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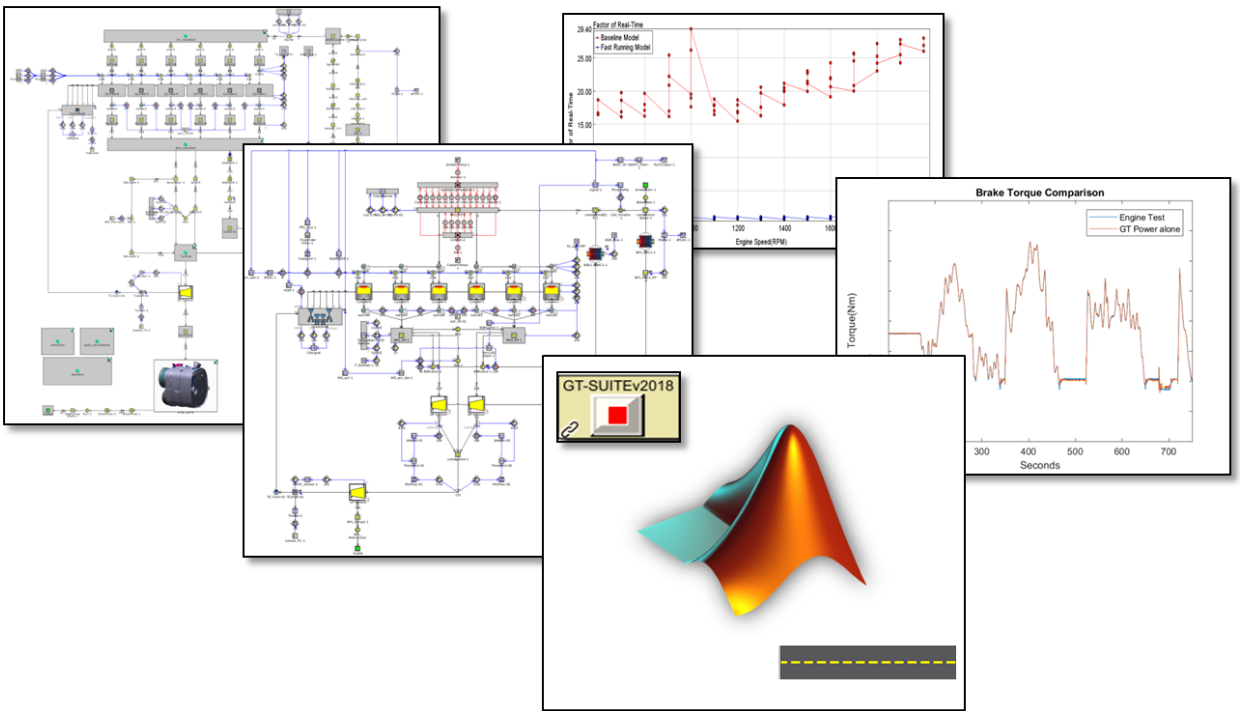


# Real Time Fast Running Engine Modelling in GT Power

## Development of the Virtual Drivetrain Simulation Environment

Master's thesis in Automotive engineering

SHASHI SHEKAR TIPPUR CHANDRASHEKAR

Department of Mechanics and Maritime Sciences
CHALMERS UNIVERSITY OF TECHNOLOGY
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Cover: Picture of the method development for Real Time GT Power and Matlab/Simulink transient co-simulation.

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# Abstract

This thesis work deals with conversion of existing engine model to a Real Time Fast Running Model (FRM) in GT Power for the Volvo Trucks Heavy Duty Diesel Engine. The existing model is a high fidelity engine model which is good to run steady state simulation, but when it comes to transients simulation, it is difficult and slower. The GT Power engine model is a physics based model that provides detailed information such as heat transfers, pressures, temperature etc. These models are not completely accurate models, but it replicates the details of the process that is involved. The Global Simulation Platform (GSP) is a tool developed by Volvo Trucks in Matlab Simulink which can perform vehicle level simulations with transient drive cycles using empirical engine plant model.

The potential of GSP can be further enhanced by replacing a empirical engine plant model with real time detailed engine models. This integrated model is advantageous in evaluating newer emission or fuel reduction concepts and also minimizing the cost for repetitive physical testing in case of a component change or a concept change, all of this can be done at the early stages of a "New Project" or even during the current project.

This report presents FRM technique as a solution to get the model running faster and the accuracy of the FRM is compared and verified with the results from the Volvo Engine Test Cell to put in to effect.

The report also involves the study of Miller cycle inlet valve closure profiles with increased Peak Cylinder Pressure and Compression Ratio. A simple hybrid system in the FRM was designed to present the application capability of the FRM in GSP.

Keywords: FRM, GSP, Volvo Trucks, Engine model, Engine test cell, Real Time, Hybrid, Power Boost.

# Abbreviations

**GT Power** - **G**amma **T**echnologies
**FRM** - **F**ast **R**unning **M**odel
**GSP** - **G**lobal **S**imulation **P**latform
**RT** - **R**eal **T**ime
**PCP** - **P**eak **C**ylinder **P**ressure
**CR** - **C**ompression **R**atio
**EU** - **E**uropean **U**nion
**EM1** and **EM2** - **E**xhaust **M**anifolds
**HTM** - **H**eat **T**ransfer **M**ultiplier
**EMS** - **E**ngine **M**anagement **S**ystem
**ECU** - **E**ngine **C**ontrol **u**nit
**EGR** - **E**xhaust **G**as **R**ecirculation
**CAC** - **C**harged **A**ir **Cooler**
**HiL** - **H**ardware in the **L**oop
**SiL** - **S**oftware in the **L**oop
**BLB** - **B**orås **L**andvetter **B**orås
**PLM** - **P**art **L**oad **M**ap
**TC** - **T**urbo **C**ompound
**LP** and **HP** - **L**ow **P**ressure and **H**igh **P**ressure
**PID** - **P**roportional **I**ntegral **D**ifferential
**VEB** - **V**olvo **E**ngine **B**raking
**VCB** - **V**olvo **C**ompression **B**rakes
**SOI** - **S**tart **O**f **I**njection
**BSFC** - **B**rake **S**pecific **F**uel **C**onsumption
**BMEP** - **B**rake **M**ean **E**ffective **P**ressure
**SOC** - **S**tate **O**f **C**harge
**EATS** - **E**xhaust **A**fter-**T**reatment **S**ystem
**vEMS** - **v**irtual **E**ngine **M**anagement **S**ystem
**vTECU** - **v**irtual **T**ransmission **C**ontrol **U**nit
**ICE** - **I**nternal **C**ombustion **E**ngine
**EM** - **E**lectric **M**achine
**CAD** - **C**rank **A**ngle **D**egrees
**LIVC** - **L**ate **I**nlet **V**alve **C**losure
**EIVC** - **E**arly **I**nlet **V**alve **C**losure
**DoE** - **D**esign of **E**xperiments

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# Contents

# 1

# Introduction

The automotive industry is taking a rapid growth with newer regulations for improving fuel efficiency and reduction in emissions. According to the European Commission, heavy duty vehicles contribute to 6% of the total EU CO2 emissions which needs to be reduced [1].

Volvo Trucks are developing engines which produce less emissions with a better fuel consumption. The subject which is treated here is the year 2025 fuel consumption and emission regulation, where the emission for example the amount of CO2 emission is expected to produce 15% lower than what it is producing in 2019 and will further continue to decrease in 2030 to 30% lower than in 2019 [1]. Hybridization is a proven technology which can be useful in reducing the emissions and improving the fuel efficiency with power-split strategies. Therefore, future powertrains require better methods for example hybridization can be an option to achieve these expected values, one way to do that for a heavy duty truck it is necessary to start with providing assist from the electric machine at necessary driving conditions. It is well known that the efficiency of the electric motor is higher than the ICE and therefore it can be said that EM is used as a main source of energy for the future powertrains, but they have their own limitations for it.

In recent years advances are enabling new technologies for recovering the lost energy, hybridization, the fuel cell concept and also to combine the characteristics of both spark-ignition and compression-ignition engines [3]. For a truck engine, there are many such engineering advancements on the subsystems as well, but the investigation on which is the best acceptable concept is still ongoing and simulations are required to investigate different design concepts and possibly narrowed down to one solution is of high priority and here, the engine model becomes the most essential part of the simulation.

Modern engines have advanced features incorporated to it like, engine braking, different stages of turbochargers, waste gate, turbo compounding techniques, free valve technology and many more, all these concepts requires a robust control logic to be operated at efficient points of the engine. The EMS (Engine Management System), Hybrid modes supervisory controller, waste gate actuation, EGR valve controllers, Fuelling controllers for torque and rail pressure are examples of the use the control systems for enhanced engine performance.

The evaluation of the controllers to serve its operation needs to be considered and modelling these controllers and testing them is beneficial in determining the behaviour of these controllers before implementing them in the physical model. GT Power is one such simulation tool for modelling different engine components along with the implementation of different controllers for efficient engine operation. The GT Power engine model used in this thesis work is a detailed model with all the features like fuel, waste-gate and EGR controllers similar to the EMS incorporated to the engines in production.

To achieve the numbers as described by the 2019 European legislation, it is necessary to have newer and faster methods for simulation. Because, through simulations, it is possible to evaluate different design concepts and model different subsystems and test them. These detailed engine models should not only run on steady state operating points, but also be able to run with time dependent transient inputs. Lastly, the models should be capable of running transient drive cycle simulations and communicate with other CAE softwares. Time for simulation of complex models are higher and the technique which was used in this thesis work to make the complex model run faster is the Fast Running Model (FRM) which will be elaborated in the upcoming chapters.

## 1.1 Background Study

This section involves the details of how FRM is used by other competitors in the industry and for what application it is used.

1. Ford uses FRM in a Hardware-in-the-Loop (HiL) rig for global driver behaviour prediction which is affected by chassis, powertrain and other hardware systems. They also use FRM for testing complex electromechanical systems [6]. They mainly use FRM for fault diagnosis, where the FRM contains the details of the failed component to be studied and the inputs to this FRM is from a controller in simulink and the data for simulation is either a steady state condition or a transient drive cycle data. Using these inputs, closed loop simulations are carried out and recommendations are given as an output for the studied failed component and also boundary conditions are given for the failed component to be tested in the engine test cell [7].
2. The study by FEV Japan Co., Ltd.,Japan present a solution by conversion of detailed models to simplified FRM plant model for HiL using co-simulation methodology. A quantitative analysis of the results are made referring to the bench test measurements. After the model is tested for steady and transient performance, it is then integrated to the FEV's HiL xMOD platform . Results of the simulations says that the pressure pulsations within the system are well captured and is mandatory for the determination of volumetric efficiency, turbocharger operation and EGR distribution. The conclusions from this article is that the engine thermodynamics and the controller behaviours have been validated with engine test bench data and the real-time capability of the model has been proven [8].
3. General motors Co. has developed FRM for HiL testing called dSPACE simulator and is verified with bench test results. The test results show that the HIL co-simulation stays consistent for most of the variables and are under 0.7-0.9 real time factor between 1000-5000 RPM. The result of this work says that the

steady state results was able to reach with Rapid-Control Prototyping (RCP) or GT Power inbuilt controllers. The transient states are achieved using different control algorithms. The main purpose of co-simulation was achieved which beneficial for RCP development and ECU verification, the inconsistancies in performance data was observed which could be because of simplifications and discretization [9].

4. Volvo Penta uses FRM as a Hardware-in-the-loop (HiL) rig which is called as VIRTEC and it stands for VIRtual TEst Cell. This VIRTEC has a computer which has the Fast Running Model which will act as an engine plant model, this FRM receives input signals from a physical ECU (Engine Control Unit) and the output from the FRM is connected to the hardwares such as injectors, valves etc. through the actuators. This system acts as an alternative to the test rig and the cost of this is 10-15% of the cost of the test rig.

## 1.2 Problem Description

Currently at Volvo Trucks, the simulation methodology for the implementation of a new engine is that the engine is modelled in GT Power along with the concept and hardware selection. The simulations are run in steady state first as the simulation results will be used for engine screening in the engine test rig and the necessary boundary conditions are provided for complex CFD calculations. These data put together result in a very large matrix that will be used to formulate a empirical or a numerical engine plant model in Matlab/Simulink which will be used for transient drive cycle simulations.

This type of methodology holds good at the later stages of the project. But, if there is a change in the strategy of replacing the existing turbo system or using new injectors which require calibration on the engine at a very early stages of a new project, then this method requires physical testing of the engine to account the changes in the empirical model in Matlab/Simulink. In short a loop is formed before a perfect final empirical engine model is developed for drive cycle simulations. This loop can be avoided by replacing the physical engine tests by Real Time Fast Running Models on one common platform that is Global Simulation Platform (GSP) which is highly beneficial during the early stages of a new project. Hence, the thesis work is about verification and replacement of the empirical engine plant model to a GT Power engine model to run transient drive cycle simulations.

The GT Power engine models are physics based models which can mimic the processes involved in a physical engine. The empirical models have set boundaries based on the results from calculations and are not allowed to extrapolate, GT Power on the other side has the ability to account the changes and tries to extrapolate and shows the true behaviour based on the changes, thereby minimizing the time and the cost of physical testing. The flowchart fig. 1.1 gives a better picture of the method development of the simulation framework.

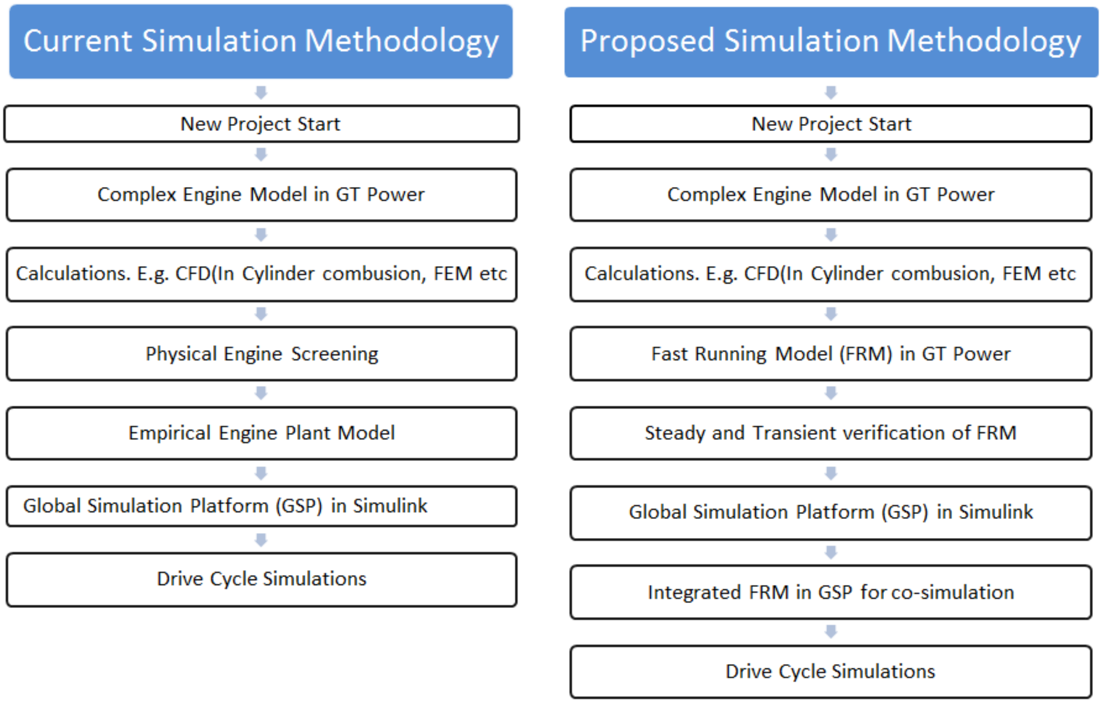



**Figure 1.1:** Difference in simulation methodology between existing and the proposed technique

Fig. 1.1 shows that the start and end point for the implementation of a new engine is the same for both the simulation methodology, but the proposed methodology is capable of avoiding the problem at hand.

## 1.3 Problem Approach

The study aims to establish a simulation methodology by developing Fast Running Engine model that can run transient drive cycles. These models can be used to study the potential of a new design concept(eg.hardware change,calibration,hybrid systems etc) or improvisation to the existing model very early in the design phase.

The work flow to achieve the simulation setup are as follows:

1. Detailed study of the baseline calibrated engine model is necessary for better understanding of the functioning of the engine model.
2. Fast Running Model (FRM) conversion from the baseline complex engine model for accuracy and speed.
    - Reduction of the model complexity leads to inaccurate models. To make the FRM accurate, Calibration and Verification is done for steady state engine operating points.

3. Testing the FRM in GT Power workbench for transient condition.
    - The results from the engine test cell is the simplest way to start the comparison of the developed FRM. The engine test cell contains input signals to the physical engine and the measured outputs.
    - Comparison of different parameters of the engine test measured results and the results from the GT Power model.
4. Integrating the GT Power FRM in a tool called Global Simulation Platform (GSP) in Matlab Simulink.
    - The FRM model in Simulink is an s-function block which reads the required input signals from the Engine Management System (EMS).
    - The comparison of results is done between the baseline GSP model and the integrated GT Power model.
5. A Simple working hybrid system is designed in GT Power FRM with a control logic for the operation of the electric motor and is integrated to GSP.
    - The input for the GT Power block in Matlab Simulink is still read from the EMS.
    - The electric motor is used as an assist to the engine at higher torque values.
    - The functioning of the Electric machine can be changed by altering the control logic in the GT Power model.

Lastly, the thesis work also involves the study of Miller cycle on the baseline complex engine model in GT Power. Miller cycle is defined as earlier or later closing of the intake valves which increases the effective expansion ratio in relation to the compression which allows advantage to be taken of increased charging at higher engine speeds [2]. Therefore an optimization was run for late inlet valve closure profiles to study the merits or de-merits of using the Miller cycle by selecting a few steady state operating points. By using the Miller cycle the effective swept volume is decreased and therefore the mass of air sent into the cylinder must be the same as the mass of air sent previously, this can be done by increasing the pressure in the intake with a slightly larger turbo system.As a result, the new turbo must be matched to the engine.

## 1.4 Scope and Limitations

WHTC is expected to be the driving cycle which will be used for the certification of vehicles. However for this thesis work, a customized drive cycle developed by Volvo Trucks which is called the BLB drive cycle (Borås-Landvetter-Borås) is used for simulation.

The problem will be limited to building FRM in GT Power of the 13L Turbo Compound heavy duty engine which is made to integrate with Matlab Simulink (GSP) to run transient drive cycles. The predictability of the FRM in transient conditions is tested with steady state Part Load Map (PLM) data first and then it is tested for transient operation with the results from the engine test cell. The integrated model is also verified by the results from the given GSP baseline model.

The results from the FRM and real truck test could be another set of comparison which was not done in this thesis work due to time constraints.

A hybrid system that is built on the FRM is based on a simple control logic, built in electric machine and battery, it is not intended to reduce fuel consumption or emission. It is just to show the capability of the FRM for modelling and evaluating different subsystems.

The report will include a minor investigation of the effect of Miller cycle implementation on the existing complex engine model for a few steady state operating points.

# 2
# GT POWER Modelling

GT-Suite is a CAE tool which is developed by Gamma Technologies for engine simulations and the extended version of GT Suite is GT Power. It is a 0D/1D/3D multi physics simulation tool and can be used to model and connect almost all the engine related components. The software consists of a set of libraries which can contribute to almost any type of industry such as mechanical,fluids, electrical, chemical and also controls. With the use of these libraries one can build almost any engineering system in one work bench.

The tool also has built-in advanced features such as DoE, optimization, combining flow volumes, parallel processing ability and many more of such features.

The engine model used in this thesis work is the D13 litre Turbo Compound engine with a simplified after treatment system. This model will serve as the baseline model for FRM simplification and calibration of certain parameters which will be explained in detail. This chapter will give a brief explanation on working of few important parts in the baseline engine model.

## 2.1 GT Power Engine Model

Firstly, let us know a few important things about the engine model used. The engine model used here is the 13 litre turbo compound Heavy Duty Diesel engine of 500hp developed by Volvo Group Trucks Technology. There are four important parts of the engine model.

1. **Charged-Air-Cooler.**
2. **Cooled EGR (Exhaust Gas Recirculation).**
3. **Double-Entry Turbocharger.**
4. **Turbo-compound.**

The principle behind using the Charged-Air-Cooler and the EGR Cooler is when the pressure from the compressor and the exhaust manifold respectively are sufficiently greater, the temperature will also be higher. Higher the temperature, lower will be the density of the air to flow through the pipes or valves. Therefore, with the cooler introduced to its path of flow the temperature will be reduced and the density of air or exhaust gas will be increased.

An internal combustion engine with more than four cylinders can decrease the overall pulse energy and increase the pumping losses. Therefore, multiple entry turbines based on the flow split, the twin-entry and the double-entry designs are used to avoid this unnecessary pumping losses. Here, A double entry turbine can be symmetrically or asymmetrically divided based on the mass flow balance at the turbine rotor. The construction has two scrolls and is often used to split the engine exhaust gas, a typical double entry turbine is as shown in fig. 2.1 [5].

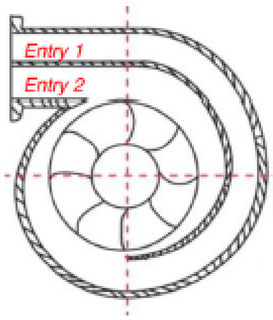



**Figure 2.1:** Double Entry turbocharger[5]

A turbo-compound device in fig. 2.2 uses two sets of turbines, one is the High Pressure (HP) turbine (T1) and the other is the Low Pressure (LP) turbine (T2). The pressure from the exhaust manifold is passed through the twin entry HP turbine which runs the compressor to produce extra boost to the engine. The exit of the HP turbine is passed to the LP turbine where this is connected to the crankshaft of the engine using a simple gear arrangement, which acts an additional power to the engine using the waste heat which was supposed to be sent to the exhaust pipes.

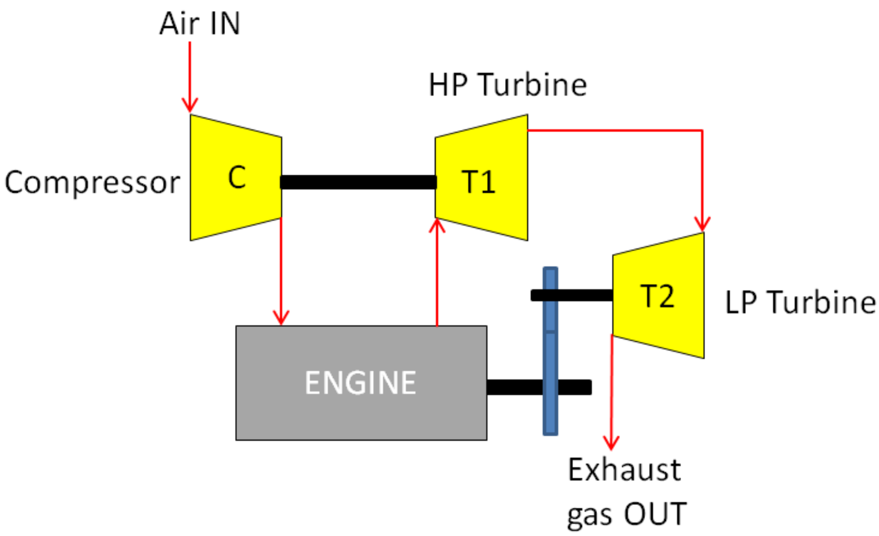



**Figure 2.2:** Turbo-Compound schematic arrangement

Most importantly the model also includes three controllers which are:

1. **Torque controller (Fueling)**
   GT Power has an inbuilt PID controller to control the amount of mass of fuel injected per cycle and the inputs for the PID controller is reference torque [Nm] and speed [RPM] from the crankshaft, the target torque [Nm] value that the engine model is supposed to produce. The controller can also include the Air-to-Fuel ratio limit as an input. With these several inputs the controller processes and send the right fuel quantity per cycle to the injectors.

2. **Waste Gate controller**
   The waste gate controller is also a PID controller for which the inputs are the pressure [kPa] from the exhaust manifold as a reference value and the target pressure [kPa]. The output from the PID controller will be in terms of waste gate valve open percentage, this percentage is then converted to equivalent orifice diameter [mm].

3. **EGR Valve controller**
   The EGR valve controller has three different controllers based on the parameters such as Fi-in, EGR mass fraction, EGR mass flow. These controller inputs are the target values which processes and gives the output in the form of percentage valve lift. The percentage valve lift is converted into equivalent orifice diameter and sent as a signal to the EGR valve for actuation.

4. **Volvo Engine Braking**
   The engine model built has a special feature called engine braking, where there is a resistance to the rotation of the crankshaft which is offered by the engine alone during braking in downhill conditions without applying manual brakes. This system works based on the timing of the exhaust valve lifts. Consider an example of a truck during the downhill drive, at a particular point if there is a braking situation the engine brake will also contribute along with the manual braking. The working principle is that the air is compressed in the compression stroke, here at this point the exhaust valves are lifted to open up where the compressed air escapes out of the cylinder thereby not producing and power from the engine. This creates a resistance to the rotation of the crankshaft and brakes the vehicle.

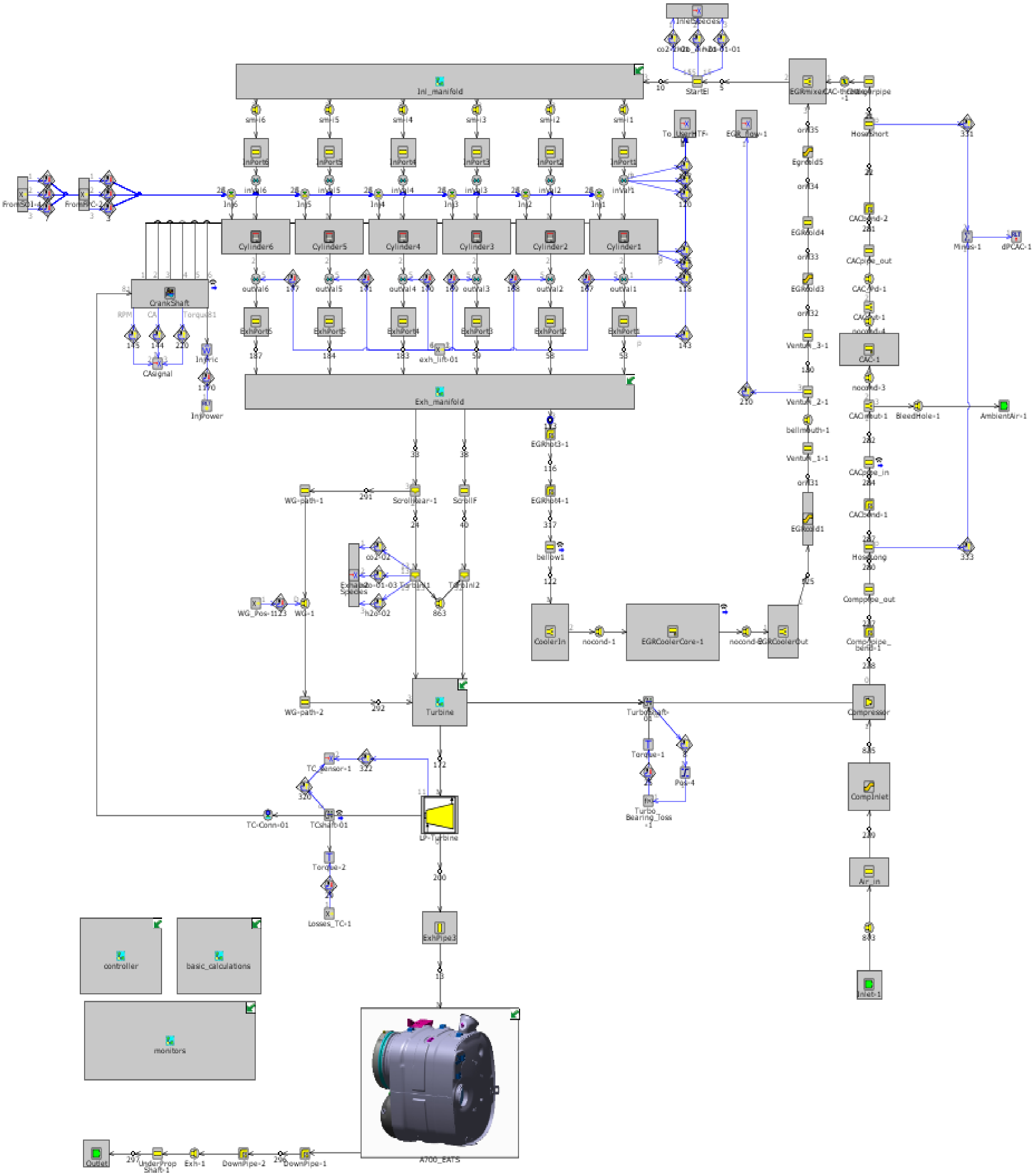



**Figure 2.3:** The detailed engine model in GT Power CAE tool

## 2.2 Fast Running Model (FRM)

Fast running models are fully-physical engine models that are designed specifically to run fast[4]. In fig. 2.3 there are different sub-assemblies which are dissolved into the main workbench to make the FRM conversion easier.

### 2.2.1 Important Definitions

1. **Discretization**
   - A pipe system is a discrete system divided into separate volumes for computation at every time step.
   - **Coarser discretization** – shorter simulation run time and accuracy compromise.
   - **Finer discretization** – Longer simulation run time with reliable accuracy.
2. **Time Step**
   - The change in time in increments where the governing equations will be solved.
3. **Real Time**
   - Real time systems are are systems which respond immediately for a given input signal or it can also be said that there is no noticeable delay between action and effect.

The models with high-fidelity run slower and it becomes difficult to run simulations in a complete system level where transient events can be performed or even when the models has to respond to real time or even faster[4]. In order to speed up the simulations, FRM is a suitable solution. There are two types of simplifying the model which is user dependant where one can choose to simplify the model for accuracy or simplify the model for speed.

The FRMs achieve fast simulation runs by two methods

1. Increasing the simulation time step size.

2. Decreasing the number of calculations per time step [4].

Which means to say that it will combine volumes and makes room for larger time step size by increasing the effective discretization length.The most important graph in conversion of the complex model into a Fast Running Model is the "Factor of real time". By this graph we can keep track of the time to simulation for each operating point after each simplification step. There will be a detailed explanation given on this in the results chapter.

# 3

# Building the Fast Running Engine Model

The previous chapter was about the baseline engine model and how the model works and also some important definitions on Fast Running Model was elaborated. In this chapter will include detailed procedure of building a Fast Running Model. This is a simple method and it usually hold good for any type of the engine model.
The inputs to the engine model are steady state operating points from an excel spreadsheet which is called as Part Load Map (PLM), this data is obtained from the engine test cell. The entire PLM is not used for carrying out the FRM conversion, but a mixture of 29 steady state operating points which include part load and full load operating points are selected manually ranging from 600RPM to 2100RPM along with a point of maximum torque of 2880Nm.

## 3.1 FRM Procedure

The first thing that needs to be done is to run the engine model for 29 selected steady state cases, after simulation the GT post processor provides information on the objects in the model which is restricting the time-step and also the number of subvolumes.

In the Post-Processor, click on the RLT Contour Map-> Pipes/FlowSplits->Flow Control->"Fraction of time-steps limited by parts".

The RLT contour map is also used to determine the number of subvolumes for each pipes that are modelled. They can be seen under RLT Contour Map-> Pipes/FlowSplits->Flow Control->"Number of Subvolumes".

The FRM procedure is started with the method of FRM tags, the entire engine model is divided into different sub system which will be dealt one after the other in an order. The FRM tags are found under Tools->"FRM Tags". The engine model contains the following sub systems and is also shown in fig. 3.1.

1. **Exhaust Manifold**
2. **Exhaust Pipes**
3. **Intake Manifold**
4. **Intake Pipes**
5. **Charged-Air-Cooler Pipes/Boost Pipes**
6. **EGR Cooler Pipes**

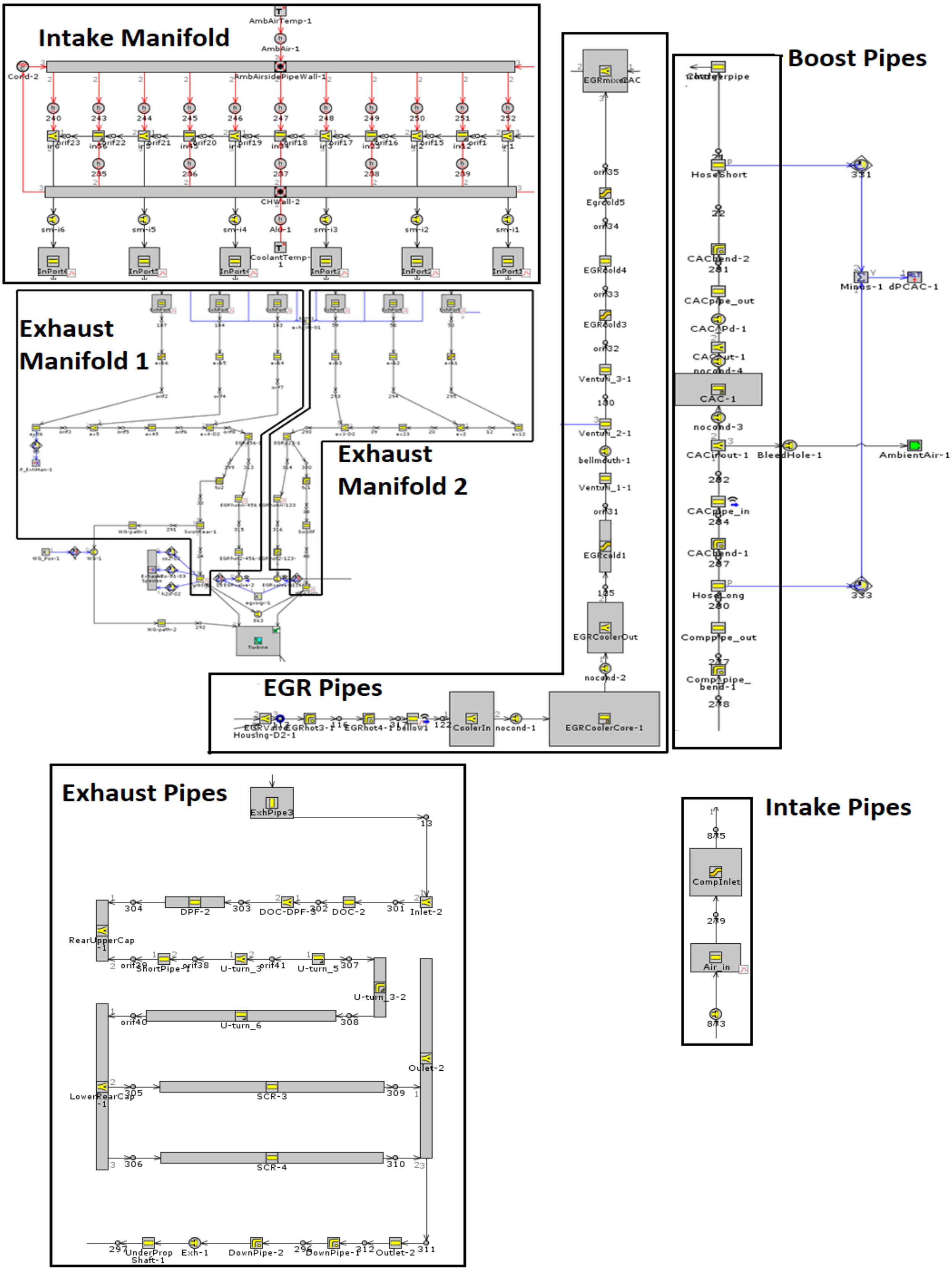



**Figure 3.1:** Picture showing only the tagged sub systems

There is an option between simplifying for accuracy and simplifying for speed, at first it is necessary to simplify the model for accuracy. However, both the options is carried out using the following three major steps.

**1. MODEL REDUCTION**
**2. CALIBRATION**
**3. RESULT**

Moving back to the tagging part, the engine is model is selected with the subsystems mentioned earlier and is divided as shown in fig. 3.1.

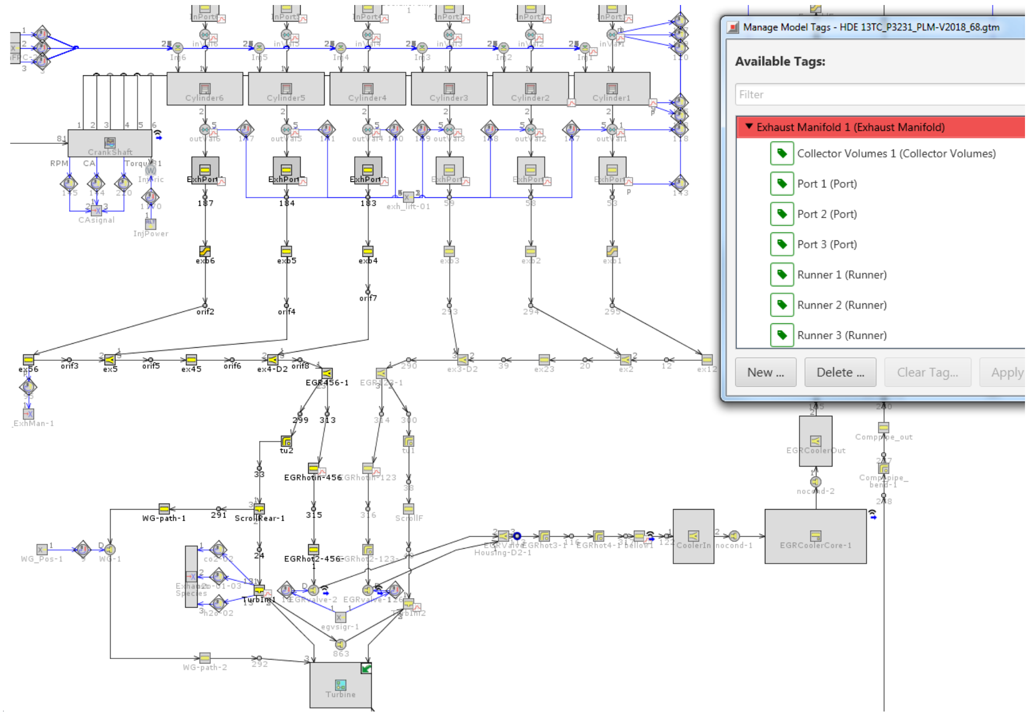



**Figure 3.2:** Picture showing the tagging window for the exhaust manifold 1

It is important to apply certain settings before going to the actual simplifications, this is done by clicking on the "Run Setup"->"FlowControl" folder and then double clicking on the "ExplicitControl" and then go to "Time Step and Solution Control Object" tab to edit and set the "Maximum Time Step to 720 CAD and then click OK and close the Run Setup [4].

Once the entire model has been tagged and the settings are applied, it is time to launch the FRM Converter under Tools. This will prompt a window to create the FRM project, here the tagged model will be specified in the "Select the Baseline Detailed Model" block and by pressing "Next" will reload the model specified for FRM Conversion. This will also prompt a dialog box as seen in fig. 3.3 called "Accuracy Checking", here it is necessary to select the RLT's to keep track on the accuracy of the model during the process of the conversion, the advantage of this is that it is possible to specify tolerances on the entered RLT's, it can be an absolute value or a percentage of the value for the tolerances.

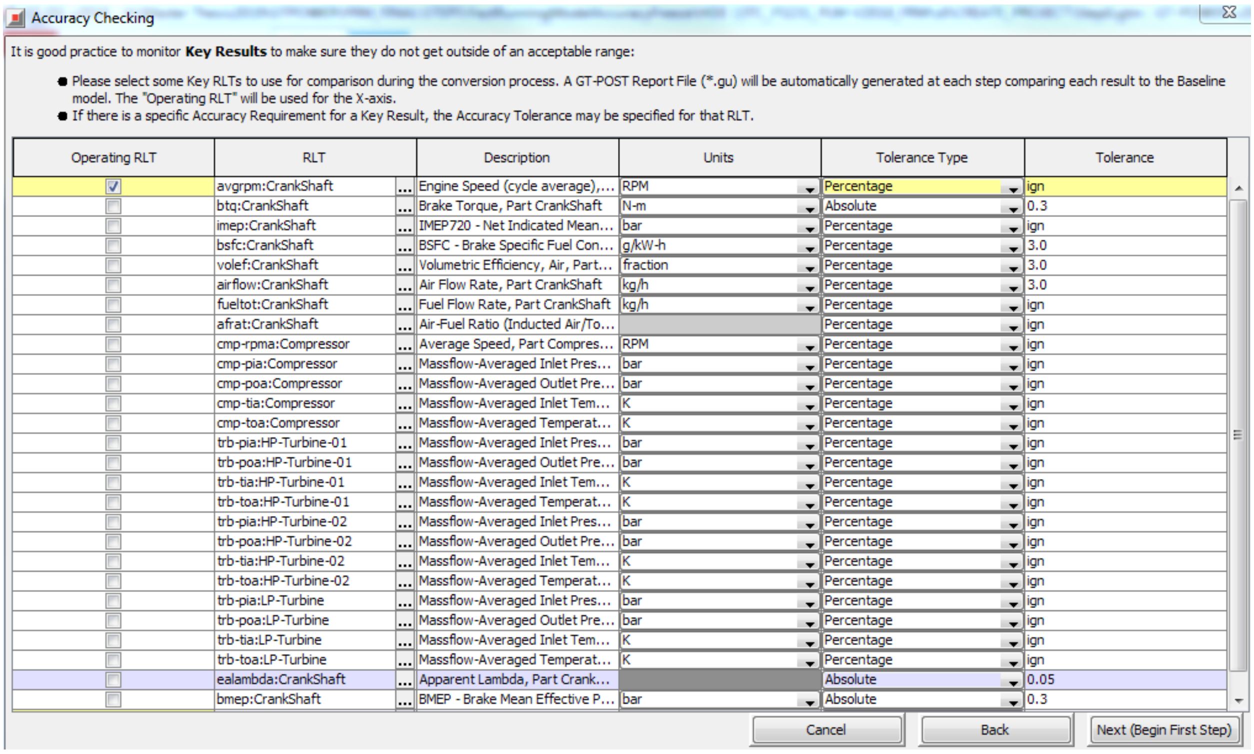



**Figure 3.3:** Accuracy Chart

### 3.1.1 Exhaust Manifold

The first subsystem to simplify is the exhaust manifold because usually for a high fidelity models, the gas velocities are usually higher, which restricts the time step[4]. Fig. 3.2 shows the tagging window of the Exhaust Manifold 1 (EM1), where it can be carefully observed that the Exhaust Manifold 1 (EM1) is highlighted while all other objects are not. This happens when the objects are grouped to one particular sub system. Likewise, the same is carried out for all other sub systems.

#### 3.1.1.1 Model reduction

The exhaust manifold is divided into two distinct parts one to the right and the other to the left which accounts three cylinders exhaust flow which actually is used to describe the flow into the two entries of the double entry turbine, orifice mouth is used in between the two exhaust manifolds which replicates as the wall between the two entries to the turbine. The tagging of the exhaust manifold should be done separately for the left (EM1) and the right (EM2) section. First the model reduction is done for the EM1 and the effect of tagging the exhaust manifold will reduce the subsystem into one flowsplit volume and pipe volumes for the three different cylinders on the EM1 which currently act as exhaust ports. The same procedure is repeated again on the right section which is also called EM2.

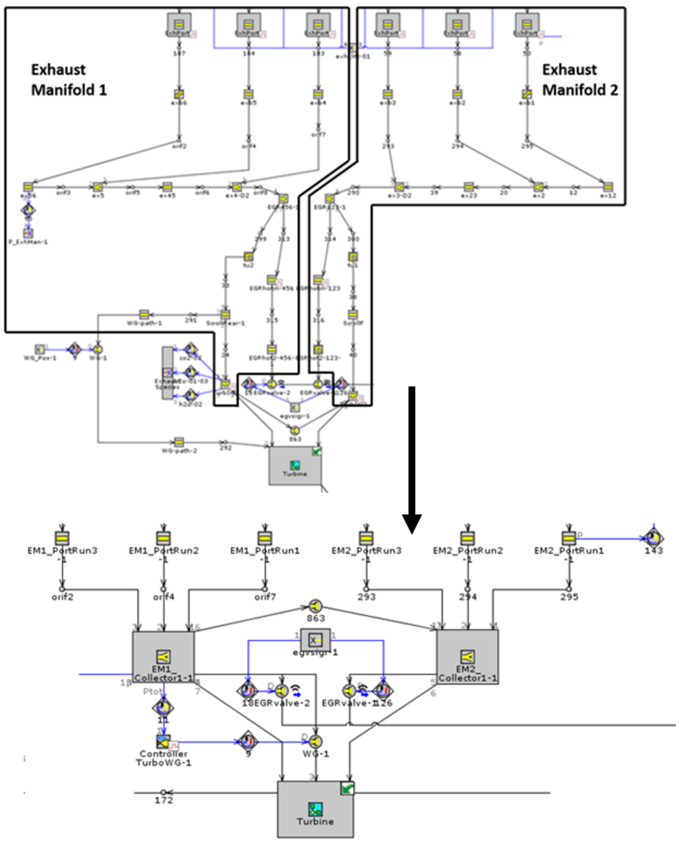



**Figure 3.4:** Picture showing the simplification of exhaust manifold

#### 3.1.1.2 Reduced model calibration

Once the model is reduced as seen in fig. 3.4, it is necessary to calibrate the model for better accuracy and good predictability. The parameters which is used for calibration are Heat Transfer Multiplier (HTM) and the Friction Multiplier (FM) entered inside the flowsplit volume. The one parameter which is given more importance is the HTM which will appear as [HTM] in the Case Setup along with the discretization length which will be automatically created as [dxe_FRM] and the value will be set to 300mm. The calibration conducted is optimization in the design optimizer turned on using Target approach, where the targeted value is the "Turbine Inlet Temperature [K]". The calibration step for the EM1 will be skipped and the model will be reduced on the right also to create EM2. Now that both the left and the right sections are completely reduced, one single HTM is used on both the EM1 and EM2, by doing this both EM1 and EM2 will be calibrated at once. The HTM value is initially set to unity and then its value is varied between 0 and 3 and the resolution to be 1% in the design optimizer. The calibration is carried out for one case which has the highest mass flow rate and an optimized value on the HTM is obtained, this HTM value is applied for all the cases and the design optimizer is turned off and the model is run. The calibration results are obtained in GT Post and the details of which will be shown in the Results chapter of the report.

### 3.1.2 Exhaust Pipes

It is always better to simplify exhaust pipes after the exhaust manifold so that the exhaust side of the engine model is completely simplified and it is also the most time taking part of the FRM conversion. The model reduction of exhaust pipes are as shown in fig. 3.5.

#### 3.1.2.1 Model reduction

The subsystem of the engine model which can be seen after the exit of the Low Pressure (LP) turbine which is also considered as the Exhaust After Treatment System (EATS) by Volvo GTT. In fig. 2.3 it can be seen that the EATS is a sub-assembly within GT Power and this sub-assembly has been dissolved to the main workbench to make it easy for the FRM conversion process.

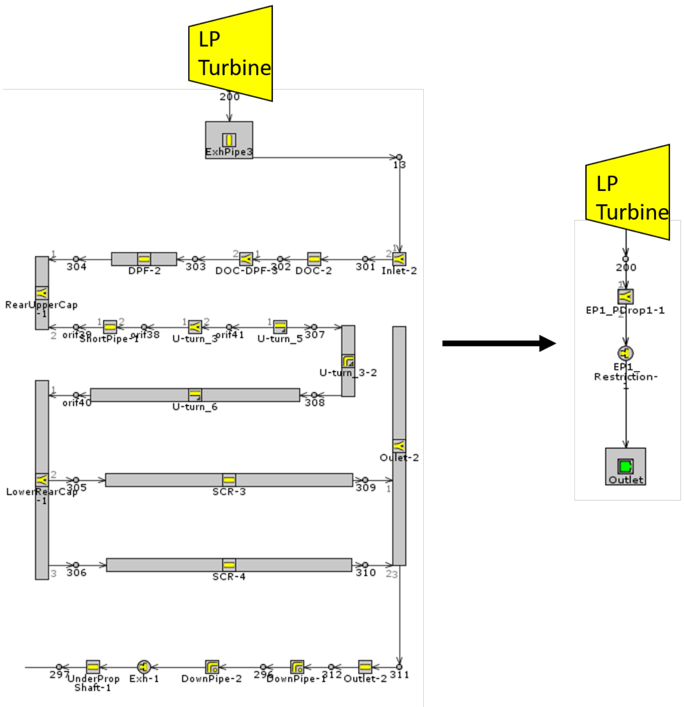



**Figure 3.5:** Picture showing the simplification of exhaust pipes

#### 3.1.2.2 Reduced model calibration

It is a regular practice to calibrate the simplified model for better predictability and accuracy. The initial steps for calibration is same as the ones mentioned in section 3.1.1.2 but, here the targeted parameter will now be the "LP Turbine Outlet Pressure" and since the calibration parameter is pressure, it is always the diameter of the orifice which is to be set as the "Parameter to be varied" with an initial value of 30mm and its value is varied between 20mm and 50mm with the resolution to be 1% in the design optimizer. It is worthwhile to mention that the orifice diameter may vary other than the mentioned numbers, it is purely dependant on the engine model.

### 3.1.3 Intake Manifold

The intake manifold is the third subsystem for the FRM conversion and the inputs to the intake manifolds are "Charger-Pipe" from the boosting subsystem and the "EGR Mixer" from the Cooled EGR subsystem.

#### 3.1.3.1 Model reduction

The entire intake manifold is tagged to form one subsystem unlike the exhaust manifold which were divided into three cylinders per section. The reduced model will have one flowsplit volume along with the intake ports which is shown in fig. 3.6.

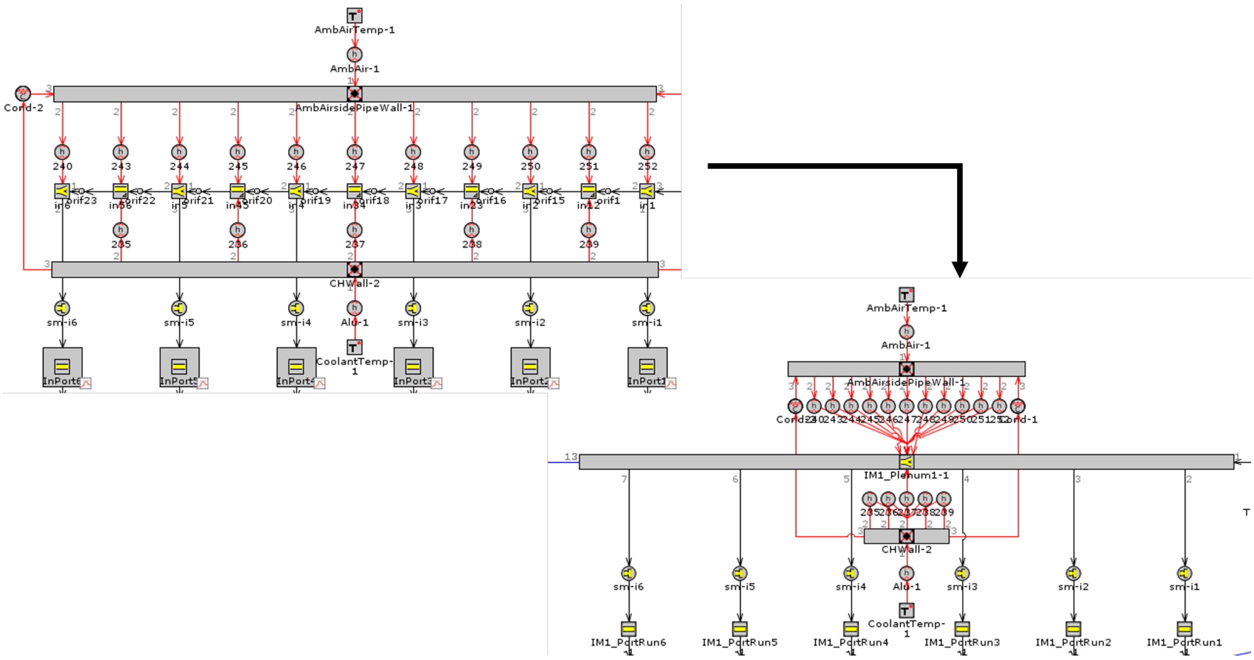


**Figure 3.6:** Picture showing the simplification of Intake Manifold

#### 3.1.3.2 Reduced model calibration

Here, in this case the targeted parameter in the design optimizer is the "Compressor Outlet Pressure" and the "Parameter to be varied" will be the [HTM] on the Intake Manifold with an initial guess of 1, the discretization length will be automatically added as 200mm by the FRM converter as [DXI_FRM]. A value on the HTM will be added to all the 21 cases and the model is run and results are read on the GT POST.

### 3.1.4 Boost Pipes

The boost pipes include the Charged-Air-Cooler pipes and the "HeatExchanger-Conn" which will help to maintain the intercooler outlet temperature.

#### 3.1.4.1 Model reduction

The object BP_CAC1-2 is the "HeatExchangerConn" and "Imposed Fluid Temperature will be replaced by the "RLTDependenceXYZ" object "CAC-HTR" which was used earlier in the CAC block from the baseline model. In the object BP1_CAC1-1 the "Imposed Wall Temperature" is set to 300K. The simplified model from the baseline model is as shown in fig. 3.7.

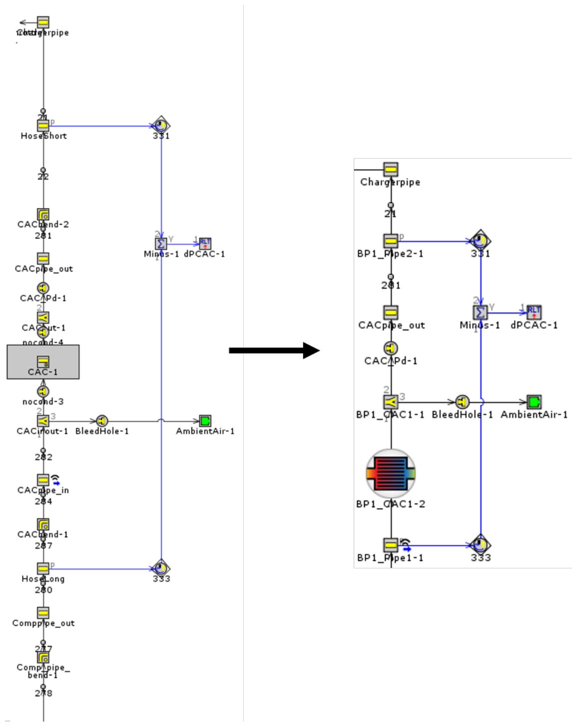



**Figure 3.7:** Picture showing the simplification of Boost Pipes

#### 3.1.4.2 Reduced model calibration

Here, in this case the targeted parameter in the design optimizer is the "Compressor Outlet Pressure" and the "Parameter to be varied" will be the diameter of the "HeatExchangerConn" object with an initial guess of 30mm and this is varied between 20mm and 40mm. It is once again worthwhile to mention that the diameter value may vary other than the mentioned numbers, it is purely dependant on the engine model. "Parameter to be varied" will be the diameter of the "HeatExchangerConn" object with an initial guess of 30mm and this is varied between 20mm and 40mm. The EGR pipes are placed adjacent to the boost pipes and the concept of cooled EGR is quite similar to the CAC where an EGR cooler is used instead of a CAC.

### 3.1.5 EGR Pipes

#### 3.1.5.1 Model reduction

The same explanation of CAC holds for the EGR Cooler where the "HeatExchangerConn" BP_CAC1-2 is replaced by the object EGR1_EGRC1-2 and the "RLTDependenceXYZ" object here is "EGR-coolerHTR" which was used earlier in the EGR Cooler Core block from the baseline model. The simplified model from the baseline model is as shown in fig. 3.8.

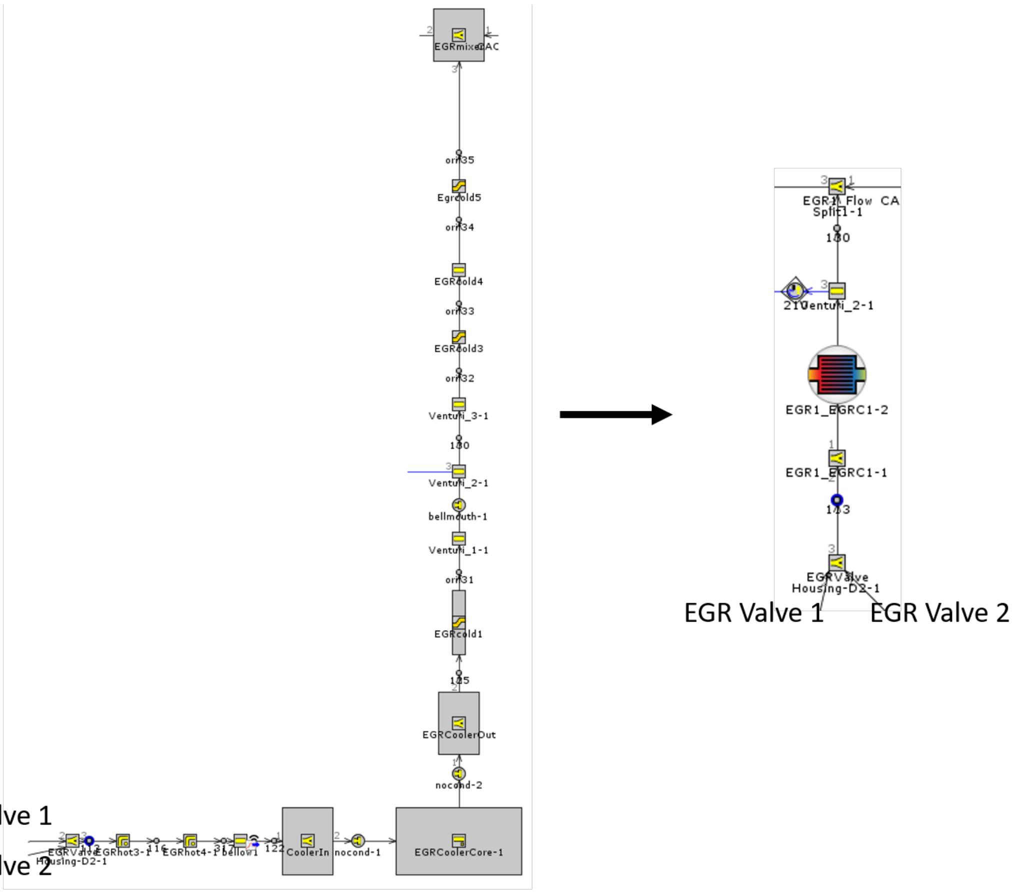



**Figure 3.8:** Picture showing the simplification of EGR Pipes

#### 3.1.5.2 Reduced model calibration

The calibration step is also similar to the calibration of the Boost pipes. Once change is that the targeted parameter in the design optimizer is "Turbine Inlet Temperature [K]" and the "Parameter to be varied" will be the diameter of the "HeatExchangerConn" object with an initial guess of 30mm and this is varied between 20mm and 60mm. The same rule of thumb holds for the selection of the diameters for the calibration.

### 3.1.6 Intake Pipes

The last and the least time taking subsystem to simplify is the intake pipes.

#### 3.1.6.1 Model reduction

The baseline model of the intake pipe contains a pipe-table which acts as a combination of several other pipes contained in it, the number of pipes inside the pipe table object "CompInlet" are four in number. This will be reduced to one big pipe volume names IP1_Pipe1-1. The reduced form of the pipe volumes is shown in fig. 3.9.

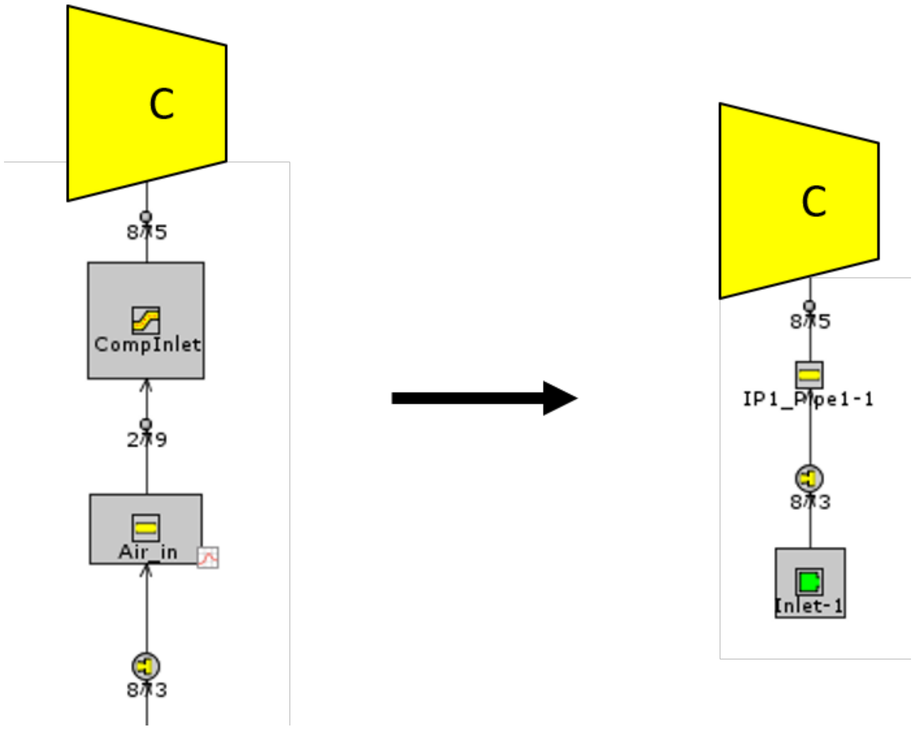



**Figure 3.9:** Picture showing the simplification of Intake Pipes

#### 3.1.6.2 Reduced model calibration

For the Intake pipes the targeted parameter in the design optimizer is the "Compressor Inlet Pressure" and the "Parameter to be varied" will be the orifice Hole Diameter with an initial guess of 90mm which will then be varied between 60mm and 110mm.

**General Note on Calibration**

For all the above subsystems, the case in the case setup used for calibration will always be the maximum air flow operating point and once the optimization is successful, the optimized value is set for all other cases with the cases turned on and the model is run to check the accuracy and predictability of the model compared with the baseline model.

### 3.1.7 Simplifying for Speed

Once after the model is simplified for accuracy, it is necessary to check the model if it is running close to Real Time (RT) or at RT and the objects which restrict the timestep. For a high-fidelity model in this case the model will not be running at RT and hence simplification for speed will be the next step in the FRM conversion procedure. In can be seen in fig. 3.10 that there are many pipe members and orifices in the intake manifold, boost pipes and the exhaust manifold side, these members will actually restrict the timestep and it is necessary to simplify those flow objects.

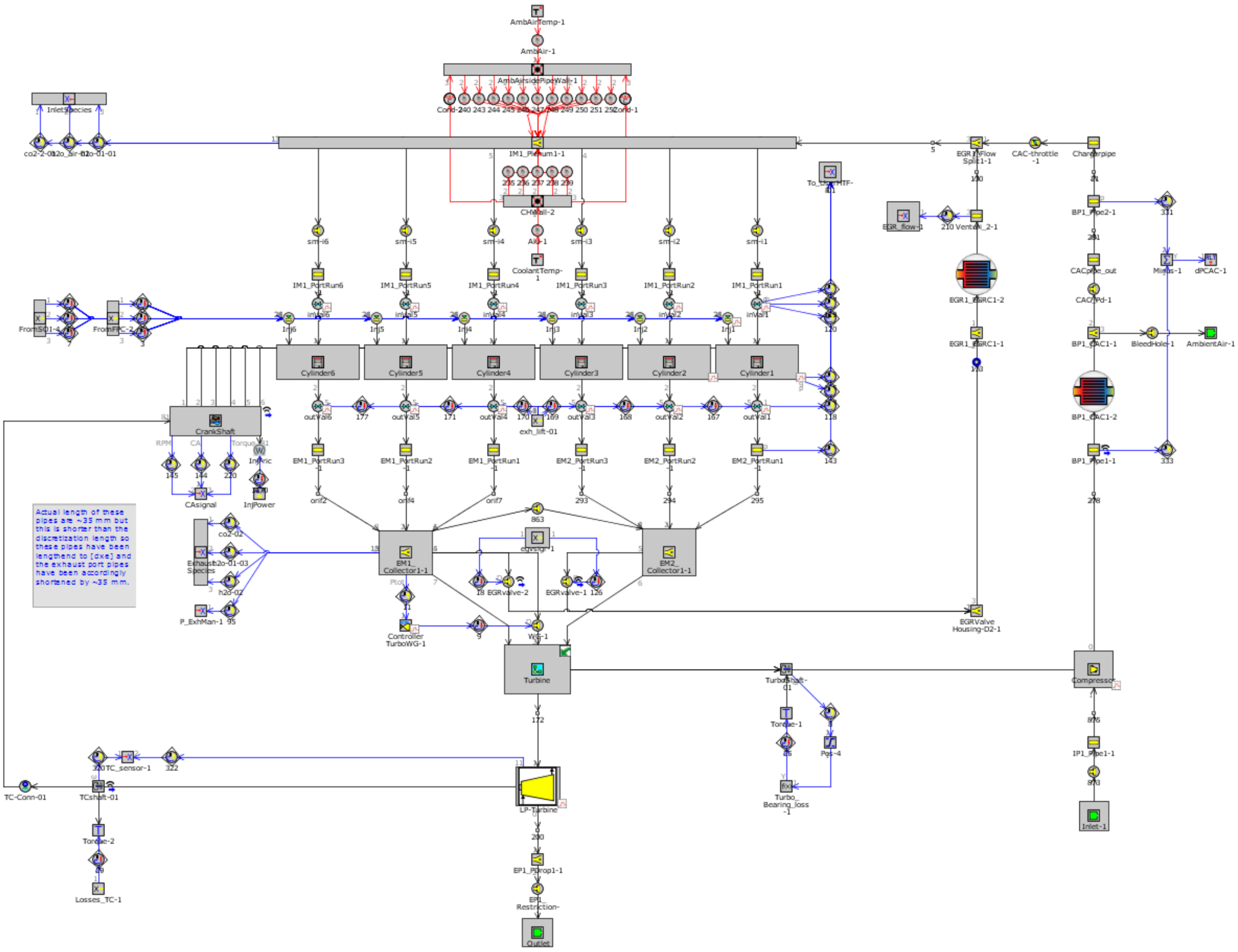


**Figure 3.10:** Picture showing the simplified version of the engine model from the base model

The FRM converter tool is launched and "simplify for speed" is selected, now the number of subsystems to simplify will be reduced to three which are:

1. **Exhaust Manifold**
2. **Intake Manifold**
3. **Charged-Air-Cooler Pipes/Boost Pipes**

The entire FRM conversion procedure which has been explained in the above sections and sub-sections will be implemented the three subsystems mentioned. Once after the model reduction and calibration of each subsystem is done, the simplified model for speed will be as shown in fig. 3.11.

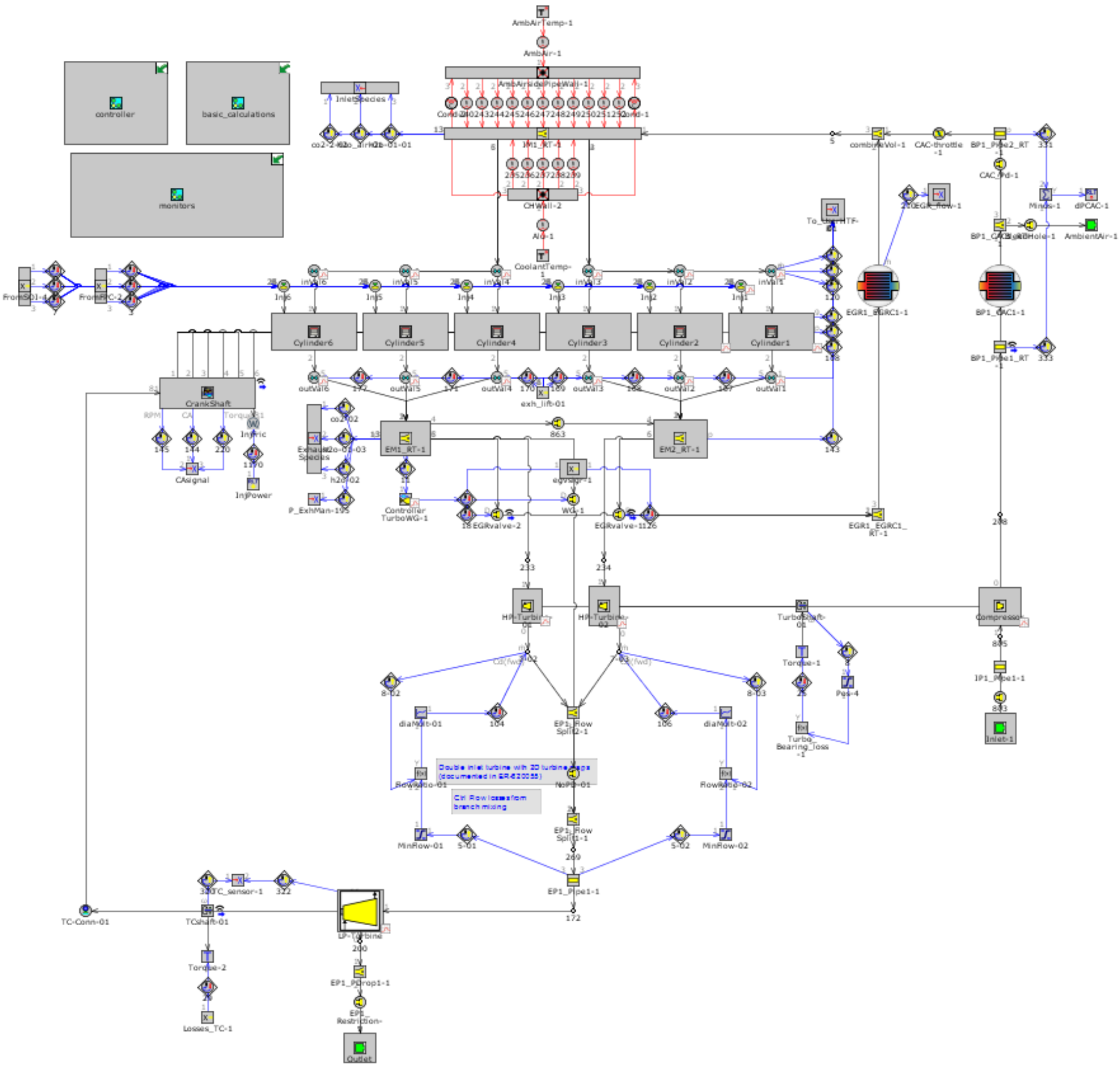



**Figure 3.11:** Picture showing the engine model which is simplified for speed

It can now be seen that the engine model has only one flowsplit volume for the intake manifold and two separate flowsplit volume for the left and right section of the exhaust manifold and also the number of flow objects on the EGR pipes,boost pipe subsystems have been decreased. It can also be seen in fig. 3.11 that the High Pressure (HP) Turbine is a sub assembly which has been dissolved to the main workbench as it has flow volumes which restrict the timestep. Once the simplification and the calibration steps are completed the model is run again and the results are checked in the GT POST to see if there are still some flow volumes restricting the timestep. In fig. 3.12, the parts which are actually restricting the timestep can be seen by going to "RLT contour Map"->"Pipes/Flowsplits"->"Flow Control"->"Fraction of timesteps restricted by parts", the colour coding will be from minimum to maximum where the minimum being the blue and the maximum being the red. In same fig. 3.12 the circles marked with red and green show that how much percentage of the timestep is actually being restricted by the parts. The one with the red circle is a pipe object inside the HP Turbine subassembly and is limiting the timestep by 99.5% and needs to be simplified, that is the reason for dissolving the HP Turbine subassembly in fig. 3.11. A similar explanation for the % holds for the circle in the green.

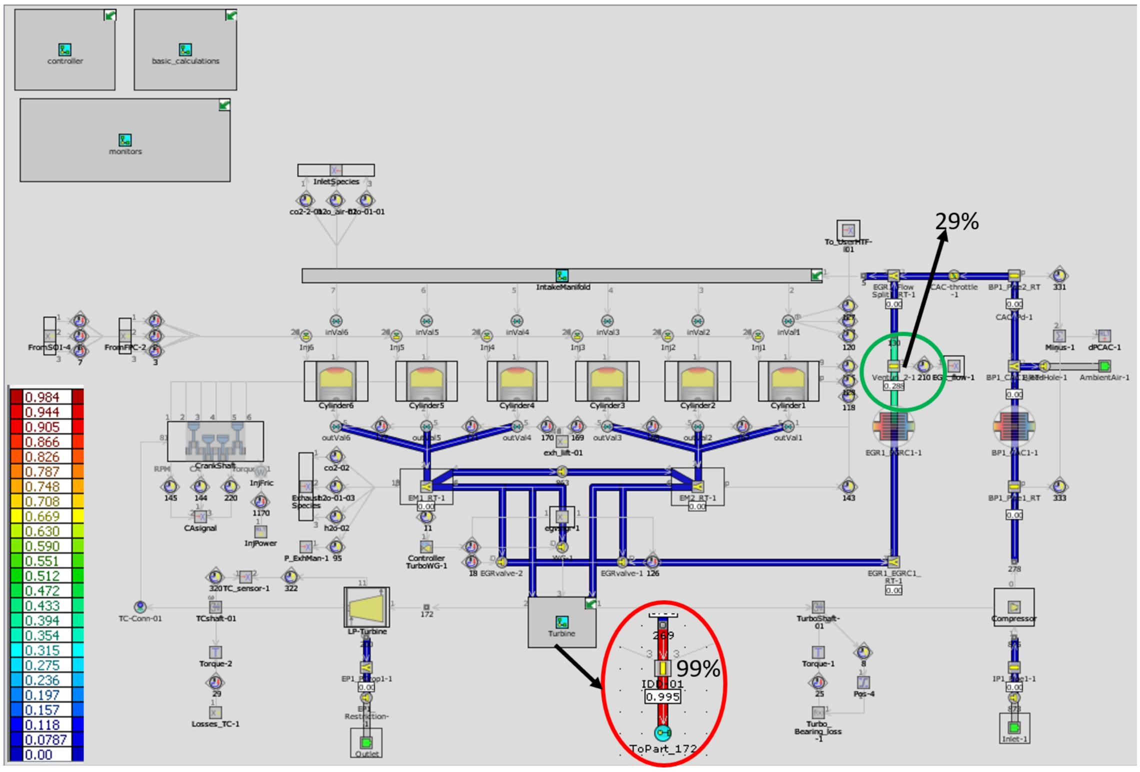



**Figure 3.12:** Picture showing the parts in the engine model which restricts the timestep

It is always possible to further simplify the model, "Simplifying for Accuracy" and "Simplifying for Speed" are two major steps which the FRM converter has as its application. In the coming section of the report, additional simplification steps will be explained further in an order.

### 3.1.8 Additional Simplifications

#### 3.1.8.1 FRM Accelerator

The FRM Accelerator is an option which can be used as an additional simplification step which is clicked on and the GT Power will automatically add the changes to the model which will actually makes the engine model run closer to Real Time (RT).

#### 3.1.8.2 Cylinder Slaving

Cylinder slaving is a straight forward easy implementation as described in fig. 3.13 and it is also a major step to reduce the factor of real time. It is done by right-clicking the cylinder object and chosen the "Edit Parent Object", in the "Advanced" tab and in the "Cylinder Slaving Option" it is required to select the "slave-RT-full-v2017" and then click "OK". Now the decision is upto the user to choose which cylinder needs to be made as a master and the slave. Here, it is chosen that "cylinder1" will be over-ridden as the Master and the rest of the five cylinders will act as the slave. The concept behind Master and slave is that the master cylinder is the only cylinder which will carry out the mathematical calculations, the rest of the five cylinders will replicate the computational results of the master cylinder.

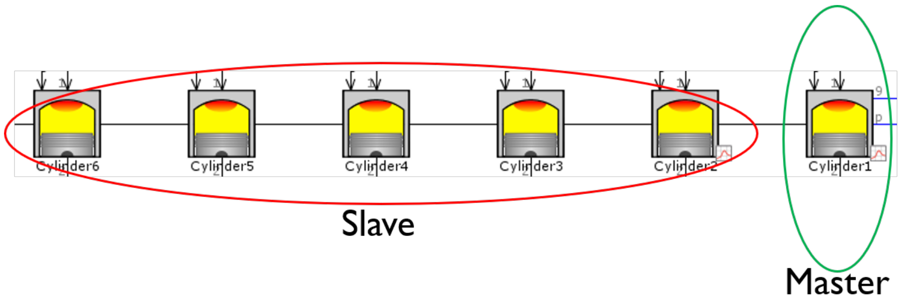



**Figure 3.13:** Picture showing the engine cylinders distinguished between the master and the slave cylinder

There is one more additional step by selecting the "Modify the Intake and Exhaust Valves"->"Output folder->"Valve Type for RLT Variable Calculations" to "intake" and "exhaust", respectively for both intake and exhaust valves.

#### 3.1.8.3 Combined flow volume approach

With the simplification steps that are applied previously, It is know from fig. 3.12 which gives the information about the objects which restricts the time step. In order to simplify these objects, it is possible to select the two or more successive objects along the flow path and combine these selected objects into one flow volume. The object can be manually selected and then "Right Click -> Combine flowsplits into subvolume", this will result in one combined volume. If needed, the accuracy of this object can be achieved through calibration by selecting the right parameter which was explained in the previous steps.

# 4

# Testing FRM for transient operation

In the previous chapter the FRM is tested to function in steady state operating points. This chapter deals with the testing of the FRM in GT Power workbench alone without adding the complexity of the inputs from the Engine Management System (EMS) in Matlab Simulink.

The best and simplest way to examine the functionality of the FRM in transient condition is to make the model to run with the input data from the engine test cell.The data is a collection of parameters which are measured through sensors placed in different parts on the physical engine. By doing this, it enables the possibility of comparing the outputs of the physical engine test with the outputs of the simulation (FRM).

## 4.1 Initial set of Modification

The fig. 4.1 and fig. 4.2 shows two types of simulation that can be carried out in GT Power. In fig. 4.1 the brake torque is plotted as a function of period and the simulation is run with a "Profile Period" instead of a "Profile Transient", this is used only for period dependent transients but not time dependent transients. The "Profile Period" does not take into account of time but will run the simulation for a specified number of cycles as constant inputs to the FRM and that could be the reason why the brake torque values exactly follow the values from the engine test cell. Taking the "Profile Period" object the time for simulation is around recorded is 4872 seconds which is not a valid duration for a BLB cycle. The complete BLB cycle which takes 3795 seconds and hence for time dependent transient simulations, "Profile Period" object is not recommended.

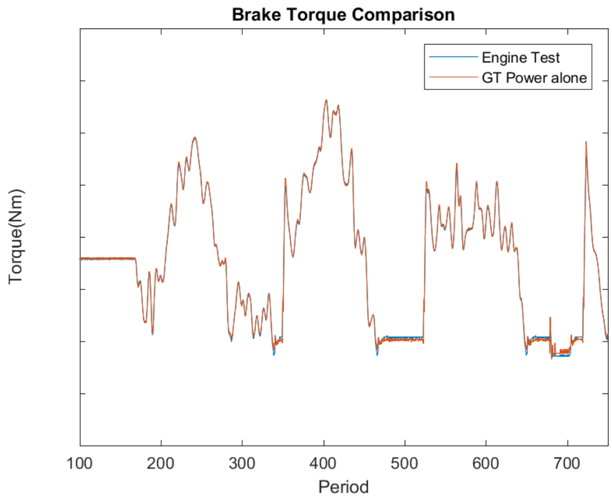



**Figure 4.1:** Brake Torque comparison in terms of period on X-axis

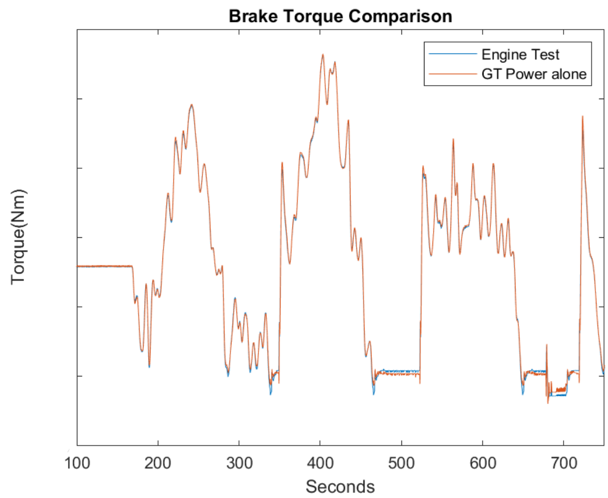



**Figure 4.2:** Brake Torque comparison in terms of time on X-axis

The data is collected in the test cell with a sampling frequency of 10 Hz, from this it is known that each value that is input to the GT Power model will have to be in a time interval of 0.1 seconds. Since the simulation is based on the run time, it is therefore necessary to change the option "Maximum Simulation Duration(Cycles)" to "Maximum Simulation Duration(Time)" in the "Run Setup" tab.
The parameters which are used a inputs to the GT Power model are as follows

1. **Charged Air Cooler Throttle position** [°C]
2. **Main fuel quantity** [mg/stroke]
3. **Injection timing** [°C]
4. **Rail Pressure** [bar]
5. **Waste gate position** [%]
6. **EGR Valve position** [%]

The above input parameters are given look up tables which reads the transient signal[**MAIN**]from the case setup. [**MAIN**] is a "ProfileTransient" object which is used when the user wishes to have a time dependent input.

**Table 4.1:** Time dependent Profile Transient object example

| time(s) | output |
|---|---|
| 0s | 1 |
| 0.1s | 2 |
| 0.2s | 3 |
| - | - |
| 982.6s | 9826 |
| - | - |
| 3795.1s | 37951 |

In table 4.1 it can be seen that the total simulation duration is 3795.1 seconds and since the input is given in the intervals of 0.1 seconds the total number of inputs accounts to a value of 37951.

**Table 4.2:** Look up table used for the part Charged Air Cooler (CAC) throttle position

| input for lookup table | CAC Throttle position(deg) |
|---|---|
| 1 | 78.0027 |
| 2 | 78.00271 |
| 3 | 78.0027 |
| - | - |
| 9826 | 11.6305 |
| - | - |
| 37951.0 | 78.0027 |

The output from the "Profile Transient" is used as an input to the look up table to fetch the corresponding value. The detailed explanation of tables 4.1 and 4.2 goes here, Consider a simulation running at time 982.6th second, table 4.1 gives an output value 9826. This value is input to the lookup table shown in table 4.2 which gives the corresponding throttle position of 11.6305 deg to the throttle object in the model. The actual working is shown as an example in fig. 4.3.

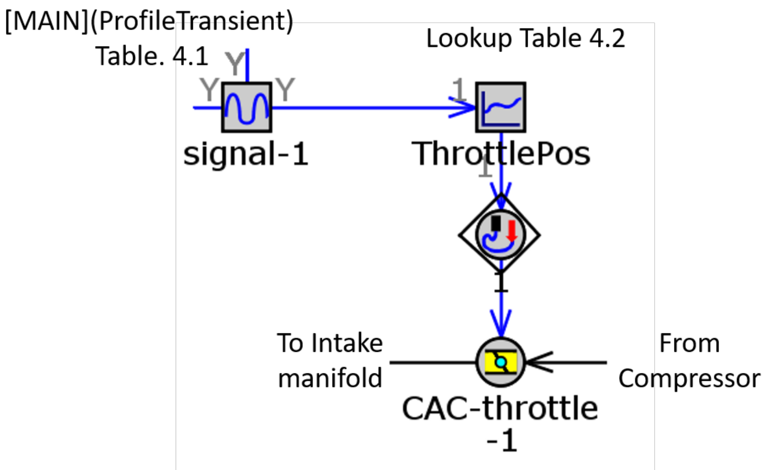



**Figure 4.3:** Example of implementing transient profile input from the engine test data to the throttle object in FRM

The values which are used as input to the FRM simulation is the EMS value which is an input from the vEMS (virtual EMS) model, so that the results are based on the fuel input from vEMS, simulated in FRM. The motivation for this is seen as an example in fig. 4.4 which shows the delay in time between the EMS signal and the measured signal. The effects of this delayed signal will have an impact on the results when comparing the outputs from the physical engine test and the simulations and hence for simulation and comparison the values from the vEMS model is taken. The explanation for the vEMS will be given in the next chapter. The FRM used for transient simulations has a few modification such as calculating the total energy consumption of the drive cycle for the post processing which can be seen as blue coloured boxes in fig. 4.5.

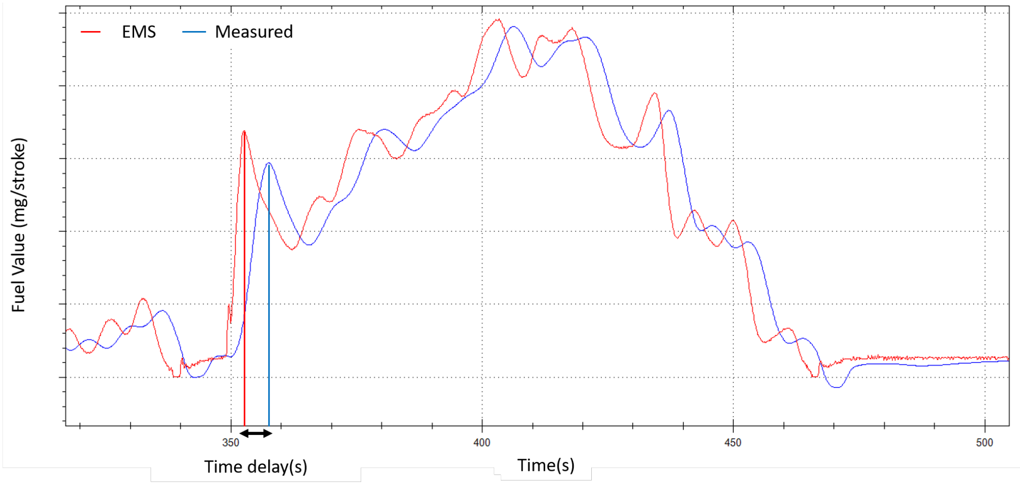



**Figure 4.4:** Graph showing a time delay between the EMS signal and measured signal

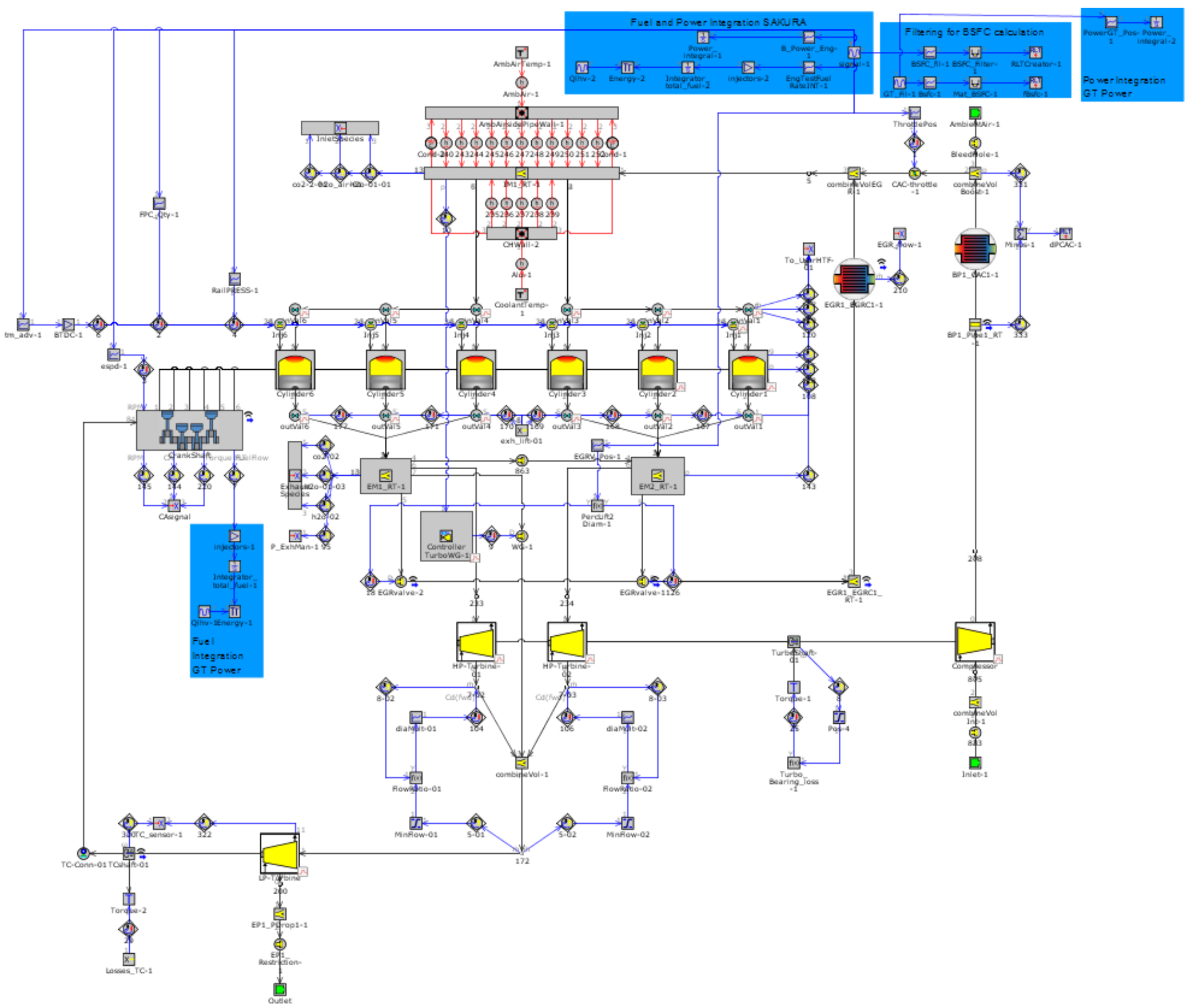



**Figure 4.5:** FRM with transient data input from the engine test cell

# 5
# Real Time FRM integration with Simulink

The engine model will currently be running in Real Time and therfore can be used to integrate the model to simulink to define a Software-in-the-Loop (SiL) type of simulation. This phase is actually the next objective of the thesis work, to be able to connect and run the real time detailed engine model along with the vehicle simulations. This is beneficial to make the model capable of running simulations with transient drive cycles for performance testing or any kind of tests. The GT Power model will be communicated with Matlab Simulink tool, this tool is what Volvo calls as Global Simulation Platform (GSP).

## 5.1 Initial set of Modification

### 5.1.1 Modifications in the GT Power model

The real time running engine model cannot be directly linked to simulink, therefore there needs to be a few important blocks that needs to be modelled in the GT Power CAE software so that it establishes proper communication with Simulink. GT Power not only allows integration with just simulink, it has a few other CAE tools with integrating capabilities. They are:

1. **AscmoModel**
2. **CarMakeInterface**
3. **CoSimInterface**
4. **FMU(Export/Import)**
5. **GTICode**
6. **MBDInterface**
7. **PythonFunction**
8. **SimulinkHarness**

These are located in the GT Library->External Model Links. Out of the external sources links mentioned above, SimulinkHarness will be extensively used to build the interconnection between Matlab/Simulink and GT Power.

The SimulinkHarness block is added to the main workbench, this is actually the start of the integration. When the SimulinkHarness block is double clicked, there will be tabs which defines the connections of signals from simulink and signals to simulink. The signals from simulink are the main signals which will actually make the GT Power model to run. Inside the SimulinkHarness block there are several tabs out of which the "Main" tab, "Inputs (from GT to Simulink)" tab and the "Outputs (from Simulink to GT)" tab are important to mention.

In the "Main" tab, the simulation type will be "run_from_simulink"
In the "Inputs (from GT to Simulink)" tab, it is possible to enter the parameters to read as an output from the GT Power model into simulink workspace.
In the "Outputs (from Simulink to GT)" tab, it is required to enter the input signals which come from the simulink model in order to run the GT Power model.

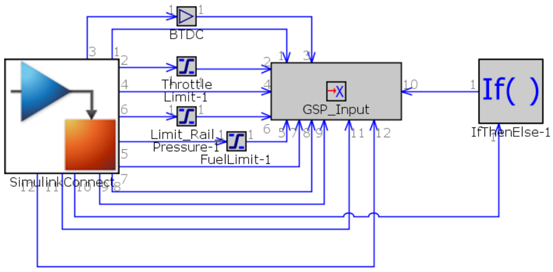



**Figure 5.1:** Path of transferring signals from simulink model to GT Power model

In fig. 5.1, it can be seen that there is a block named "GSP_Input" which is actually a "SendSignal" block in GT power which will store the signals from the "SimulinkHarness" block in the form of "Signal Names" which can be received anywhere in the engine model with a block called "ReceiveSignals", it is important to note that the "Signal Names" given in the "ReceiveSignals" block should exactly match the "Signal Names" used in the SendSignal" block.These are the important modifications on the FRM before proceeding for the integration with simulink.

## 5.1.2 Modifications in Simulink

The GT Power model is built as a s-function block in simulink which will actually replicate the real time engine model as a simulink block, the s-function will be found under the Simulink folder inside the GTI installation folder, it can be dragged on to the simulink workbench. In this thesis in fig. 5.2, as a trial, a new simulink model is built with the GT Power s-function and the input to the GT Power model is given as constants.

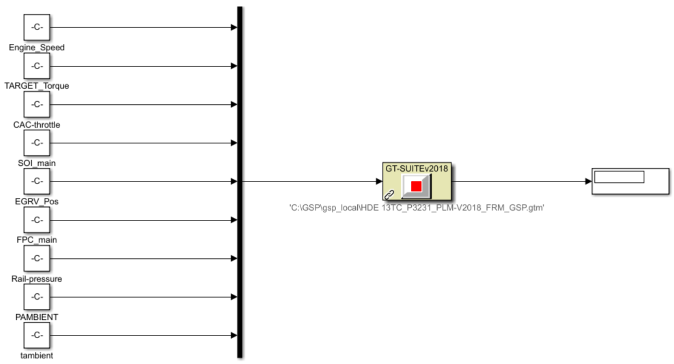



**Figure 5.2:** An example of a simple simulink and GT Power integrated model

Initially this integrated model is run by retaining the controllers built previously in GT Power. This is the first step considered in this thesis work to actually test the functioning of the integrated model which can be seen in fig. 5.2. The output for the example model was chosen to be the brake torque from the crankshaft object which is read in simulink as a display. The GT Power block in simulink can include any number of inputs and gives out any number of outputs, but have to be specified in the block, fig. 5.3 shows that the path for the GT Power engine model must specified (.gtm) and the number of inputs in the MUX creator should be equal to the number of inputs on the GT Power block, the same holds for output also. The timestep for the engine plant is 20ms which is entered in the s-function as 0.02s.

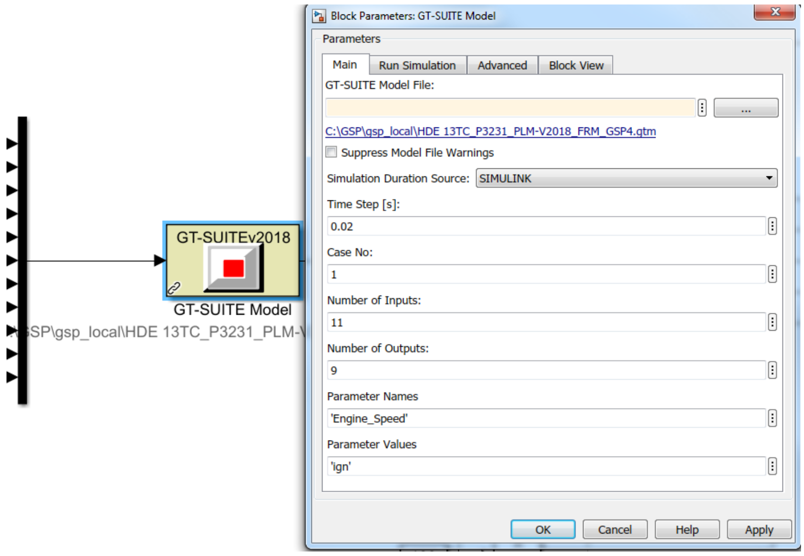



**Figure 5.3:** Settings required for the s-function to work

## 5.2 Global Simulation Platform

Global Simulation Platform shortly called GSP is a tool developed by Volvo Trucks to perform vehicle level simulations virtually keeping the Software-in-the-loop. It consists of softwares such as vTECU and vEMS which will replicate the functional softwares employed in the real truck, vTECU is a system for the control of the transmission subsystem of the truck and vEMS is a system to control the combustion of the IC engine by providing signals to the actuator which controls the engine operation. There are simplified models similar to the working of vTECU and vEMS, but with lesser level of complexity which eventually results in the model being less accurate when compared to the detailed model, these models are termed as "LITE Models". The main simulation environment looks like fig. 5.4. It consists of a driver model, the road environment and the vehicle model, they collectively simulate and gives the required output. To keep track of the simulation, it is possible to see the distance travelled, the gear in which the vehicle is running currently, the appropriate vehicle speed and finally the % progress of the simulation.

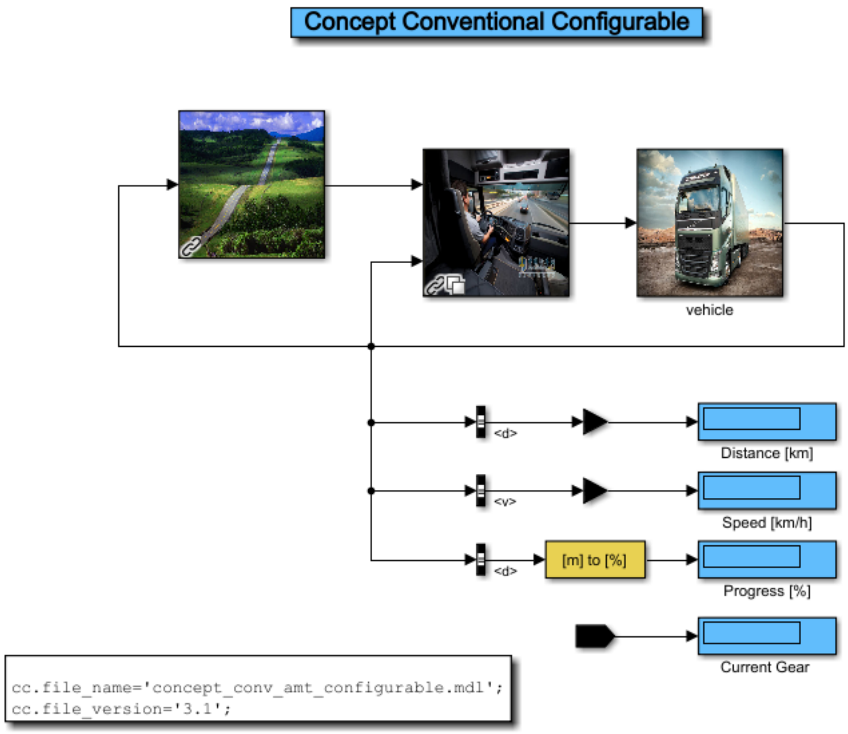



**Figure 5.4:** GSP main simulation environment

The vehicle model in GSP contains the following components:

1. **Engine model**
2. **Exhaust After Treatment System model**
3. **Electric starter motor**
4. **Auxiliary components such as pumps and fans**
5. **AMT Clutch model**
6. **AMT gearbox model**
7. **Rear axle**
8. **Tire model and lastly**
9. **The chassis**

Fig. 5.5 shows the main layout of the GSP tool along with an empirical model of the engine which the Volvo Trucks call it as "Diesel Engine Work Horse", this thesis focuses on the engine plant model which can be seen as a red circle highlighted in fig. 5.5. The basic idea is to replace the content of the engine plant model with the GT Power model (s-function).

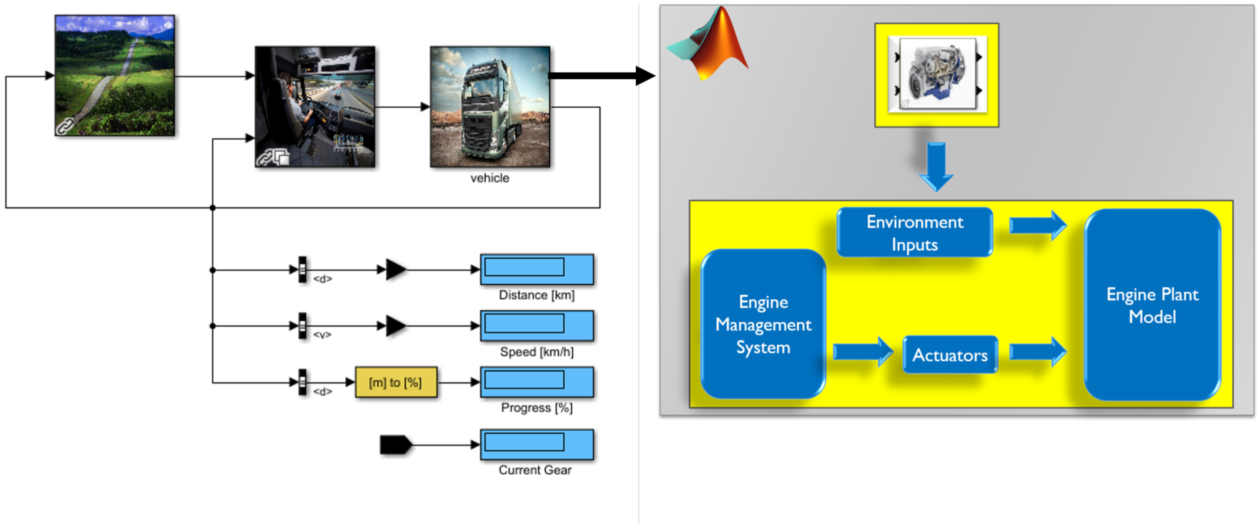



**Figure 5.5:** Picture describing the overview of the Global Simulation Platform and the engine plant model in simulink

In fig. 5.5 it can be seen that the inputs to the engine plant model is coming from the Engine Management System (EMS) through the actuators. The output of the ECU can be altered to the user requirements and the same parameters should be recorded by the actuators to actually transmit the signals to the engine plant model. The base model before the GT Power engine model is integrated is shown in fig. 5.6 and the inputs from the EMS which are required for it to function are as follows:

1. **Starter motor signal**
2. **Volvo Compression Brake signal**
3. **Volvo Engine Brake signal**
4. **Torque set value signal**
5. **Main quantity of fuel**
6. **Vehicle speed from the wheel sensor**
7. **Ambient pressure and temperature from the external sensor**

The vehicle speed is converted into engine speed [RPM] and the signals are sent to the necessary subsystems. The output of the engine model can be sent to a "BUS" Creator, which can be read in the Matlab workspace once after the simulation is complete. This can be seen with the name "LogOut" represented as a green block in the bottom right most corner in fig. 5.6. It is a very important block which will be extensively used when the GT Power model is integrated.

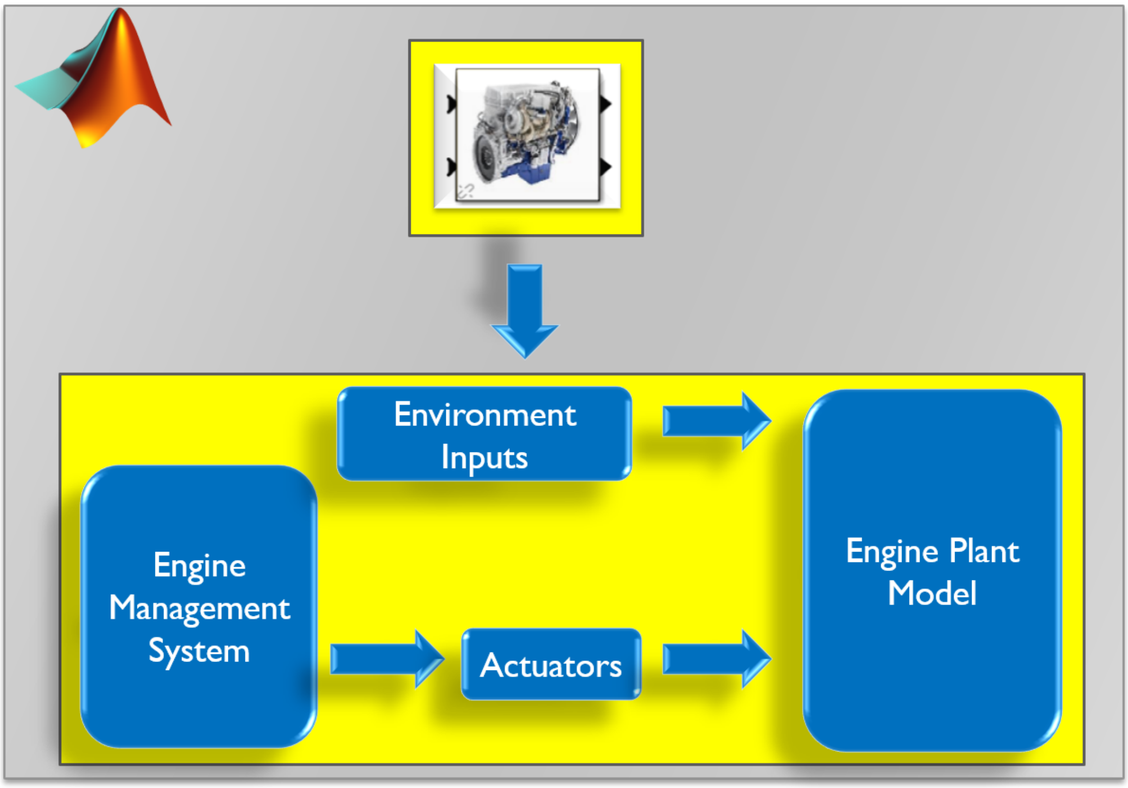



**Figure 5.6:** Engine plant model previously used for simulation

Reading the signals from the EMS to the engine plant model is the most important step before setting up the GT model for integration. The concept which was explained previously in fig. 5.2 and fig. 5.3 is applied here, but the only changes will be the number of inputs and outputs to read. Comparing fig. 5.9 and fig. 5.6, it can be seen that the VCB and the VEB subsytems are removed from the model because to make the initial model simple the VCB functionality has been removed and the GT Power model itself has VEB functionality and therefore it is removed from the GSP model. The "Boost_Pressure subsystem is retained because it is a necessary requirement for the GSP model to run in transient conditions. GT Power has the ambient pressure and temperature requirement on the units and therefore a couple of multiplication factors are introduced before the signals are sent into the GT Power model s-function. The input parameters which are read out from the EMS through the actuators are:

1. **Rail pressure [bar]**
2. **Main quantity of fuel [mg/stroke]**
3. **Volvo Engine Brake signal(Logic implemented)**
4. **Waste Gate position demand**
5. **EGR valve opening position demand**
6. **Torque set value signal(as a reference)**
7. **Vehicle speed from the wheel sensor**
8. **Ambient pressure and temperature from the external sensor**

The outputs from the GT Power model is connected to the "LogOut" so that the results are read out from the Matlab workspace.
The outputs are as follows:

1. **Brake Torque output [Nm]**
2. **Brake specific NOx emission [g/kWh]**
3. **Brake specific CO2 emission [g/kWh]**
4. **Engine exhaust temperature [K]**
5. **Brake Specific Fuel Consumption [g/kWh]**
6. **Air Flow Rate [kg/s]**

During the simulation tests after integration, it could be seen that the parameter "EMS_Control_EBRK" will be "zero" most of the times in the drive cycle and the VEB will be active when the value of "EMS_Control_EBRK" will be greater than "Zero", therefore a simple control logic was implemented for the VEB to get activated on the engine model in GT Power. An "ifthenelse" block is added in the GT Power model which is as explained in fig. 5.7.

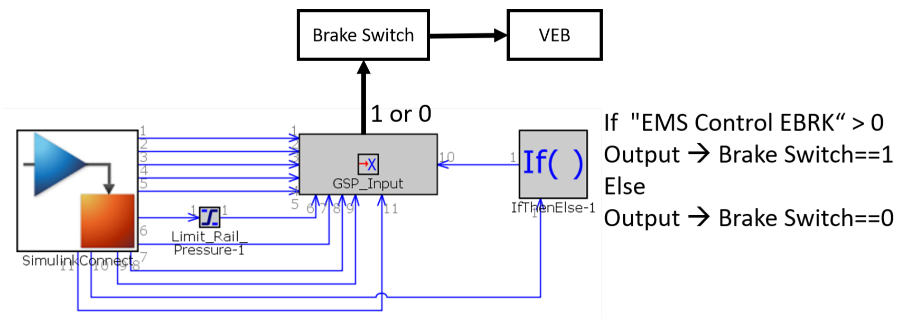



**Figure 5.7:** Control logic in GT Power for VEB activation

The main quantity of the fuel [mg/stroke] is actually a converted signal from the quantity of the fuel in [mg/s], to make the GT Power model less complicated, the post and the pilot pulses on the injector is replaced by one main pulse quantity and the SOI (Start of Injection) timing. This is an input to the GT block in simulink. It is good to note that the right signals are sent in to get a proper output from the integrated model.

For the conversion from the [mg/s] to [mg/stroke] equation 5.1 is used and the same has been represented in simulink as shown in fig. 5.1.

$$Fuel(mg/stroke) = \frac{Fuel(mg/s) * 60(s)}{3 * EngineSpeed(RPM)} \tag{5.1}$$

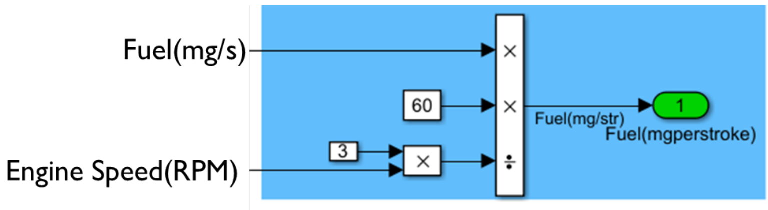



**Figure 5.8:** Conversion of the fuel quantity from mg/s to mg/stroke

Constant value of 3 in equation 5.1 mean that the fuel will be injected in 3 cylinders out of 6 cylinders at a time in one cycle and 60 is a constant that converts engine speed from [RPM] to [RPS]. Resulting in the main quantity of fuel in [mg/stroke] on the injectors.

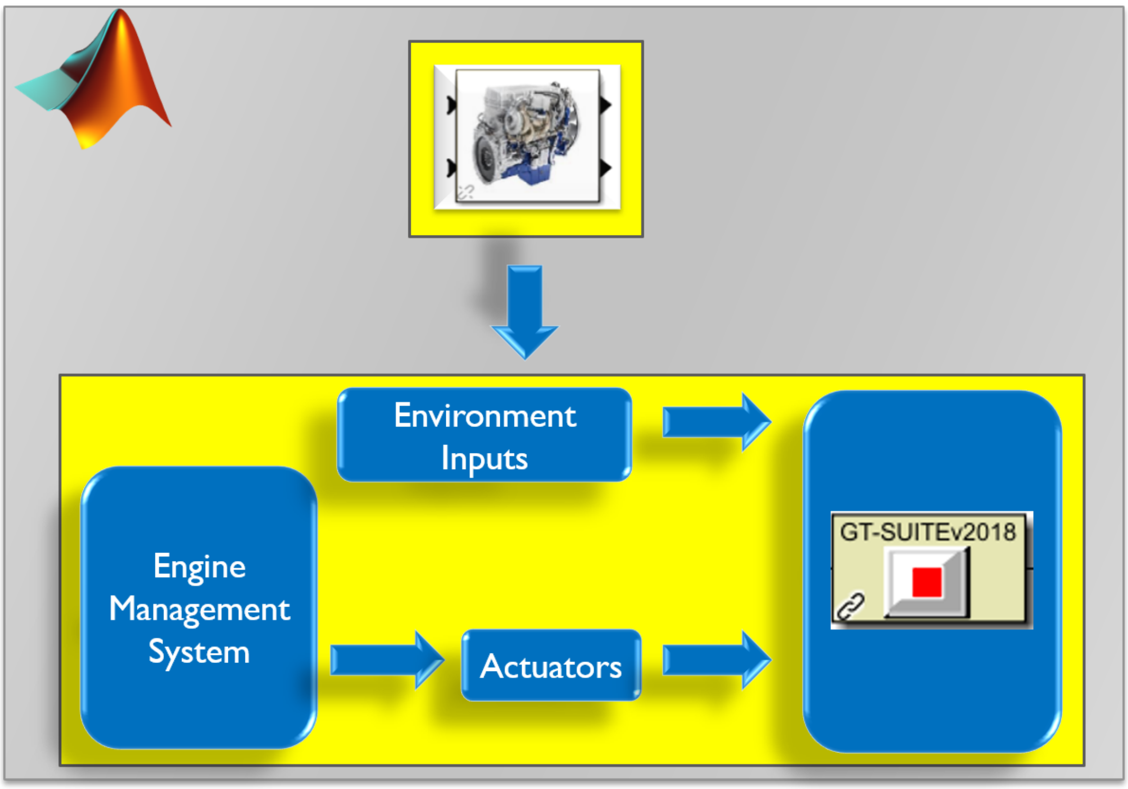



**Figure 5.9:** Engine plant replaced by GT Power s-function for simulation

# 6

# Results

This section shows the results for the calibration step when simplifying for accuracy, speed and the manual simplification steps in the FRM conversion. The comparison plots between the results of the data received from the engine test cell and the GT Power integrated model will be shown, which will suggests how much off the model is when compared to the data obtained by running a physical engine in the test rig.

The results of the application of the FRM to build a simple hybrid system will also be explained, but the results are just to show the working of the hybrid system and to also show the potential of the FRM to handle new hardware components in the model. The hybrid control logic used does not contribute in reducing fuel consumption or emissions.

Further,the optimization results of the late closing of the inlet valve which is simulated on the "Original" engine model (not the FRM) will be covered in detail with their respective graphs. The results will suggest an optimum valve lift profile which has a BSFC benefit compared to the baseline model.

## 6.1 FRM calibration results

The FRM calibration results include step wise accuracy plots for the 21 different operating points of the engine model and there will also be plots of the Part Load Map operating points as a Factor of Real Time, these plots suggest how far it is from the Real Time which gives a picture of how much more the model needs to be simplified.

### 6.1.1 Results of "Simplification for accuracy"

The comparison between the baseline "Original" model and the FRM is done on 21 different operating points which were at first selected for simplification and calibration.During the process of simplification it is expected that the model will no longer behave the same way as the previous "Original" model as some of the vital objects which makes the model to predict better will be taken away or simplified as one single volume and this volume may not capture the same physics as the "Original". The model still has some complexity built into it which can capture the phenomena. Here, fig. 6.1 shows the step where the exhaust manifold will be simplified, it can be seen that for the selected 21 different operating points the temperature data is seen overlapping with each other which explains the accuracy in calibration. The results of simplification of each tagged sub-model from the "Original" model will be explained in an orderly manner in the appendix chapter. For a good calibration step, it is always better to have proper correction factors such as Heat Transfer Multipliers and the Orifice diameters so that the simplified model behaves similar to the "Original" model.

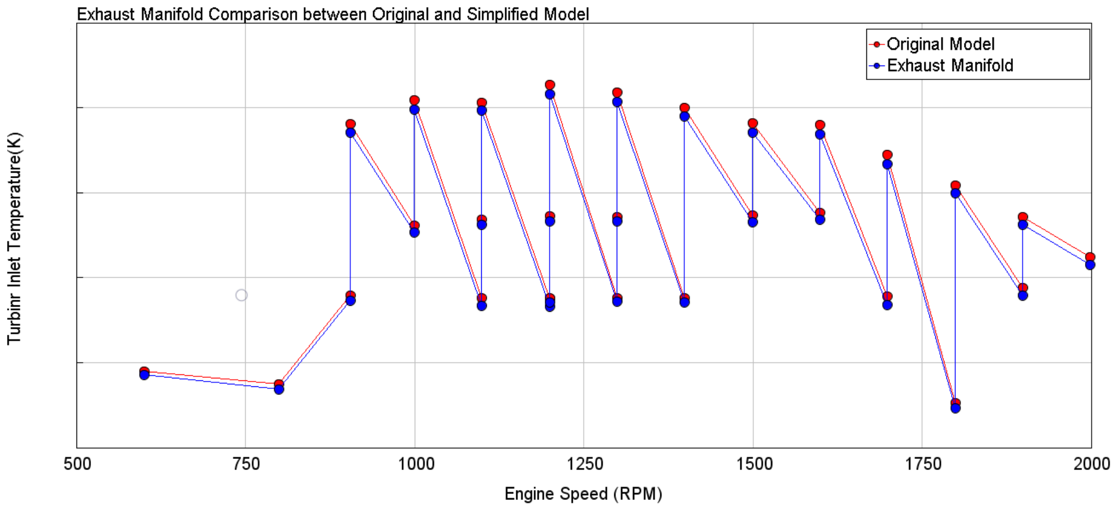



**Figure 6.1:** Comparison of turbine inlet temperature after simplification for accuracy of exhaust manifold sub-model

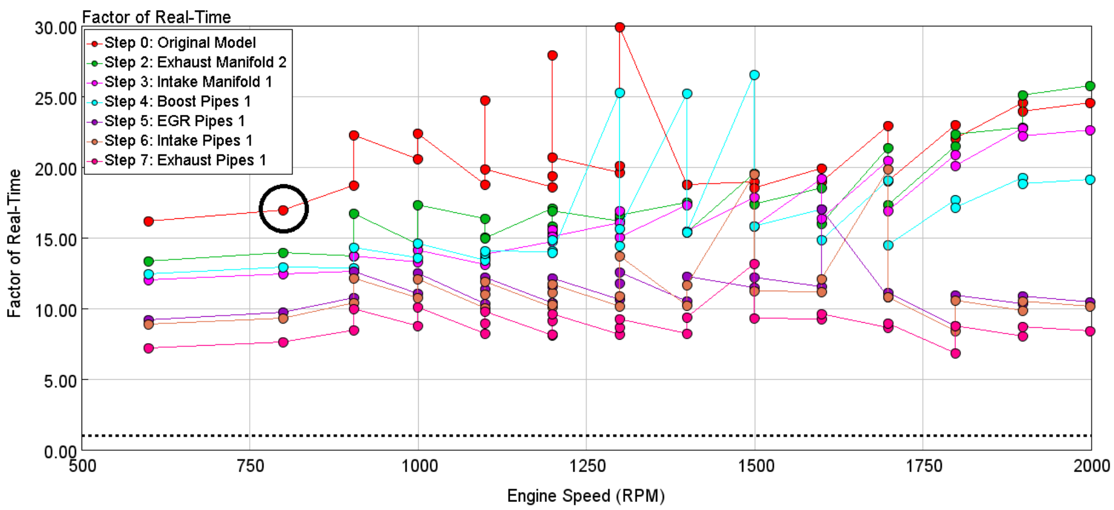



**Figure 6.2:** Factor of Real Time after simplification for accuracy of tagged sub-models

Fig. 6.2 shows the variation of the operating points in terms of factor of Real Time, this is an important graph to look into during the simplification process as the graph gives the information of the real-time factor of each of the operating point in a decreasing order after simplification of each of the tagged sub-model. Consider an operating point at around 780 RPM, the value of the factor is approximately 16 which explains that the original model at that operating point is running 17 times slower than Real-Time. So, after each of the tagged sub-model is simplified, there is a decrease in the factor by almost half of the previous value, that is 7.5. The explanation holds for each of the operating points shown in fig. 6.2.

### 6.1.2 Results of "Simplification for speed"

In the "Simplification for Speed" the Boost pipes and the exhaust pipes can no longer be further simplified in the FRM conversion procedure. However, they will be considered in the manual simplification steps if any of those objects are a restriction the time step. Therefore, in this step the boost and the exhaust pipes are not considered for simplification and calibration. The detailed explanation is previously discussed in the subsection 3.1.6.

The graphs shown in fig. 6.3 show similar trend as explained in the previous paragraph of "Simplification for accuracy". The graph also suggest that there is a fair overlap with the "Original" model and the simplified model. However,in the graph of intake manifold and EGR pipes there seems to have a loss of accuracy on one or couple of operating points which will be explained in the appendix.

There can also be another reason for the inaccuracy which will explained using another graph in the appendix.

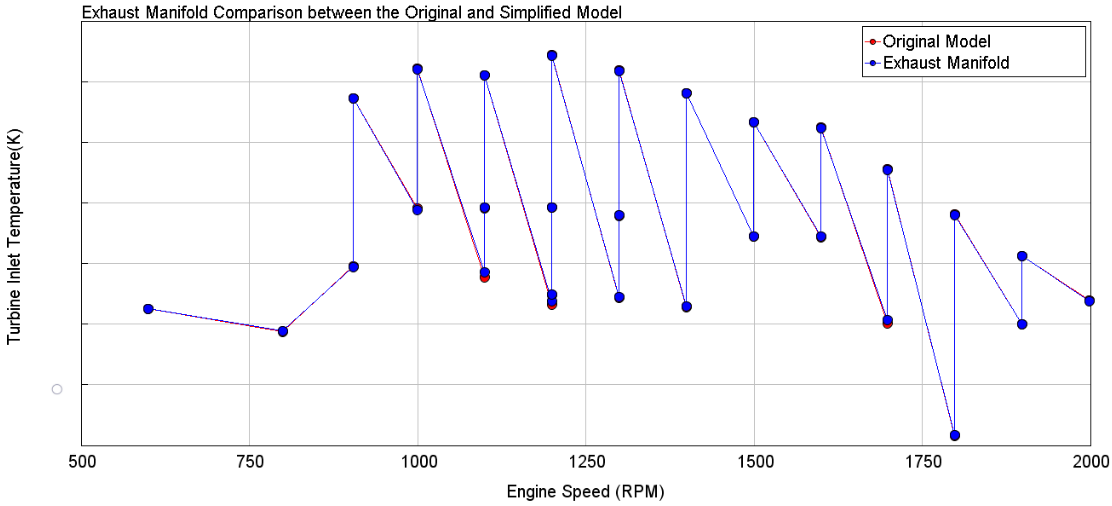



**Figure 6.3:** Comparison of turbine inlet temperature after simplification for speed of exhaust manifold sub-model

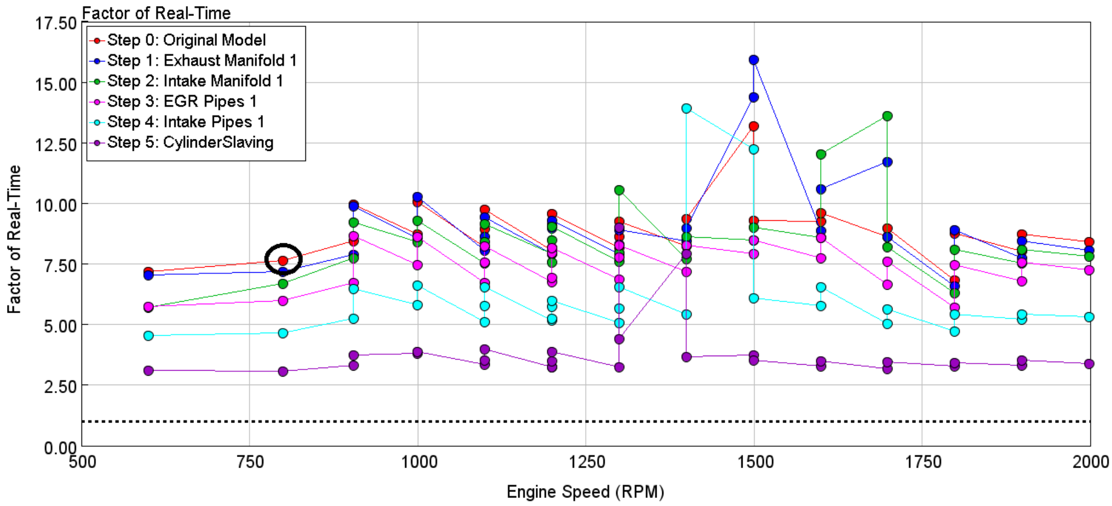



**Figure 6.4:** Factor of Real Time after simplification for speed of tagged sub-models

The most important graph that needs to be studied at this point after the second simplification step is the factor of real time.

The same operating point is that was selected in the previous simplification step is selected for analysis. It can be observed that the operating point has now reduced after simplifying for speed from a factor of around 16 to a factor of around 5 which can be seen as "Intake Pipes 1". There was an additional simplification which was carried out which is called "Cylinder Slaving" explained in the conversion chapter previously. Through cylinder slaving the factor of real time in fig. 6.4 was reduced to 3 from 5 which means the model is still running 3 times slower than RT. Therefore, manual simplification which is explained as "Combine flow volume approach" are done by combining flow objects to flow volumes and calibrating if necessary, by doing this the FRM will be running in 1xRT.

## 6.2 Verification of FRM in Steady state and transient simulation

The verification of the performance was made for a few steady state points used for calibration. Now, the FRM is tested to perform for the entire PLM data for steady state verification. For transient state, the verification was done by running the BLB drive cycle in GT Power FRM and comparing the results with the results of BLB cycle run on a physical engine in the engine test cell.

### 6.2.1 Results of FRM under Steady State operation

The graph of brake torque shown in fig 6.5 show the comparison between the baseline "Original" model and the Fast Running Model of all the operating points in the PLM data. The important parameters which are selected for discussion are the brake torque and the exhaust temperature of the gases exiting the Low Pressure(LP) turbine. The inlet temperature of the turbine, outlet pressure from the compressor and the maximum pressure inside the cylinder are explained in the appendix.

In fig. 6.5 the operating points of the baseline and the FRM are overlapping each other which indicates the working capability of the torque controller(fueling), however, there can be seen a couple of points at lower engine speeds which are not coinciding with the baseline operating points.

Due to the reduction of the complexity of the model in FRM, fig. 6.6 of exhaust temperature show a similar type of trend as a baseline model but it can be seen that at lower torque regions on all engine speeds the values of the exhaust temperature has an offset and tend to deviate from the baseline model, the reason for this deviation will be explained in the later part of the report when explaining the exhaust temperature during transient state operation.

Lastly in fig. 6.7 the connected RED dots are the real time factor of the baseline model and the BLUE dots are the values of the real time factor of the FRM. The complex model operating points which was running 15-30 times slower than real time, after the simplification the same operating points are running in 1xRT or even faster than 1xRT.

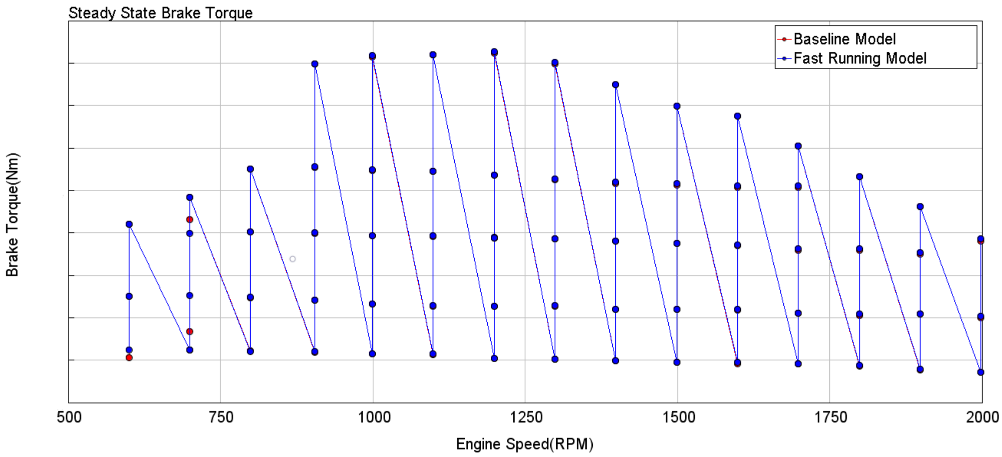



**Figure 6.5:** Comparison of brake torque between the complex base model and the FRM

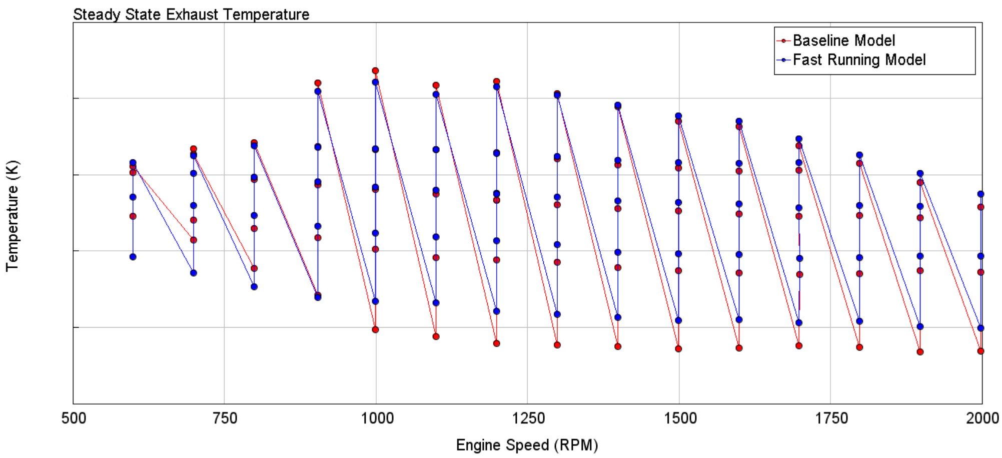



**Figure 6.6:** Comparison of exhaust temperature between the complex base model and the FRM

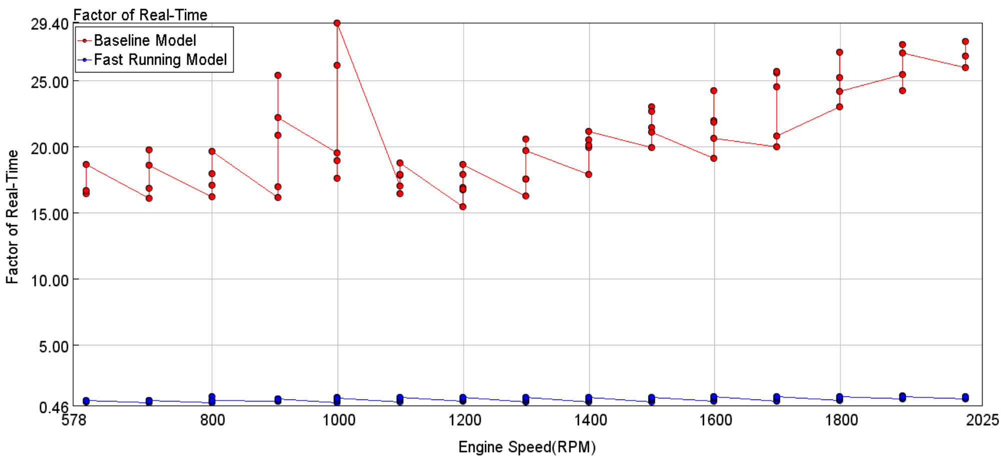



**Figure 6.7:** Factor of Real Time after the complex model is simplified to FRM

### 6.2.2 Results of FRM under transient operation

The FRM is now verified under steady state operating points. This subsection deals with the comparison of results obtained from the transient simulation of the FRM.

The comparison is between the simulation results and the physical engine test results. Parameters such as "Brake Torque [Nm], Brake Specific Fuel Consumption [g/kWh]", "Boost Pressure [kPa]", "Turbocharger speed [RPS]", "Inlet Air Flow Rate [g/s]", "Compressor Pressure Outlet [kPa]", "Exhaust Temperature [°C]","Fuel Flow Rate [g/s]" are compared as graphs with time on the X-axis in seconds.

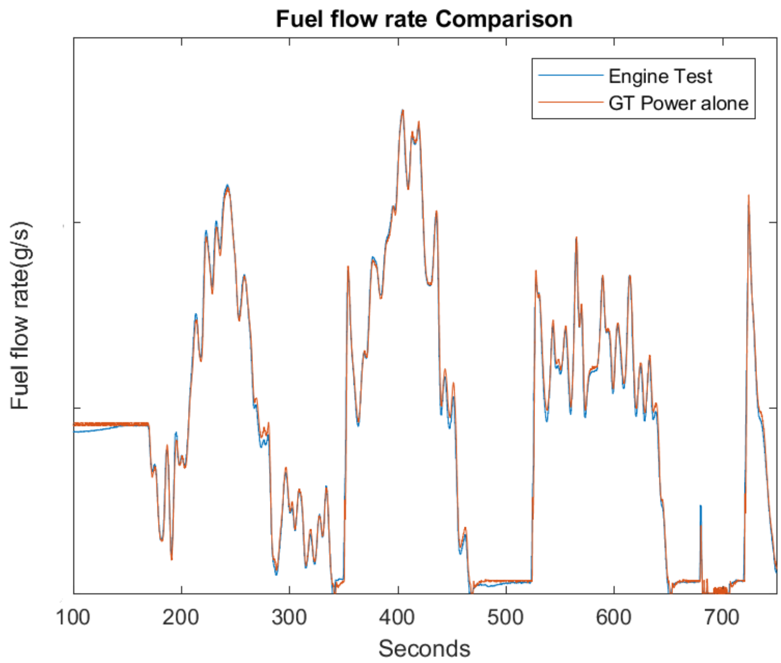



**Figure 6.8:** Comparison of Fuel Flow Rate from results of FRM and Engine test

The fuel flow is an important parameter which will decide the engine behaviour. The torque is generated based on the fuel input and for the FRM to perform better, the fuel signal from the EMS in [mg/stroke] is used as an input to the injectors for simulation.

The "Fuel Flow Rate" is a parameter is checked in the "Crankshaft" block in GT Post Processor. This will be used as a comparison to the engine test data. Observing Fig. 6.8 the values of GT Power simulation seems to follow the values of the engine test data, the trend looks similar, without any time delay to injectors in GT Power. The delay could mainly occur because of the type of injectors used in the engine test might have different tolerances and might have calibrated differently to the ones used in simulation. And also the measurement equipment will be at a distance apart and there might be a minimum time consumption for the signal to reach the equipment.

The amount of brake torque generated will be equal to the amount of fuel supplied. In fig. 6.8 the right amount of fuel quantity is supplied by the injectors and therefore the right value of torque is being generated as shown in the fig. 6.9. The model always has a loss of accuracy at low load operations and most of the simulations will be performed with those low load operating points, fig. 6.11 shows the difference in the value of brake torque at a certain time period of 35 seconds from 1410 to 1445 seconds of the drive cycle. The percentage error in the deviation of the brake torque during low load is approximately 2-3%.

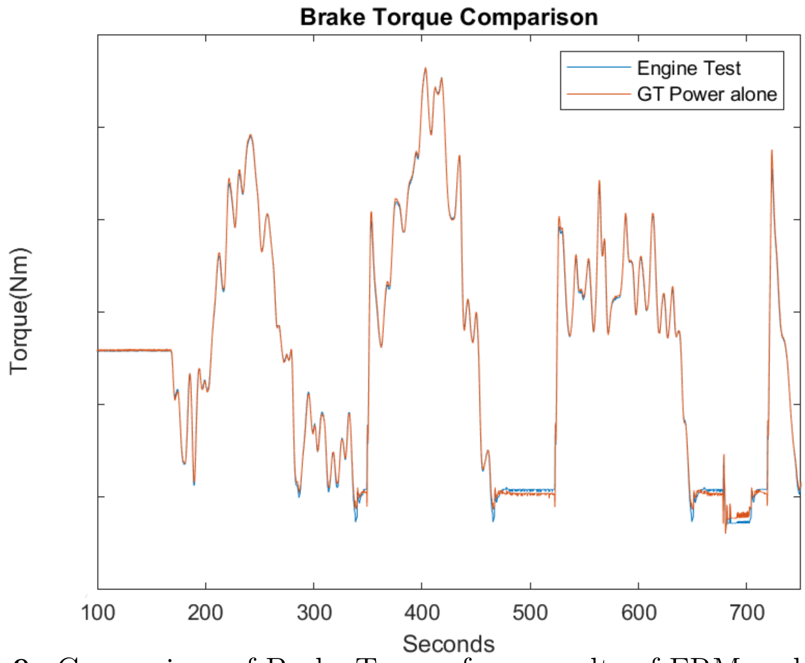



**Figure 6.9:** Comparison of Brake Torque from results of FRM and Engine test

In fig. 6.10 during the period of 600 to 1300 seconds, at this event it is seen that the value of the brake torque to be deviating drastically to the fuel input, the reason being that the Fast Running model that is used to run the transient simulations in the GT Power platform alone doesn't have the engine braking mode activated and a similar behaviour can be seen in a few other parts of the drive cycle.

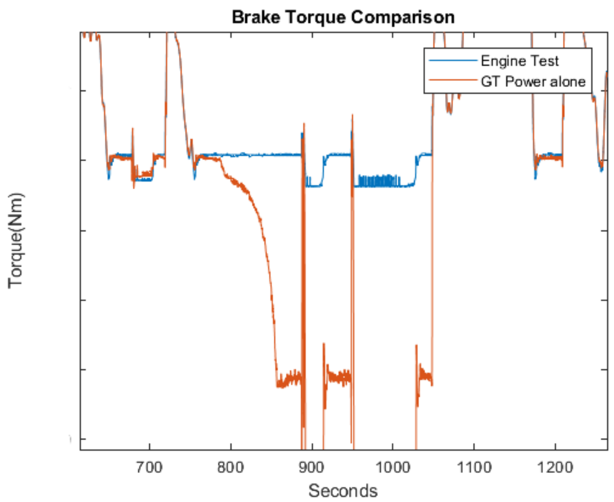



**Figure 6.10:** Brake Torque value deviation from the Engine Test value

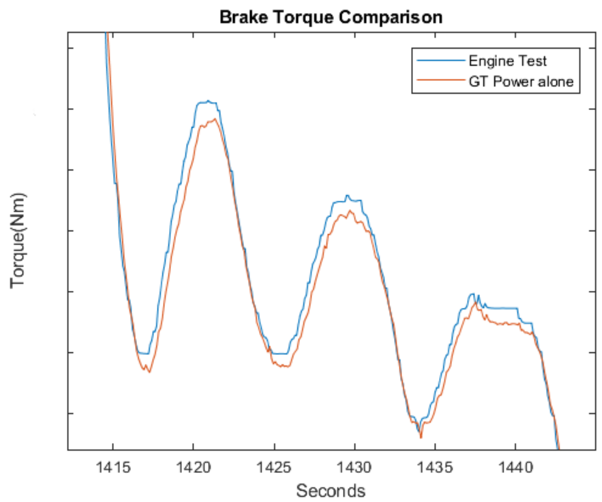



**Figure 6.11:** Brake Torque difference at low load operations

The other reason being, at operating points when the engine brake is active, it can be seen in fig. 6.12 throttle position has low opening angle which results in creating pressure in the intake manifold to be less than 1 which in turn makes the pressure ratio in the low pressure turbine to be less than 1 as shown in fig. 6.13 and fig. 6.14 which is not valid and thus shows the drastic behaviour in fig. 6.10.

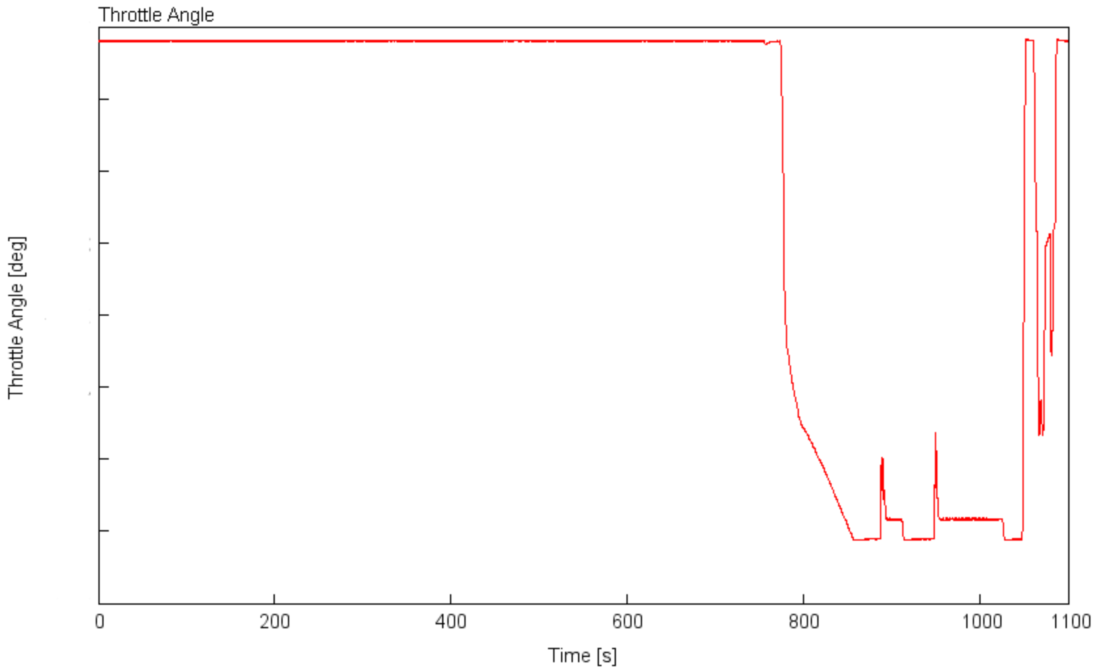



**Figure 6.12:** Throttle position behaviour with and without active engine brake

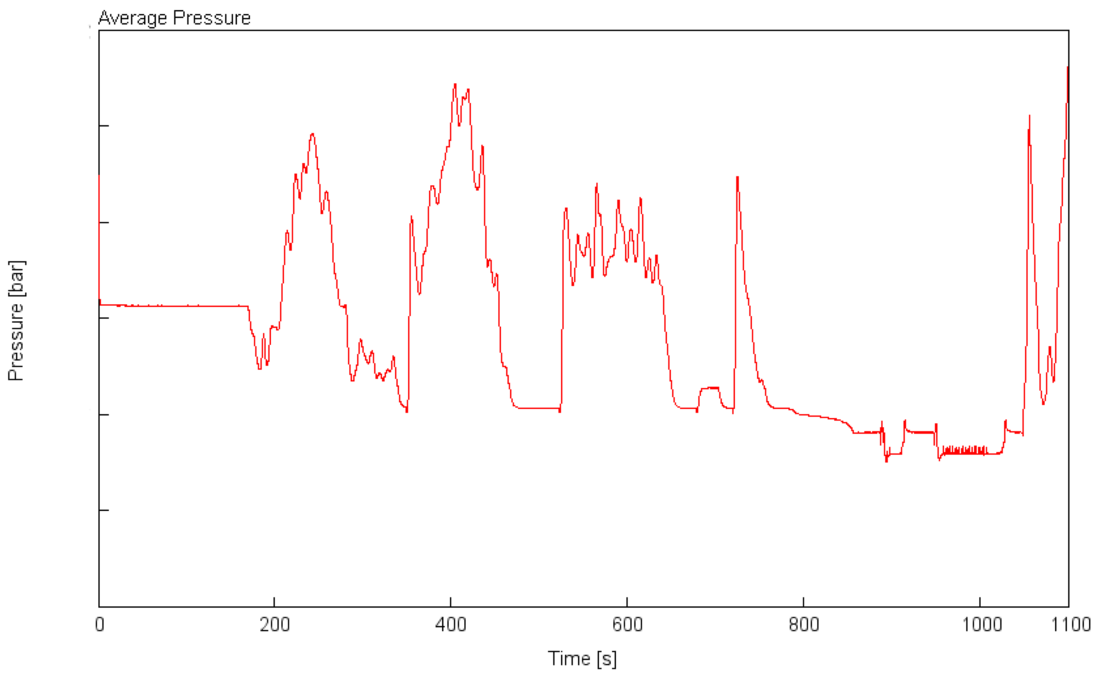



**Figure 6.13:** Average pressure at the intake manifold

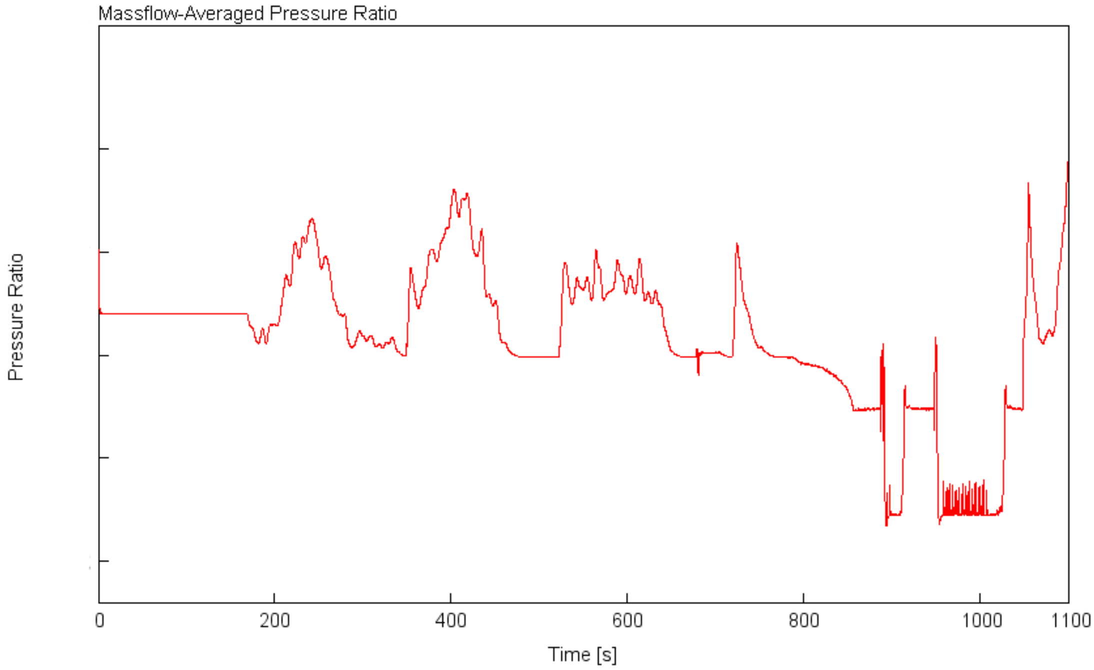



**Figure 6.14:** Average pressure ratio at the low pressure turbine

The graphs shown can prove that the throttle position change during the engine brake is causing pressure ratio problems. The value for the throttle position when seen GSP is a constant value of 100 which means to say it is completely open for the entire cycle. For simulation purpose in GT Power alone, a constant throttle opening value of around 78 degrees can be maintained.

Now, since the fuel and the torque is overlapping with each other and are required for the calculation of the Brake Specific Fuel Consumption. The fig. 6.15 is a Y-Y axes plot which has the values of the BSFC on one of the Y axis and the engine speed represented as a red curve on the second Y axis.

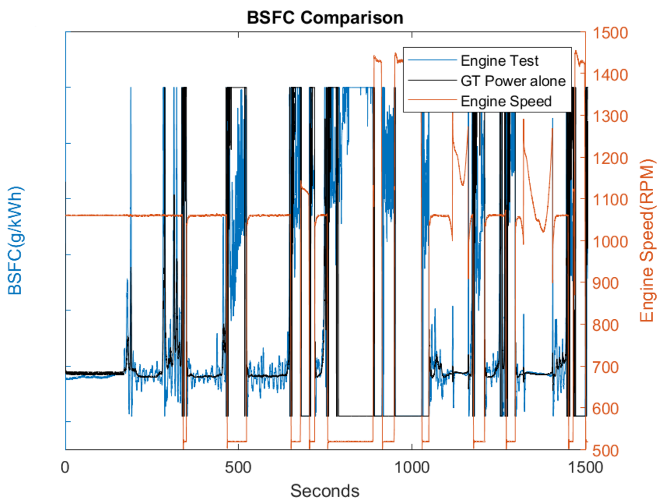



**Figure 6.15:** unfiltered and Scaled BSFC value comparison plot

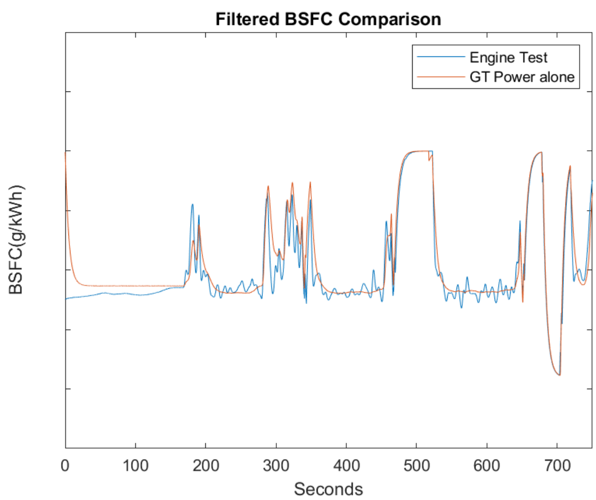



**Figure 6.16:** BSFC comparison plot filtered signal

The black curves represent the results of FRM simulations and the blue curves represent the results from the engine test rig. At certain events in the drive cycle when the engine is running in idle speeds will result in higher values of BSFC. One thing to note in fig. 6.15 is that the signals from the GT Power is a smooth curve and the signals from the engine tests has spikes in its measurements during every time instant and it becomes difficult to arrive at any conclusions. A first order filter block is used in GT Power to filter the disturbances and the filtered plot is shown in fig. 6.16. Even after filtering the signals there are some values which are overlapping with each other but, the time instances for example between 500 to 700 seconds the disturbances can still be seen.

Alternatively, the work around to conclude on the BSFC is to calculate the total energy consumption of the entire drive cycle. The energy is calculated by integrating the positive power values from the engine test rig and from the GT Power simulations. So, this is a measure of how much energy is spent by the engine to complete the BLB drive cycle. It can be seen in fig. 6.17 that the energy consumption for both physical engine test and FRM simulations are the same.

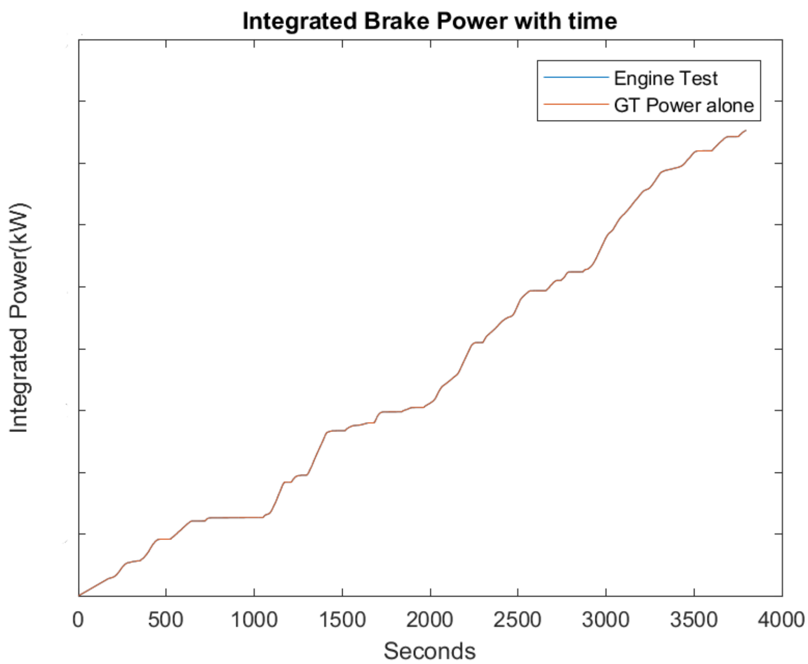



**Figure 6.17:** Integrated Brake Power comparison between engine test and FRM simulation

Another way to verify is to measure the fuel input using a GT Power inbuilt fuel controller or a suitable PID controller. These controller requires the brake torque as a target input parameter and the fuel in mg/stroke will be the output. Here we can actually record the amount of fuel that needs to be supplied in order to reach the target torque value. The conclusion here will be to see if the fuel energy of both engine test and FRM simulation is the same or not.

Now, the plots such as boost pressure, turbo speed and the air flow rate will be discussed in detail. These parameters are essential for the combustion process and also to check the predictability of the FRM in transient operations.

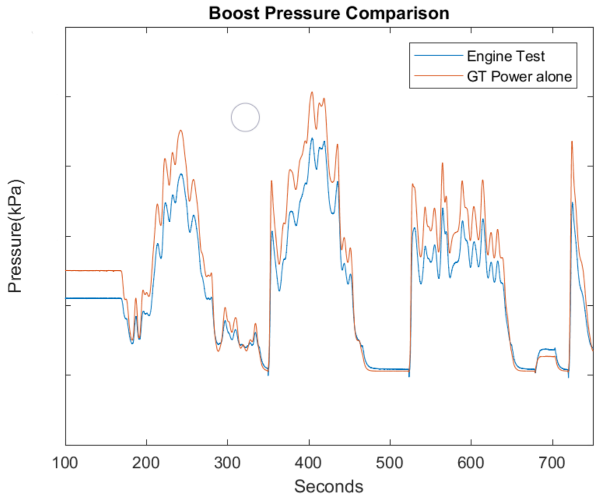



**Figure 6.18:** Boost Pressure values deviating engine test results

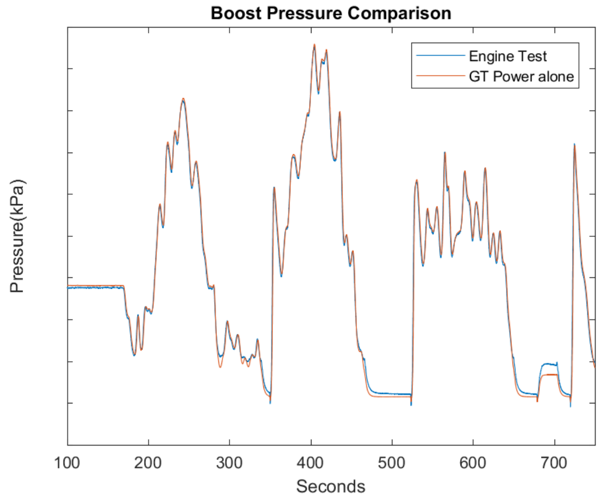



**Figure 6.19:** Boost Pressure correction made to follow the values of Engine test

Fig. 6.18 and fig. 6.19 are comparison plots of boost pressure. In fig. 6.18 it can be seen that the boost pressure values have a clear offset with the values measured in the engine test cell.

The boost pressure is regulated using a waste gate, this waste gate is modelled as an orifice and this object receives a transient input as per the logic shown in fig. 4.3 in chapter 4. This transient inputs are constant values and it does not take into account of the boost pressure in the intake manifold. Without a reference input from the intake manifold, The model will behave as it is seen in fig. 6.18.

To solve this problem, a GT Power inbuilt waste gate controller is been added to the model. The controller requires a reference input from the intake manifold and a target signal of the boost pressure. The target signal is a signal input from the EMS. Based on the target signal and the reference signal, the right value of diameter is calculated and is sent to the waste gate which is in the form of an orifice. With the right value of the diameter of the orifice, the value of the boost pressure from the simulations overlap with the results from the engine test cell.

The explanation for the turbo speed in fig. 6.20 and the air flow rate in fig. 6.21 is same as the explanation for the boost pressure. Because these parameters are dependent on each other. The amount of boost pressure relates to how fast the turbo is spinning and the faster it spins, the higher will be the density of the air going into the cylinder.

In fig. 6.20 at around 480 to 550 seconds in the drive cycle, it can be observed that there is an offset. The reason for this is that the turbine efficiency in the turbine map drops at lower turbo speeds. Although it can be observed in fig. 6.21 that the air flow rate are not completely overlapping with each other, the difference in the value of the flow rate is very minimum and which can be seen on the Y-axis.

The three parameters explained here had a clear offset in the values without the use of a waste gate controller but upon using the waste gate controller block in GT, the results of both the simulations and engine test are in co-ordination with each other.

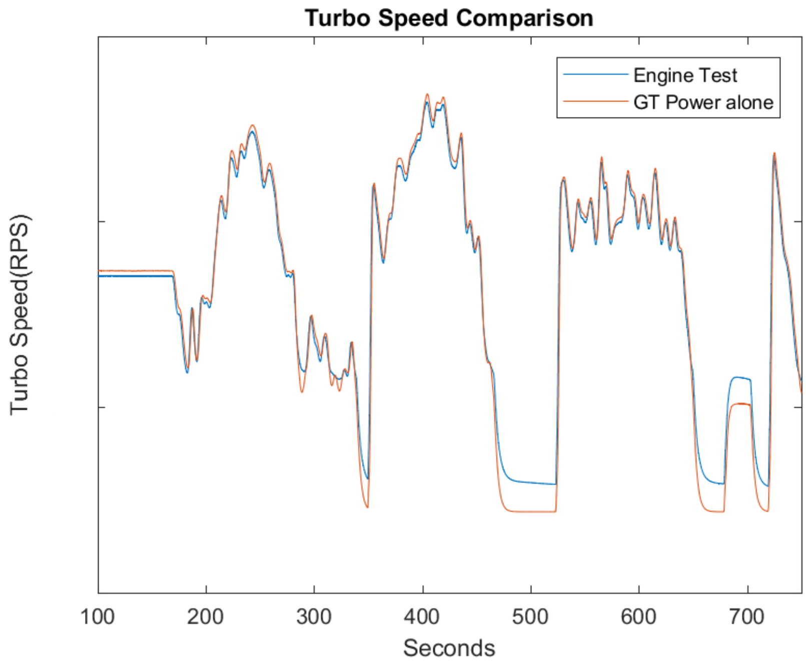



**Figure 6.20:** Comparison of Turbocharger Speed from results of FRM and Engine test

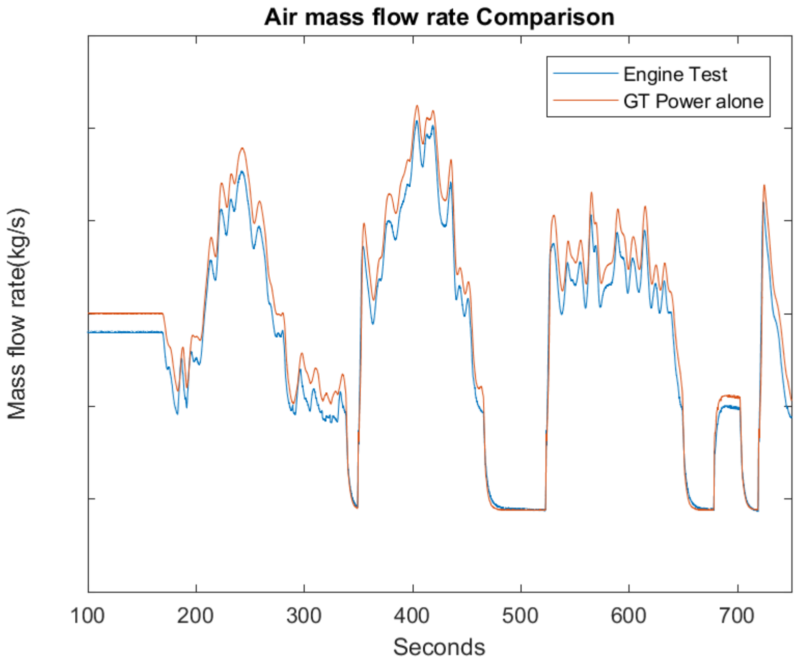



**Figure 6.21:** Comparison of Air mass flow rate from results of FRM and Engine test

Exhaust temperature is the most important parameter to compare, here, in fig. 6.22 shows the exhaust temperature before the Low Pressure Turbine unit which seems to be again deviating at lower torque regions when compared to the results of the engine test cell. The accuracy of this object needs more attention, one thing to start the investigation here is by looking at the turbine maps and adding the right value of the turbine inertia for both Low Pressure and High Pressure turbine.

In fig. 6.23 the trend of the exhaust temperature measured from the engine test cells looks to be filtered out and the temperature values from the simulation has a lot of spikes similar to the ones seen in fig. 6.22 and therefore does not overlap with each other.

The object where the temperature is measured is a combined flow split object which is calibrated to show considerable amount of accuracy on the, but even after calibration the behaviour is deviating and the object needs a thermal mass or a thermal network to be built on the object.

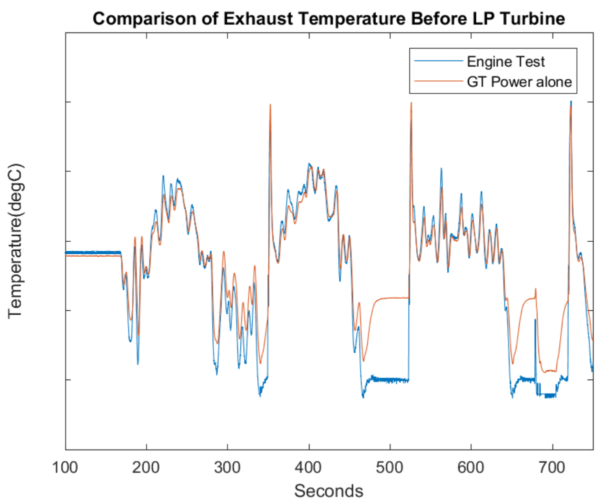



**Figure 6.22:** Comparison of Exhaust Temperature before the TC unit

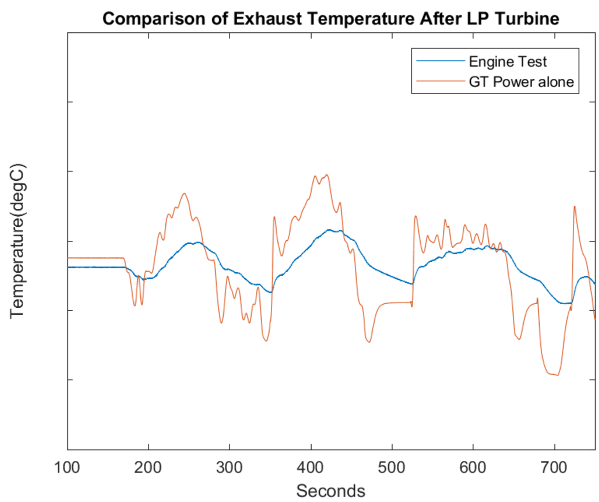



**Figure 6.23:** Comparison of Exhaust Temperature after the TC unit

The thermal network is already built on the intake manifold and a similar kind can be built on the exhaust pipe. The advantage of building a thermal network is that it is possible to add the details of the exhaust pipe such as the pipe length from the Turbo Compound (TC) unit till the Exhaust After-Treatment System (EATS), pipe thickness and the diameters.

## 6.3 Verification of integrated FRM with GSP

This subsection shows the performance of the FRM when coupled with the GSP as seen before in fig. 5.9. Since the FRM is tested for steady state and transient state conditions, it is now important to connect the required transient inputs to the FRM from the EMS and run the simulations from simulink. Here, the comparison is between the GSP simulations with "D-Eng" as the engine plant model and GSP simulations with FRM as engine plant model. There are not too many parameters that are compared as the results of the FRM is already compared with the physical engine test cell results. This is more or less the comparison of one model with another model.

The first thing to make sure is that both model's fuel input is the same. Therefore, it can be observed in fig. 6.24 that both the fuel quantities are overlapping with each other.

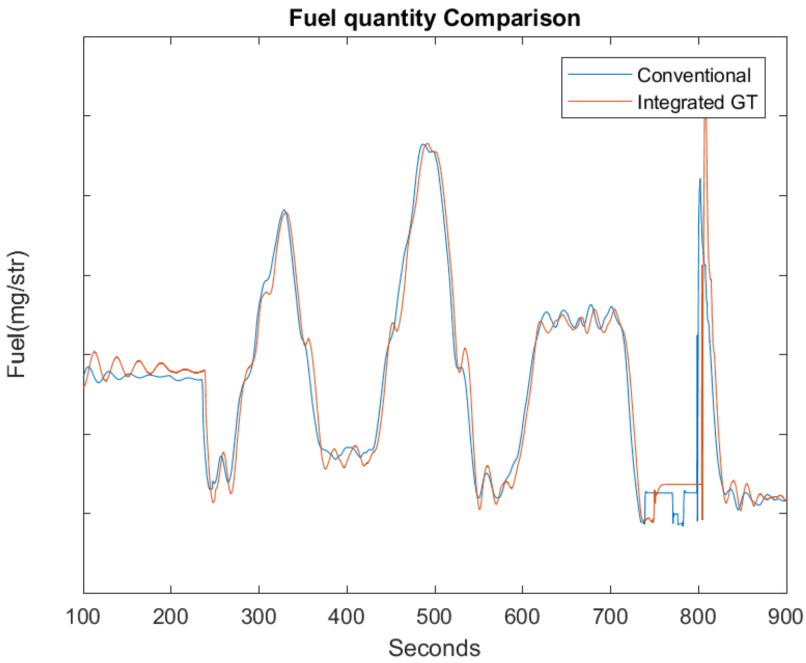
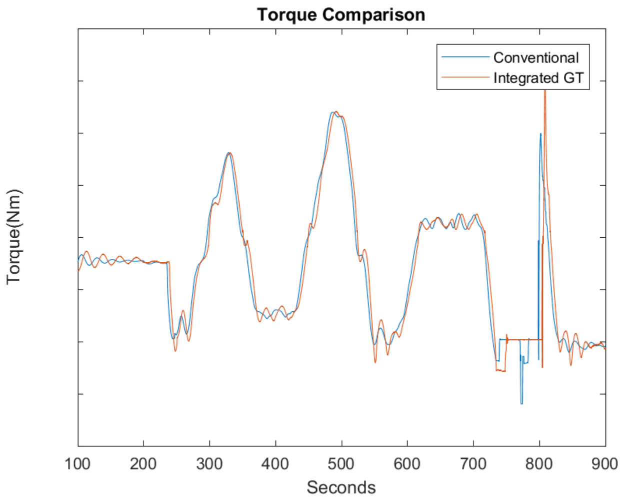



**Figure 6.24:** Comparison of Fuel input from results of Integrated FRM in GSP and Conventional GSP

It is now known from transient simulations that the amount of torque output depends on the input fuel and hence it can be seen in fig. 6.25 the same type of behaviour of the brake torque which was seen in the previous section.

Previously, In chapter 5 under the section "Global Simulation Platform", the logic of how the Volvo Engine brake was activated has been explained in fig. 5.7 So, now if we take the plot of the brake torque at the same interval as taken in fig. 6.10 and compare with the results from the integrated model with the Engine brake active, the drastic behaviour of the torque cannot be seen which means that the integrated model is responding to the logic written. The comparison can be seen in fig. 6.26.



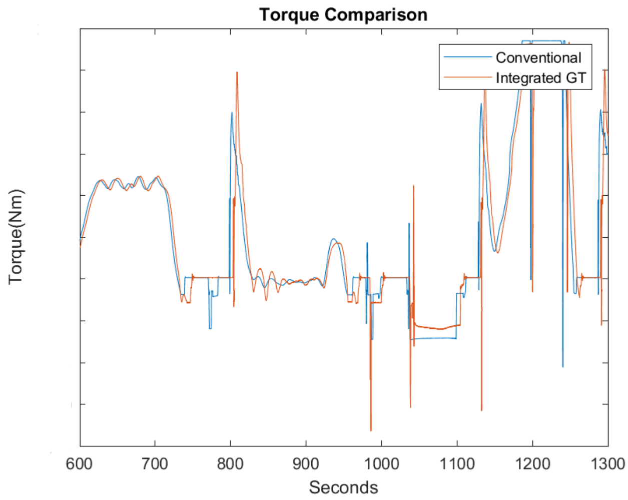
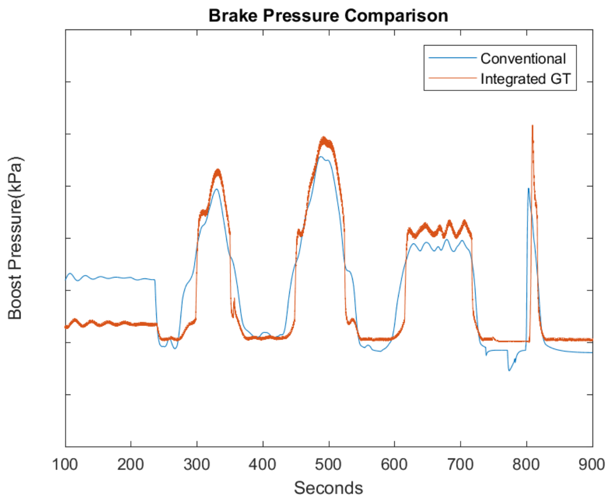




**Figure 6.25:** Brake Torque comparison of two engine plant models

**Figure 6.26:** Brake Torque behaviour during engine braking

The boost pressure data from the Conventional GSP model was recorded and then compared with the boost pressure from the GT Power output. It was seen that again without the use of waste gate controller there was a definite offset in the values and this offset was eliminated by the use of the controller, it can be observed when comparing fig. 6.27 and fig. 6.28.



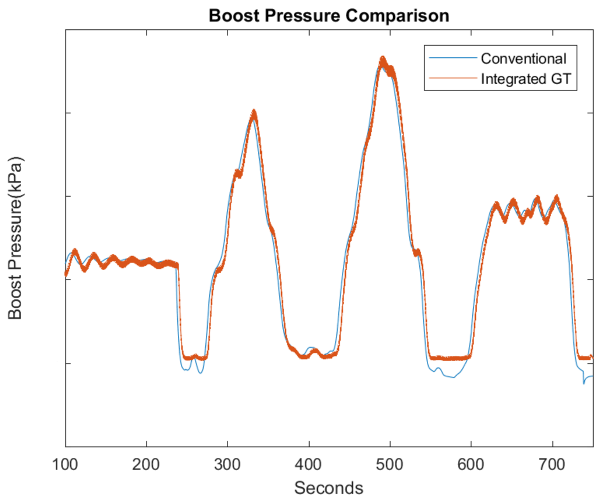




**Figure 6.27:** Boost Pressure comparison without waste gate controller

**Figure 6.28:** Boost Pressure comparison with a waste gate controller

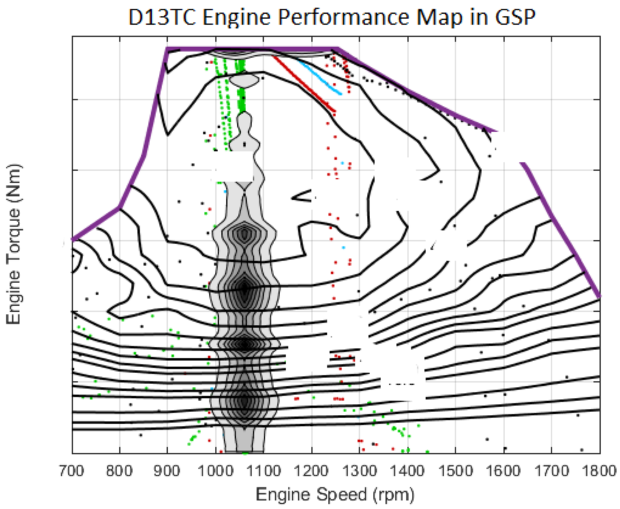



**Figure 6.29:** Engine Performance map showing engine operating regions of the BLB cycle from the GSP model

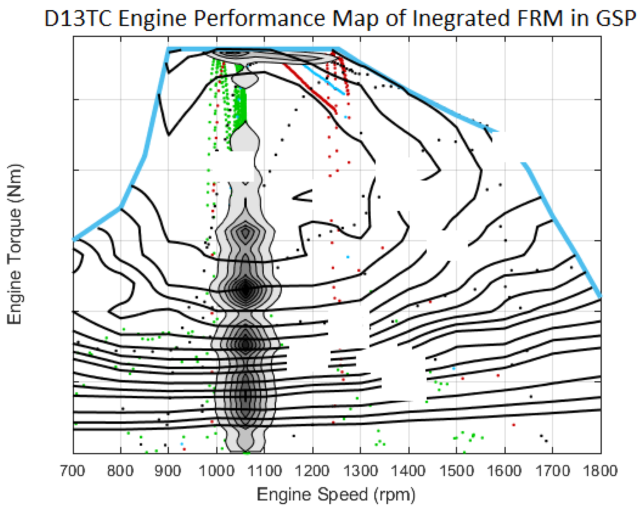



**Figure 6.30:** Engine Performance map showing engine operating regions of the BLB cycle from the Integrated GSP model

Fig. 6.29 and fig. 6.30 is the engine performance map comparison between the integrated model and the GSP model. Fig. 6.29 is the performance map obtained at the end of simulation of the complete BLB cycle and fig. 6.30 is the performance map obtained at the end of integrated model simulations. It can be seen that both these maps look very similar to each other and the numbers inside the map which are seen along the contour lines are the indicated brake efficiency values of the engine at different engine speed and torque points. At around 1000RPM to 1200RPM, a cloud of operating points on different engine speeds ranging from 0 to maximum torque of 2856 Nm is seen on fig. 6.29, it gives the information about where the engine is mostly run in the BLB drive cycle. A similar pattern can also be seen in fig. 6.30.

## 6.4 Miller Cycle Investigation results

Apart from the method development process described in all the previous chapters and sections, a short investigation on the Miller cycle was made on the complex baseline engine model to know if the concept can actually work on the existing TC engine. For simulations, Design of Experiments (DoE) was implemented with the parameters such as Rail Pressure, Injection Timing, Dwell at Maximum lift and Compression Ratio. The parameters that were varied and their limits will be shown in the form of a table 6.1.

**Table 6.1:** Table describing the parameters involved for DoE and its limits

| DoE Parameters | Value limits to vary |
|---|---|
| Rail Pressure | 400 to 2800(bar) |
| Injection Timing | +3 to -10(deg) |
| Compression Ratio | 16.5 to 22 |
| Dwell from 596 CAD | 15 to 54(CAD) |
| Peak Cylinder Pressure | 220 to 270(bar) |

A total of six Steady State Operating Points were chosen from the performance map generated using the results of the GSP simulations and the window for the selection of operating points is as shown in fig. 6.31 with a total of 2160 experiments.

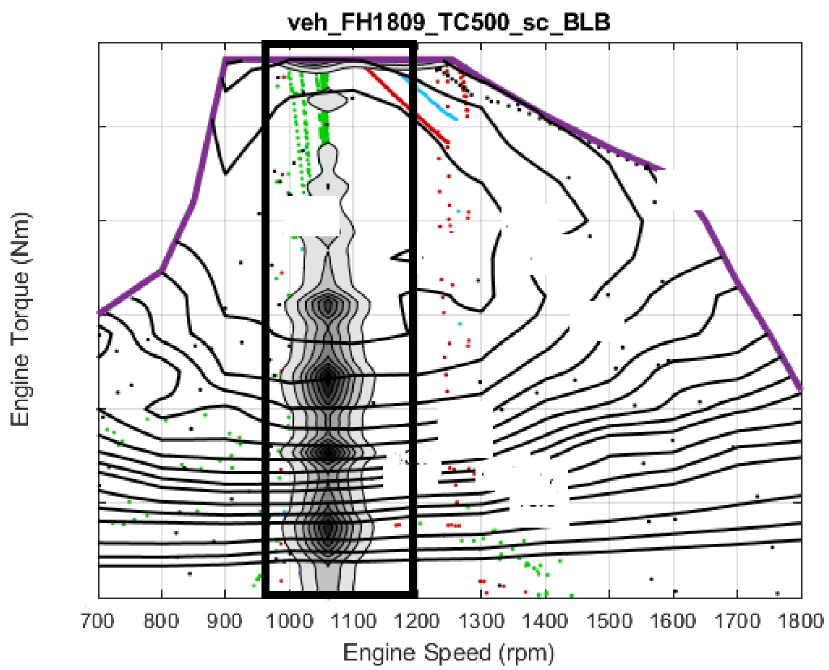



**Figure 6.31:** Window of operating points selected for DoE

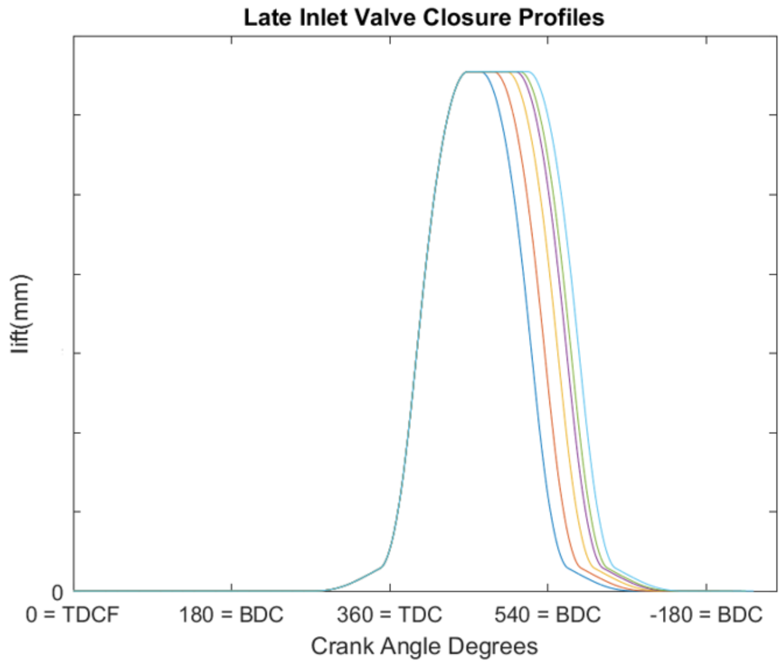



**Figure 6.32:** Miller cycle Inlet valve closure profiles investigated

The Miller cycle was investigated for various late inlet valve closure (LIVC) profiles from the baseline CAD of 596 as shown in the fig. 6.32. Five to six LIVC profiles were tested and the results of the Peak Cylinder Pressure (PCP) of 270 bar and with a Compression Ratio (CR) of 22 are presented in the table 6.2

From table 6.2 it is evident that with the use of Miller cycle at a PCP of 270 bar, CR of 22, Dwell of 40.5 there is a 0.3% to 2.6% BSFC benefit.

**Table 6.2:** The results of the DoE for a PCP of 270bar, Dwell of 40.5 CAD and a CR of 22

| Engine Speed | Brake Torque | Baseline BSFC | Miller BSFC | Percent decrease |
|---|---|---|---|---|
| 1099 | 2879.647 | 182.88 | 182.25 | 0.342 |
| 1199 | 2889.627 | 184.74 | 180.36 | 2.371 |
| 1299 | 2790.566 | 183.83 | 178.91 | 2.676 |
| 1199 | 503.678 | 202.86 | 201.36 | 0.708 |
| 1199 | 1148.26 | 188.66 | 187.91 | 0.362 |
| 1199 | 1740.531 | 185.45 | 183.21 | 1.205 |

# 7
# Applications

The Fast Running Model is tested and verified for transient operation. This technique can be applied to Hardware-in-the-Loop rigs which can be useful in fault diagnosis, global driver predictability. It can also be applied as a engine plant model in bench tests and Virtual testing of engine components such as injectors, valves etc which is similar to Volvo Penta's VIRTEC and FEV's HiL xMOD.

This fast running engine model will be useful for the development of MPC (Model Predictive Control) which is advantageous in analysing the road data ahead and calibrating the engine to suit the demand. Here, in this thesis a hybrid system is presented as an application by which it is possible to test the functioning of the FRM along with the complexity of the electric machine.

## 7.1 Hybrid system results

As an application to the use of FRM, a simple hybrid system was modelled with the inbuilt electric motor and battery models as shown in fig. 7.1. The electric motor is connected to the engine crankshaft just like the way the TC unit is connected. Here, instead of a gear mechanism a simple clutch is modelled to engage and disengage the electric motor based on the driving demand. A supervisory control logic is written in order to shift the mode from only ICE to ICE plus electric assist which acts as a power boost mode. The system here is designed to show the capability of the FRM to accept new hardware components for simulation or to account for the changes to the existing model. This system is used as an hybrid engine plant model for integrated simulations in GSP. The model is not designed to reduce fuel consumption or emission. In this section the results of the working of the hybrid system control logic will be discussed.

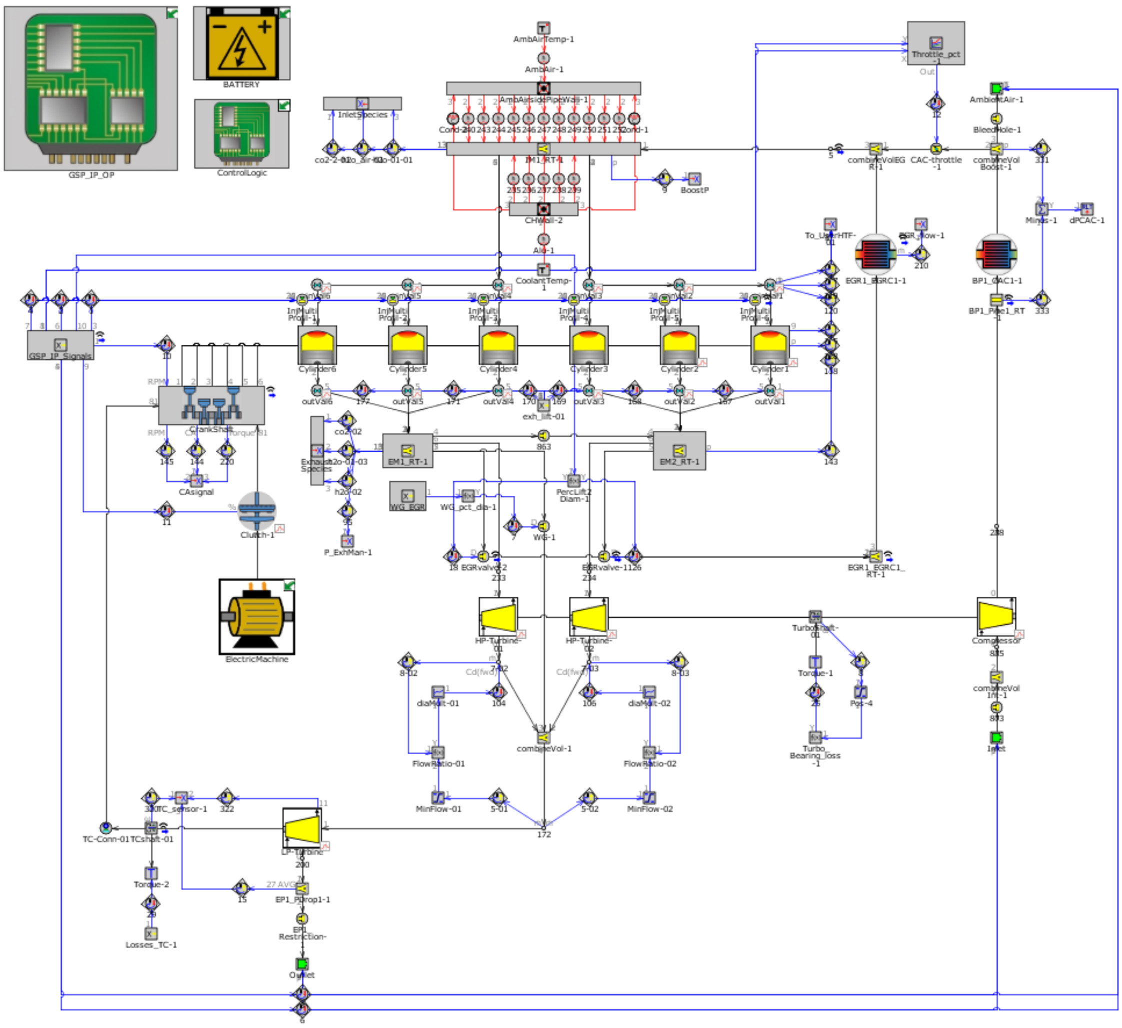


**Figure 7.1:** Layout of the FRM with a built simple hybrid system

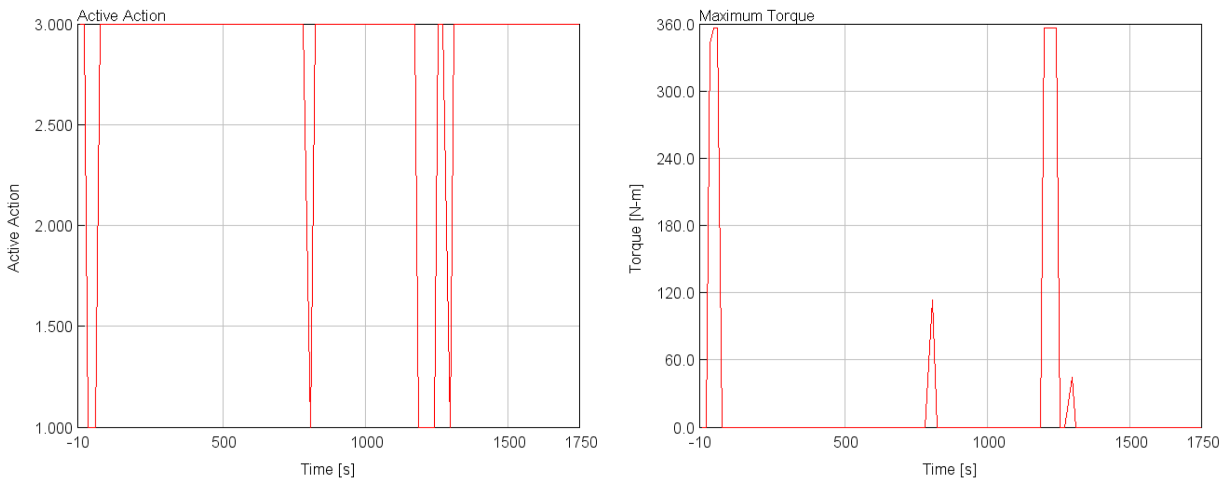



**Figure 7.2:** Plot describing the electric motor activation and the requested torque value

To verify the working of the supervisory controller, in fig. 7.2 the Active Action plot gives the information about the time instances in the drive cycle when the electric machine is active and when the vehicle is running only on ICE mode. In the Y axis of the active action plot, the value ONE represents that the vehicle is driving in electric assist or the power boost mode, the value TWO is the condition for engine braking in hybrid mode and the value THREE is the in only ICE mode. It can be seen that most of the time in the drive cycle the mode will be in THREE which is only ICE mode and there are a few instances in the drive cycle which goes to mode ONE.

The plot next to the active action is the motor torque request plot and it can be seen that whenever the driving mode is ONE, the electric motor torque request is created and there is a ramp up of torque. This can be verified by looking into the battery state of charge (SOC) plot in fig. 7.3, it gives the information about the use of the battery whenever the electric machine is switched on.

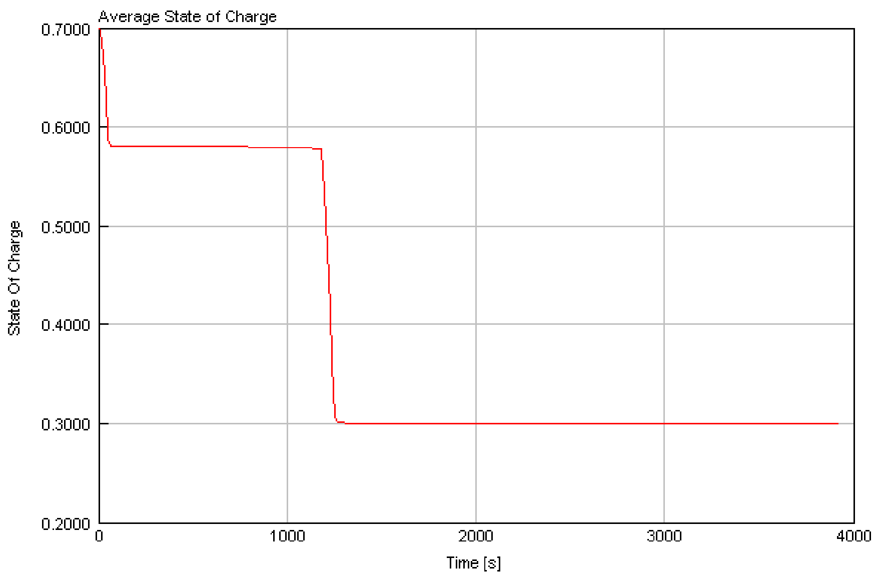



**Figure 7.3:** Battery State of Charge for the entire BLB cycle

# 8 Conclusions and Recommendations

## 8.1 Conclusions

This thesis work has demonstrated the process of conversion of complex engine plant model to Fast Running Engine models for Global Simulation Platform system integration and verification. The FRM is now running in Real Time and are used for drive train level simulations.

The conversion procedure of the complex model to FRM in GT Power has a few simplification steps which are easy to conduct and the method is also effective in predicting the performance parameters in a combustion process.

The FRM is first verified with inputs from the steady state PLM data which suggests that the FRM follows the behaviour of the results from the original model except for exhaust temperature at lower torque regions.

Open loop simulations are run with the transient inputs from the BLB cycle on the GT Power FRM, the results from this simulations is been compared to the results from the engine test cell on the same drive cycle. Other than the exhaust temperature measured after the TC unit, most of the plots of performance parameters follow the physical engine test results.

Since the model is validated for transient inputs, the model is then integrated into the drive train closed loop simulations by replacing the existing empirical model to a GT Power FRM.

The integrated FRM engine model works well within the simulink environment with the inputs from a virtual EMS. Both in open loop and closed loop simulations the exhaust temperature plots were inconsistant. In order to improve the consistancy of the exhaust temperature the exhaust pipe requires a thermal network. The trend seems to be the same, but it needs to be accurate to have the right amount of temperature to keep the Exhaust After-Treatment System(EATS) warmer.

Hence, a co-simulation system with a Real Time detailed Fast Running Engine Model is established to perform drive train level closed loop simulations. The proposed method development flowchart shown in fig. 1.1 is advantageous during the early stages of a new project which eventually reduce the time and cost for physical testing.

Upon investigating the Miller cycle for different LIVC profiles on selected steady state operating points, there seems to be a benefit of 0.3%-2.6% in the BSFC values.

## 8.2 Recommendations for future work

From the simulations, the inconsistancies of the exhaust temperature was observed and the solution for this inconsistancy is to build a thermal network on the exhaust pipe after the TC unit. With the thermal network it is possible to add the characteristic details of a physical exhaust pipe such as the length of the pipe from the LP Turbine outlet to the entry of EATS, the thickness and diameter of the pipe which is actually used in production.

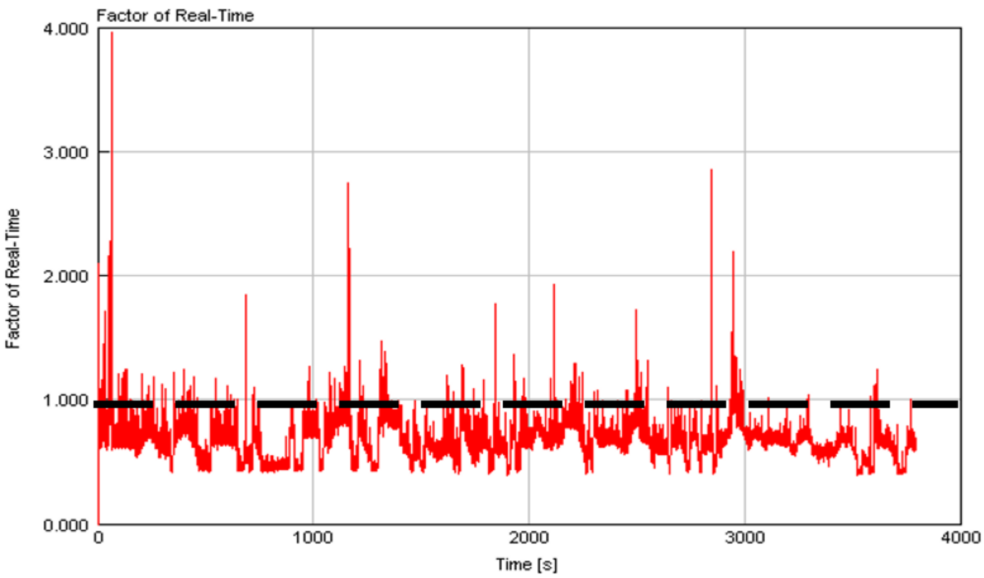



**Figure 8.1:** Factor of Real Time plot for the entire BLB drive cycle in GT Power

Fig. 8.1 shows the real time factor when the transient simulations are run on the BLB drive cycle with a GT Power license. It can be observed from fig. 8.1 that most of the operating points are below the Real Time margin that is the black dotted line. There are a few operating points which are above the RT margin. To avoid the operating points to be running above the RT margin, it is recommended to use RT License in GT Power for simulations, the reason being, the RT License restrict the operating points to not go beyond the RT margin.

The engine braking was not modelled in the open loop transient simulations and that was the reason why there was an aggressive behaviour of the brake torque in fig. 6.10. A suitable logic is required for the engine brake to be active.

The FRM used for transient simulations is built using a single main pulse for injection,it is necessary to model the pre and post injection which will be beneficial for reduction of noise and emission respectively.

Since there was a BSFC benefit seen in the Miller cycle investigation, it is known that with the BSFC benefit there is a trade off on the NOx emissions because of the fact of diesel dilemma. Therefore, a detailed investigation on Miller cycle is required.

Now that the Fast Running engine model can be used for co-simulations in GSP, it is recommended to apply the settings of the Miller cycle to the FRM and run drive train level simulations to test for transient condition.

Using the BSFC benefits from Miller cycle engine there is a possibility of improvising or redesign the built hybrid system as the BMEP of the engine reduces when the Miller cycle is implemented, which can be compensated with the torque from the electric machine.

# Bibliography


[1] G. Erbach, 'CO2 emission standards for heavy-duty vehicles', 2019, [Online]. Available: http://www.europarl.europa.eu/thinktank/en/document.html (January 2019)

[2] Heywood, John B. Internal Combustion Engine Fundamentals. New York :McGraw-Hill, 1988.

[3] R. M. Wagner, 'Engines of the future', December 11 2015, Oak Ridge National Laboratory.[Online]. Available: https://www.asme.org/topics-resources/content/engines-of-the-future, (January 2019)

[4] Gamma Technologies, GT Suite version 2019, 'Engine Performance Tutorials', Tutorial 9 : Building a Fast Running Engine Model (FRM).

[5] Romagnoli, Alessandro and D. Copeland, Colin and Seiler, Martin and Costall, Aaron and Ricardo, Martinez-Botas and Rajoo, Srithar. (2012). Comparison Between the Steady Performance of Double-Entry and Twin-Entry Turbocharger Turbines. Journal of Engineering for Gas Turbines and Power. 135. 10.1115/1.4006566.

[6] Liu, Y., Hong, S., and Ge, T., "Real-Time Hardware-in-the-Loop Simulation for Drivability Development," SAE Technical Paper 2017-01-0005, 2017, doi:10.4271/2017-01-0005.

[7] R. Gumaste, 'Application of GT-POWER to Model Failed Parts for On-Board Diagnostics', 2018, Cummins Inc. [Online], Available: https://www.gtisoft.com/wp-content/uploads/2018/11/Application-of-GT-POWER-to-Model-Failed-Parts-for-On-Board-Diagnostics-Rohan-Gumaste-Cummins.pdf (March 2019)

[8] Xia, F., Lee, S., Andert, J.,Kampmeier, A. et al., "Crank-Angle Resolved Real-Time Engine Modelling: A Seamless Transfer from Concept Design to HiL Testing," SAE Int. J. Engines 11(6):1385–1398, 2018, doi:10.4271/2018-01-1245.

[9] Wu, H. and Li, M., "A Hardware-in-the-Loop (HIL) Bench Test of a GT-Power Fast Running Model for Rapid Control Prototyping (RCP) Verification," SAE Technical Paper 2016-01-0549, 2016, doi:10.4271/2016-01-0549.

# A

# Appendix 1

The chapter explains few other plots of the results during the simplification and also a few other parameters for comparison in steady and transient condition. The plots will be shown in the same order as discussed in the results section of the report.

## A.1 Results of "Simplification for accuracy"

As it is known that the comparison between the baseline "Original" model and the FRM is done on 21 different operating points, the exhaust manifold simplification result are explained using fig. 6.1 in the Results chapter. Here, the plots of other simplification steps are shown from fig. A.1 to fig. A.4, the reduced flow volume may not capture the same physics as the "Original" and can be seen that the first three steps fig. 6.1, fig. A.1 and fig. A.2, the model still has some complexity built into it which can capture the phenomena and makes the operating points to overlap with each other. But gradually in fig. A.3 and fig. A.4, when the model starts to absorb a few of its objects, the accuracy seems to drop a bit. This may not be a huge impact in the "Simplify for accuracy" step but later in the process the objects will be further simplified. Therefore, the Heat Transfer Multipliers and the Orifice diameters are tuned such a way that the model behaves the same way as the complex model. The factor of real time plot is the same as shown in fig. 6.2.

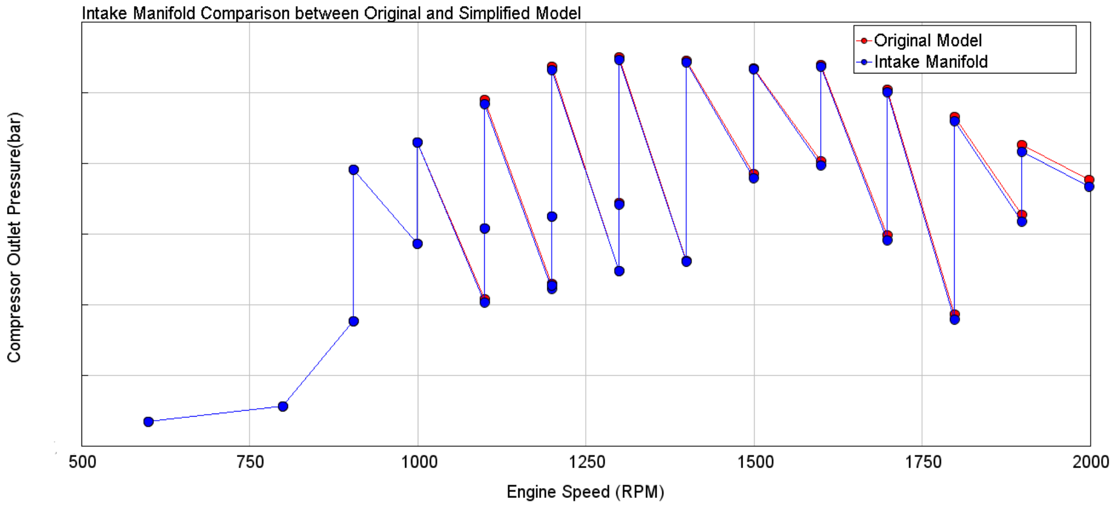



**Figure A.1:** Comparison of compressor outlet pressure after simplification for accuracy of intake manifold sub-model

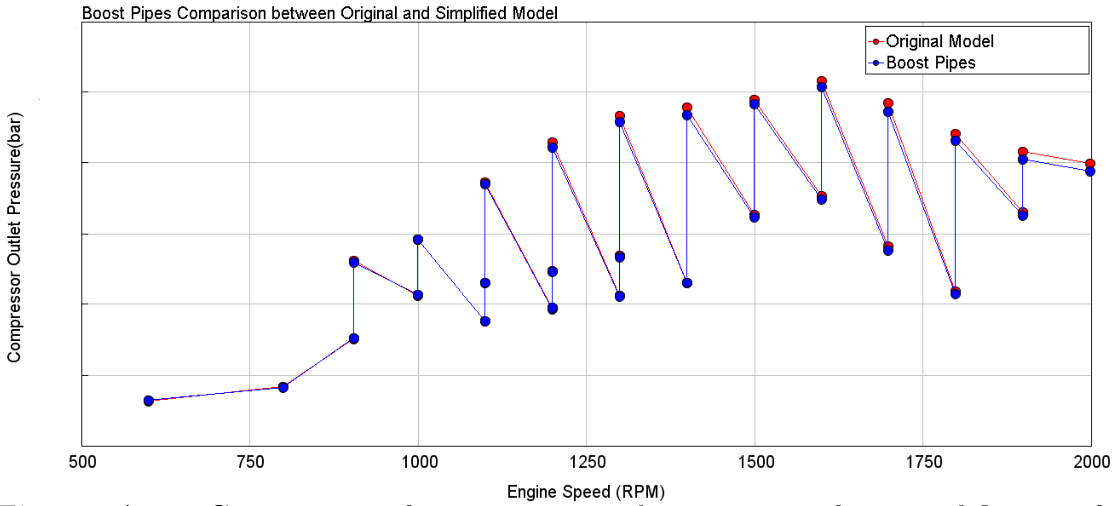



**Figure A.2:** Comparison of compressor outlet pressure after simplification for accuracy of boost pipes sub-model

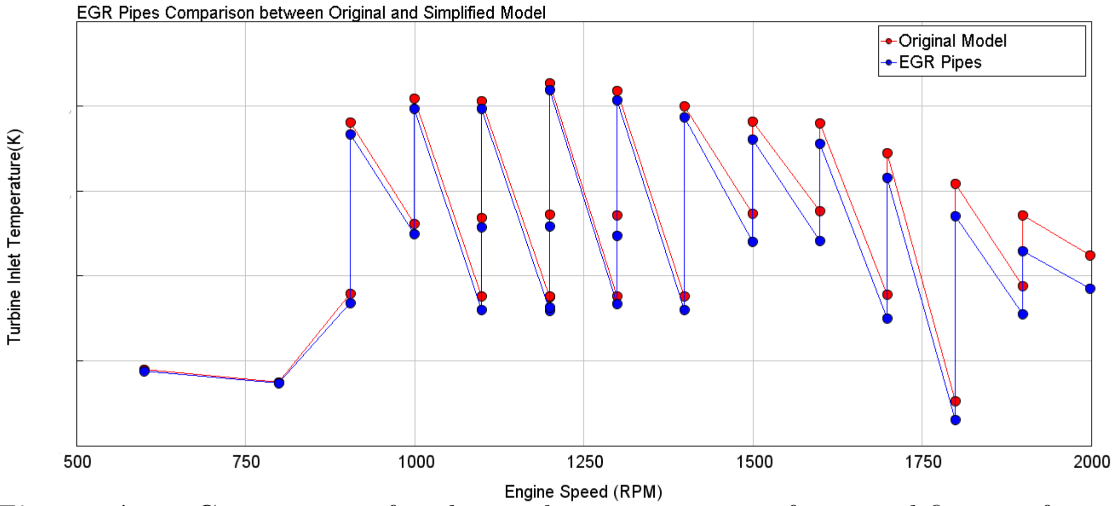



**Figure A.3:** Comparison of turbine inlet temperature after simplification for accuracy of EGR pipes sub-model

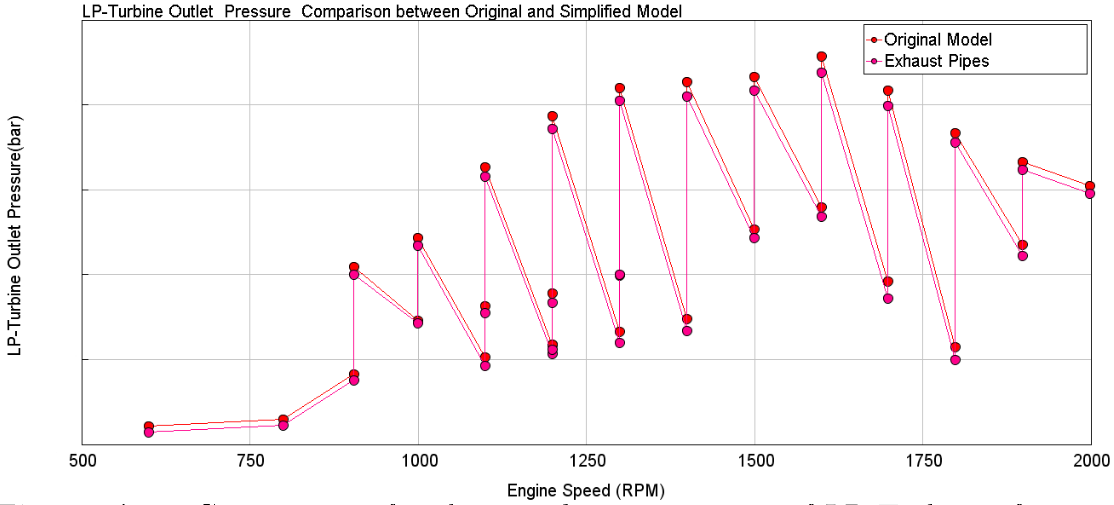



**Figure A.4:** Comparison of turbine outlet temperature of LP Turbine after simplification for accuracy of exhaust pipes sub-model

### A.1.1 Results of "Simplification for speed"

Similar to "Simplification for Accuracy" the remaining results of "Simplification for Accuracy" will also be shown. It can be seen that the operating points after simplification are fairly overlapping with each other which indicates good calibration. The graphs shown from fig. A.5 to fig. A.7 show similar trend as explained in the previous paragraph of "Simplification for accuracy". These graphs also suggest that there is a fair overlap with the "Original" model and the simplified model. However, it can be observed in fig. A.5 and fig. A.7 there seems to have a loss of accuracy on the first operating point which is at lower engine speed. This is mainly because the model with simplified flow volume is unable to capture the physics at really low engine speeds. A similar kind of loss of accuracy can be observed at higher engine speeds in fig. A.6.

There can also be another reason for this, while doing the calibration for 21 different operating points, it is known that based on the subsystem the Orifice diameters and the Heat Transfer Multipliers (HTM) are selected and the values will be different for each of the 21 different operating points. Then a trend of these values are studied and one common value of the diameter and the HTM is selected to represent all the 21 operating points, since it is physically not possible to have different HTM's and the orifice diameters for different operating point in the real engine. This selected common value may not be a suitable factor for all the selected operating point to actually match the Temperature for example or any other specific parameter, this can be more evident in fig. A.6. The graph of factor of real time has been explained using fig. 6.4.

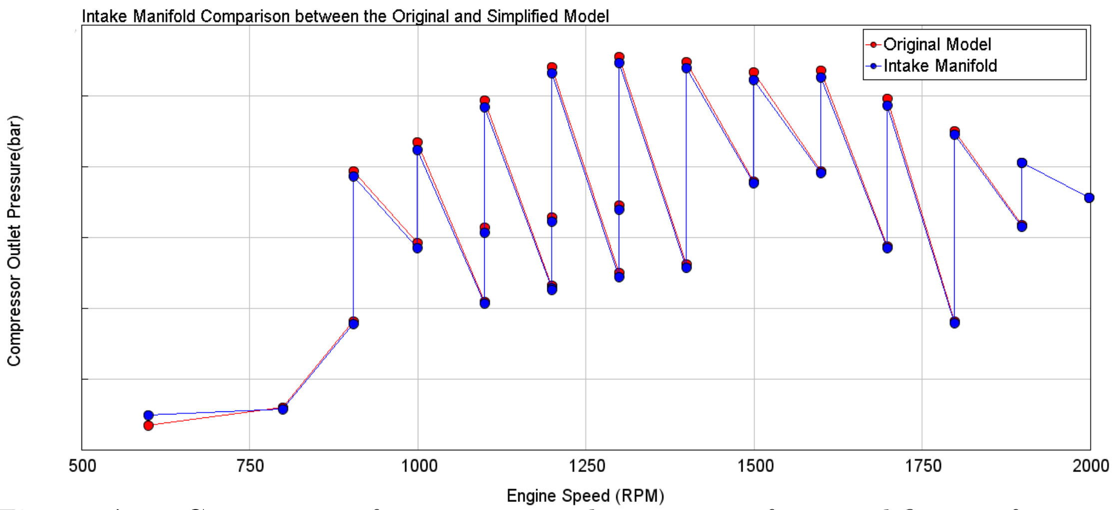



**Figure A.5:** Comparison of compressor outlet pressure after simplification for speed of intake manifold sub-model

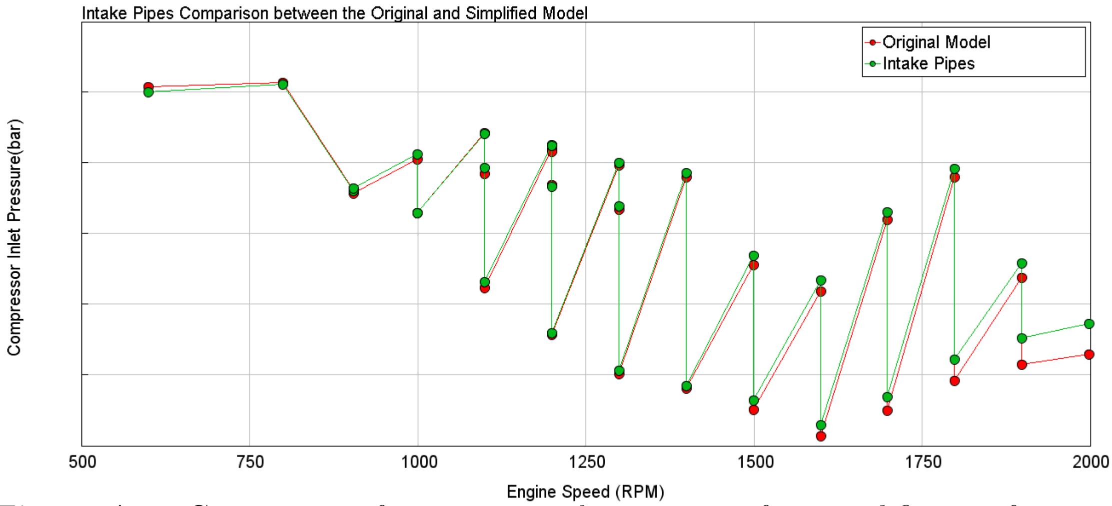



**Figure A.6:** Comparison of compressor inlet pressure after simplification for speed of intake pipes sub-model

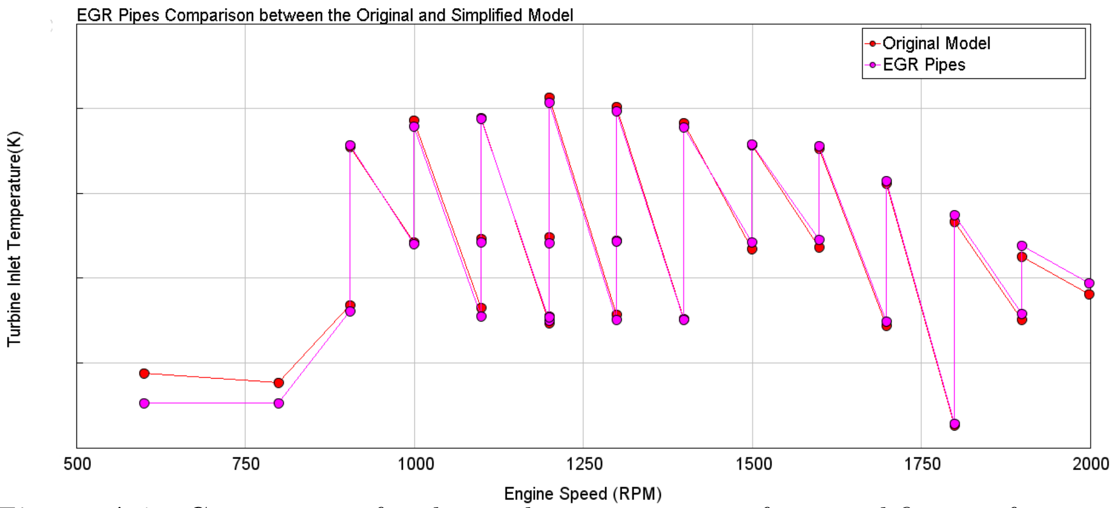



**Figure A.7:** Comparison of turbine inlet temperature after simplification for speed of EGR pipes sub-model

## A.2 Verification of FRM in Steady state and transient simulation

The plots of brake torque and exhaust temperature were already discussed in the results chapter. Here, the results of turbine inlet temperature, peak cylinder pressure and the compressor pressure will be shown. Under steady state, from fig. A.8 to fig. A.10 have a good level of accuracy. Fig. A.8 shows that the FRM values have a very little offset at lower engine speeds less than 1000 RPM, but later on the FRM behaves the same way as that of the behaviour of the original model.

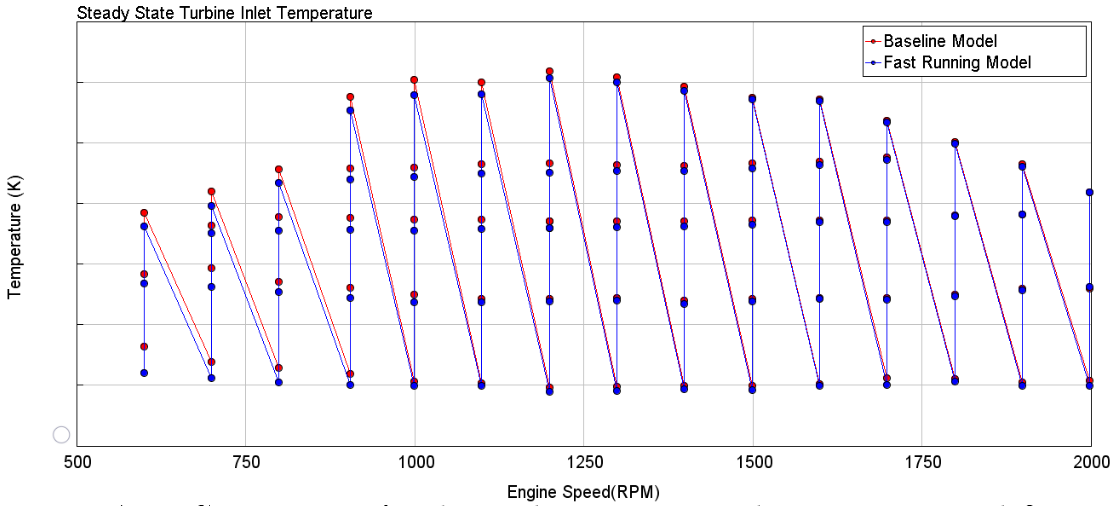



**Figure A.8:** Comparison of turbine inlet temperature between FRM and Original model

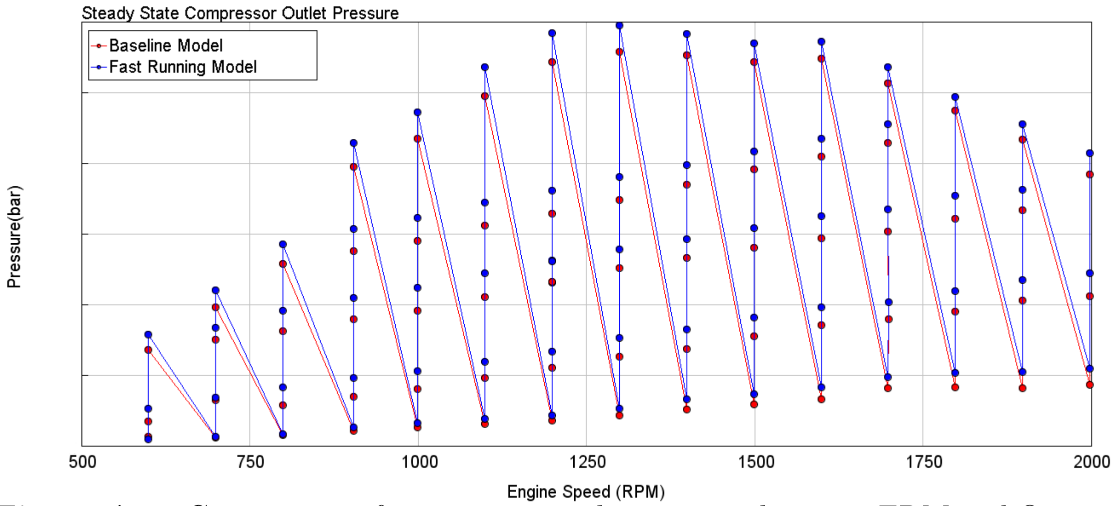



**Figure A.9:** Comparison of compressor outlet pressure between FRM and Original model

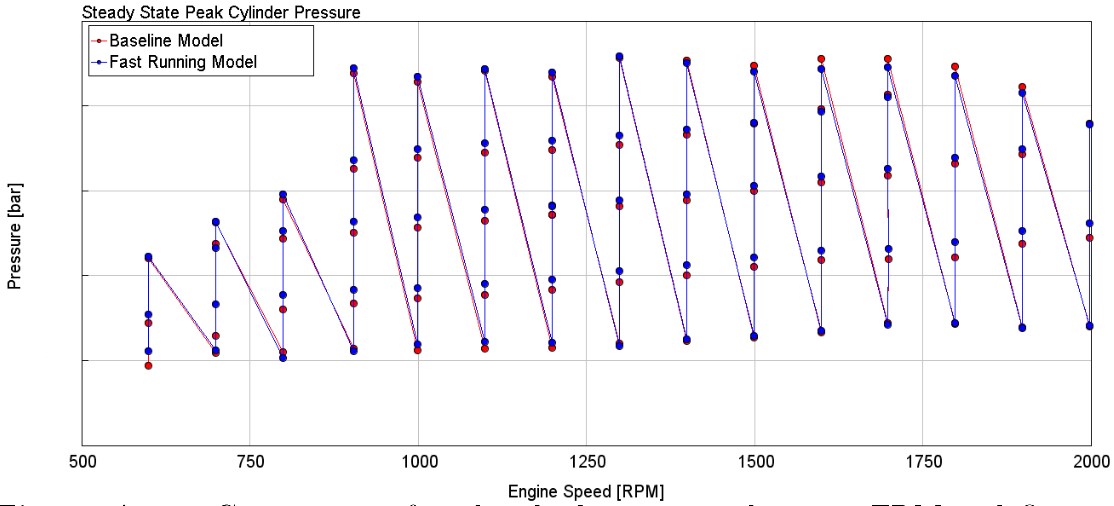



**Figure A.10:** Comparison of peak cylinder pressure between FRM and Original model

In fig. A.9 it can be seen that the difference in boost pressure is a constant offset between the values of FRM and the original model which is observed to have a minimum difference and the pressure prediction at low torque regions is fairly accurate. Fig. A.10 is the plot of maximum pressure inside the cylinder, it is good to notice here is that the values obtained from steady state simulations does not exceed the maximum pressure value which is around 220 bar and also the pressure values are overlapping with each other. This means to say that the prediction of air pressure, air quantity and also the fuel quantity in steady state are accurate using FRM.

## A.3 Model Accuracy Band

In chapter 3, "Building the Fast Running Engine Model" fig. 3.3 was used to define the accuracy limits for the FRM. Here, fig. A.11 and fig. A.12 are studied to check if the FRM simulated values are within the limits or not. Fig. A.11 is the result of the "Simplification for Accuracy" step, the operating points of each of the simplified sub-model is overlapped in one single plot and these operating points are within the accuracy band. Fig. A.12 shows the data of the operating points from the "Simplification for Speed" step to final manual simplification step. Majority of the operating points lie within the accuracy band but, from all the four plots it can be observed that at very low engine speed at around 600 RPM the value deviates and goes beyond the accuracy limits. There are also a few operating points at maximum engine speed that are outside the accuracy band.

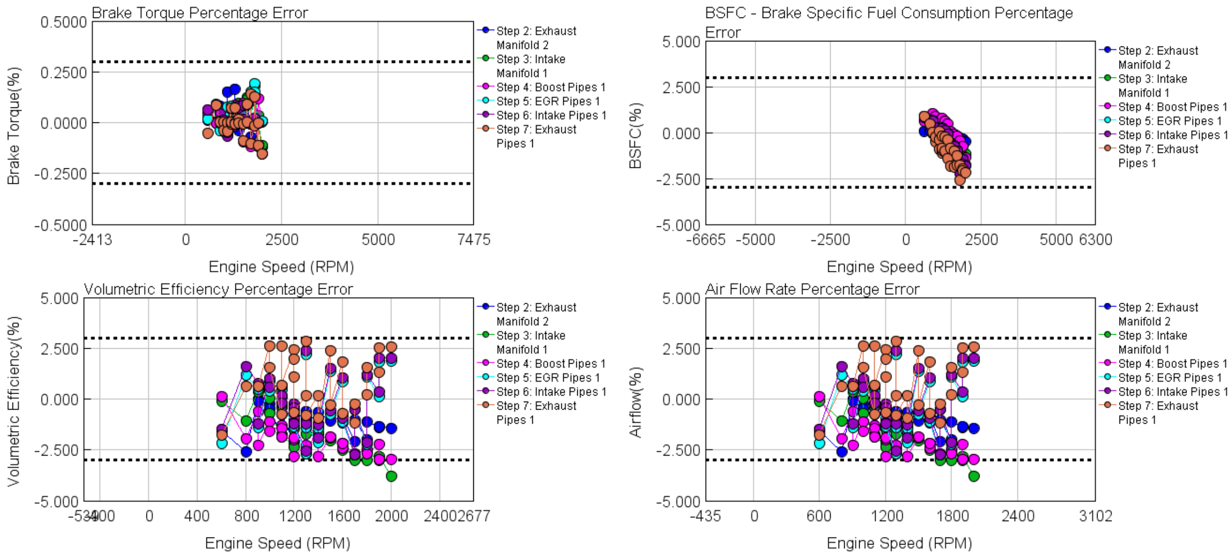



**Figure A.11:** Accuracy limits for simplification for accuracy

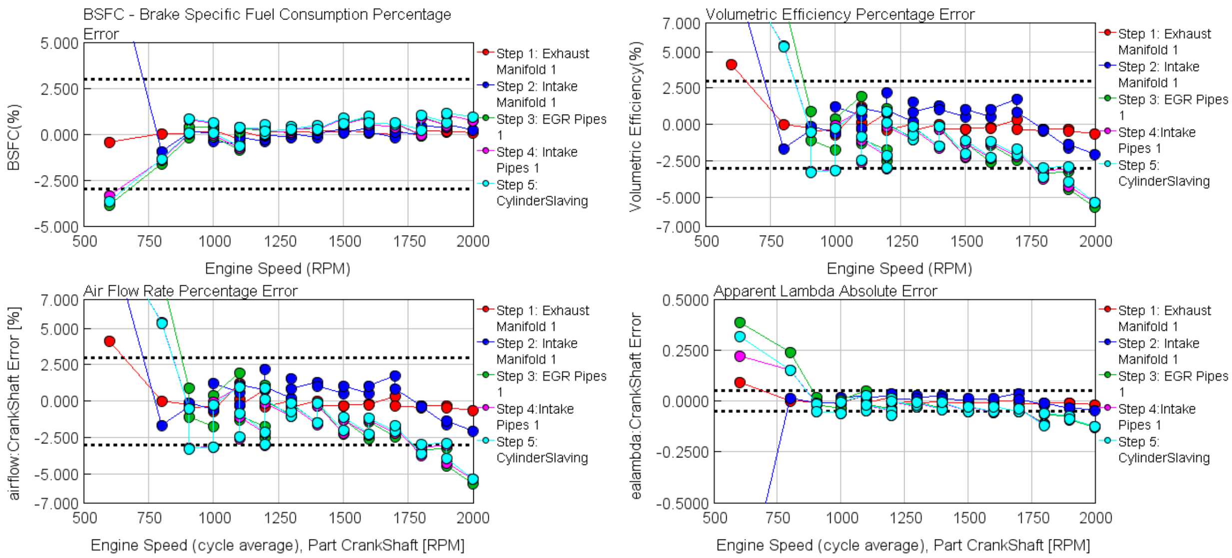



**Figure A.12:** Accuracy limits from simplification of speed to final manual simplification