporation (VCC) use FKFS combustion model for the development of the gasoline engines. Using a different combustion system requires recalibration of the combustion model to develop new concepts.

The aim of the thesis is to calibrate and validate the two predictive combustion models, SI Turb and FKFS in GT-Power and comparison of the models in order to assist future development of Gasoline engines at VCC. Predicting capabilities of the two calibrated combustion models is evaluated by its ability to predict the main operating parameters such as IMEP, Peak pressure, CA at 50% burn fraction and Ignition delay.

Calibration of the model is done against the test data obtained from a single cylinder test rig and the data was validated for different errors before using it for calibration. The predicting capabilities for FKFS combustion model is good for the complete engine map, whereas SITurb combustion model predicts well only for the calibrated range.

Keywords: Gasoline combustion, Burn rate, Predictive Combustion modelling, Three Pressure analysis, SITurb, FKFS, Calibration, Validation, GT-Power.

Acknowledgements

The thesis is carried out with a focus on calibrating the two predictive combustion models in GT-power for a direct injection gasoline engine. The work is carried out at Volvo Car Corporation from January 2016 to June 2016.

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MANJUNATH NAGAPPA, ADITYA HEBBUR RAMESHBABU

Gothenburg, September 2016

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List of Abbreviations

BDC	Bottom Dead Center
IMEP	Indicated Mean Effective Pressure
PMEP	Mean Effective Pressure of exhaust stroke minus the intake stroke
BSFC	Brake Specific Fuel Consumption
CA	Crank Angle
CAD	Crank Angle Degrees
CO	Carbon monoxide
CPOA	Cylinder Pressure Only Analysis
GDI	Gasoline Direct Injection
ECU	Electronic Control Unit
EGR	Exhaust Gas Recirculation
HC	HydroCarbon
IVC	Intake Valve Closing
EVC	Exhaust Valve Closing
WOT	Wide open throttle
IVO	Intake Valve Opening
LHV	Lower Heating Value
NOx	NO (Nitric Oxide) and/or NO_2 (Nitrogen Dioxide)
OEM	Original Equipment Manufacturer
SOC	Start of Combustion
SOI	Start of Injection
TDC	Top Dead Center
TPA	Three Pressure Analysis
PFI	Port fuel Injection
VVT	Variable Valve Timing
MOP	Maximum Opening Position
VCC	Volvo Car Corporation
FKFS	Forschungsin-stitut für Kraftfahrwesen und Fahrzeugmotoren Stuttgart
OEM	Original equipment manufacturer
SITurb	Spark Ignited Turbulent Flame model
CAE	Computer Aided Engineering

1
INTRODUCTION

This chapter gives a brief introduction to the background that motivated the project to take form. It also contains the goal as well as objectives, scope and boundaries of the project.

1.1 Background

With rising fuel prices and stricter emission regulations, OEMs are constantly working on improving the performance, fuel efficiency and reducing emissions of the internal combustion engines. Gas exchange simulation tools plays a very important role in the development of new concepts. Simulation tools helps to reduce the lead time significantly in developing new concepts.

The engine performance is dependent on the combustion efficiency which is analysed through burn rate. But measuring the burn rate during the combustion is a difficult task. One of the major tool developers Gamma Technologies have developed a predictive combustion model which predicts the burn rate using the cylinder pressure. Calibration of the models is important for obtaining reliable results. VCC use FKFS combustion model for the development of the gasoline engines. Using a new combustion system requires recalibration of the combustion model to develop new concepts.

1.2 Aim

The aim of this thesis is to calibrate and validate two predictive combustion models in GT-Power using SI Turb and FKFS combustion models and comparison of the models in order to assist future development of gasoline engines at VCC.

1.3 Objectives

The thesis will focus on:

- Literature survey on different combustion models for a direct injection gasoline engine.

- Data acquisition from the engine test rig for different operating points.
- Modelling of a single cylinder direct injection gasoline engine.
- Pressure trace analysis using two different combustion models.
- Calibration of the two combustion models against operating points.
- Validation and comparing the results of two models to investigate the predicting capabilities.

1.4 Scope

A GT – Power model of a single cylinder direct injection gasoline engine has to be modelled using the dimensions obtained from the test rig. Data acquisition is done for different operating points at the test rig. The GT – Power model is used to perform the pressure trace analysis for two different types of combustion models. The two combustion models are then calibrated and validated separately against the test data. Results will be based on the simulations in GT – Power.

The two combustion models used is developed by Gamma Technologies and FKFS. VCC use FKFS combustion model for the development of gasoline engines. The thesis is limited only to evaluate the two combustion models for a single cylinder direct injection gasoline engine.

1.5 Boundaries

- The GT – Power model is specific to a single combustion system which includes intake and exhaust ports and the cylinder.
- The model created is calibrated for a high performance tuned single cylinder engine.
- Selection of the engine operating points is based on the availability of time to run the engine in the test rig.
- Prediction of the knocking is not performed due time constraint and it was out of the thesis scope.

2

THEORY

The first part of this section introduces to the gasoline engine basics and the variable valve timing technology. The second part provides a information about the combustion modelling in GT – Power. The third part explains on how to perform the validation of the test data obtained from a single cylinder test rig. The final part gives information about the burn rate analysis using GT – Power.

2.1 Gasoline Engine Basics

The gasoline engine is a spark ignition (SI) internal combustion engine which operates based on the Otto cycle. In the SI engine, a spark is produced by a spark plug which ignites the charge inside the cylinder. The combustion phenomena in the SI engine is homogeneous unlike the CI engine which is heterogeneous. There are 2 types of commonly used gasoline engines, Port Fuel Injection (PFI) and Gasoline Direct Injection (GDI) engine. Currently most of the OEM's are manufacturing GDI engines due to its advantages over PFI engines. The limitation of the SI engine is that a higher compression ratio cannot be achieved due to knocking tendency.

2.1.1 Direct injection Gasoline Engine

GDI is a type of fuel injection method in an SI engine in which fuel is injected directly into the combustion chamber. Intake air is drawn into the cylinder through the intake port, and the fuel is injected at very high pressure and the air/fuel mixture is formed inside the cylinder. A high pressure fuel helps in creating finer fuel particles leading to a better combustion.

The main advantage of the GDI engine is improved swirl/tumble effect creation and better cooling inside the cylinder which helps to achieve higher compression ratio. This leads to a better efficiency, thereby reducing fuel consumption [19]. GDI technology is very advantageous for downsizing and turbocharging because of which all the OEM's are into this technology. In a GDI system, emission levels can be controlled accurately by an ECU (Engine Control Unit) based on the amount of fuel injected into the cylinder, hence giving a better control of the emission levels[19].

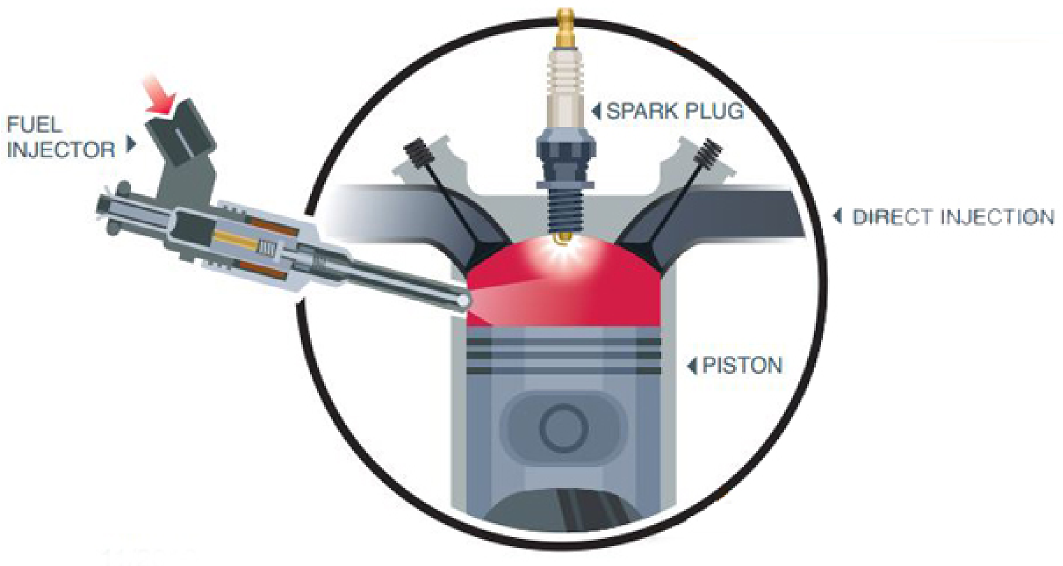

Figure 2.1: Gasoline Direct Injection [13]

2.1.2 Combustion in Gasoline Engine

The flame propagation is one of the main factor which influences the combustion in the gasoline engines. In the gasoline engines, the fuel and air are mixed inside the port or inside the cylinder based on the type of engine. The combustion process in a gasoline engine can be imagined in two stages, growth and development of the flame and flame propagation throughout the cylinder [20].

The flame development in the cylinder is initiated through inducing a spark with a spark plug. The flame propagation inside the cylinder is affected by the following parameters,

- Fuel-air ratio – The velocity of the flame depends on the quality of the charge. If the mixture is lean or rich, the velocity of flame reduces compared to the stoichiometric mixture.
- Compression ratio – The combustion process is accelerated more with the increase in compression ratio because of an increase in the temperature.
- Intake air temperature and pressure – An increase in the intake temperature and pressure increase the flame speed.

- Turbulence – The turbulent motion of the mixture helps in better mixing of fuel and air. The quality of mixture influences the velocity of the flame front.

- Engine speed – The Turbulent flame speed increases almost linearly with the engine speed.

Apart from the above explained parameters, spark timing, combustion chamber deposits and a few other parameters influence the combustion process. If compression ratio, AFR ratio, spark timing and Intake air temperature parameters are not controlled in the engine it can lead to abnormal combustion [20]. The two types of abnormal combustion identified in the gasoline engine are knock and surface ignition.

Knocking is a result of auto ignition of a portion of air-fuel and residual gas mixture ahead of the advancing flame. As the flame propagates through the combustion chamber, the unburned mixture ahead of the flame gets compressed leading to increase in pressure and temperature. This causes auto ignition of the end gas releasing a large part of chemical energy leading to high pressure frequency oscillations which produces sharp metallic noise called knock. Figure 2.2 shows the knocking effect against different spark timing. In the long run, knock effects are not desirable as it ruins the piston and reduces the engine life [1].

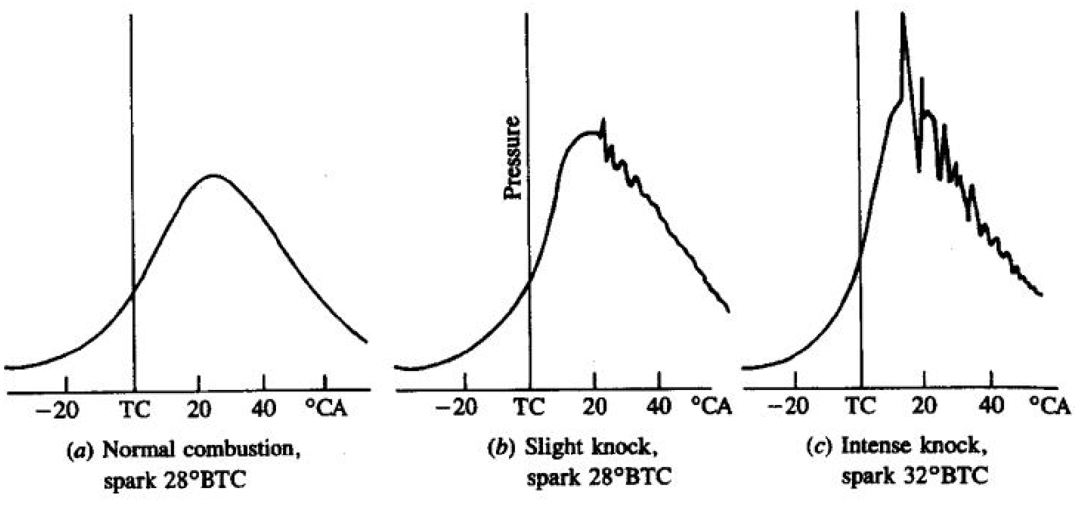

Figure 2.2: Knocking effect at different spark timings [1]

The other important abnormal combustion is surface ignition. It is the ignition of the fuel-air charge by overheated valves, spark plugs, glowing combustion chamber deposits or any other hot spot in the combustion chamber. Surface ignition may result in knock [1].

The following parameters characterize the energy release aspects of a combustion:

- **Flame development angle** ($\Delta\theta_d$) – The crank angle interval between the spark discharge and time during which major amount of fuel chemical energy has been released.

- **Rapid burning angle** ($\Delta\theta_b$) – The crank angle interval between the end of the flame development stage and the end of the flame propagation process.

- **Overall burning angle** ($\Delta\theta_o$) – The duration of the overall burning process. It is the sum of the flame development angle and the rapid burning angle.

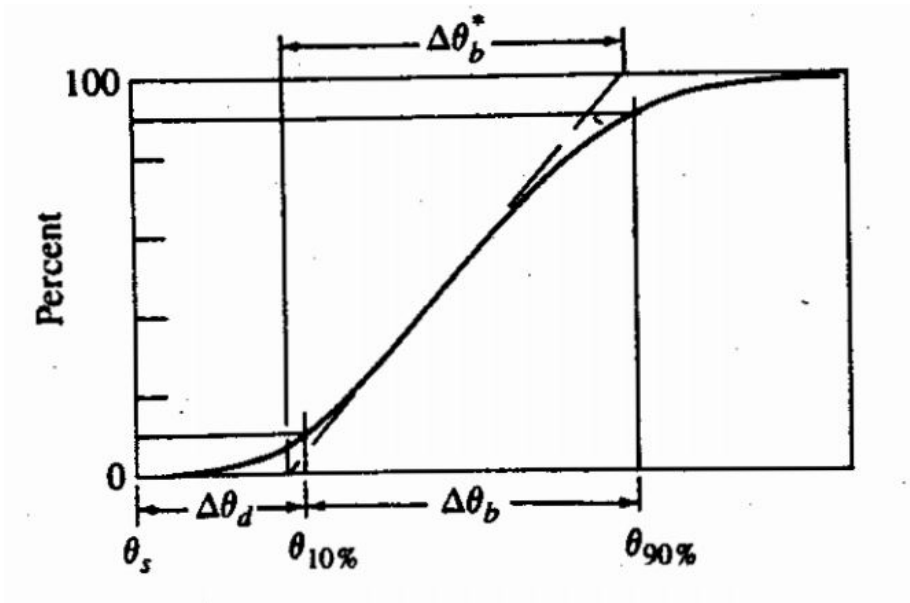

Figure 2.3: Characterization of energy release curve [1]

The gasoline engines usually operates in three combustion modes, lean burn, stoichiometric and full power output [14].

- Lean burn mode – Light load or reduced speed operating points use this mode of operation. The fuel is injected at the latter stage of the compression stroke.

- Stoichiometric mode – Moderate load conditions use this mode of operation. The fuel is injected during the intake stroke and this leads to a homogeneous fuel-air mixture. The stoichiometric operation mode results in a better combustion and reduced emissions.

- Full power mode – This combustion mode is used during the full load and accelerating. The fuel is injected during the intake stroke and the fuel-air mixture is slightly richer than stoichiometric mixture.

2.1.3 Variable Valve Timing (VVT)

The valve is one of the primary component of an engine. The efficiency and performance of an engine can be controlled by the valve timing by regulating the flow of air/fuel into the combustion chamber. Each engine has its own set of values for valve timing for an optimum performance. The valve timing plays a very important role in controlling the quality of the emissions. The basic method of controlling valve timing is by a simple fixed mechanical connection between the crankshaft and cam gear. The major disadvantage of using this method is that the valve opening timing can be controlled with respect to the speed and load sometimes but sometimes it can't be controlled. There is always a compromise between low torque and high power.

By using VVT, the engines performance and efficiency can be increased at every load and speed. Large overlap between the EVC and IVO can be an advantage for the high performance engines. By the use of VVT, the pumping losses of the engine can be reduced by 30% at part load condition [21]

Intake Valve Effect

Better combustion and performance can be achieved when the intake valve is closed before BDC for high loads with wide open throttle (WOT). Whereas, closing of the intake valve late proves advantageous when running at a part load at high engine speeds, but at lower speeds the pumping losses increases considerably [21].

Exhaust Valve Effect

Varying the exhaust valve timing affects the exhaust gas temperatures and hence the emissions, in particular hydrocarbons. Early opening of the exhaust valve leads to incomplete work extraction from the combustion gases but helps in better scavenging of the exhaust gases [21].

2.1.4 Valve Overlap effects

Valve overlap is beneficial only when pressure waves are considered for intake and exhaust system for a specific load and speed. Ideally, the valve overlap gives better scavenging ability. The overlap duration is dependent of the speed and load.

At high loads and speeds, increased overlap helps the pressure waves in the exhaust manifold supports intake of a fresh charge. Very large overlap leads to a higher emissions as the fresh charge escapes into the exhaust manifold.

Large overlap can also lead to a higher EGR, which is useful at the part load conditions, this can also lead to a instability in the combustion process [15].

The earlier the valve overlap, the larger is the flow of exhaust gases in to the intake manifold. The later the valve overlap, the higher is the flow of exhaust gases into the cylinder. Proper timing of the overlap leads to a better volumetric efficiency, which means better mixing and hence results in higher torque output. There could be a possibility of a secondary wave front developed before the spark, this could be a case of improper valve overlap [15].

2.2 Combustion Modelling

The simulation models are of great advantage and a substitute when real-life testing is expensive, time consuming, difficult or not feasible. The models are very beneficial while performing iterative tests and investigating the influence of different parameters on the results. The main advantages with a simulation models are that once the right model is created it will be efficient enough to use it for simulations over actual testing in the rig as it requires calibration, set up time etc.and also changing the design or any parameter would be very easy compared to doing it in the test rig.

The drawbacks of the models would be the time consumption for setting up the model and also the simulation time. The models might not yield the same results as the test rig results due to the several factors but calibrating and validating the models against the test rig data would yield close enough results which can be relied upon.

GT-Power provides a different combustion modeling alternatives and depending on the intended use of the completed simulation it is possible to choose between predictive and non-predictive models.

2.2.1 Predicting and Non-Predictive Combustion Models

In GT-Power for a given engine type and simulation requirement different combustion modelling alternatives are available namely Predictive and Non Predicting Combustion models. It is important to understand when each model is appropriate.

A non-predictive combustion model the burn rate is imposed initially as a function of the crank angle. The simulation follows this prescribed burn rate assuming sufficient fuel is available regardless of the cylinder conditions. Therefore, the residual fraction and the injection timing doesn’t have any influence on the burn rate during the simulation. Hence non-predictive combustion model can be used for studying the parameters which has a very small influence on the burn rate [10]. When studying the variable which is greatly influenced by the burn rate non-predictive combustion models are not desired and these situations can be handled by the predictive combustion models.

In the predictive combustion models burn rate changes accordingly to any change in a parameter which has influence on it. The burn rate is calculated based on the measured input data from the test rig hence the predictive combustion model yields more accurate results. But the computation time of these models is significantly higher because of the added complexity in calculations and also predictive models requires calibration against measured data to provide accurate results which is time consuming [10]. Hence implementing the right model based on the requirement is very important.

2.2.2 Predictive Combustion models

GT-Power has developed four different predictive combustion models based on the engine type namely,

- Spark-Ignition Turbulent Flame Model (SITurb)
- Direct-Injection Diesel Multi-Pulse Model (DI-Pulse)
- Direct-Injection Diesel Jet Model (DI-Jet)
- Homogeneous Charge Compression Ignition Model (HCCI)

Out of the above four predictive models, Spark-Ignition Turbulent Flame Model (SITurb) is used for simulation of a gasoline engine. This model is as fast as non-predictive models but with little more longer run-time for typical engine models [22].

2.2.3 Spark-Ignition Turbulent Flame Model (SITurb)

The SITurb combustion model predicts the burn rate for homogeneous charge, spark ignited engines. This combustion model works based on a two zone, entrainment and burn up model [10]. The prediction in this model considers the cylinder geometry, spark locations and timing, air motion and fuel properties. The mass entrainment rate into the flame front and the burn rate is governed by the following equations [10]:

$$\frac{dM_e}{dt} = \rho_e A_e (S_T + S_L) \tag{2.1}$$

$$\frac{dM_b}{dt} = \frac{(M_e - M_b)}{\tau} \tag{2.2}$$

$$\tau = \frac{\lambda}{S_L} \tag{2.3}$$

where,
M_e = Entrained mass of unburned mixture
t = Time
ρ_u = Unburned density
A_e = Entrainment surface area at the edge of flame front
S_T = Turbulent flame speed

S_L = Laminar flame speed
M_b = Burned mass
τ = Time constant
λ = Taylor microscale length

Equation 2.1 states that the unburned mixture is entrained into the flame front through the flame area at a rate proportional to the sum of turbulent and laminar flame speeds. Equation 2.2 shows that the burn rate is proportional to the amount of unburned mixture behind the flame front, divided by a time constant which is calculated by dividing the Taylor Microscale by the laminar flame speed as shown in equation 2.3.

The turbulence intensity and the length scale can be obtained in results by defining the flow reference object in the model. The model can be calibrated against the measured data by adjusting the effects of the turbulence intensity and the length scale on calculation of the turbulent flame speed and Taylor micro-scale length using the multipliers.The combustion chamber geometry should be defined to capture the turbulent effect for proper calibration. Figure 2.5 shows the different zones of flame and equations used to define it.

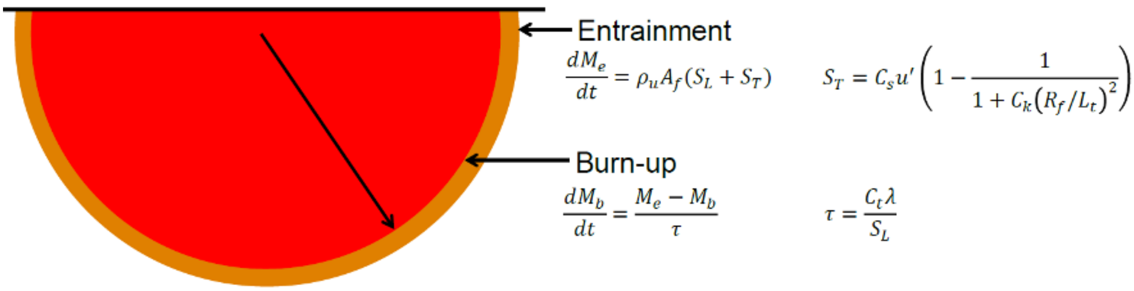

Figure 2.4: SITurb Model [22]

2.2.4 FKFS model

The modelling of FKFS combustion model remains similar to that of the SI Turb model in GT-Power. The only difference between the two are the combustion object defined in the model. The calculation of FKFS cylinder model starts from IVC by considering the initial parameters such as pressure, temperature and gas compositions from GT-Power. The calculations of the pressure, temperature and gas compositions are completed and handed over to GT-Power when EVO.

As the calculations of FKFS Cylinder object start at IVC, an accurate value must be chosen when defining the start of cycle. If the Crank-Angle value for the start of cycle chosen is later than IVC, then the calculations are performed from IVC to the chosen value is handled by the GT-Power and then the calculated conditions are used by FKFS for further calculations. FKFS follows two zone combustion method.

Two Zone Combustion

The combustion process is divided in to three parts originally,

- Zone with unburned fuel with a homogeneous mixture of air, fuel and residual gases
- A zone with burned exhaust gas
- Flame front dividing the above two zones

For the process of combustion in the FKFS cylinder and calculation of the burn rate, the model divides the combustion process into two regions, the unburned zone and the burned zone. The unburned zone is the zone where the fuel and air is mixed to form a homogeneous mixture. The burned zone is where the actual combustion occurs.

There is also a penetration zone, the combustion process always follows a turbulent flow field. In the region of the flame front the unburned mixture mixes with the burned mixture and the unburned mixture is consumed by the flame.

From turbulence theory, in a turbulent flow field, magnitude of the turbulent scale varies. This maximum region of the turbulence is defined by the integral length scale. The average of the turbulence region size l_T is defined by the Taylor length. The turbulence regions are separated by the vortex sheets with thickness I_k Kolmogorov length (it refers to the order of magnitude of smallest turbulence model in flow field).

The penetration velocity u_E is the sum of the turbulence fluctuation velocity u_{turb} and the laminar burning velocity s_L which is represented by Equation 2.4,

$$u_E = u_{turb} + S_L \tag{2.4}$$

The change in penetration mass flow of unburned mixture into flame zone is signified by Eq 2.5 as shown below,

$$\frac{dm_E}{dt} = \rho_{uv} * A_{fl} * u_E \tag{2.5}$$

where,

A_{fl}-Flame front surface
ρ_{uv}-Density in the unburned zone

The heat release rate and the changing of zone from the unburned to the burned zone is as follows,

$$\frac{dm_v}{dt} = -\frac{dm_v}{dt} = \frac{dQ_B}{d\varphi} * \frac{1}{H_u} * \frac{m_F}{\tau_1} \tag{2.6}$$

where,

m_f-mass of flame zone
T_1-characteristic combustion time

The characteristic burning time is the ratio of the Taylor length and the laminar burning velocity. The Taylor length is a function of the turbulence fluctuation velocity and the turbulent kinematic viscosity,

$$l_T = fac_{taylor} * \frac{v_T * l}{u_{Turb}} \tag{2.7}$$

With high residual gas fraction, the fuel mass is not completely converted to the energy close to the end of the combustion as the modelled Taylor length is really high and the laminar flow velocity is moderately small. With respect to the real conditions, in the simulation the the above mentioned phenomena occurred slightly early and was adjusted to the value at $\phi_{TurbStart}$ and starts to take effect with a high residual gas volumes (about 25-30%).

Residual Gas Fractions

FKFS definition of burned gas is that mass which represents the combustion products (CO_2, H_2O and CO, H_2). This shows the relevance to reality, burned mass fraction is the mass of oxygen that is consumed along with the burned products formed because of the oxygen. Whereas, if there is any mass that does not react with the oxygen to form combustion products then this mass is defined as the unburned mass fraction.

2.2.5 QDM Charge Motion: (C_u)

The use of k-epsilon equation in relation to cylinder turbulence was setup for quasi dimensional requirements. Bargande put forth the use of IVC as the start value for the empirical calculations of ordinary differential equations. This considers the volumetric efficiency, intake valve lifts and the cylinder conditions such as intake pressure and temperature.

The response of an engine at the high pressure region is very sensitive to change in the turbulence. The turbulence intensity is based on the kernal flame development and ignition. It's effects compared to the laminar flame speed has a wider range of consequences in affecting the engine emissions and the efficiency.

The charge motion multiplier is used for defining the turbulence when the tumble or swirl motion is used to create the homogeneous mixture. For this modelling of turbulence to function efficiently the tumble/swirl coefficients are to be provided in the valves along with discharge coefficient for the respective port geometry. Tuning of the Turbulent fluctuating velocity can be done by varying this parameter to obtain the correct combination of the turbulence and the heat release. Higher the value of C_u, shorter would be the combustion duration and vice versa

2.3 One-Cylinder Test Rig

The data obtained from the test cell needs to be validated before using it in any simulations, particularly when performing the heat release analysis; since the raw data obtained have common errors such as encoder phasing, pegging error and thermal shock [3].

The accuracy of the combustion model calibration is greatly influenced by the quality of the test data. Hence the errors in the test data must be verified and this is carried out through a data analysis tool known as Concerto which is developed by AVL.

AVL Concerto is a platform for data processing which is used for multiple purposes such as evaluation of test bed data, combustion analysis data, exhaust gas analysis etc [2]. User defined macros can be defined in the software to visualize the data. The data obtained from the test rig was validated through this software using user defined scripts.

An introduction about the errors found in the measured pressure data are explained below,

2.3.1 Encoder Phasing

Encoder phasing is the offset in the signals between the cylinder pressure measuring sensor and the crank angle encoder. The crank angle deviation influences the thermodynamic analysis. The basic influence of the crank angle error is on the indicated mean effective pressure (IMEP) and is as shown in Figure 2.6.

If the piston TDC position is too early, then the cylinder pressure trace is shifted to the right side i.e after the TDC position which leads to low pressure development reading when piston moves up and excess pressure when the piston moves down. This results in an extended post combustion phase and increase in heat release rate. The above scenario is the opposite when the TDC is too late. The effects of angle offset can be be compared in terms of IMEP, which becomes greater when the TDC is too early and vice-versa [5]. An error of 0.6 degrees or less is acceptable in order to get accurate results [4].

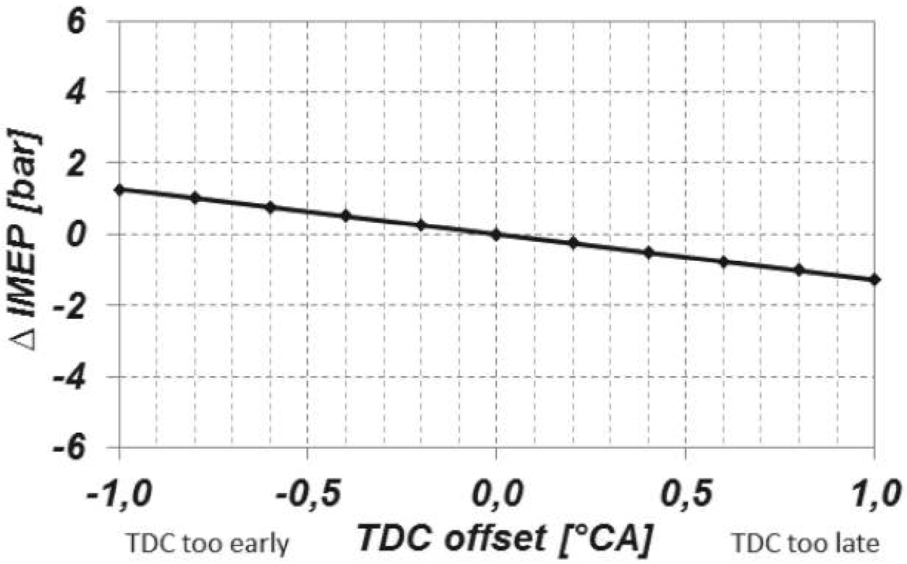

Figure 2.5: Encoder phasing influence on IMEP [5]

There are two commonly used methods for determining the encoder phasing. The first one is using a TDC sensor, which is mounted on the cylinder head through an existing feature such as spark plug or glow plug hole. When the engine is motored and as the piston moves towards sensor tip, the capacitive field strength increases and produces a voltage signal proportional to the distance between the probe and the piston. A key advantage of using this technique is that the accuracy is not influenced by the measurement of cylinder pressure, transducer gain, thermal shock and pegging[3]. Even though this method gives accurate result it is not commonly used because the sensor and set up is expensive.

The second method used for determining the TDC position is through using the motored engine pressure trace. In this method, the crank position sensor is calibrated against the measured motored cylinder pressure of the engine by finding the crank position at cylinder peak pressure [6]. Cylinder pressure obtained by a motored cycle has crank angle phasing information. The maximum pressure inside the cylinder is close to TDC, hence the calibration of the crank angle sensor is easier. The maximum cylinder pressure and the TDC position cannot be accurate in the same position because of losses due to heat transfer, blow-by and crevice effects hence the maximum pressure occur bit earlier [4].

2.3.2 Pegging error

A piezoelectric transducer are commonly used for measuring the pressure in the engine cylinder, intake and exhaust. The transducers produce a voltage relative to a change in the pressure level and the charge is converted to voltage and amplified. To obtain the measured pressure, the overall gain and the signal offset should be known. The gain will be constant for a given transducer. The signal offset keeps changing for different cycles and this should be calculated which is referred to as signal pegging. There are many methods used for pegging the pressure but there are two commonly used method [23]. Incorrect pressure pegging affects the parameters such as burned mass fraction, heat release rate, polytropic coefficient etc. Hence, it is very important to peg the pressure through a correct method.

The first commonly used method is by fitting the polytropic compression coefficient. The polytropic coefficient affects the instantaneous heat release rate and the specific heat ratio of the surrounding heat exchange. The polytropic coefficient for gasoline is 1.32 – 1.40 [9]. The polytropic coefficient for the pegged pressure should lie within the range for right pressure measurement [1].

The second method used for pegging is by referencing the absolute pressure measured by the inlet pressure sensor to the output of the cylinder pressure sensor measurement during the intake valve opening time of the engine cycle. The offset between these pressures should be around or smaller than 200 millibar [22].

2.3.3 Thermal Shock

Thermal shock is one of the major issues with piezoelectric transducers. The transducers are exposed to combustion gases where there are large temperature fluctuations. Due to this, the diaphragm inside the transducer is expanded and contracted. This causes a difference in measuring the pressure, hence leading to an error in parameters derived from this pressure and IMEP is affected by about 10%. Thermal shock has been found to be most significant at low engine speeds, high loads and advanced ignition timings [7].

Kistler Company[5] who are one of the pioneers in manufacturing these transducers has developed an equation to compensate thermal shock as show in Equation 2.4,

$$IMEP_{corr} = IMEP_{meas} + F * x * P_{max} + Offset \tag{2.8}$$

where

$$F = 0.000133(\frac{N}{1000})^2 - 0.002(\frac{N}{1000}) + 0.0105 \tag{2.9}$$

$$Offset = 0.12(\frac{N}{1000}) \tag{2.10}$$

One way to evaluate the thermal shock is to analyse the exhaust pressure after the combustion. The average exhaust absolute pressure (AEAP) is checked between 240 to 320 degree after firing TDC and the standard deviation of the AEAP should be within 4 kPa [3].

2.3.4 Data Filtering

Data filtering is a very important step due to the following reasons,

- Filtering of data is a simple process of eliminating the unwanted/very sensitive information from the raw data.

- Filtering of the data obtained from test rig is very important, as the raw data might have discrepancies and filtering helps in obtaining an accurate and precise results when simulations are performed.

- Two types of filter methodology is used to “clean” the data for intake, exhaust and cylinder pressure.

2.3.4.1 Types of Filters

Filters are numerous, so choosing a particular filter for the specific data is a challenge in itself. In this project, an IEEE filter was used for the intake and the exhaust pressures and a Savitzky–Golay filter for the cylinder pressure.

IEEE Filter

The below mentioned filters are integrated in GT-Power and the order of the filter can be used according to the smoothing requirements. One of the following can be chosen:

- IEEE-10 – This is a low smoothing filter which indicates that a 10th order IEEE low pass filters is used.

- IEEE-24 – This filter’s smoothing is better than that of IEEE-10 and it indicates that a 24th order IEEE low pass filter is used.

- IEEE-54 – The highest smoothing ordered filter with 54th order low pass filter smoothing and provides more smoothing that IEEE-24.

Savitzky–Golay filter

Savitzky-Golay is a digital smoothening technique that is generally used to smooth out a noisy signal whose frequency band is large. With respect to this, a Savitzky-Golay filter has better filtering abilities when compared to a averaging FIR (Finite Impulse Response) filters; FIR filters tends to delete the unwanted information that is out of the required bandwidth of frequency. The Savitzky-Golay filter retains all the information of the signal, including the high frequency content [18].

Savitzky-Golay calculates using a process called convolution. It fits the successive sub-sets of neighbouring data points with a low degree polynomial by the method of linear least-square. With an equally spread out data points, an analytical solution can be obtained in the form of single set convolution coefficients by a least-square method, this can be applied to all the data subsets to provide an estimate of a smooth signal. This is an optimum filter that minimizes the least-square error in matching the polynomial to follow the noisy signal [18].

2.4 Engine Burn rate analysis

The engine burn rate is a very important factor influencing the combustion phenomena inside the cylinder. Burn rate is the instantaneous rate of fuel consumption during the combustion process inside the cylinder. Calibration of a combustion model depends on the burn rate specified, but measuring the burn rate during the on-going combustion process is a difficult task and hence the cylinder pressure is measured in the test rig.

In GT-Power simulations, a two-zone combustion methodology is followed in which the combustion cylinder is divided into two zones, the burned zone and the unburned zone. A fresh charge is injected into the unburned zone and the burn rate is calculated as the rate at which fuel and air molecules are transferred from the unburned zone to the burned zone and to form chemical reactions.

The combustion process in GT-Power is controlled by the burn rate [10]. Gamma Technologies has developed a forward and reverse run method to overcome this problem. During the reverse run, the burn rate is calculated using the cylinder pressure as input which reproduces the same cylinder pressure in forward run.

In GT-Power reverse calculations are performed in a similar fashion as used in the standard forward run. At each time step, the amount of fuel transferred from the unburned zone to the burned zone is repeated until the measured cylinder pressure is achieved. During the forward run, cylinder pressure is calculated using the burn rate. The fuel and air molecules are transferred from the unburned zone to the burned zone as specified by the burn rate and the cylinder pressure is the resultant of energy released from combustion. Both forward and reverse run calculations use all of the same equations as two zone combustion process [10].

2.4.1 Burn rate analysis methods

In GT-Power, there are two methods for estimating the burn rate namely, the Three Pressure Analysis (TPA) and Cylinder Pressure Only analysis (CPOA) [10]. Both of these methods follow the reverse run method which requires the cylinder pressure as input. In this thesis, TPA is used for finding the burn rate as explained below.

2.4.1.1 Three pressure analysis (TPA)

The name three pressure analysis is obtained from the pressures that are required as input to the model, namely, the cylinder Pressure, the intake pressure and the exhaust pressure. To be able to use this method, a complete model of engine has to be modeled as these instantaneous pressures are given as input to the model.

The advantage of this method is that the trapping ratio and the residual fractions are calculated from the simulation. Hence, these quantities need not be measured in the test rig as it is a difficult task [10].

3

METHODOLOGY

In this section, the work flow describes the method to calibrate the FKFS and SITurb combustion models in detail. The work flow chart is shown in Figure 3.1.

The first step was data acquisition from the single cylinder test rig. Due to time constraints, the data was obtained for 19 operating points and these points include full load conditions for entire engine speed range and few points at a low speed part load conditions. The selection of these points were based on the suggestion by FKFS and Gamma Technologies such that these operating points should reflect the real time engine operating conditions.

The data recorded from the test rig included crank resolved intake, exhaust and cylinder pressures, fuel flow and air flow measurements, lambda, emissions (CO, HC and NO_x) and variable valve time settings. After data acquisition, the data was validated in AVL Concerto before using it for the simulations. In data validation process, the data was checked for three different errors which commonly occur in the test rig, Encoder phasing, Pegging error and Thermal shock error. After checking the data for these errors it was then used for the simulation.

The next step was to model the single cylinder direct injection engine in GT – Power. The geometrical dimension data, material properties and injector rate map details were obtained from VCC. For capturing the pressure fluctuations correctly 3D files of intake and exhaust port geometries were also received and were converted to GT – Power flow objects using GEM 3D.

Once the model was set up, data of 15 operating points was given as a input to perform TPA. The main inputs for TPA were the three crank resolved pressures, intake, exhaust and cylinder pressure, mass of fuel injected, VVT settings and emission values. TPA performs automated consistency checks to make sure that the input data is correct. The main advantage of performing TPA is to obtain the results for the trapped quantities and the volumetric efficiencies which is not possible to obtain at the test rig.

After TPA, the calibration of the combustion model is carried out. The model is calibrated for 15 operating points. Optimization tools are set up to obtain optimum calibration multipliers.

The next step after calibration is to check the prediction ability of the models, and are analysed by validating it for 4 operating points. The above methods are discussed briefly in the below section.

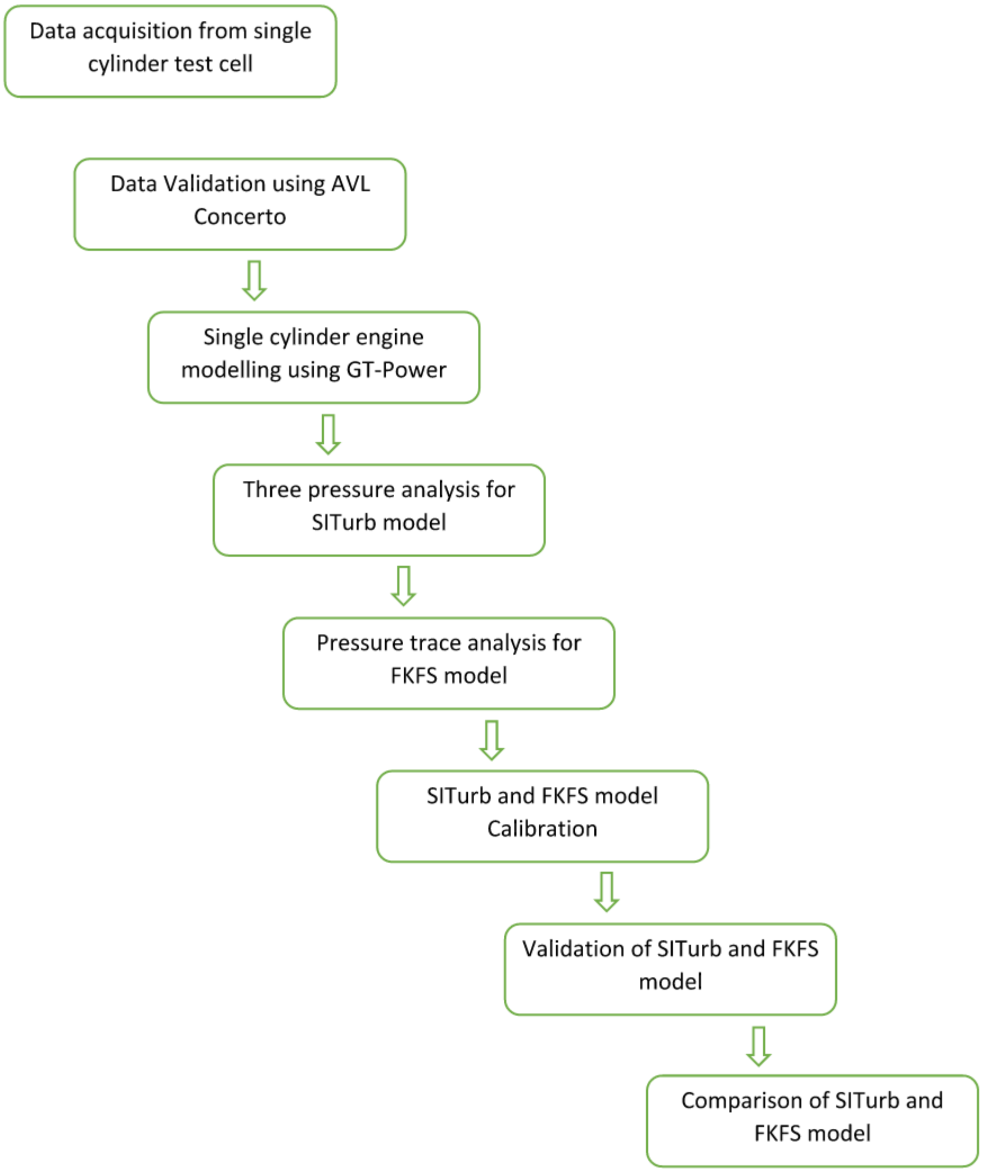

Figure 3.1: Flow chart of the work flow

3.1 Single cylinder test cell

The calibration process depend completely on the quality of the data obtained. The test runs were carried out at the single cylinder test rig at VCC. The data required for modelling and calibration process such as intake and exhaust pressure, cylinder pressure, emission values and so on are measured in the rig for all operating points.

Gamma Technologies recommend at least 25 operating points which is spread across the engine speed range for better prediction of the burn rate for all the operating range.

Due to time constraints, the engine was run at 19 different operating points varying from low speed-low load to high speed-high load point. The operating points are based on the real time scenarios of engine operating range and the points are shown in figure 3.2. Out of 19 operating points, 15 points were used for calibrating the model and the remaining 4 points were used for validation of the calibrated model.

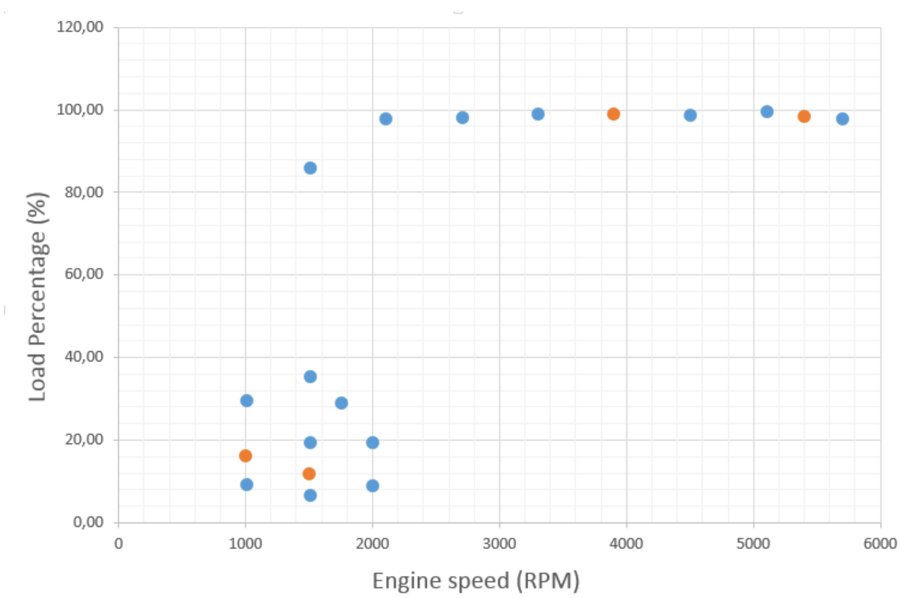

Figure 3.2: Operating points used for calibration (blue) and validation (orange)

3.2 Test data validation

The data recorded from the test rig should be validated because of its great influence on the PTA accuracy and calibration. For correct data measurement, calibration of all the sensors used in the test rig is very important. As there will be some discrepancies while recording the data, it is very essential to perform the validation of the test data. The most common errors that tend to happen at the rig during data recording are, encoder error, pegging error and thermal shock. The obtained data is verified for these errors using the software, AVL Concerto.

3.2.1 Encoder phasing

As explained in section 2.3.1, encoder phasing is a very important error that needs to be verified and there two methods to perform this. Since the TDC sensor was not available at the VCC test rig, motored pressure trace was analysed to verify the encoder phasing. At VCC, a test rig stability check is performed before performing any test runs and the motored pressure trace for 2000 rpm was obtained. As shown in figure 3.3, the peak pressure was occurring within the error limit of 0.6 degree before the compression TDC.

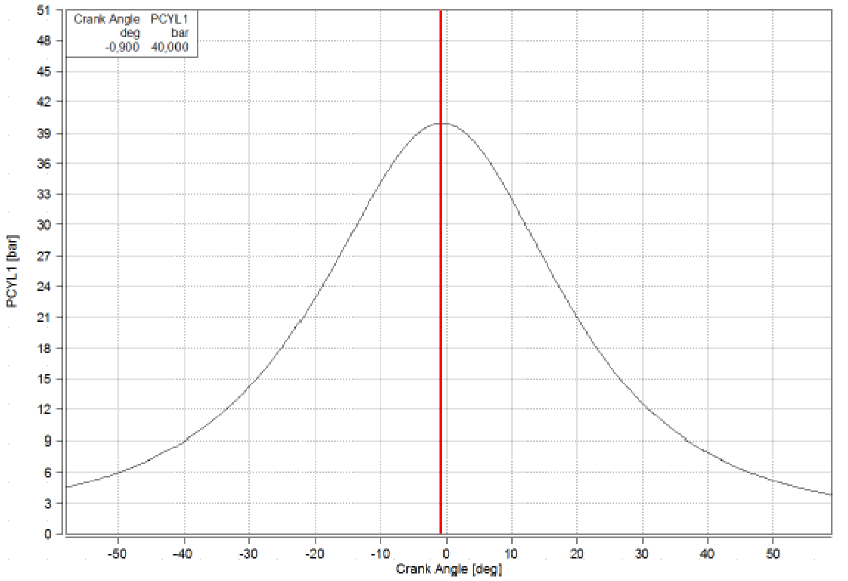

Figure 3.3: Motored Pressure Trace for encoder phasing

3.2.2 Pegging error

As explained in section 2.3.2, pegging error is verified through two methods. Both the methods are used in this thesis to verify the pegging error. First method is verifying the polytropic coefficient to be within the recommended limits. For all the operating points, the polyropic coefficient was found to be between 1.32 to 1.4 at -90 to -35 CAD before compression TDC.

Second method is analysing the cylinder pressure and the intake pressure trace between +/- 5 CAD of intake BDC. The difference between the two pressures was found to be within the limit of 200 milli bar for part load points and result for one of the case is as shown in the figure 3.4. For full load points, the difference was outside the error limit and this is due to the error in pegging during calibration of the sensors. Figure 3.5 shows result of one of the cases of the full load points and the difference between the pressure traces are out of the suggested limit.

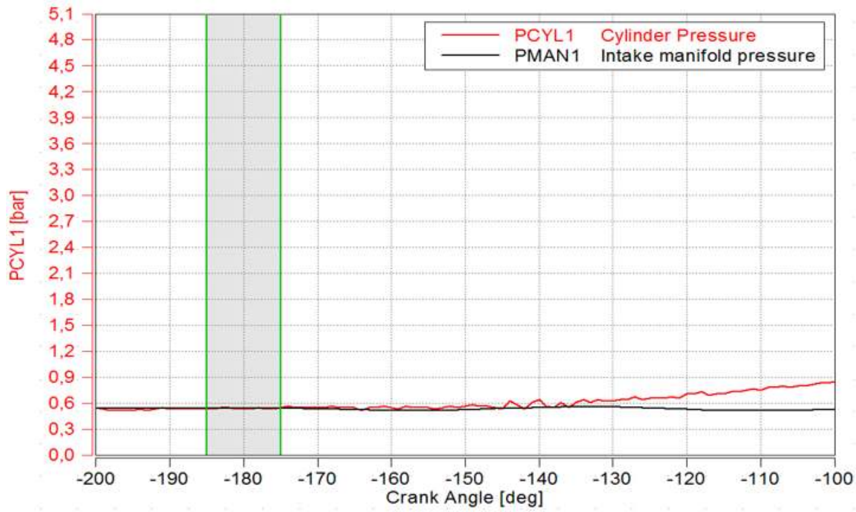

Figure 3.4: Cylinder pressure vs Intake manifold pressure for correct pegging

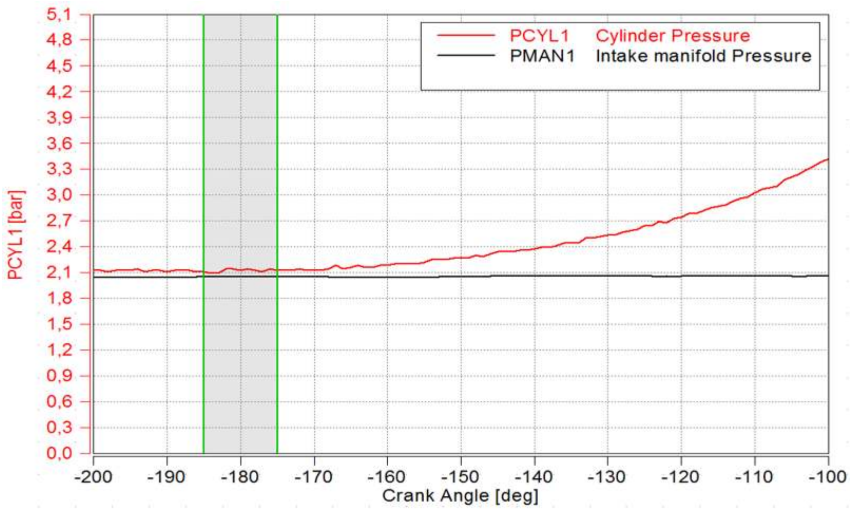

Figure 3.5: Cylinder pressure vs Intake manifold pressure for incorrect pegging

3.2.3 Thermal shock

As explained in 2.3.3, thermal shock is a very common error occurring due to high temperature in the engine. This error is verified by analysing the AEAP.

The AEAP was calculated between 220 to 320 CAD after firing TDC and the standard deviation of the AEAP was verified to be within 4kPa for few cases and one of the case result is as shown in Figure 3.6. As mentioned in section 3.2.2, the pressure sensors were not calibrated for high temperatures because of which the standard deviation of AEAP was out of range for a few cases and one of the case result is as shown in figure 3.7 which shows the standard deviation of the AEAP is more than 4kPa which is outside the error limit.

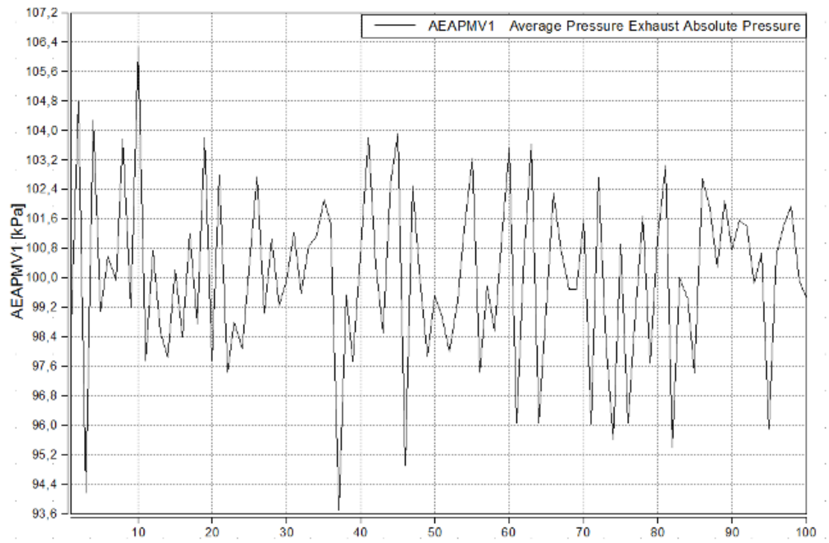

Figure 3.6: Average exhaust absolute pressure for 100 cycles for calibrated point

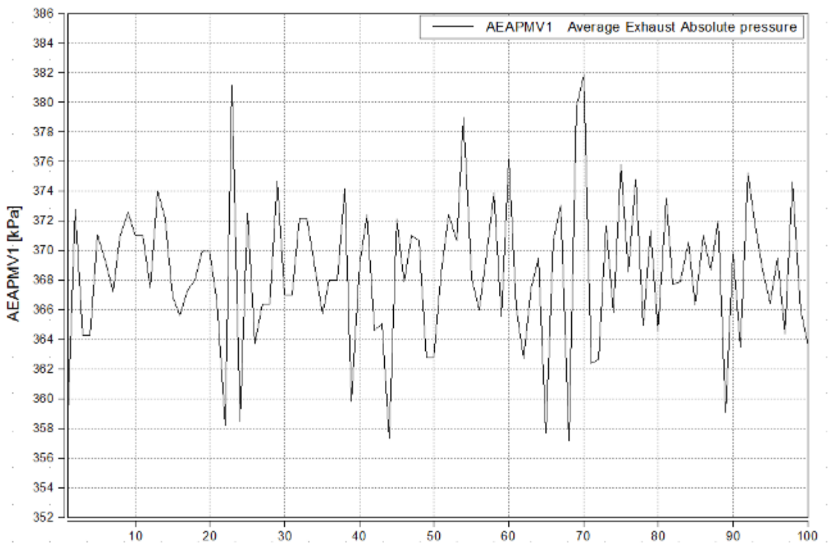

Figure 3.7: Average exhaust absolute pressure for 100 cycles for not calibrated point

3.2.4 Data Filtering

As explained in section 2.3.4, filtering of the pressure traces were done to filter out the discrepancies to obtain better results. In SITurb model, there are different integrated filters which can be used directly while feeding the pressure traces into the model. Whereas, in FKFS model there are no integrated filters so the pressure traces have to be filtered externally. Savitzky golay filter with a fourth order polynomial and a frame size of 121 was used to filter the pressure trace. Figure 3.8 shows the filtered and unfiltered cylinder pressure during the EVO phase.

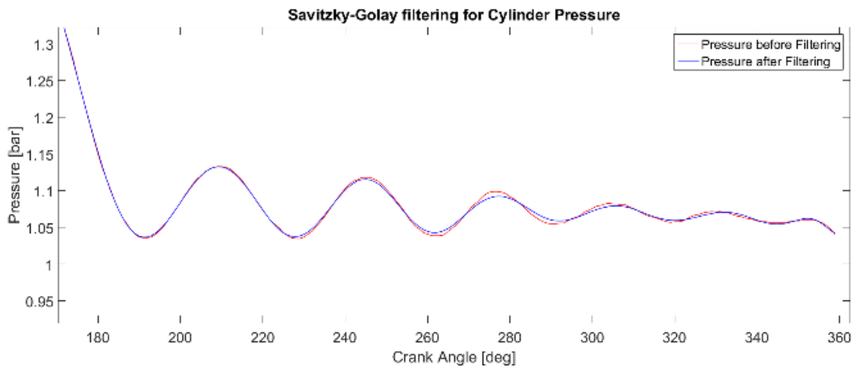

Figure 3.8: Cylinder pressure filtering during EVO

3.3 Modelling of Single cylinder engine

A model of single cylinder direct injection gasoline engine is modelled in GT-Power. The model includes a 1:1 scaled dimensions that are taken from the 3D models and from the measurements taken in the test rig. The ports and the runners of the single cylinder engine in GT-Power are modeled using GEM-3D. This method of modeling provides a better capturing of pressure fluctuations than using the GT-Power integrated ports in the software.

The same ports geometry is used for both SITurb and FKFS only the combustion cylinder differ from each other. Initially, the FKFS model was created using all the geometrical dimensions and material properties obtained from the test rig. Later, the FKFS cylinder was replaced with SITurb cylinder retaining all other connections same as in the FKFS model to have fair comparison between the models.

3.4 Three Pressure Analysis for SITurb model

TPA is essential when predicting the burn rate based on the measured pressure traces for gasoline engines. The main advantage of performing TPA is to obtain trapping ratio and the residual fraction which are given out as results from the simulations. The trapped quantities at the IVC are required for calibrating the combustion model. But the downside of TPA is that the calculations are slower and more time consuming as the simulation runs until convergence is reached [10].

To perform TPA the following data are required,

- Crank angle resolved intake port pressure
- Crank angle resolved exhaust port pressure
- Crank angle resolved cylinder pressure
- Intake Port Temperature
- Air mass flow rate
- Spark Timing or CA of 50% Burn Fraction

Since there are possibility of some error in cylinder pressure and other measurement data that is used as input to the TPA. GT-Power performs consistency checks to support the quality of the input data and the following are the consistency checks performed [10],

- **Reasonable IMEP** – IMEP is calculated by integrating the cylinder pressure profile and it should be greater than the BMEP which is obtained from the measurement. BMEP is equal to algebraic sum of IMEP and FMEP. If IMEP and PMEP is not reasonable, then it indicates an error in measured pressure trace data obtained from the test rig.

- **Pressure Smoothing** – Measured cylinder pressure at the test rig will have noise in the signal and this will cause error in the simulation. To avoid this, raw pressure is smoothed using a low pass filter. An RMS pressure value is reported in pressure analysis result which indicates error between raw and smoothed pressure curves. An error will be flagged if the RMS pressure is greater than 0.02 which indicates there is data loss while smoothing the curve.

- **Cumulative Burn During Compression** – During compression stroke, the apparent burn rate is calculated by integrating burn rate up to the designated start of analysis and the value should be close to zero. During this period, there should be no fuel burning, so any calculated fuel that is burned shows an indication of error in the input data and an error is flagged if the cumulative burn during compression is greater than 2% of the total fuel.

- **Fraction of Fuel Injected Late** – In direct injection models, any time before and after the injection event, if there is insufficient fuel in the cylinder, the insufficient fuel is tracked and integrated over the cycle. The amount of missing fuel is reported in pressure analysis results and this value should be always zero. If the fraction exceeds 0.02 then an error is flagged.

- **LHV Multiplier** – The Latent heating value (LHV) multiplier indicates the error in the cumulative burn rate. Ideally, the value should be 1 and if the LHV adjustment required is more than +/- 5%, an error is flagged.

- **Combustion efficiency comparison to target** – Combustion efficiency indicates the burned fuel fraction inside the cylinder. The LHV is adjusted to match the combustion efficiency with the target value. If the LHV change required is more than 5% then an error is flagged.

- **Apparent Indicated Efficiency** – During the pressure trace analysis, if the calculated indicated efficiency is greater than 45%, then it indicates an error in the input data.

- **Air Mass at IVC** – In TPA simulations, it is possible to compare the trapped air mass at IVC from the simulation to the measured air mass from the test rig. If the difference in the value is greater than 5% then it indicates an error in the valve timing settings.

- **Fuel Mass** – In TPA simulations, total fuel injected from the simulation is compared to the measured fuel mass from the test and if the values differ by greater than 5%, then the amount of fuel injected is incorrected.

- **Fuel Air Ratio** – In TPA simulations, simulated air-fuel ratio is compared to the measured value obtained from the test cell. If the deviation exceeds +/-5%, an error is flagged.

The above mentioned consistency checks helps finding the error in the input data. The advantage of the consistency checks is that it indicates the problems specifically so that correcting the respective data would result in solving the error which reduces the time in finding the cause.

The TPA analysis was performed on all operating points. The measured and the simulated pressure curves were matched visually. Along with this, the consistency checks were also verified. In some of the cases there were few consistency checks error which could be caused due to variable valve timing settings, compression ratio adjustment and heat transfer multipliers in the ports.

3.5 Pressure Trace Analysis for FKFS and SITurb model

Pressure trace analysis for the FKFS combustion model is similar to the SITurb model. Input parameters for the FKFS model and SITurb model is same apart from the CA of 50% burned fraction values that are given as input to the FKFS model instead of spark timing as in the SITurb model.

As a part of consistency check, FKFS performs zero level correction in which it checks for the pegging error at IVC and if there is any error then it automatically shifts the pressure curve to overcome the error. This is one of the advantages of the FKFS model and referred as 100% iteration value in the model.

The pressure trace analysis for both the models is similar and corrections are done for both the models to fix the consistency checks and to match the measured and the simulated results.

3.5.1 Incorrect gauge pressure

During pressure measurement in the test rig, an ambient pressure or gauge pressure is used to convert measured signal into absolute pressure and if the gauge pressure has some error then it affects the whole measurement profile by a factor proportional to the error.

The gauge pressure error can be identified by studying the log P- log V diagram. Figure 3.9 shows pressure error of +/- 0.2 bar. The pumping loop mismatch can be reasoned by the difference in pressure measurements at intake and exhaust ends. Incorrect VVT settings could also lead to the mismatch of pumping loop. In the logP-logV diagram, the low pressure region slope decreases significantly from beginning to end of compression and the slope increases during compression for high pressure region. The pressure shift was manually specified after initial evaluation of the measured pressure data [10].

3.5.2 Intake and exhaust pressure scaling

During data validation, it was found that the pressure sensors were not calibrated for high temperatures. Due to this reason, the simulation results were under predicting as shown in figure 3.10. The intake and exhaust pressure trace data were carefully analysed and scaling of the same was done to match the test data results.

The pressure is measured at two positions in the test rig at VCC. Slow pressure measurement was taken in the vessel and the fast/crank angle resolved pressure measurement was taken at the respective runners. There always existed a difference in pressure measurement when the average of the fast sensor was compared with the respective slow measurement.

A linear trend with respect to speed was observed in the pressure difference .i.e., higher the engine speed, higher was the difference in pressures.

Thus, a scaling factor was calculated using the below formula,

$$Scaling factor = \frac{(P_{vessel} - P_{fastaverage})}{P_{vessel}} \tag{3.1}$$

Figure 3.11 shows the simulated cylinder pressure matching with the measured pressure after scaling the intake and exhaust pressure. The fast sensor pressure data is multiplied by this scaling factor to obtain a reasonably correct pressure. Figure 3.11 shows the simulated cylinder pressure after the intake and exhaust pressure have been corrected by using the scaling factor.

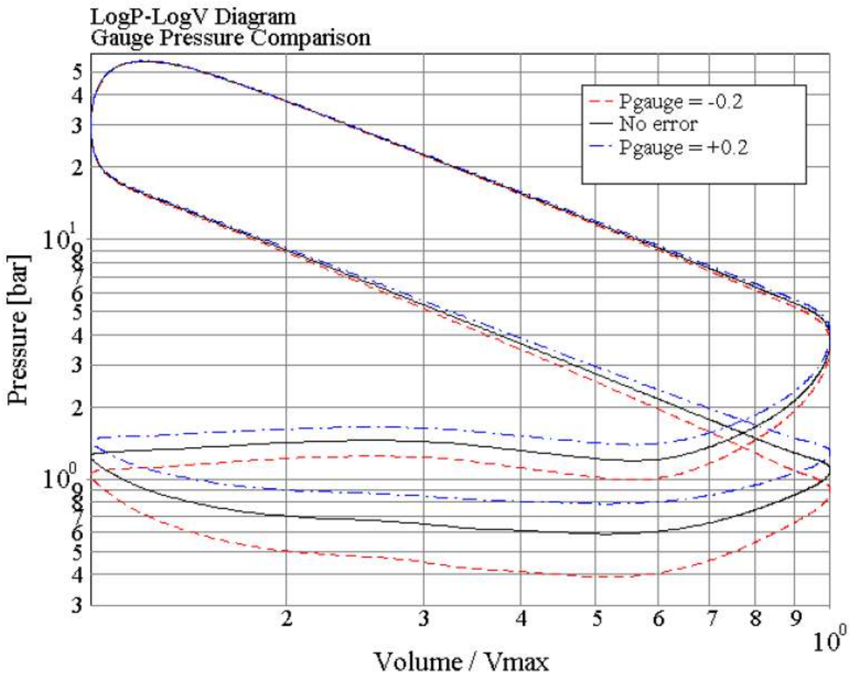

Figure 3.9: Incorrect gauge pressure [10]

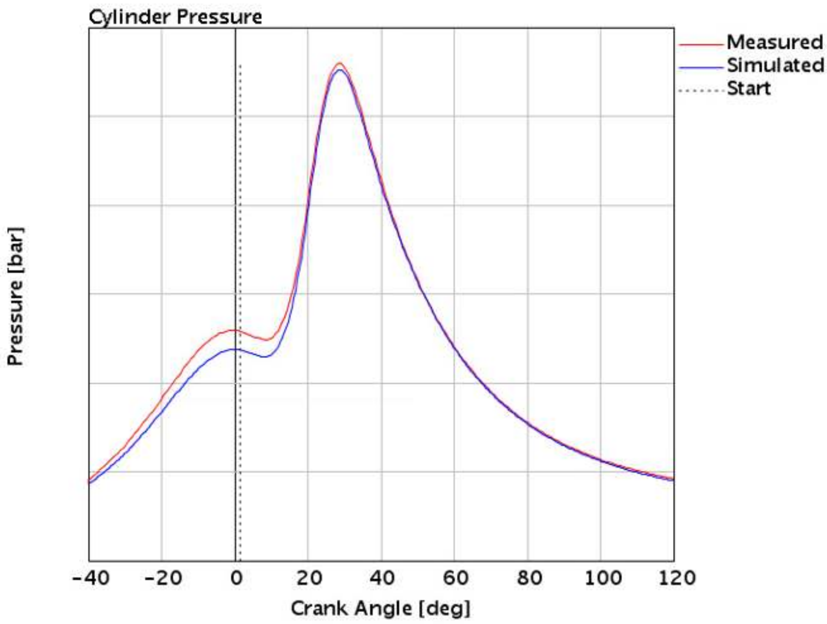

Figure 3.10: Measured vs Simulated cylinder pressure without intake and exhaust pressure scaling

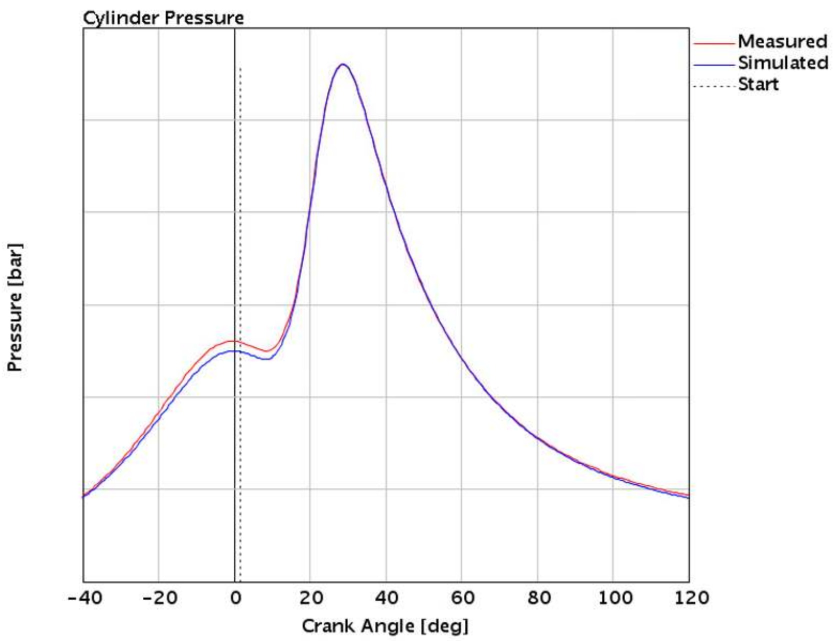

Figure 3.11: Measured vs Simulated cylinder pressure with intake and exhaust pressure scaling

3.5.3 Valve timing settings

The values of VVT settings used in the TPA model is from the test rig measurements. With the original values the TPA results such as air flow rate, trapped quantities and cylinder pressure were off from the test rig data and this could be because of the valve lash, inertia and slack in the belt of camshaft belt drive that control the VVT. To compensate these errors VVT settings were varied within the limit of +/- 5 degrees, with several iterations the correct VVT settings values were identified to match the TPA results with the test rig measured values. .

3.5.4 Compression ratio adjustment

The geometric compression ratio for the engine is assumed to be constant but the dynamic compression ratio varies with different load and speed conditions. Measuring the dynamic compression ratio for each operating point at the test cell is very difficult, hence TPA simulation is used to find the dynamic compression ratio. In TPA analysis, compression ratio is iterated until the measured and simulated pressure curves along with log P-log V diagrams are matched. Figure 3.12 and Figure 3.13 shows one of the cases where compression ratio is adjusted by reducing 0.4 from the absolute value used in the test rig to match the pressure curves. For final calibration model, the average of the dynamic ratio value is used for easy model scalability.

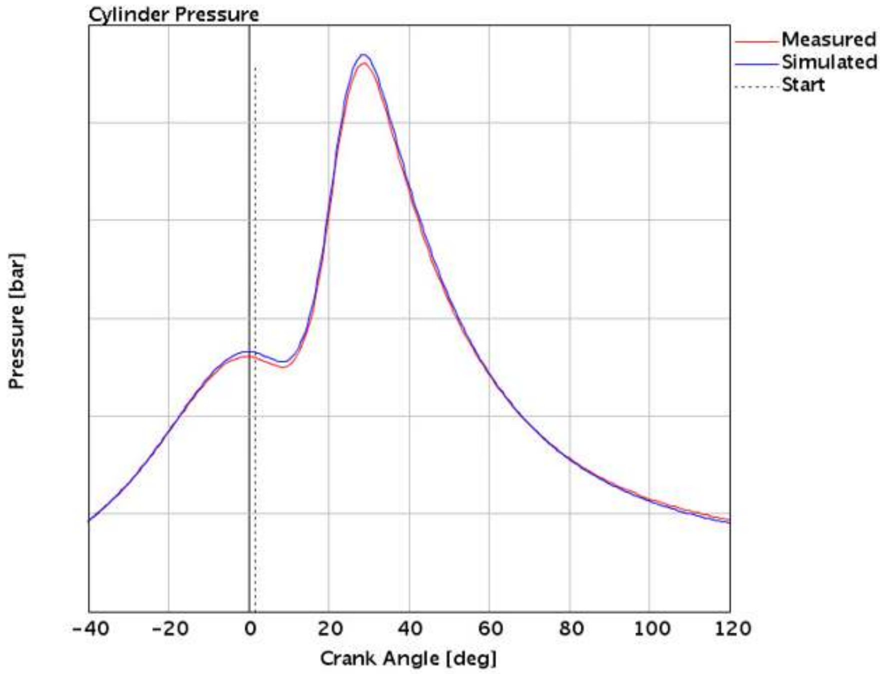

Figure 3.12: Measured vs Simulated cylinder pressure with absolute compression ratio value

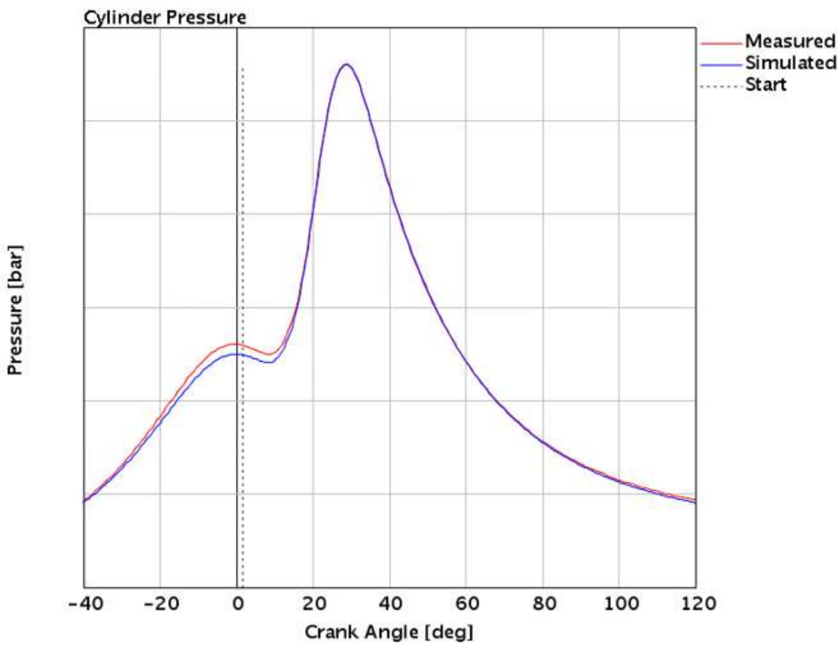

Figure 3.13: Measured vs Simulated cylinder pressure with compression ratio value reduced by 0.4

3.6 Combustion model Calibration

After TPA, the next step is to calibrate the combustion model. The goal of the combustion model calibration is to obtain single set of model multipliers that will present best possible match that would be applicable for wide range of operating points.

Out of 19 operating points obtained from the test rig, 15 points are used for calibrating the model and rest 4 points are used for validation of the calibrated model to predict the combustion characteristics.

Each combustion model has a set of multipliers that are tuned using optimizer tool to obtain the correct set of values to have the best fit of the curve when compared to the results from TPA. The results of the TPA were checked for error limits, if these are satisfied then the SITurb model is ran through optimizing tool called Direct Optimizer and Joint Optimizer for FKFS model to obtain the correct set of multiplier values. These values determine the process of combustion right from turbulent mixing to the flame front propagation. Each of the multiplier are discussed in the below subsections.

3.6.1 SITurb Calibration Model

The four multipliers for calibrating the SITurb combustion model are namely,

- Dilution multiplier
- Flame kernel growth multiplier
- Taylor factor multiplier
- Turbulent flame speed multiplier

Gamma technologies recommend sweep optimization using Direct optimizer to find the best multiplier values for all operating points. The model set up for the calibration includes only cylinder and cranktrain and this will perform closed volume pressure analysis, i.e. no gas exchange is included. The model will provide a comparison of cylinder pressure and burn rate between measured and predictive combustion model.

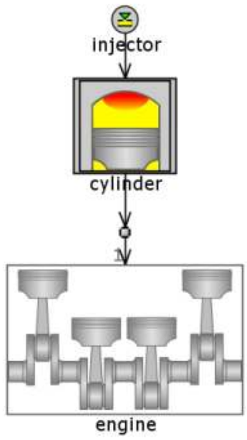

Figure 3.14: Calibration model in GT-Power

The quality of the calibration was analysed by comparing the measured and predicted burn rate plots and they should match closely for all operating points. Apart from comparing the burn rate curves, Gamma technologies suggests to compare the results of the parameters as show in table 3.1 and the error limits were suggested by Gamma Technologies.

Table 3.1: Recommend error limit for SITurb calibration

Parameter	Recommended limit
IMEP (%)	+/-5
CA50 (deg)	+/-3
Peak cylinder pressure (bar)	+/-5
Ignition delay (deg)	+/-3

3.6.2 Dilution multiplier

Multiplier used to scale the effect of dilution (residuals and EGR) on the laminar flame speed or the name of a dependency reference object defined. Increasing this value will reduce the effect of dilution on the laminar flame speed and thus increases the burn rate and vice versa. Figure 3.15 and 3.16 shows the influence of different dilution multiplier values on cylinder pressure and burn rate.

3.6.3 Flame kernel growth multiplier

This multiplier is used to scale the calculated value of the growth rate of the flame kernel. This variable influences the ignition delay. Larger numbers shorten the delay, advancing the transition from laminar combustion to turbulent combustion. Figure

3.17 and 3.18 shows the influence of different dilution multiplier values on cylinder pressure and burn rate.

3.6.4 Turbulent flame speed multiplier

Multiplier is used to scale the calculated turbulent flame speed or the name of a dependency reference object defined by the user. This variable influences the overall duration of combustion. Larger numbers increase the speed of combustion. Figure 3.19 and 3.20 shows the influence of different dilution multiplier values on cylinder pressure and burn rate.

3.6.5 Taylor Length Scale Multiplier

This is used to scale the calculated value of the "Taylor microscale" of turbulence or the name of a dependency reference object defined by the user. The "Taylor microscale" modifies the time constant of combustion of fuel/air mixture entrained into the flame zone by changing the thickness of the plume. This multiplier mostly influences the tail part of the combustion and is relatively insensitive. Figure 3.21 and 3.22 shows the influence of different dilution multiplier values on cylinder pressure and burn rate.

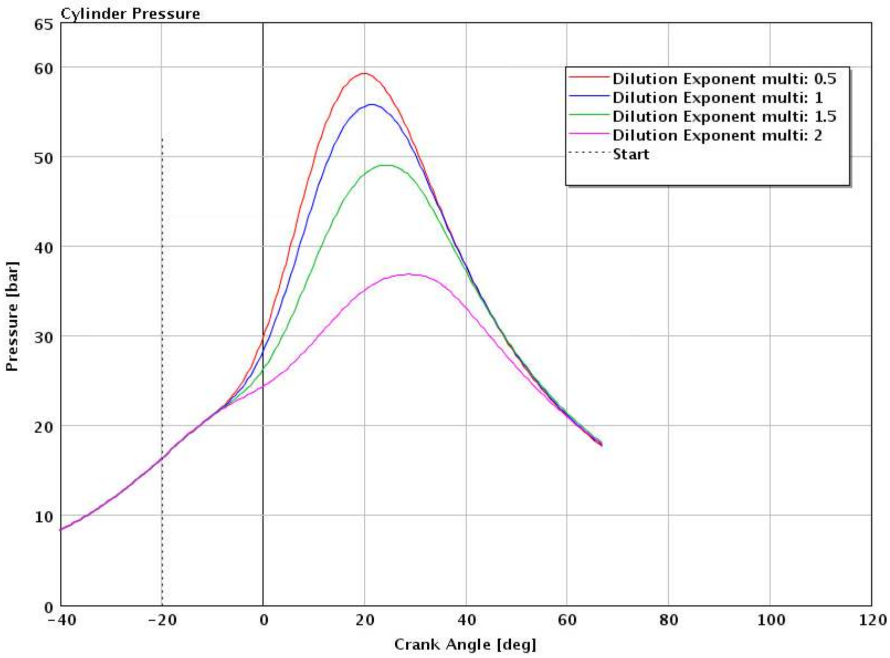

Figure 3.15: Cylinder pressure for different Dilution multiplier

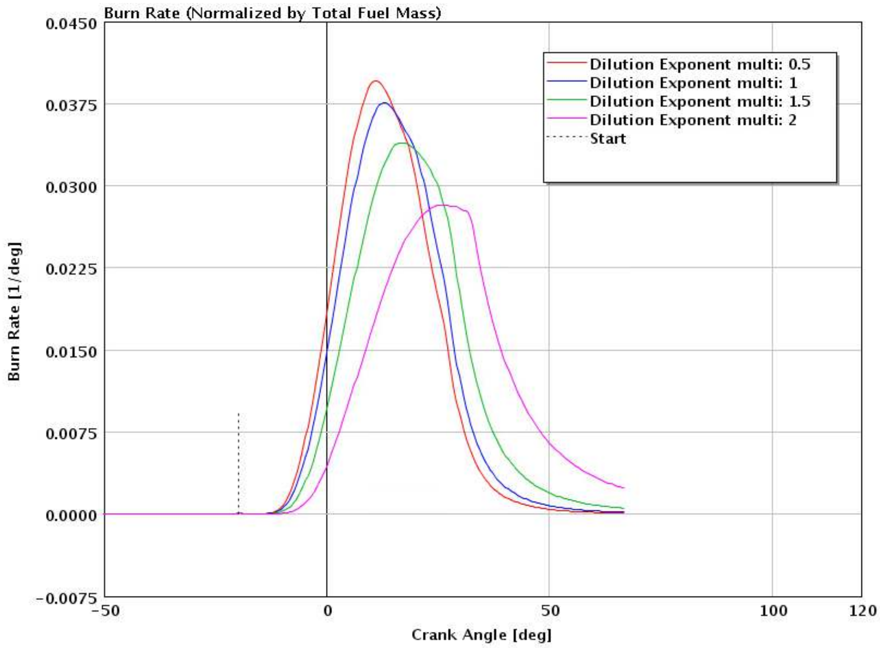

Figure 3.16: Burn rate for different Dilution multiplier

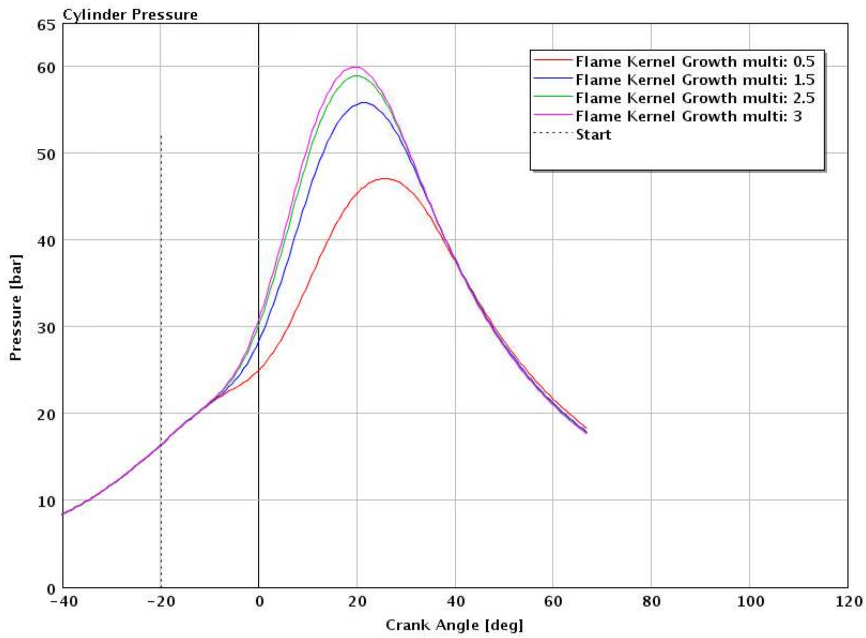

Figure 3.17: Cylinder pressure for different Flamer kernel growth multiplier

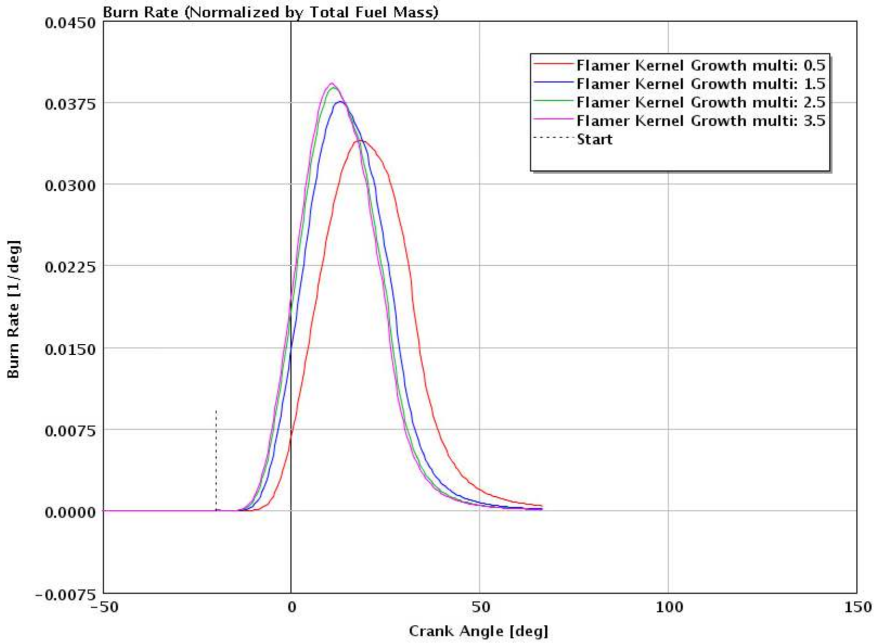

Figure 3.18: Burn rate for different Flamer kernel growth multiplier

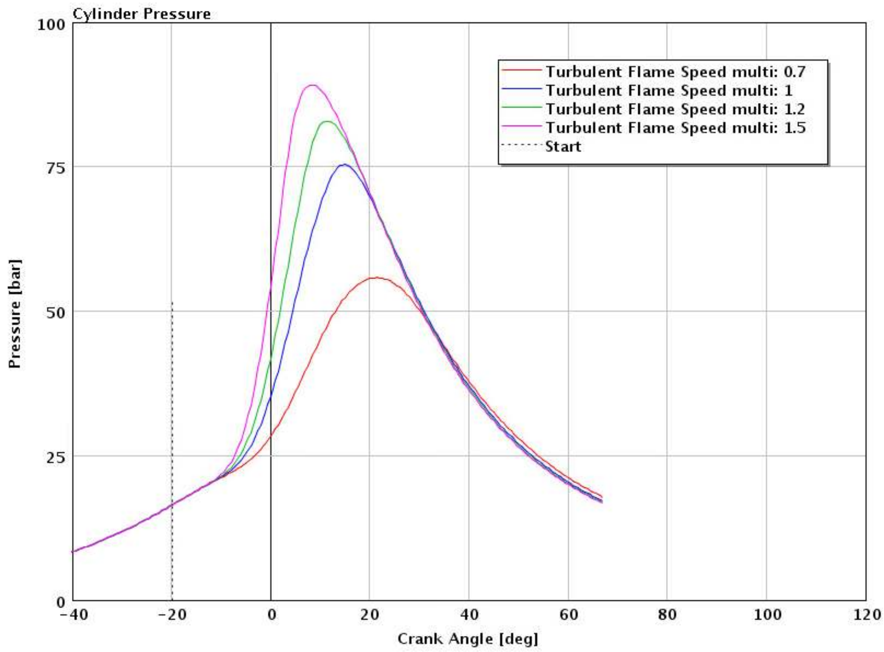

Figure 3.19: Cylinder pressure for different Turbulent flame speed multiplier

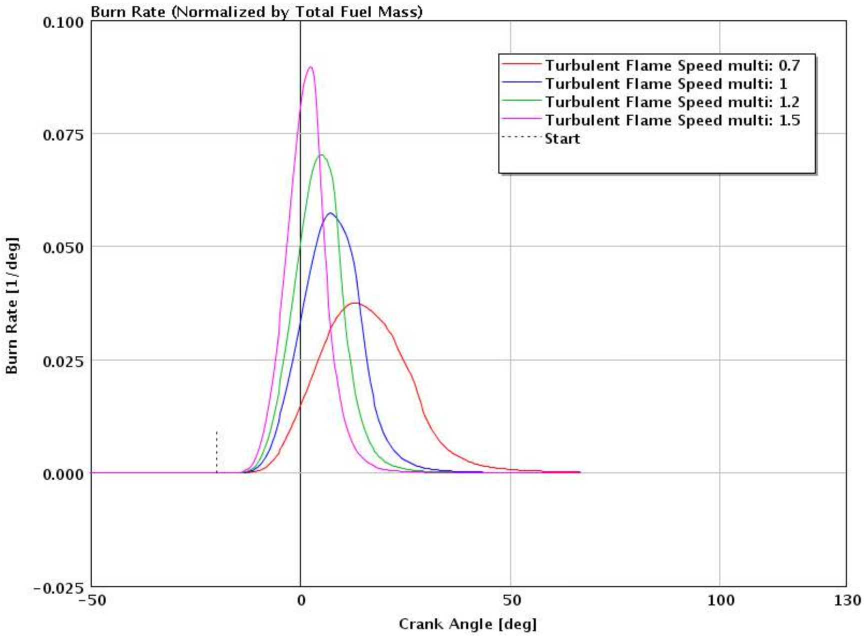

Figure 3.20: Burn rate for different Turbulent flame speed multiplier

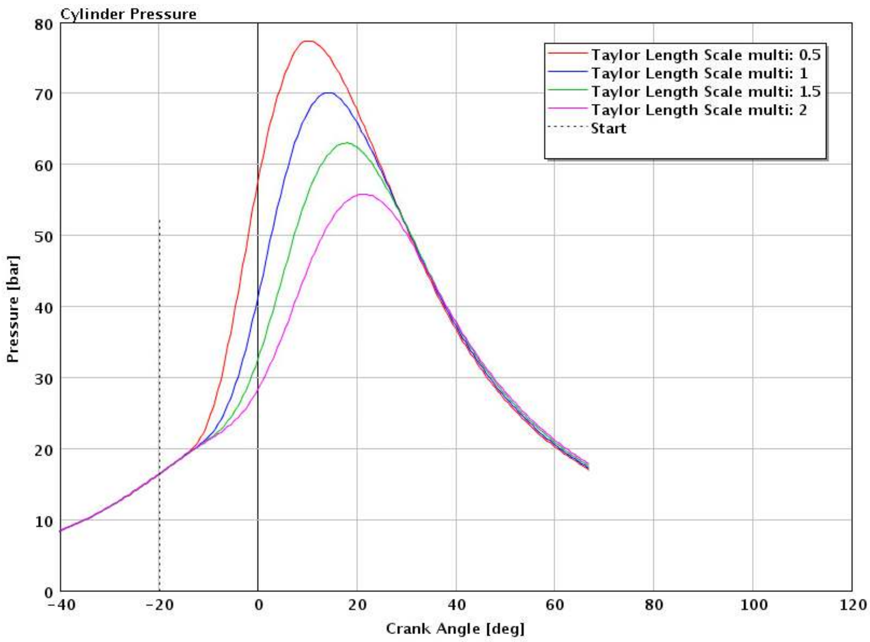

Figure 3.21: Cylinder pressure for different Taylor length scale multiplier

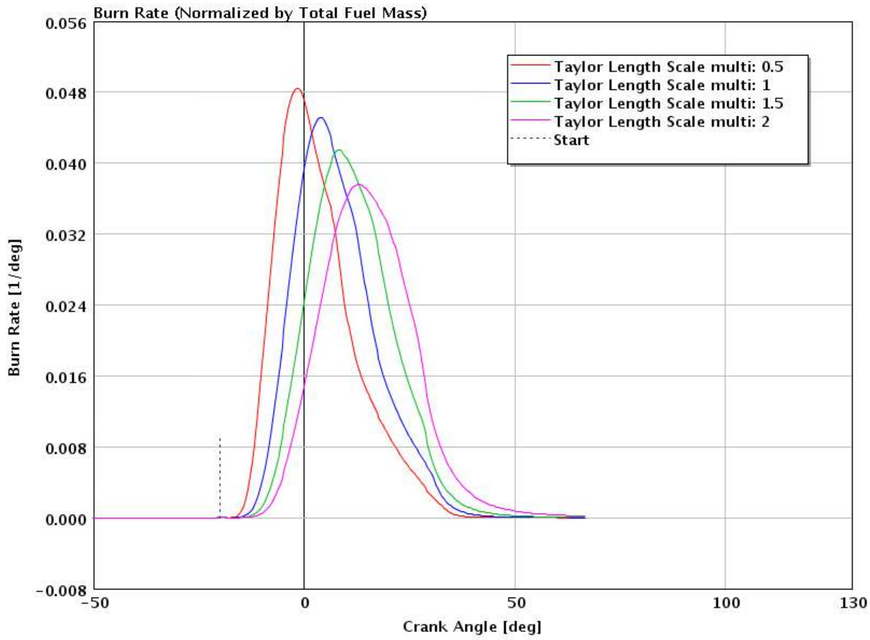

Figure 3.22: Burn rate for different Taylor length scale multiplier

3.7 FKFS Combustion model Calibration

After the pressure trace analysis is completed, the FKFS model needs to be calibrated. The calibration method followed for FKFS is slightly different compared to the SITurb model. FKFS model has only one calibration parameter which is Quasi Dimension charge motion (C_u). The calibration model is same as the pressure trace analysis model and there is no need to isolate the cylinder, cranktrain and injector as its done for SITurb calibration model.

The FKFS model has options where TPA and optimizer can be turned ON and OFF depending on the purpose of the simulation. It also has an inbuilt optimizer in the combustion model which should be turned ON while running the calibration process. The optimization tool used for the calculation of charge motion multiplier (C_u) is "Joint Optimization". To run this particular optimizer, the TPA has to be turned ON. It uses the results of TPA as input automatically to run the optimizer. This type of optimizer yields a single value that is optimized for the complete range of operating points of the engine.

Turbulence Modeling

QDM Charge motion – This is a turbulence modeling multiplier. For the turbulence calculation, the model takes the course of the gas exchange into account and enables the combustion model to respond, for example, to variable cam timing or a variable valve lift. The calculations of turbulence starts at the IVO. The turbulence model can be represented by the equation 3.2 that shows a change in specific turbulent kinetic energy as a difference between a product term and dissipation ε.

$$\frac{dk}{dt} = (\frac{dk}{dt})_{prod} - \varepsilon \tag{3.2}$$

There are several factors that influence the creation of turbulence. In FKFS model, the following scenarios are considered for calculation of turbulence.

- Turbulence production due to inflow air
- Turbulence due to piston motion
- Turbulence production due to charge motion or tumble
- Turbulence due to change in density
- Dissipation due to cylinder geometry and cylinder inflow
- Dissipation due to wall influence
- Determination of the turbulent fluctuating velocity u_{Turb} - The turbulence kinetic energy (k) obtained from turbulence model will be used to calculate the turbulent fluctuating velocity that is utilized in entrainment model. This equation also contains the calibration variable (c_u) that allows to tune the mixture of turbulence and heat release model[25]. This is given the below equation

$$u_{turb} = c_u * \sqrt{\frac{2}{3} * k} \tag{3.3}$$

3.8 SITurb and FKFS Prediction

The calibrated combustion model predicting capabilities are validated by comparing the results with test data collected for operating points from the low speed to the high speed at different load conditions. Out of 19 operating points, 15 points were used for calibrating the combustion model and the rest 4 points are used for the validation. The operating points used for validations are as shown in table 3.2.

Table 3.2: Operating points for SITurb and FKFS prediction

Sl No.	Speed (RPM)	Load (%)
2	1000	16.16
6	1500	11.81
16	3900	98.4
19	5400	100

4

RESULTS AND DISCUSSION

In this section results and discussion of the test data verification, pressure trace analysis, calibration of the combustion models and predicting capabilities of the calibrated models are presented.

4.1 Test data verification

In this section the results of data quality validation are discussed. As explained in section 2.3, the common errors tend to occur in the test data which consists of Encoder error, Pegging error and Thermal shock error.

4.1.1 Encoder phasing

The encoder error was verified by analysing the motored pressure trace obtained at 2000 rpm and the peak cylinder pressure was found to occur at 0.4 CAD before TDC. The error limit recommended was 0.6 CAD and the result obtained is within the limits. Though the result is within the limits it can't be guaranteed about no encoder error through the motored pressure analysis method instead as explained in section 2.3.1 following the TDC sensor method would yield more accurate and reliable results but unfortunately this method is not available at VCC test cell.

4.1.2 Pegging error

Out of two methods used for verifying the pegging error, the polytropic exponent for all the cases was found to be within the recommended limit of 1.32 to 1.4 for the gasoline engines during the compression stroke between -90 to -40 before the TDC as shown in the figure 4.1. The polytropic exponent indicates the type of gas mixture, cylinder wall heat transfer and also between the air fuel mixture and burned gases.

The second method for checking pegging error is analysed by checking the pressure difference between cylinder pressure trace and intake pressure within a window of +/- 5 CAD at intake BDC. The results for part load cases were found be within the limit of 200 milli bar. Since the pressure sensors were incorrectly pegged while running the engine for full load points the results for full load points are out the error limit.

From the pressure trace analysis, it was found that the correct pegging of the sensors was very important to yield correct results. Because of incorrect pegging for full load cases the intake and exhaust pressure traces were pegged with the scaling factor to obtain correct results as explained in the section 3.5.2.

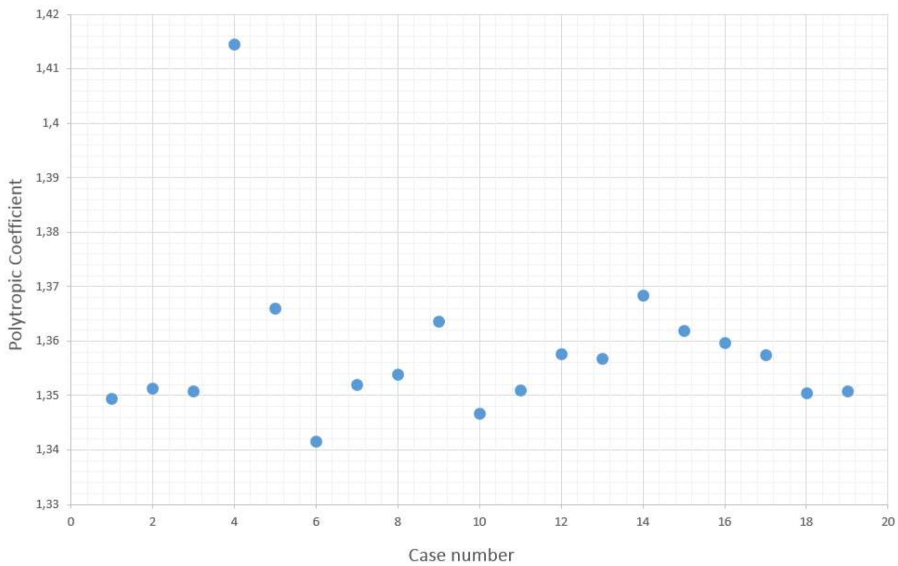

Figure 4.1: Polytropic index for all the cases

4.1.3 Thermal shock

Thermal shock is verified for all the cases by analysing the average exhaust absolute pressure (AEAP) which is calculated between the window of 220 and 320 CAD after firing TDC. The standard deviation of the AEAP for 100 cycles was calculated and found that for the part load points the value were within the limits and since the pressure sensors were not calibrated while running for full load points the standard deviation value was out of the recommended value of 4 kPa. The results of both part load and full load cases are presented in the section 3.2.3. Thermal shock error results an offset in the pressure measurement which leads to a false results. Scaling factor for intake and exhaust pressure was used to compensate for this error.

4.2 Pressure Trace Analysis

The TPA was performed for both SITurb and FKFS model for all the 19 operating points. The findings are discussed in the sections 4.2.1 to 4.2.3.

4.2.1 Pressure trace matching

Matching the measured and simulated cylinder pressure traces is one of the primary goal of the TPA. For easy comparison same settings and parameter values were used for both SITurb and FKFS model during pressure trace analysis. The cylinder pressure traces for all the cases were visually verified and found to be matching. Figure 4.2 and 4.3 shows one of the good and poor matching case for part load points and similarly figure 4.4 and 4.5 shows for full load points. As explained in section 3.5.2, due to calibration error of sensors the intake and exhaust pressure traces were scaled with a factor value obtained through equation 3.1 to match the measured and simulated pressure traces and air flow rate values. Apart from scaling the pressure traces, compression ratio and variable valve timing settings were changed within the specified limits to fine tune the pressure trace matching.

As shown in figure 4.5, the simulated pressure from the FKFS model has better curve fitting compared to the SITurb model during the combustion end phase this is because of the difference in the heat transfer model used. The FKFS model uses 'Bergende' heat transfer model which accounts the heat transfer during the mixing turbulence effects inside the cylinder which is not the same case in 'Woschni' heat transfer model used by the SITurb model. Also the difference in the peak pressure is because of the average compression ratio value used for all the cases as explained in section 3.5.4.

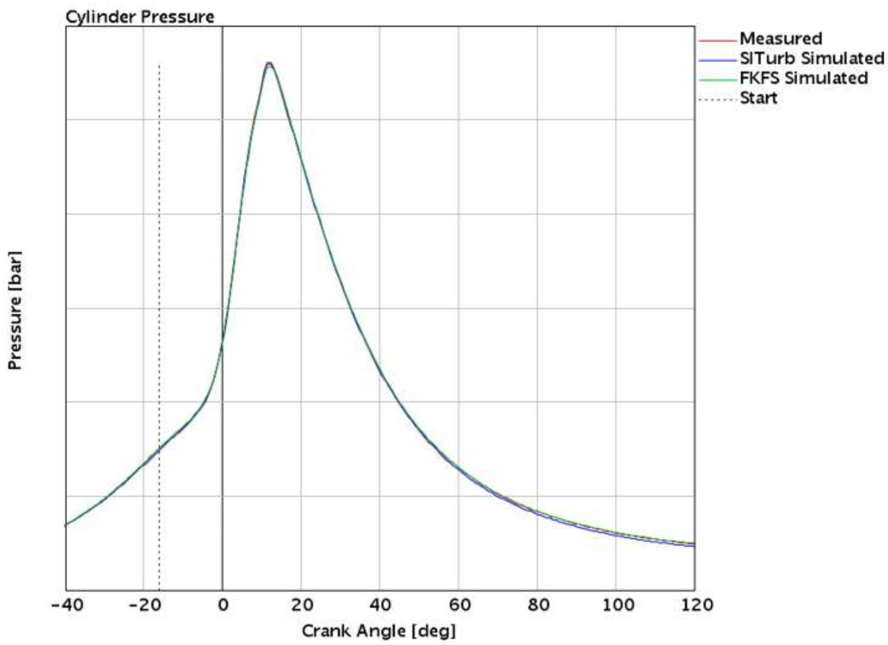

Figure 4.2: Measured vs Simulated pressure trace for best matching case at part load point

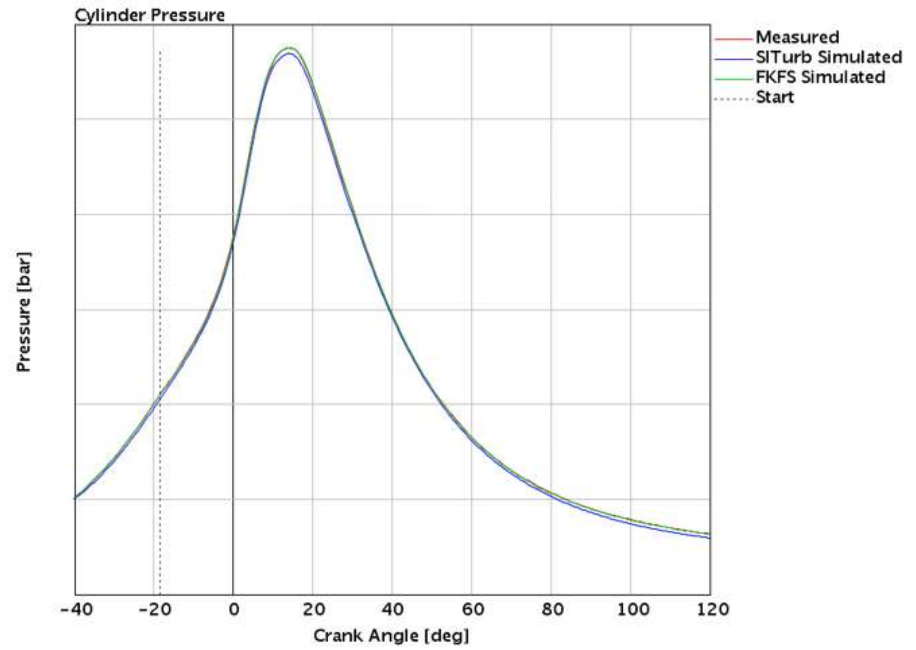

Figure 4.3: Measured vs Simulated pressure trace for poor matching case at part load point

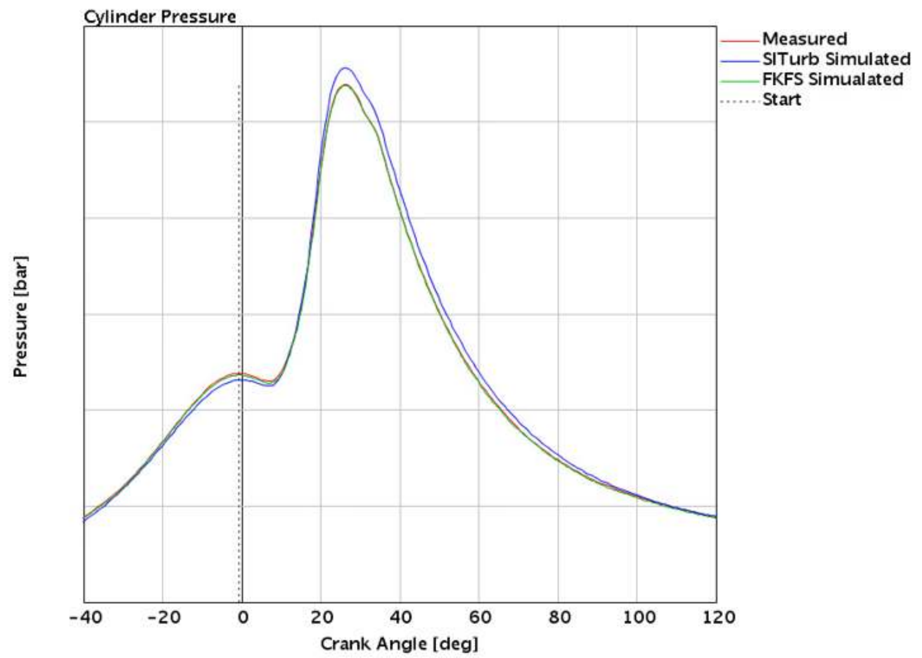

Figure 4.4: Measured vs Simulated pressure trace for best matching case at full load point

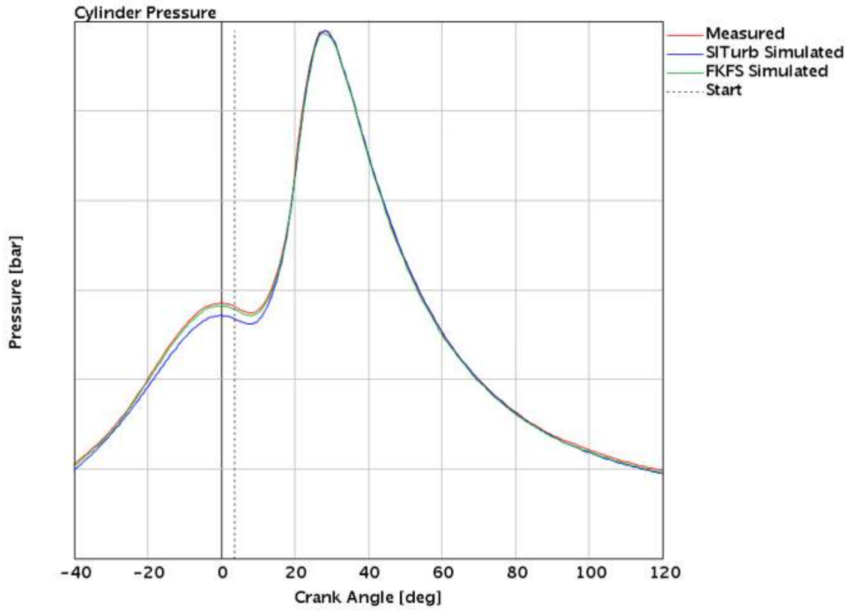

Figure 4.5: Measured vs Simulated pressure trace for poor matching case at full load point

4.2.2 Consistency checks

As explained in section 3.4, consistency checks help to analyse the data quality used for simulation. During three pressure analysis different consistency error checks were flagged and most of the errors were fixed through Scaling factor, VVT and compression ratio adjustment as explained in section 3.5. But even after matching the measured and simulated pressure traces and results for all the points few minor errors still existed as shown in table 4.1.

Table 4.1: Consistency checks summary

Consistency check	Number of cases
Pressure smoothing	5
Large LHV change required	3

Pressure smoothing error was showing for 5 cases during TPA analysis and this was because of the filtering option used. Gamma technologies suggested to use IEEE54 filter but this filter was removing the data points almost to the half number and since FKFS model is more sensitive to the input data, Savitzky golay filter was used to filter the cylinder pressure trace which tries to retain as many number of points without losing much data. This error was ignored and moved on with the calibration since the error didn't had any significant effect on the results.

The LHV multiplier error indicates error in the input pressure traces, spark timing, amount of fuel injected etc. Gamma technologies suggest the LHV values should be within 0.95 and 1.05, all the cases values were within the suggested limit except for few cases. But even for these cases which are out of the error limit was not much deviated from the limit, the values were almost close to the error limit. Hence the consistency check error was ignored and moved on to calibration with the obtained results.

Apart from the two consistence check errors mentioned in the table 4.1, there were no other errors for all the cases. Since the simulated and measured pressure traces and results were matching well for all the operating points, calibration process was carried out.

4.2.3 IMEP, Air flow rate and Volumetric efficiency variation

IMEP is one of the primary result which is verified to be within the limits during PTA. Gamma technologies suggest the error limit between the measured and simulated IMEP should be within +/-5% and the results are calculated and as shown in figure 4.6. The IMEP variation was found to within the error limit for all the cases and this indicates the pressure trace analysis performed yielded satisfactory results which can be used for calibration process. Any error in the IMEP would indicate the error in the cylinder pressure input and IMEP values are very important to be within in the limits since the simulated cylinder pressure and burn rate are greatly influenced by these values. Since calibration of predictive combustion is carried out in the thesis great importance is given in keeping the IMEP values within the limits.

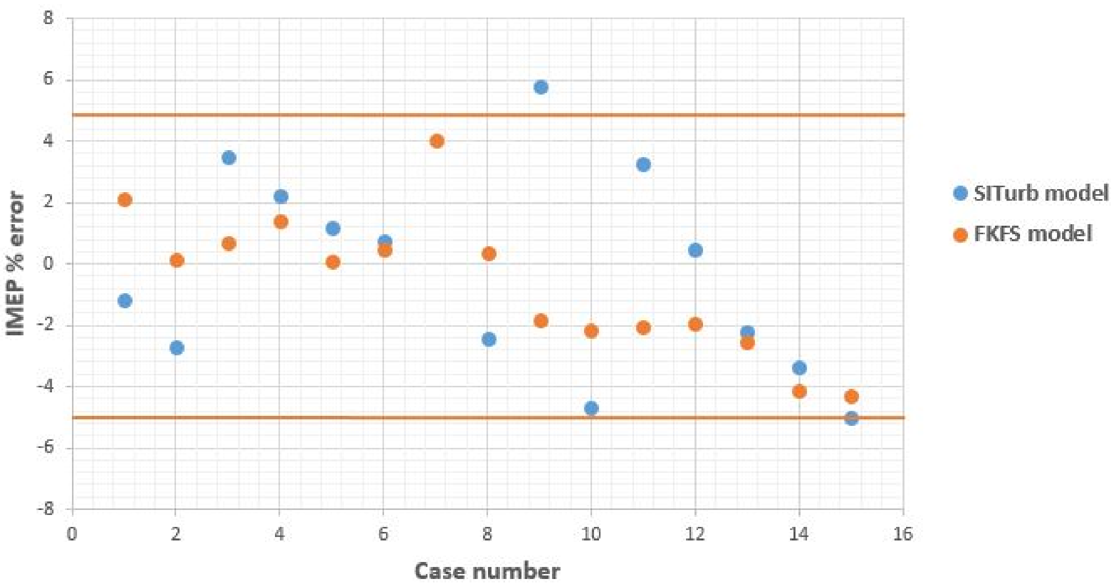

Figure 4.6: IMEP variation results of SITurb and FKFS model

Figure 4.7 and 4.8 shows a plot of air flow rate and volumetric percentage error. The error limit for both the calibration models is recommended to be within +/-2% and +/-3%. It is verifies that most of the engine operating points lie within the range of error limits except for a few points this could be because of the error in the intake and exhaust pressure traces measurement.

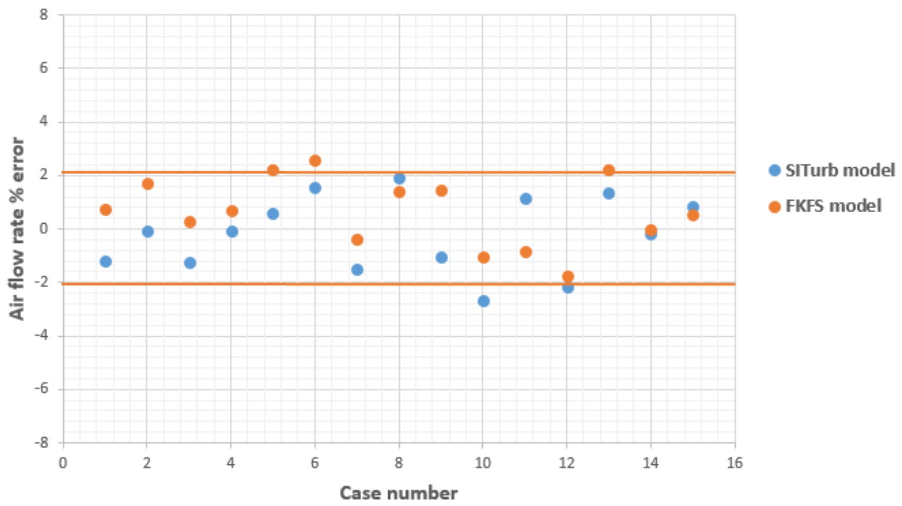

Figure 4.7: Air flow rate variation results of SITurb and FKFS model

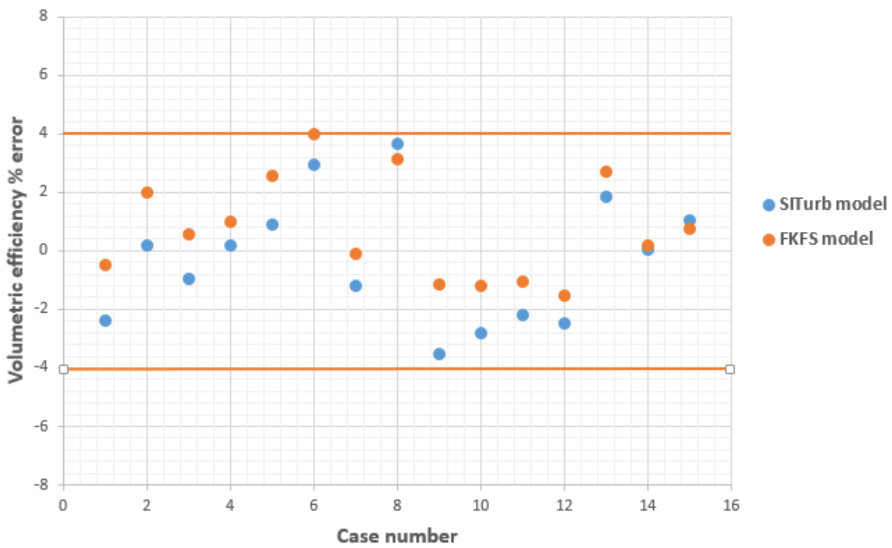

Figure 4.8: Volumetric efficiency variation results of SITurb and FKFS model

4.3 SITurb and FKFS Calibration

After obtaining satisfactory results from the TPA, the calibration process was carried out. 15 points were used for the calibration process and 4 points were used for the validation purpose. Optimized multipliers are obtained through direct optimization method suggested by Gamma Technologies and through joint optimization method for FKFS model.

Figure 4.9 and 4.10 shows the cylinder pressure and burn rate results of the best cases and figure 4.11 and 4.12 shows the results of the poor cases. From the calibration results it is observed that the prediction for all the part load points were good compared to the full load points. The possible reason for this is the number of operating points used for calibration is populated more in part load region compared to full load region and also due to the lack of operating points at mid load region. The calibration multipliers obtained seem to represent more like part load points, hence the prediction is good at part load compared to full load points. Apart from the above reason, poor prediction at full load cases is due to the wrong intake and exhaust pressure values obtained from incorrect calibrated pressure sensors. The scaling factor for the intake and exhaust pressure used yield satisfactory results but not accurate results because the pressure values are crank resolved values and scaling factor was multiplied for the entire pressure trace which does not account for any pressure fluctuations in between. Intake and exhaust pressure values greatly influences the volumetric efficiency output which is the key input to the calibration model.

A small variation in the volumetric efficiency would affect the prediction curve drastically. Figure 4.4 shows the influence of scaling factor on the intake pressure trace. During the compression stroke at the TDC there is an offset between the measured and the simulated pressure trace which indicates that there is no right amount of mass inside the cylinder during the compression stroke and this could be corrected with increasing the intake pressure scaling factor but since the SITurb model is compared with the FKFS model the scaling factor was kept constant for both the models. Apart from the visual comparison, the calibration results are verified by the error limits suggested by Gamma Technologies and for most of the cases the results were in the limits except for few cases at full load points.

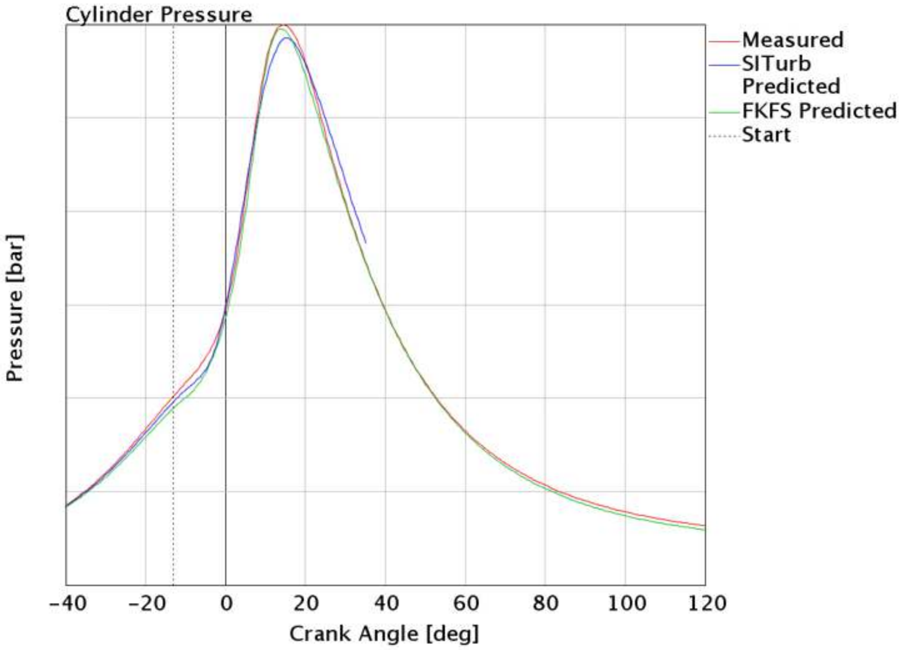

Figure 4.9: Cylinder pressure trace for one of the good calibrated point

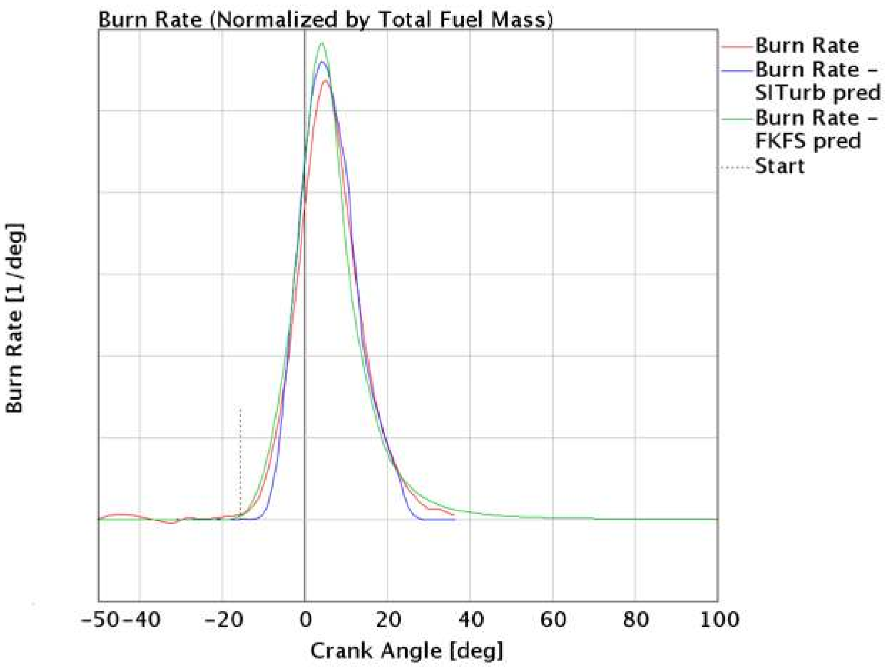

Figure 4.10: Burn rate trace for one of the good calibrated point

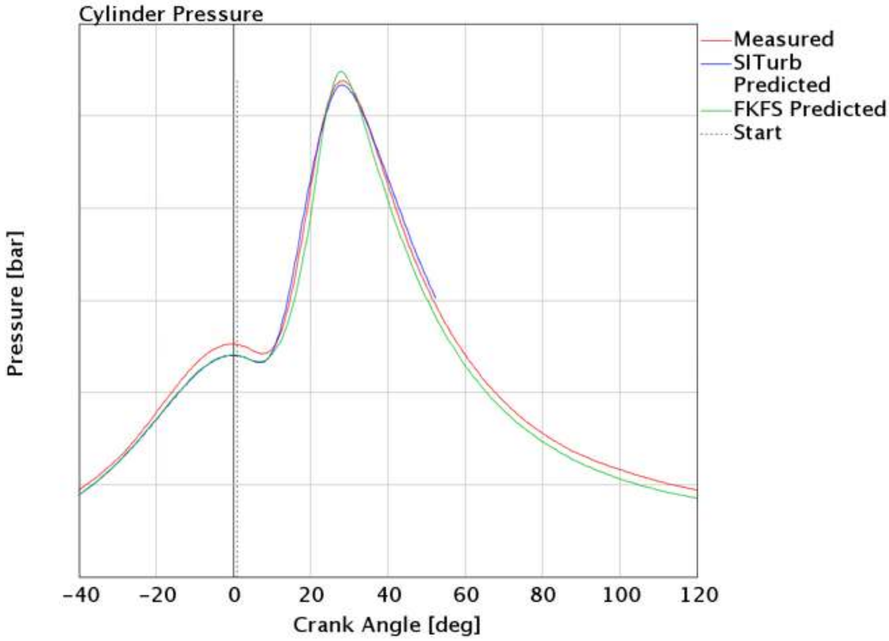

Figure 4.11: Cylinder pressure trace for one of the poor calibrated point

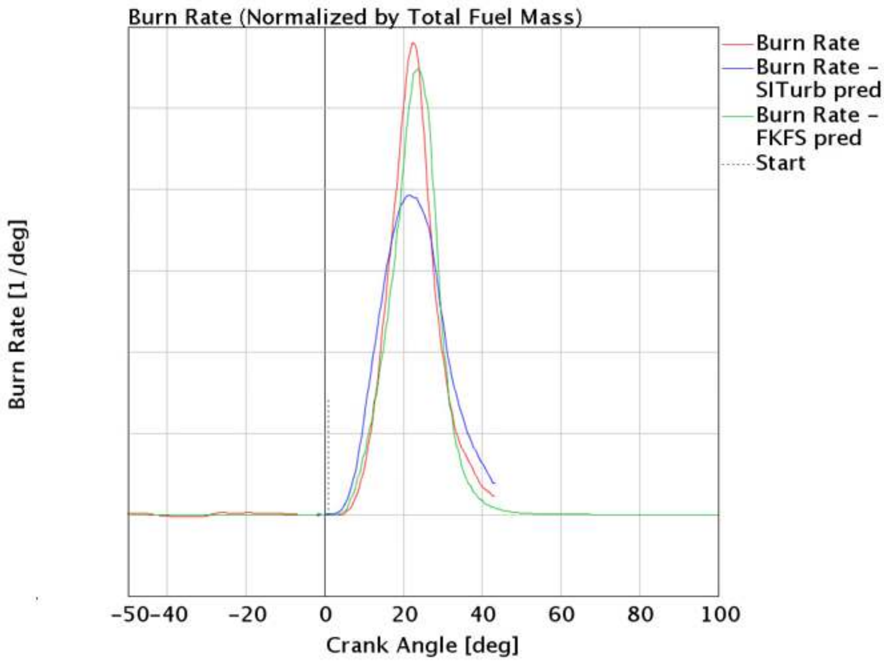

Figure 4.12: Burn rate trace for one of the poor calibrated point

RLT parameters obtained from the both calibrated models are verified to be within the suggested limits except for very few operating points. This could be because of the average compression ratio used and also due to difference in both the models as SITurb uses spark timing and FKFS uses CA50 values. Since CA50 values obtained from the test is a calculated value it consists of a small error like one or two degrees. As overall comparison the results seems to be quite satisfactory.

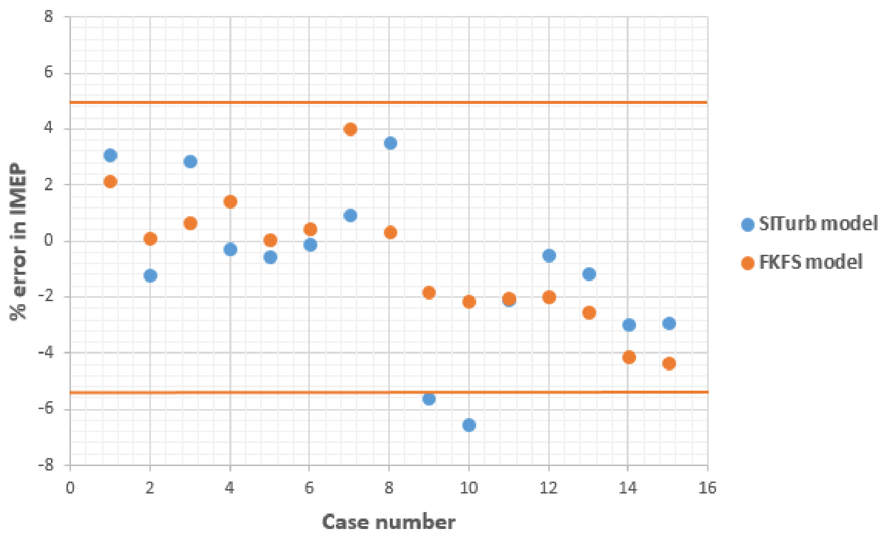

Figure 4.13: IMEP verification for calibrated SITurb and FKFS model

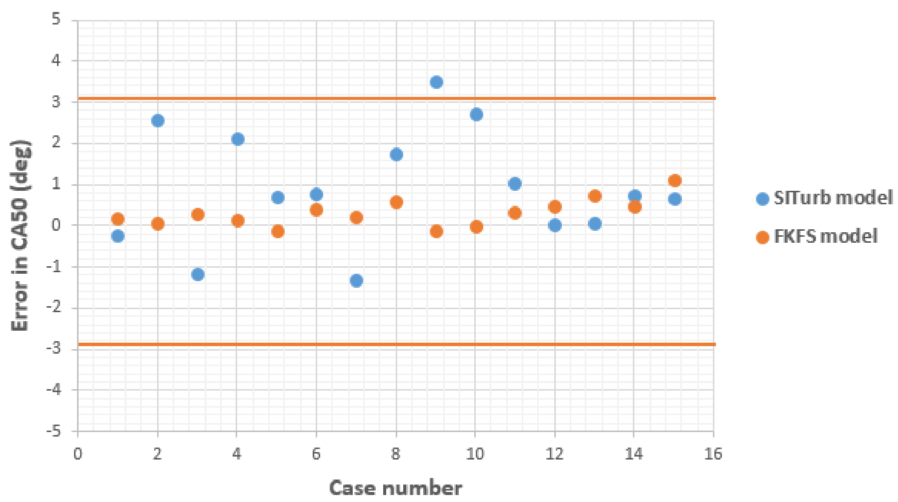

Figure 4.14: CA50 verification for calibrated SITurb and FKFS model

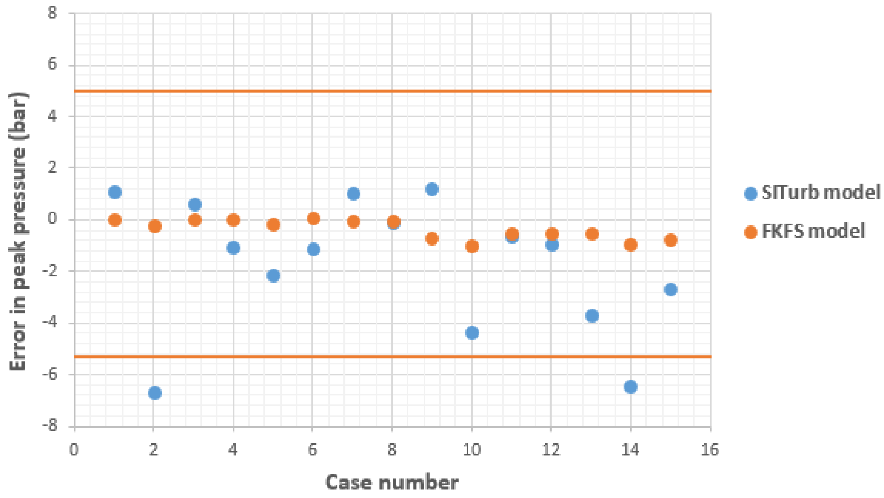

Figure 4.15: Peak pressure verification for calibrated SITurb and FKFS model

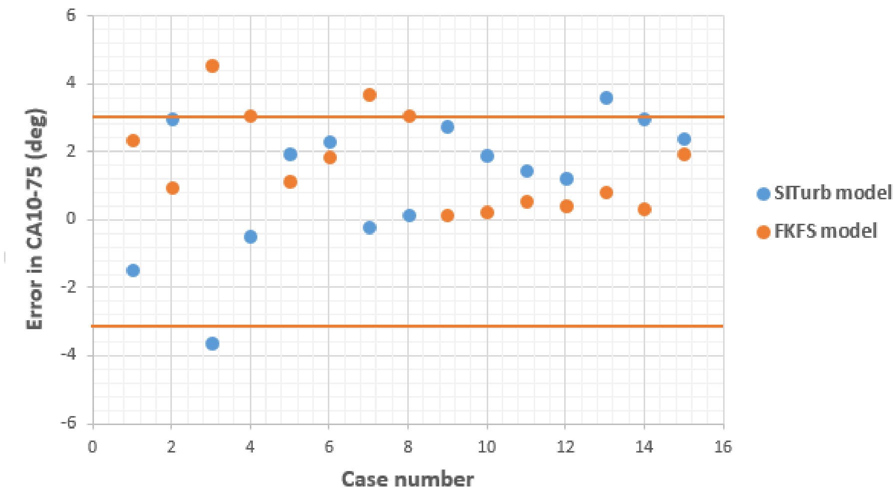

Figure 4.16: CA10-75 verification for calibrated SITurb and FKFS model

4.4 SITurb and FKFS Prediction

The calibrated models predicting capabilities are validated through 4 operating points as explained in section 3.8. The points used for validation includes two points at low load at low speed and two points at high load mid speed and high speed as shown in figure 4.17.

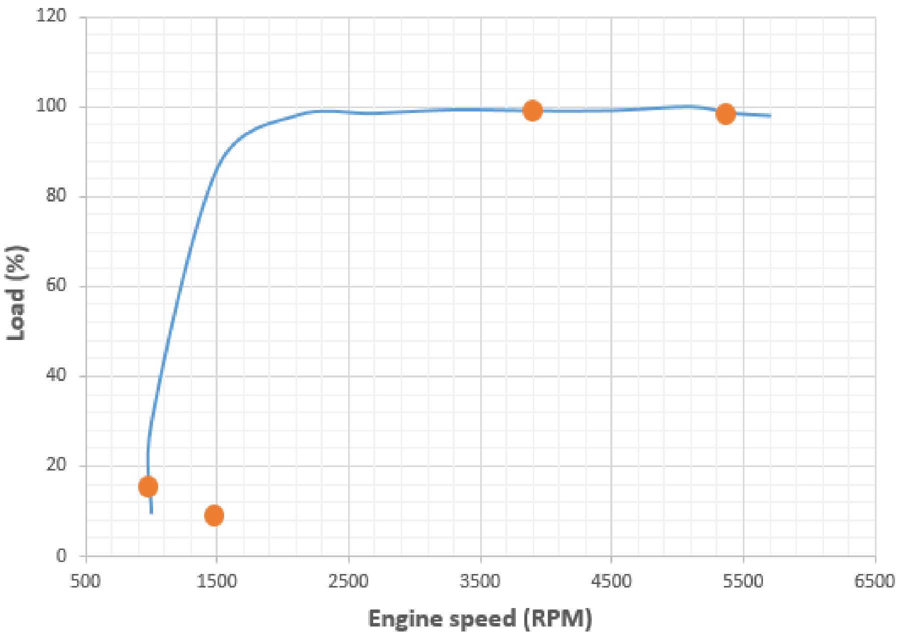

Figure 4.17: Operating points (orange) used for validation of SITurb and FKFS model

The results of the validations points were visually verified for the cylinder pressure and burn rate traces. It is verified against the error limit suggested by Gamma Technologies and FKFS. The results are are as shown in the table 4.2.

The SITurb model prediction was good at part load points as the calibration multipliers obtained were more representing the part load cases. The operating points used for calibrating the model are more populated at part load region and few cases were available at full load region as shown in figure 3.2. Gamma Technologies suggest to have minimum of 25 operating points which is spread evenly across the engine map for good calibration of the model. As explained in section 3.5.2, the pressure trace data obtained for the full load cases was from incorrect calibrated sensors and for part load cases the error was fixed and the data is more reliable hence the calibration for these points yielded better results.

For full load cases the intake and exhaust pressure data was scaled to minimise the error due to fault in sensor calibration at the test rig. With this scaling factor pressure fluctuations varying with load and speed can't be captured. This could be one of the reason for having offset in the simulated and measured cylinder pressure at the compression TDC which can be seen in figure 4.20 and 4.22. For full load cases as shown in from figure 4.20 to 4.23, apart from the offset near the compression TDC the prediction in the combustion region is satisfactorily accepting.

Other possible explanations to the offset near compression TDC is, Blow-by could also be one possible explanation as certain amount of volume and pressure is lost due to the escape of combustion products through piston rings and head gasket. There were problem pertaining to worn head gaskets in the test rig. Blow-by cannot be estimated as it is not measured in the test rig at VCC. A dynamic compression ratio varies for each operating point with speed and load. In this model, an average compression ratio is used for calibrating the model and to eliminate the possible suspicion of variation of the results obtained from calibration.

The pressure at compression TDC in FKFS has a small variation from measured cylinder pressure compared to SITurb model. This is due to availability of a sophisticated tool in FKFS called 100% iteration which can be used while running TPA only. The functioning of this tools starts at IVC by measuring the difference in simulated and measured cylinder pressure. The difference is compensated by pegging the intake pressure to the cylinder pressure. The total energy is calculated by forward run (total heat release is calculated from measured cylinder pressure). To get the same total energy, the FKFS model varies exactly the amount of fuel and air mass required to match that total energy. Subsequently in the reverse run, an accurate cylinder pressure curve is simulated which matches really well when compared to measured cylinder pressure and is shown in Figure 4.4.

Table 4.2: SITurb model error limit verification for validation points

Parameter	Error limit	1000	1500	3900	5400
IMEP (%)	+/-5	1.08	0.027	0.08	-1.61
CA50 (deg)	+/-3	0.62	0.61	-0.35	0.43
Peak pressure (bar)	+/-5	0.64	2.22	-2.28	-1.93
CA10-75	+/-3	-3.1	3.4	0.55	0.56

Table 4.3: FKFS model error limit verification for validation points

Parameter	Error limit	1000	1500	3900	5400
IMEP (%)	+/-5	0.15	-3.71	-3.3	-4.66
CA50 (deg)	+/-3	0.2	3.22	0.59	0.64
Peak pressure (bar)	+/-5	-0.34	-4.69	-0.31	-0.22
CA10-75	+/-3	-2.34	3.49	2.68	-3.14

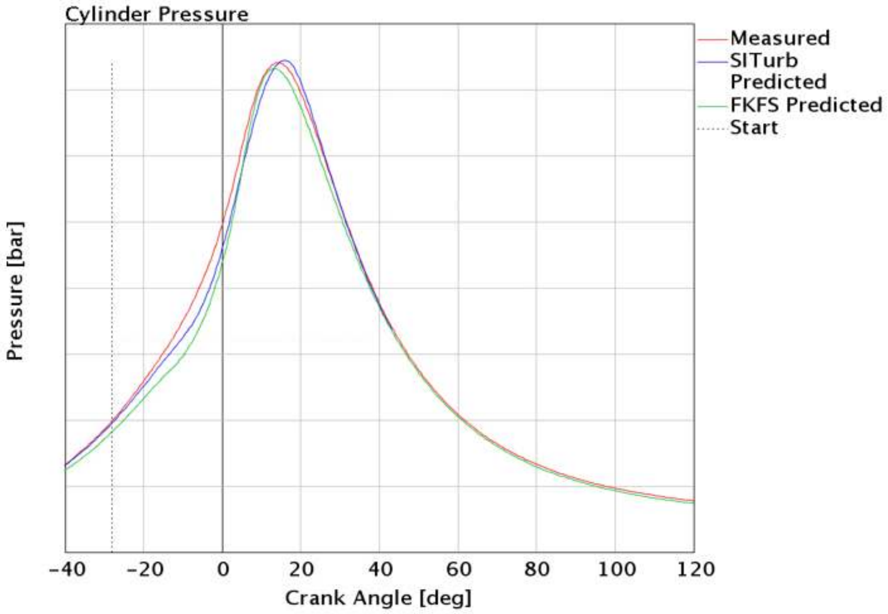

Figure 4.18: Cylinder pressure prediction for low load point

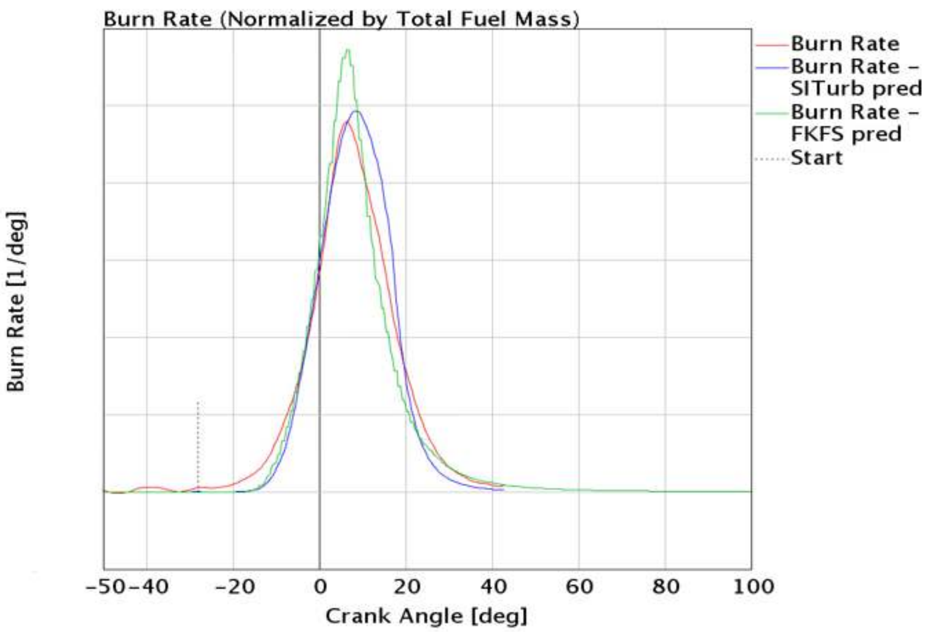

Figure 4.19: Burn rate prediction for low load point

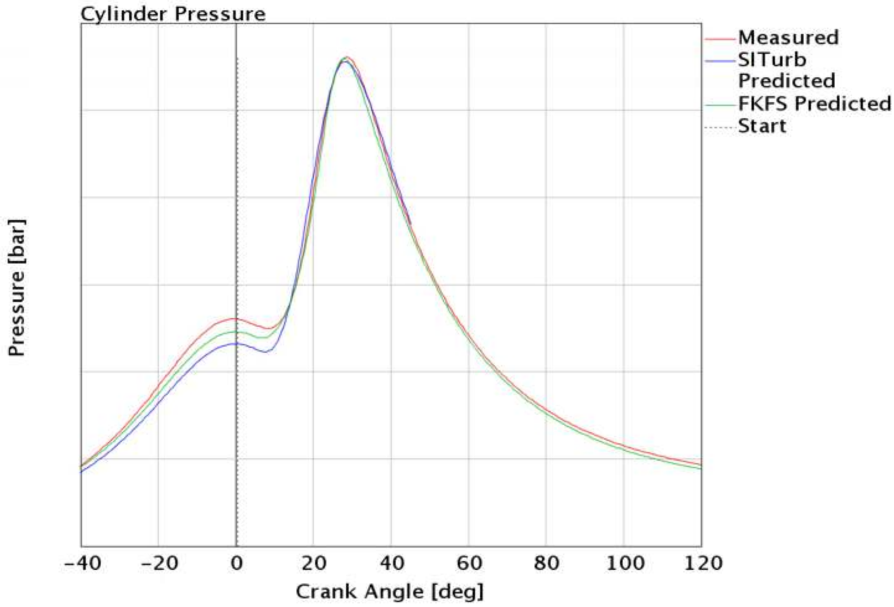

Figure 4.20: Cylinder pressure prediction for mid load point

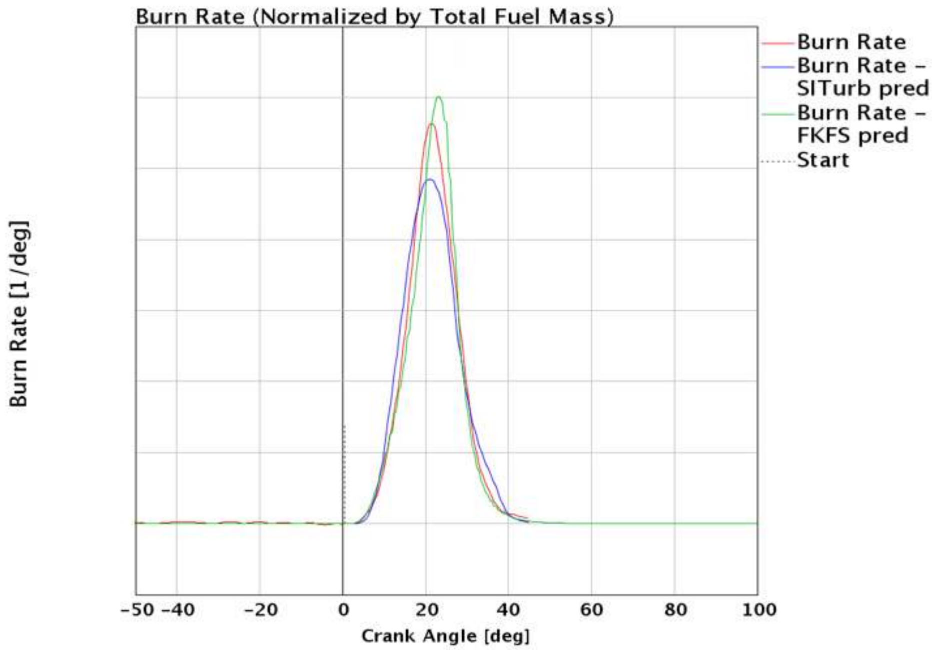

Figure 4.21: Burn rate prediction for mid load point

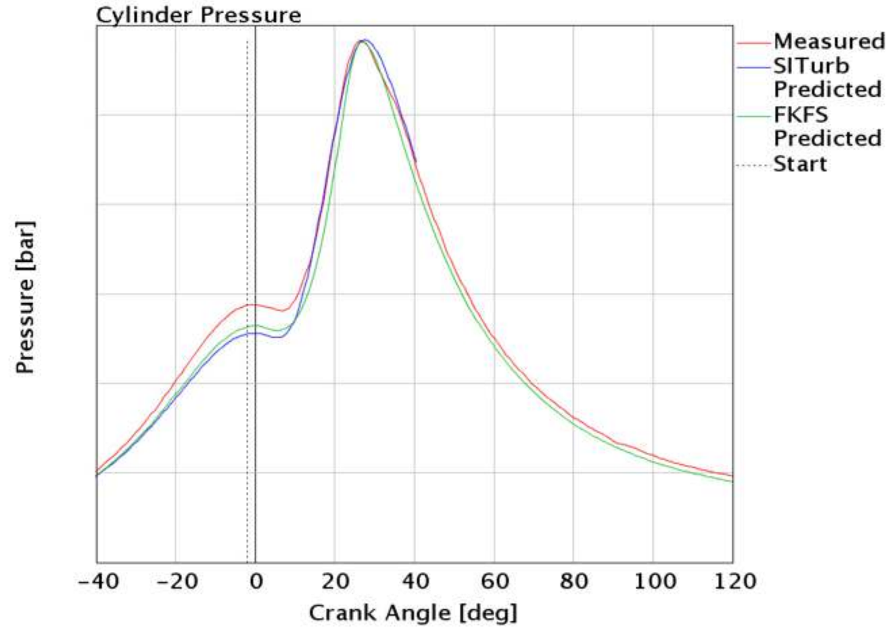

Figure 4.22: Cylinder pressure prediction for high load point

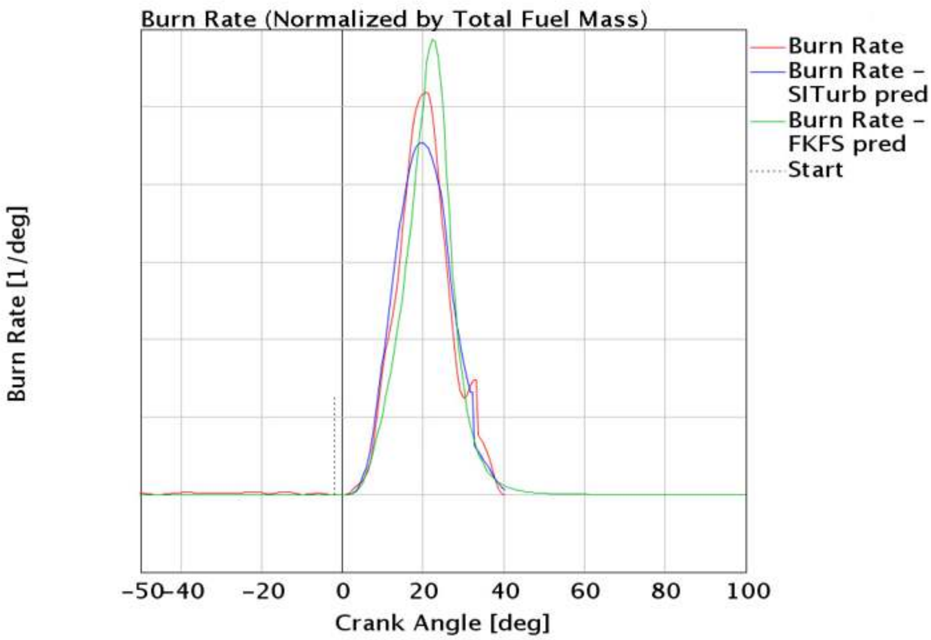

Figure 4.23: Burn rate prediction for high load point

4.5 Differences between SITurb and FKFS Predictive combustion models

This section discusses about the differences between SITurb and FKFS combustion calibration model for Gasoline combustion calibration. The differences are as discussed below,

- Required number Operating points for Calibration
  - **FKFS** - This model requires about 12-14 operating points spread across full load and part load region of the engine.
  - **SITurb** - It requires about 25-200 operating points spread across evenly on the complete engine map.

- Data Filtering
  - **FKFS** - The data filtering has to be performed externally either in AVL Concerto or Matlab
  - **SITurb** - The data filtering for pressure can be done using one of the low pass filter that is built in GT-Power (IEEE10, IEEE24, IEEE54).

- Importing of data into the model
  - **FKFS** - The FKFS model requires that the cylinder pressure has to be in the ASCII format.
  - **SITurb** - This model accepts cylinder pressure in excel or in ASCII formats.

- Input parameters
  - **FKFS** - Crank angle of 50% burn fraction is given as input from which the spark/ignition timing is back calculated from where the combustion process starts.
  - **SITurb** - It uses the spark timing to start the combustion process.

- Zero line correction
  - **FKFS** - This has a tool that automatically corrects the pegging error at IVC.
  - **SITurb** - An automatic pressure shifting tool is available, but the results were not as accurate as seen in FKFS cylinder.

- Cylinder Wall Heat Transfer Modeling
  - **FKFS** - This model offers a number of heat transfer models-Bargende, Woschni, Woschni-Huber and Hohenberg.
  - **SITurb** - SITurb offers only two wall heat transfer model-Woschni and Woschni-GT.

- Energy Balance - 100% Iteration
  - **FKFS** - Automatic Energy balancing option is available in FKFS during TPA and is discussed in section 4.4.
  - **SITurb** - This option is not available for SITurb.

- Heat Transfer Multipliers in the cylinder
  - **FKFS** - It has two heat transfer multipliers, one for low pressure cycle and another for high pressure cycle.
  - **SITurb** - It has an overall heat transfer multiplier in the cylinder.

- CPOA Model
  - **FKFS** - No model for verification of TPA results
  - **SITurb** - CPOA model can be used as a method of verification for TPA results obtained.

- Parameter Input for Calibration Model
  - **FKFS** - No separate inputs are required to setup the calibration model. There is an option called "Optimization", just type yes in the column and run the model.
  - **SITurb** - Results such as volumetric efficiency and trapping ratio along with piston temperatures are to be setup in an isolated model with injector, cylinder and crank-train.

- Calibration Model
  - **FKFS** - This model has only one parameter/multiplier to calibrate.
  - **SITurb** - This has four multipliers to calibrate the model.

- Turbulence Modeling
  - **FKFS** - The model accounts the complete course of gas exchange and thereby enabling the combustion model to react to variable valve timing and tumble.
  - **SITurb** - SITurb model does not take into account the variation in variable valve timing.

- Simulation Time
  - **FKFS** - The simulation time required to calibrate the model is very less.
  - **SITurb** - The time required to calibrate using this model is high compared to FKFS model. Sometimes it might take upto one day to obtain the calibration multipliers.

- Optimization tool
  - **FKFS** - In joint optimization, calibrating the complete model to obtain a single multiplier values that fits all the engine operating points. In single optimization, each point on the engine map has its own calibration multiplier value.

  - **SITurb** - The SITurb model follows direct optimization method where the calibration multipliers are optimized based on the key operating parameters.

- Effect of multipliers burn rate or cylinder pressure curve
  - **FKFS** - Only one parameter which affects the combustion, quite difficult to focus on the problem when analysis is superficial.
  - **SITurb** - Four parameters define each of the factors affecting the cylinder pressure or the burn rate, analysis can be made based on each multiplier to check for the error.

- Calibration Results extraction
  - **FKFS** - The result files are stored external to GT and it needs to be extracted from the cluster which is real hassle.
  - **SITurb** - The results from this model is readily available in GT-Post.

- Process of Post Processing
  - **FKFS** - As the result files for this model is stored externally, additional effort has to be put in to compare the results of PTA, calibration and validation.
  - **SITurb** - All the results are available in GT-Post.

5 CONCLUSION

The aim was to compare, calibrate and validate two predictive combustion models, FKFS and SITurb in GT-Power. Data from single cylinder engine data were acquired for 19 operating points from an engine test bed at VCC. The engine data acquired was focused on the parameters, VVT, tumble, 50% burn fraction and compression ratio. The data obtained was checked for standard error limits using the tool AVL Concerto. An error in the measurement of the crank angle resolved (also known as fast sensor measurement) of the intake and exhaust pressures was observed and this error was due to improper calibration of the pressure sensors at intake and exhaust runners. This error was fixed by using a correction/scaling factor, where the average of the fast sensor measurement data was scaled to the pressure measurement taken at the intake and exhaust vessel. A linear trend was observed which increased with increasing engine speed. Apart from the pressures, the rest of the data were within the acceptable range. The scaling method is just an approximation method, but to obtain accurate predicting model, the pressure sensors have to be studied or calibrated to withstand the thermal shock and other external factors.

After fixing the intake and exhaust pressure, a PTA was performed for FKFS and SITurb combustion model to obtain the trapped quantities and the volumetric efficiency. The TPA results obtained proved to be satisfactory for both the combustion models. The next step was to perform the calibration for the two models, FKFS and SITurb and was performed using 15 engine operating points. The results obtained from calibration were checked for error limits and were satisfactory except for the cylinder pressure at compression TDC where there was a small deviation. This was explained by the improper data measurement of intake pressure as the sensors were not calibrated. This problem is verified as this behavior is persistent in both the models.

Finally, the validation of the combustion model is performed by using the calibration multipliers for the 4 validation points. The prediction results for SITurb and FKFS were sufficiently equal and satisfactory except for the cylinder pressure at compression TDC. The possible reason for this variation can be explained with the same reason of improper data measurement of intake pressure as provided for calibration.

To conclude, both the calibrated models predicts with good accuracy for the entire range of engine map. The major advantage of using FKFS model is that it can handle variation in VVT, tumble and slightly the compression ratio. whereas, the SITurb model does not account for variation in VVT and tumble. The time required to calibrate FKFS model is significantly less compared to SITurb model calibration and also in terms of the number of operating points required to calibrate the model.

From the comparison of the results, it might not be able to conclude which of the two models have the better predicting capabilities. The differences exists in the logic and accuracy of each sub part of modeling methods that are implemented. For better models comparison similar methods are to be used in calculating the key parameters. Due to time constraints such detailed modeling variation could not been analysed.

6 FUTURE SCOPE

In this section the possible improvements and the future work at Volvo Cars that can be carried out for the thesis is discussed. The recommendations include test cell, engine model, SITurb and FKFS models for improving the quality of calibration.

6.1 Test cell

Pressure sensors

The sensors used at the test rig should be calibrated correctly before using at the test rig. As explained in the section 2.3.1, incorrect pegging of the sensors led to false pressure measurement at the intake and the exhaust manifolds and this resulted in wrong pressure trace analysis results. As explained in section 2.3.3, thermal shock affects the measurement values at high temperatures leading to false results. Hence sensors used at the exhaust should be verified that it is compensated for thermal shock error.

The pressure sensors positioning in the intake and exhaust manifold is very important because the instantaneous pressure measurement should be able to capture the pressure fluctuations in the manifold which affects the flow parameter results. Hence the position of the sensors should be selected after the geometry change in the manifolds because the change in geometry will affect the pressure values and this should be captured. Also positioning of the thermocouples and lambda sensors should be after the pressure sensors position because if they are placed before they might cause interference for the flow which will lead to difference in pressure measurement which in-turn lead to error in pressure trace analysis

TDC Sensor

Encoder error significantly affects the simulation results and currently at the test rig motored pressure curve method is used verify the encoder which is not the accurate method. Also TDC sensor method is easier to perform compared to motored pressure method. Hence as explained in section 2.3.1, TDC sensors should be used for encoder phasing which is the more accurate way of doing. Though TDC sensors are expensive and sensitive to handle it is recommended for better results.

Lambda measurement

The different lambda measurement done at the test rig, one is direct measurement in the exhaust using a lambda sensor and another one calculated using the emission values. Before using these lambda values in the simulation they should be verified as there might be chances of error due to malfunction of lambda senor or due to error in the formula used in concerto to calculate the lambda value. Hence the different lambda measurements done should be verified against each other.

Spark timing recording

Correct spark timing is very important while calibrating the SITurb model. Spark timing and start of combustion values obtained from the test rig can be wrong due to calibration faults. Hence recording the right spark time is very important in the test rig, so it is recommended to verify the spark timings from the ECU and also the formula used to calculate CA50 in AVL Concerto.

Valve maximum lift opening position

The maximum lift position for variable valve timing settings used in test rig was unknown and it was difficult to find the exact values of it through the simulations because of uncertainty in results and there was no air flow data from the test rig. Hence verifying the results against the test data was difficult. The maximum opening position of the valves should be provided by the valves supplier for obtaining correct simulation results.

Air flow measurement

Air flow rate values from the test are very important because during TPA simulated air flow rate values should be almost same as the measured value to have correct combustion inside the cylinder and this affects all the parameters. Since the air flow rate values were not available from the rig it was calculated from the lambda and the flow rate values available from the test rig. It is recommended to use air flow meter at the test rig for air flow measurement.

6.2 Engine model

Geometry and Material properties

The modelling of the engine in GT-Power, all the dimensions such as the thickness of the cylinder, ports, piston, valve etc and also the material properties of different components. Some of the values are assumed with default values because of the data unavailability. Hence before modeling the engine in GT-Power all necessary data should be gathered from all the departments at VCC so that the model yields accurate results.

Fuel and coolant properties

The correct fuel and coolant properties should be used in the model as it affects the combustion efficiency and volumetric efficiency during simulation. Hence fuel and coolant properties required for GT-Power modelling should be acquired from the suppliers.

Injector model

In the thesis injector rate map used was from generation 1 engine, since the results were satisfactory it was continued with it but the injector rate map of actual injector used at the test rig should be used because small difference in fuel injection quantity would cause error in the results. Hence the injector rate map from the injector suppliers should be acquired for modeling.

Cooling Water Pump

Water pump used for water cooling circulation speed map is required to define in the temperature solver for the cylinder and for the thesis general speed map was used. Since the pump speed varies with speed and load correct pump speed map should be defined in the solver for more accurate results.

Blow-by

Blow-by is not measured in the test rig but the value affects the compression ratio during the simulation as explained in section 3.5.4. It is recommended to include the blow-by in the model which reduces the time in adjusting the compression ratio for simulation and also yield more accurate results.

6.3 SITurb Calibration and Validation

TPA was performed but the validation of the TPA results wasn't performed due to time constraints and also due to issues in the test rig caused delay in data acquisition. Hence it is recommended to perform Cylinder only pressure analysis to validate the residual fractions and trapped quantities obtained from TPA as correct.

Gamma technologies suggest to use minimum of 25 operating points spread evenly across the engine map for good calibration. But if the there is time constraint that it can't be ran at all region of engine map and only at real time operating region such as mid load at low speed and high load and high speed, if possible the calibration algorithm should be updated to handle these scenarios or try to reduce the number of points to be used for calibration to save the time.

The multipliers used for the SITurb calibration primarily works on optimizing the burn rate but other key parameters such IMEP, CA50 etc should also be considered as function for optimization because this would help to reach optimized results quickly.

In the thesis model is not calibrated to predict knock due to time constraint and its strongly recommended to calibrate the model for knock calibration as well since in gasoline engines knock sensitivity is one of the very important factor to predict.

6.4 FKFS model calibration and Validation

FKFS calibration model predicts the burn rate and cylinder pressure accurately. The model is highly sophisticated and has advanced options for calibration of combustion, knock and emissions. Due to time restrictions, knock and estimation of emissions could not be performed. It could be recommended to further calibrate the model for knock and emissions and to see how the combustion prediction varies with these changes.

Advanced options for performing PTA and Optimizer are available in FKFS combustion model. Exploring all the available option might yield much better results than what is obtained in this thesis. A detailed investigation can be performed by calibrating the model for all the engine operating points and studying the results would be interesting how it varies from the results that are obtained in this thesis.

It is a suggestion to FKFS to provide an option for filtering the data in the model instead of performing it externally. When the simulations are performed in LINUX by running on distributed cluster, the result files are stored in the cluster and these files have to be manually fetched from the cluster and this is a very tedious process. It would be helpful if FKFS could upgrade their model which can fetch the results from the cluster automatically and include them in GT-Post so that the results can be analysed easily.

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A

Appendix 1

Matlab script used for filtering the cylinder pressure using Savitzky-golay filter

```
clc
clear all
close all

file = load('1000_1.txt');
X = [file(:,1) file(:,2)];

filter = sgolayfilt(X,11,101);

figure(1)
plot(X(:,1),X(:,2),'r')
 hold on;
plot(filter(:,1),filter(:,2),'b')

xlabel('Crank Angle [deg]')
ylabel('Pressure [bar]')
legend('Pressure before Filtering','Pressure after Filtering')
```

Figure A.1: Matlab script for Svatizky-golay filter

Matlab script for verifying the pegging error

```
% intake and exhaust pressures
raw_in_ex_pres = xlsread( '1000_1_INTAKE_EXHAUST');
raw_cyl_pres = load('1000_1.txt');

in_pres = [raw_in_ex_pres(:,1) raw_in_ex_pres(:,3)];
ex_pres = [raw_in_ex_pres(:,1) raw_in_ex_pres(:,2)];

intake = raw_in_ex_pres(:,3);

%cylinder pressure
cyl = [raw_cyl_pres(:,1) raw_cyl_pres(:,2)];
cyl2 = raw_cyl_pres(:,2);

x = -175;
x1 = -185;

%Finding the points on the graph to check pegging error at -185 & -175
index1 = find(cyl(:,1)==-185);
index = find(cyl(:,1)==-175);
Y_cyl = cyl2(index);
Y_cyl1 = cyl2(index1);
Y_intake = intake(index);
Y_intake1 = intake(index1);

%Plots of all three pressures
figure()
% plot(in_pres(:,1),in_pres(:,2),'r',ex_pres(:,1),ex_pres(:,2),'b')

plot(in_pres(:,1),in_pres(:,2),'r','LineWidth',2)
hold on;
plot(cyl(:,1),cyl(:,2),'g','LineWidth',2)
hold on
line([x x],ylim)
hold on;
line([x1 x1],ylim)
hold on;
plot(x,Y_cyl,'o',x1,Y_cyl1,'x')
hold on;
plot(x,Y_intake,'v',x1,Y_intake1,'s')
legend('Intake Manifold Pressure','Cylinder Pressure')
axis([-200 -120 0 5])
xlabel('Crank Angle [deg]','FontSize',20)
ylabel('Pressures [bar]','FontSize',20)
title('Cylinde Pressure, Intake Manifold Pressure against Crank angle','FontSize',20)

% Difference in values at -185 & -175 btw intake and cylinder pressure
diff_175 = abs(Y_cyl-Y_intake)*1000
diff1_185 = abs(Y_cyl1-Y_intake1)*1000
```

Figure A.2: Matlab script for pegging error verification