POLITECNICO DI TORINO

In partnership with:

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Master's Degree in

MECHANICAL ENGINEERING

Impact of VCR con-rod switching time on fuel consumption

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Abstract

Variable Compression Ratio (VCR) Engines are among the most valid technologies for emissions reduction. Being emissions targets increasingly stringent, beside the vehicle powertrain electrification or hybridization, it will be essential to exploit the internal combustion engine potential. VCR engines allow to maximize the efficiency of combustion process by operating at low loads with higher compression ratios, compared to a base engine with fixed one. Further benefits are achieved at high loads, where a lower compression ratio can be used in order to face knock and improve fuel economy. There are several studies on the different Compression Ratios switching technologies and on how to refine the mechanisms and maximize the benefits. Taking as a fact the improvement of combustion efficiency, in both high loads and low loads operation, this Thesis Work aims at establishing how much the switching time impacts on the fuel economy.

In order to provide such a result, a complete versatile and parametric model in *Matlab* has been created: given as Input the switching time, the vehicle specifications, the fuel maps corresponding to the two different compression ratios and a Driving Cycle, the model computes the percentage decrease of fuel economy with respect to instantaneous switch case.

Different scenarios have been analyzed so to verify the correct working of the code and evaluate the combined effects of other factors as type of transmission and number of switches in the driving cycle.

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Chapter 1

Introduction

As well known, nowadays the main concern in automotive industry is about pollutant emissions. All the actual researches and new technologies are focused on reducing the vehicle environmental impact. The legislations are becoming increasingly stringent requiring an average fleet fuel consumption significantly reduced with respect to previous regulations.

On 8 November 2017, the European Commission submitted a legislative proposal setting new CO_2 emission standards for Light-Duty Vehicles (LDC) in the European Union for the period after 2020. This proposal, in addition to bonuses for whom respect the threshold, has a 30% target reduction in CO_2 emissions by 2030 with respect to 2021 levels. An intermediate requirement of 15% reduction has been imposed by 2025 in order to check if the reduction is possible and to speed up the process and reach the targets as soon as possible.

The transition to this new framework is based on the already established EU targets for 2020/2021 of 95 g CO_2/km for passenger cars and 147 g CO_2/km for light commercial vehicles, both of which are based on the NEDC (New European Driving Cycle).

The other main target remains to move towards a regulation that includes all the countries since, in the past, several critics raised because of different testing cycles and thresholds. All the new targets are then set taking as reference the Worldwide harmonized Light vehicles Test Procedure (WLTP) combined with more reliable emissions tests in real driving conditions ('Real Driving Emissions' - RDE). Being these procedures phased-in in these years, all the fleet wide targets have been set as percentage reduction and not as an absolute value.

Thus far the scenario presented is pretty challenging and it won't be possible to achieve such targets with powertrains using conventional Internal Combustion Engine (ICE) or with Electric Vehicles (EV), since the costs are still too high and a commercial spread won't be possible in the short-time. From here raises the need to focus all the attention and technologies on fully exploit the Internal Combustion Engine (ICE) potential, improving its efficiency and guaranteeing a very high performance level, low fuel consumption and low total cost. There are already several well-known strategies, technologies and constructive solutions operating in this field, as:

- Direct Fuel Injection (Common Rail Direct Injection);
- Supercharging (Supercharger and Turbocharger);
- Variable Valve Actuation;
- Nitrogen Oxide Injection;
- Inter- and After- Cooler systems;
- Low friction designs, materials and lubricants;

Beside which the **Variable Compression Ratio engines** stands. These systems represent a valid alternative offering significant improvements in fuel economy reduction and Break Mean Effective Pressure (BMEP) and also have a good developed technological status.

Overlooking the details for a subsequent deeper analysis, the basic working principle of VCR Engines is to geometrically modify the chamber volume so to change compression rate during the operation. The benefits are experienced on two sides:

- Using an high compression ratio at **medium-low loads** corresponds to the downsizing strategy. Even if in this case the operation is performed by using higher compression rates, the performances are improved since with an equivalent smaller engine there are reduced pumping and friction losses in addition to reduction of gases heat transfer with the same power. Thus the efficiency in the low region operating points is substantially improved, leading to a better fuel economy.
- One of downsizing major limitations is knock (spontaneous ignition) at **high loads**. There are several techniques adopted in standard engines to mitigate knock (cooling, mixture enrichment, direct fuel injection) but they usually lead to an increase in pollutant emissions and need of tradeoff with efficiency. This problem is completely solved in VCR Engines since it is possible to decrease the compression rate in the interested operating zone, allowing the engine to work below knock whatever the load.

Moreover, VCR Engines are compatible with other technologies as Variable Valve Actuation and Atkinson Cycle, making possible a significant reduction in emissions together with an improved fuel economy (up to 40% when combined with turbocharging).
In this latter case the benefits are so high since VCR reduces turbo lag effect by decreasing the compression ratio (better inlet condition at turbine inlet); simultaneously it provides an optimized catalyst operation due to control of combustion parameters (temperature, pressure, rate of expansion), allowing to change the exhaust gases temperature depending on the engine condition (cold start, regime) and making catalytic efficiency as high as possible.

It is worth to highlight that at optimal conditions VCR engines can lead to a 8% fuel consumption decrease by themselves.

1.1 Overview of VCR Systems

A chamber geometry modification remains the sticking point of Compression Ratio variation. It can be performed in several ways, by adding an additional volume or varying the pistons Top Dead Centers positions. Different studies have been conducted on these technologies, offering a wide range of possibilities.

The main Variable Compression Ratios engines can be classified as reported in Figure 1.1:

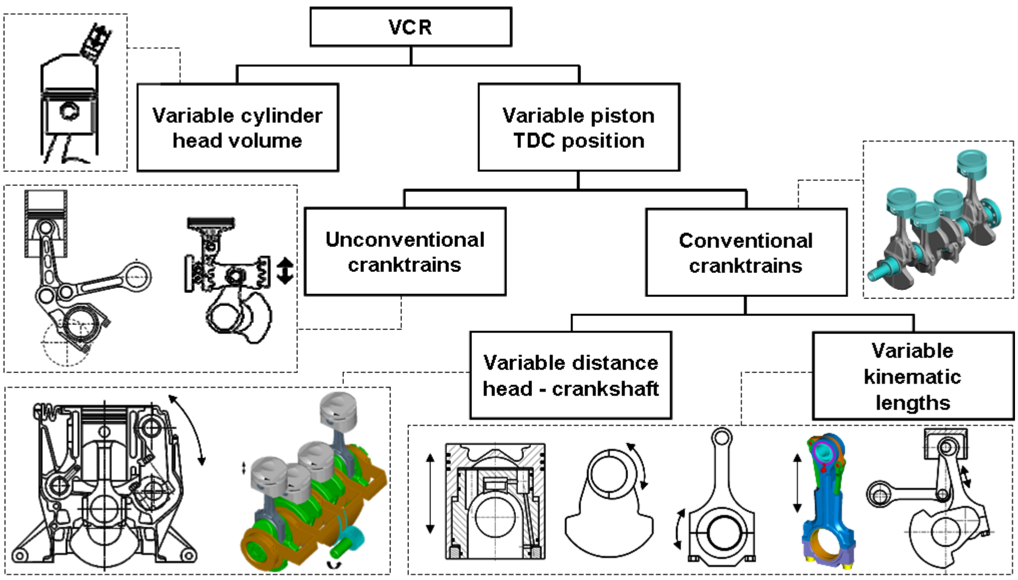


Figure 1.1: Classification of VCR systems

Getting to the choice moment, being the number of mechanical devices capable to perform the modulation pretty high, some parameters as engine architecture modification needed, suitability for engine operation, production, costs must be taken in account. Classifying the technologies as reported in the image 1.1, the main advantages and disadvantages are the following:

- Unconventional cranktrains: the CR variation is continuous, going through a wide range of values, hence a precise control and a better exploitation of the system is possible. On the other hand significant space in transversal direction is required, needing a complete replacement of the engine and a new design stage for the architecture and leading to higher costs for manufacturing process.

- Systems where the distance between the crankshaft and the cylinder head is varied also allow a continuously variable CR; compared to previous type they show the advantage tofutilizing the existing cranktrain with no additional modifications.

- The cylinder head and barrel compound position can be varied with translatory or rotary motion but, again, the base engine design has to be extensively modified. These systems require a complete engine architecture revolution, necessitating also an intake and exhaust systems coupled movement, thus making the costs and the packaging unfeasible.

- Eccentric bearing crankshaft, still providing a continuous variable CR, requires the introduction of a compensating gear for the forces generated. So even if no substantial modifications are needed, new losses will be introduced.

- Concluding, there are the systems involving the cinematically effective lengths modulation: even if the CR variation is not continuous but discrete (the system can take only two CR values), the costs will be very low and no modifications to engine architecture will be needed.

As a consequence, the length-adjustable con-rod with eccentric piston pin, most commonly known as 2-step VCR system, can be considered the best option because of the low cost manufacturability, the easy integrability and modularity into common engine architectures and the enhanced turbocharging. The values of fuel consumption and BMEP benefits are almost comparable to continuous variable CR system and overcome the only drawback of slightly increased reciprocating mass.

1.2 Two-Stage VCR System

The working principle of the system is based on the rotation of the eccentric sleeve that makes the connecting rod length vary. The mechanism is activated by the forces acting on the piston head that, through the wrist pin, are translated in torque acting on the sleeve. The moment, during the cycle, takes both positive and negative values, making possible precise adjustments.
This pretty simple device is the key feature of 2-stage VCR system, allowing a low cost, easy manufacturing, and avoiding the usage of expensive actuators or complex control systems.

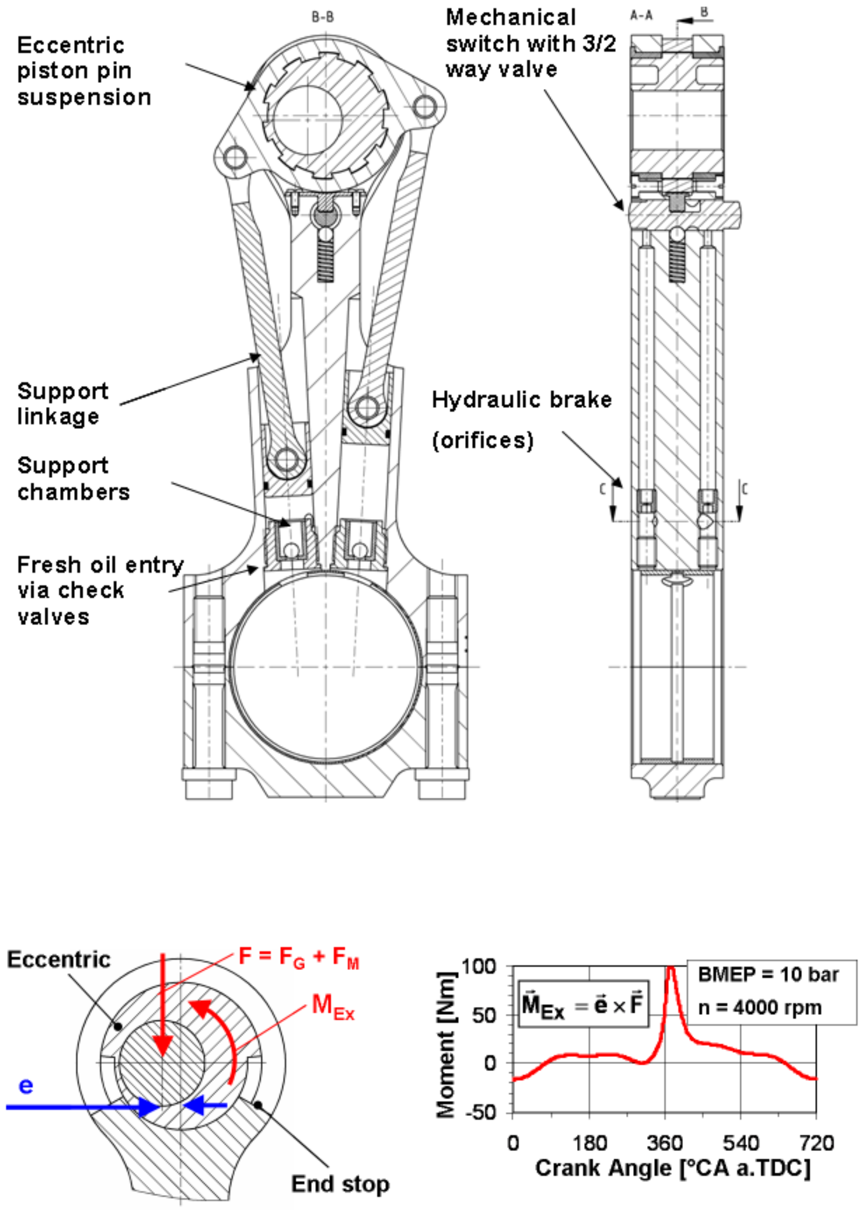


Figure 1.2: Two-Stage VCR System

The piston support includes two hydraulic chambers connected to the crank case by a 3/2 check valve and to the oil circuit by two check valves. When the connection between one of the support cylinders and the crank chamber is made, the oil flows outside and the support piston move downwards; in the meantime fresh oil supply is provided in the second chamber through the check valve, positioning the second piston in the upper part. In this way the systems come to a free-wheeling configuration and the eccentric can rotate only in one direction.

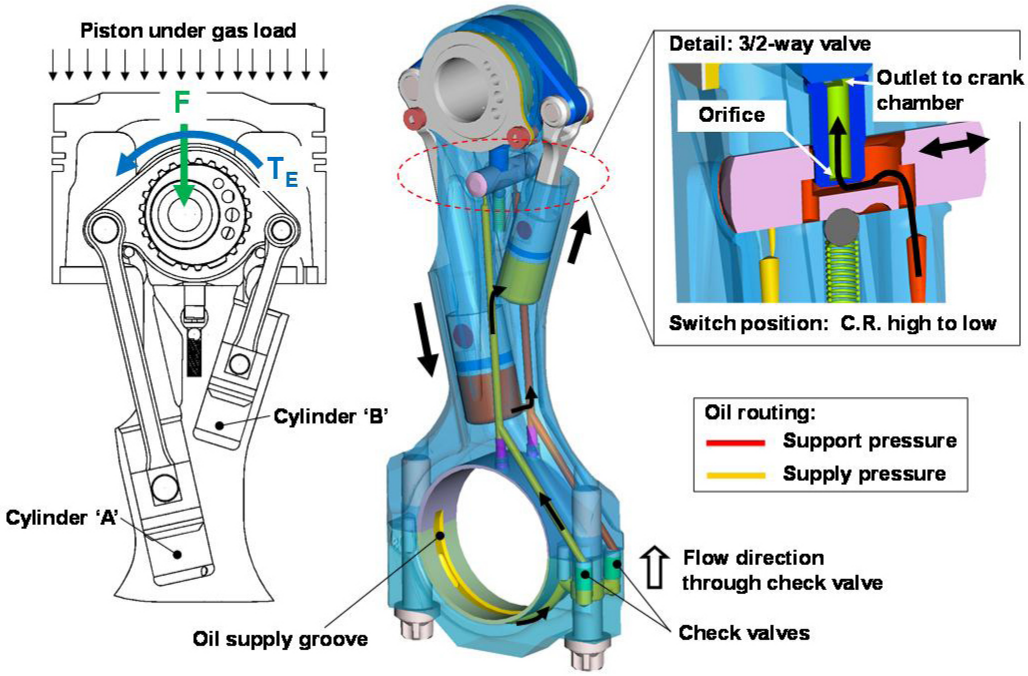


Figure 1.3: Two-Stage VCR System - Valves working principle

Even if there are several possibilities, the 3/2 valve is actuated by the curved (cam) disks, that are pushed towards the valve movement direction. Being an integrated part of the connecting rod, the valve curves in specific location and is moved axially: lock position is actuated. The camshaft is positioned depending on the engine arrangement.
As anticipated moments in both directions act on the mechanism so to allow a proper adjustment, that will take some cycles though. The time required depends on the hydraulic resistance, in turn controlled by orifices acting like hydraulic brakes. The process should be designed so to not take a lot of time, so to avoid knock moving from part load to full load and exploit as soon as possible the efficiency of the VCR system. On the other hand it cannot be too fast otherwise cavitation in the support chamber or high impact can occur at support pistons.

Concluding this section it is worth to remember that is possible to perform a Two-Stage VCR System also with an eccentric connecting rod big end bore or a with a telescopic conrod shank. These latter cases are not considered because of minor use with respect to the solution just analyzed.

1.2.1 Response Time

In opposition to the driving force coming from the combustion chamber the resistances in connecting rod mechanics together with the fiction losses in the hydraulic brake should be taken in account. Hence the major part of resistance can be attributed to the orifices, showing a low dependency from oil temperature.

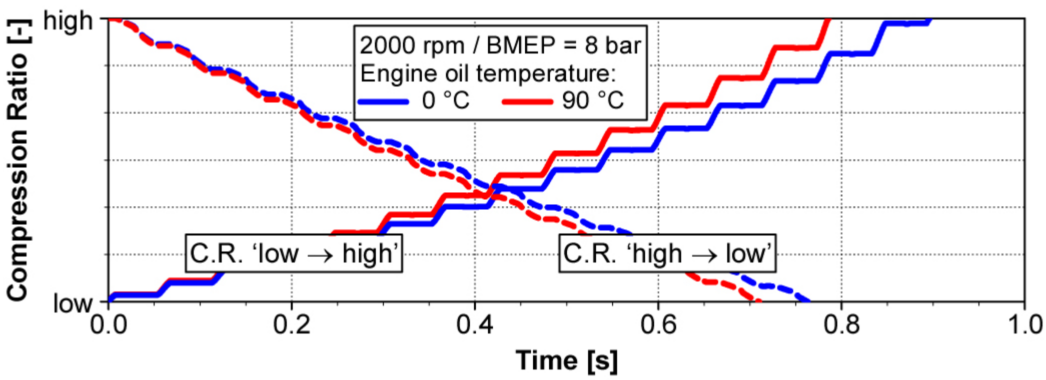


Figure 1.4: Influence of engine oil temperature on compression ratio actuation time

Considering the low-to-high and high-to-low switching process compared at same engine operating conditions it can be noticed how the difference between cold start and regime it's only 6%.

In order to better identify the switching time response and all the parameters that can affect the outcome, the same situation has been analyzed at a load step from part load to full load. The simulation, coming from an external research, has been conducted at 2000rpm in 6th gear. The switching starts exactly when the step is performed:

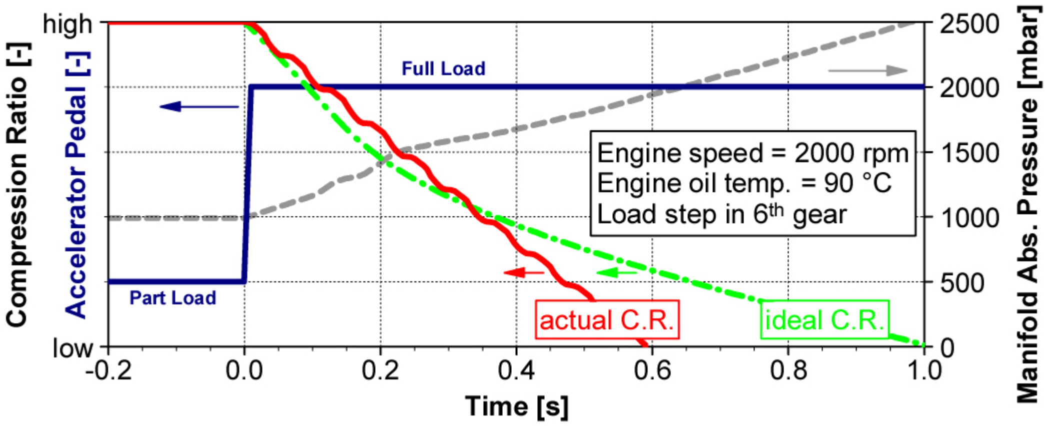


Figure 1.5: CR actuation and boost pressure build-up after positive load step

The switch is much faster than boost, hence further reduction in response time would not result in further improvement but only in torque reduction.

Anyway in the *Matlab* model has been introduced a further parameter accounting both for time needed from the signal to be sent from control system to actuation one and for time response of the mechanisms. In this case the quantities are expressed as a fixed time interval that can be mapped and further impact on total switching time is registered. The results of fuel consumption variation will be analyzed in the conclusive part of the thesis work.

1.2.2 Assumptions on Switching Time

As mentioned before, the mechanism working principle is based on the forces acting on piston head, that can be identified as gases pressure and inertia. Usually the analysis is conducted by tracing the cylinder pressure signals: in this way it is possible to isolate the switching events following the different peak pressure levels.

This method has some drawbacks though, since it comes with poor accuracy and the identification of effective switching time will be difficult. A more precise analysis can be conducted adapting eddy current sensors near the liner in two different positions.

In general VCR con-rod system shows pretty high switching velocities that need to be quantified for the need of this study. More than an exact value it is necessary to understand in which terms the switching time should be expressed: if in seconds (fixed time interval) or with a fixed number of engine cycles.

As the gas and inertia forces vary with engine speed, the switching time is not constant and is more closely related to number of cycles. In practice, neither is the switching time constant nor are the number of cycles. However, based on the design of the con-rod, it is more closely related to the number of engine cycles. So, in this study, the switching time will be expressed as a fixed number of engine cycles: this is a simplification but it is more representative of the actual operation than picking a fixed time.

Basically in the model created the switching duration will be among the parameters to be introduced and considered as a variable. In this way it will be possible to get a valuation of fuel consumption to switch duration.

Chapter 2

Engine Fuel Maps Generation

Before going to the core of this thesis work, consisting in the switching logic implementation and consequent impact on fuel consumption, it will be necessary to obtain the two fuel maps corresponding, respectively, to High and Low Compression Ratio.

Being the work intended for publication, it was not possible to use already existing maps for confidentiality reasons. Hence, through the GT-Power software, a 4 Cylinder SI engine was simulated, at the two different compression ratios, in order to compute the Brake Specific Fuel Consumption in a wide range of operating points allowing a subsequent precise interpolation.

After a quick overview of the GT-power model and the assumption and simplifications introduced in the model, the results will be shown in this chapter.

2.1 GT-Power

GT-Power is a powerful tool that allows one-dimensional simulation of any kind of Internal Combustion Engine. The modeling format in GT-ISE uses an object-oriented structure. This structure is comprised of a three-level hierarchy: templates, objects, parts. Templates are provided which contain the unfilled attributes needed by the models within the program. The templates are made into objects, and when component and connection objects are placed on the project map with a drag-and-drop operation they become parts and inherit their values from their parent objects. These parts may call reference objects like air and fuel properties, heat transfer objects, combustion objects, etc. During the course of building the model many reference objects will be used, and most are automatically imported into the project at the time they are needed. Moreover the graphical layout is very intuitive and allows a clear and faster understanding of the model.

Depending on the results needed all the libraries are divided into main categories as controls, acoustics or vehicle driveline. For this work the engine performance section has been investigated. Starting from the tutorials provided by the software developers, a model has been chosen and properly modified so to perform the desired simulation.

At this stage, the engine has been considered decoupled from the vehicle, as a test-rig. After the vehicle has been chosen so that the specifications, as displacement, fuel supply and so on, were matching the ones inserted in the software. In the future practical usage of the *Matlab* model this adaptation won't be necessary since real fuel maps, obtained with real tests, will come together with the vehicle.

2.1.1 Basic 4 Cylinders SI Engine

The construction of the model starts considering a Basic Cylinder Engine that then will be turned into a 4 cylinders inline spark-ignited engine. The base components of the single cylinder engine will be copied three times and an intake and exhaust manifold will be introduced to the system. The base geometry of the manifold will be outlined and discretization will be discussed.

The 1 cylinder base engine has been modeled accounting the following components:

- Envirorment Component
  - Intake Environment
  - Exhaust Environment
- Round Pipe Component
  - Intake Runner
  - Exhaust Runner
  - Intake Port
  - Exhaust Port
- Valve Cam Connection Component
  - Intake Valve
  - Exhaust Valve
- SI Injector
- Cylinder Component
- Cranktrain Component

In the library section the templates can be found organized in a hierarchy structure, under the main classes Flow, Mechanical and General.

The graphical layout is reported hereafter:

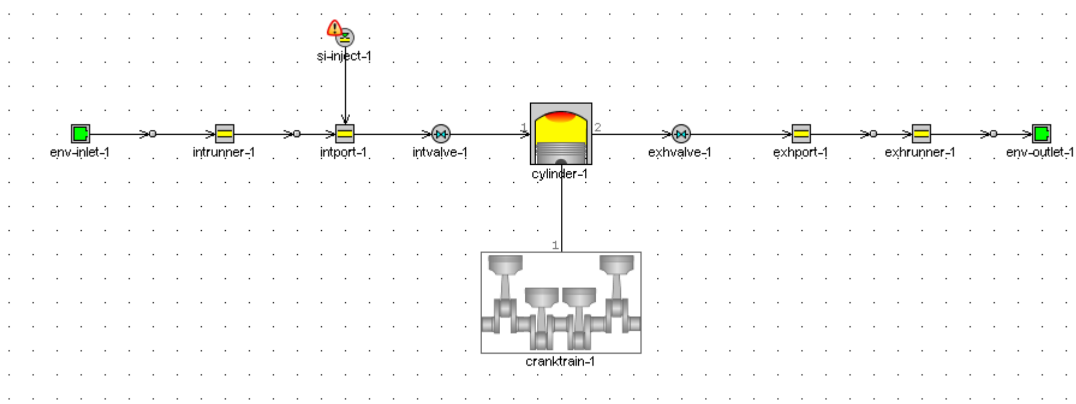


Figure 2.1: Basic 1 Cylinders SI Engine GT-Power Map Layout

All the elements properties have been set with the default values reported in the software tutorials and no modification were applied.

It is worth to notice that an error appear when some setting has not been imposed properly or information are missing. In this case it shows up at the SI injector since the component will need a Part Name for the Air Mass Flow Rate Sensor. It can be easily solved useing an RLT (ResuLT) to sense the mass flow through a selected orifice. An RLT can come from a single engine cycle, typically an average, maximum, or minimum value.

Moving to the 4 cylinders, the procedure involves the copy and paste operation of the 1 cylinder and adding intake and exhaust manifold.

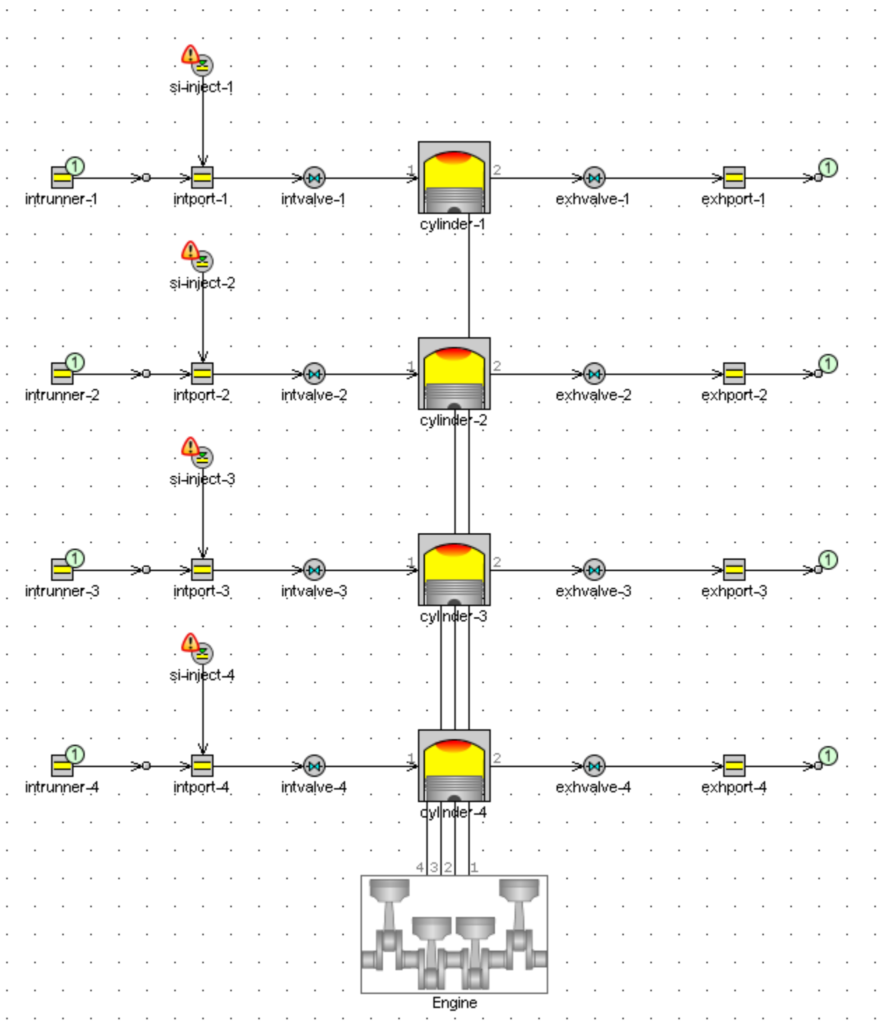


Figure 2.2: 4 Cylinders SI Engine GT-Power Map Layout - Step 1

As can be noticed the Cranktrain element is in common for all the four cylinders. An important step in building a multi-cylinder engine is making certain that the cylinders are connected to the 'EngineCrankTrain' part correctly. If the cylinders are defined as 1-4 numbered from top to bottom on the project map, the cylinders should be connected to the 'EngineCrankTrain' in that order.

The intake system model is made of an inlet pipe, a volume for the air cleaner, a throttle valve, and a log-style manifold. The four runners will come off the manifold and connect to the intake ports.

The manifolds, both at intake and at exhaust, will consist of a series of general flowsplits and pipes. Starting from drawing of the manifold an approximation method used to discretize the geometry. The discretization consists in dividing a physical volume into smaller sub-volumes.

Referring to the exhaust case the process is resumed in the following images:

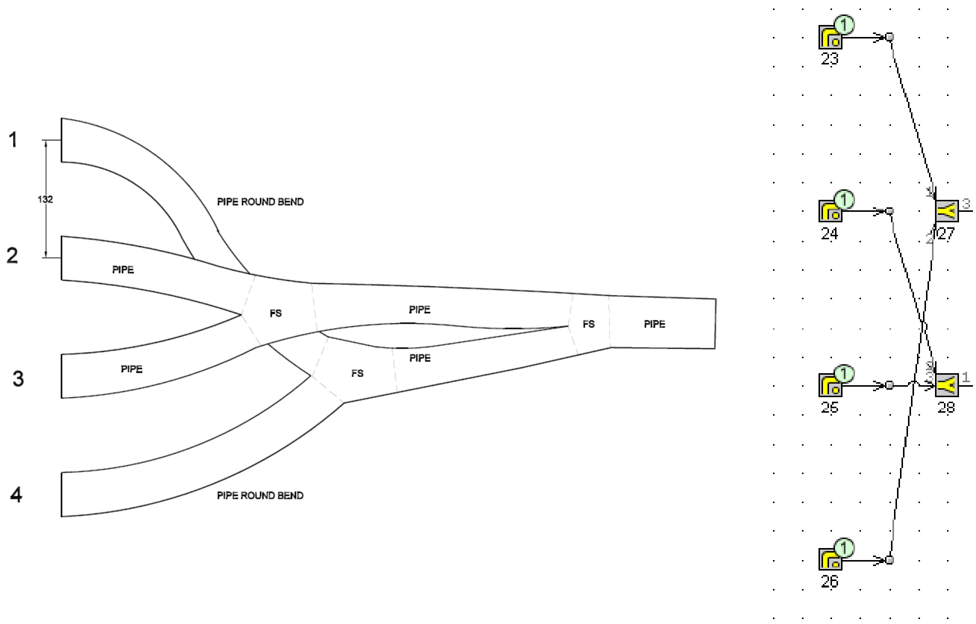


Figure 2.3: Exhaust Manifold Discretization

Once both intake and exhaust systyems have been added, the model of a 4-Cylinders SI engine provided as GT-power default model is obtained. In order to meet the purposes of this thesis work some adjustment to this basic model are needed.

2.1.2 Throttle Controller

In first place the throttle valve, substituting the orifice connection present in the base model, should be connected to a controller. In this way it will be possible to perform a load targeting during the engine operation: at a given engine speed the controller will change the throttle angle of the intake valve so that the load reach pre-specified values. Thanks to this process then a variety of speed and load combinations will be covered and Brake Specific Fuel Consumption will be computed in each one of them. At the end the engine fuel map is obtained in GT-Post through interpolation.

In the GT-Power library it's present a template called *"Throttle Controller"* performing what just described. In order to proper set all the properties of the element an investigation has been necessary. Basically it is a PID controller with specific input requirements for a valve control. In general, a PID controller is used to control an input to a given system so that the output of that system reaches a target value (it can be used to target almost any quantity). Being so generic, some efforts are required to calibrate its "gains" so that the convergence on the output value can be obtained. Moreover it is required that the target is reached:

1. In a stable manner - a steady output signal so that there is not oscillation around the target;
2. In a prompt manner (i.e. reasonable time scale) so that excessive computational time is not required waiting for the signal to move.

Since in this simulation the number of cases will be pretty high (around 60) a fast control is needed. For this reason, GT-SUITE provides specialized templates, known as model-based controllers, which can replace a PID controller for a few common engine control situations, among these it is possible to find the Throttle Controller element inserted in the project.

Inside the controller a variable element named [*target_load*] has been introduced: in this way a variable appearing in the GT-Power Case Set-up has been created. When setting the simulation cases it will be possible to specify, at a given RPM, the desired load. As can be seen (Figure 2.4) in fact the Throttle controller element must be connected to the 'EngineCrankTrain' element from where reads the values needed.

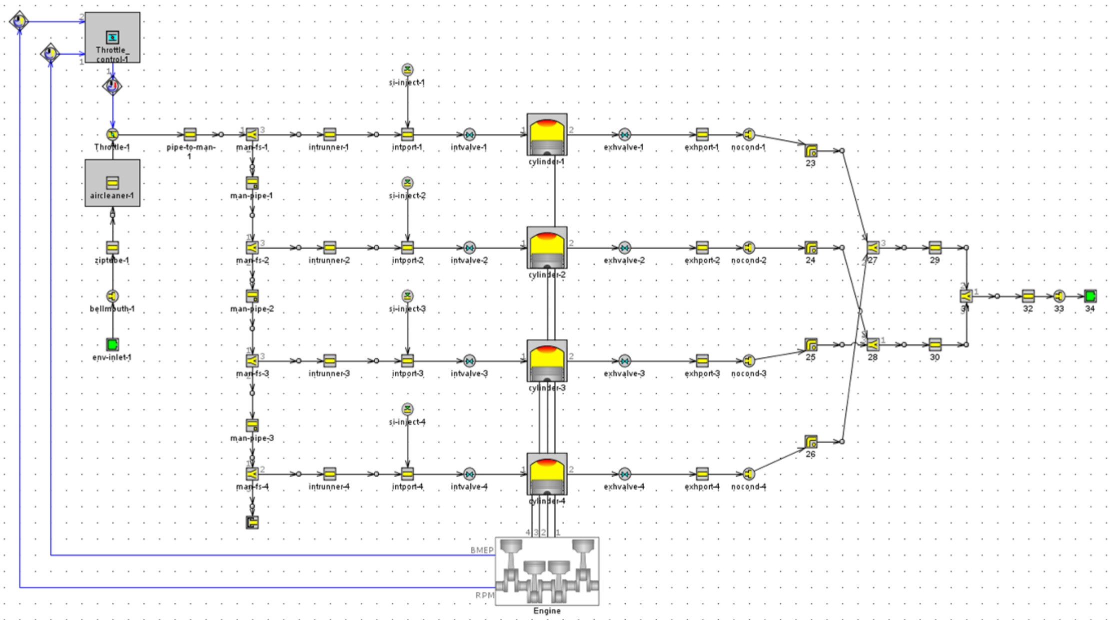


Figure 2.4: 4 Cylinders SI Engine GT-Power Map Layout

The Throttle Control element performs an Open-Loop control, so compare the value of actual BMEP with the user-specified one for that case and varies the throttle angle consequently, up to convergence.

2.2 Assumptions on Combustion Model

When Knock appears one of the first way to face it is to adapt the combustion phasing. Being in this case the Compression Ratio variable and a whole map of operating points considered, some data about CA50 variation with speed and load at different CRs had to be found in the literature.

As a reference experimental data obtained testing a conventional fixed CR GTDI engine were used. The baseline CR was 10:1. Experiments were conducted at 8:1 and 9:1 by means of replacing the connecting rods with shorter rods and using the baseline pistons. Higher CRs, 11.9:1 and 13:1, were achieved by using the production connecting rods with prototype pistons.

It should be highlighted that this data have been used with the only purpose of extracting an overall trend of CA50 retard and Burn Duration when varying the three parameters Compression Ratio, Engine Speed and Engine Load. The values introduced in the GT-power model are not equal to experimental ones and have been exaggerated, then too far removed from reality.

The reference data were available in the following shape:

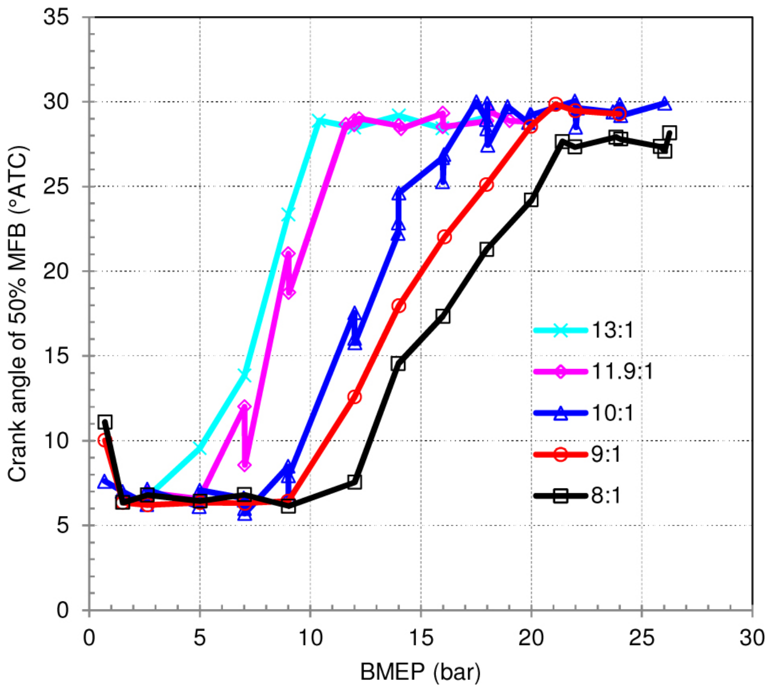


Figure 2.5: Measured knock-limited CA50 versus BMEP including all points, 2000 rpm, 91 RON fuel

So fixing the engine speed and increasing load at different compression ratios. The experimental data considered, in an effort to avoid low speed pre-ignition, show an amount of spark retard constrained to CA50<30°ATC. Once knock-limited CA50 reached 30°ATC at stoichiometric A/F, enrichment was used to mitigate knock up to the enrichment limit of 0.75 lambda. In this case instead, the A/F ratio has been not modified, increasing retard up to 50°ATC. This value is obviously not correct and impossible to find in real engine operations, still it has been used to represent the high knock limitation for the high compression ratio.

The main purposes at this stage remains to obtain two plausible fuel maps in order to proceed at the main object of this thesis. Again, if used with real maps, the results obtained will be exact.

Assuming to use a value of Low Compression Ratio 10 and High Compression Ratio 13, the values of CA50 and Burn duration are reported in the following tables. Experiments conducted at different Engine Speed were considered and interpolation extrapolation tools used.

		BMEP [*bar*]					
		2	4	6	8	10	WOT
Eng Speed [*rpm*]	1000	7	9	12	15	28	30
	2000	7	8	9,6	10	23	25
	3000	7	7	7,1	9	18	20
	4000	7	7	7,4	9	15	17
	5000	7	9	9,1	9,5	13	15
	6000	7	9	9,5	9,7	11	13

Table 2.1: CA50 (°ATC) - Compression Ratio 10

		BMEP [*bar*]					
		2	4	6	8	10	WOT
Eng Speed [*rpm*]	1000	39	37,5	35,3	33,7	32	31
	2000	30	27,5	26	24	22,7	21,7
	3000	28,2	25	24,5	22,5	20	19
	4000	29,5	26,5	25,8	23,5	21,5	20,5
	5000	31,5	28,5	27	24,2	21,8	20,8
	6000	34	30,5	28,7	25	22,7	21,7

Table 2.2: 10-90% MFB duration (°) - Compression Ratio 10

On the other hand, when the compression ratio increases:

		BMEP [*bar*]					
		2	4	6	8	10	WOT
Eng Speed [*rpm*]	1000	8	11	15	38	45	50
	2000	8	10	12,6	33	40	45
	3000	8	9	10,1	28	35	40
	4000	8	9	10,4	25	32	37
	5000	8	11	12,1	23	30	35
	6000	8	11	12,5	21	28	33

Table 2.3: CA50 (°ATC) - Compression Ratio 13

		BMEP [*bar*]					
		2	4	6	8	10	WOT
Eng Speed [*rpm*]	1000	40,75	40,5	40	38,5	36,7	33,8
	2000	31,75	30,5	34	33	29,5	29,4
	3000	29,95	28	31	27	25,7	25
	4000	31,25	29,5	31,5	28	32,5	26
	5000	33,25	31,5	32	29	28	26,5
	6000	35,75	33,5	35	31	29	27,5

Table 2.4: 10-90% MFB duration (°) - Compression Ratio 13

This section is based on the premise to have the same engine hardware, with the only difference being in the CR which will physically show up as a squish height difference. In terms of operating, the valve timings are kept the same and the simulations are running at the same speed and loads. At the lower loads, the CA50 is in the 7-9 range which is essentially MBT. In theory one could have set CA50=8 for all cases below 6 bar, but the difference in load for a 1 deg change in CA50 at MBT is negligible. When knock is encountered on an actual engine, the first response is to retard spark and thus CA50. As the simulation is running with a prescribed combustion model, the CA50 is varied instead of spark. If a predictive combustion model was available, then spark would be set and the knock model would have been retarding the spark as needed to avoid knock. In the absence of a predictive combustion model, the next best option was to use the prescribed combustion (Wiebe) model. As knock starts at lower loads at higher compression ratios, CA50s start retarding sooner (lower load) at CR 13 than CR 10.

In order to have a qualitative trend behavior of CA50 and Burn Duration with speed and loads experimental data regressions have been taken as reference from a paper. For this study purposes it is also very important to understand how this parameters vary with compression ratio along with load and speed. Moreover, it is worth to highlight that the numerical values obtained from experimental data have been used only as a reference and not exactly reported: in first place because not all the value needed were present, in second place because a more aggressive, and in a practical and theoretical way "overstated", retard is needed for the purposes of the study.

2.2.1 SI-Wiebe Combustion Model

By analyzing more in depth the engine simulated, some clarifications about the combustion process are needed at this point. The model used, SI-Wiebe, is a simple, non-predictive way to define combustion in GT-POWER. Since it is a non-predictive model, simulations that use objects created with this template run very fast. The attributes for SI-Wiebe typically only define combustion for one operating condition, then it will often be necessary to change the attributes for different operating conditions. The attributes control can be accomplished several ways: each one of these features describing the combustion can be made a parameter and changed from case to case.

Several Dependency Reference Objects are allowed for the attributes. If one knows how the attributes vary with speed and load, an RLT Dependence using an XYZ map can be set up. It's here that the two dependency map derived from experimental data came into play.

The two tables have been inserted into the model and results analyzed.

2.3 Simulation Routine and Results

GT-Power had to be resorted since it was necessary to generate two fuel maps of the same engine at different discrete compression ratios. Hence two identic models had to be created, with the only variation in Cranktrain element, were compression ratio has been varied. Together with this modification, the connecting rod lentgh have been varied so to follow the system physics.

The values inserted are reported in Table 2.5:

CR [/]	Con-rod length $[mm]$
10	175
13	177,8

Table 2.5: Compression Ratios and Con-rod lengths in GT-Power models

After setting up the model and al inserting all the dependencies, the process can be outlined in the followinf way:

- Start the simulation in a set of engine speed in the range of $[800 \div 6000]rpm$ with step of $500rpm$ (except for the first one that is $200rpm$) imposing a constant Throttle Angle of 90° (Throttle Controller not used). In this way it will be possible to compute the full load curve corresponding to the two different compression ratios.

- Once the maximum loads that the engine can reach have been computed, a further discretization is done. For each engine speed mentioned above a set of operating points has been computed increasing the load from $0barBMEP$ to full load with a step of 2 bar bmep. Each point is reached thanks to the Throttle Controller.

- At the end in GT-Post the fuel surface maps are obtained.

Each of this step has been performed in two different models, each one accounting for a different compression ratio.

The full load curves obtained in the first step are reported in the Figure 2.6.

The higher difference between the full load curves can be spotted at low engine speeds. At 1000 RPM, the engine will be more knock limited at WOT as there is more physical time for the fuel-air mixture to autoignite. Being just based on CA50 retard a 1.5 bar BMEP gap can be plausible for instance having an aggressive CA50 at CR10 and very retarded CA50 at CR13.

In order to have a graphical impact of the assumptions made on combustion, simulations at Wide Open Throttle have been conducted on models accounting for a fixed CA50 in the whole map. The results are compared in figure 2.7.

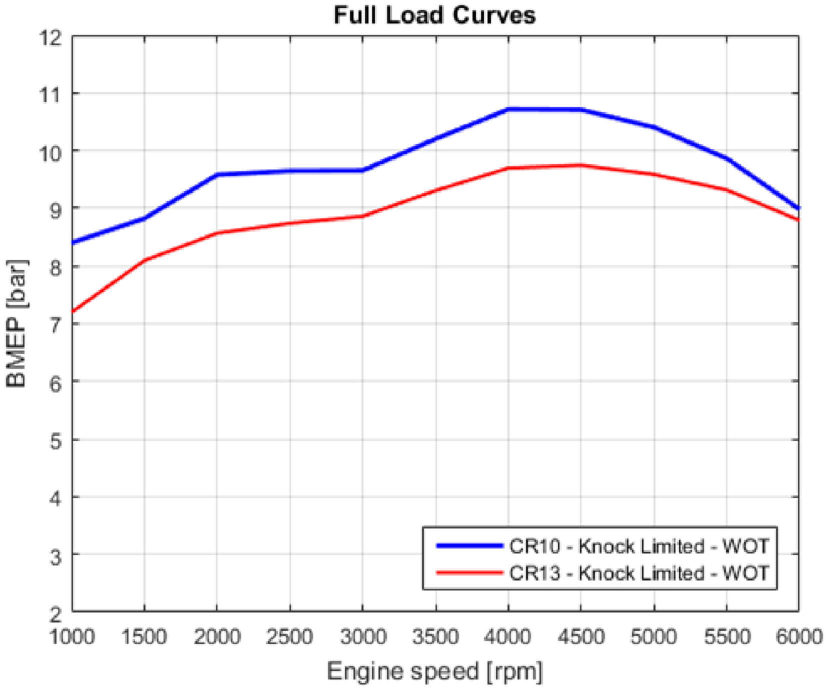


Figure 2.6: Full Load Curves

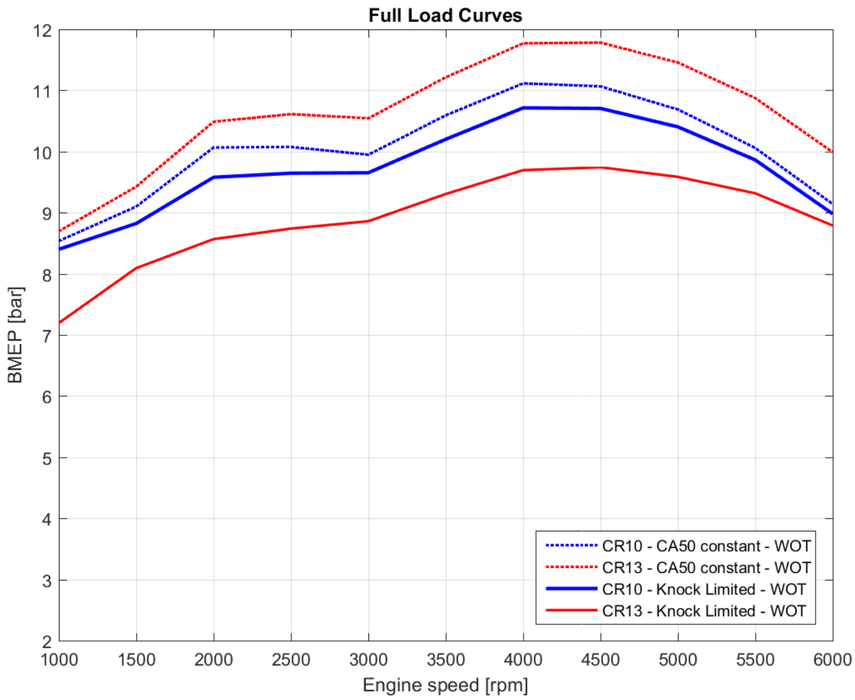


Figure 2.7: Full Load Curves - Comparison between fixed and variable CA50

As can be noticed the effects are stronger on the High Compression Ratio, especially at low engine speeds.

It should be remembered that when the comparison between the two fuel maps is made, it is the same engine operating at two different compression ratios. Also, the comparison is made at the same speed-load points, they are not different operating points from the perspective of vehicle operation. For the vehicle to move, it is the speed-load and hence power that matters, not the actual in-cylinder conditions. The difference in in-cylinder conditions matters separately, in a second stage, when the dedicated OEM team tries and decides what is a more efficient way to operate the engine. For the loads that cannot be reached at CR13, no comparison is made, the CR 10 data for higher loads should be used as the engine cannot operate there at CR 13.

It is worth to highlight that the engine simulated is Naturally Aspired, no turbocharging is included, for this reason the load achieved are pretty low with respect to standard values.

Considering all the disclaimers, the resulting fuel maps are reported in Figures 2.8 and 2.9.

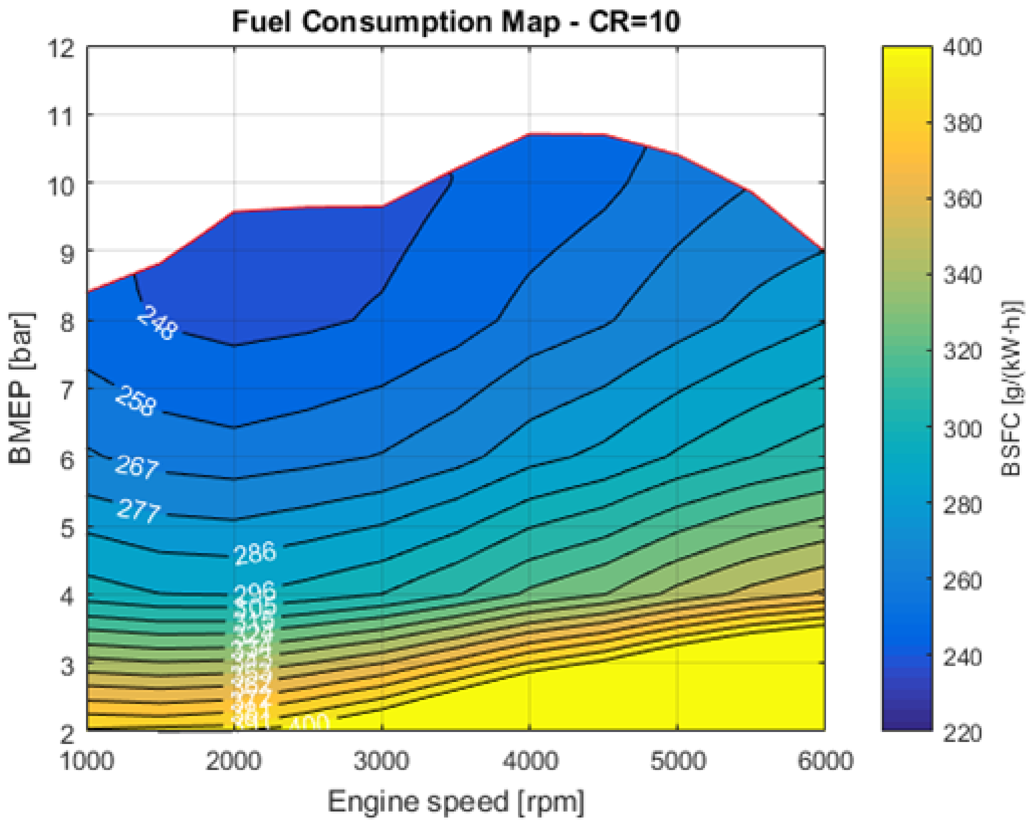


Figure 2.8: Brake Specific Fuel Consumption Map - CR10

The effects of the unrealistically increased spark retard can be spotted in the High Compression Ratio map: the best efficiency - low consumption - island is moved downwards, since at high loads the effect of knock and delayed combustion are experienced.

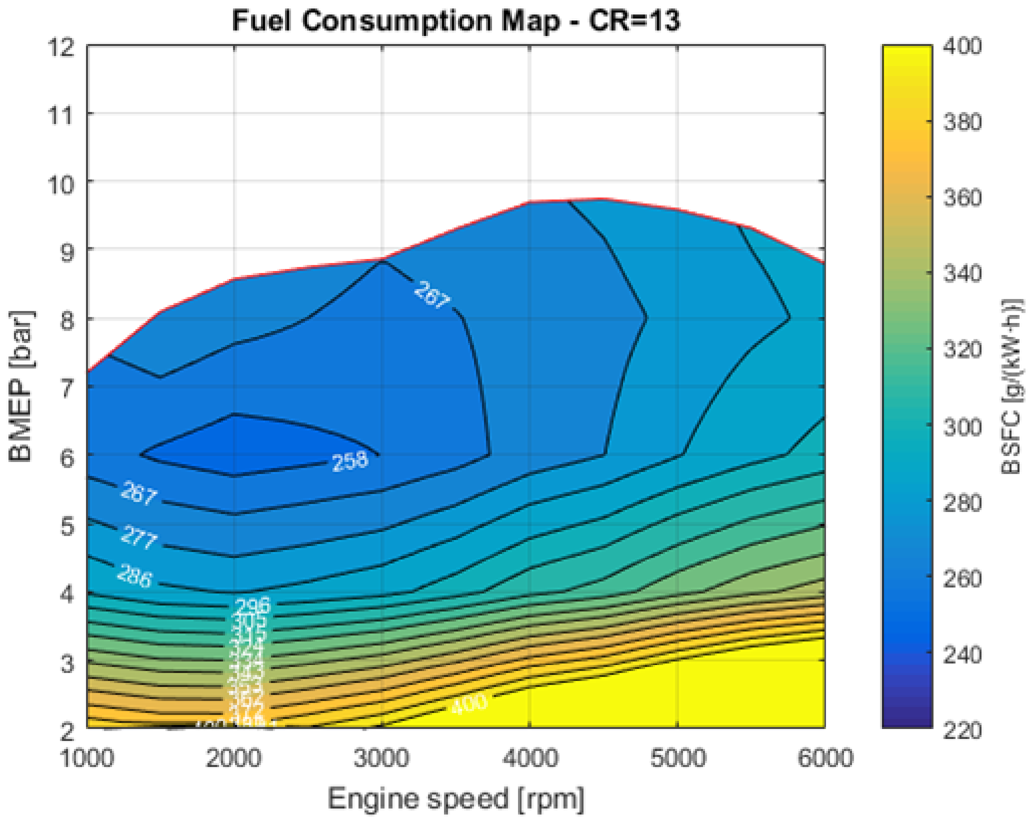


Figure 2.9: Brake Specific Fuel Consumption Map - CR13

2.3.1 Cylinder Indicated Efficiency

As far as the efficiency is concerned, VCR systems can show quite penalized values also at low loads for several reasons, as:

1. The higher Compression Ratio increases the losses, thermal, crevices (their volume increase) and mechanical (due to higher pressures, thus leading in a worse overall thermal efficiency;
2. Also in this condition it is required a spark timing adjustment in order to maintain an adequate airflow level;
3. Over-expanded exhaust gases are at a pressure below atmospheric one before Exhaust Valve Opening.

On the other hand, at high loads, where the low compression ratio is operating, there is the effect of delayed spark due to knock limit.

For these reasons, it is interesting to look at Indicated Thermal Efficiency of the cylinders of the GT-Power model. Results have been obtained in the GT-Post environment. A remark should be made: even taking in account all the considerations mentioned before, the maximum efficiency in the model case show an odd behavior at High Compression Ratio. Looking at the Cylinder Indicated Efficiency of CR=13 (Image 2.10), the maximum is reached in an isle around $6 bar BMEP$, then at a pretty low level. The explanation of such a trend lays again in the unrealistically high values of spark retard, making the combustion efficiency too low above 6bar BMEP.

At Low Compression Ratio instead, the map (Image 2.9) as a more reasonable shape.

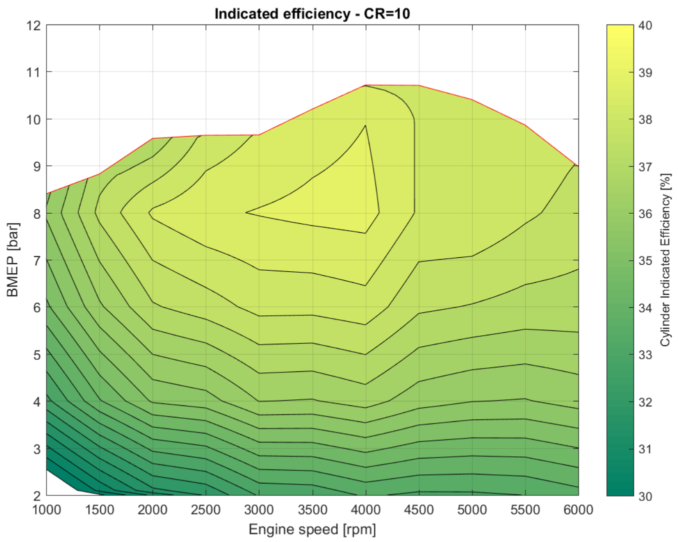


Figure 2.10: Cylinder Indicated Efficiency - CR10

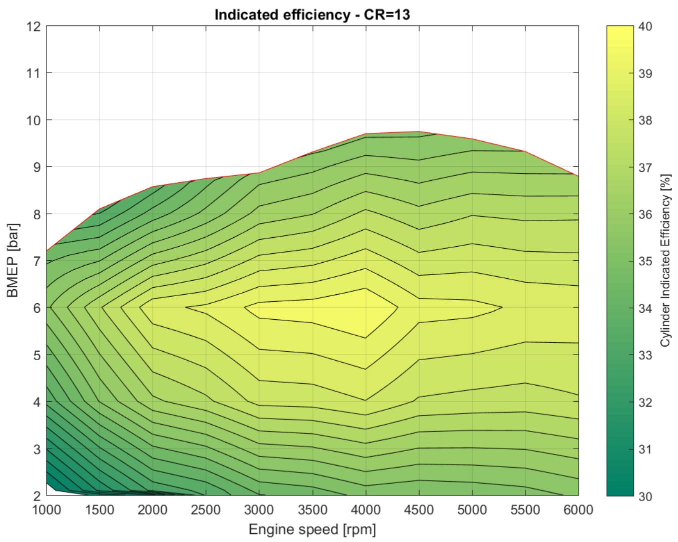


Figure 2.11: Cylinder Indicated Efficiency - CR13

Chapter 3

Introduction to the *Matlab* Code

Before turning attention to the transmission and switching time penalty models, a preamble on inputs, code requirements and general set-up is necessary.

As anticipated, main purpose of this thesis work is to write a versatile code that, given some inputs, operates as a "black box" giving as an output the percentage impact of switching time on fuel economy. It is worth to highlight that the benefit of a Variable Compression Ratio over a Fixed Compression Ratio it is undisputed, the comparison is needed at the level Instantaneous Switching versus Non-Instantaneous Switching.

The main characteristics desired in the code are:

- **Versatility**: the model should be parametric, meaning that should give reliable outputs varying the input. It wouldn't be fair to have a model constrained to a specific kind of inputs.

- **Non-predictivity**: meaning that, when a driving cycle is assigned, the model cannot control the switching logic knowing all the schedule. Even if among the inputs, the whole driving cycle is specified, all the computations should be made just considering the following point, coinciding with the pedal request instant per instant.

- **Computational speed**: the code should be able to give the result in a reasonable time. Hence should be written so to perform the computation for each point and cumulate the results in some minutes.

Such requirements have been taken in account during the work. After a first raw approach, a post-processing has been done so to ensure a dynamic and effective model.

3.1 Input Data

As far as the **inputs** are concerned, the data should be gathered in three excel files that the *Matlab* code reads. The first one should contain all the information on Fuel Maps, so a discrete number of points organized as engine speed, engine load and corresponding BSFC; Full Load Curve points should be included too.

The second file contains all the chosen vehicle characteristics. Concluding, the last file includes the points of the driving cycle. It is possible to organize the data in multiple sheets: once renamed, it will be possible to easily switch from one case to another through a dialogue box (as the one showed in Figure 3.1) in the *Matlab* code.

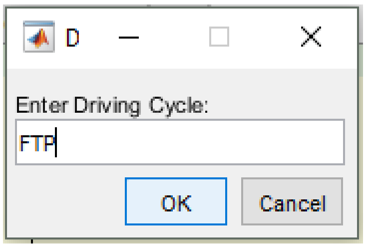


Figure 3.1: Example of *Matlab* code dialogue box

Additional parameters are then required when running the model. In order to have a clear view, all the inputs required are reported here after:

- To be uploaded in the excel file:
  - Fuel Maps for High Compression Ratio and Low Compression Ratio;
  - Vehicle Characteristics;
  - Driving Cycles different from testing procedure one (the standard have already been uploaded).
- Required when running the model through a dialogue box:
  - Engine cycles needed for switch compression ratio;
  - Select one of the driving cycles uploaded.

A remark should be made on how the Fuel Maps are handled in the code. Basically what *Matlab* does is to read a discrete number of points from the Excel File. This data set comes from GT-Power cases, but in general can have any source, both experimental on the test-grid and simulation.

Once the data set is acquired, there is an interpolation process: basically a function is created through the *scatteredInterpolant* command. To this latter is assigned a name, depending on if the High Compression Ratio or Low Compression Ratio Map is considered, and it will be used several times since it will provide a BSFC value for any point given as input.

A grid of point is then generated: engine speed goes from $800rpm$ to $6000rpm$ with a step of $500rpm$ (except for the first that is $200rpm$), while BMEP goes from $0bar$ to $12bar$ with a step of $0.1bar$. A dedicated, separate function, named *limit_ curve_ bmep* is adopted in order to delete from the mesh all the points above a line. Thus giving as an input the vector containing all the speed points, the vector containing all the load points and the coordinates of full load curve, the output is the grid limited to the WOT line.

The function capable of interpolating BSFC in every point is then used to compute the Fuel Map in the grid of points just generated. Again the process should be made two times, one for each compression ratio.

3.2 Switching Line

At this point it is necessary to compute the line separating the two operating areas of the compression ratios. Referring to the theory behind the VCR logic explained in Chapter 1, in the upper area will operate the Low Compression Ratio, while the High one in the bottom part.

The abovementioned curve is named Switching Line. In order to found it the grid of points below the Compression Ratio 13 Full Load Curve is considered. In other words both Fuel Maps are considered up to the WOT curve of high compression ratio, since above it the CR 13 cannot operate due to knock limitations and no comparison is needed.

Known the Fuel Consumption of both compression ratios in the same operating points the following difference is computed in each one of them:

$$\Delta BSFC = (BSFC)_{CR=13} - (BSFC)_{CR=10} \qquad \left[\frac{g}{kW \cdot h}\right] \tag{3.1}$$

Being the difference defined in this way it will mean that:

$$\begin{cases} \text{High CR is better,} & \text{if } \Delta BSFC < 0 \\ \text{Low CR is better,} & \text{if } \Delta BSFC > 0 \end{cases}$$

being the lowest fuel consumption among the two the target. Once defined the $\Delta BSFC$ in each point of the grid it will be necessary to find exactly the locus of points where such a difference is null.

$$\text{Switching Line} = \textit{locus}\,(\Delta BSFC = 0)$$

This process has been done through two *for loops* in *Matlab*: the first *for loop* iterates on engine speed, the second *for loop* iterates on the loads; they are following a hierarchy since one it is inside the other. An outline of the algorithm is reported hereafter:

- Fix an engine speed;
- For each couple of loads at the abovementioned speed check the $\Delta BSFC$;
- When the values of $\Delta BSFC$ have opposite signs save the two corresponding BMEP values. For sure the zero will be crossed at a load included in this interval;
- Through a weight average find the ordinate where null $\Delta BSFC$ is registered;
- Repeat the process for the following engine speed of the grid.

The Switching Line is defined connecting the discrete number of points resulting from this process. The comparison is graphically represented in Figure 3.2: being the reference to Equation 3.1, in red is represented the area where Low Compression Ratio is better, in green where High Compression Ratio gives a lower BSFC. Also the intensity of the difference is plotted.

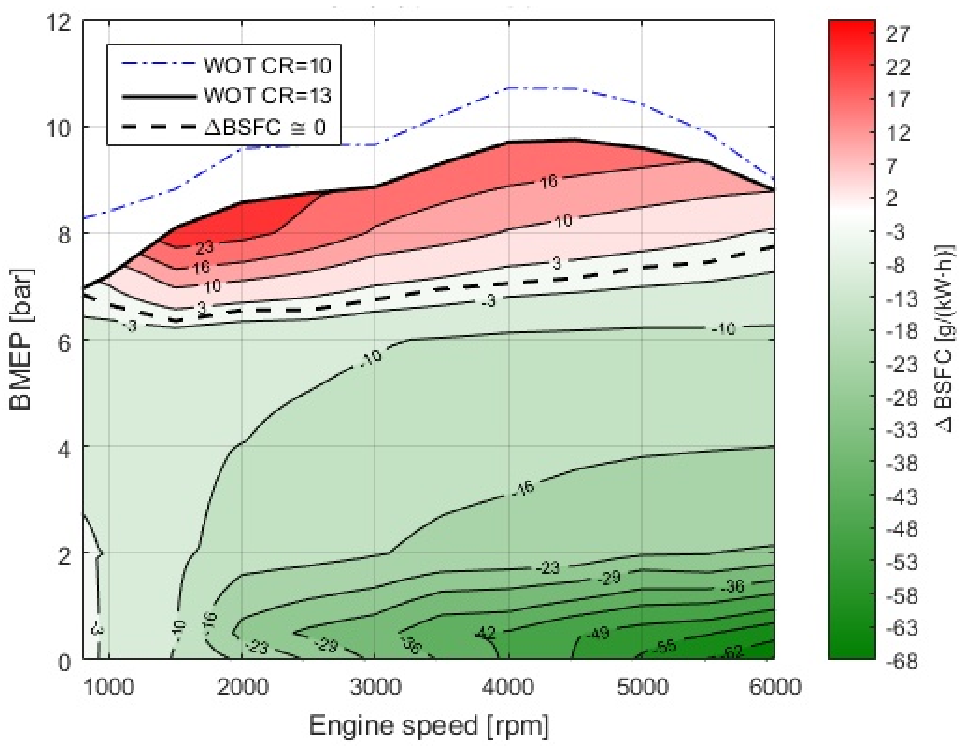


Figure 3.2: BSFC Difference

Once Defined the line it will be necessary to blend the two fuel maps, creating the unique map of the Variable Compression Engine. A process similar to the one done during the Fuel Maps import should be done, limiting this time the grid of CR 13 point to the Switching Line with the dedicated *limit_ curve_ bmep* function. The results is plotted in the Figure 3.3.

An important **remark** should be done at this point: there could be cases where, once the switching is started, the time available before the following pedal request is not sufficient to complete the switch. In this case, when a new condition is imposed, two scenarios can be present: or the switch can continue in the same direction or it will be necessary to come back. As a fact, when the pedal request occurs, the con-rod length is neither at Low Compression Ratio neither at the High condition position. Hence the switching line will, in reality, is not fixed, but it will be moving in transient condition. In other words the switching line plotted in Figure 3.3 is valid for the instantaneous switching; in the intermediate cases it is shifting. Concluding, the results provided in the following sections are still exact, simply the intermediate cases that can occur won't be plotted.

It is worth to highlight how up to this point have been provided information only about the engine. This means that, in order to get results about the Optimal Switching Line, a test rig is sufficient. The code it is completely still independent from vehicle or driving cycle, that will come into play shortly.

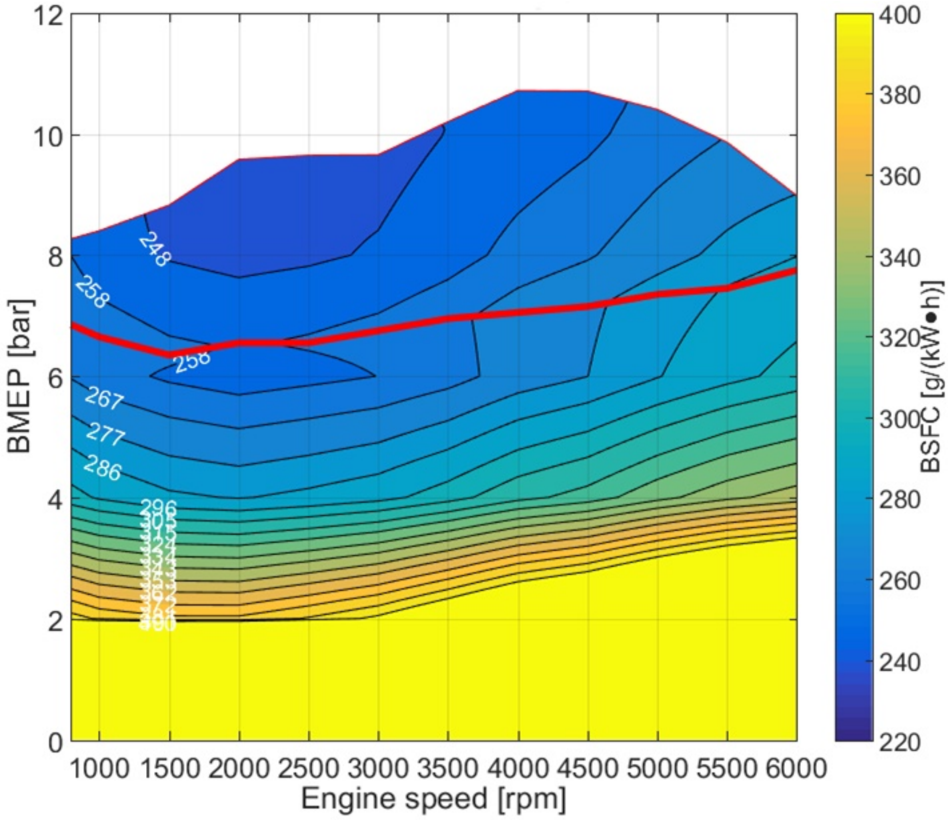


Figure 3.3: Blended Fuel Maps - VCR Engine Fuel Map

Chapter 4

Vehicle Choice

At this point it will be necessary to move from engine level to vehicle level. As anticipated several times, the *Matlab* Code should not depend on input, hence the choice it is completely arbitrary. However, in this particular case, an adjustment is necessary. It was not possible in fact to obtain fuel maps of a specific real engine, and so it has been necessary to resort to GT-Power to generate them: in order to obtain coherent results the vehicle picked should have at least the same displacement of the one simulated in GT-Power. If the data inserted are more precise and coming from experimental test this won't be necessary.

The vehicle and all its characteristics have been found in the EPA[1] database. More additional data as tire sizes and transmission ratios (for the manual transmission case, see chapter 7) have been taken from other sources.

All the data needed and used are reported in the following table:

Model Year	2018
Certifying Manufacturer Name	FCA US LLC
Certified Testgroup	JCRXJ01.45P0
Certified Evaporative Family	JCRXR0137PK0
Represented Test Vehicle Make	Jeep
Represented Test Vehicle Model	Renegade 4x4
Vehicle ID	V6BUJ8467
Vehicle Configuration #	0
Displacement (L)	1,4
Curb Weight (lbs.)	3214
Gross Veh. Weight Rating (lbs)	4586
Equivalent Test Weight (lbs.)	3500
Test Drive Code	F

[1] United States Environmental Protection Agency

Test Drive Description	2-Wheel Drive, Front
Transmission Type Code	M
Transmission Type Description	Manual
Transmission Type, If 'Other'	
Number of Gears	6
Transmission Lockup Y/N	N
Creeper Gear Y/N	N
Vehicle Fuel Category Code	
Vehicle Fuel Category Desc	
Hybrid Y/N	
Set Coefficient A (lbf)	10,22
Set Coefficient B (lbf/mph)	0,0236
Set Coefficient C (lbf/mph^2)	0,02516
Target Coefficient A (lbf)	27,39
Target Coefficient B (lbf/mph)	0,468
Target Coefficient C (lbf/mph^2)	0,02307
Test Number	JCRX10047314
Test Originator	MFR
Test Procedure Code	3
Test Procedure Description	HWFE
Test Fuel Code	46
Test Fuel Description	CARB LEV3 E10 RegularGasoline
Certification/In-Use Code	C
Vehicle Class Code	LDV
Vehicle Class Description	LDV/Passenger Car
Certification Region Code	FA
Certification Region Desc.	Federal
Emission Standard Level Code	T3B125
Emission Standard Level Desc	Federal Tier 3 Bin 125
Useful Life Miles (k)	120
Test Result/Emission Name	CREE
Rounded Emission Result (g/mi)	999

Certification Level (g/mi)	999
Emission Standard (g/mi)	999.9
Additive Deterioration Factor	0
Multiplc. Deterioration Factor	1
Uses NMOG/NMHC RATIO	N
NMOG/NMHC Ratio	
Upward Diesel Adjust. Factor	
Downward Diesel Adjust. Factor	
Reactivity Factor	
Carline Models Covered	FIAT: 500L
Wheel radius	351,5 (215/45R17 x 7.5 ET46-1.0 %)
Final drive ratio	4,334

Table 4.1: Jeep Renegade 2018 - EPA Vehicle characteristics

In addition to the particular attention that should be given to the displacement an additional remark is needed about the engine charging. As it can be seen the vehicle chosen has a turbocharged engine while the GT-Power model is Naturally Aspirated. This means that in reality the vehicle can reach operating points that are not included in the map generated with the simulation. In the following chapters will be explained how this problem has been handled, however it must be clear that the *Matlab* Model differs from the reality for this reason. Again the discrepancy won't show up if the fuel maps are from experimental data of the real engine.

Also the Coast Down coefficients cover an important role since they will be used to move from the driving cycle to the power demand at the engine.

Concluding it can be noticed how the model of the vehicle has a Manual transmission. Another exception is made in this respect since, for a first approach, a Continuous Variable Transmission is considered. The motivation and all the related information are analyzed in-depth in the following chapter.

4.1 Power request over the driving cycle

Once uploaded the vehicle data in the *Matlab* code, as a first check the power request over a driving cycle has been computed. As will be reported better in the following chapters, the power is computed through the coast down coefficients.

The logic has been developed so to consider each instant of the driving cycle separately: in other words to the code it is not allowed to know all operating points, but it computes the following one while operating, when operating at a given engine speed and load, the model computes the next operating point starting from the pedal (speed) request for the next one.

In this case an exception has been made, and all the driving cycle has been considered as known for the whole procedure. In each point the power request has been computed as:

$$a_{veh} = \frac{v_{veh,t_{i+1}} - v_{veh,t_i}}{t_{i+1} - t_i} \qquad \left[\frac{ft}{s^2}\right] \tag{4.1}$$

$$F_{res} = A + B \cdot v_{veh} + C \cdot v_{veh}^2 \qquad [lbf] \tag{4.2}$$

$$P_{wheels} = \frac{(F_{res} + a_{veh} \cdot M_{app}) * v_{veh,t_{i+1}}}{\eta \cdot 1000} \qquad [kW] \tag{4.3}$$

The results have been reported in Figure 4.1.

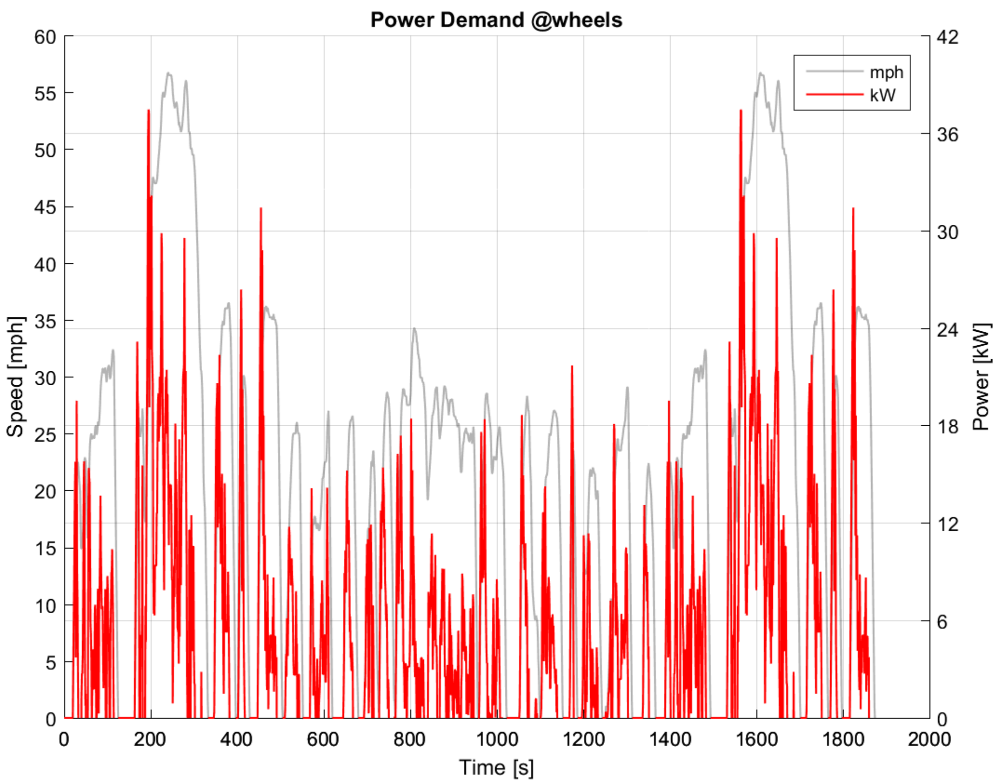


Figure 4.1: Power request at wheels over the FTP-75 driving cycle

The values obtained have been compared with the ones coming from a real dyno test performed on the vehicle in order to double-check the proper working of the *Matlab* model. However, these latter cannot be published for confidentiality reasons.

Chapter 5

Vehicle with Continuously Variable Transmission

As a first approach, the vehicle is considered with a Continuously Variable Transmission. This choice has been made in order to start from a simpler case allowing to properly develop the Switching Fuel Penalty logic. The transmission is in fact considered when it is needed to translate quantities measured at the wheels at engine level. Being the CVT efficiency considered close to 1, and being the choice of engine operating points more intuitive, this first approximation has been done.

CVT goes under the class of automatic transmissions and allows a seamless variation of transmission ratio between two limit values, generally corresponding to higher and lower gear ratios of equivalent manual transmission. The advantages of such a transmission lays in the fact that it allows the engine to work at an optimal speed, providing a substantial fuel consumption reduction. A different point of view can be considered: the CVT, given a power request, allows the engine to operate in the point in the map giving the same power but at the lowest Brake Specific Fuel Consumption.

The values of the Gear Ratios of the Jeep Renegade with manual transmission are reported in the Table 5.1.

Type	Manual
Gears	6
Final drive ratio	4,334
Gear shift #1 transmission ratio	4,15
Gear shift #2 transmission ratio	2,12
Gear shift #3 transmission ratio	1,36
Gear shift #4 transmission ratio	0,98
Gear shift #5 transmission ratio	0,76
Gear shift #6 transmission ratio	0,62

Table 5.1: Vehicle Transmission Specifications

As abovementioned, referring to this values, the two extremities of CVT operating area will be found by multiplying the final drive ratio times the minimum and maximum gear ratio. This procedure is not exact, since it is not referring to a real existing transmission. All the computation are made on basic engine principle basis and theory knowledge. Thus, in the CVT considered, the transmission ratio can oscillate continuously between:

$\tau_{max,(CVT)}$	17,9861
$\tau_{min,(CVT)}$	2,68708

Table 5.2: Equivalent CVT limit values

By plotting the comparative effectiveness of vehicle advancement using the CVT transmission, it is possible to spot the CVT operating area between the two lines, corresponding respectively to the values reported in Table 5.2.

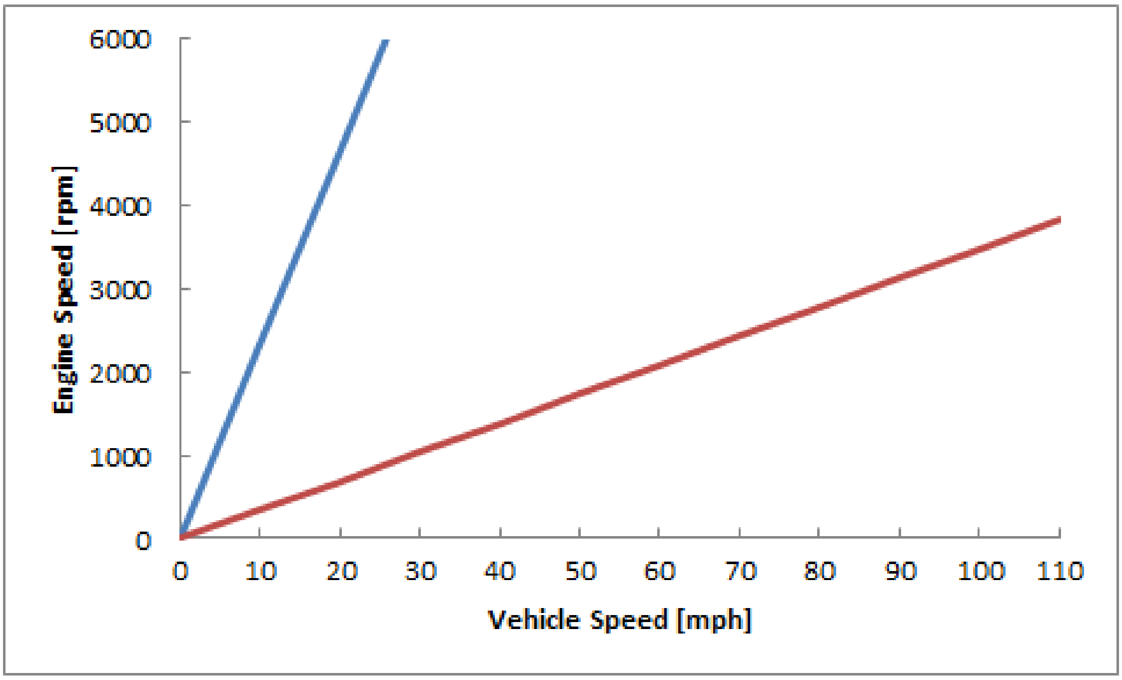


Figure 5.1: Speeds diagram for CVT

In general, whatever engine speed is present in the idle, left zone, it is reconduced to a constant value in the working area.

The engine speed won't be equal in all the operating points of the map obviously, but it will follow the trend of the **Optimal Operating Line**. This latter is the line connecting, increasing the power, all the point with minimum BSFC. CVT operates so that all to operating points are close to the abovementioned locus.

Assumption

Being just a first approach, an additional simplification on transmission efficiency $\eta = 100\%$ has been done. By the way, being the code parametric, if different values of efficiency are required, it will be necessary a change in the parameters initialization part at the beginning of the code.

5.1 Procedure

When all the inputs have been set properly it is possible to go through the main part of the code. Being non-predictivity required, all the passages that will be analyzed now will be performed for each driving cycle step, or better for each interval between one pedal request and the following. Assuming that the vehicle is considered in an instant t_i, the computation start when there is a pedal request for the instant t_{i+1}.

In first place, when the driver push the pedal, the information is given to the code as: a vehicle speed must be reached by a defined time interval. Through this information and coast down coefficients, it is possible to compute the power request at the wheels:

$$a_{veh} = \frac{v_{veh,t_{i+1}} - v_{veh,t_i}}{t_{i+1} - t_i} \qquad \left[\frac{ft}{s^2}\right] \tag{5.1}$$

$$F_{res} = A + B \cdot v_{veh} + C \cdot v_{veh}^2 \qquad [lbf] \tag{5.2}$$

$$P_{wheels} = \frac{(F_{res} + a_{veh} \cdot M_{app}) * v_{veh,t_{i+1}}}{\eta \cdot 1000} \qquad [kW] \tag{5.3}$$

where:

$$\begin{aligned}
a_{veh} &= \text{Vehicle acceleration} \\
v_{veh} &= \text{Vehicle speed} \\
F_{res} &= \text{Sum of resistance forces acting on the vehicle} \\
A, B, C &= \text{Coast down coefficients} \\
P &= \text{Power demand} \\
M_{app} &= 1,03 \cdot M_e, \text{ where } M_e \text{ is the equivalent test weight [lbs]} \\
\eta &= \text{Transmission Efficiency}
\end{aligned}$$

As a consequence of the assumption made on efficiency, if the CVT is considered ideal then power at wheels level will be equal to the one at engine level:

$$P_{engine} = P_{wheels} \tag{5.4}$$

Now the engine **operating point** must be found. The general expression used in order to move from vehicle speed to engine speed is:

$$enginespeed = \frac{v_{veh} \cdot 60}{2\pi \cdot r_{wheel}} \cdot \tau_{final} \cdot \tau_{gear} \qquad [rpm] \tag{5.5}$$

But in the CVT the transmission ratio is not fixed, so it is not known. The procedure has been developed then in the following way:

- An average fixed value has been imposed for τ;
- A dummy couple of engine speed and load giving the power request is found;

- The dummy operating point, together with the engine displacement, is given as an input to a dedicated created function called *cp_line*. The output of this latter is a vector containing all the other set of operating points in the map giving the same power.

5.1.1 Constant Power Lines

The target at this point is locating in the map the correspondent Constant Power Line. This computation is perofrmed in the following way:

- Two vectors are considered: in one are included the engine speed going from $800rpm$ to $6000rpm$ with a step of $10rpm$; in the other the engine loads going from $0bar$ to $12bar$ with a step of $0.1rpm$

- For each combination of these points the power is computed as:

$$P = \frac{[rpm] \cdot [BMEP] \cdot 1200}{V_{tot}} \tag{5.6}$$

- This value is compared with the one given as an input to the function: if the value computed is equal to the reference one **within an error of 3%**, then the corresponding engine speed and load are saved;

- All the combinations giving, approximatively, a power level equal to input ones are transmitted as outputs.

In order to check if the code was properly working a set of power levels, going from $4kW$ to $40kW$ with a step of $2kW$, has been given as input and constant power lines obtained are reported in Figure 5.2.

It is worth to notice that this procedure does not depend on the fuel maps: the constant power lines have been found just knowing the engine displacemet.

For each step of the driving cycle a specific line should be found and extracted. In order to reduce computation times, as a trial, a *structure* in matlab containing a discrete number of power levels and correspondin operating points arleady attached has been created. In this way the code, instead of computing, had simply to search the desired power. But, being the number of levels required too high, and being the computation heavy too, this latter method has resulted slower than the one explained.

5.1.2 Optimal Operating Point

Once the set of possible combination of engine speeds and loads giving the desired power is found, it will be necessary to pick one of them, generally chosing the one with minimum fuel consumption.

In this case, however, the CVT is connected to a VCR engine, thus the optimal operating point will be the one giving the minimum BSFC among the two compression ratios.

The optimal operating point can be found by:

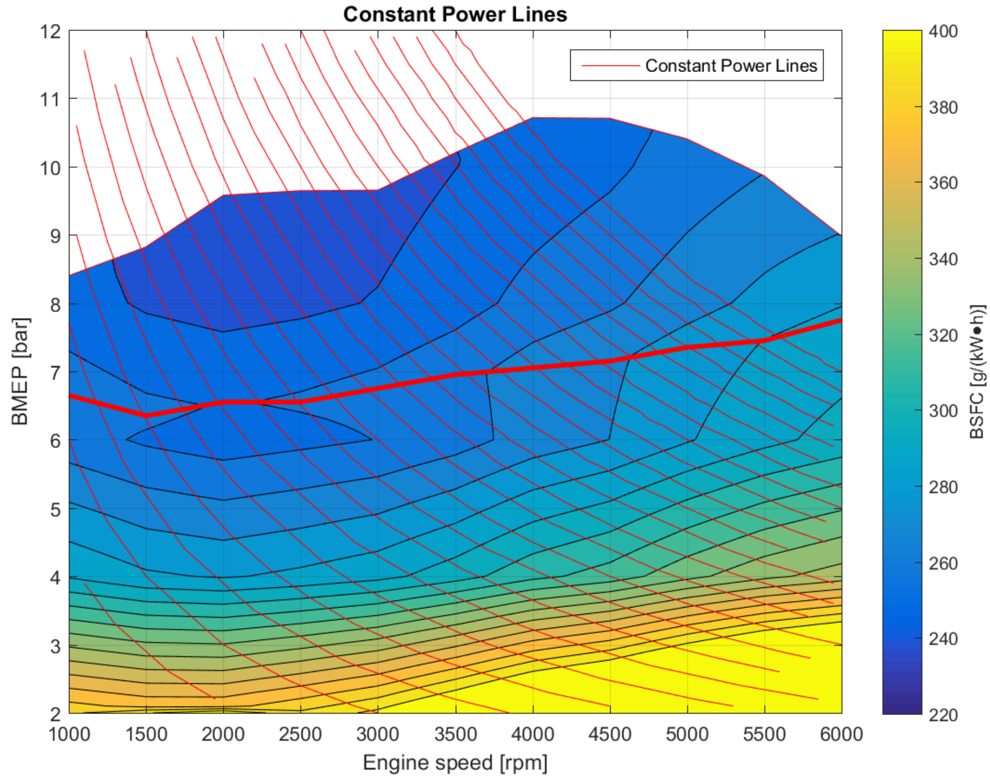


Figure 5.2: Constant Power Lines example

- Considering the points of the Constant Power Line only up to Low Compression Ratio WOT curve:
  - Interpolate and compute in these points the BSFC (CR=10 Map);
  - Find the point with minimum BSFC at Low Compression Ratio;
- Considering the points of the Constant Power Line only up to High Compression Ratio WOT curve:
  - Interpolate and compute in these points the BSFC (CR=13 Map);
  - Find the point with minimum BSFC at High Compression Ratio;
- Compare the two points and find the absolute minimum.

If this procedure is done for the set of Constant Power Lines used for the example showed in Figure 5.2, several Optimal Operating Points will be obtained. When connecting them a Optimal Operating Line (OOL) is obtained, and it will represent the locus of points with lowest BSFC. The result is reported in Figure 5.3.

As can be seen the line is discontinuous in some areas, due to the fact that Constant Power Lines have been found as a spline between a discrete number of points: the BSFC is evaluated just in the coordinates provided in the vector, the curve is not defined as a continuous function.

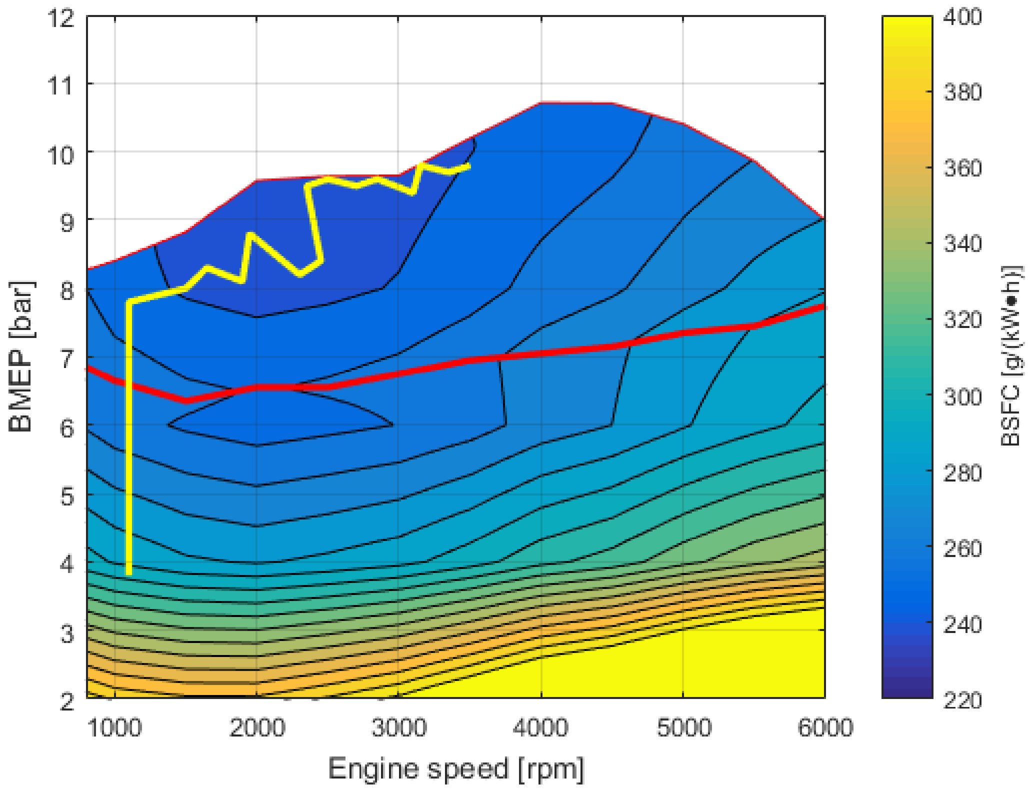


Figure 5.3: CVT Optimal Operating Line

An additional important remark on **CVT Minimum Speed** should be made. Among the parameters to be set in the code initialization section, also this value should be specified. Basically, for reasons related to CVT working principles the transmission do not allow engine speeds below a certain threshold. In this case the CVT minimum speed is **1100rpm**. If this is imposed as a constraint, the code won't consider in the comparison a point with engine speed lower than the minimum.

As a check, when the algorithm is applied to the driving cycle, all the operating points should be in the neighborood of the OOL, otherwise something is not correct.

5.1.3 Need of Compression Ratio Switching

Almost all the information about the next operating point are known. The only data missing is how much fuel is required to perform this little step of the cycle, that is also the most important since at the end the total fuel consumption is needed.

It is at this at this point that Switching Logic should be taken in account: considering the Compression Ratio at which the engine is working at instant t_i and Compression Ratio giving the minimum BSFC for next point, the code take in account if switch is needed in the interval.

If the two CRs are equal, no switch is needed and the quantity of fuel injected can be computed as:

$$m_{fuel} = \frac{BSFC_{min} \cdot P_{engine} \cdot \Delta t}{3600} \qquad [g] \tag{5.7}$$

Otherwise a separate function named *switch_logic* should be recalled and will give mass of fuel injected considering that in the meantime compression ratio is changed. In this latter case the fuel economy deterioration with switching is accounted. It is explained in detail in Chapter 6.

5.1.4 Results arrangement

Inside the code all the information about each point of the driving cycle are collected in a matrix were each row corresponds to a time instant, while each column contain the following quantities:

- time $[s]$
- engine speed $[rpm]$
- engine load $[bar]$
- engine power $[kW]$
- torque $[Nm]$
- BSFC $[g/kWh]$
- fuel injected between that instant and the previous (m_fuel) $[g]$
- optimal compression ratio
- actual compression ratio

After that the first row is initialized with idle quantities, this matrix is also used to provide data during computation about the operating point preceding the one considered.

In case of instantaneous switching (engine cycles needed for switch = 0), if the Federal Test Procedure is considered as reference cycle, the operating points of the engine are reported in Figure 5.4

The points showed are not sequential, it will be necessary to cross the switching line several times. As a confirmation of the hypothesis on CVT, all the operating points are in the neighboroud of the Optimal Operating Line.

In the low load zone, below 2 bar BMEP, there is a set of out trending points: majority of them have almost null power, so won't weight on the fuel consumption.

The results on fuel consumption, and related impact of switching, will be reported in the following chapter, after that a clear explanation of what happens when compression ratio is changing is given.

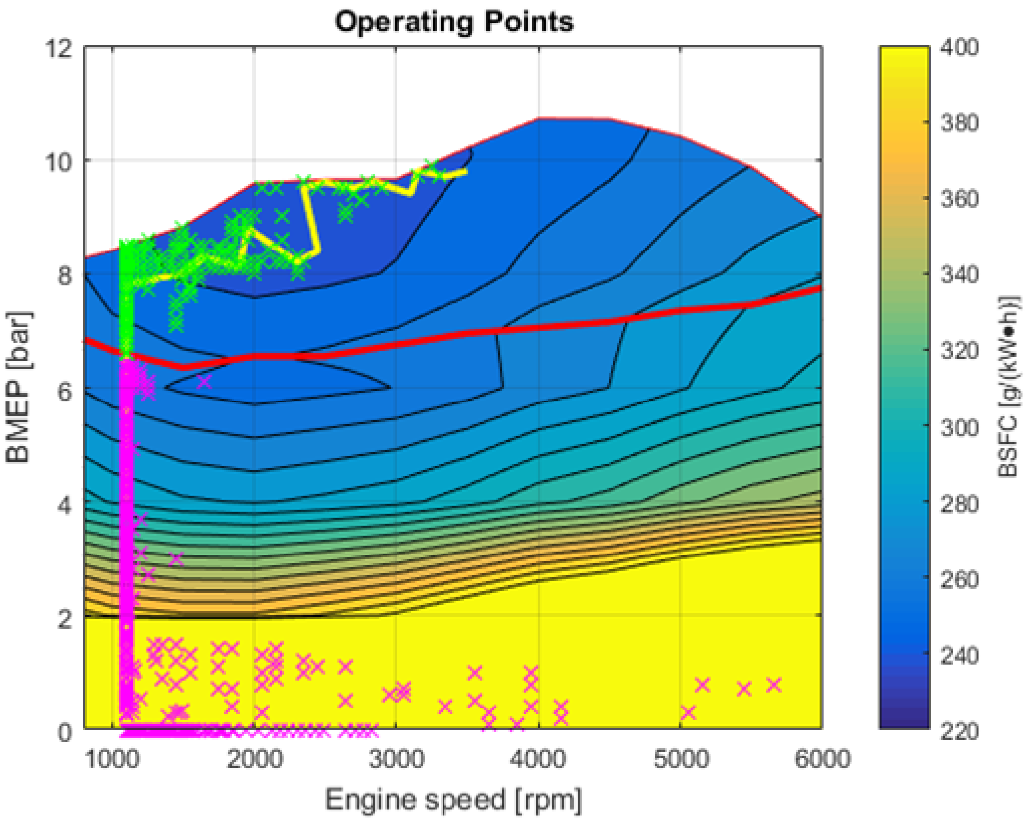


Figure 5.4: Operating Points on FTP cycle with CVT

Chapter 6

Switching Logic - Fuel Economy

In this chapter will be explained the core function of the *Matlab* model, since it is the one estimating the impact of switching time on fuel economy. This part is equal for the CVT and the Manual Transmission, that will be analyzed after. Some results coming from CVT will be used in order to understand the various stage of this function.

The basic operation performed is the following: it is known that a switch from high-to-low or from low-to-high is needed within a certain time interval. Since the fuel maps are known, also the BSFC at starting compression ratio and at target optimal operating point are known. The only exception is that in this case the switching is not instantaneous, so there will be a lot of intermediate points where the compression ratio is neither low or high and where BSFC are different both from extremities ones.

This problem has been dealt through two simultaneous discretization processes:

1. Discretization in time: the interval of time, available from the instant in which the engine is operating and the one in which is required to perform the pedal request, is subdivided in steps.

2. Linear Discretization of BSFC: Knowing both values of Fuel Consumption, in the two time instants, the difference is subdivided in steps computed through the switching time. Depending on the engine cycles needed for switch, it can take more or less steps to complete the switching.

Hence each one of these discretization steps corresponds to a little variation of the con-rod length, that is assuming different compression ratios in-between the two limit values. In every discretization step the BSFC and then mass of fuel injected are computed, then leading to the total mass of fuel injected in the time interval. A qualitative outline of the process is reported in Figure 6.1.

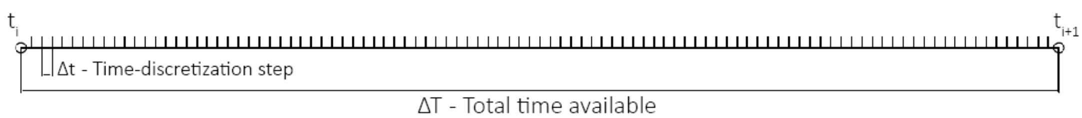


Figure 6.1: Time Discretization in Switch Penalty Function

Being the attention focused on the difference between instantaneous (ideal) and non-instantaneous (real) switch, several cases have been taken in account. In particular, this function takes, among others, the time needed for switch as input: if this latter is nil then it means that the ideal case is considered. On the other hand a set of switching times, expressed in engine cycles, has been considered so to obtain a trend with slower switching as result. In particular the cases considered are reported in Table 6.1.

Engine cycles needed for switch	5	10	15	20	25	30	35	40	50	100

Table 6.1: Different switching time cases

6.1 Function Input and Output

The algorithm used is pretty complex, and it must account for a lot of scenarios that can occur when switching.

In particular main concern is about the switching duration: if the switch is too long and cannot be completed, enough infortmation should be exchanged with the code. In first place because if the target point is not reached the engine load and speed and power won't be the ones plotted but intermediate ones. Hence when computing the total fuel consumption, the correct level reached in each point should be considered.

In order to perform properly all the computations these are all the information needed and then computed by the function:

- INPUT
  1. $[rpm_i]$ — Engine speed at which switch starts;
  2. $[BSFC_0{}^*]$ — Brake specific fuel consumption of the Operating Point at t_i $[g/kWh]$;
  3. $[BSFC_1{}^*]$ — Brake specific fuel consumption of the Operating Point at t_{i+1} $[g/kWh]$;
  4. $[P_d]$ — Power request $[kW]$;
  5. $[c_s]$ — Engine cycles needed for a complete CR switch $[enginecycles]$;
  6. $[\Delta T]$ — Driving Cycle step, time interval $(t_{i+1} - t_i)$ $[s]$;
  7. $[c_r]$ — In the case the previous switch has not been completed how many cycles are left $[enginecycles]$;
  8. $[CR_0]$ — Actual compression ratio at t_i;
  9. $[CR_0{}^*]$ — Optimal compression ratio at t_i;
  10. $[CR_1{}^*]$ — Optimal compression ratio at t_{i+1};
- OUTPUT
  1. $[m_f]$ — Mass of fuel injected in the interval $[g]$;

2. $[CR_1]$ — Intermediate CR in the case the switch is not completed;
3. $[BSFC_1]$ — Intermediate BSFC in the case the switch is not completed $[g/kWh]$;
4. $[c'_r]$ — Cycles needed to complete the switching $[enginecycles]$;

6.2 Initialization

Before going to the effective fuel consumption computation, the function needs to understand in which scenario the mechanism is working. In first place it is necessary a coherence among the quantities compared and involved. It is possible in fact to express all the processes in seconds, engine cycles or discretization steps. There are trivial relations connecting each one of them, the procedure followed is reported hereafter. In first place the engine cycles needed for switch are expressed, through the engine speed, in seconds:

- One engine cycle duration:

$$t_0 = \frac{60}{rpm_i} \qquad [{}^{s}/_{cycles}] \tag{6.1}$$

- Complete switch duration:

$$T_s = c_s \cdot t_0 \qquad [s] \tag{6.2}$$

Where adjective *complete*, referred to switch, has been expressly specified since it is not said that the process can be completed in the time interval.

At the same time it is necessary to explicit both time interval and BSFC gap as a set of values. In other words, two mutually connected linear discretization operations are needed: they are performed by firstly subdividing the time available in steps of Δt and then using the resulting quantity for the ΔBSFC.

As far as the choice of Δt is concerned, a tradeoff is needed: the value cannot be too high since at high engine speeds the compression ratio switching is very fast, and the step should be small enough to appreciate it. On the other hand, if the value chosen is too small, the requirements on high computational speed fall. In this function a default value has been assigned:

$$\Delta t = 0,01 \quad [s]$$

Consequently:

$$steps_t = \frac{T_s}{\Delta t} \tag{6.3}$$

At this point, the controller needs to do some checks on the effective time available and on the status of previous switch before moving to BSFC discretization.

Known the time available and the switch duratin four scenarios are possible:

(a) Previous switch has been completed, ΔT is sufficient to complete the current switching;

(b) Previous switch has been completed, ΔT is not sufficient to complete the current switching;

(c) Previous switch has not been completed, ΔT is sufficient to complete the current switching, in both directions. In fact, if the con-rod length is in an intermediate position, it can occurr that, depending on the pedal request:

- it is necessary to go back to starting compression ratio;
- it is possible to continue switching in the same direction.

(d) Previous switch has not been completed, ΔT is not sufficient to complete the current switching. This latter case is improbable but must be taken in account for completeness reasons. Again the possibilities mentioned in (c) should be considered.

As a consequence of discretization processes, a stair behavior is obtained: to each time step Δt corresponds an increment or decrement of BSFC. Hence in general:

$$BSFC_j = BSFC_0 + BSFC_i \qquad \text{for} j \in [0; step_t] \tag{6.4}$$

Now for each scenario the logic is explained. Obviously, increasing the number of engine cycles needed for swich it is easy to move from one case to the others.

Scenario (a)

In this case at each one of the discretization time instants an intermediate BSFC is registered; the increment (decrement), depending on the sign, has been computed as:

$$BSFC_i = \frac{BSFC_1{}^* - BSFC_0{}^*}{steps_t} \qquad [g/kW \cdot h] \tag{6.5}$$

What matters are the starting and final value of BSFC: both are the optimal ones. The two discretization processes result in a step trend from when switching start to end point.

Scenario (b)

When the time interval available is too short, the mechanism, driven by gas and inertia forces, won't be able to reach the target length. What happens in reality at this scenario, is that, being the compression ratio different from optimal one, an operating point different from desired one is reached. Then engine speed and load will slightly differ from computed ones and consequently the power. For this reason has been previously mentioned that the operating point map is plotted for ideal, instantaneous switching, case.

However, in order to obtain a functional and coherent model, the intermediate points, even if not represented in the map (it wouldn't be possible to appreciate the difference), are taken in account in the computations. For this reason, in addition to the fuel consumption in the interval, intermediate compression ratio, BSFC and cycle remaining to complete the switching are given as output. These latter three parameters are left unchanged to default settings if the switch is completed; otherwise some values are attributed and used as input for the subsequent time interval.

Being the switching time fixed the BSFC increment or decrement steps will be again computed as:

$$BSFC_i = \frac{BSFC_1{}^* - BSFC_0{}^*}{steps_t} \qquad [g/kW \cdot h] \tag{6.6}$$

with the exception that $CR_1{}^*$, and consequently $BSFC_1{}^*$, cannot be reached. Instead two intermediate values CR_1 and $BSFC_1$ are present at the end of time interval. This latter will be the starting point for next discretization.

$$\begin{cases} CR_{0,t}|_{\Delta T_{i+1}} = CR_{1,t+1}|_{\Delta T_i} \\ BSFC_{0,t}|_{\Delta T_{i+1}} = BSFC_{1,t+1}|_{\Delta T_i} \end{cases} \tag{6.7}$$

In addition also the number of engine cycles is saved, the reasons will be explained shortly.

Scenario (c)

If the previous switch, being too slow, i.e. at low engine speeds, is not completed, the initial $BSFC_0 \neq BSFC_0{}^*$ for the function won't be the optimal one. For this reason, when the mechanism has not completed the length variation, it is necessary to save the intermediate compression ratio $CR_t = CR_1 \neq CR_1{}^*$ reached and corresponding BSFC and include it among outputs. Then, once ascertained that the switch can be completed, another check is needed:

- if $CR_t{}^* = CR_{t+1}{}^*$ it means that the switch can continue in the same direction, and the steps will be computed as:
  - Additional time needed to complete the switch

$$T_r = c_r \cdot t_0 \qquad [s] \tag{6.8}$$

  - Additional steps needed to complete the switch

$$steps_r = \frac{T_r}{\Delta t} \tag{6.9}$$

  - BSFC increment (decrement) value

$$BSFC_i = \frac{BSFC_1{}^* - BSFC_0}{steps_r} \qquad [g/kW \cdot h] \tag{6.10}$$

- if ${CR_t}^* \neq {CR_{t+1}}^*$ it means that the mechanism has to go back to starting position. Then:
  - Engine cycles for switching already performed in previous time interval:
$$c_b = c_s - c_r \qquad [enginecycles] \tag{6.11}$$
  - Time needed to go back to initial con-rod length
$$T_b = c_b \cdot t_0 \qquad [s] \tag{6.12}$$
  - Steps needed
$$steps_b = \frac{T_b}{\Delta t} \tag{6.13}$$
  - BSFC increment (decrement) value
$$BSFC_i = \frac{{BSFC_1}^* - BSFC_0}{steps_b} \qquad [g/kW \cdot h] \tag{6.14}$$

It is worth to highlight the dependency from t_0: if the engine speed is changing from one interval to the other, all the switching durations will change too.

Scenario (d)

This latter case is a combination of scenarios (b) and (c), all the considerations made in this respect should be taken in account.

As anticipated, it is improbable to operate the mechanism in such conditions but, being the analysis conducted up to 100 engine cycles needed to switch, it had to be taken in account.

6.3 Fuel consumption

At this point the computation is focused on the further smaller discretization time steps Δt. For each one of them the mass of fuel injected during thw switch can be computed, referring to 6.4, as:

$$m_{f,j} = \frac{BSFC_j \cdot P_d \cdot \Delta t}{3600} \qquad [g] \qquad \text{for} j \in [0; step_t] \tag{6.15}$$

Then, in order to obtain the output fuel consumption in the time interval ΔT, the sum of all contribution should be computed. This process, though, is not trivial. During the analysis several attempts have been necessary before getting to the procedure providing the correct results. The main problem was encountered in the definition of the instant where switching starts.

In the following sections a detailed analysis of the attempts is reported, lingering on all the reasoning behind. For a clearer explanation reference to CVT results will be made.

6.3.1 Attempt 1 - Switch performed as soon as possible

In the first development stage, the switching logic function has been inserted in the CVT model code, in order to evaluate the impact of switching time on fuel economy. No particular attention was given to the starting and target BSFC values. Simply, in the cases where the switching was needed, the quantity of fuel injected in the time interval was computed by summing all the contributions coming from the time discretization steps. By defining:

$$\begin{cases} T_0 = t_i & \text{Instant where switching starts;} \\ T_s = T_0 + \Delta t \cdot steps_t & \text{Instant where switching ends;} \\ T_1 = T_0 + \Delta T = t_{i+1} & \text{Instant where pedal request must be fulfilled;} \end{cases}$$

The fuel consumption in the time interval is:

$$m_f = \left(\sum_{k=0}^{steps_t} (BSFC_0{}^* + k \cdot BSFC_i) \cdot \Delta t + BSFC_1{}^* (T_{end} - Ts) \right) \cdot \frac{P_d}{3600} \qquad [g] \tag{6.16}$$

In case of instantaneous switch the number of steps to change the con-rod length is one, hence the CR after 0.01s changes value. In this case the fuel consumption is:

$$m_f = (BSFC_0{}^* \cdot \Delta t + BSFC_1{}^* (\Delta T - \Delta t)) \cdot \frac{P_d}{3600} \qquad [g] \tag{6.17}$$

For a better understanding the situation is plotted in the Figure 6.2.

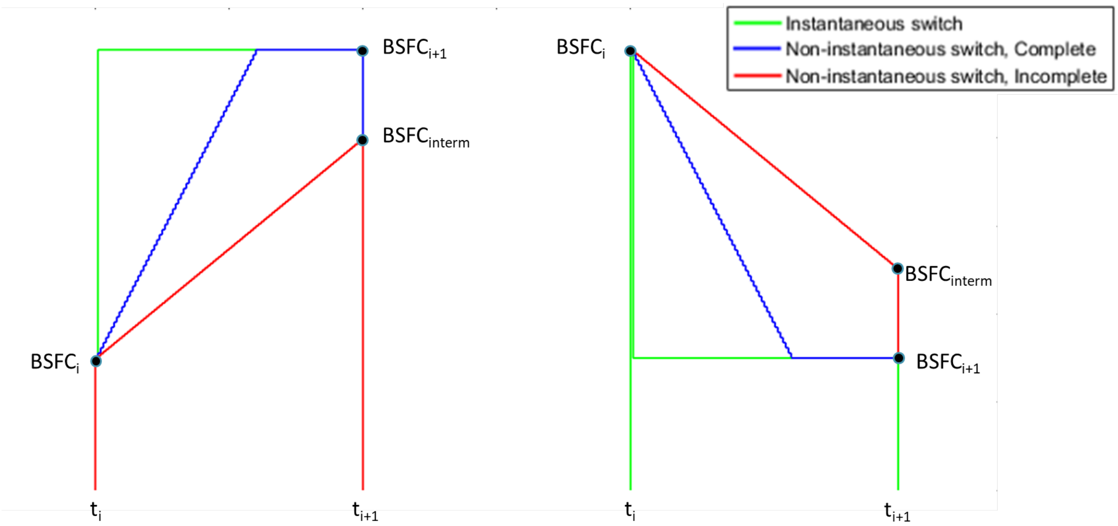


Figure 6.2: Logic Layout - Attempt 1

Application to CVT

When inserting the switching logic function inside the main CVT code, it is possible to compute the overall fuel consumption on the cycle by accounting each contribution of time intervals between one pedal request and the other.

In this way it is possible to compute both the instantaneous fuel economy of the vehicle and the average fuel economy on the driving cycle. In first place two vectors are defined:

1. **Mass of fuel injected** Quantity of fuel injected in each second of the driving cycle;

2. **Cumulative fuel consumption** Vector obtained summing to each element of mass of fuel injected vector the previous one:

$$m_{fuel,cum}|_i = m_{fuel}|_i + m_{fuel,cum}|_{i-1} \qquad [g] \tag{6.18}$$

3. **Instantaneous fuel economy** Each element represent the fuel economy assuming that that consumption is fixed for the cycle. The elements are computed as:

$$V_{fuel,i} = \frac{\dot{m}_{f,i} \cdot T_{dc}}{1000 \cdot \rho_{fuel}} \qquad [gal] \tag{6.19}$$

$$FE_{inst} = \frac{d_{dc}}{V_{fuel}} \qquad [mpg] \tag{6.20}$$

   where T_{dc} and d_{dc} represent, respectively, the total driving cycle duration and distance covered.

At the sime time, by summing all the elements of the cumulative fuel consumption vector, a value $V_{fuel,tot}$ can be obtained, representing the total amount of fuel needed to go through the driving cycle. Consequently the average fuel economy on the driving cycle can be computed as:

$$FE_{av} = \frac{d_{dc}}{V_{fuel,tot}} \qquad [mpg] \tag{6.21}$$

By plotting in the same graph both the instantaneous fuel economy vector, varying with time, and the average value the Figure 6.3 is obtained for the **instantaneous switching** case.

The same plot can be obtained for different values of switching time. If a difference is made between each one of them and the instantaneous case the Figure 6.4 is obtained.

$$\Delta FE_{inst} = FE_{inst}|_{inst.switching} - FE_{inst}|_{non-inst.switching} \tag{6.22}$$

For the instantaneous switching case it is expected the higher fuel economy and, consequently, the lower fuel consumption, since there is no fuel penalty associated with con-rod length variation. Hence, by defining the difference as in Equation 6.22, only positive values of the delta should be present. It is clear that something is

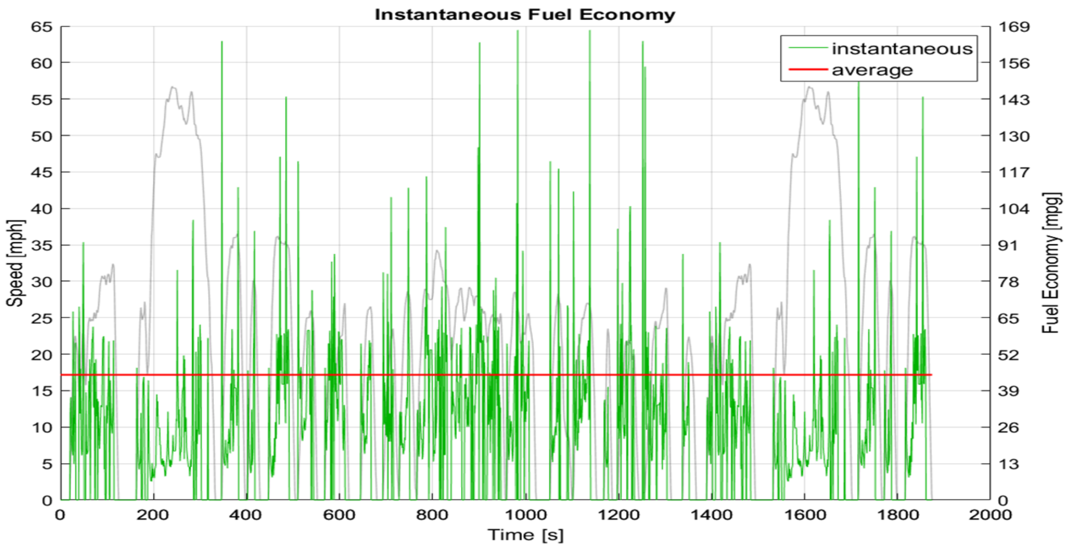


Figure 6.3: Instantaneous and Average Fuel Economy over FTP cycle - CVT - Instantaneous switching

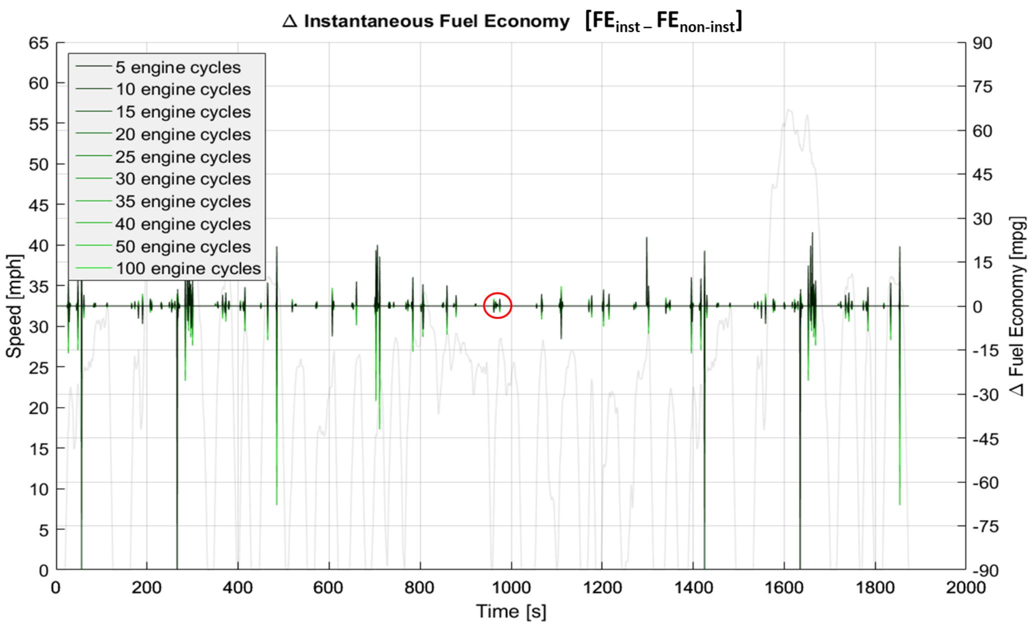


Figure 6.4: Instantaneous Fuel Economy Difference between Instantaneous and Non-Instantaneous cases

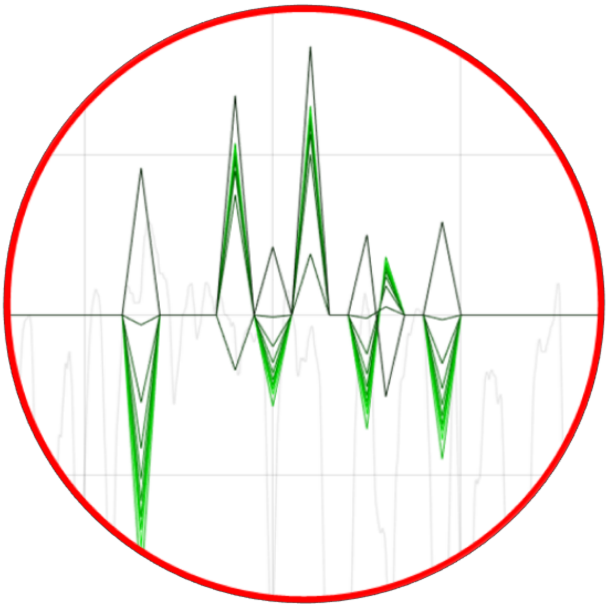


Figure 6.5: Instantaneous Fuel Economy Difference between Instantaneous and Non-Instantaneous cases - Zoom

not working since the non-instantaneous switch show higher values of fuel economy then the instantaneous one in some cases.

The reason of this behavior can be found looking at Figure 6.2: if the fuel consumption is represented by the area under the lines representing the switching, it can be noticed how when the $BSFC_{t+1}$ is higher than the $BSFC_t$, the area corresponding to instantaneous switch is higher compared with the non-instantaneous one. In other words, it is like the engine is moving to an higher consumption operating point as fast as possible.

It is worth to highlight that the starting and target BSFC points are fixed, prescribed by the CVT logic, hence they do not depend on the VCR mechanism. The changing in CR makes possible that the BSFC is the lower between the two possibilities, however both of them can be higher than the starting one.

If the instantaneous BSFC over the driving cycle is plotted against time, in the three cases of Compression Ratio 10 only, Compression Ratio 13 only and Variable Compression Ratio Figure 6.6 is obtained. Being the discretization steps of 0.01s and the overall driving cycle duration of about 2000s, it is not possible to aprreciate a lot from this graph. Hence a zoom on a time interval of about 5s is made (Figure 6.7).

Here it is possible to check how the variable compression ratio makes always pick the lowest BSFC between the two possible conditions. By the way it is not said that these are lower than the previous instant one.

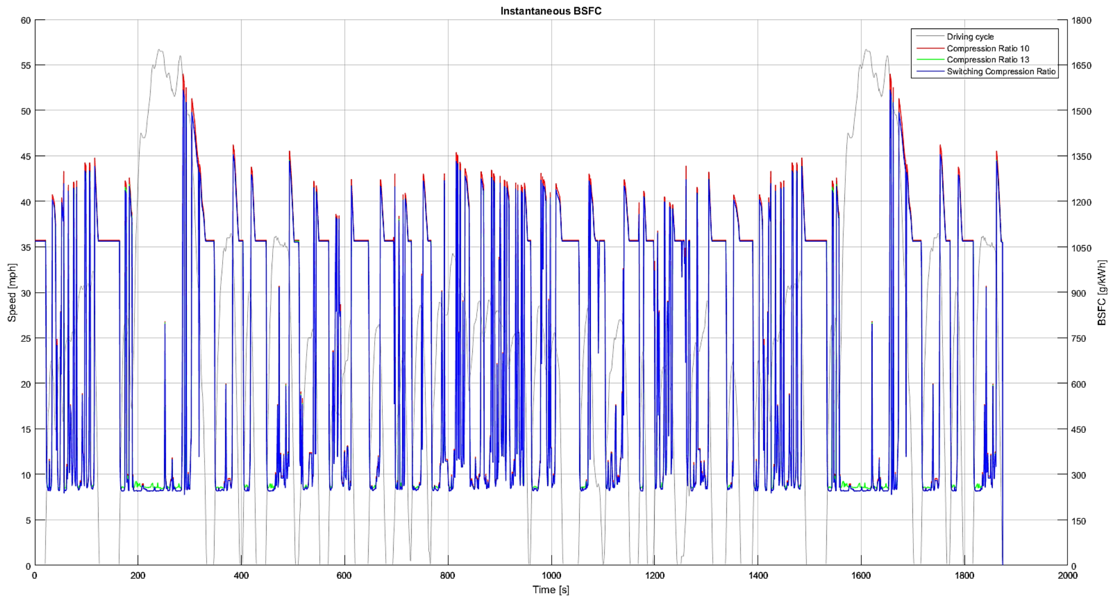


Figure 6.6: Instantaneous BSFC at CR 10, CR 13 and VCR

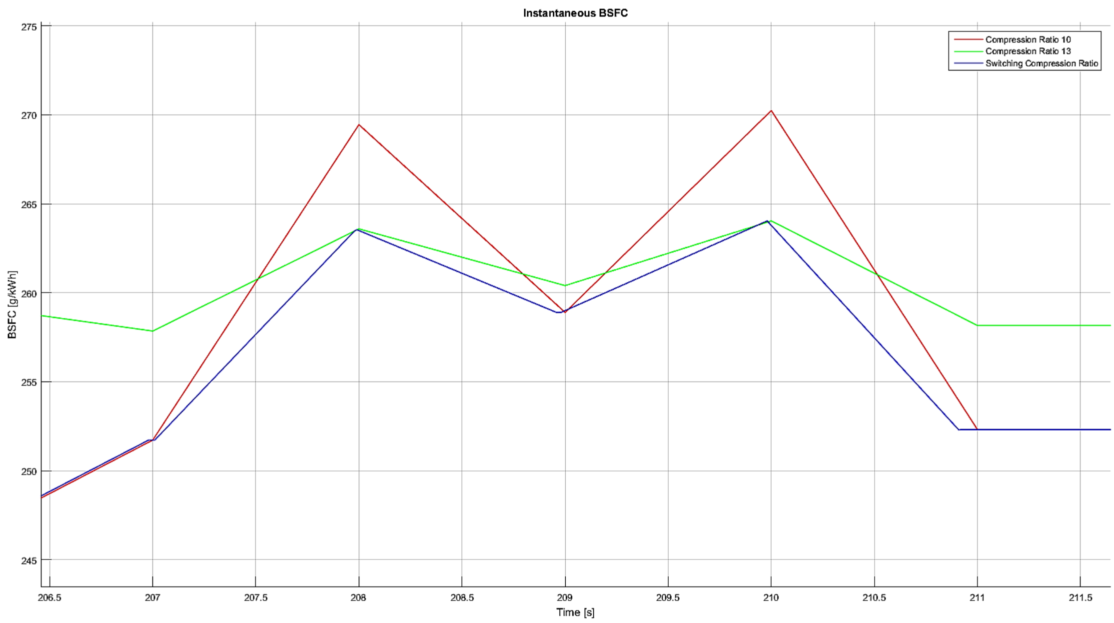


Figure 6.7: Instantaneous BSFC at CR 10, CR 13 and VCR - Zoom

6.3.2 Attempt 2 - Instantaneous switch depending on target BSFC - Non-instantaneous switch performed as soon as possible

At this point a kind of pre-check on following BSFC is needed. Being the main discrepancy noticed in the comparison between instantaneous and non-instantaneous switch, the function has been modified as follows. In first place it detects if the engine is working in ideal switching conditions or real. If ideal, meaning infinitively fast switch, a check on BSFC is done.

$$\begin{cases} \text{if } BSFC_{t+1} \geq BSFC_t & \text{Instantaneous switch occurs as late as possible} \\ \text{if } BSFC_{t+1} < BSFC_t & \text{Instantaneous switch occurs as soon as possible} \end{cases}$$

In this way it has been ensured that the instantaneous switch has always a lower fuel consumption compared to the non-instantaneous case. The logic of the controller is reported in Figure 6.8. Looking at the areas under the curves, the green line shows, in both cases, the lowest fuel consumption.

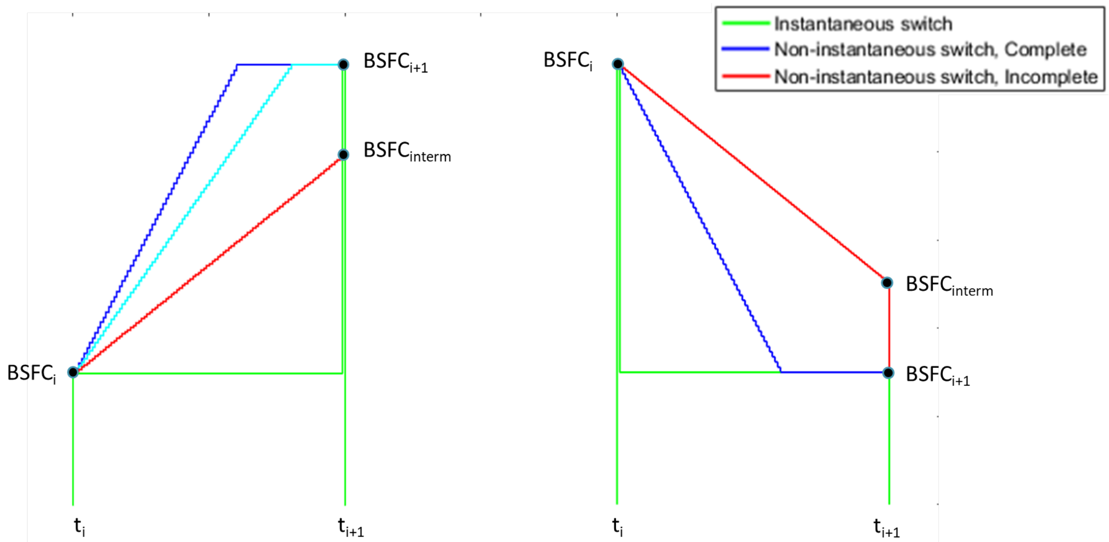


Figure 6.8: Logic Layout - Attempt 2

It follows that, for the case in which $BSFC_{t+1} \geq BSFC_t$, the Equation 6.17 should be modified as:

$$m_f = (BSFC_0{}^* \cdot (\Delta T - \Delta t) + BSFC_1{}^* \cdot \Delta t) \cdot \frac{P_d}{3600} \qquad [g] \qquad (6.23)$$

While the one for non-instantaneous switch (Equation 6.16), in both cases, remains unvaried.

Application to CVT

Again, referring to the vehicle on the FTP driving cycle, the results with a CVT are taken in account. The only difference with previous case lays in the way in control operates within the switching fuel penalty function.

Referring to the instantaneous fuel economy in each operating point of the driving cycle the difference between instantaneous and several non-instantaneous compression ratio switching is considered. The result is plotted in 6.9.

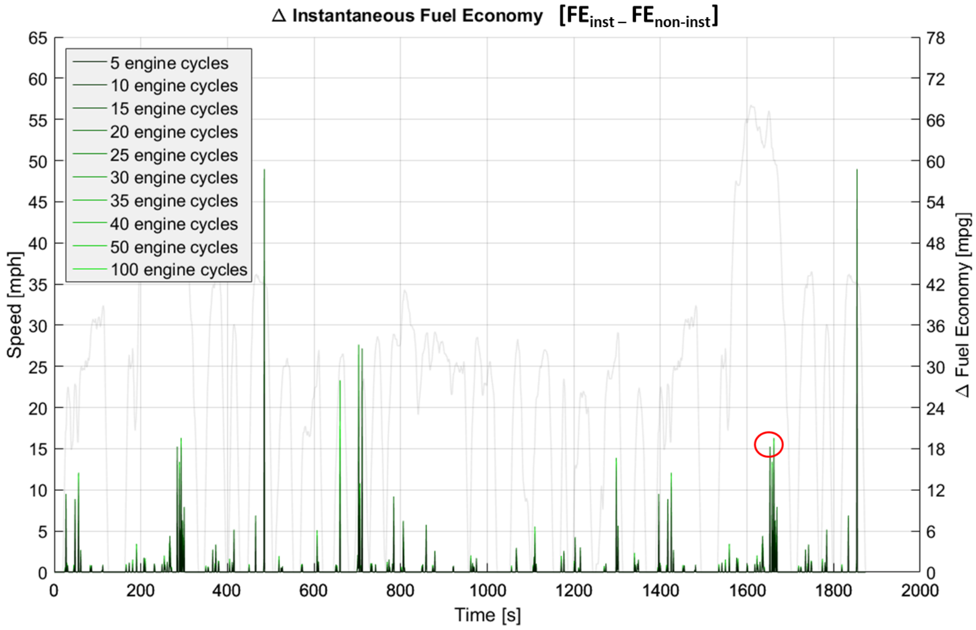


Figure 6.9: Instantaneous Fuel Economy Difference between Instantaneous and Non-Instantaneous cases

As it can be noticed all the negative part disappeared, confirming that now the instantaneous theoretical switch is always better than the real one. But, when zooming on the peaks, another problem can be spotted: in some cases the slower switching is associated to an higher fuel economy (lower fuel consumption). Again, by looking at Figure 6.8, it can be noticed how the area under the incomplete / slower switch is smaller compared to the faster one. This occurs when $BSFC_{t+1} \geq BSFC_t$ since it is as the engine is moving to an operating point with higher BSFC faster when, instead, could have exploited longer the lower one.

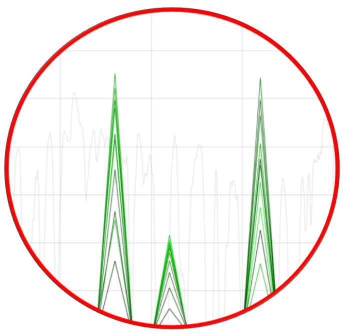


Figure 6.10: Instantaneous Fuel Economy Difference between Instantaneous and Non-Instantaneous cases - Zoom

6.3.3 Attempt 3 - Switching timing depending on target BSFC

As anticipated, if the next operating point shows an higher fuel consumption, the engine should remain at the starting one as long as possible; for this reason the controller should give the signal to start the switch as late as possible, still ensuring that the process can be completed.

$$\begin{cases} \text{if } BSFC_{t+1} \geq BSFC_t & \text{Sswitch occurs as late as possible} \\ \text{if } BSFC_{t+1} < BSFC_t & \text{Switch occurs as soon as possible} \end{cases}$$

Hence, in the first case, the starting switching time is computed as:

$$T_0' = \Delta T - T_s \neq T_0 \equiv t_i \tag{6.24}$$

The following behavior is plotted in Figure 6.11. In this case the slower switching will have a larger area under the curve.

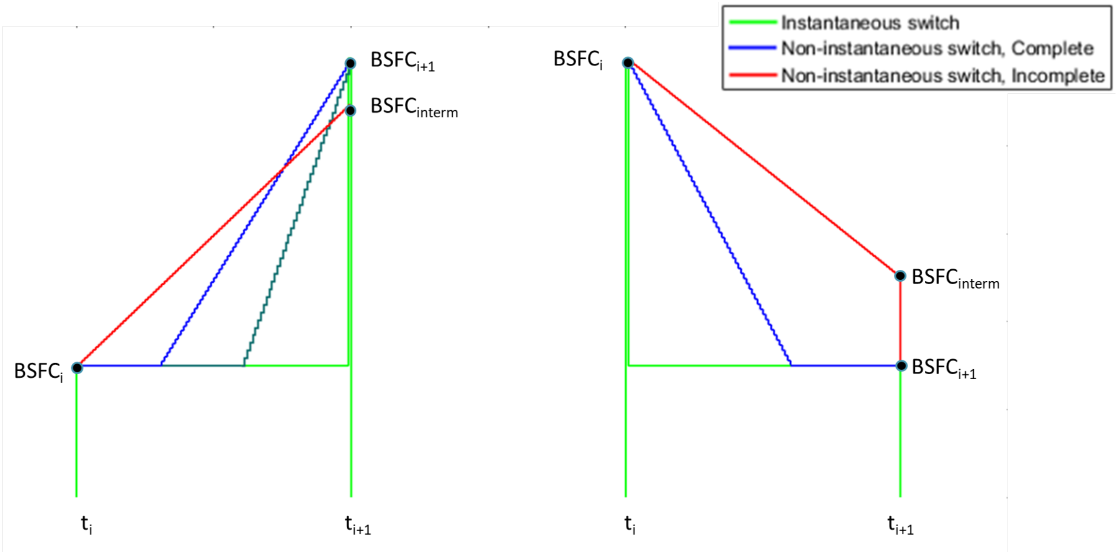


Figure 6.11: Logic Layout - Attempt 3

This lead to a different way to compute BSFC in the interval in the case $BSFC_{t+1} \geq BSFC_t$.

$$m_f = \left(BSFC_0{}^* \left(T_0' - T_0\right) + \sum_{k=0}^{steps_t} \left(BSFC_0{}^* + k \cdot BSFC_i\right) \cdot \Delta t \right) \cdot \frac{P_d}{3600} \quad [g] \tag{6.25}$$

Particular attention should be given to incomplete switch: following the logic explained, if a switch is extremely slow the red line in Figure 6.11 will go down, approaching the horizontal one. Hence, the area, after reaching a maximum value, starts decreasing. Anyway, in the cases analyzed, even when have been considered 100 engine cycles for a complete switch, the slower switching has resulted with an higher fuel consumption. Thus, in practical applications, it is almost impossible to find such slow processes leading to this model drawback. This error in fact arises from the mathematical model built for the switching process.

For clarity reasons this remote chance had to be reported. However, this final attempt has been considered as the best in order to represent the fuel penalty due to switching time and used as final.

6.4 Vehicle with CVT - Impact of Switching Time on Fuel Economy

The main references for results analysis are the three vectors defined in Section 6.3.1. In first place the cumulative fuel consumption is considered and plotted, for each case, in the Figure 6.12. As can be seen the curve shift upwards increasing the number of engine cycles needed for switch, meaning that more fuel is necessary to go through the driving cycle.

Also the abscissa where increases are registered vary increasing the switching time: for more time-requiring processes it can occur that more incomplete switching are present and a different approach should be used with respect to previous cases.

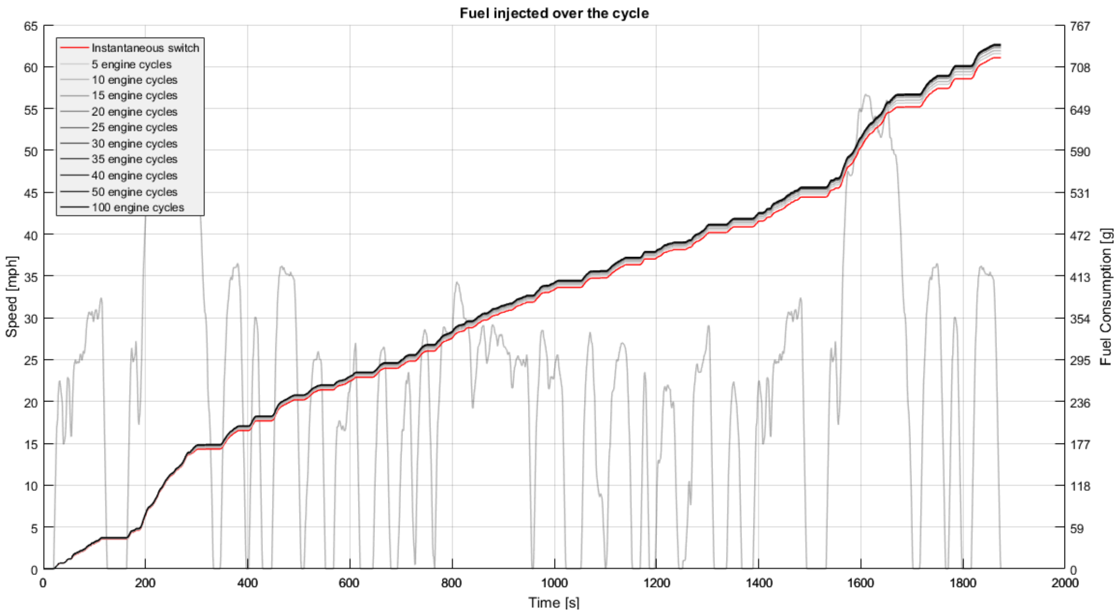


Figure 6.12: Cumulative Injected over the FTP Driving Cycle - CVT

As can be noticed the plot has been overlapped to the driving cycle. The higher slope and increase in fuel consumption is registered at high acceleration-deceleration zone. At this point, in addiction to the higher power required, the switching line is crossed several times requiring more switching events.

Another interesting way to appreciate the results is looking at the average fuel economy over the same driving cycle in the different cases. In this way it is possible to compute the percentage decrease in fuel economy due to the penalty coming from switching time. The results are plotted in the Figure 6.13.

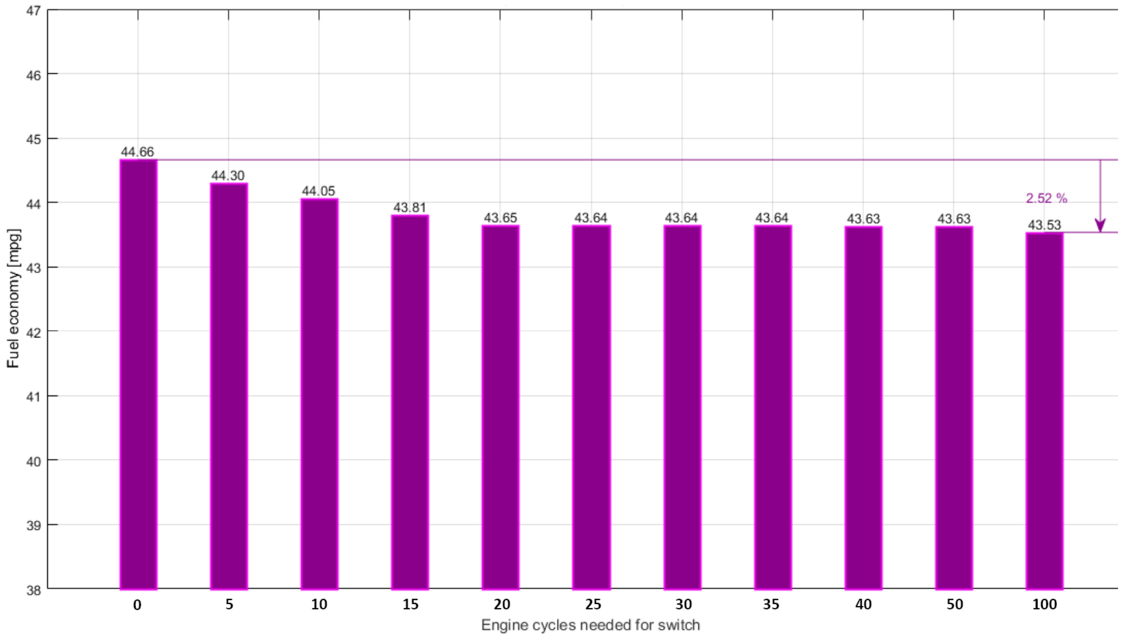


Figure 6.13: Average Fuel Economy Comparison - CVT

The percentage decrease is computed between instantaneous switch and 100 engine cycles switch duration. For the smaller values of switching time the decrease is about decimals. In the Chapter 8 will be analyzed which are the parameters affecting this number.

6.5 Flow charts explaining the logic

In order to give to the reader a clear picture on how the fuel penalty associated with switching is computed, the function has been resumed in the following flow charts. In this way an intuitive overview is provided.

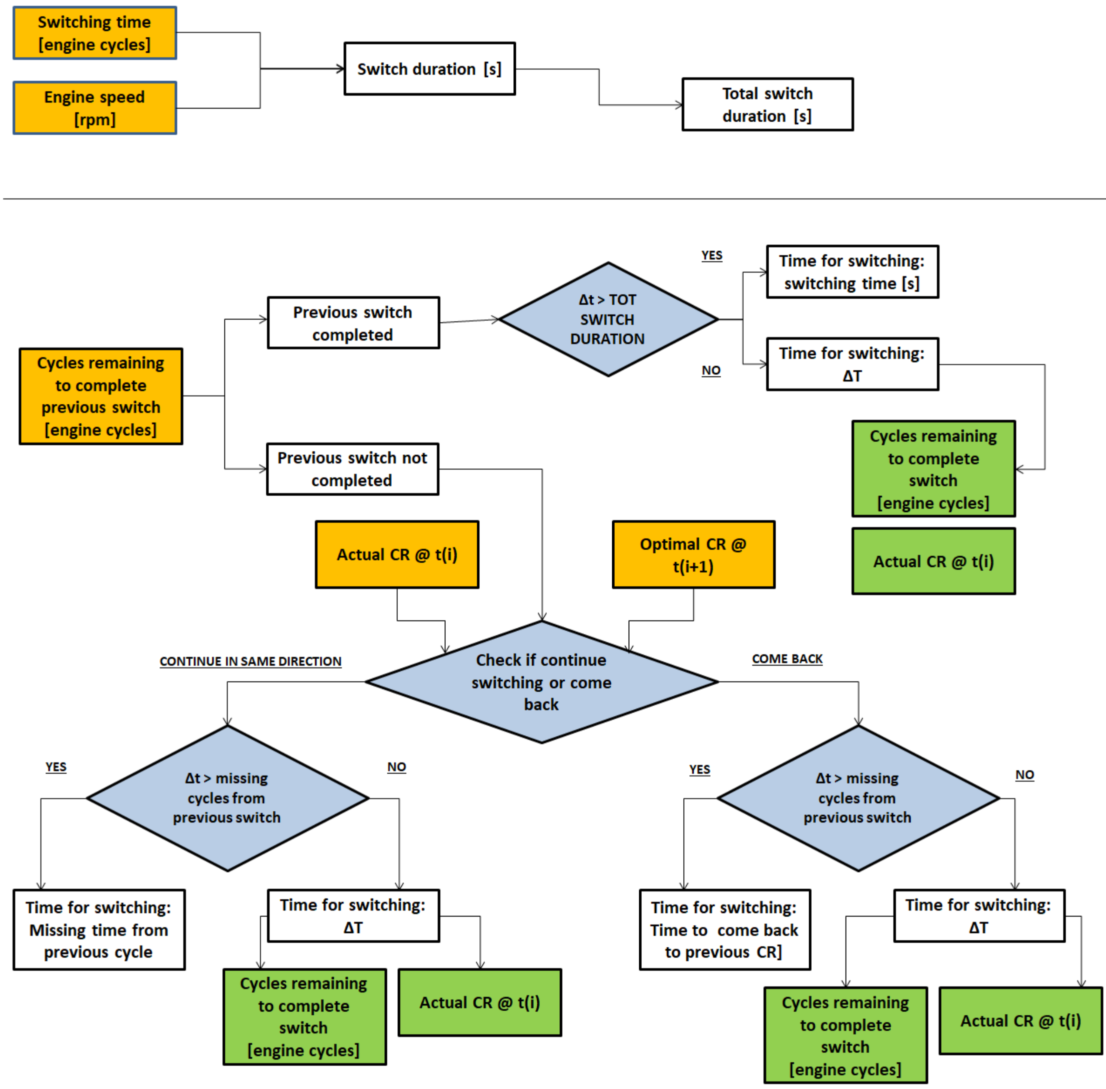


Figure 6.14: Switching Fuel Penalty Function - Flow Chart - Part 1

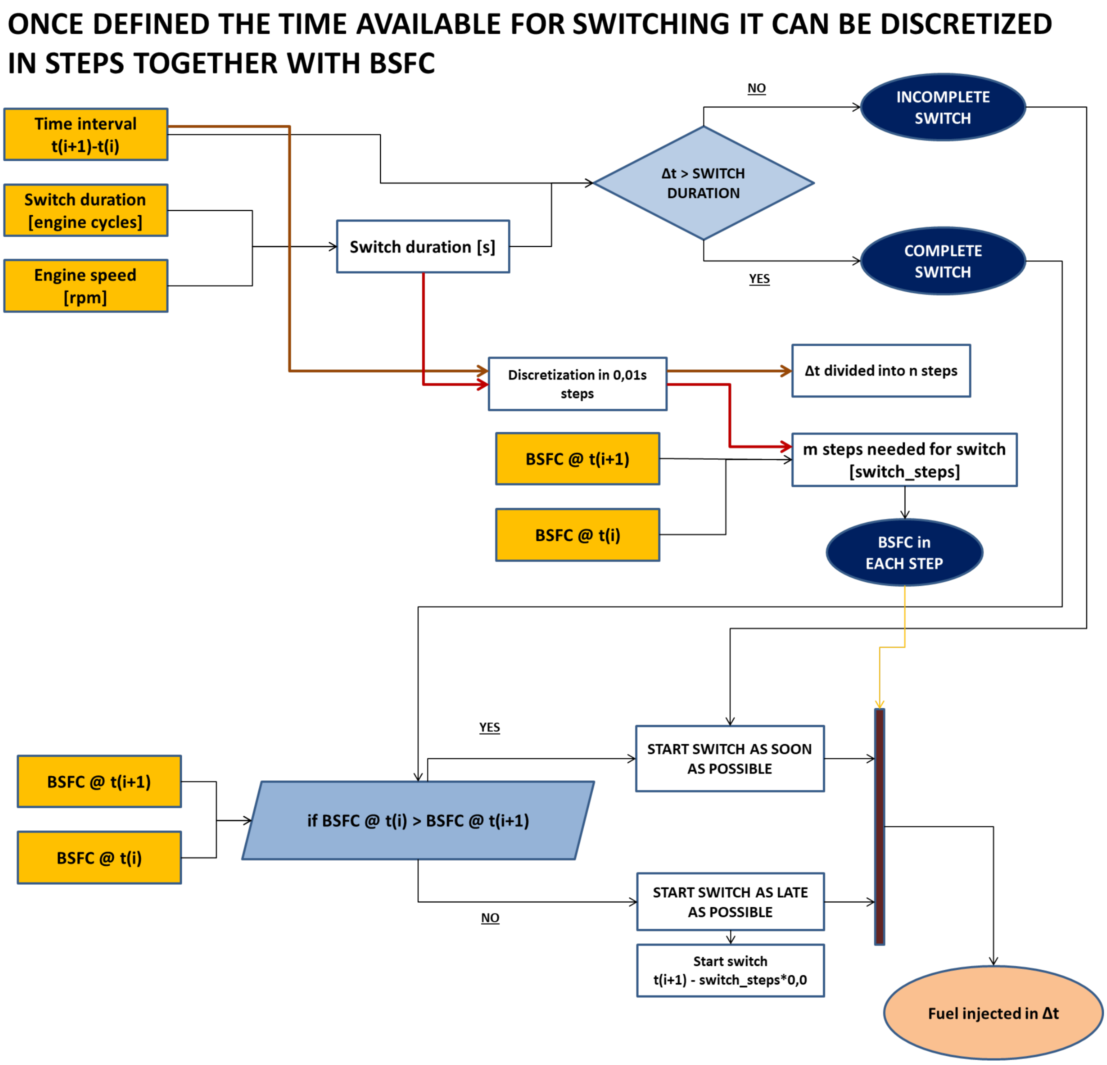


Figure 6.15: Switching Fuel Penalty Function - Flow Chart - Part 2

Chapter 7

Vehicle with Manual Transmission

Coming to the final case, the one closest to the reality, the manual transmission is considered. The term manual should be considered with care since no fuel penalties for gear shifting or delay due to driver response to the cycle have been considered. Then, being such simplifications adopted, there is a fine line between the term manual and automatic 6 ratios transmission. By the way, being all the values al referred to the Jeep Renegade, that uses a manual transmission, this terminology has been chosen.

Moreover the code has been written so that if in a second analysis penalties for gear shift are needed can be easily inserted. In this study the losses have been assumed null.

With the term "manual" it is intended a transmission with fixed gear ratios that, starting from vehicle speed allows speed reduction, from wheel level to engine level, in a step-way, differing in this way from CVT. For this reason all the environment in which the Switching Logic is placed should be changed, and a second code will be necessary. The introduction part where all the data are imported and Switching Line found remains practically unvaried. The main difference, on the other hand, lays in how the operating points on the map are chosen.

There is a driving logic to go through the driving cycle: once defined the first idle point, the manual transmission logic should be able to compute which is the best gear ratio for next point, in order to approximately reproduce the behavior of a real driver. This choice must be made simultaneously with the Compression Ratio, increasing of one the order of the problem.

Another important difference with respect to CVT model, is that in this case the efficiency of the transmission is not ideal, thus different from 100%.

Values of transmission ratios and efficiencies are reported in Tables 7.1 and 7.2

Minimum Engine Speed needed for shift

As a further requirement, among the input, has been inserted a parameter that indicates which is the minimum engine speed that have to be reached for shift the gear. Two cases have been analyzed and the impact on fuel economy, together with the compression ratio switching has been evaluated.

Type	Manual
Gears	6
Final drive ratio	4,334
Gear shift #1 transmission ratio	4,15
Gear shift #2 transmission ratio	2,12
Gear shift #3 transmission ratio	1,36
Gear shift #4 transmission ratio	0,98
Gear shift #5 transmission ratio	0,76
Gear shift #6 transmission ratio	0,62

Table 7.1: Manual Transmission Specifications

Final drive efficiency	1
Gear shift #1 efficiency	0,93
Gear shift #2 efficiency	0,93
Gear shift #3 efficiency	0,93
Gear shift #4 efficiency	0,93
Gear shift #5 efficiency	0,93
Gear shift #6 efficiency	0,93

Table 7.2: Manual Transmission - Efficiencies

7.1 Procedure

Again due to the non-predictivity of the code the code should be based only on the pedal request for the next point. Summarizing the point in a list, for each time instant:

1. Read the pedal request for next point;
2. Compute the power at wheels;
3. For each one of the gear
   (a) Compute the power at engine considering the efficiency of the transmission gear;
   (b) Compute the rpm at that gear considering the transmission ratio;
   (c) Compute the bmep associated;
   (d)
   $$\begin{cases} \text{if RPM} < 800 & \text{engine in IDLE} \\ \text{if BMEP above WOT} & \text{need to move the point;} \end{cases}$$
4. Once all the 6 operating points corresponding to each gear have been computed starting from vehicle speed:
   (a) Compute the BSFC at Low Compression Ratio for each one of them;
   (b) Compute the BSFC at High Compression Ratio for each one of them;

(c) Compare them so to have the best combination of gear and compression ratio for next point;

(d) Apply the switch logic if required;

(e) Take as output the fuel injected in the time step.

Analuzing in-depth each step, after the pedal request is done the power can be computed as:

$$a_{veh} = \frac{v_{veh,t_{i+1}} - v_{veh,t_i}}{t_{i+1} - t_i} \qquad \left[\frac{ft}{s^2}\right] \tag{7.1}$$

$$F_{res} = A + B \cdot v_{veh} + C \cdot v_{veh}^2 \qquad [lbf] \tag{7.2}$$

$$P_{wheels} = \frac{(F_{res} + a_{veh} \cdot M_{app}) * v_{veh,t_{i+1}}}{\eta \cdot 1000} \qquad [kW] \tag{7.3}$$

where:

$$\begin{aligned}
a_{veh} &= \text{Vehicle acceleration} \\
v_{veh} &= \text{Vehicle speed} \\
F_{res} &= \text{Sum of resistance forces acting on the vehicle} \\
A, B, C &= \text{Coast down coefficients} \\
P &= \text{Power demand} \\
M_{app} &= 1,03 \cdot M_e, \text{ where } M_e \text{ is the equivalent test weight [lbs]} \\
\eta &= \text{Transmission Efficiency}
\end{aligned}$$

Regenerative braking is not considered, hence if negative values of power, during braking or decelrating, showed up, they have been imposed zero.

In this case, being the efficiency different from one, the power at engine level differs from the one at the wheels:

$$P_{engine} = \frac{P_{wheels}}{\eta_{final} \cdot \eta_{geat}} \tag{7.4}$$

Hence, the power demand at the engine, looking at the values in Table 7.2, will be the same for each gear ratio.

In this case the engine speed can be computed exactly starting from the following relation:

$$enginespeed = \frac{v_{veh} \cdot 60}{2\pi \cdot r_{wheel}} \cdot \tau_{final} \cdot \tau_{gear} \qquad [rpm] \tag{7.5}$$

and, through power, also load is uniquely defined:

$$engineload = 1200 \cdot \frac{P_{engine}}{[rpm] \cdot V_{tot}} \qquad [bar] \tag{7.6}$$

If this operation is made for each gear - considering every time different transmission ratio and efficiency - six combinations of speed and load are defined. The values should be anyway adapted to the map available, that is not engine's real one.

- if the values are too low the engine goes in the idle condition, so a fixed engine speed of $800rpm$ is imposed;
- the fuel maps used represent a Naturally Aspired engine, thus there will be a lot of points that cannot be reached since the vehicle has got a Turbocharged engine and can reach significantly higher loads. In order to not lose all the information on these points, the code goes in open wide throttle and provides, at that vehicle speed, the maximum torque possible. It won't be the value asked by the driving cycle, but the maximum reachable by the engine in that condition. There is basically a **Power Reduction** with respect to driving cycle. Point in which Power Request cannot be fulfilled are highlighted in the final table containing all the result on the driving cycle.

At the end of the process a vector containing gears and corresponding operating point is generated. So now BSFC come into play since among the six possibilities, the combination showing lower fuel consumption will be chosen. Another constraint is imposed: in practice not all the combination can be chosen since it has to be considered the gear in which the transmission is at the instant t_i; starting from this condition the driver has three possibilities:

1. Up-shift;
2. Down-shift;
3. Stay at the same gear;

Moreover, if both the engine speed at t_i the one at t_{i+1} are below the minimum engine speed needed for shift, the choice it is reduced to one option, that is to stay at the same gear.

Another case that requires attention is deceleration. A double check has been imposed on power demand and vehicle speed coming from pedal request: if both are decreasing or nil, it means that the driver is braking or decelerating, thus should start shift-down. Doing this it is possible to avoid situation where the code keeps the same gear during the deceleration and then is suddenly required to go to idle.

Through a series of *if conditions* in *Matlab* it is possible to exaclty understand in which case the engine is operating and select the possible combinations for next point. When they are reduced from six to a number included between one and three, Brake Specific Fuel Consumption is interpolated in each one of them for each compression ratio.

Again the absolute miniumum should be found, and it would be encountered at a precise combination of gear and compression ratio.

Some cases, for example at high loads, includes zones where the High Compression Ratio cannot work due to knock limitations. The corresponding point at Low CR will still be taken in account and, if the BSFC shown is lower than all the other possibilities, it will be chosen.

7.2 Results

Again considering the Federal Test Procedure as reference Driving Cycle, and imposing a minimum engine speed needed for shift equal to 1100 rpm, the set of operating points measured for instantaneous switch is reported in Figure 7.1.

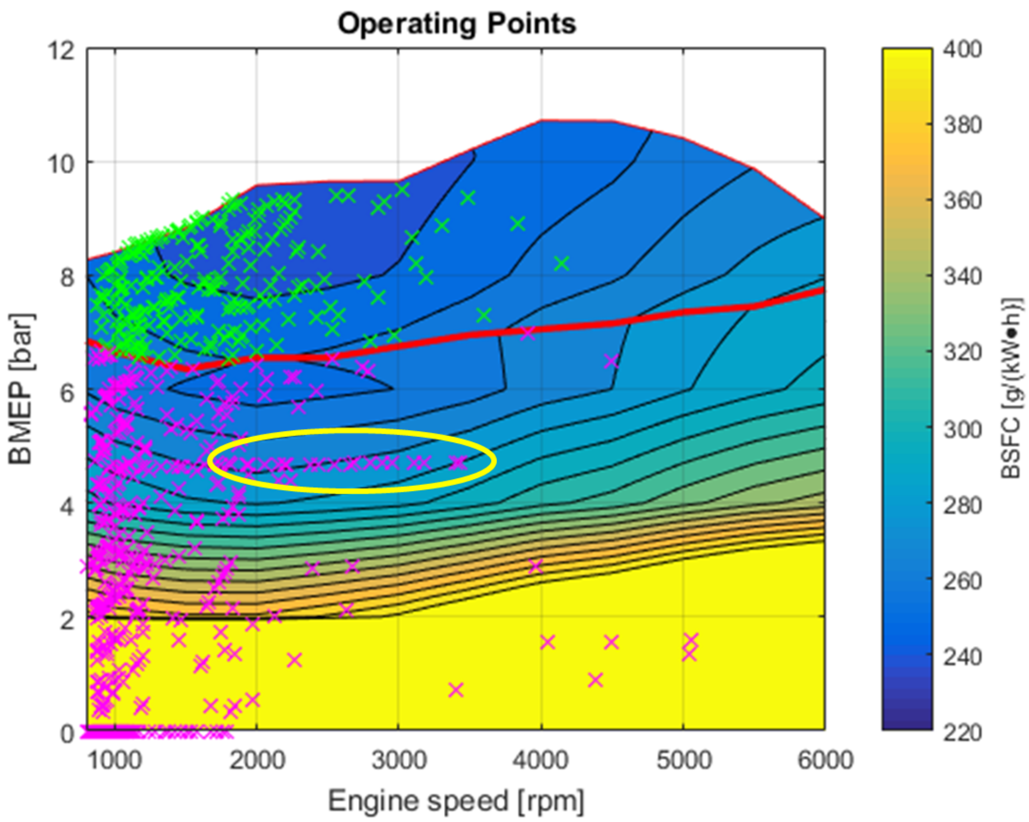


Figure 7.1: Operating Points on FTP cycle with Manual Transmission Logic

As can be noticed in the figure there is an odd concentration of points around 5 bar BMEP. This problem has been investigated. In the process has been realized a matrix containing all the information on the driving cycle, registered step per step.

- time $[s]$
- power at wheels $[kW]$
- power at engine $[kW]$
- engine speed $[rpm]$
- engine load $[bar]$
- gear
- BSFC $[g/kWh]$

- fuel injected between that instant and the previous (m_fuel) [g]
- Optimal compression ratio
- Actual compression ratio

Referring to this latter, all the points in the area of interest have been extracted in order to figure out what they had in common. Basically al this operating points show up after idle conditions. What happens is:

1. Start from idle;
2. Go in first gear (low speed (around $1000 \div 2000 rpm$) and load around 5 bar);
3. At this point, if there was a real driver it would be logic to go in second gear for third point but the code keeps at gear 1 because the BSFC at gear 2 is slightly higher with rspect to BSFC at gear 1 (about decimals);
4. when this really small gap is overcame the driving continues normally.

Some of interested operating point details are showed in Figure 7.2

TIME	PWR @ wheels	PWR @ engine	rpm	bmep	gear	BSFC	m fuel	CR	
164	0	0	800	0	0	1056.10199	0	13	
165	7.530349992	8.01101063	1476.9556	4.64914487	1	277.922159	0.61845482	13	BSFC gear 2
166	11.35065791	12.07516799	2215.4334	4.67183712	1	275.586075	0.92437448	13	275.821
167	15.22091005	16.1924575	2953.9112	4.69860072	1	280.310703	1.26081087	13	
347	0	0	800	0	0	1056.10199	0	13	
348	4.8919878	5.204242341	962.258951	4.63573672	1	285.110722	0.41216258	13	
349	8.683335747	9.237591221	1700.73675	4.65559137	1	276.514638	0.70953589	13	
350	12.5177337	13.31673798	2439.21455	4.67951736	1	276.850865	1.02409734	13	
351	16.40507337	17.45220571	3177.69235	4.70751471	1	282.228084	1.36819516	13	
403	0	0	800	0	0	1056.10199	0	13	
404	6.725464702	7.154749683	1320.30879	4.64485476	1	280.170175	0.55681874	13	
405	10.53645005	11.20898942	2058.78659	4.6666834	1	274.846435	0.85576411	13	
406	14.39528138	15.31412913	2797.26439	4.69258338	1	279.53617	1.18912583	13	
448	0	0	800	0	0	1056.10199	0	13	
449	7.530349992	8.01101063	1476.9556	4.64914487	1	277.922159	0.61845482	13	
450	11.35065791	12.07516799	2215.4334	4.67183712	1	275.586075	0.92437448	13	
451	15.22091005	16.1924575	2953.9112	4.69860072	1	280.310703	1.26081087	13	

Figure 7.2: Some Operating Points belonging to the odd concentration area

On one side this confirms that the code always pick the lowest BSFC, correctly. On the other hand, given the not exact shape of the fuel maps and a control logic for manual transmission not advanced, this odd area of points forms.

7.3 Vehicle with Manual Transmission - Impact of Switching Time on Fuel Economy

Again, an increasing number of cycles needed for switching has been considered so to evaluate the impact on fuel consumption. The code, except for the different logic adopted for best operating point choice, is identical to the CVT one. Hence, at the end, the results are gathered in specific vectors within the *Matlab* code; the mass of fuel injected, cumulative fuel consumption and instantaneous fuel economy are considered for each step of the driving cycle. Other interesting parameters are contained in the *driving_schedule* matrix and checked when details are needed.

By looking at the cumulative fuel consumption when increasing the number of engine cycles needed for switch reported in Figure 7.3, it is possible to notice how there is a slightly lower raising of the curves with respect to the CVT case.

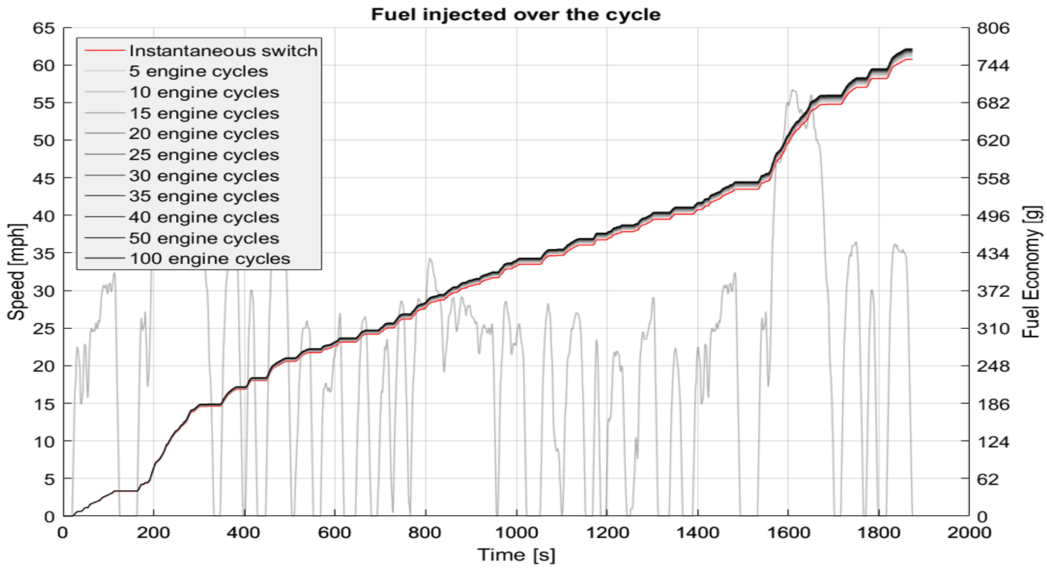


Figure 7.3: Cumulative Injected over the Driving Cycle - Manual Transmission

A further confirmation comes from the driving cycle average fuel economy results. In fact there is an approximatively 0.2% difference in the overall percentage decrease due to compression ratio switching. The trend can be spotted in Figure 7.4.

Comparing the results with the ones reported in Figures 6.12 and 6.13 the following considerations should be made:

- The manual transmission entails an higher fuel consumption being the choice of operating point governed by the six fixed transmission ratios. In other words, the engine cannot operate in the point with minimum BSFC on the constant power line but on a discrete number of speed and load combinations, usually showing an higher BSFC with respect to CVT;
- On the other hand, the impact of con-rod length variation on total fuel economy is less strong. Being the number of high-to-low and low-to-high

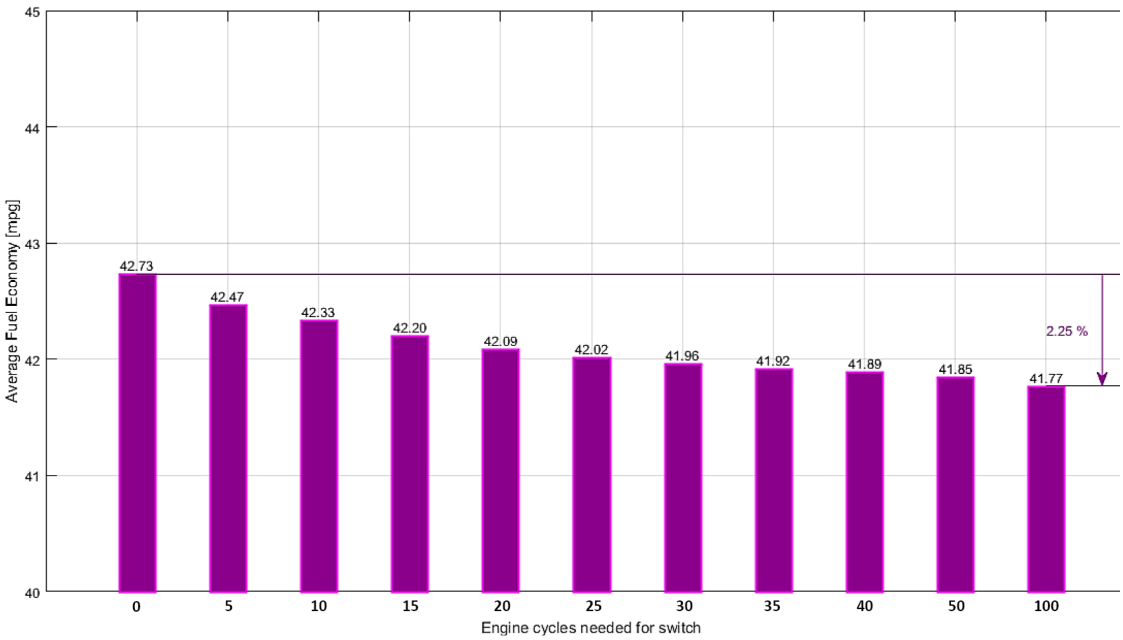


Figure 7.4: Average Fuel Economy Comparison - Manual Transmission

switches similar[1], the reasons of such a behavior should be attributed to other phenomena. In particular, when using Manual Transmission the benefit of Variable Compression Ratio is less pronounced with respect to CVT; even when considering instantaneous switch the gain in fuel economy is lower. Hence, when considering the impact of switching time, the effects on average fuel economy are not so evident because in general VCR is less effective.

7.3.1 Increase of minimum engine speed needed for shift

As additional parameter in Manual Transmission case the minimum engine speed needed for shift has been inserted. Basically this value represent the engine speed above which the gear can be changed; in other words if the engine is operating at a speed below this threshold and the pedal request for the next point falls within the same region, the gear must remain the same. The introduction of this constraint resembles the logic to an Automatic Transmission.

For analysis purposes the minimum engine speed has been increased from $1100rpm$ to $1400rpm$, the map of operating points is reported in figure 7.5

As can be noticed no substantial changes to the map can be seen except for a slight shift of the operating points below 1100rpm towards higher engine speeds. This occurs since the engine is forced to reach higher rpm before shifting. Consequently higher fuel consumption, and lower fuel economy, are expected when the minimum engine speed needed for shift is increased. By looking at Figures 7.6 and 7.7, the following considerations should be made:

- The average fuel economy increase when incresing the minimum engine speed

[1]By looking at the two codes, the switch is required 92 times with Continuously Variable Transmission and 94 with Manual Transmission over the FTP driving cycle.

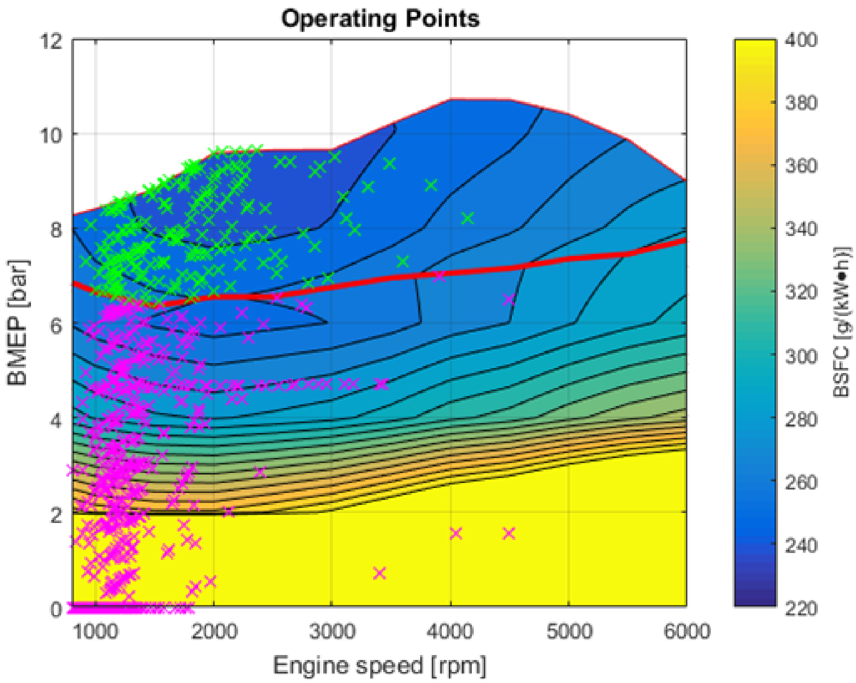


Figure 7.5: Operating Points on FTP cycle with Manual Transmission - Increased minimum engine speed needed for shift 1400rpm

needed for shift, since the engine cannot go to the gear providing the lower BSFC if the speed is below the threshold;

- There is an increment of the impact of switching time on fuel economy since the percentage decrease of average fuel economy, between Instantaneous Switch and 100 engine cycles needed for switch, varies as reported in Table 7.3.

Minimum engine speed needed for shift [rpm]	Percentage decrease of fuel economy due to non-instantaneous switch
1100	2,25%
1400	2,61%

Table 7.3: Percentage decrease of fuel economy due to non-instantaneous switch varying minimum engine speed needed for shift

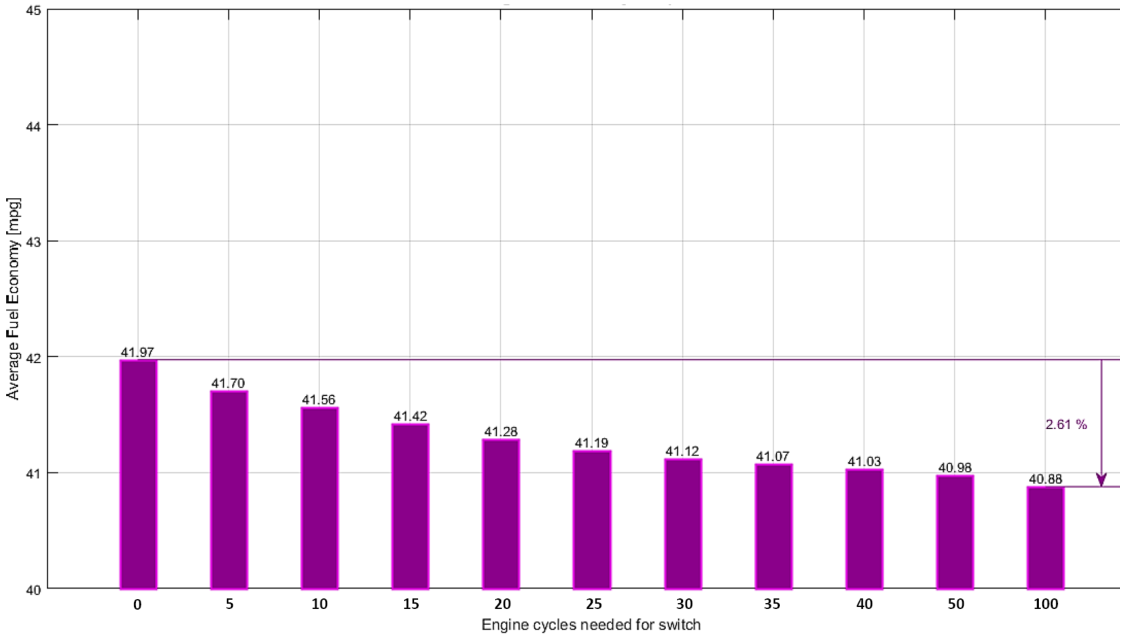
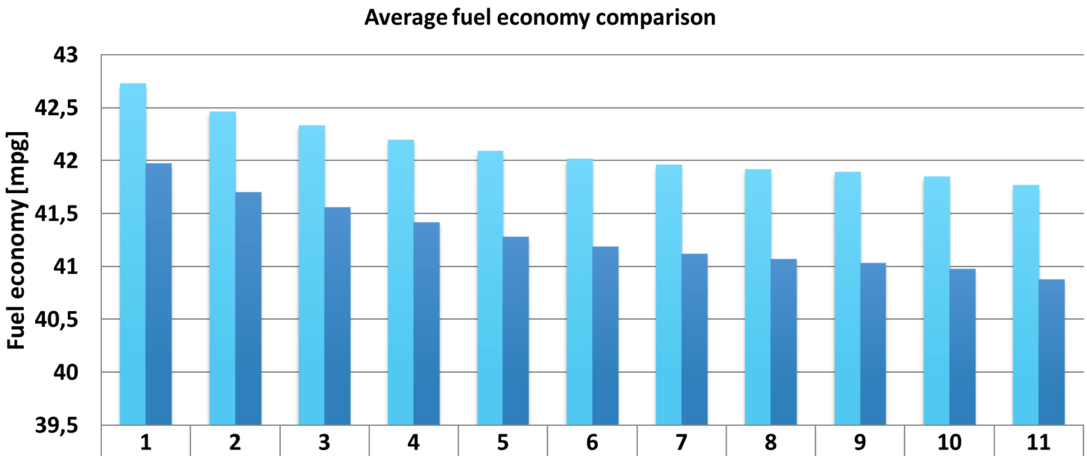


Figure 7.6: Average Fuel Economy Comparison - Manual Transmission



	1	2	3	4	5	6	7	8	9	10	11
1100	42,7315	42,4681	42,3342	42,2012	42,0912	42,0193	41,962	41,9225	41,893	41,8512	41,7716
1400	41,9738	41,702	41,5606	41,4199	41,2834	41,1912	41,1217	41,0718	41,0339	40,9809	40,8788

Figure 7.7: Comparison between average fuel economy over the FTP cycle for different switching times at minimum engine speed needed for shift respectively of 1100rpm and 1400rpm

Chapter 8

Influence of other parameters

8.1 Driving cycle

In order to have a wider perspective of the results, different driving cycles have been inserted as input and resulting average fuel economies compared. This process has been used also as a check on the versatility of the code, for vehicle both with manual and continuously variable transmission. The driving cycles considered are the Emission Test Cycles commonly used nowadays in the different countries. They have been chosen just as reference because easily accessible, but it is worth to highlight again how the code can work by simply inserting in the dedicated excel file a driving schedule in the form vehicle speed at each time.

Four different cycles have been considered, that together with the different transmission gave eight different scenarios:

- FTP-75 (Federal Test Procedure): Transient test cycle for cars and light duty trucks derived from the FTP-72. Used for emission certification testing of cars and light duty trucks in the USA.

- HWFET (Highway Fuel Economy Test): Test cycle reproducing on chassis dynamometer the highway conditions, considered in order to evaluate how the VCR switching behaves in such conditions.

- NEDC (New European Driving Cycle): A combined chassis dynamometer test used for emission testing and certification in Europe. It is composed of four ECE Urban Driving Cycles, simulating city driving, and one Extra Urban Driving Cycle (EUDC), simulating highway driving conditions. The NEDC includes the cold-start phase.

- WLTP (Worldwide harmonized Light vehicles Test Procedure) Chassis dynamometer test cycles for light-duty vehicles faced in in this years and defining a global harmonized standard for determining the levels of pollutants.

Even if they are really popular cycles, a description is needed in order to give a background to the numerical results that follow.

Before moving to the average fuel economy at different switching times and consequent percentage decrease in the values, it is worth again to remember one important simplification made inside the code. In particular when referring to WLTP, very high power levels are required and a wide variety of operating points is investigated. In this case the GT-Power model used for Fuel Maps creation constitutes a strong limit being not turbocharged. When the power request cannot be met, the code automatically operates the engine at wide open throttle giving the maximum power available in that conditions. Hence, in this scenario, it is not possible to fulfil the request coming from driving cycle and the consequent results will be affected. By looking at Figure 8.1 representing the engine operating points on the WLTP driving cycle when considering the vehicle with manual transmission, it is possible to notice how many points operates on the Wide Open Throttle (or Full Load) curve.

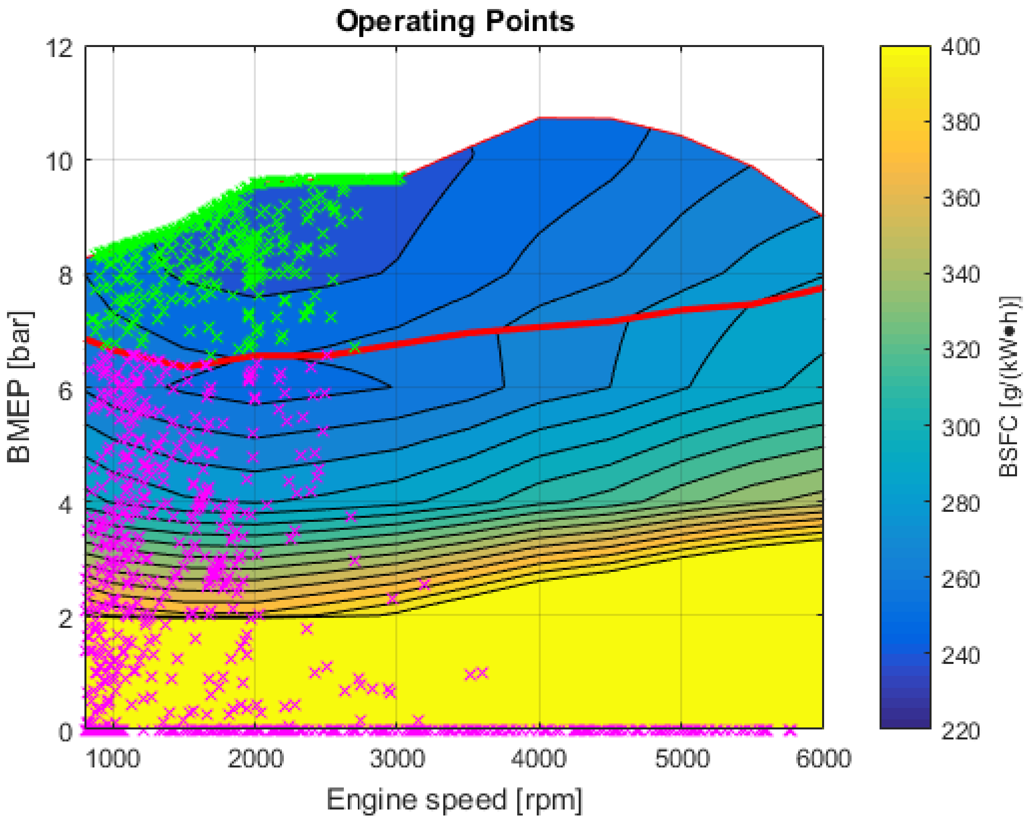


Figure 8.1: Operating Points on WLTP cycle with Manual Transmission

At each one of them there is a different condition from the one that would be the real one, obtained with engine test stand or a proper model. Thus the results coming will be affected by this simplification. On the other hand, when inserting the actual fuel maps coming from the considered vehicle engine the numerical results will be correct. Once clarified this point, it is possible to see how the switching time impacts differently when considering several driving cycles.

In first place the results for vehicle with CVT are showed in Figure 8.2, Figure 8.3 and Figure 8.4, respectively for HWFET, NEDC and WLTP.

The results coming from the vehicle with Manual Transmission are reported in Figure 8.5, Figure 8.6 and Figure 8.7, respectively for HWFET, NEDC and WLTP.

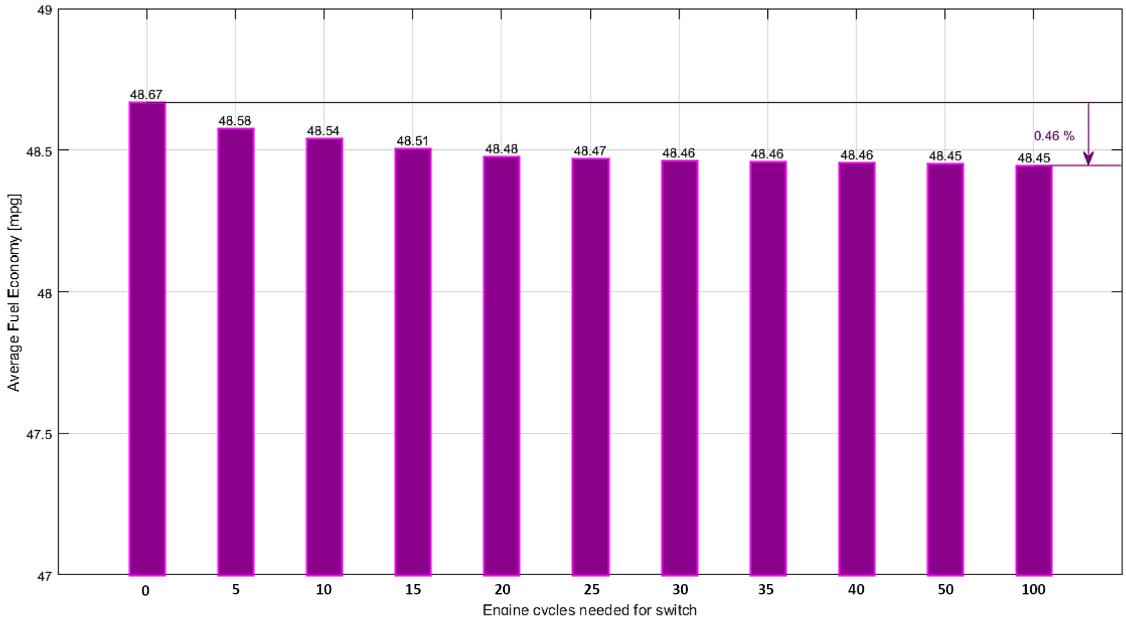


Figure 8.2: Comparison between average fuel economy over the HWFET driving cycle for different switching times - Vehicle with CVT

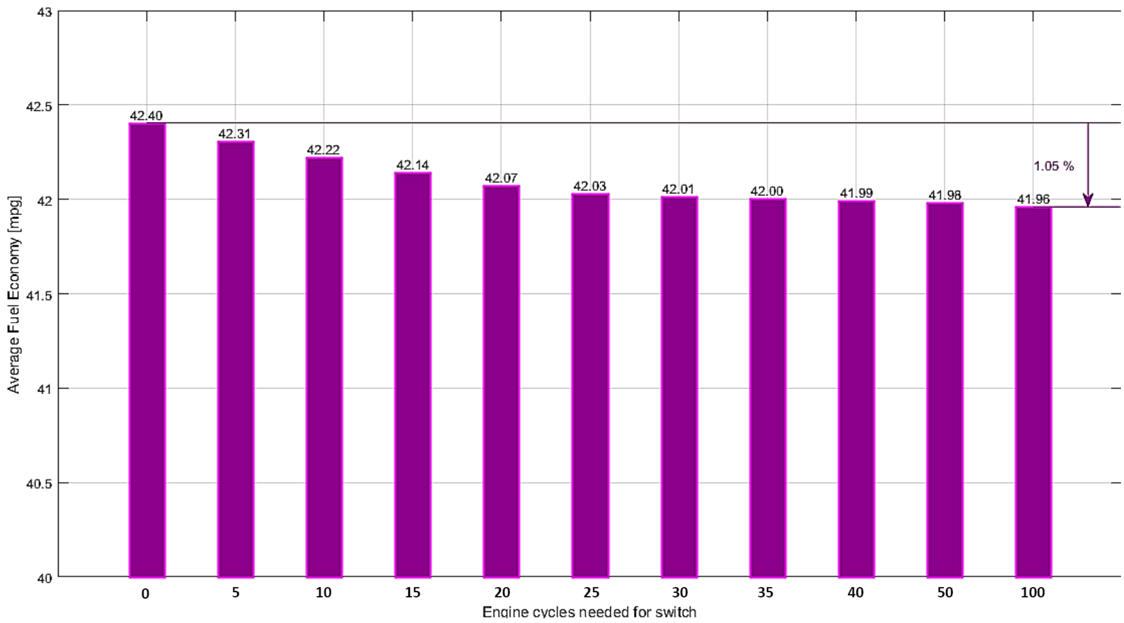


Figure 8.3: Comparison between average fuel economy over the NEDC driving cycle for different switching times - Vehicle with CVT

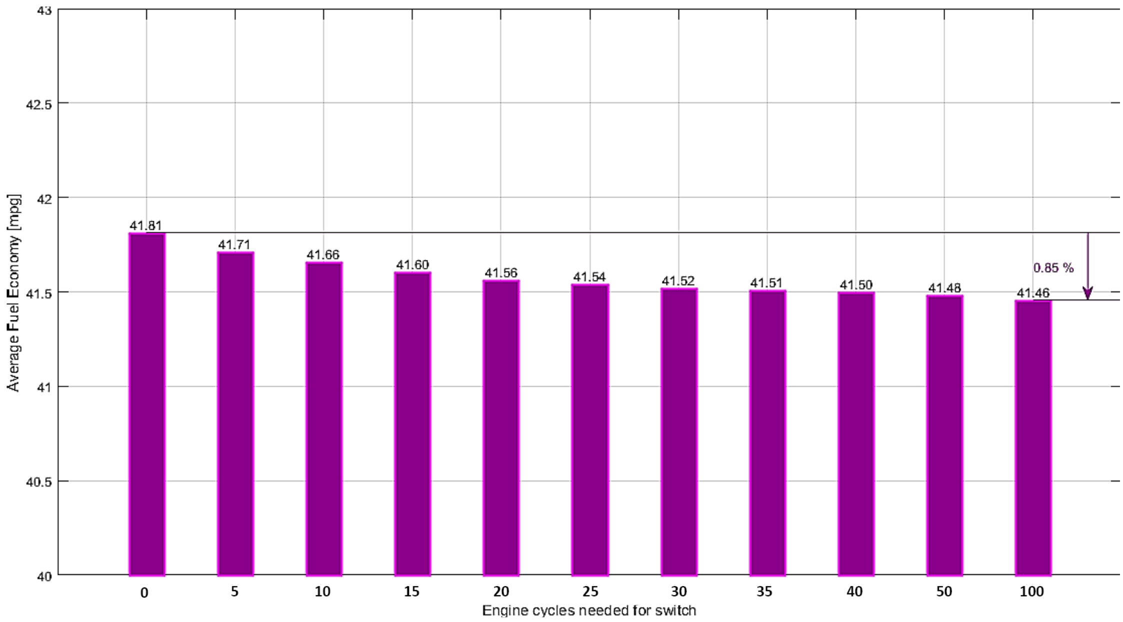


Figure 8.4: Comparison between average fuel economy over the WLTP driving cycle for different switching times - Vehicle with CVT

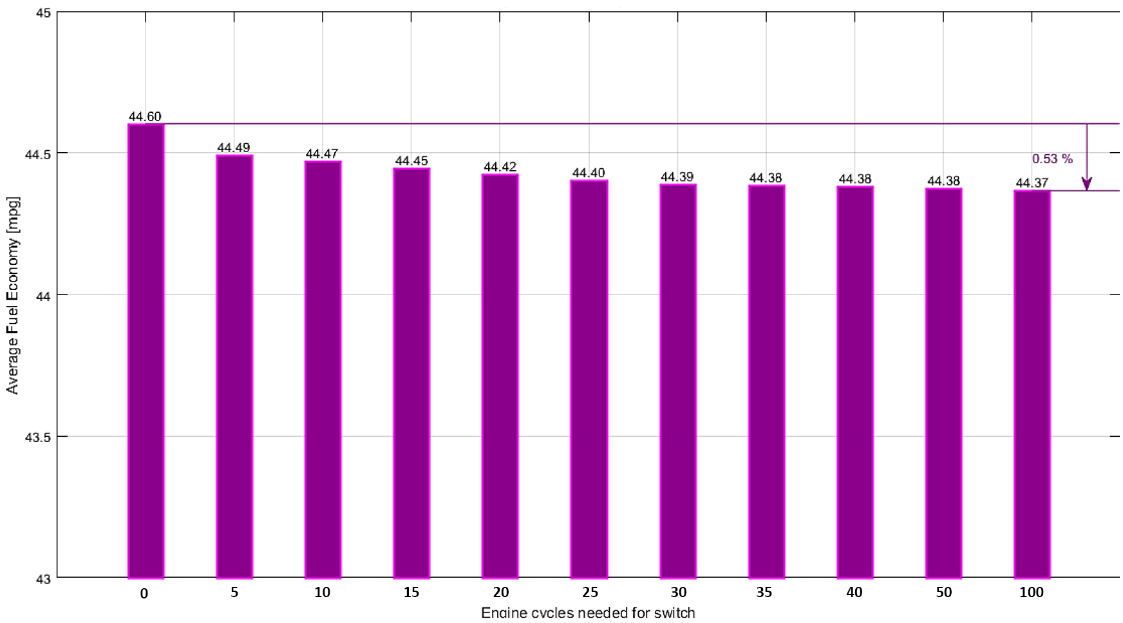


Figure 8.5: Comparison between average fuel economy over the HWFET driving cycle for different switching times - Vehicle with Manual Transmission

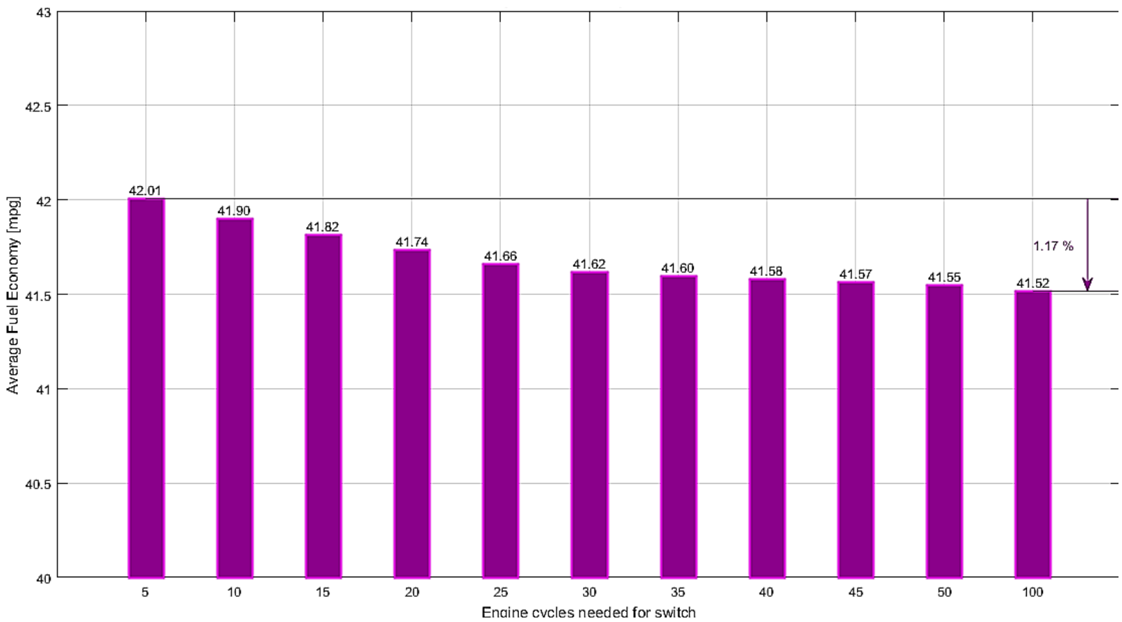


Figure 8.6: Comparison between average fuel economy over the NEDC driving cycle for different switching times - Vehicle with Manual Transmission

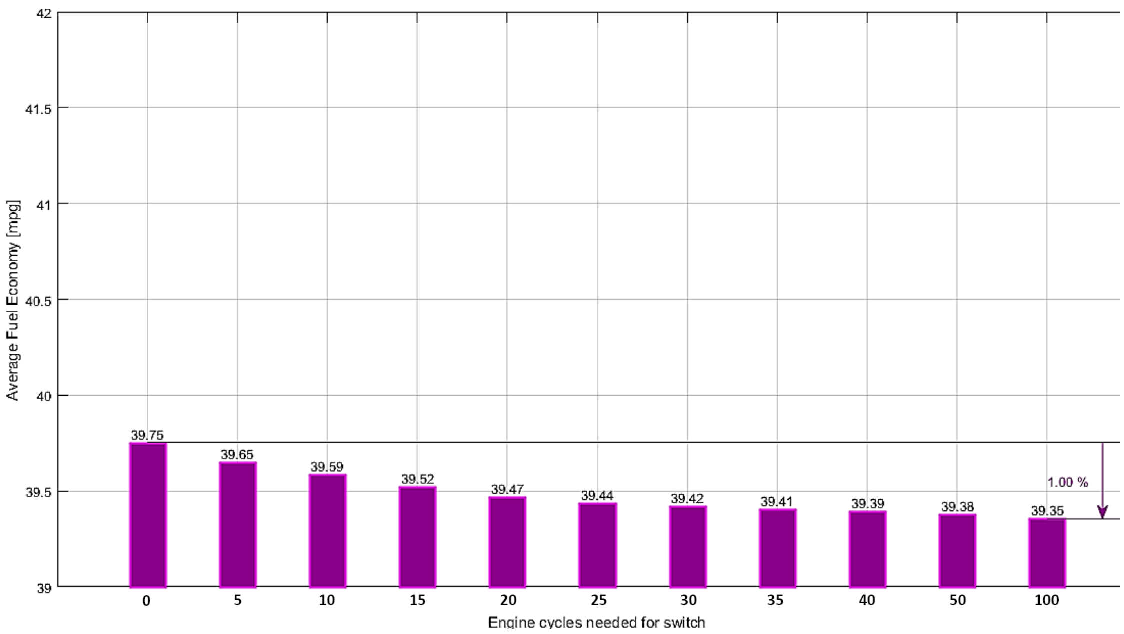


Figure 8.7: Comparison between average fuel economy over the WLTP driving cycle for different switching times - Vehicle with Manual Transmission

For sake of clarity and for a more intuitive representation the percentage decrease have been gathered in Figure 8.8 and Table 8.1.

	Manual Transmission	**CVT**
FTP	2,61%	2,52%
HWFET	0,53%	0,46%
NEDC	1,12%	1,05%
WLTP	1,00%	0,85%

Table 8.1: Numerical values of percentage decrease of average fuel economy [mpg] between Instantaneous Switch and 100 engine cycles needed for switch

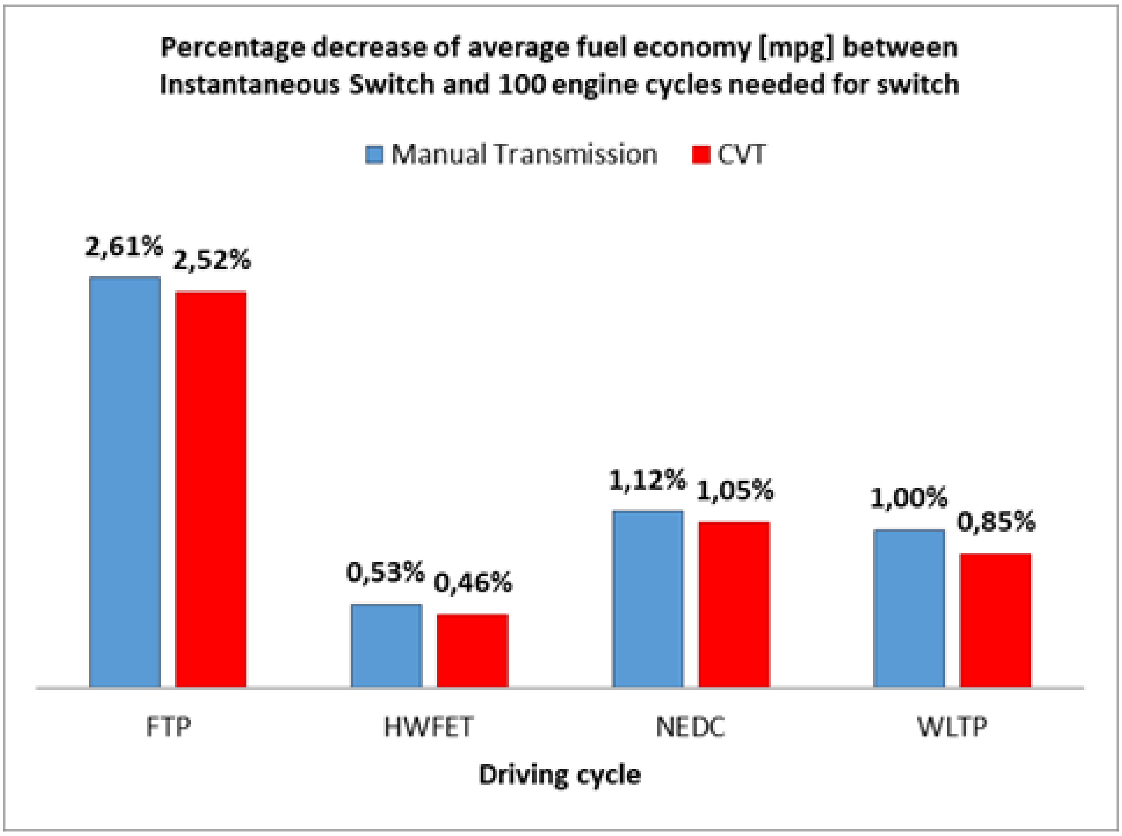


Figure 8.8: Percentage decrease of average fuel economy [mpg] between Instantaneous Switch and 100 engine cycles needed for switch

Looking at the results the following considerations can be made:

- The impact of non-instantaneous switch on average fuel economy is always slightly higher for manual transmission, if considering the minimum engine speed needed for shift equal to $1400 rpm$. It is worth to highlight that this percentage it is not linked to the fact that CVT has, in general, a lower fuel consumption because points with lowest BSFC are picked. This aspect rather leads to an overall higher average fuel economy that can be spotted comparing Figure 8.5, Figure 8.6, Figure 8.7 with Figure 8.2, Figure 8.3 and Figure 8.4 respectively. The percentage decrease value it is not affected by this phenomena, it comes from the number of incomplete or slow switches. Being the operating points chosen differently depending on the type of transmission, it means that the stricter chose in Manual transmission (dictated by the six gear ratios), leads to a series of events that make the non-instantaneous switch more important.

On the other hand this behavior is not absolute, in fact, if decreasing the minimum engine speed to 1100rpm, the trend is inverted in the case of FTP-75, leading to an higher impact for vehicle with CVT.

- As far as the impact of driving cycle is concerned, the analysis confirmed the correct working principle of the *Matlab* code since the higher the number of compression ratio switches required, the higher the impact on fuel economy. The total number of switches is reported in Table 8.2.

	CVT		**Manual Transmission**	
	htl and lth no of switches	**total no of switches**	htl and lth no of switches	**total no of switches**
FTP	92	184	94	188
HWFET	39	78	48	96
NEDC	17	34	21	42
WLTP	62	124	67	134

Table 8.2: Number of compression ratio switches associated to each driving cycle

Where htl and lth stands respectively for High-to-Low and Low-to-High, referred to compression ratios. The two numbers usually correspond since at the beginning the engine starts at low loads and engine speed where the con-rod is in the high compression ratio position. The same can be said for when the vehicle stops and the driving cycle is concluded. Hence every time that the switching line is crossed in one direction, a second one in the opposite one should be expected. The htl and lth number of switch can differ at maximum of one, if the analysis starts or stops in unconventional conditions.

By looking simultaneously at Table 8.2 and Figure 8.8 it can be noticed how the higher impact of non-instantaneous switching is corresponding to the FTP-75 cycle, since it shows the higher number of switching needed. On the other hand, the HWFET, being a cycle conducted on highway where the engine operating points are mainly gathered in the high part of the fuel map, the impact is pretty low.

An anomaly can be noticed on the NEDC since, even if it shows a small number of switches required, lead to a pretty high fuel penalty compared with the other cycles. The cause of such a behavior lays in the structure of the driving cycle and the incomplete switches. Also the transmssion model affects this result.

8.2 Control Delay

In the final stage has been required the introduction of a further level of complexity given by the control delay. This feature adapts more the code to the reality, giving the chance to the user to vary a new parameter and evaluating its effect on fuel consumption.

With control delay is indicated the time necessary to the control unit to send the signal to the mechanism to start the switch. Even if the model is referred to a 2-step Variable Compression Ratio engine with variable con-rod length and eccentric pin, this parameter is used in several others VCR engines systems. Again, it is required, through a dialogue windows showed when running the model, to insert a time interval in seconds representing this gap. A modification of the switching logic has been then necessary since the interval available for switch is reduced.

In order to have an intuitive plot of how the logic should be changed, a layout is reported in Figure 8.9.

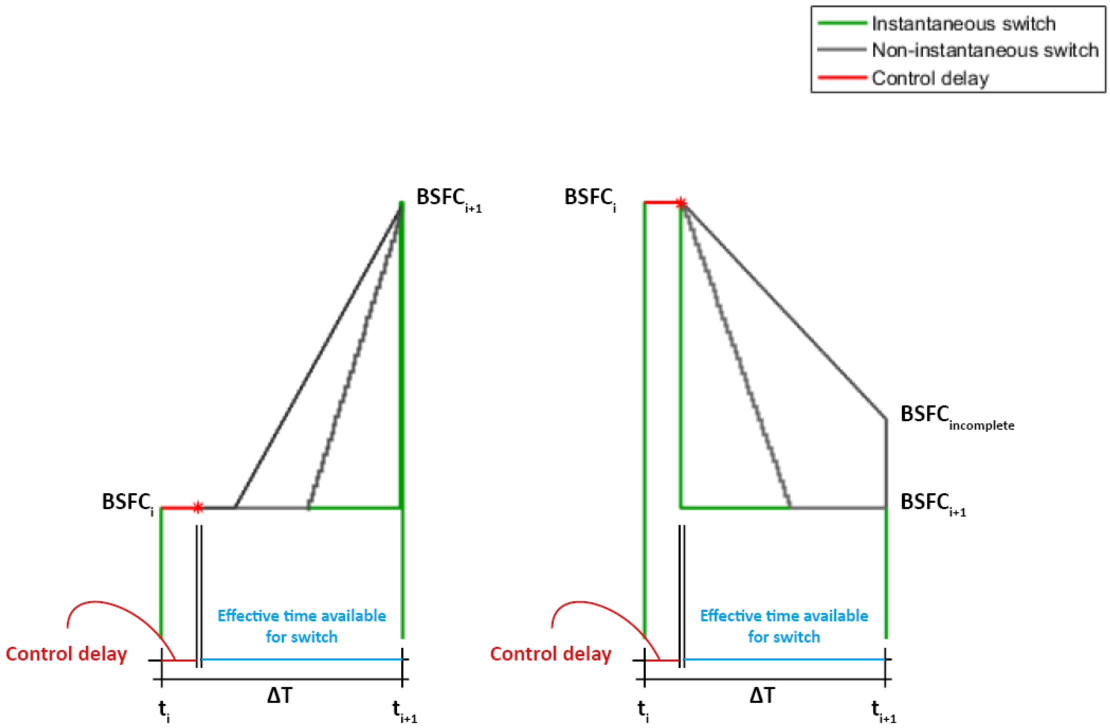


Figure 8.9: Introduction of control delay in the switching logic

In the Figure 8.9 the control delay has been exaggerated so to give to the reader a clear idea of what is happening. In real working of the mechanism the time interval needed for the signal to go from the controller to the actuator is really small, and it won't possible to appreciate it on the graph.

Basically, before starting changing the length in any of the two directions, some time is needed to start the switch. This delay should be considered also in the case of instantaneous switch. In general it is possible to subdivide the total time included between the point where the engine is operating and the one in which the pedal request should be fulfilled as:

$$\Delta T = \text{Control Delay } + \Delta T^* \tag{8.1}$$

Where with ΔT^* is indicated the effective time available for switching.

In the case where $BSFC_{t+1} > BSFC_t$, the switch should be started with a further delay in order to not generate a fuel penalty for faster switches. Hence the optimal time for starting changing the con-rod length is:

$$T_0' = \Delta T - \text{Control Delay } - T_s = \Delta T^* - T_s \tag{8.2}$$

On the other hand, when $BSFC_{t+1} < BSFC_t$, a fuel penalty is introduced since the engine operates at higher BSFC t for the time necessary to the signal to arrive to the actuator. In this case the switch starts at:

$$T_0' = T_0 + \text{Control Delay } \equiv t_i + \text{Control Delay} \tag{8.3}$$

In order to have an approximate idea of the impact of Control Delay on the fuel penalty related to the compression ratio switching three cases have been analyzed (reported in Table 8.3).

	Control delay
Case A	**0.005s**
Case B	**0.010s**
Case C	**0.020s**

Table 8.3: Control delays analyzed

As far as Case A is concerned, particular attention should be given to the time value. Inside the switching penalty logic there is a discretization process of the time interval in steps of 0.01s: a control delay of 0.005s is not compatible since it is smaller than the above mentioned step. Hence a change in the parameterization is needed; on the other hand an halving of discretization step would lead to a substantial increase of computation times, making the requirements on the code fall.

As far as Case B and Case C are concerned, the evaluations have been made on all the possible cases, changing both transmission and driving cycle. However, no appreciable changes in the average fuel economy over the cycle have been found.

In this case, the average fuel economy is a more effective parameter than the percentage decrease between instantaneous and non-instantaneous switch since, in this latter case, both parameters of the comparison are affected in the same way by the control delay. Hence, when computing the percentage decrease, no difference can be spotted.

Being the control delay covering a really small amount of time on the overall cycle, the effects are present but not aprreciable.

Concluding, this parameter has been inserted within the model in order to give to the user a more complete code, closer to the reality.

The final version of the switching penalty logic is resumed in the flow charts showed in Figure 8.10 and 8.11.

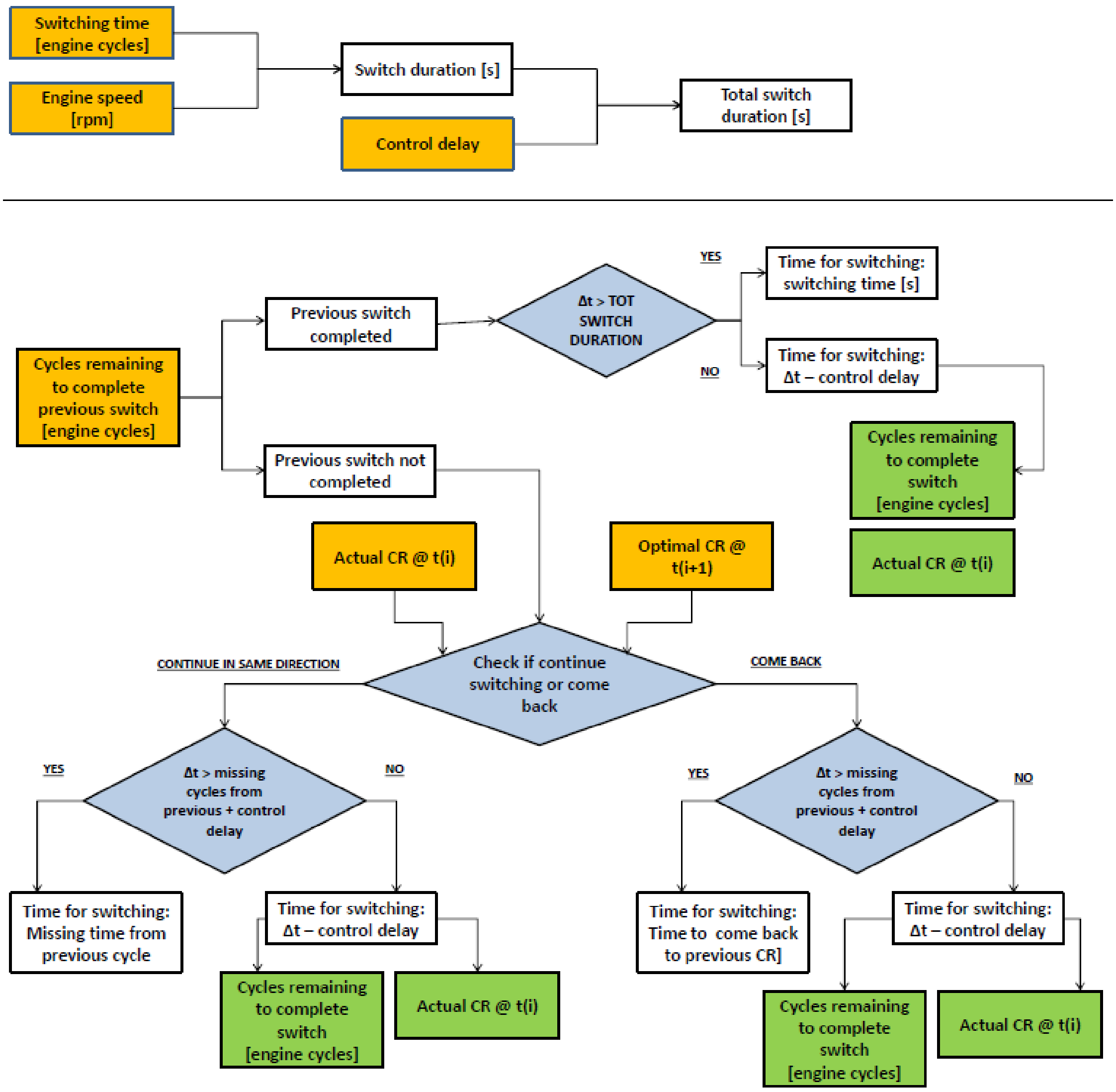


Figure 8.10: Switching Fuel Penalty Function - Flow Chart - Part 1

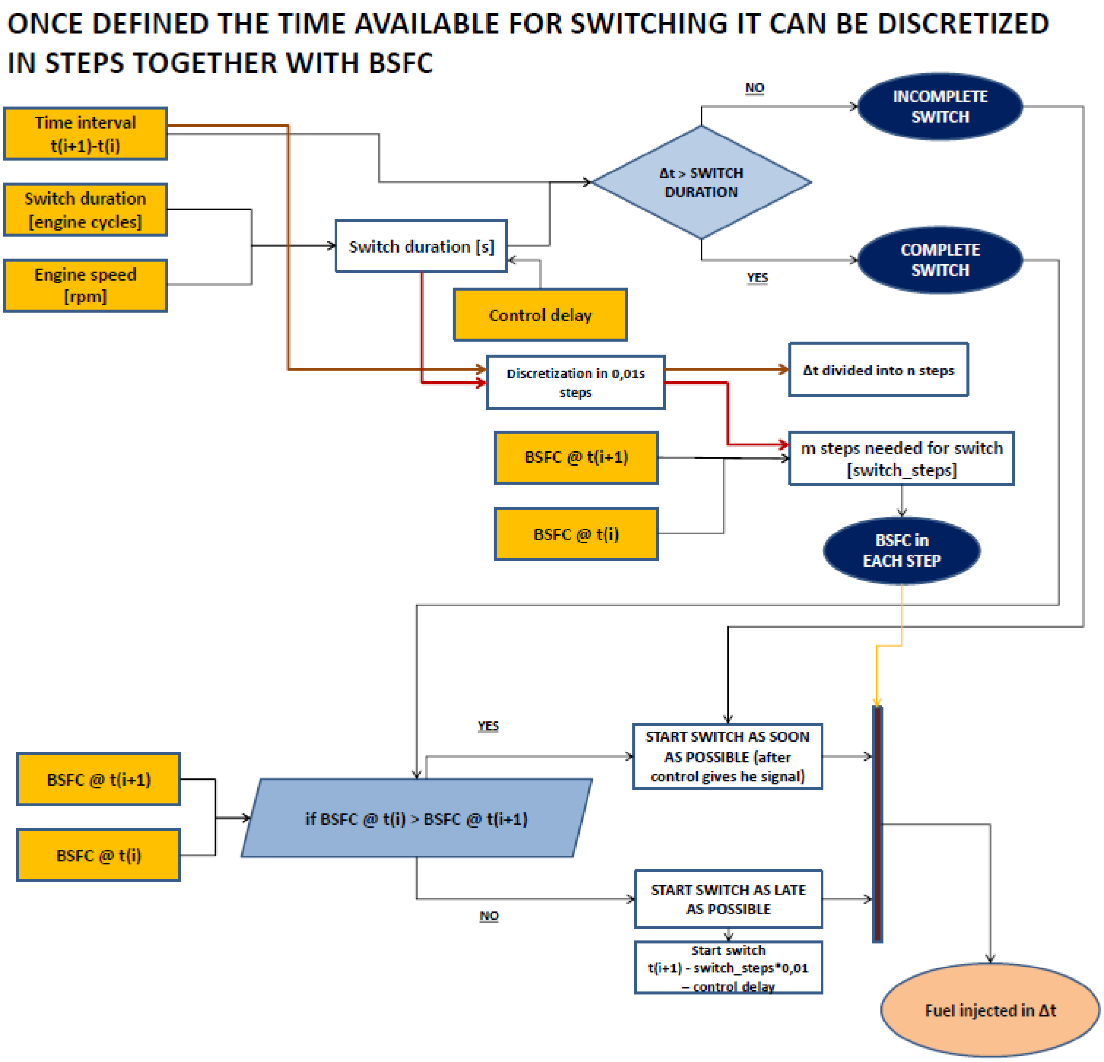


Figure 8.11: Switching Fuel Penalty Function - Flow Chart - Part 2

Chapter 9

Conclusions

Through this Master Thesis work, a mathematical representation of the fuel penalty behind compression ratio switching has been developed. Starting from an introductory part necessary to lay the foundation for such a model, a *Matlab* function capable of computing the fuel penalty associated to non-instantaneous switch between two subsequent operating point has been created. Inputs and outputs have been widely explained through the chapters: has been proved how by giving specifications only about the vehicle, the engine (fuel maps) and the driving cycle the code is able to provide in a prompt way the fuel consumption and the impact, expressed as a percentage, of the switching time.

The main focus of the whole project has been to create a model able to represent the distinctive and fundamental characteristics of the switching times in a 2-Stage Variable Compression Ratio and how they are linked to fuel consumption. The target has been reached in a pretty linear and clear way, through two discretization processes of both time and Brake Specific Fuel Consumption.

Particular attention has been given to the controller, so to represent in a proper way the problem logic: a slower switching must show an higher fuel consumption with respect to instantaneous one. Moreover, high efforts have been made in order to ensure that all the scenarios were considered. Turned out that several factors as BSFC of next operating, previous incomplete switches and control times had to be taken in account. In order to get to this result several attempt on the switching logic and modeling have been conducted. The code obtained at the end turned out to follow all the requirements desired in the preliminary study; in particular:

- The slower the switch the higher the fuel consumption;
- The higher the number of switches the higher the fuel consumption;
- Low computational times;
- Logic working just on the following operating point, no need for the code to know all the driving cycle in advance;
- No need for the user to change internal parameters.

Limitations

The numerical results reported are anyway affected by some bias, for several reasons. In first place, it hasn't been possible to give as an input to the code the real fuel maps of the engine: all the data available were referring to engines with fixed compression ratio or containing confidential information. Hence, it has been necessary to generate the two fuel maps of the same engine with two different compression ratios on GT-Power, process that led to assumptions, deviating the results from reality.

In addition, the engine simulated was not turbocharged making not possible to reach some power levels required from the driving cycles in specific conditions. On the other hand, the vehicle considered has turbocharging. The overall fuel consumption and switching strategy have been strongly affected by this restriction but, being the main focus the switching logic, the limitation has been overlooked.

In the end, also the modeling of Manual Transmission included some simplifications. No fuel penalties or delays in gear shifting have been inserted, therefore the behavior obtained is closer to an Automatic Transmission. Efforts have been made in order to make the code decision-making as close as possible to reality, imposing a gradual gear shifting, understanding when braking or deceleration is occurring, detecting idle conditions. The logic obtained has got nevertheless some aspects that cannot be corrected unless the whole driving and gear schedule is known in advance.

9.1 Future developments

Being this Thesis Work the first version of such a model, it can be refined and more characteristics can be added. Hereafter some of the main adjustments that can be made in order to get results with higher reliability are reported:

- Giving as input fuel maps obtained experimentally or, more in general, referred to the specific engine. In this way the numbers obtained as result can be analyzed on a practical basis;

- If the maps are generated through simulations an effective knock-predictive model should be inserted in order to understand how much the high compression ratio is disadvantageous and which are its limits.

- A map of switching times changing depending on the operating points can be generated and given as input. In other words, instead inserting a fixed duration expressed as function of engine cycles, an estimations of switching times on the operating map can be conducted and then given as input. Being the switching times correct and not assumed, the results will have more accuracy;

- Regenerative braking can be accounted since it impacts on fuel economy too;

- A better model of what happens when the switch is incomplete can be developed: during the intermediate steps, when the con-rod is neither fully extended or contracted, the switching line will move slightly, hence the controller should take in account this phenomena.

Concluding, it is worth to highlight again how the main target has been to provide a model that can be used by third parties for a deeper analysis or for a fast and qualitative idea of the non-instantaneous switching impact on the fuel consumption. All the intended requirements have been met in the developing of the code, and a fully-functioning and completely parametric *Matlab* function has been created. It can be inserted in other *Simulink* or *Matlab* models and, if the inputs are reliable, it will compute the fuel consumption due to compression ratio switching times in an exact way, following the mathematics explained.

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