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INVESTIGATING THE FEASIBILITY OF DOWNSIZING BY HAVING TWO SMALLER ENGINES IN AN AUTOMOBILE

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By

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April 2, 2015

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ABSTRACT

In this competitive era, there is an increasing demand to address the problem of powertrain matching in automobiles, one amongst the significant technologies out in the market that addresses such a problem, is cylinder deactivation technology. While this technology mitigates the part load losses in an automobile, due to its method of operation, is subjected to frictional losses from the deactivated cylinders, which hinder the additional benefit that could otherwise be capitalised on, to improve the fuel economy. For this reason, a new strategy is presented through this thesis, which eliminates the above discussed frictional losses even while addressing the part load performance of the engine.

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LIST OF ABBREVIATIONS/SYMBOLS

A/f Ratio	Air Fuel ratio
BMEF	Brake Mean Effective Pressure
BSFC	Brake Specific Fuel consumption
CCE	Cylinder Cut-out
CDA	Cylinder Deactivation
degC	degrees centigrade
EPA	Environmental Protection Agency
FTP	Federal test procedure
HWFET	Highway Fuel Economy Test
ISFC	Indicated Specific Fuel Consumption
MPG	Miles per gallon
PV	Pressure Volume
RPM	Revolutions per minute
VE	Volumetric Efficiency
WOT	Wide open throttle
V_{TDC}	Volume at Top Dead Centre
V_{BDC}	Volume at Bottom Dead Centre

1 Introduction

Globally, demand for automobiles with higher power, lower fuel consumption and lesser emission is on the rise. Automobile manufacturers, to stay current and to improve their product appeal, have been trying to enhance the running performance of a vehicle but are finding it difficult to cope up with the current fuel economy standards that are getting more stringent day by day. One of the most promising areas for increasing the fuel economy of automobiles lies in the area of decreasing the brake specific fuel consumption.

Brake specific fuel consumption, is the amount of fuel consumed per unit power produced in a vehicle, it gives a basic understanding of how efficiently the fuel is being utilized by the engine to produce power.

Ever since the automobile was introduced, a major challenge the engine designers have been trying to address is to decrease the brake specific fuel consumption in all operating conditions in an automobile.

The nature of the driving cycle plays a vital role in utilizing the brake specific consumption to its potential. Contemporarily, most people experience drive cycles that do not even set out to utilise the present-day engines at anything even close to their brake specific fuel consumption potential. This can be explained in detail by the problem statement below:

1.1 Problem Statement

Let us consider a naturally aspirated car, having the following specifications:

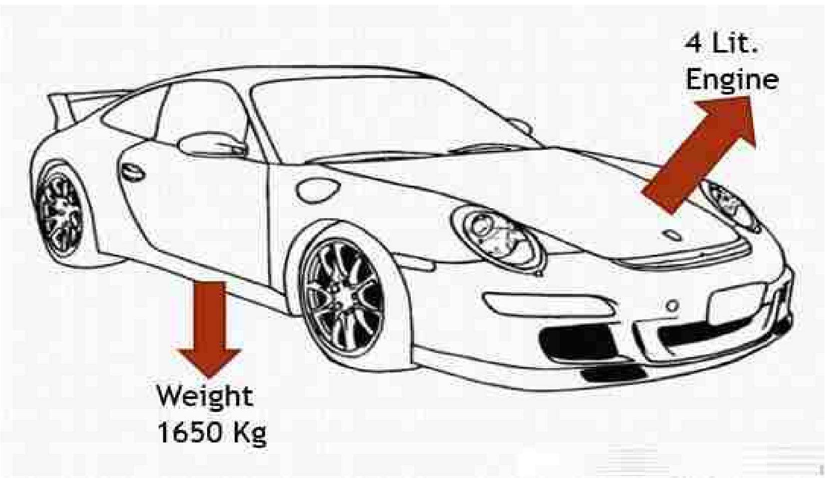

Figure 1.1.1: Car Specifications [9]

- Fuel Consumption City: 20 MPG
- Fuel Consumption Highway: 28 MPG
- Average City/Highway Fuel Consumption: 24 MPG
- Corresponding Average BSFC: 0.400 g/W-hr
- Optimum BSFC: 0.280 g/W-hr

When the car travels through city and highway conditions, the average fuel consumption is about 24 MPG, this corresponds to an average brake specific fuel consumption of about 0.400 g/W-hr but the average brake specific fuel consumption for this vehicle is around 0.280 g/W-hr [8].

Let us look at the driving time distribution graph to get a detailed picture of what is happening with the brake specific fuel consumption.

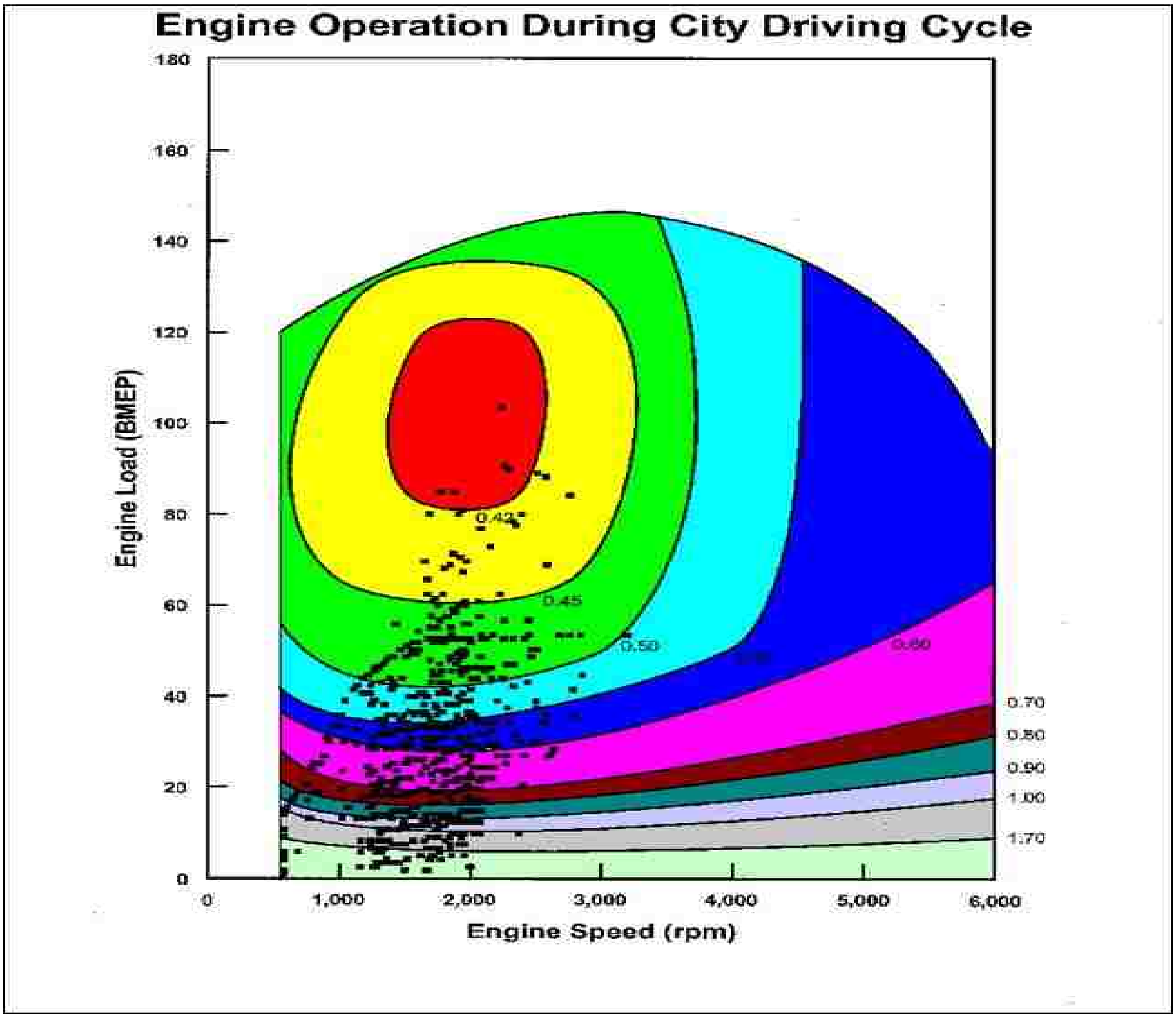

Figure 1.1.2: Driving time distribution [17]

It is clearly understood from the driving time distribution map that the typical road load values are nowhere close to utilizing the optimum brake specific fuel consumption of this car.

This can be further explained by looking into the engine speed vs torque map, where in, the brake specific fuel consumption values expressed in g/W-hr are placed as a matrix of values that are expressed as a function of engine torque and speed.

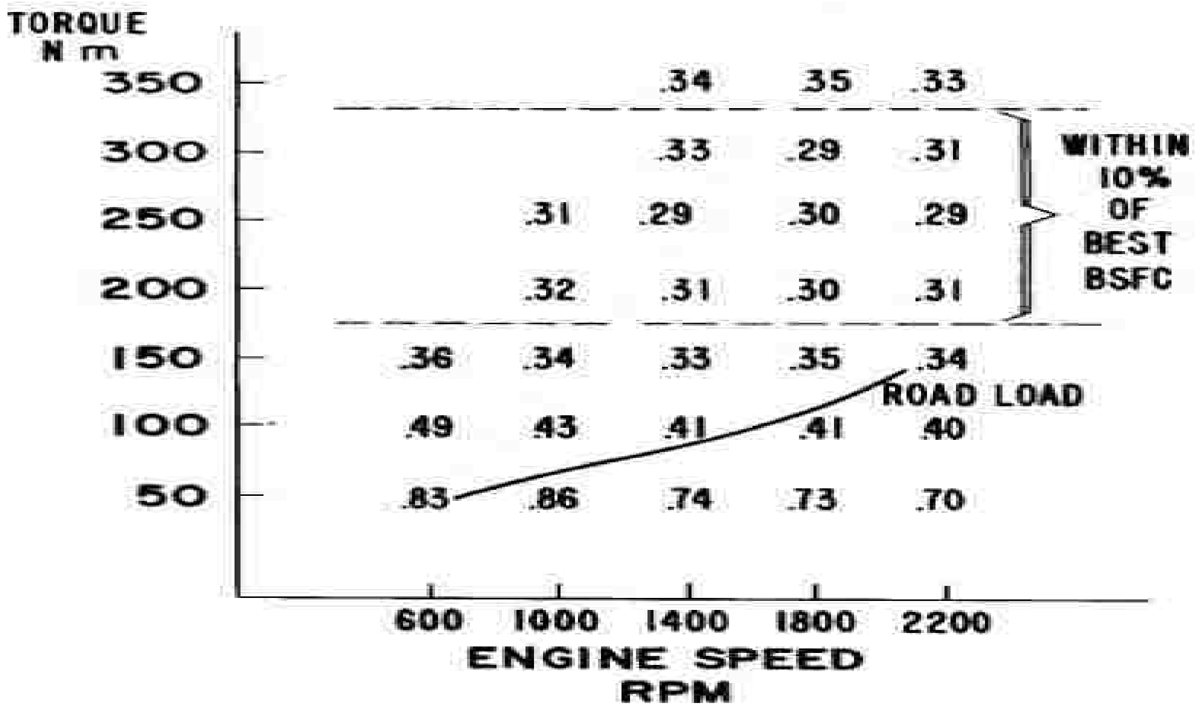

Figure 1.1.3: Effect of Road load on engine performance [8]

It can be observed that there is reasonably a broad band range for the engine to operate within 10% of its best brake specific fuel consumption values. If the vehicle could be used in such a manner that it operates within the 10% of the best brake specific fuel consumptions values during the entire driving cycle, it would achieve a mileage of about 37 MPG.

From the above map, it is understood that the 10% of the best brake specific fuel consumption band only begins at about twice the road load values. So, this problem of trying to use the engine in an optimum manner by designing it in such a way that the engine runs as efficiently as possible throughout the entire driving cycles is called a powertrain matching problem.

The only solution to such a problem would be to improve the brake specific fuel consumption over a wide operating range [8].

Let us try to understand the factors that hamper the brake specific fuel consumption over wide operating ranges.

During part load conditions, spark ignited engines restrict the flow of fresh charge into the engine by using a throttle. When the throttle valve is partially open, the air pressure inside the intake manifold is below atmospheric pressure and the piston, has to work against the partially opened throttle valve caused manifold depression in order to induct fresh air charge. This negative work done by the piston is called as pumping work and the loss incurred from this phenomenon is called as throttling loss.

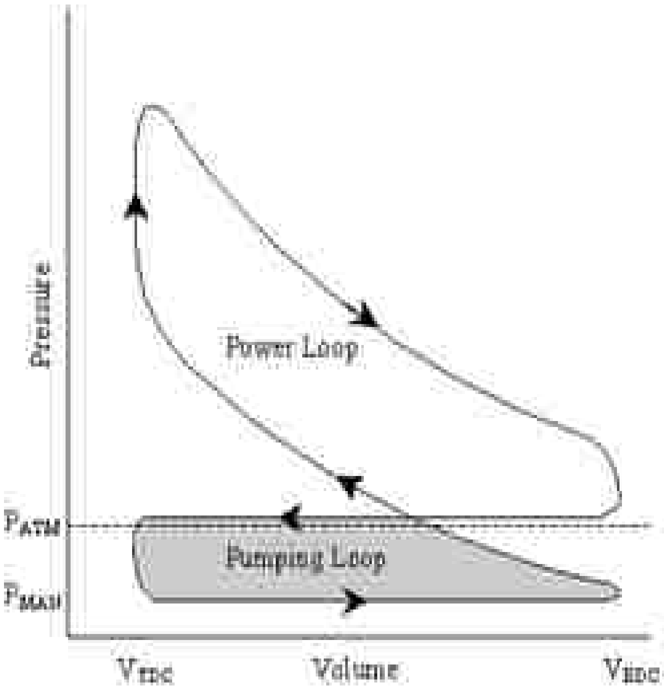

Figure 1.1.4: Engine pumping losses [10]

For a given engine output, if it is made possible to reduce the size of the pumping loop, which means less amount of power being lost in the induction process and a corresponding reduction in the area of the power loop is noticed. Hence, this reduction in power loop proportionately reduces the fresh charge requirement for a given engine output and the efficiency is improved.

It is also noticed that the indicated fuel consumption does not change as drastically as the brake specific fuel consumption changes in response to the variations in load. Shown in the Figure 1.1.5, is a graph of the changes in specific fuel consumption over wide rpm or intake manifold pressures. It is observed that the indicated specific fuel consumption changes only about 15%, while the brake specific fuel consumption changes about a 100% in response to changes in load.

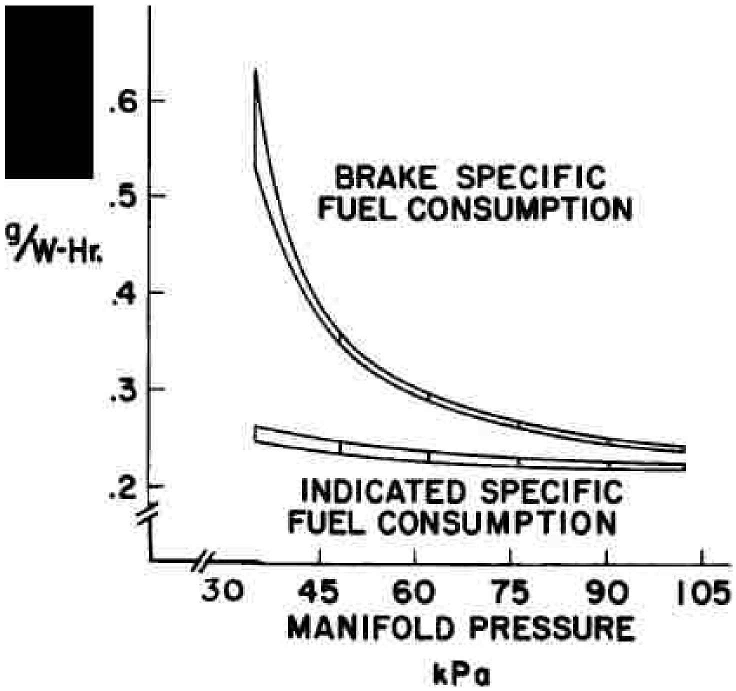

Figure 1.1.5: Variation in specific fuel consumption [8]

This implies that the task of improving on the BSFC is more a task of trying to cut down on the losses than trying to improve the Basic Combustion Efficiency.

In the next chapter, contemporary methods of improving the fuel economy by addressing the above discussed issues are briefly analysed.

2 Cylinder Deactivation Technology

2.1 Introduction

As discussed in the previous chapter, to eliminate the part load losses or the pumping losses in a vehicle, for a given output, the area of the pumping loop, in the PV diagram has to be reduced. The pumping work can be reduced by making the pressure in the intake manifold as close to atmospheric as possible, in other words, eliminating the pressure difference that exists between the intake manifold and outside environment leads to reduction of pumping losses in an engine.

Now to reduce this pressure difference, the throttle opening has to be increased, but the engine's power output is directly proportional to the throttle valve opening, so any increase in the throttle opening to reduce the pumping loss, would lead to an increase in the power output of the engine. Any abrupt change in the power output is not desirable to the driver. For this reason, depending on the need, a few cylinders in the engine are sometimes turned off before the throttle opening is increased to eliminate the part load losses. In this way the remainder of the active cylinders need to produce more power to compensate for the power lost from the deactivated cylinders and require the throttle valve to open wider and eliminate the pumping losses, which is the main objective of the cylinder deactivation technology.

Cylinder Deactivation Technology, is a means of improving the fuel economy in gasoline engines. Reduce the part load losses by varying the displacement of the engine, this is accomplished by having a control on number of active cylinders and cutting out or deactivating the cylinders that are not wanted during the part load operation.

Deactivation of cylinders can be done by keeping the intake and exhaust valves closed, either before the suction stroke or after the compression stroke. When deactivated, the gasses that get trapped in the combustion chamber are subjected to compression and expansion due to the up and down movement of the piston during the deactivation phase, this creates an air spring inside the combustion chamber which has virtually, no additional load on the engine.

This transition while activating and deactivating the cylinders is made seamless by making small changes to the ignition timing, camshaft timing and also, the throttle valve opening, all these modifications are controlled by sophisticated electronic control systems.

2.2 Types of Cylinder Deactivation Technologies

Cylinder deactivation technology can be classified into two kinds, depending on the type of valve train.

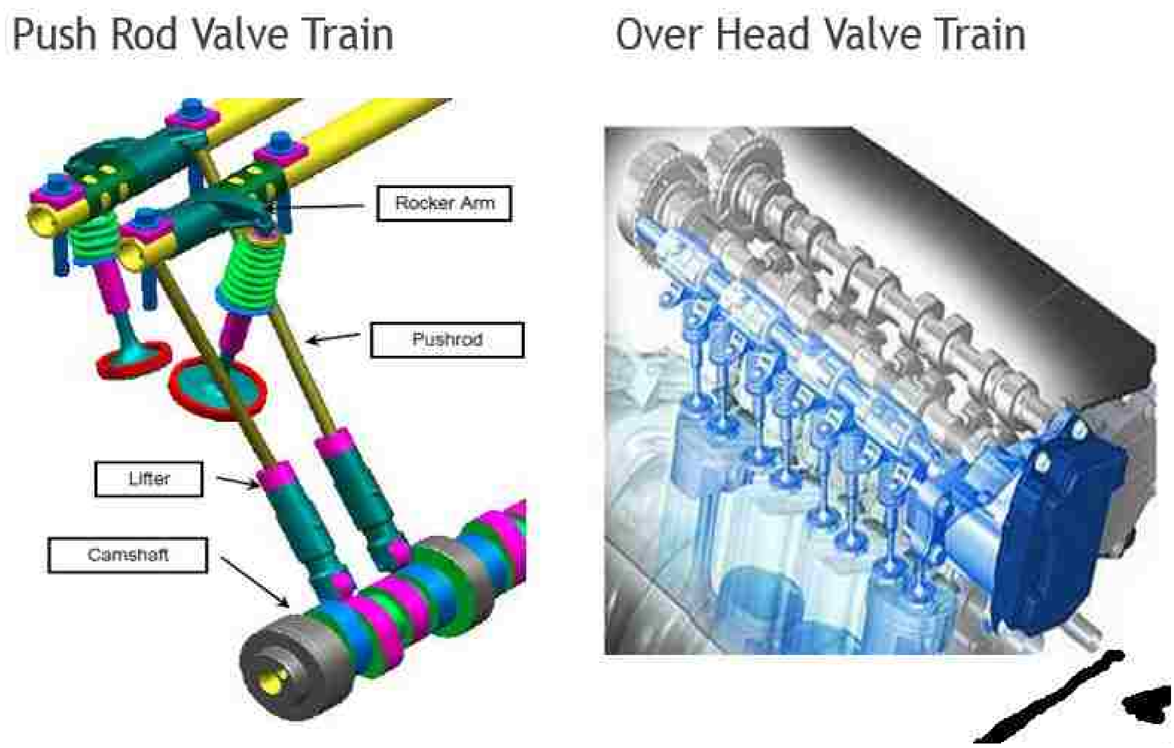

Figure 2.2.1: Types of deactivating mechanisms [4]

For push road valve trains, the deactivation is initiated at the lifter by decoupling the lifter and camshaft. A hydraulically operated pin is used to break the link between the lifter and camshaft.

For overhead valve trains, the deactivation is initiated at the rocker arm, again by the use of solenoid controlled hydraulically operated pin to engage and break the link with the remainder of the valve train.

It can be understood that this cylinder deactivation technology runs the cylinders at higher volumetric efficiency zones by increasing the throttle valve opening, eliminates the part load losses and also, at the same time reduces the valve train friction. The extent to which the valve train friction gets reduced depends on the mechanism of valve train operation.

Given below are the current OEM strategies and claimed economy benefits obtained from the use of cylinder deactivation technology.

	Chrysler Multi Displacement (MDS)	**GM Displacement on Demand (DOD)**	**Honda Variable Cylinder Management (VCM)**	**Mercedes Benz**	**Volkswagen**
Valve Train	Push Rod Design	Push Rod Design	Overhead Valve train design	Overhead Valve train design	Multi Piece cam shaft design
Decoupling location	Lifter	Lifter	Rocker Arm	Rocker Arm	Rocker Arm
Decoupling force	Oil Pressure	Oil Pressure	Oil Pressure	Oil Pressure	Electro Magnetic Actuator
Decoupling controlled by	Solenoid Valve	Solenoid Valve	Solenoid Valve	Solenoid Valve	
Claimed Benefit	City 7% improvement Steady CRUISE 20%	5.5% - 7.5% Fuel economy improvement	About 7-10% fuel economy improvement	About 7% fuel economy improvement	About 7% fuel economy improvement

Table 2.2.1: Current technologies [4, 7, 11]

The United States Environmental Protection Agency (EPA) claims that on an average, the Cylinder Deactivation Technology improves the fuel economy by 7.5%.

2.3 Disadvantages

Besides having an improvement in the fuel economy, the usage of this technology has the following drawbacks:

- The engine cools down unevenly, when the cylinders are deactivated, the exhaust gas is entrapped in the combustion chamber, the heat generated from compression and expansion of this exhaust gas sets in an uneven pattern in the thermal model of the engine and also makes these deactivated cylinders cool down slowly.

- A large quantity of the confined gas creates an air spring inside the combustion chamber, leads to producing different pressures inside this chamber and would further lead to producing greater irregularities in forces on the crankshaft.
- Care should be taken to ascertain that no vacuum or suction force is produce inside the combustion chamber, as this would cause the crankcase engine oil to be drawn into the chamber.
- Even though the cylinders are deactivated, there is power loss incurred from the reciprocating pistons, this loss accounts to about 65-80% of the frictional mean effective pressure in an engine [7].

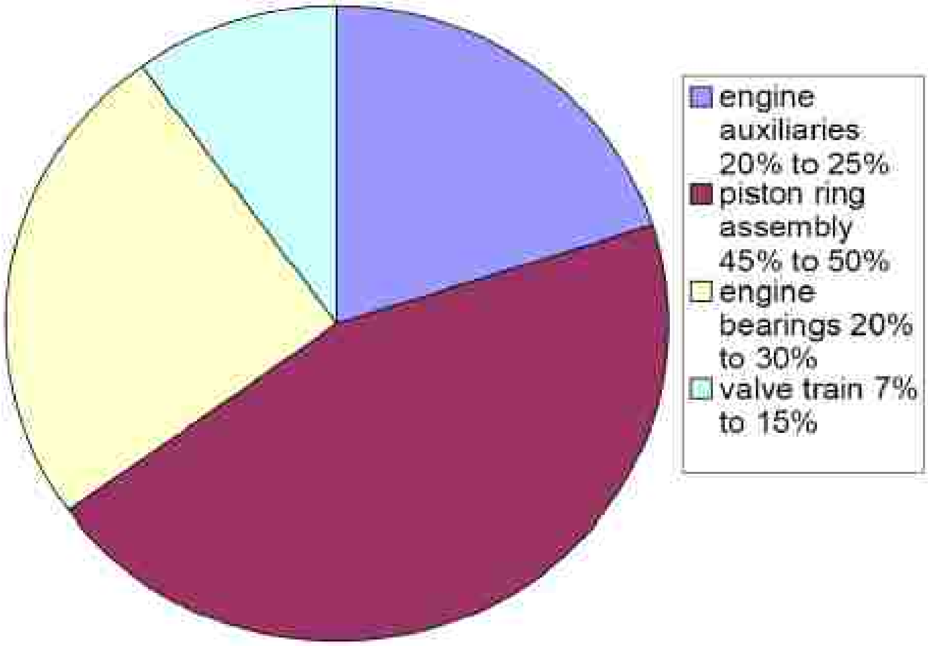

Figure 2.3.1: Frictional losses in an engine [7]

- It is only this 7-15% valve train friction, shown in the above pie chart, the cylinder deactivation technology manages to save.
- A major chunk of loss is incurred through the friction from piston assembly and the bearings, this loss in total sums up to 65-80% of the friction mean effective pressure in an engine.
- In order to analyse the variation in friction mean effective pressure over a wide range of engine load, as depicted in Table 2.3.1, friction mean effective pressure represented as a function of rpm and brake mean effective pressure, is taken into consideration.

RPM	BMEP (Bar)	FMEP (Bar)
6000	10.044	2.19334
5500	10.4799	2.07223
5000	10.7896	1.94741
4500	11.1176	1.83086
4000	11.5497	1.71931
3500	11.2842	1.57739
3000	10.6337	1.42645
2500	10.3781	1.2733
2000	10.1414	1.12335
1500	9.99601	0.979237
1000	8.755	0.809341
800	8.15763	0.748192

Table 2.3.1: Engine data at wide open throttle (WOT)

- The above data is collected during the wide open throttle conditions (WOT).
- It can be inferred that the frictional mean effective pressure is about 10-20% of the brake mean effective pressure, when the speed ranges from idle to maximum rpm.
- For example, consider an 8-cylinder engine, running at 3000 rpm, which deactivates four of its cylinders, at wide open throttle. It can be inferred from the above figure that each cylinder produces 10.6 bars of brake pressure and 1.43 bars of frictional pressure. When we sum up the frictional loss from all the 8 engines, it accounts to 11.3 bars, which is equal to the power produced by a single cylinder at the same time.
- This means that even if you use the cylinder deactivation technology, the frictional loss is so high that it eats up the entire power produced by a single cylinder running at the same instance of time.

For this reason, there is a need to curb this frictional loss while trying to improve the part load performance to experience a maximum benefit.

In in the next chapter, a new strategy is developed to curb the frictional loss while improving the part load performance of the engine.

3 Proposed New Strategy

For reasons explained in the previous chapter, there is an increasing need to develop a fuel saving strategy that averts the piston motion while addressing the problem of powertrain matching.

This is only possible if the motion of the piston is ceased while the throttle position is being varied. For the motion to be ceased, the piston or crank shaft should possess the ability to disengage when required.

Therefore, this requirement lead to the development of a separate engine strategy, instead of a single large engine, two separate smaller engines would be connected to the powertrain and one of the engines would turn off when not required. This would completely decouple the engine from the powertrain and hence, the frictional losses are curbed while improving the part load performance.

For example, in case a vehicle has an 8-cyliner engine that has a cubic capacity of 4 litres, it could be replaced with two 4-cylinder engines, having a cubic capacity of 2 litres.

These separate engines can be connected in two possible ways.

3.1 Parallel Connection:

Engines can be connected in parallel by the use of a planetary gear system. Shown above, is an example of the planetary gear system layout. It consists of a ring gear, sun gear and a set of planet gears that connect by a carrier. This gear system is used in many systems, mostly in the hybrid electric vehicles, where the ring, the sun and the carrier or a combination of these can be the driven while another shaft is the output. A realistic model has losses due to friction, but modelling these losses is very complicated.

The efficiency of a planetary gear system depends on which gears are driven and the ratio of teeth between different gears. Devising a model that includes this friction model at all configurations is quite complicated. It is also understood that the input power passes through a sequence of gear meshes before it comes out through an output shaft of a planetary gear, the number of gear meshes depends on the combination of driven gear and the output shafts.

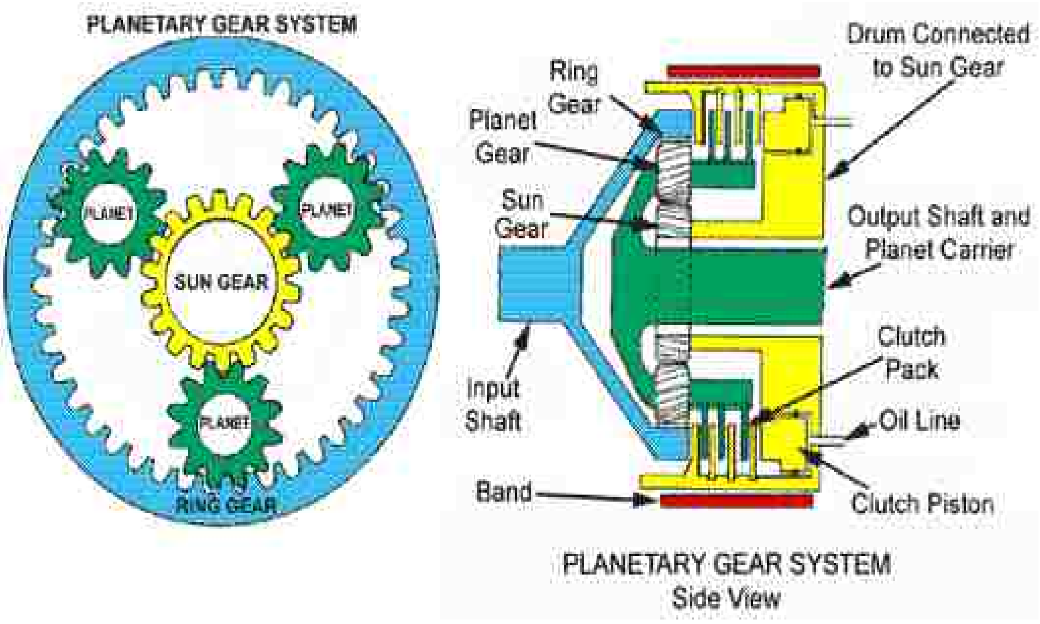

Figure 3.1.1: Planetary gear system layout [12]

Every gear mesh has about 2-3% loss in power, so in case we use the planetary gear, due to the sequence of gear meshes involved in this model, the output power is only about 94-95% of the input power, or in other words, the usage of this model causes about 5-6% loss in the power. To avoid the above stated power loss, a method described below is chosen for connecting the two smaller separate engines.

3.2 Series Connection:

A friction clutch as shown in the figure below, is used to connect the engines in series.

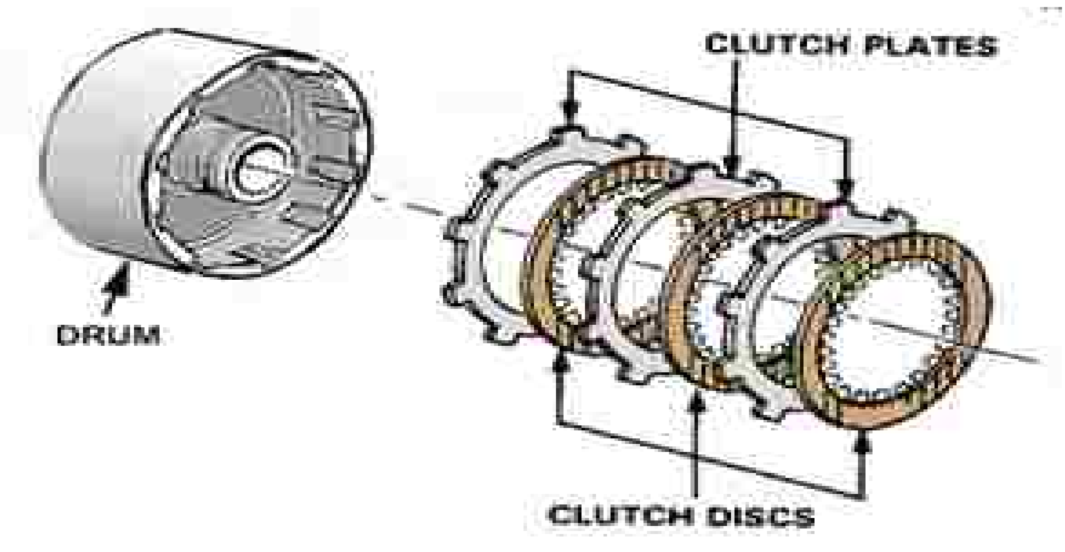

Figure 3.2.1: Friction clutch components [24]

The clutches are connected to the flywheel of the engine, so a minor modification has to be made to the engine that sits in the front, for this engine a flywheel has to be attached to the crank shaft and connects on the rear side, i.e., on the side that is on the other end of the conventional flywheel location.

The flywheel is equipped with a friction surface, and this is pressed against the friction surface of the clutch discs, this makes the flywheel lock with the clutch and rotate as one unit.

This additional flywheel does not need to have any teeth on its circumference, as it would not be connected to the starter motor, a friction clutch is attached to this additional flywheel and at the other end of this friction clutch is the flywheel of the second engine. The layout of this model is shown in the image below

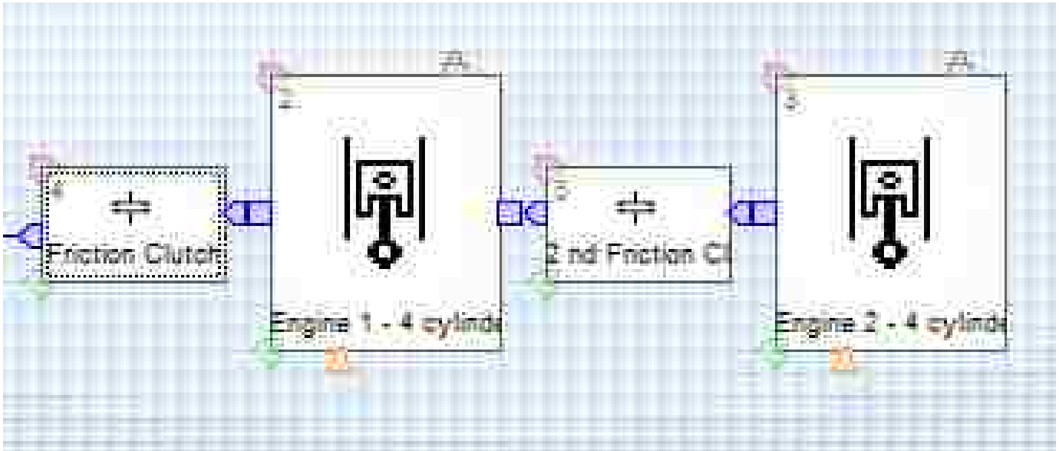

Figure 3.2.2: Separate engine strategy

This can be correlated to the cylinder deactivation technology, when the cylinders are deactivated in the CDA engine, in this model, the second engine is turned off and the second friction clutch disengages the motion between the two engines.

This way, the frictional losses are eliminated while improving the part load performance of the vehicle.

4 Methodology

The proposed separate engine strategy is simulated and compared against the conventional engine as well as cylinder deactivation engine and the fuel economy benefit if any, observed.

The following models are built for this project:

- A 4-cylinder conventional engine
- An 8-cylinder conventional engine
- An 8-cylinder cylinder deactivation compatible engine

The following steps are taken to design and simulate in this project:

- Firstly, the engine models are built in AVL BOOST software by defining the following basic input data
    - bore, stroke, number of cylinders, connecting rod length
    - numbering of cylinders, principle arrangement of manifolds
    - compression ratio, firing order and firing intervals
    - number of valves, inner valve seat diameters
    - valve lift curves
- The simulated engine data from the first step, is input to the Vehicle model in AVL CRUISE.
- Modifications to the powertrain in case of the proposed separated engine technology is accommodated by:
    - Adding the moments of the additional flywheel to the first engine model
    - Adding the weights of the additional components, used to accommodate this proposed separate engine strategy, to the vehicle model
- Running all the vehicle models through drive cycles, which simulate the real life driving conditions.
- Finally, comparing and analysing the results.

5 Engine Model Creation

5.1 AVL BOOST

BOOST is used for simulating a wide range of engines, spark ignited, compression ignited, 2-stroke or 4-stroke. The applications range from small capacity engines such as motorcycles up to very large engines used for marine propulsion.

BOOST comes with an interactive pre-processor that helps in preparing the input data for the main simulation program. An interactive post-processor supports the results analysis. The pre-processor of AVL BOOST is equipped with a model editor. The computational model of an engine is defined by using the element tree to select the required elements and these are connected by piped elements. Availability of a wide range of elements make BOOST easier to model complex engine configurations.

The BOOST program also provides optimized simulation algorithms for all the elements. The pipe flow is treated as one-dimensional. This implies that the attributes related to flow such as flow velocity, temperatures and pressures are obtained by solving the gas dynamic equations which represent the mean values over the pipe cross-sections.

5.2 Creating the Model

AVL BOOST is used to create the following engine models.

- A 4-Cylinder Conventional Engine
- An 8-Cylinder Conventional Engine
- An 8-Cylinder Engine having 4 cylinders deactivated

The first step to design an engine is to identify the required components, place them in the working area and finally, complete the setup by providing the required connections in between the components.

The following Figure 5.2.1, is a 4 cylinder conventional engine model after the completion of the initial setup:

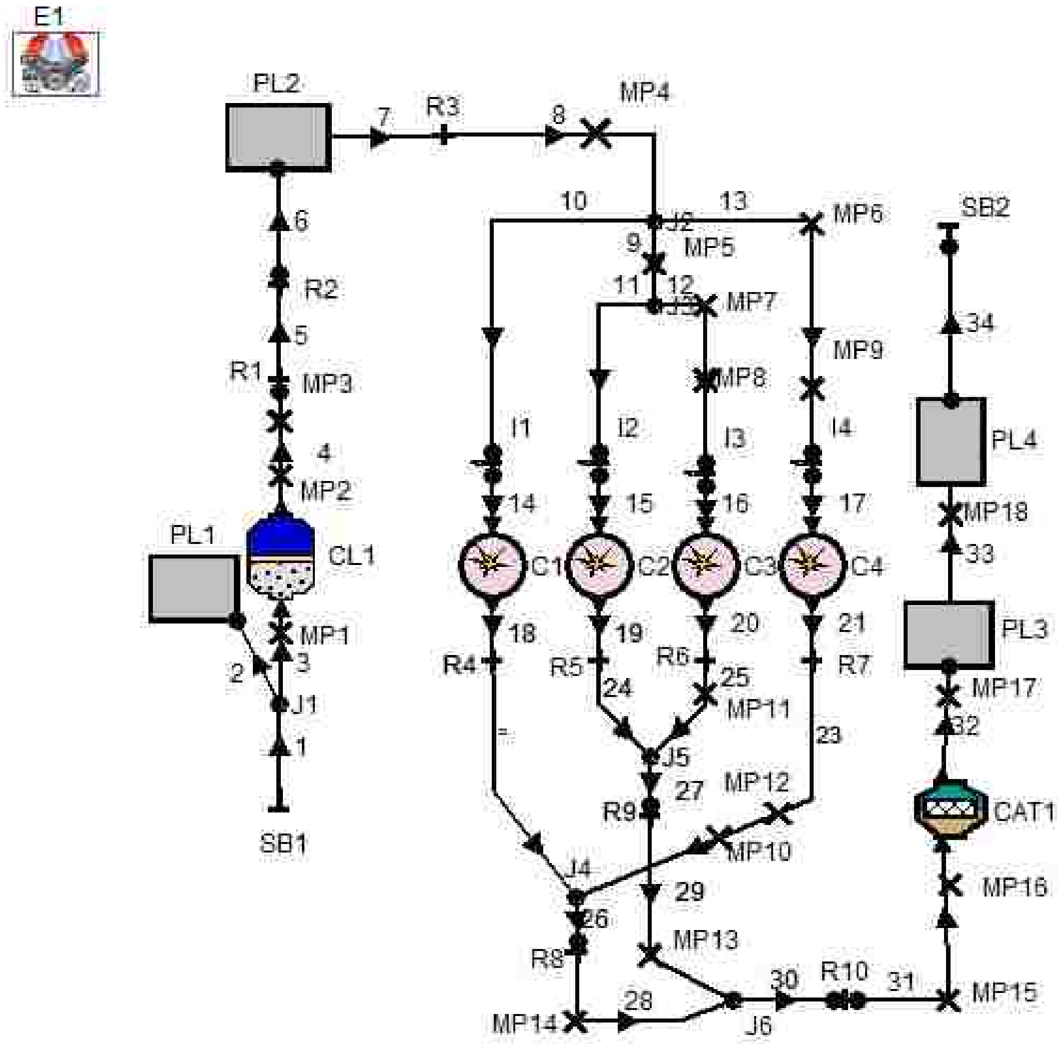

Figure 5.2.1: 4-cylinder engine model

5.3 Defining the Input Data

In Boost it is required to specify the general input data prior to the input of any other element data. The general input data regarding the simulation is as follows:

Cycle Simulation	
Simulation Interval: End of Simulation	40 Cycles
Convergence Control	Activate

Spatial Pipe Discretization: Average Cell Size	30mm

Table 5.3.1: Cycle Simulation data

The simulation interval is set to 40 cycles to make sure that the results converge to provide a reliable data.

5.3.1 System Boundary 1 or the Inlet Environment

In the first step to model an engine, we define the inlet boundary conditions. In this model System Boundary 1, element is used to do this. The following data is used to build the inlet boundary conditions:

Pressure	0.995 bar
Gas Temperature	30.85 deg C
Fuel Vapour	0
Combustion Products	0
A/F Ratio	10000

Table 5.3.2: System boundary 1

5.3.2 System Boundary 2 or the Outlet Environment

The boundary conditions used for the outlet are as follows:

Pressure	0.995 bar
Gas Temperature	676.85 deg C
Fuel Vapour	0
Combustion Products	1
A/F Ratio	14.3

Table 5.3.3: System Boundary 2

5.3.3 Engine

While defining the engine, the engine speed is set as a parameter value, these values are later entered as steps of 500 through to the maximum engine speed, which is 6000 in this case. This is done to obtain the engine data for various rpm values. The following data is used to build the engine:

[redacted] Engine Firing Order

The following firing order is considered for a 4-cylinder engine:

Cylinder	Firing Angle (Degrees)
1	0
2	540
3	180
4	360

Table 5.3.4: 4-cylinder engine firing order

The following firing order is considered for an 8-cylinder engine:

Cylinder	Firing Angle (Degrees)
1	0
2	180
3	450
4	630
5	270
6	90
7	540
8	360

Table 5.3.5: 8-cylinder engine firing order

[redacted] Friction

Engine friction affects the work output as well as fuel economy characteristics of the engine. Engine friction data is entered as a function of rpm, the following data is used to define the engine friction for a 4-cylinder engine:

Engine Speed RPM	Friction Mean Effective Pressure (FMEP) bar
800	0.6
6000	2.89

Table 5.3.6: 4-Cylinder Engine Friction

The following friction data is used for an 8-cylinder engine:

Engine Speed RPM	Friction Mean Effective Pressure (FMEP) bar
800	0.6
6000	3.2

Table 5.3.7: 8-Cylinder Engine Friction

5.3.4 Cylinder

Next comes building of the engine cylinder element that covers basic dimensions of the cylinder and crank train such as stroke, bore, compression ratio, connecting road length, piston pin offset, etc.., also, information regarding combustion characteristics, scavenging process and the valve or port specifications for the attached pipes have to be specified.

The piston motion is computed from the lengths of stroke, connecting rod and piston pin offset. The direction of positive offset is computed as the rotational direction of crankshaft at the top dead center.

The relative piston position is calculated as the distance of piston from the top dead center position relative to full stroke. A zero degree crank angle corresponds to the firing at top dead center of a selected cylinder.

While computing the blow-by from this model, average crankcase pressure and the effective blow-by gas has to be specified. Cylinder, crank case conditions and effective flow area are used to calculate the actual blow-by mass. To calculate the effective flow area, cylinder circumference and effective blow-by gap are considered.

The following data is used to build the engine:

Bore	86 mm
Stroke	86 mm
Compression Ratio	10.5
Con-Rod Length	143.5 mm
Piston Pin Offset	0 mm
Effective Blow By Gap	0 mm
Mean Crankcase Press.	1 bar

Table 5.3.8: Engine data

The scavenging model used in this project is perfect mixing, this means that the gas going into the cylinder is immediately mixed with the contents of the cylinder. Also, the gas going out of the cylinder has the same mixture composition as the gas that was in the cylinder. This perfect mixing model is considered to be a standard model to simulate the scavenging in a 4-stroke engine.

The initial conditions at exhaust valve opening are taken as follows:

Pressure	5 bar
Temperature	726.85 deg C
A/Fuel Ratio	14.3
Combustion Products	1

Table 5.3.9: Initial Conditions

The simulation of the conditions inside the cylinder are started with the opening of exhaust valve and not before.

[redacted] Combustion

A combustion model can either be described as predictive or non-predictive.

Predictive Model: A combustion model where in, the burn rate is computed from various inputs, such as gas pressure, fuel, temperature, equivalence ratio and residual fraction. This burn rate is then applied in the combustion simulation.

Non-Predictive Model: A combustion model where in, the burn rate is directly imposed as a simulation input. In a non-predictive model, cylinder pressure or residual fraction have no effect on burn rate. Fuel and air will burn at a prescribed rate.

Predictive Combustion	Non-Predictive Combustion
The burn rate is predicted from various inputs such as fuel, gas pressure, temperature, equivalence ratio... etc.	The burn rate is imposed by the user
Self-adjusting for transient conditions	Not affected by operating conditions
No experimental data needed	Required experimental combustion data at all operating points
Slow Computation	Fast Computation
Higher accuracy of results	Lower Accuracy of results

Table 5.3.10 Predictive combustion Vs Non Predictive combustion

As the accuracy of results is high, a predictive combustion model, Vibe, is used to define the combustion model.

The Vibe function is a very convenient method for describing the heat release characteristics. It is defined by the start and duration of combustion, a shape parameter 'm' and the parameter 'a'. These values can be specified either as constant values or dependent on engine speed (in rpm) and engine load (expressed as BMEP in bar). The values considered in building the model are taken from the example models in the software’s user guide.

Start of Combustion	-5 deg
Combustion Duration	47 deg
Shaping Parameter m	1.6

Parameter a	6.9

Table 5.3.11: Combustion Model

The heat release characteristic of spark ignition engines, with homogeneous mixture distribution inside the cylinder, is essentially determined by the speed of flame propagation and combustion chamber's shape. A very high flame propagation speed can be obtained with a combination of high turbulence levels and high compression ratio inside the cylinder. For precise and accurate engine simulations the actual heat release characteristic of the engine, (which can be obtained by an analysis of the measured cylinder pressure history), should be matched as accurately as possible. To get an estimate on the required combustion duration to achieve a certain crank angle interval between 10% and 90% mass fraction burned (MFB), the following chart (Figure 5.3.1) is used [14]:

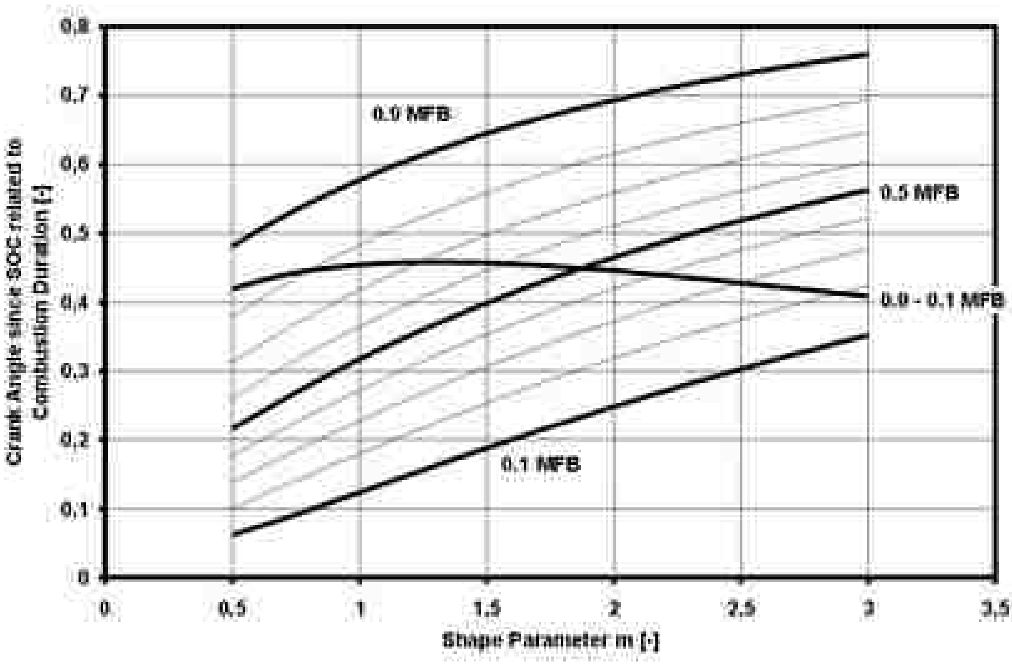

Figure 5.3.1: Crank Angle related to Combustion Duration [14]

For example:

"*A shape parameter of 1.5 is selected and the duration between 10% and 90% MFB is 30 degrees CRA. The crank angle interval between 10% and 90% MFB related to the combustion duration is 0.46 (from the graph). Hence the combustion duration is 30/0.46 = 65 degrees CRA. The point of 50% MFB is at 10 degrees CRA ATDC. According to the graph the location* *of 50 % MFB after combustion start related to the combustion duration is 0.4. Thus the combustion start is calculated from 10 - 65 * 0.4 = -16 = 16 degrees BTDC*" [14].

[redacted] Valves

Valve dimensions at the intake and exhaust are specified as follows:

Intake	
Inner valve seat diameter	43.84 mm
Valve clearance	0 mm
Scaling factor for effective flow area	1.72
Exhaust	
Inner valve seat diameter	36.77 mm
Valve clearance	0 mm
Scaling factor for effective flow area	1.242

Table 5.3.12: Valve dimensional data

The data, used to define the valve lift, is show as curve below, having crank angle as function of the valve lift.

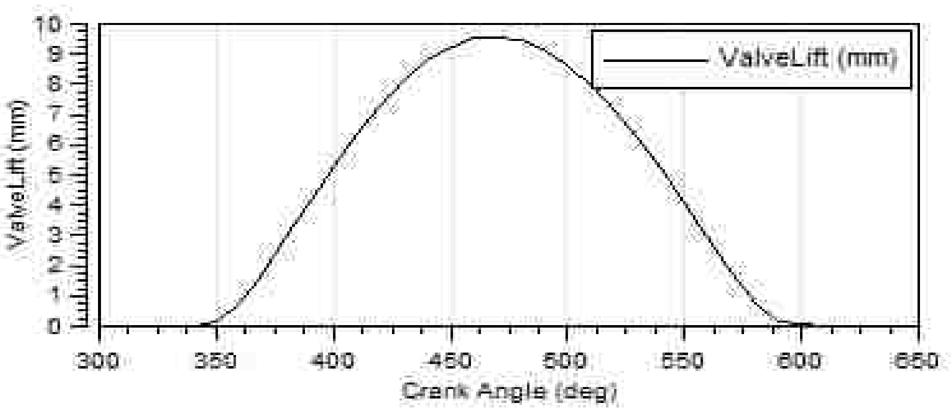

Figure 5.3.2: Valve lift curve [14]

[redacted] Heat Transfer

In addition to the heat transfer coefficient provided by the heat transfer model, the surface areas and wall temperatures of the piston, cylinder head and liner must be specified.

The wall temperatures are defined as the mean temperature over the surface.

A calibration factor for each surface may be used to increase or to reduce the heat transfer.

For the surface areas the following values are considered:

5.3.4.3.1 Piston:

SI engines: Surface area is approximately equal to the bore area.

5.3.4.3.2 Cylinder Head:

SI engines: Surface area is approximately 1.1 times the bore area.

5.3.4.3.3 Liner with Piston at TDC:

The area may be calculated from an estimated piston to head clearance times the circumference of the cylinder. The mean effective thickness of the piston, the liner and the fire deck of the cylinder head together with the heat capacity determine the thermal inertia of the combustion chamber walls. The conductivity is required to calculate the temperature difference between the surface facing the combustion chamber and the surface facing the coolant.

The heat capacity is the product of the density and the specific heat of the material.

For the heat transfer to the coolant (head and liner) and engine oil (piston), an average heat transfer coefficient and the temperature of the medium must be specified [14].

To build the heat transfer model, all the surface areas of piston, cylinder head and liners are defined as follows:

Heat Transfer	
Cylinder Model	Woschni 1978
Piston	
Surface Area	5809 mm^2
Wall Temperature	226.85 degC
Piston Calibration Factor	1
Cylinder Head	
Surface Area	7550 m^2

Wall Temperature	256.85 degC
Head Calibration Factor	1
Liner	
Surface Area when piston at TDC	270 mm^2
Wall Temperature when piston at TDC	161.85 degC
Wall Temperature when piston at BDC	151.85 degC
Liner Calibration Factor	1
Combustion System	Direct Ignition
In-cylinder Swirl Ratio	0

Table 5.3.13: Heat transfer model

5.3.5 Injector

An injector was defined by specifying the air fuel ratio to 13.34 and a continuous injection method is selected as the injector model.

5.3.6 Air Cleaner

The total volume of the air cleaner, the collector volumes of the collectors at inlet and outlet and the filter element's length have to be define during air cleaner modelling. The following data is considered to build an air cleaner:

Air Cleaner	
Geometrical Properties	
Total Air Cleaner Volume	8.7 litres
Inlet Collector Volume	3.0 litres
Outlet Collector Volume	4.3 litres
Length of Filter Element	300 mm
Friction Specification	Target Pressure Drop

Target Pressure Drop	
Mass Flow	0.13 kg/s
Target Pressure Drop	0.008 bar
Inlet Pressure	1 bar
Inlet Air Temperature	19.85 degC

Table 5.3.14: Air cleaner data

The filter element's length is used to model the time required by the pressure wave to travel throughout the air cleaner.

The performance of the air cleaner is measured by the reference mass flow and the target pressure drop, which is defined as the pressure difference at the inlet pipe and outlet pipe, at the reference mass flow and the inlet air conditions such as temperature and pressure.

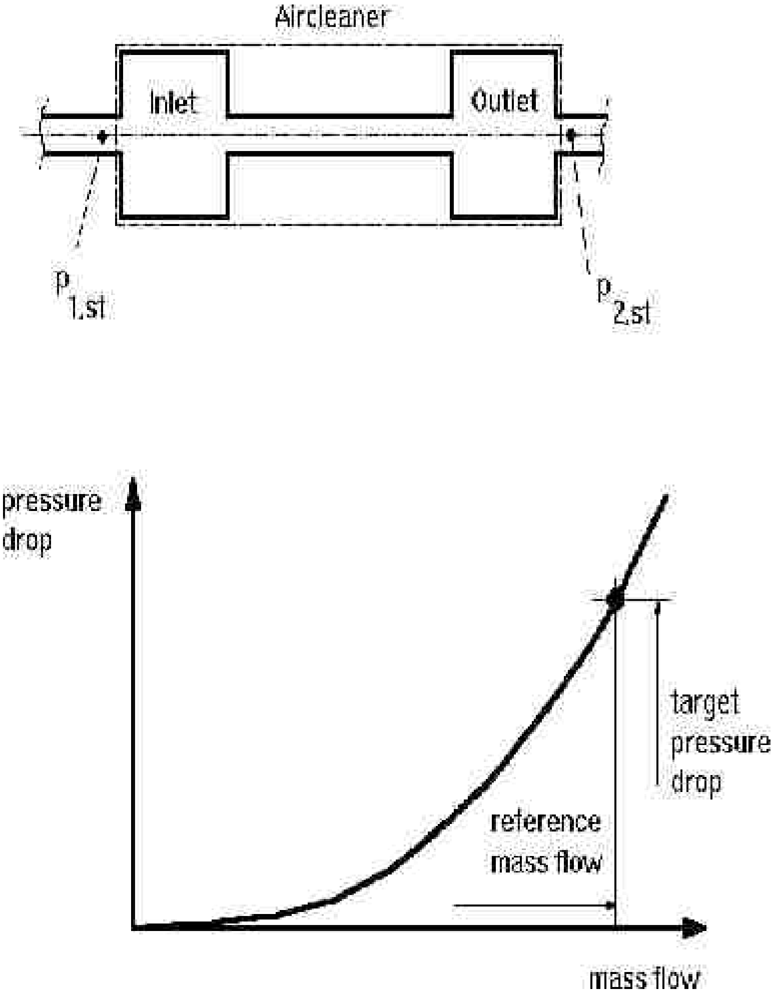

Figure 5.3.3: Steady State Air Cleaner Performance

With reference to the above information, the program adjusts the wall friction loss in the model accordingly.

5.3.7 Junctions

The refined model requires flow coefficients for each flow path in each possible flow pattern. For the three-way junction, this adds up to two times six flow coefficients. The following Figure 5.3.4 shows the qualitative trend of these flow coefficient versus the ratio of the mass flow in a single branch to the mass flow in the common branch for a joining flow pattern

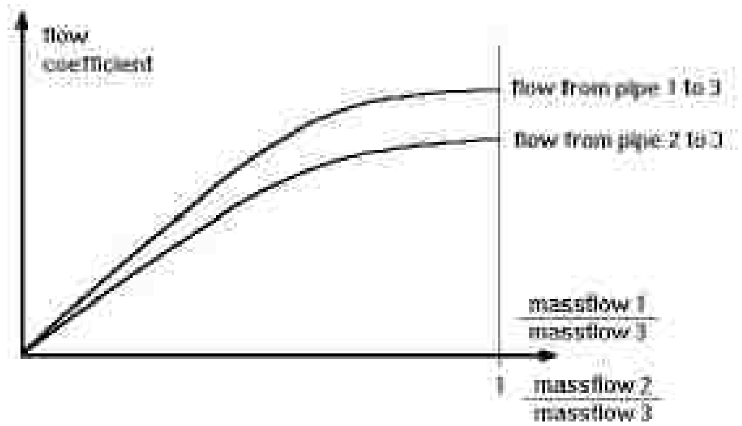

Figure 5.3.4: Flow coefficients of a junction [14]

The actual values depend on the geometry of the junction, i.e. the area ratio and the angle between the pipes. BOOST interpolates suitable flow coefficients for the considered junction from a database (RVALF.CAT) delivered with the BOOST code. The database contains the flow coefficients of six junctions, covering a wide range of area ratios and angles. The data were obtained by measurements on a steady state flow test rig. The file RVALF.CAT is a formatted ASCII file [14].

5.3.8 Exhaust Connections

The exhaust connections are important to drive away all the burnt products inside the combustion chamber, in case there are two cylinders firing subsequently in an engine, then these two exhaust runners should not be interlinked at least in the first stage of connections.

It is recommended to keep all the subsequently firing cylinders as far as possible, so that there are no back pressures generated from the collision of newly generated exhaust gas pulse with an existing exhaust gas pulse, which is already present in the exhaust runner as a result of the previous power stroke in the first cylinder, this affects the efficiency of the engine to drive away the combustion products from the chamber.

Therefore, as depicted in the Figure 5.3.5 below, the exhaust connections in an 8-cylinder engine are modelled to eliminate the back pressure.

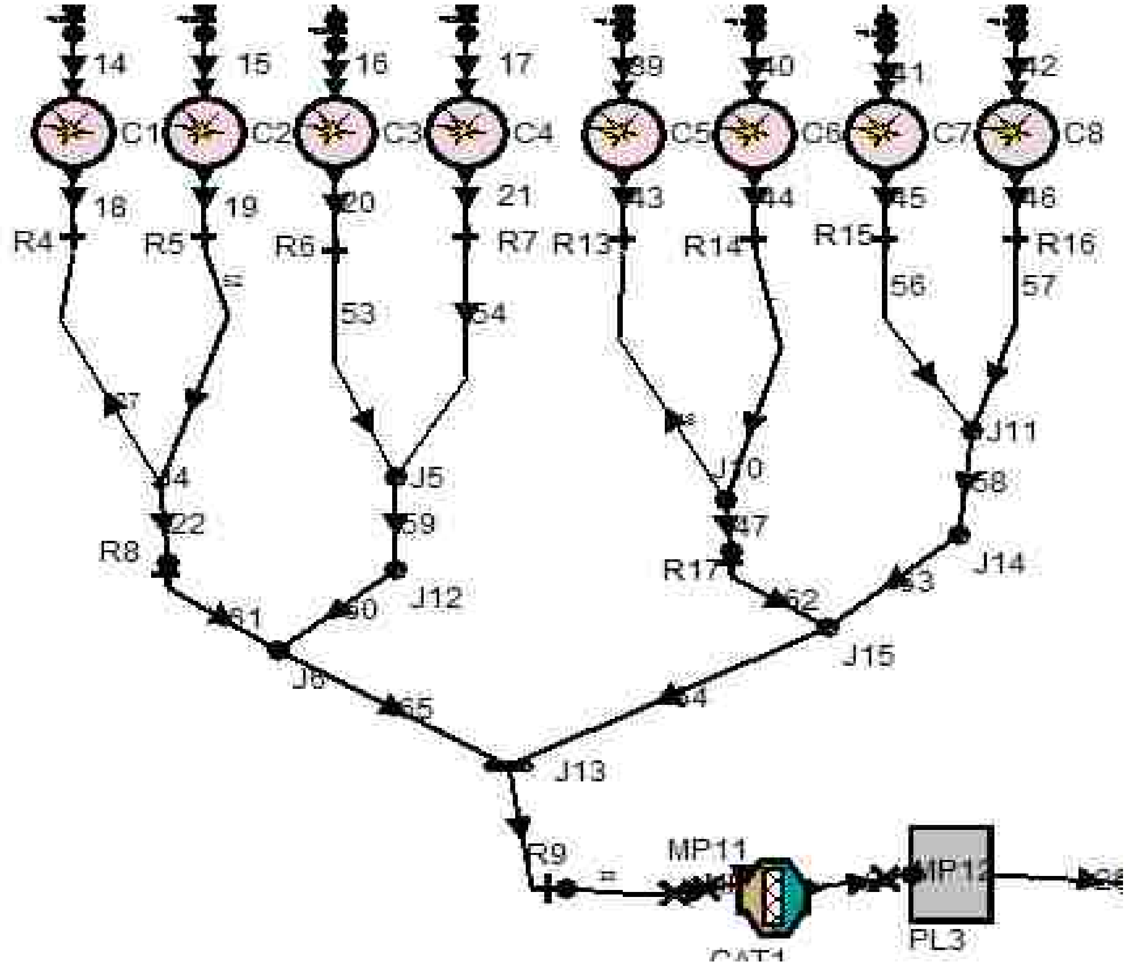

Figure 5.3.5: Exhaust Connections

The firing order for this engine is 1-6-2-5-8-3-7-4, it can be observed that in the first stage of exhaust connections, no two subsequently firing cylinders are located close to each other.

5.3.9 Optimization

The plenum dimensions play an important role in determining the volumetric efficiency of the engine. The engines are modelled to produce maximum torque at 4000 rpm, the volumetric efficiency of the engine is a mere reflection of the torque curve, so the plenum volume is optimized in a way that it produces maximum volumetric efficiency at 4000 rpm.

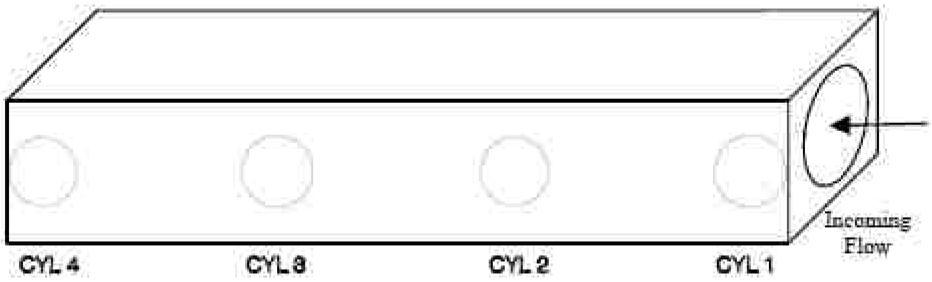

Figure 5.3.6: Plenum for a 4-cyliner engine model

As shown in the Figure 5.3.6, the plenum can be defined by volume or diameter and length, the air that leaves the plenum passes through pipes to the combustion chamber. Therefore, care must be taken to build the plenum model.

The optimiser has different functions, the user can either specify the exact value of a variable required or just say maximum value or minimum value of the selected variable to be optimised. In this case, the goal of optimization is set to maximum value, engine torque is taken as the target variable to be optimised, the plenum diameters and length are given as parameters to be varied in order to achieve the goal of maximum engine torque. Engine rpm is fixed at 4000, during this optimization process.

The limits of the parameter to be varied have to be defined. In case, the optimization result has these parameter values on the extremes of initially defined limits, a new optimization process is initiated again by, considering the result from the previous iteration, as a mean value to the new optimization process, which gets repeated until all the results do not fall on the extreme ends of the limits defined for the parameters to be varied.

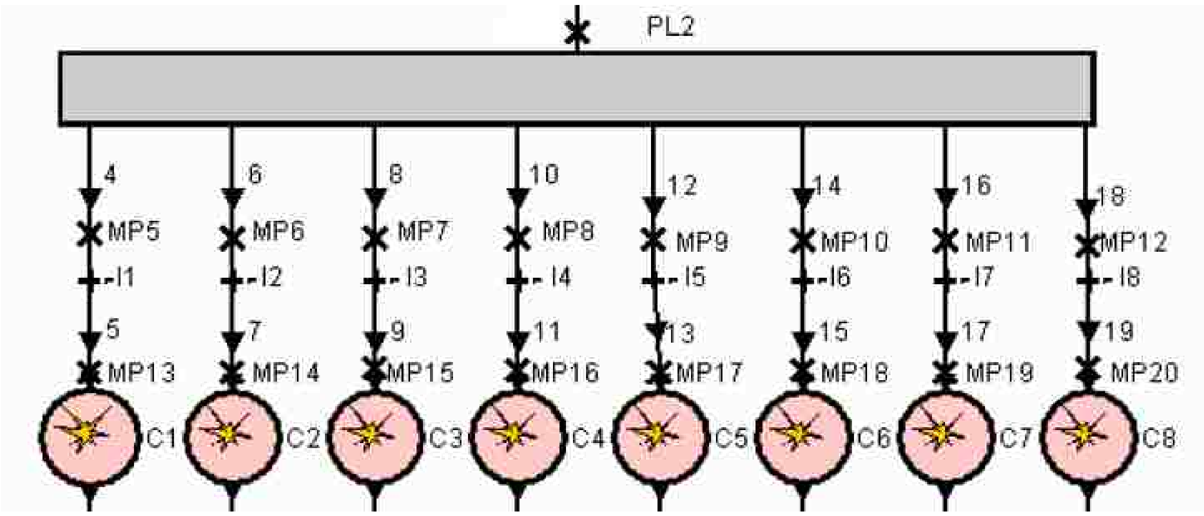

Figure 5.3.7: Plenum for an 8-cylinder engine model

5.3.10 Running the simulation

Engine speed is set as a parameter and it is considered in steps of 500 from idle speed through to maximum engine speed redline. The following results are obtained from simulation.

[redacted] Simulation data of a 4-cylinder engine

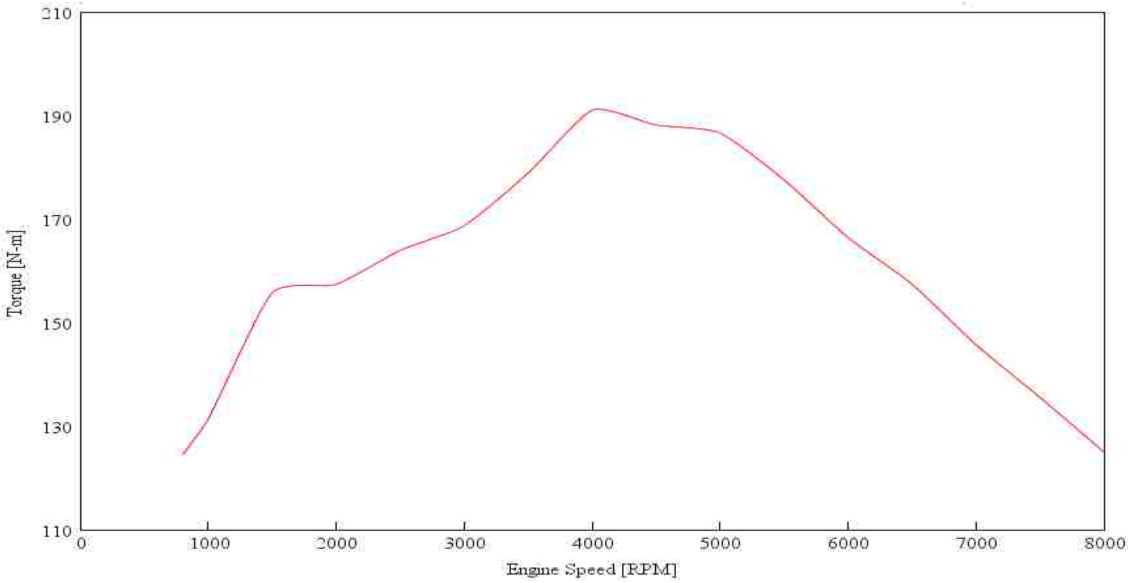

Figure 5.3.8: 4-Cylinder Engine Brake Torque

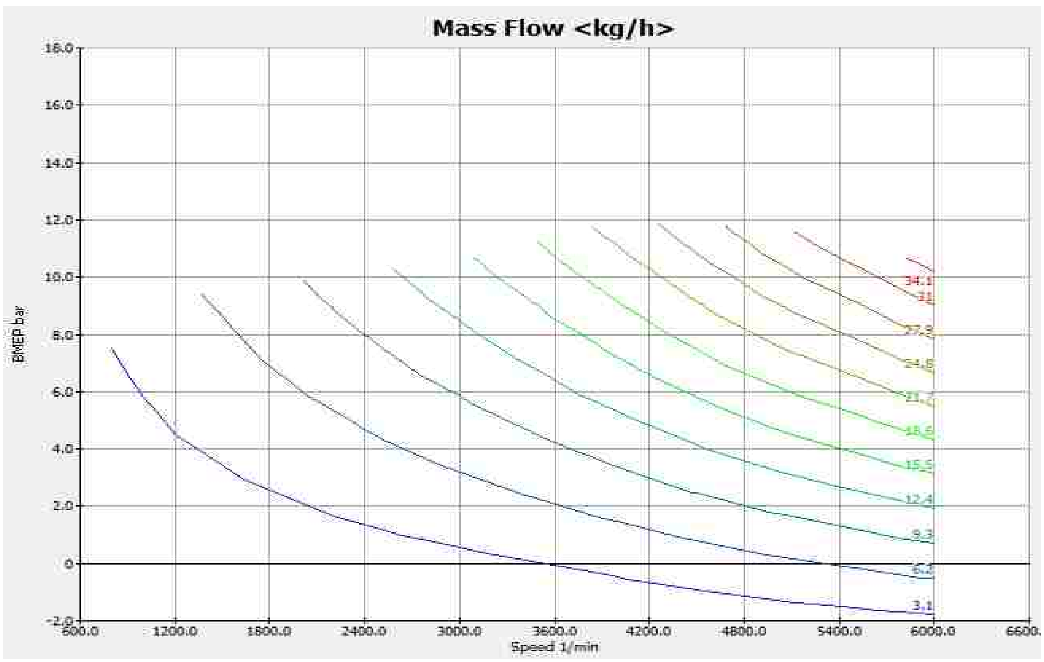

Figure 5.3.9: 4-Cylinder Engine Fuel Consumption Map

[redacted] Simulation data of a 8-cylinder engine

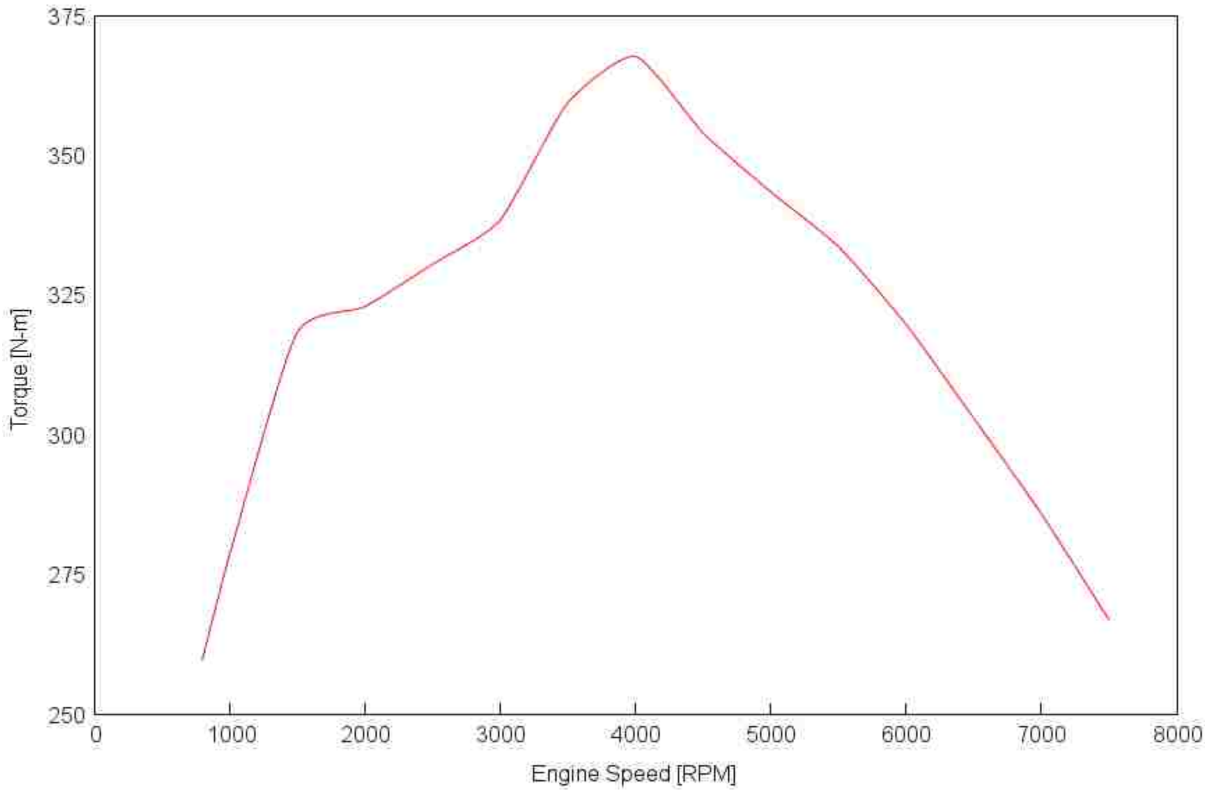

Figure 5.3.10: 8-Cylinder Engine Brake Torque

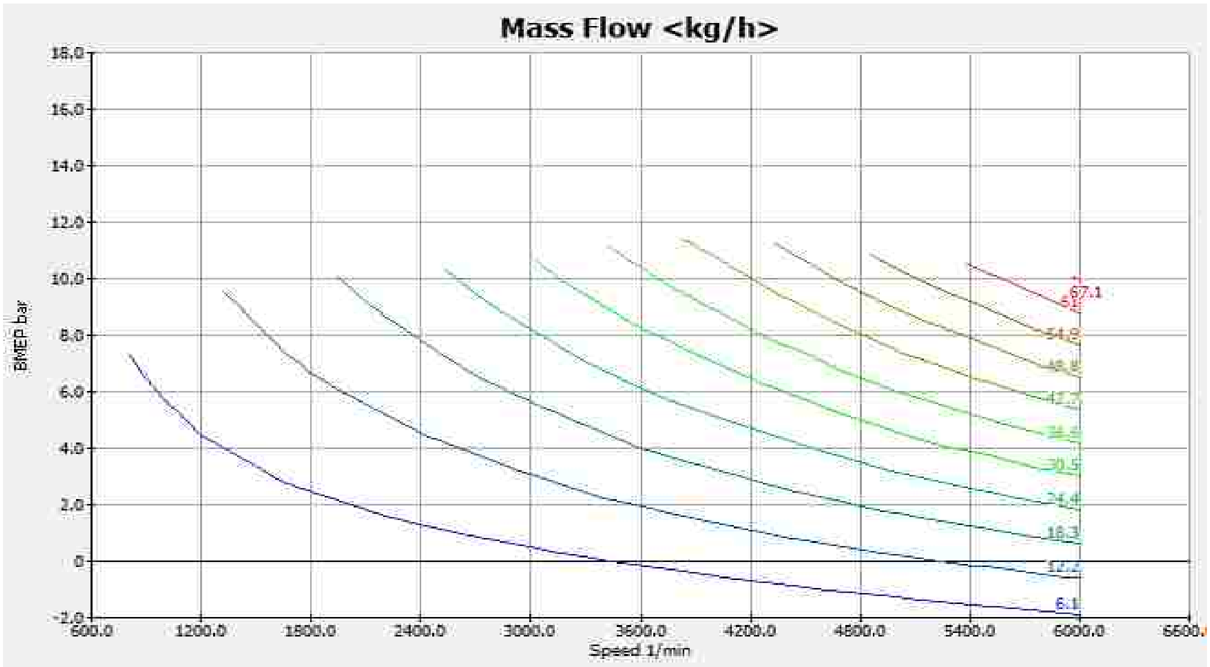

Figure 5.3.11: 8-Cylinder Engine Fuel Map

[redacted] Simulation data of CDA- engine

While simulating the cylinder deactivation engine the following steps are taken:

- The cylinders to be deactivated are identified by:
    - Understanding the firing order of the engine and making sure that the power stroke is evenly distributed, throughout the crank rotation, for each thermodynamic cycle that is every 720 degrees.
    - The firing order for the eight cylinder engine is 1-6-2-5-8-3-7-4.
    - Now subsequent firing cylinders should not be deactivated in order to evenly distribute the power stroke.
    - For the same reason, initially, cylinder 1 is considered for deactivation.
    - It is then followed by alternate firing cylinders for deactivation, which are 2-8-7.
    - So the cylinders, 1-2-7-8 are deactivated and the cylinders 3-4-5-6 keep running through this simulation.
- After identifying the cylinders to be deactivated, their corresponding valve lift is defined to be zero and the following results are obtained.

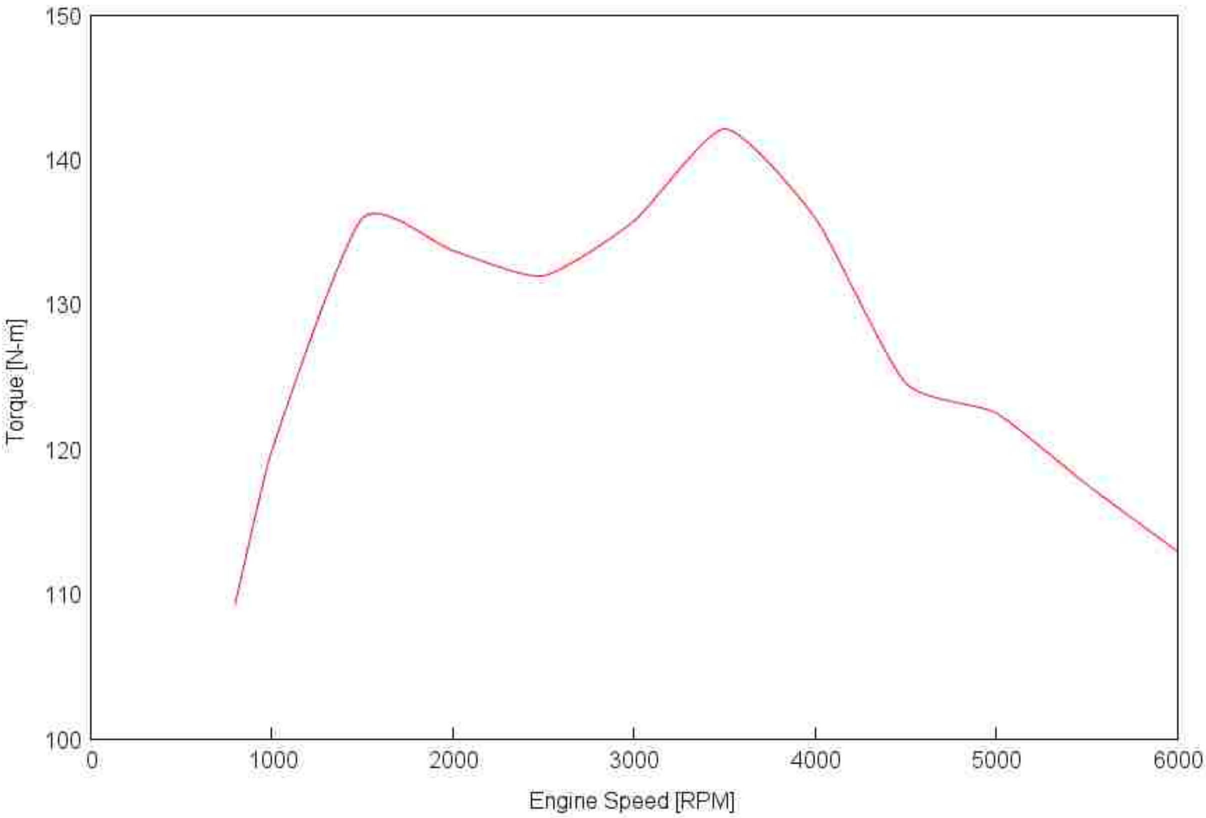

Figure 5.3.12: CDA Engine Brake Torque

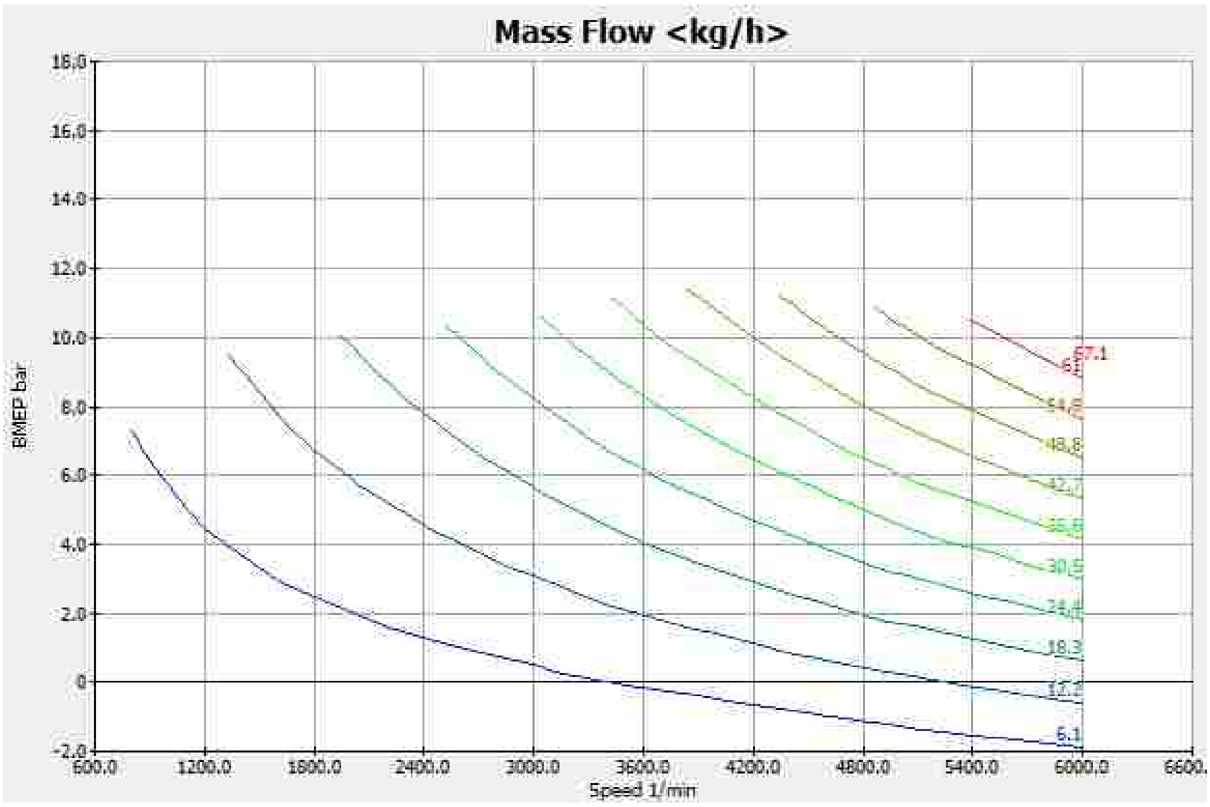

Figure 5.3.13: CDA Engine Fuel Map

6 Running Simulation Using AVL CRUISE

6.1 AVL CRUISE

AVL CRUISE is a system level simulation software used to simulate vehicle and powertrain. It supports day to day tasks in a vehicle, powertrain analysis, through the entire development phase, right from concept planning until the launch and beyond. Its application range covers conventional drivelines through to advanced hybrid powertrain systems and purely electric drivelines. It provides a streamlined workflow for various kinds of parameter optimizations and component matching which aid the user in attainable and practical solutions. It has organised interface, highly advanced data management scheme, also well equipped with system integration and data communication which have established CRUISE as a world leader and is being used by major OEM's and suppliers.

To summarise, CRUISE is used in engine development and drivetrain to calculate and optimize the following:

- Fuel Consumption
- Driving Performance
- Transmission ratios
- Braking Performance

6.2 CRUISE Workflow

The following steps are referred for the CRUISE workflow:

- Create project/version
- Create vehicle model
- Input the data in components
- Create connections: Energetic
- Create connections: Informational
- Create task folders and add tasks
- Set up calculation
- Run calculation
- View and evaluate results

6.3 Creating the Vehicle Model

The following components are placed in the work area and each of them are defined individually

- Vehicle
- Engine
- Clutch
- Gear Box
- Single Ratio Transmission
- Differential
- 4 Brakes
- 4 Wheels
- Cockpit
- Monitor

6.3.1 Defining the Input Data for each component

[redacted] Vehicle

It is the main object in the model. It contains the general data of the vehicle, such as weights and nominal dimensions. Every model requires only one vehicle component in the model. The dimensions and load state form the base for calculating dynamic wheel loads and road resistances for different road runs. Wheel loads are calculated considering motion from the effects of acceleration, aerodynamic drag and rolling resistance. A passenger car from the example model is used to build the vehicle with the following data:

Type	Manual Front Wheel Drive
Gas Tank Volume	0.075m^3
Vehicle Body Dimensions	
Distance form hitch to front axle	3300 mm
Wheel Base	2650 mm
Height of support point at bench test	100 mm

Load Dependent Characteristics [mm]					
Load State	Distance of Gravity Center	Height of Gravity Center	Height of Hitch	Tire Inflation front axle	Tire inflation pressure Rear Axle
Empty	1250	552	400	2.0	2.2
Half	1255	545	390	2.0	2.2
Full	1263	530	368	2.0	2.2
Nominal Weight					
Curb Weight	1650 Kgs				
Gross Weight	1850 Kgs				
Air Coefficient					
Frontal Area	1.88 m^2				
Drag Coefficient	0.32				

Table 6.3.1: Vehicle data

[redacted] Engine

It contains the model for combustion engine. The used has to define the characteristic curves for motoring curve, full load and fuel consumption. An engine was modelled by a structure of curves and maps as follows

Engine type	Gasoline
No of Cylinders	4/8
Number of Strokes	4
Cylinders arrangement	Inline
Charger	Without
Engine Displacement	2000/4000 cm^3
Engine Working Temperature	80 C

Idle Speed	800 rpm
Maximum Speed	6000 rpm
Inertia Moment	0.1225 kg*m^2
Response Time	0.1s
Fuel Type	
Fuel Type petrol	
Heating Value	44000.0 KJ/Kg
Specific Carbon Content	0.86

Table 6.3.2: Engine data

We also have to provide the full load characteristic, motoring curve and the fuel consumption map to completely define the engine in CRUISE.

Full Load Characteristics: is taken from the engine modelled in AVL Boost, it is the maximum torque/BMEP/power produced by the engine at various rpm steps and wide open throttle (WOT).

Motoring Curve: is used to analyse the frictional power in an engine, it is a curve plotted against cylinder pressure and crank angle when no firing occurs in an engine. The cylinder pressure built in this case is a result of the compression stroke in an engine. This data is again obtained from the engine model in AVL BOOST.

Fuel Consumption Map: is a 3 dimensional map of the fuel consumed in an engine that takes into consideration the various rpm values on x-axes, various BMEPs for each rpm value, on y-axes and the fuel consumed for each BMEP, on z-axes. This is even obtained from the engine model in AVL BOOST.

[redacted] Clutch

Idling, transition to motion and power flow interruption are made possible by the used of clutch. When the vehicle is transitioning to motion, the clutch slips to compensate for the difference in the rotational speeds of the engine and the drivetrain. The following data was used to model the clutch.

Type	Friction Clutch
Inertia Moment In	0.01 kg*m²
Inertia Moment Out	0.01 kg*m²
Model Type	Simple
Maximum transferable Torque	350 Nm

Table 6.3.3: Friction Clutch data

The pressure force for the clutch has been defined as a function of clutch release, when the clutch release is zero, it means that the pressure pads are exerting maximum force on the clutch plates, if the pressure force is one, this means that the driver has stepped on the clutch, it disengages the engine from the powertrain and the pressure force exerted by the pressure pad on the clutch plates is zero, the same can be seen in the form of numerical values in the table below.

Clutch Release (%)	Pressure force (N)
0	5000
33	2250
66	750
100	0

Table 6.3.4: Clutch release as a function of pressure force

6.3.1.3.1 Working Principle

The clutch model considers standard slip and stick friction values, that friction value used in the computation depends on whether the clutch is locked or unlocked. A set of equations run in the background to find the unknowns in the following order to obtain the final power loss from the clutch:

- A fictive radius called mean effective radius which determines the radius in which the frictional force acts is computed initially

- The actual frictional coefficient is then calculated from the relative angular velocity and a function of stick and slip frictional coefficients
- Later, the transmitted torque in a clutch is calculated by interpolating the actual clamping force from the actual clutch release map and by multiplying the obtained value with the above calculated values
- Finally, the power loss in a clutch is evaluated from the difference in powers at the inlet and output of the clutch

[redacted] Gear Box

A gear box enables the engine to run as close as possible to its best performance rpm value, it is equipped with underdrive and overdrive gear ratios which alter the toque and speed of the prime mover output shaft to fulfil the acceleration demands by the drive. The manual gear box is modelled with the following data.

Type		5 Speed Gear Box			
Gear Ratio Table					
Gear	Transmission Ratio	Inertia Moment In (Kg*m^2)	Inertia Moment Out (Kg*m^2)	Number of Teeth Input	Number of Teeth Output
0	1	0.0015	0.005	10	10
1	3.62	0.0015	0.005	50	181
2	2.22	0.0015	0.005	50	111
3	1.51	0.0015	0.005	100	151
4	1.08	0.0015	0.005	25	27
5	0.85	0.0015	0.005	20	17

Table 6.3.5: Gear box data

[redacted] Single Ratio transmission

This is the final drive unit ratio, it is considered to be separate from the main gear train, usually within the differential and on the other side of the differential. The following details are used to model the final drive.

Type	Final Drive
Location	Front
Transmission Ratio	3.0
Inertia Moment In	0.008 kg*m²
Inertia Moment Out	0.015 kg*m²
Efficiency	0.97

Table 6.3.6: Final drive data

[redacted] Wheels

The vehicle is linked to the road through the wheels. This component considers the influence of many variables that effect the rolling state. The wheel load factor, slip factor, friction coefficient aid in computing the longitudinal circumferential tire force. The wheel load, dynamic rolling radius and the rolling drag coefficient form the basis for calculating the moment of rolling drag.

MICHELIN model is used for computing the detailed rolling resistance. The data used for all the four wheels is the same, the following information is used to define the wheel.

Inertia Moment	0.51 kg*m²
Wheel Slip	
Friction Coefficient of Tire	0.95
Reference Wheel Load	2500 N
Wheel Load Correction Coefficient	0.02
Rolling Radius	
Static Rolling Radius	305 mm
Circumference	1916.37 mm
Dynamic Rolling Radius	312 mm
Circumference	1960.35 mm

Table 6.3.7: Wheel data

[redacted] Brake

The braking torque is calculated using the braking dimensions and input brake pressure. The data for the front two brakes and also the data for the rear two brakes is the same. The following data is used to model the brakes. Cockpit component delivers the required braking pressure.

When the vehicle is in stand still mode, the calculation time is reduced by reducing the degrees of freedom. In case a small velocity threshold is reached, the equation system is reverted back to an unreduced basic one and checks if the instantaneous compulsive force is smaller than the braking torque, then the vehicle movement is supressed as long as the above condition holds good.

	Front Disc Brake	Rear Disc Brake
	Front right / Front left	Rear Right / Rear left
Brake Piston Surface	1800 mm²	1500 mm²
Friction Coefficient	0.25	0.25
Specific Brake Factor	1	1
Effective Friction Radius	130	110
Efficiency	0.99	0.99
Inertial Moment	0.02 kg*m²	0.015 kg*m²

Table 6.3.8: Brake data

[redacted] Differential

When the car takes a turn, the inner wheels are at different turning radius than the outer wheels, this makes the wheels to rotate at different speeds and as a result, slip and wear out over time, in order to eliminate this effect while turning a differential is being used in car. The main functions of a differential are to transmit the power to the wheels, act as a final gear reduction in a vehicle and accommodate the differential speeds of the wheels while turning. The following data was used to model the differential.

Type	Differential
Location	Front end
Differential Lock	Unlocked
Torque Split Factor	1
Inertia Moment In	0.015 kg*m²
Inertia Moment Out	0.015 kg*m²

Table 6.3.9: Differential data

[redacted] Cockpit

This component links the drive to the vehicle. The connections in this component are made through the Data Bus. Linking is done by communicating the driver pedal positions to other components and at the same time, providing the driver, information regarding the vehicle such as vehicle acceleration and velocity. The pedal characteristic such as clutch pedal characteristic is related to the pedal positions to deliver required information to corresponding indicators such as clutch pedal release.

Component	Cockpit
Shift Mode	Manual
Number of Gears	Forward: 5 Reverse: 1
Maximum Brake Force	100 N
Number of Retarder Steps	0
Brake Light Switch Threshold	1.0 %

Table 6.3.10: Cockpit data

The acceleration pedal characteristic and brake pedal characteristic are defined linearly proportional as shown in the graph below, for acceleration pedal characteristic graph is defined as acceleration pedal travel (%) vs load signal (%) on x and y axes respectively. The brake pedal characteristic map has specific brake pedal force (%) on x axes and brake pressure (bar) y axes.

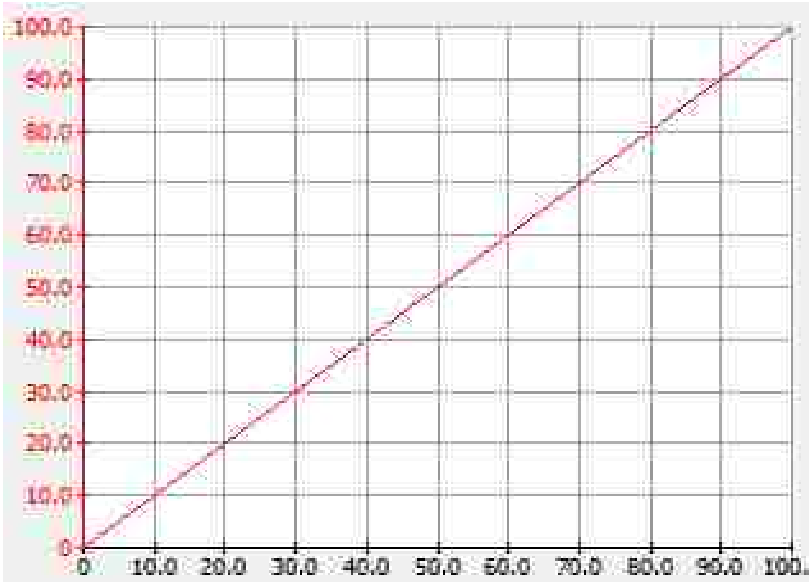

Figure 6.3.1: Acceleration pedal travel (%) vs load signal (%)

6.3.2 Informational Connections

In the vehicle system, the Data bus makes all the necessary informational connections for connecting different models with information flow. Given below is an example how different components make informational connection via Data Bus.

Component Requirement	Input Information From	Component Delivering	Output Information
Brake	Brake Pressure	Cockpit	Brake Pressure
Clutch-Friction Clutch	Desired Clutch Release	Cockpit	Desired Clutch Release
	Gear Indicator	Gear Box	Current Gear
	Operational Control	Engine	Operational Control
Cockpit	Speed	Engine	Engine Speed
Engine	Load Signal	Cockpit	Load Signal

	Start Switch	Cockpit	Start Switch
Gear Box	Desired Gear	Cockpit	Desired Gear

Table 6.3.11: Data bus connections

6.3.3 Vehicle Models Created

To compare the proposed Separate Engine Strategy with Cylinder Deactivation strategy and the Conventional Engine Model, the following models have been created in CRUISE.

Model Created	Key Component	Components added
Conventional 8 Cylinder Engine	8 Cylinder Engine, 4000 cc	None
CDA Engine	8 Cylinder Conventional Engine, 4000 cc	Cylinder Cut-Out Engine, 4000 cc
Separate Engine Strategy	4 Cylinder Engine, 2000 cc	Friction Clutch

Table 6.3.12: Vehicle models created

The modifications done to the conventional model are briefly discussed in the control strategy section below.

Given below is, an 8-cylinder conventional engine model, having a cubic capacity of 4000 cc, the layout of the vehicle after creating and connecting the required components in CRUISE.

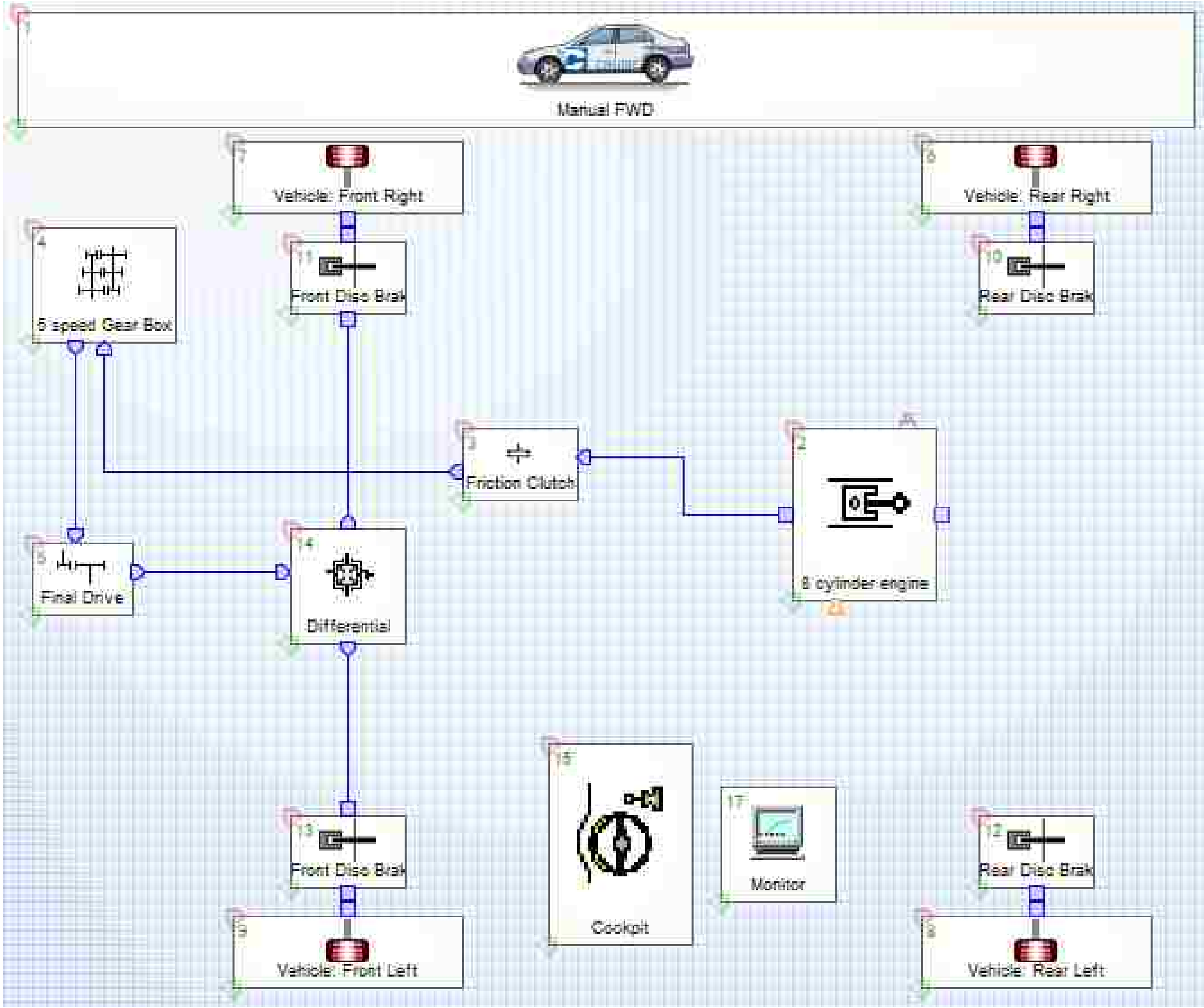

Figure 6.3.2: 8-Cylinder conventional engine model

6.4 Control Strategy

A control algorithm is devised to deactivate and reactivate the cylinders in Cylinder Deactivation Engine (CDA Engine) and in case of separate engines, algorithm is used to turn on and off the second engine.

6.4.1 Cylinder Deactivation Engine (CDA Engine)

CRUISE uses Cylinder Cut-out Engine (CCE) to accommodate the cylinder deactivation technology. This CCE represents a cylinder cut-out, it is directly connected to the main engine, through the use of activation control, a decision can be made on which engine is activated. The signal for the cut-out engine is 2 and for the main engine is 1. When the CCE engine is activated, some data such as full load characteristic and engine maps are taken from CCE engine and the remaining data such as motoring curve are all taken for main engine.

Using AVL Boost, data required by the CCE Engine is generated by defining the valve lift to be zero for 4 alternate cylinders in an inline 8 cylinder engine and the corresponding full load characteristics and fuel consumption maps are generated through simulation. The weight of the additional components added to the system in order to accommodate the CDA technology is very small when compared with the weight of the vehicle and hence, can be neglected.

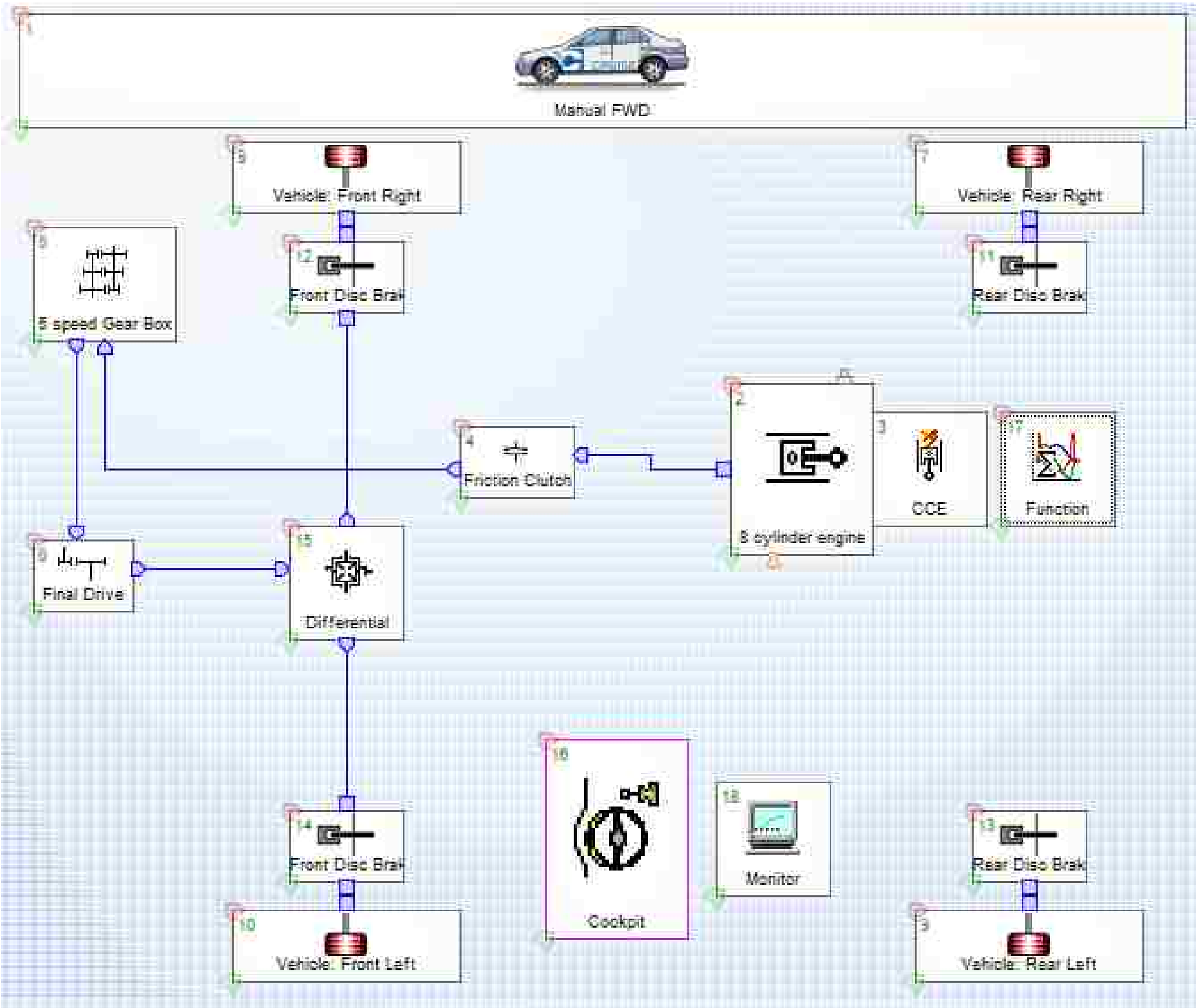

Figure 6.4.1: 8-cylinder CDA engine model

The function component keeps checking the boundary conditions for the operation of each engine, then communicates with the engine and cockpit via the corresponding results obtained from control strategy.

[redacted] Control Algorithm Code

A function component is used to calculate the user defined functions. The program can be written in C-Programming style. Up to 99 function inputs and outputs can be defined in Function component. The function component only interprets the C code during simulation and does not compile it. After computing the function value, it can be used as an output value via the Data Bus. The logic for cylinder deactivation in this project is as follows:

The parameters that establish boundaries for the operation of CDA System are as follows:

- Vehicle Speed > 30 Km/hr
- Engine Speed > 1000 rpm and < 4000 rpm
- Acceleration < 0
- Gear State >= 3

Cylinder deactivation is usually done during decelerations because during accelerations it is very likely that at some point the active cylinders will not be able to provide the required power demanded by the driver and it is also difficult to achieve a smooth transition phase for reactivation of the cylinders while the vehicle is accelerating. NVH constraints prevent the vehicle from deactivating the cylinders during idling conditions and also in the first and second gear.

Once all the above conditions are satisfied, the Function component sends an input signal "2" to the engine. This activates the CCE Engine, the full load characteristics and engine maps from CCE are considered for computing the fuel economy during the drive cycle simulation.

In order to prevent from frequent deactivation and reactivation, the fluctuation in the vehicle velocity during phase changes is considered and a hysteresis has been accordingly added to the code by slightly increasing the acceleration limits for reactivation, this takes care of any acceleration fluctuation during the event of deactivation.

6.4.2 Separate Engine Strategy

In this case, two, 4 cylinder engines having a cubic capacity of 2000cc have been added to the powertrain. A friction clutch connects these two engines in series. The first engine has another flywheel on the other end where the second engine is connected. This modification to a standard engine is incorporated in the model by, making an assumption that 50% of the moment of inertia in an engine comes from the flywheel, and adding this additional moment incurred from the flywheel located at the rear end of the engine to the moment of a standard engine model. Also, adding the weight of additional components such as flywheel and clutch to the overall weight of the vehicle, accommodates all the modifications done to the standard vehicle model.

The moment of the flywheel and clutch assembly can also be obtained from any OEM's sales brochure, for this project, the moments and weights of a flywheel manufactured by Tilton Engineering are taken into consideration, it has a maximum torque capacity of 650 Nm, the weight and moment are as follows [15]:

- Weight: 5.5 Kgs
- MOI: 0.03 Kg*m^2

The control strategy used in this case is the same as the code used in Cylinder Deactivation Engine, the only difference being the signal output from Function component, communicated upon satisfying the operational boundary parameters.

In this case, the engines are connected by the use of a clutch in series. So, a signal to turn off the second engine, is always accompanied by a signal to disengage the clutch. The RPM input to the Function component is provided by the Engine 1 and upon checking for the conditions of separate engine operation, the Function component sends signals to the 2nd engine to shut down or reactivate, and to engage or disengage the friction clutch present in between the engines.

The model for Separate Engine Strategy is shown in the following Figure 6.4.2:

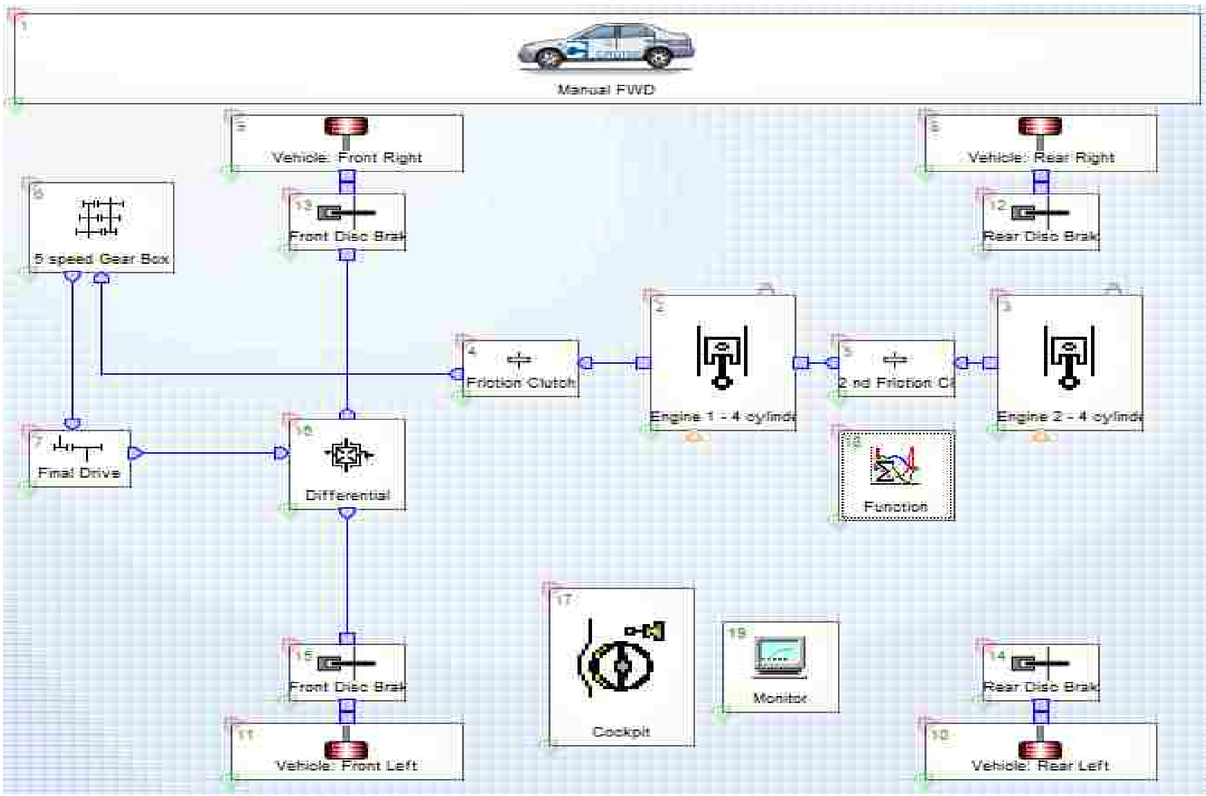

Figure 6.4.2: Separate Engine Strategy

6.5 Adding Tasks and Solvers

After the model for the vehicle is well defined, a task is assigned to the model, Cycle Run task is used for calculating the fuel consumption.

6.5.1 Cycle Run

It is a computational task for analysing the fuel consumption of the model. In this task, the behaviour of the model is observed when it follows a predefined velocity profile.

[redacted] Velocity Profile/Drive Cycle

The velocity profile simulates the real life driving conditions in a vehicle. It is defined as a profile that is entered as a function of time or distance, in other words, the velocity profile represents a set of vehicle points versus time.

The driving cycles can be classified into two kinds, firstly, modal cycles, these are a compilation of linear acceleration and constant velocity periods, hence, cannot represent a real life driver behaviour, whereas, the transient cycles accommodate many velocity variations, and represent a typical on-road real life driving behaviour. European standard NEDC or Japanese drive cycles are typical examples of modal cycles and American driving cycles such as FTP-75 and Highway Fuel Economy Test Cycle (HWFET) are examples of transient cycles.

The driving profiles considered in this project are city (FTP-75) and highway (HWFET) driving conditions defined by United States Environmental Protection Agency (EPA).

6.5.1.1.1 Federal Test Procedure cycle (FTP-75)

This has been developed by Environment Protection Agency to represent a typical commuting cycle which includes parts of urban driving conditions with frequent stops and a portion of highway driving.

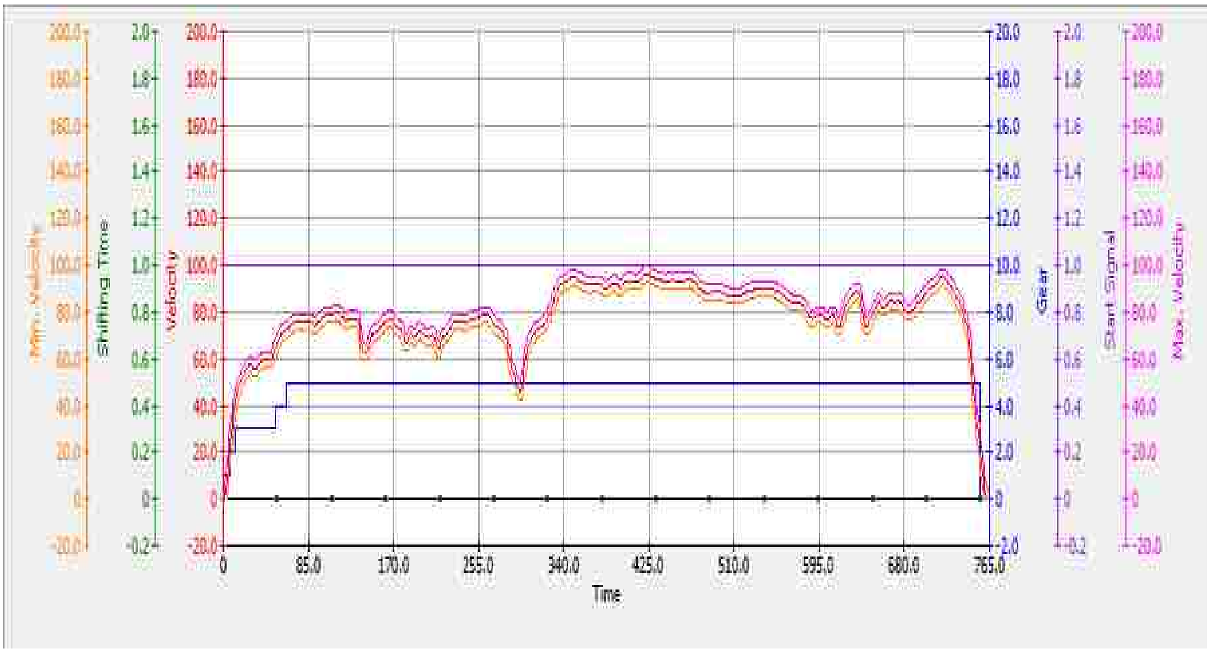

Figure 6.5.1: Federal Test Procedure cycle velocity profile (FTP-75)

Driving Cycle	Distance	Duration	Average Velocity
FTP-75	17.77 km	1874 s	34.1 km/hr

Table 6.5.1: Federal Test Procedure cycle data (FTP-75) [16].

6.5.1.1.2 Highway Fuel Economy Test Cycle (HWFET)

This is used to compute the fuel economy by simulating highway driving conditions.

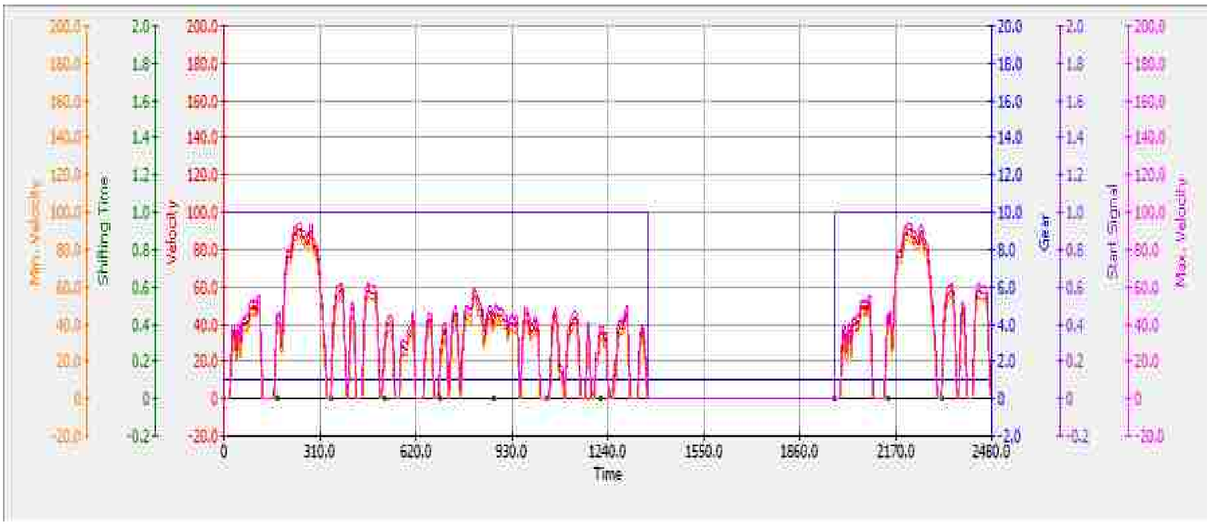

Figure 6.5.2: Highway Fuel Economy Test Cycle velocity profile (HWFET)

Driving Cycle	Distance	Duration	Average Velocity
HWFET	16.45 km	765 s	77.7 Km/hr

Table 6.5.2: Highway Fuel Economy Test Cycle data (HWFET) [16].

6.6 Choosing the Solver

The solver is component in a software package that determines the time, required for next simulation step, also applies numerical methods to solve a set a differential equations which represent the model.

Solvers can be broadly classified into two kinds, fixed step solvers and variable step solvers. Both the solvers calculate the next simulation time by adding up the current simulation time and a quantity called as step size. The former has the step size constant throughout the entire simulation and leads to large simulation times, whereas, in the latter, the step size depends on the model dynamics and varies from step to step. The variable step solver shortens the simulation time by altering the step size and at the same time complying with the error tolerance specified by the user. Controlling the error by varying the step size can stabilize the integration, hence it is important to estimate the error to gain some confidence in the simulation. Considering all the reasons stated above, a variable Step Bulirsch-Stoer Solver with an error estimator has been used in this project.

In AVL CRUISE, modelling a vehicle by modules with various connections leads to a system of non-linear ordinary differential equations which are of the form below, and are to be solved.

$$\dot{y} = f(t, y) \tag{1}$$

$$y(0) = y_0 \tag{2}$$

"Hidden in the system are possible changes of the number of unknowns, gear changes, opening and closing of connections. That is why the system function f may be very unsmooth" [13].

[redacted] Discretization

"The Bulirsch-Stoer approach discretizes the Ordinary Differential Equation (ODE) System with an implicit midpoint rule:

$$y_{n+1} - y_{n-1} = 2hf\left(\frac{y_{n+1} + y_{n-1}}{2}\right) \tag{3}$$

With the equidistant time step size h for an integration step with local grids.

By linearizing this equation the result is the semi-implicit midpoint rule" [13].

$$\left[1 - h\frac{\partial f}{\partial y}\right] \cdot y_{n+1} = \left[1 + h\frac{\partial f}{\partial y}\right] \cdot y_{n-1} + 2h\left[f(y_n) - \frac{\partial f}{\partial y} \cdot y_n\right] \tag{4}$$

[redacted] Smoothness

"For calculating a time step, the Bulirsch-Stoer approach performs the time integration step with different equidistant local steps and uses extrapolation for extrapolating the step size to zero. The extrapolation rests on smoothness assumptions for the function f and the solution vector y.

In contrast to this the integral formulation and the implicit discretization is applicable for functions and solutions with less smoothness. The nonlinear equation is solved by reformulating it to an optimization problem in which regularization by line search and trust region are applied to detect and handle appropriately critical situations" [13].

[redacted] Step Size Control

"The Bulirsch-Stoer algorithm uses information from the extrapolation for the assumption of the next step size length. This is an a posteriori approach. If the function f changes rapidly and especially if switches occur, this approach tends to be very small step sizes" [13].

[redacted] Solution of Equations on Local Level

"As shown the Bulirsch-Stoer approach leads to the semi-implicit midpoint rule. For a fixed step size h the matrix on the left hand side $\left[1 - h\frac{\partial f}{\partial y}\right]$ *remains the same so that one LU-decomposition is performed and then only forward-backward substitutions are needed. Therefore one Jacobian* $\frac{\partial f}{\partial y}$ *has to be calculated, then for the extrapolation a number of LU-decomposition is performed and afterwards the iteration is performed. But especially with a fine local time step, many function evaluations of f are needed.*

Now for the calculation of each search direction we have to solve one linear system, which is more time consuming than just a forward-backward substitution, but we calculate the time step with fewer function evaluation f. The size of the matrix is comparable small and the computing time for one evaluation of f is comparable large, so that this approach is considerable faster and still handles the instabilities more accurately" [13].

7 Results and Discussion

7.1 Analysing the part load performance

It is observed that running the engine on a separate cylinder mode is more beneficial than cylinder deactivation mode.

To compare these two strategies during part load conditions, for the same engine speed (rpm) and a specific power output, the attributes of different components are compared in the Table 7.1.1.

When the corresponding fuel saving strategy is active, a few attributes are taken at the active cylinder while others at the engine and are compared as follows:

For a Brake Power of 20.8 kW and Engine Speed 2500 rpm			
	Conventional 8-Cylinder Engine	***Cylinder Deactivation Engine***	***Separate Engine***
Engine Displacement	4000 cc	4000 cc	2000 cc
No of Cylinders Active/Deactivated	All active	4 active/ 4 deactivated	4 active/ 4 deactivated
Air Flow @cylinder	187.7 mg/cycle	364.8 mg/cycle	282.6 mg/cycle
Fuel Flow rate @cylinder	14.6 mg/cycle	28.6 mg/cycle	22.67 mg/cycle
Indicated Specific Fuel Consumption (ISFC) @cylinder	280.22 g/kWh	247.59 g/kWh	264.16 g/kWh
Volumetric Efficiency @cylinder	32.18%	62.54%	49%
Pumping Mean Effective Pressure @cylinder	0.61bar	0.27 bar	0.42 bar

Indicated Mean Effective Pressure @cylinder	4.36 bar	8.59 bar	6.6 bar
Indicated Efficiency @cylinder	29.40%	33.30%	31.18%
Burn Residual % @cylinder	11.03%	5.32%	8.08%
Throttle angle	7.3	8	10.5
Volumetric Efficiency @Engine	30.95%	29.52%	48.14%
Torque @Engine	79.55 Nm	79.55 Nm	79.55 Nm
Brake Mean Effective Pressure @Engine	2.5 bar	2.5 bar	5 bar
Brake Specific Fuel Consumption @Engine	420 g/kWh	405.4 g/kWh	326.7 g/kWh

Table 7.1.1: Comparing the engine performances for the same power and rpm

The following can be deduced from analysing the above data:

- All the data is taken when the corresponding fuel saving strategy in each engine is active.
- For the cylinder deactivation engine, when only 4-cylinders are active, to produce the same power as the remaining ones, it has to burn more fuel, in order to compensate for two things, firstly, no combustion taking place in the deactivated engines and secondly, the frictional losses incurred from the deactivated cylinders.
- For the same reason, it has to produce more power that eventually suffers losses to stay on par with separate engine strategy, therefore, it burns more fuel and this is reflected in the above data by a comparative increased flow rates such as air flow and fuel flow rate.
- The increased flow rate results in an increased volumetric efficiency of the cylinder deactivation engine and this in turn leads to a comparatively higher indicated mean effective pressure and efficiency.
- When running at same rpm, the volumetric efficiency is directly related to the throttle valve opening, hence, a comparatively higher volumetric efficiency is observed in the separate engine technology.

- Finally, to produce the same amount of power, as the fuel consumed is the lowest for the separate engine strategy, this results in lower brake mean effective pressure for this strategy.

7.2 Simulating through the drive cycles

7.2.1 Trial run

Before running the final simulation, a trial run is performed to check for the authenticity of the engine models built. In this trail run, the separate engine strategy model is made to run through the drive cycles without deactivating any engine. This means that the two 4 litre engines keep running throughout the drive cycle and the clutch connecting the engines is also engaged throughout the drive cycle. The results are shown in the table below:

	8-Cylinder Conventional Engine (MPG)	Separate Engine Strategy without deactivating the second engine (MPG)
City, FTP-75	20.04	19.64
Highway, HWFET	27.95	27.49

Table 7.2.1: Trial run

It is observed that the separate engine strategy has slightly lesser mileage, this reduction can be attributed towards moments and weight of the components added to the vehicle by the use of flywheel and clutch, in order to accommodate the separate engine strategy. Thus, it can be deduced that the data used is reliable and two similar entities are being compared in the next phase of simulation.

7.2.2 Final Simulation

Given in the Table 7.2.2 below, is the mileage obtained from simulating the models through city and highway drive cycles.

	8-Cylinder Conventional Engine (MPG)	Cylinder Deactivation Engine (MPG)	Separate Engine Strategy (MPG)
City, FTP-75	20.04	21.07	21.34
Highway, HWFET	27.95	30.40	31.08

Table 7.2.2: Drive cycle simulation results

Given in the Table 7.2.3 below, is the improvement in fuel consumption over conventional 8-cylinder engine, expressed in percentages:

	8-Cylinder Cylinder Deactivation Engine	Separate Engine Strategy
City, FTP-75	5.1%	6.5%
Highway, HWFET	8.8%	11.2%

Table 7.2.3: Mileage improvement over conventional engine

Given in the Table 7.2.4 below, is the comparative benefit obtained from use of separate engine strategy over cylinder deactivation technology, expressed in percentages:

	Separate Engine Strategy
City, FTP-75	1.2%
Highway, HWFET	3.5%

Table 7.2.4: mileage improvement over cylinder deactivation

In order to analyse brake specific fuel consumption over various drive cycles, the corresponding driving time distribution maps for all the engines are considered in the next section.

The maps represent the driving time as a collection of points through the entire cycle run simulation. The graph contains torque produced on y-axes, rpm on x-axes, and the brake specific fuel consumption can be seen as contour circles in the plot area.

[redacted] Conventional 8-Cylinder Engine

Highway drive cycle

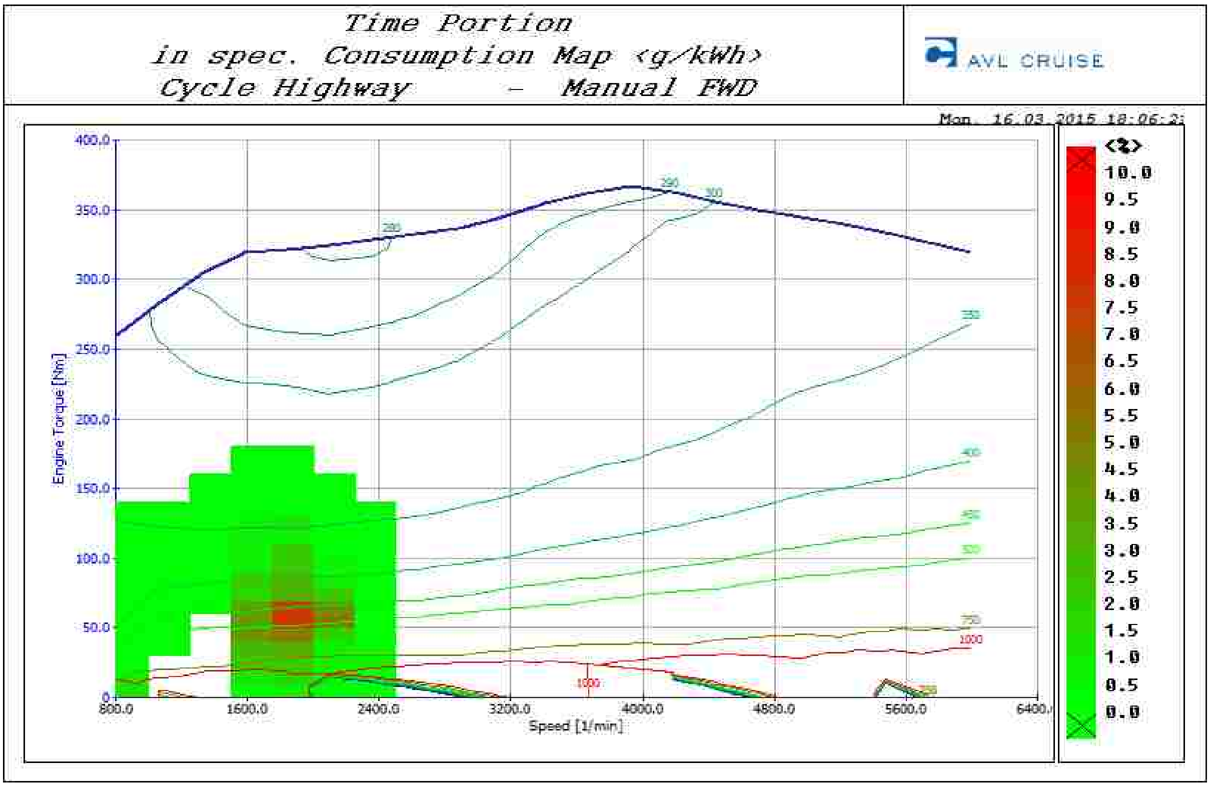

Figure 7.2.1: Driving Time Distribution for conventional 8-Cylinder Engine in highway cycle

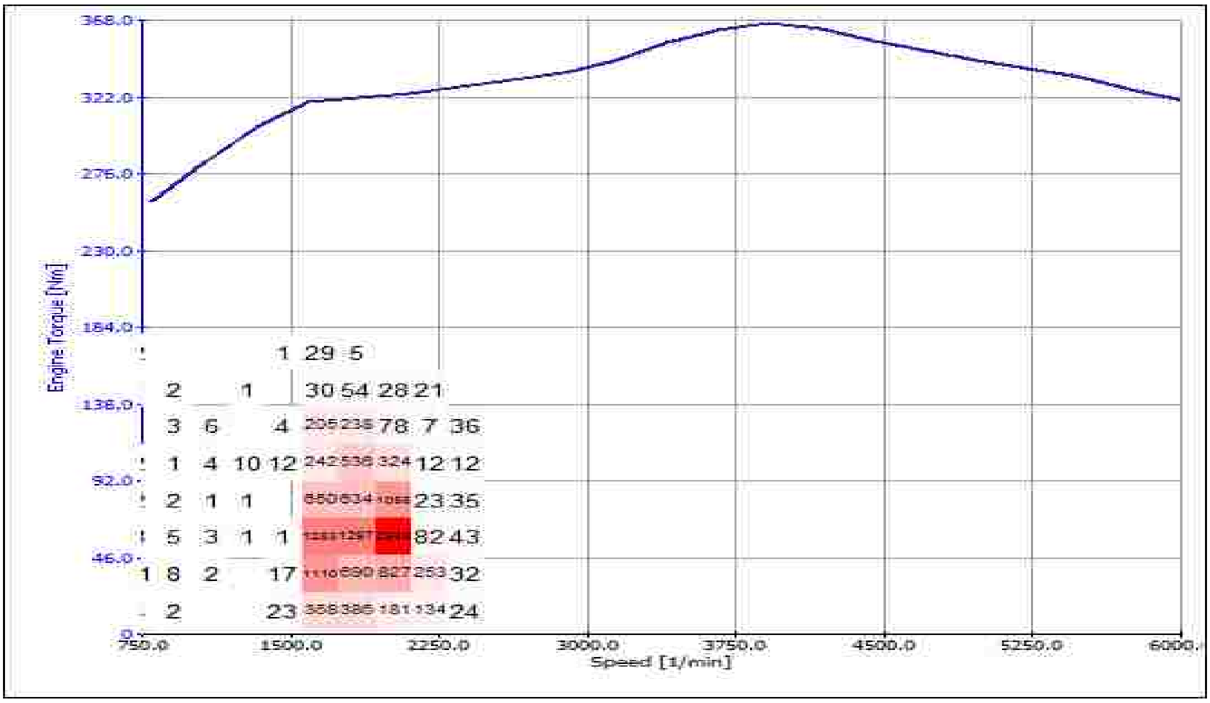

Figure 7.2.2: Driving Time Abundance for conventional 8-Cylinder Engine in highway cycle

[redacted] Cylinder Deactivation Engine

Highway drive cycle

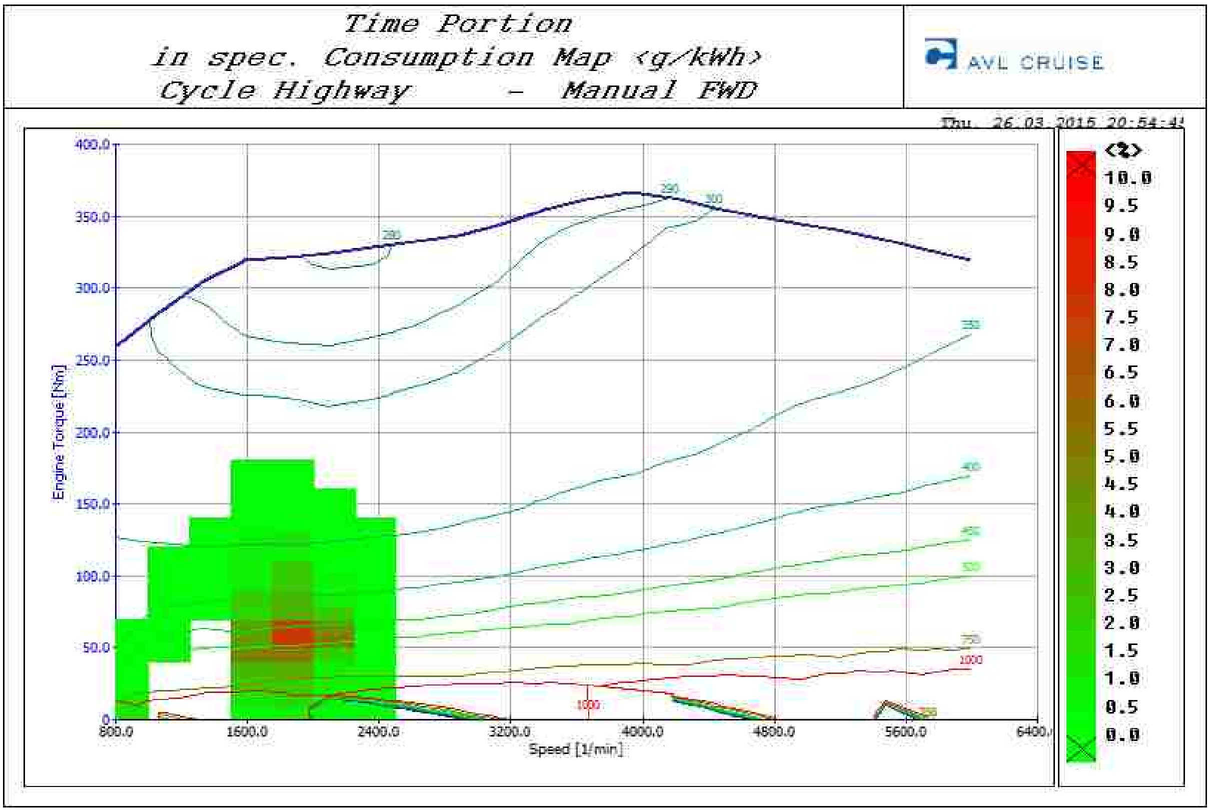

Figure 7.2.3: Driving time distribution for Cylinder Deactivation Engine in highway cycle

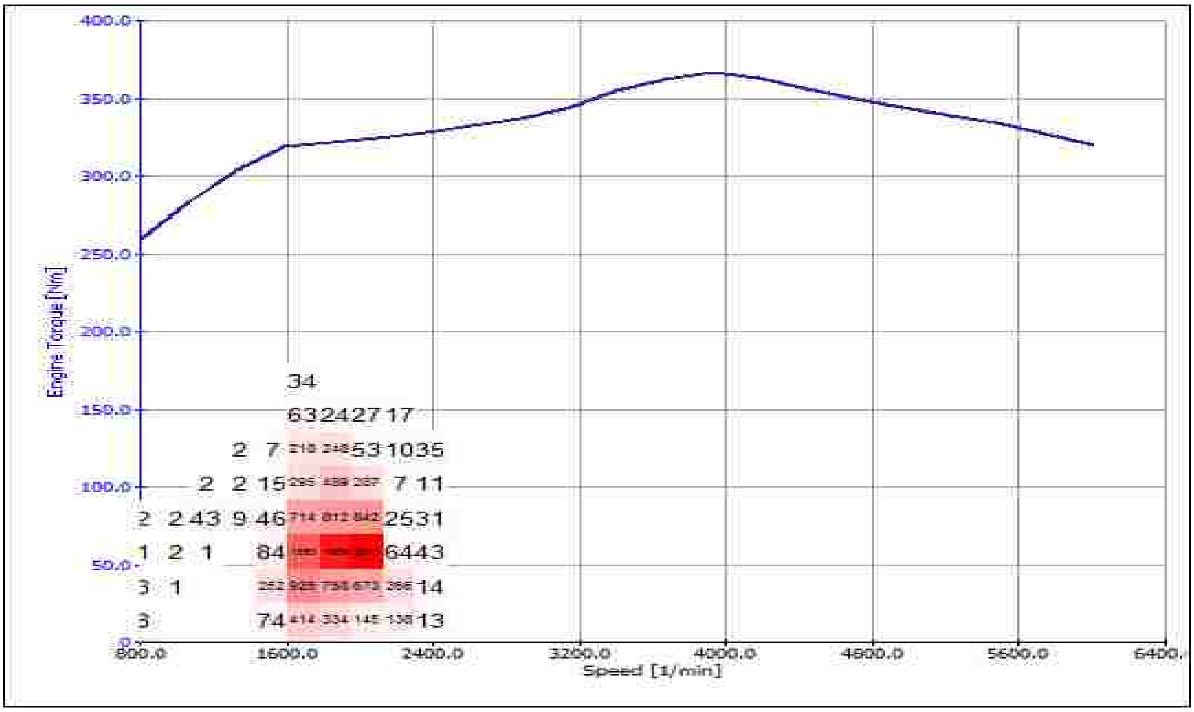

Figure 7.2.4: Driving time abundance for Cylinder Deactivation Engine in highway cycle

[redacted] Separate Engine Strategy

Highway drive cycle

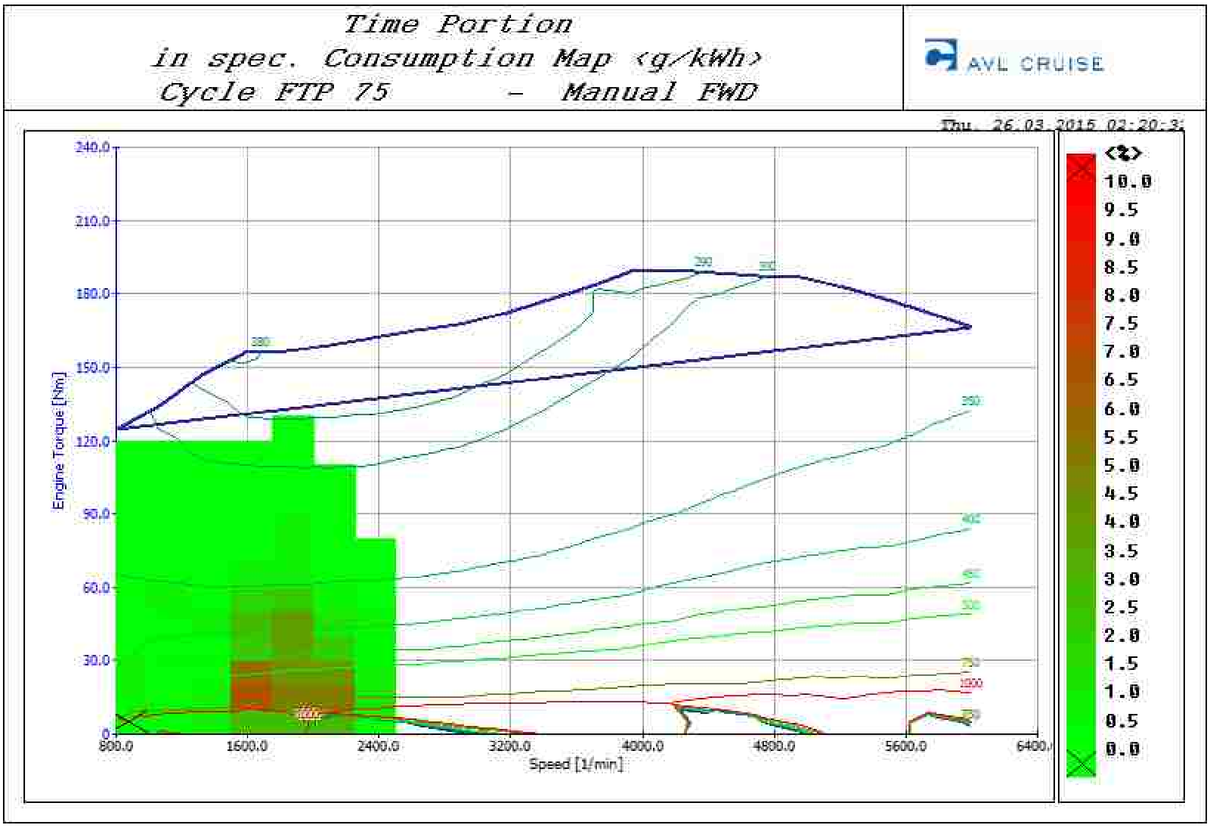

Figure 7.2.5: Driving time distribution for Separate Engine Strategy in highway cycle

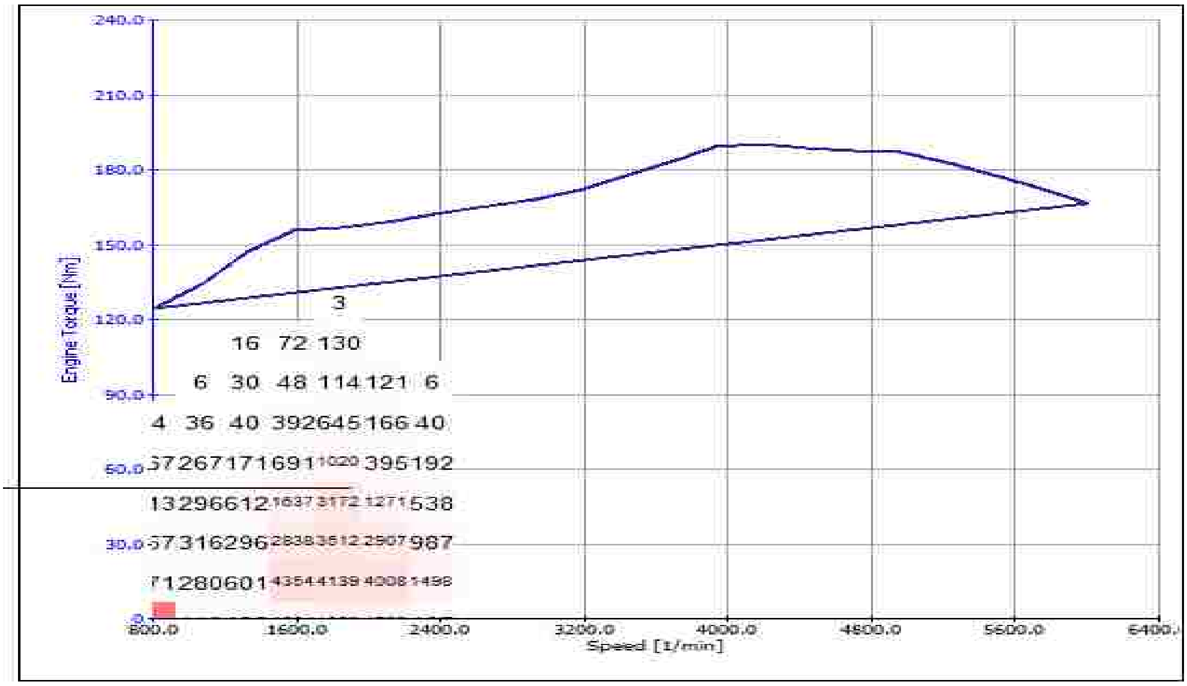

Figure 7.2.6: Driving Time Abundance for Separate Engine Strategy in highway cycle

From the above results, the following observations can be made:

The data generated is reliable and the engines considered are comparable entities. This is apparent from the trail run results where, the separate engine strategy when both the engines are kept running throughout the entire drive cycle, has a slightly higher fuel consumption when compared with a conventional engine due to the components added to connect the engines in series.

The highway cycle simulation generates better results as the engine runs more on open throttle while traveling at high speeds and also speed fluctuations, gear changes are comparatively less in a highway drive cycle.

The separate engine strategy has a better overall mileage because, it saves on the frictional losses which are otherwise experienced in the cylinder deactivation engine. This saving of friction in tern leads to better brake specific fuel consumption values and hence better mileage over cylinder deactivation technology.

The percentage of improvement in the fuel consumption by the use of separate engine strategy is higher in the highway when compared to city drive cycle. This is because the highway drive cycle maintains higher velocities and the decelerations are gradual, where as in the city driving conditions, the decelerations are very steep and result in rapid gear changes, as the control algorithm checks for vehicle speed, gear state and deceleration, this results in lesser time for the fuel saving strategy to capitalise on the driving conditions. Hence, the highway conditions are ideal for fuel saving strategy to set in for a longer duration, and therefore, better mileage improvement is observed in case of highway driving simulations. This is clearly evident from the driving time abundance graphs where, the top row having better brake specific fuel consumption points are more in number for a separate engine strategy, when compared to the driving time abundance points of other strategies.

[redacted] Driving time distribution vs fuel flow

CDA Engine

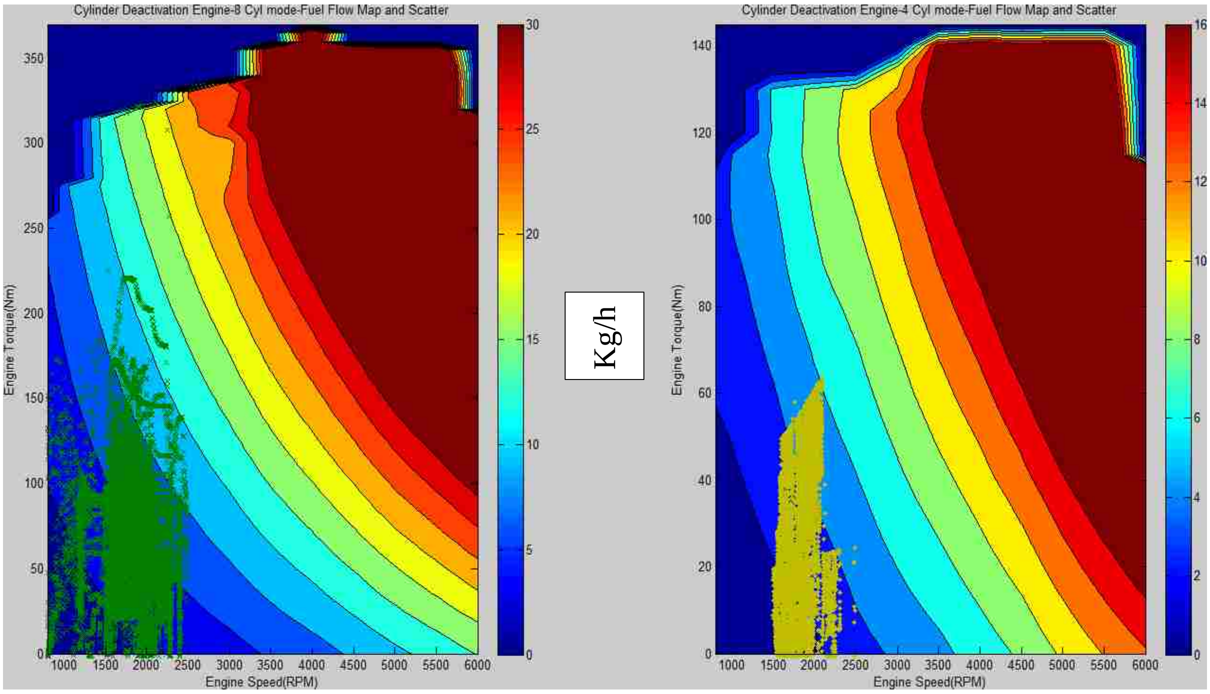

Figure 7.2.7: CDA Engine fuel flow

Separate Engine

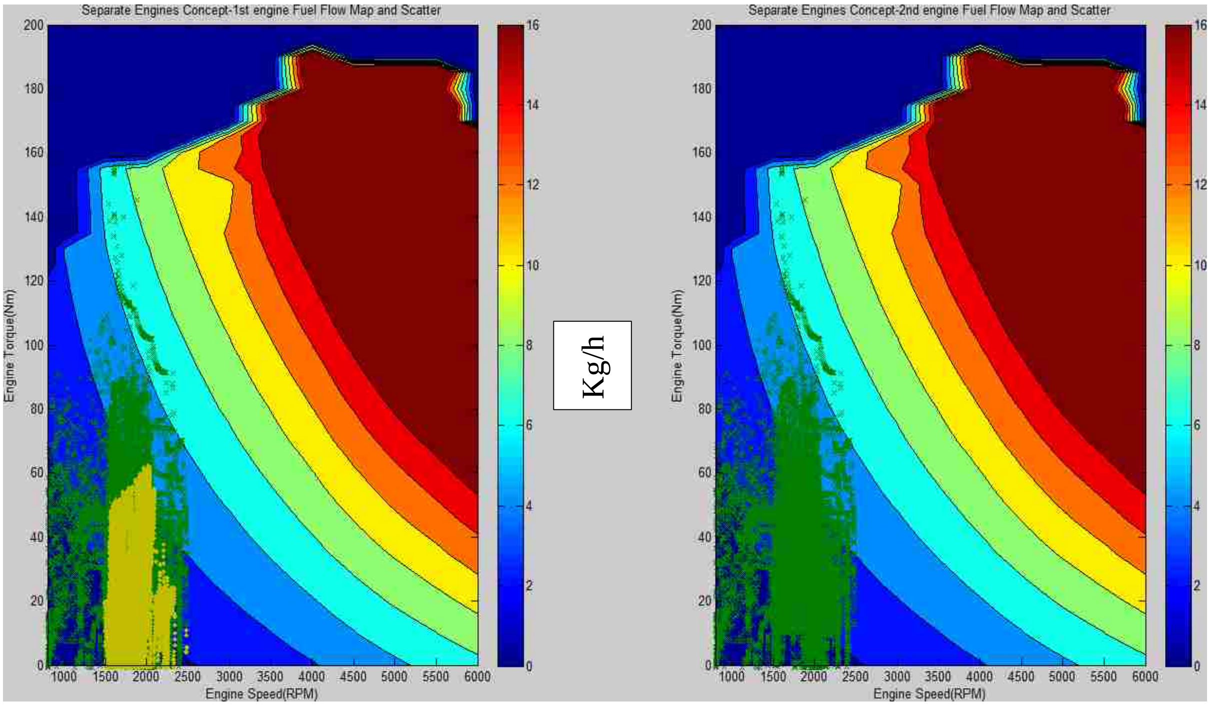

Figure 7.2.8: Separate Engine fuel flow

It can be observed from the Figure 7.2.7, the green points on the fuel map on the left side are a result of normal driving mode, and when the CDA mode is activated, the data is represented as yellow colour points visible on the right side of the Figure 7.2.7.

Just in the same way, the yellow colour points in the Figure 7.2.8, are a result of separate engine operation on the first engine and the decoupled engine operating fuel points are marked in green and are seen on the right side of the Figure 7.2.8.

It can be observed from the Figure 7.2.7 and Figure 7.2.8, that the yellow colour points in both the figures, which represent the fuel data with respect to engine rpm, operate in between 1500-2500 rpm, and at a maximum torque of 60 Nm. This implies that the control strategy used in both the cases is the same.

The fuel saving by the use of separate engine technology, can be calculated from the Figure 7.2.9, for example, consider the car to run at 1600 rpm and produce a power of 50 Nm, the corresponding fuel saving obtained, when this car operates using separate engine strategy would be 0.5 Kg/hr.

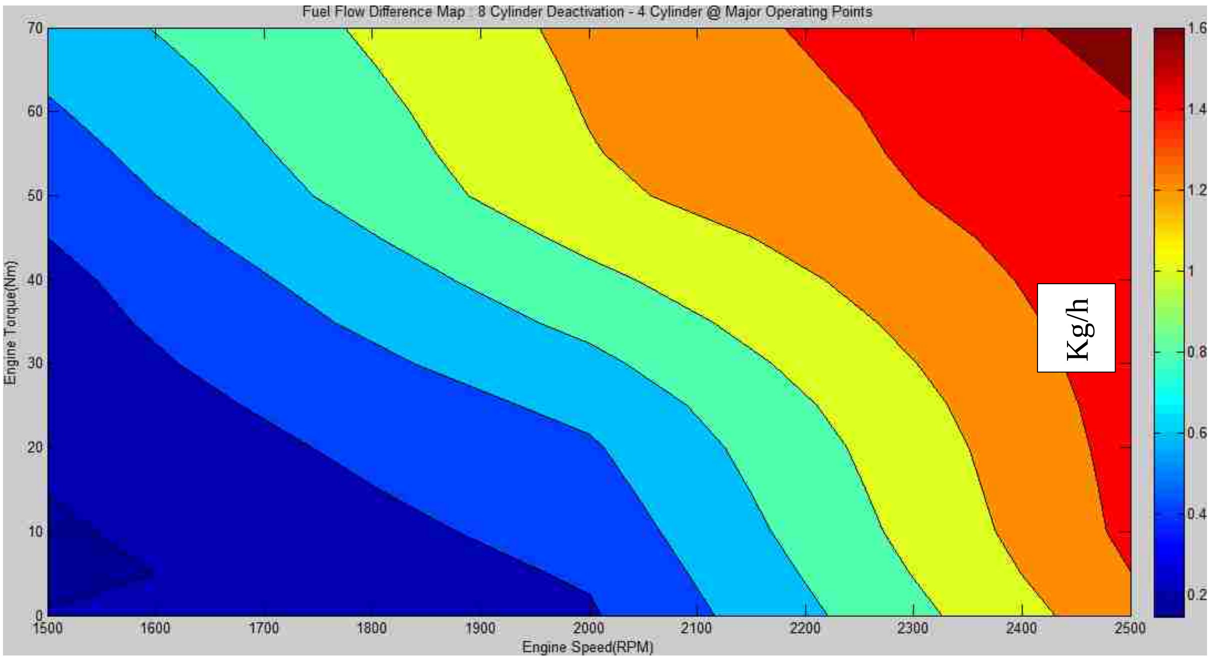

Figure 7.2.9: Fuel benefit from separate engine operation

8 Conclusions

The basic conclusions that can be deduced from this project are:

- Even though cylinder deactivation addresses the powertrain matching problem, it has a major setback involving frictional losses even when the cylinders are deactivated. This hinders the additional benefit obtained from the use of such a technology.
- The frictional losses incurred in the cylinder deactivation engine account to about 65-80% of the frictional mean effective pressure generated by the engine. So, these losses have to be curbed in order to get a maximum benefit.
- A separate engine strategy would eliminate the frictional losses incurred in a cylinder deactivation engine while addressing the problem of powertrain matching.
- From the Table 7.2.2, it can be inferred that during part loading conditions, the separate engine strategy has the best brake specific fuel consumption.
- In city driving conditions, a separate engine strategy has a mileage improvement of 6.5% over conventional engine and 1.2% over cylinder deactivation engine.
- In highway driving conditions, a separate engine strategy has a mileage improvement of 11.2% over conventional engine and 3.5% over cylinder deactivation engine.
- From the driving point of view, a maximum benefit can be obtained from using this separate engine strategy on a steady CRUISE.
- Form the application point of view, a maximum benefit can be observed when this technology is used in a muscle car or a pickup truck which experience severe powertrain matching problems.

9 Future Work

In a separate engine strategy, the second engine turns off when the vehicle complies with a set of conditions defined. A major limiting condition is the activation of such a technology only during deceleration, both engines keep running all the time, only when the vehicle decelerates and other set of conditions are satisfied, the second engine turns off.

Instead of turn off the second engine intermittently, It would be interesting to check for the feasibly of turning on the engine intermittently. So, an extension of this project would be to investigate the feasibility of supplying power only on demand. The first engine keeps running throughout the entire drive cycle, the second engine only kicks-in when there is an additional power demand form the driver.

There has to be no difference in terms of acceleration in a vehicle having this proposed technology and a conventional vehicle. For the same gas pedal positions, both these vehicles have to accelerate in the same way. In order to accomplish this, a continuously variable transmission has to be used in the vehicle which runs using this proposed technology. The gear ratios have to be varied depending on how many engines are being used at that particular instance.

While running on a smaller engine, varying the gear ratios on the continuously variable transmission and running engine on higher rpms, meets the acceleration demands of the driver. This allows the first engine to run as close as possible to the optimum operating line and thus, the efficiency gets improved.

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