GT-Power Modeling of a 6-Cylinder Natural Gas Engine and Investigation of the Possible Performance Improvements by Studying the Miller Cycle

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To Huma!

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Abstract

This master thesis has been conducted at the Division of Combustion Engines under the Department of Energy Sciences, Faculty of Engineering at the Lund University.

The purpose of this master thesis project is to create a computer based model of a Natural Gas Engine in order to raise its potential for updating a Volvo bus as well as creating a study of the improvement efficiency of the Miller Cycle.

In the last 100 years, there has been enormous progress in the internal combustion engine field. The emission of reactive Hydrocarbon from light duty vehicles is approximately. 70 percent lower in natural gas operation than with petrol operation. Emissions of greenhouse gases are 15-25 percent lower with in natural gas operation than from corresponding petrol vehicles.

Environmental improvement by reducing emissions as well as alternative energy issues will become more and more important in the future. Diesel engines have a higher efficiency level compared to gasoline engines, but the problem is that their NO_x and Soot emissions are higher than other types of engines. That's one of the main reasons why Natural Gas engines became more and more important over the last few years.

New gas buses that are run with vehicle gas emit approximately 50% less NO_x than corresponding diesel buses, something that can to contribute substantially to creating better air environment in the big towns. Moreover, the emission is reduced of carcinogenic pollutants and particles. [1]

The gas engine project at Lund University is exploring the extension of the performance, fuel efficiency and the stability of a spark ignition (SI) natural gas engine. The engine is a partly modified (modified from a single point to multi-port injection system), 6-cylinder turbocharged (9.4 liter) Volvo engine, which is equipped with EGR valves and an EGR cooler. The engine runs on a Stoichiometric mixture of air and fuel associated with the cost effective 3-way after treatment system.

The work presented in this thesis has focused on studying, with simulation, an engine model for new gas buses that is run with natural gas and a study with Miller Cycle. This research has been conducted in a 1-D gas exchange simulation program called GT-Power with an engine model originating from VOLVO. As well, during the verification of this model, testing cases were modified to produce different tests.

Nomenclature

ABDC	After Bottom Dead Center
BDC	Bottom Dead Center
BMEP	Brake Mean Effective Pressure
BSFC	Brake Specific Fuel Consumption
BTDC	Before Top Dead Center
CA50	Crank Angle of 50 % heat released
CAD	Crank Angle Degree
CI	Compression ignition
CO	Carbon monoxide
CO2	Carbon Dioxide
CR	Compression Ratio
DI	Direct Injection
EGR	Exhaust Gas Recirculation
EIVC	Early Intake Valve Closing
EVC	Exhaust Valve Closing
EVO	Exhaust Valve Opening
IMEP	Indicated Mean Effective Pressure
IVC	Inlet Valve Closing
IVO	Inlet Valve Opening
LIVC	Late Intake Valve Closing
MBT	Maximum Brake Torque
Nm	Newton meter
NOx	Nitrogen oxides

QLHV	Lower heating value
SI	Spark Ignition
TDC	Top Dead Center
WOT	Wide Open Throttle

Greek symbols:

λ	Air/fuel equivalence ratio
ϕ	Fuel/air equivalence ratio
η	Efficiency

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1. Introduction

1.1 Background

There are more than 1 million vehicles that run on natural gas in the world today. The most common gas engine in Sweden is a Volvo 9.6 liter diesel engine, converted to natural gas operation.

Recently, environmental improvements (CO_2, NO_X and ozone reduction) and energy issues have become more and more important as worldwide concerns. Natural gas is a good alternative fuel to improve these problems because of its plentiful sources and clean burning characteristics. Since the fuel system of CNG (Compressed Natural Gas) vehicles is completely closed, fuel evaporative emissions are practically eliminated. The evaporative emission control system commonly used in gasoline vehicles is unnecessary for CNG vehicles. In addition, un-used total reactive organic gas (ROG) emissions from fuel storage and refueling of CNG vehicles are small [3]. Therefore, CNG vehicles are very beneficial for environmental protection.

1.2 Objectives:

Natural gas has been the subject of growing interest as a low emission alternative to conventional automotive engine fuels.

The objectives of this thesis are to develop a GT-Powered model of the engine and investigate the possible performance improvements by studying Miller-Cycle.

1.3 Method:

The entire work is studied and evaluated in a 1-D simulation program called GT-Power, which is a software used to simulate engine capabilities. Computer simulation is a good way to evaluate if the research is worth investing in or not. Real experiments usually are costly and simulations are much cheaper. The engine model used in this work is a 6-cylinder Spark Ignition engine with a 1-stage supercharger, Exhaust Gas Recirculation (EGR) system and an Inter -Cooler system.

Since running the engine with the Miller Cycle needs variable valve timing (in order to reduce the effective compression ratio) and an appropriate turbocharger (or supercharger), modeling of the engine can be a cost effective and appropriate investigation tool. Modeling of the engine will be done with a GT Power environment, which is a powerful simulation tool for designing and building advanced engines.

2. The Internal COMBUSTION FUNDAMENTALS

2.1 Working principle

All the internal combustion principles that are dealt with in depth in this work are of four stroke types. Although of different types of combustion, the cycle can still be divided into the same four parts and repeated continuously as long as the engine runs. The four stroke cycle in an SI engine can be seen in Figure 1, where the piston and valve movement during (a) the intake, (b) compression, (c) expansion, and (d) exhaust stroke are shown.

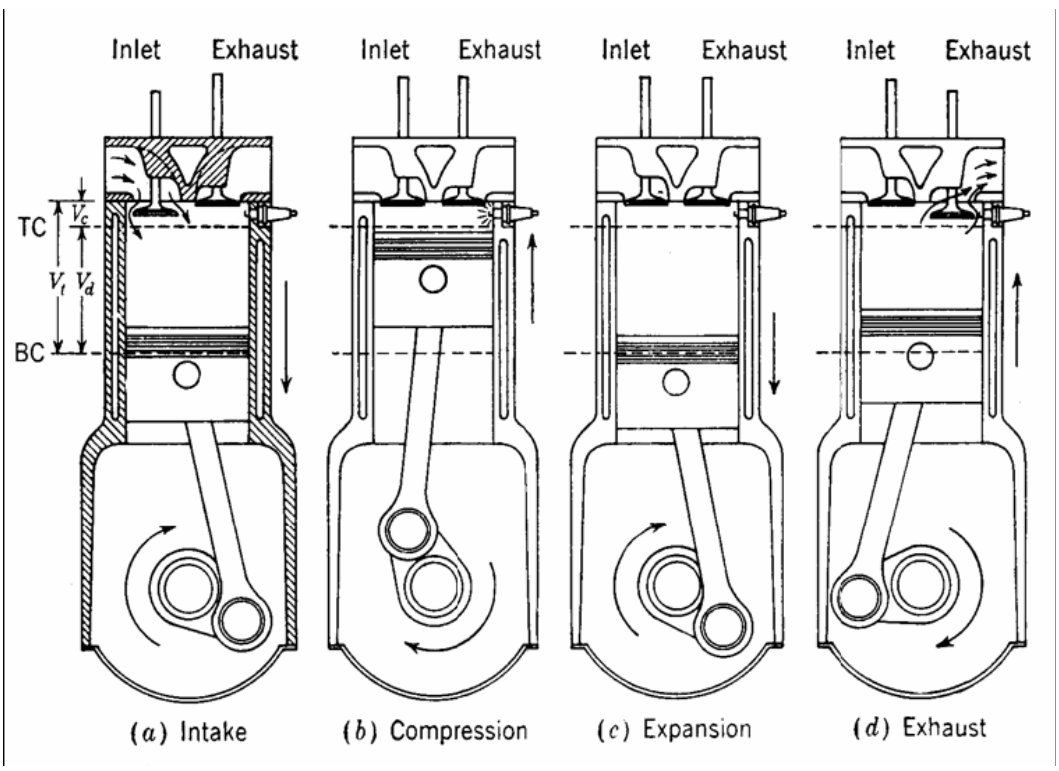


Figure 1: The four stroke principle for an SI-engine [2]

1. Intake stroke. During the intake stroke, the air and fuel is inducted into the engine through the open intake valves as the piston moves towards its lower position, bottom dead centre (BDC).

2. Compression stroke. In the compression stroke both intake and exhaust valves are closed and the charge is compressed as the piston moves towards its upper position, top dead centre (TDC).

3. Expansion stroke. Close to TDC the air/fuel mixture is ignited, differently depending on the combustion principle.

The combustion process usually occurs in the last part of the compression stroke and continues some time into the expansion stroke.

During the expansion stroke the gases, burned or unburned, are expanded and work is produced.

4. **Exhaust stroke**. During the exhaust stroke, the exhaust valves are open and the piston pushes the burned gases out from the cylinder. The four stroke described are repeated continuously as long as the engine is running.

Figure 2 shows a P_V diagram of an SI engine.

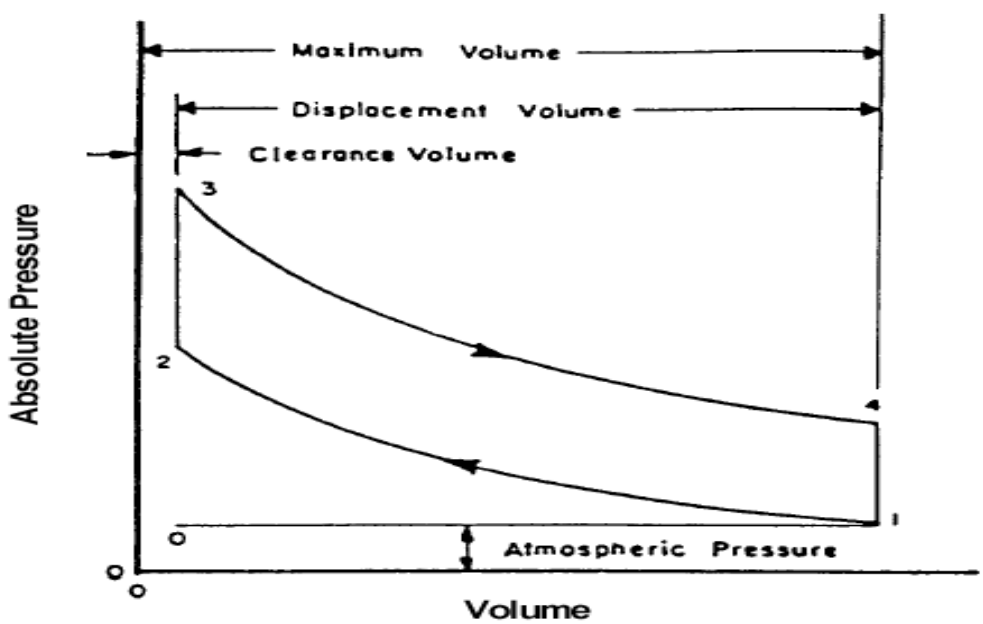


Figure2: Pressure vs. volume of the four strokes

- Point 0 to 1 – Intake
- Point 1 to 2 – Adiabatic Compression
- Point 2 to 3 – Constant-Volume Heat Input
- Point 3 to 4 – Adiabatic Expansion
- Point 4 to 1 – Blow Down
- Point 1 to 0 – Exhaust

2.2 The Spark Ignition Engine

In port fuel injected (PFI) SI engines, fuel and air are mixed in the intake manifold. The air/fuel mixture is then inducted into the cylinder during the intake stroke. As the fuel is mixed with air, the fuel is atomized, vaporized, and mixed with the air creating a fairly homogeneous charge. The charge is compressed during the compression stroke. Close to TDC the charge is ignited by a spark discharge, creating a turbulent flame which propagates through the combustion chamber, until it reaches the walls where it is quenched.

The load of the engine is controlled by throttling of the intake air, changing the flow rate of air into the engine. The mixture is kept close to Stoichiometric proportions at all loads, which results in high output power. At Stoichiometric mixture proportions, just enough air is supplied to combust all the fuel, neither more nor less. At part load, throttling of the air is necessary and leads to increased pumping losses since the engine has to produce work in order to induct the air/fuel mixture into the cylinder. This results in decreased part-load efficiency for the SI engine. The efficiency directly affects the fuel consumption, which results in higher carbon dioxide (CO2) emissions. The positive side of SI engines is the use of 3-way catalysts, which decreases substantially the emissions of carbon monoxide (CO), nitrogen oxides (NOx) and unburned hydrocarbons.

SI engines are mostly used in passenger cars due to high power density and low emissions, but with the penalty of high fuel consumption.

3. Generalities about Natural Gas Engines

A Compressed Natural Gas Engine works in similar as an SI Gasoline Engine, which has been described in the previous section.

3.1. Why is the Natural Gas Engine interesting?

The Compressed Natural Gas (CNG) Engine is a substitute for the gasoline (petrol) or diesel fuel engine.

Why are engines operated with natural gas? There are many reasons why natural gas is a good fuel. Natural gas exists in large quantities all over the world. The high methane content, CH4, (approximately 90%, depending on supplier) in natural gas makes the emission of the greenhouse gas carbon dioxide (CO2) lower than with petrol. A high octane rating allows for a higher compression ratio and thus increased efficiency. The high methane content means that most of the emitted hydrocarbon is methane, which is less harmful than more complex hydrocarbon from petrol compounds. It also emits less soot and particles than with diesel combustion.

This is cleaner fuel than other fossil fuels; also, natural gas is the cleanest burning fossil fuel since natural gas consists mostly of methane, which is a simple hydrocarbon. Coal and oil, the other fossil fuels, are more chemically complicated than natural gas, and when combusted, they release a variety of potentially harmful chemicals into the air.

While both are stored forms of natural gas, the key difference is that CNG is in compressed form, while LNG is in liquefied form. So, CNG has a lower cost of production and storage compared to LNG because it doesn’t require an expensive cooling process and cryogenic tanks. But CNG requires a much larger volume to store the same mass of natural gas.

Consequently, CNG can be used in Otto-cycle and modified Diesel cycle engines using a spark plug and a lower compression ratio.

There are many advantages in using a Natural Gas Engine:

A naturally aspirated natural gas engine would have lower power density than the corresponding petrol engine due to the lower density of the gaseous fuel. On the other hand, the high octane rating of natural gas allows a higher compression ratio and more turbocharging. These factors outweigh the density penalty of the gaseous fuel.

Secondly, Natural Gas Engine CNG price increases are a derivative of gasoline price increases and its price is in a range of 1/3 to 1/2 compared to gasoline in Europe. Third, methane is a

strong greenhouse gas. The High Hydrogen/Carbon ratio means that less CO2 is formed during the combustion compared to the same engine which is working with gasoline.

But there are also some drawbacks. Compressed Natural Gas vehicles require a greater fuel storage space than in conventional gasoline vehicles. This makes it difficult to design smaller vehicles that people are accustomed to operating.

In response to high fuel prices and environmental concerns, compressed natural gas is starting to be used more and more frequently in trucks and buses because the storage capacity is not a problem with these types of vehicles.

3.2 Lean Burn and Stoichiometric operations

A lean Burn Engine is characterized by operation with excess air and, a high lambda valve.

Lambda is defined as the ratio of the actual air quantity relative to the ideal Stoichiometric required quantity.

$$A = \frac{A/F_{actual}}{A/F_{st}} \qquad \text{Eq. 1}$$

A/F_{actual} = the actual air/fuel ratio

A/F_{st} = the Stoichiometric air/fuel ratio, just enough air to oxidize all the fuel.

Normal λ values in SI engines are between 0.9 and 1.4 (lean limit). The cycle-to-cycle variations increase rapidly at the lean limit, as well as the HC and CO emissions. The spark does not have enough energy to ignite the mixture, or the combustion becomes so slow and cold that it is quenched.

With this type of operation, a lower exhaust temperature is observed, which could be a benefit when working with a natural gas engine, which is actually a modified diesel engine. Higher efficiency is obtained with lean operations than stoichiometric operations because the throttle will be more open for the same load (so less pumping losses) and there will be less heat losses during the combustion (due to the colder combustion temperature).

The test engine of this study operates at stoichiometric conditions. A stoichiometric engine operates at λ =1. The reason for this is that a three-way catalyst can then be used to reduce HC, NOX and CO simultaneously.

The positive side of SI engines is the use of 3-way catalysts, which substantially decrease the emissions of carbon monoxide (CO), nitrogen oxides (NOx) and unburned hydrocarbons. A modern spark ignition (SI) engine with a three-way catalyst emits very low amounts of hazardous emissions, mostly water and carbon dioxide (CO2), if driven according to the certifying cycle. CO2, which is a greenhouse gas, can be reduced in various ways; e.g. by improving fuel economy, using a fuel with a higher hydrogen to carbon ratio (H/C) or using a renewable fuel.

One of the main disadvantages is lower efficiency, because there are pumping losses at part load due to the throttle and because there are more heat losses due to the higher combustion temperature. This drawback can be reduced by running the engine with stoichiometric mixture diluted with exhaust gas.

3.3 Natural Gas DATA

Natural gas fuel exists naturally in the earth's crust and consists mostly of methane (CH4). Other components are ethane (C2 H6), propane (C3H8), butane (C4H10), nitrogen (N2), carbon dioxide (CO2) and oxygen (O2). The composition varies between different sources, and also in time.

The composition of natural gas, which varies slightly over time, is shown in table 1. The lower heating value is 48,4 MJ/kg.

Table 1 shows the natural gas composition.

Compositions	%	Structure	a	b	A	B
Methane	89,84	CH4	1	4	89,84	359,36
Ethan	5,82	C2H6	2	6	11,64	34,92
Propane	2,33	C3H8	3	8	6,99	18,64
I-Butane	0,38	C4H10	4	10	1,52	3,8
N-Butane	0,52	C4H10	4	10	2,08	5,2
I-Pentane	0,11	C5H12	5	12	0,55	1,32
N-Pentane	0,07	C5H12	5	12	0,35	0,84
Hexane	0,05	C6H14	6	14	0,3	0,7
Nitrogen	0,27	N2	0	0	0	0
CO_2	0,6	CO2	1	0	0,6	0
Sum	99,99				113,87	424,78
					1,1387	**4,2478**

Table 1: The composition of the natural gas used for this study

4. EXPERIMENTAL APPARATUS

The objective of this study is create a model natural gas engine with GT-Power and compare the model with an authentic engine to investigate higher performance potentials, then run it with the Miller Cycle for improvement efficiency.

4.1 Engine Specification

The multi-cylinder engine is of the same type as the single-cylinder engine, and the engine is a six-cylinder 9.4- liter turbocharged engine. The experimental engine was originally a

diesel engine from Volvo which was converted to a natural gas engine. (See Table 2 for specifications.) The engine is equipped with a short route cooled EGR system and also a turbocharger with waste gate.

Displacement	**9,4 L**
Number of Cylinder	**6**
Bore	**120 mm**
Stroke	**138 mm**
Compression Ratio	**10,5:1**
Number of valves	**4/cylinders**
Ignition sequence	**1-5-3-6-2-4**
Fuel	**Natural gas**
Valve Lift Intake	**40 mm**
Valve Lift Exhaust	**41 mm**

Table 2: specifications of the engine

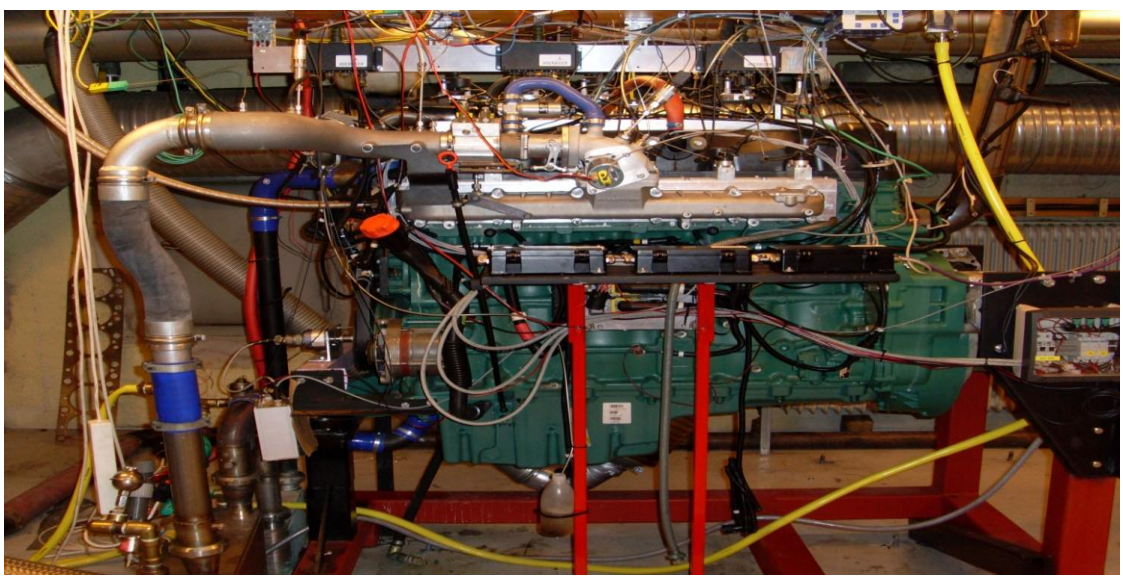


Figure 3: General view of the Volvo M9G Engine

4.2 The Test Rig

To start the engine, the motor is coupled to an AC motor equipped with a dynamometer and a control unit. It can motor or brake the engine in order to achieve the required engine speed (Figure 3).

Figure 4: View of the engine coupled to the dynamometer

4.2.1 Multi-Port Injection System

The port injection system for the test bench engine is supplied with natural gas at 4.6 bars, So the port injection system is equipped with 12 injectors (2 per cylinder) in order to cover the whole load range, as shown in Figure 5. This also makes it possible to operate the engine with two different gaseous fuels simultaneously. An extension of the intake ports prevents cross breathing of natural gas between cylinders at high loads. The total volume of each intake port is slightly larger than half the displacement volume per cylinder. [11]

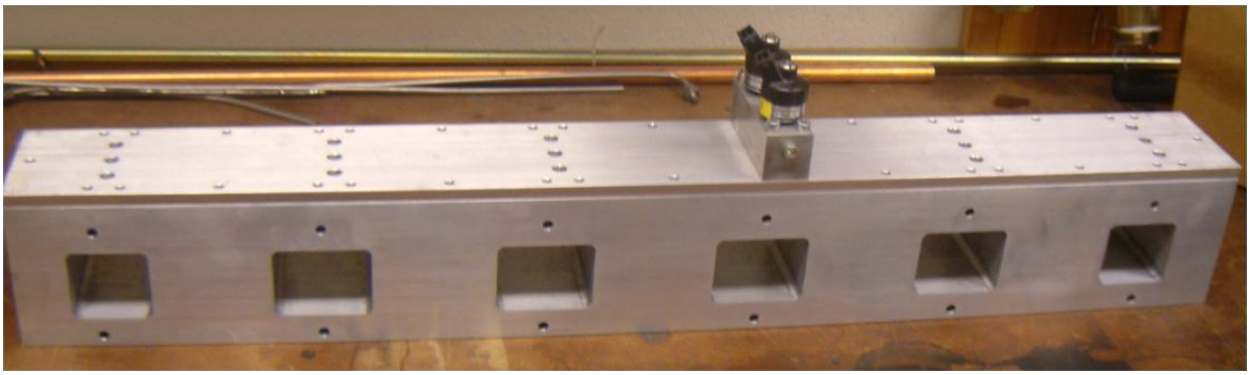


Figure 5: The principle of the port injection system. [11]

4.3 Engine Control System

The engine control system is the brain of the engine system. The most difficult part in converting a diesel engine into natural gas SI operation is the design and optimization of the control system. The engine performance is strongly dependent on the design of the control system. The fuel and ignition system are equally important to the operating characteristics. A high power ignition system is needed at lean operation, to deliver sufficient energy so the mixture ignites. A master PC based on GNU/Linux operating system is used as a control system. It communicates with three cylinder-control modules (CCM) for cylinder-individual control of ignition and fuel via CAN communication. (See Figure 6.)

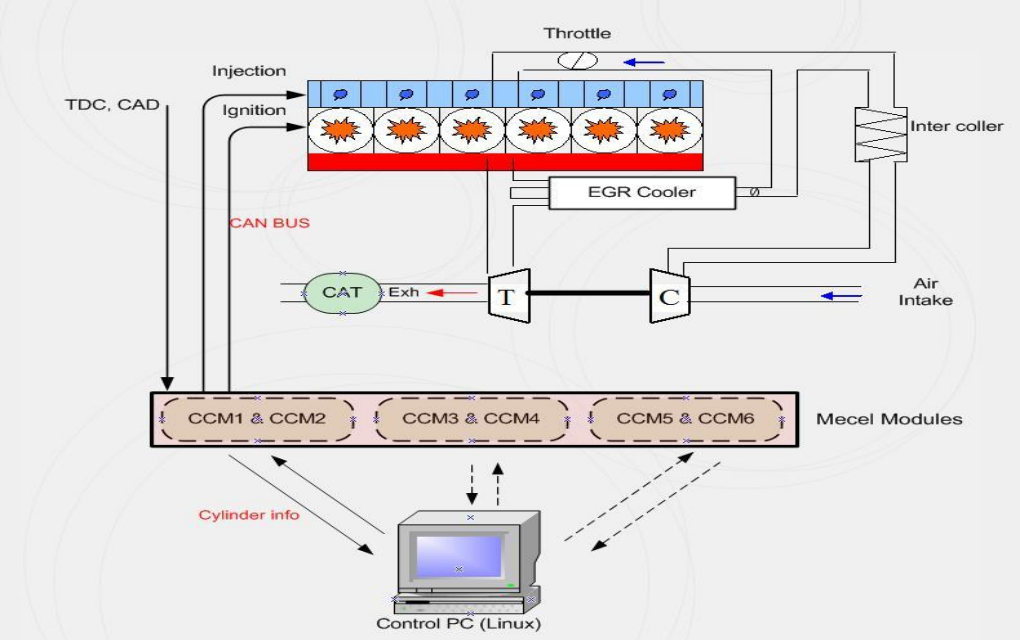


Figure 6: The engine and its control system. [11]

4.4 EGR System

A long-route cooled EGR (Exhaust gas recirculation) is used in the engine. An exhaust-gas heat exchanger is used to cool the EGR. Water from a buffer tank, with water maintained at a constant temperature, is circulated through the heat exchanger to the EGR temperature (approximately 60° C). Both hot and cold water is connected to the buffer tank. A throttle on the inlet of the exhaust-gas side of the EGR cooler controls the amount of EGR delivered to the engine. A throttle at the end of the exhaust pipe is used to further increase the amount of EGR (If the EGR-throttle is fully open and not enough EGR is delivered). The amount of EGR is computed according to [5].

$$\%EGR = \frac{CO_{2\ inlet}}{CO_{2\ exhaust}} * 100[\%\text{-vol}] \qquad \text{Eq. 2}$$

Where CO_2 inlet is compensated for the injected fuel according to:

$$CO2,_{\text{inlet}} = \frac{CO_{2,inlet,measured}}{1+\left(\frac{1}{AF_S.\lambda}.\frac{M_{AIR}}{M_{FUEL}}\right)} = \frac{n_{air}}{n_{air}+n_{fuel}}\ CO_{2,inlet,measured} \qquad \text{Eq. 3}$$

Where:

$CO_{2,\ inlet\ measured}$= the measured connection of CO_2 in the intake, before the fuel is injected.

AF_s = stoichiometric air/fuel ratio

M_{AIR} = molecular weight for air

M_{FUEL} = molecular weight for the fuel

m_{Air} = moles of air

m_{Fuel} = moles of natural gas (fuel)

5. Description of the software

A brief description of GT-Power [4] will be given in the following chapter.

5.1 GT-POWER

GT-Power is a 1-D- simulation program from Gamma Technology, which simulates pressure, temperature and mass flow in different parts. This program is a part the main program GT-Suite.

GT-Power is designed for steady state and transient simulations suitable for engine/power train control analysis and can be used to simulate all kinds of I.C engines. The software uses one dimensional gas dynamics to represent the flow and heat transfer in the components of the engine model. The user constructs the model by dragging and dropping objects in the graphical user interface GT-SUITE, where the component database offers a broad range of engine components. After linking the components with connection objects the user may define properties for each component, setting up simulation options such as convergence criteria and specify desired output plots before running the simulation.

In GT-Power one can build models of engines that are very close to reality. To make the model work as the real engine, it is most crucial to imitate the real engine down to the smallest pipe and angle. To model the engine in GT-Power there are objects like cylinders, crankcases, pipes, turbochargers and so on, that are easy to modify by desire. For these objects there is sometimes a need for reference, which describes the object more in detail to the program.

5.2 Engine model

5.2.1 Design

The engine model used in this master thesis is a 6 cylinders turbocharger SI engines that is run with natural gas used for buses. The engine was fitted with a turbocharger, an intercooler and an EGR-system.

5.2.2 Parts and function

5.2.2.1 Turbocharger

The turbocharger consists of three main parts that the efficiency, the turbine, the compressor and the bearings. The turbine and compressor are mounted on the same shaft and rotate with the same angle velocity. The turbine is driven by the energy available in the exhaust gases and the compressor increases the inlet air density prior to each cylinder. A waste gate valve, which bypasses exhaust gases around the turbine, controls the boost pressure.

The maximum power output an engine can deliver is limited by the amount of fuel that can be burned efficiently in the cylinder. The mass of fuel that can be burned is depended of the amount of fresh air that is inducted during each cycle. By using a turbocharger more mass of air can be inducted into each cylinder. The fuel economy of a turbocharger engine is influenced by the same factors as the naturally aspirated engine. The compression ratio is of great importance, a turbocharged engine makes a reduction of the geometric compression ratio compared to a naturally aspirated spark ignition engine necessary. There is relation between the boost pressure and the geometric and effective compression ratio. A high boost pressure requires a low geometric compression ratio, which leads to an increase in the fuel consumption [9].

5.2.2.2 Intercooler

In the engine the intercooler is used to cool the air that comes from the compressor, because when using a turbocharger the air density increase and so the inlet temperature will increase, also. The intercooler is an air-air cross flow heat exchanger. The cooler consists of a big volume with many small pipes where the internal air is flowing through. In the model, the main part of the intercooler model will consist of a multiple pipe object. The wall temperature can be adjusted in the main folder of the intercooler in the cooler part properties. These items can be adjusted to match a certain experimental pressure and temperature drop. Figure 7 shows the modeling of the intercooler.

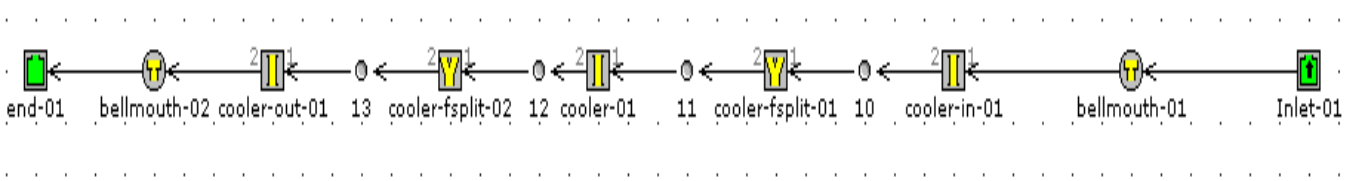


Figure 7: The Modeling Intercooler

5.2.2.3 EGR-system

Some of the exhausts that come out from the engine are recycled through the EGR-system. The EGR flow is regulated by valves. In this system the gas is cooled by a water-air upstream flow heat exchanger and is then, together with the inlet air, led back into the engine. EGR- systems are used to lower the temperature in the cylinders, and through that reduce NOX formation. In addition, exhaust gas recirculation can also help to reduce the fuel consumption in the part load range. The amount of recirculation exhaust gas has to be varied depending on engine load and thus the model includes an EGR valve.

The model for the EGR system shown in Figure 8 was obtained from Volvo Powertrain.

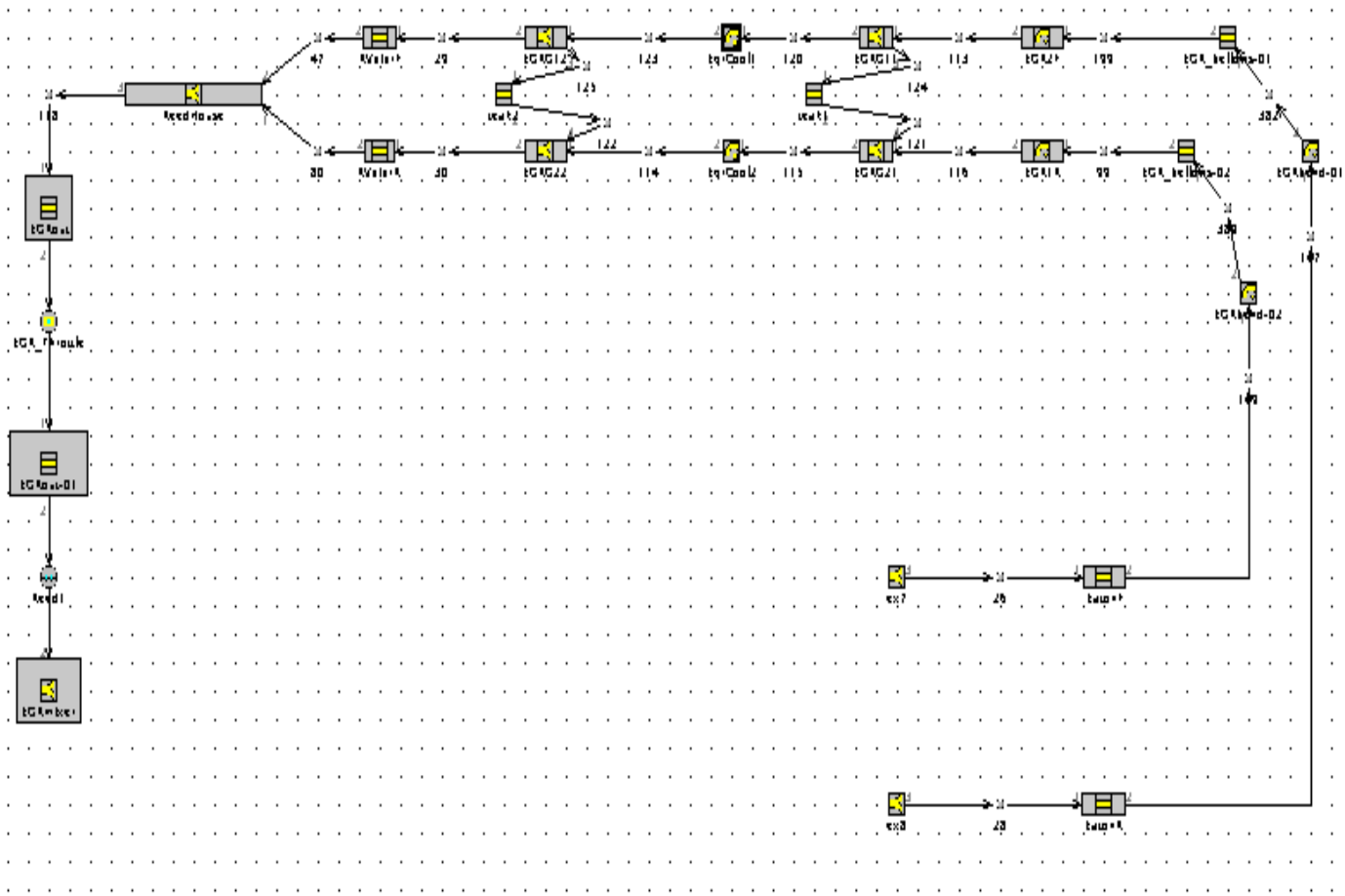


Figure 8: The modeling of the EGR- system

This is a general view of the EGR- system. The most important parameter to adjust is the EGR Throttle Discharge Coefficient in order to have the right percentage of exhaust gases when opening the EGR valve.

5.3 The original engine model

The original engine model is made to simulate a 6 -cylinders turbocharged SI engine with EGR, and a intercooler.

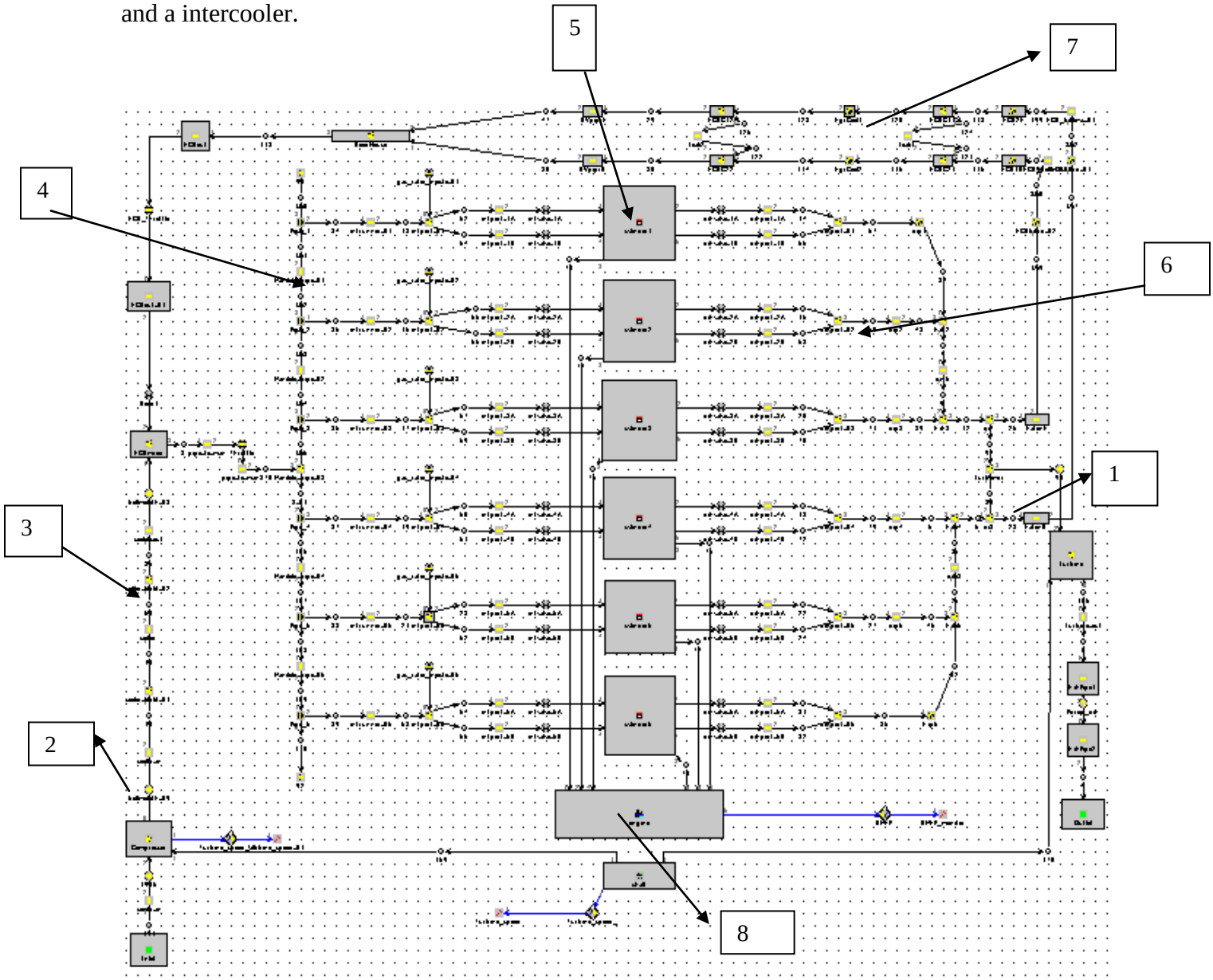


Figure 9: The verified engine model of GT-Power. 1: Turbine, 2: Compressor, 3: Intercooler, 4: Inlet manifold, 5: Cylinder, 6: Exhaust manifold, 7: EGR-system, 8: Shaft.

6. MILLER CYCLE

6.1 The Miller Cycle

The Miller cycle [7] represents a modification of the Otto cycle where the compression process is shortened relative to the expansion process by early or late intake valve closing in order to achieve higher efficiency and power output.

The Miller Cycle was first proposed by R.H. Miller in 1947. The proposal was for the use of early intake valve closing (EIVC), in order to provide internal cooling before compression, therefore to reduce the compression work. Miller further proposed increasing the boost of the inlet charge to compensate for the reduced inlet duration. The cycle that Miller proposed is a cold cycle which has allowed an increase in engine performance with increase of the knocking threshold. At that time, the Miller Cycle was focused on improving the thermal efficiency of the engine. This is still the aim. Since the Miller Cycle is a cold cycle, there is the possibility of applying it to reduce the combustion temperatures in engines thus reducing the NOx formation and emissions. [8]

The goal of the Miller Cycle is to run the thermodynamic cycle at a higher expansion ratio to improve the efficiency (lower exhaust temperature) through EIVC or LIVC.

The Miller System is particularly attractive for natural gas engines, since compression temperatures may be reduced, allowing much higher compressor pressure ratios, or engine compression ratios, to be used without causing combustion knock. Another advantage as a result of the lower cylinder temperatures during engine process is the lower thermal loading of the engine. The lower cylinder temperatures also bring an advantage to a small NO_x formation with itself, which is more important from a environmental point of view (Watson and Janota, 1982; Pischinger et al., 1989). [9]

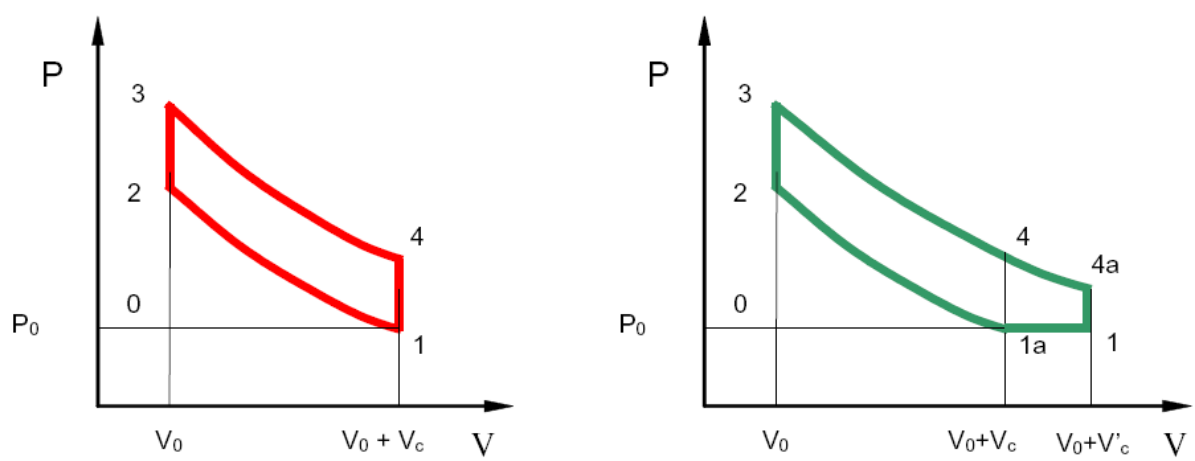
6.2. The concept of the Miller Cycle

6.2.1 Description of the Miller Cycle

For the Miller Cycle, the expansion-ratio exceeds its compression-ratio [10], which is the effective expansion stroke of the engine is longer than the compression stroke.

A comparison of the standard Otto Cycle with the Miller Cycle is shown in Figure 10. We can assume the cylinder pressure at the starting point 0 is P_0, and the volume is V_0, the sweep Volume of the cylinder for the Otto Cycle is V'_c and for the Miller Cycle is V'_c. As shown in Fig. 10a, the work processes of the Otto Cycle are: intake process 0→ 1, compression process 1→ 2, combustion and expansion process 2 → 3→ 4, and exhaust process 4 →1→ 0. For the cycle, the compression- ratio is identical to the expansion-ratio; a higher expansion-ratio causes a higher compression-ratio.

However, the Miller Cycle allows the compression and expansion ratios to be preset independently, as shown in Fig.10b. The work processes are: intake process 0 → 1a → 1; then an additional "intake blow-back" process 1→1a, which is the main difference between the Miller Cycle and the Otto Cycle; compression process 1a→2; combustion and expansion process 2→ 3→ 4 → 4a; and exhaust process 4a→ 1→ 1a→ 0. From the P-V diagram of the Miller Cycle, it can be seen that higher engine efficiency is expected with an increased expansion-ratio because more heat is changed to mechanical power. This was the original idea behind the Miller Cycle. [8]



(a) Otto cycle (b) Miller cycle

Figure 10: a comparison of the standard Otto Cycle with the Miller Cycle [8]

6.2.2 Main methods to realize the Miller Cycle

There are three main methods to realize a Miller cycle in practice [8]: (a) installing a rotating valve between intake manifold and intake valve (on the cylinder head) to control the intake air quantity – this is called the early rotary- valve closing (ERVC): (b) this is called referred to as the closing the intake valve before the termination of the intake stroke- early intake valve closing(EIVC); and (c) keeping the intake valve open during a portion of the compression stroke, thus rejecting part of the charge and reducing the net compression ratio – late intake valve closing (LIVC- as show fig 11).

For this experimental study, the LIVC and EIVC of the Miller Cycle were selected. Three versions of the LIVC and two versions of the EIVC Miller Cycle were designed and tested; the detailed parameters are presented in Section result.

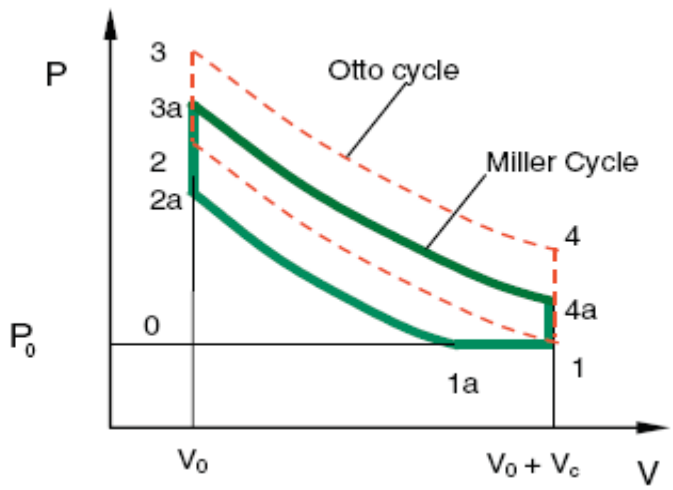


FIGURE 11: A comparison P-V diagram of the Otto Cycle and the Miller Cycle. [8]

6.2.3 Analysis of questions concerning the Miller Cycle

Which modifications should be analyzed with regard to converting the engine to the Miller Cycle?

When converting or building an engine to work with the Miller Cycle, different variables shall be analyzed. A first approach is to determine if the late or early inlet valve closing shall be applied. Regarding that the valve timing is going to be altered and a turbocharger with high boost pressure will be chosen.

The turbocharger and the valves shall be matched appropriately according to the turbo manufacturer. This implicates that the cam lobe profiles may have to be altered. A higher intake charge pressure needs to be matched with new valve opening characteristics. When converting an engine the valve train system may also need some modifications to achieve proper valve operation, such as valve lift, valve seating velocity and valve spring constant. Also, the ignition timing may be altered because the intake air is cooler and therefore a greater knock margin has been accomplished.

A comparison between early and late Miller Cycles.

A negative effect when applying the LIVC is that the intake charge air temperature is increased because of the backflow. The backflow occurs when the pressure generated by the piston is higher than that of the turbo charger. This dictates that the piston forces some inlet air back to the manifold. A reason for concern is that the increased inlet air temperature can increase the risk of engine knock.

EIVC can have another particular negative effect: with IVC a vacuum is created in the cylinder room causing a suction of lubricant oil on the cylinder wall up by the piston. Concerning the turbocharger, there is a higher demand placed when applying EIVC. LIVC hasn’t as high a demand on the turbo and can at the same time also have an increased compression ratio compared to EIVC.

The EIVC requires a more extensive modification due to changes on the camshaft, valve train etc. This is because the IVC is open during a shorter time so the valves open/closing time is decreased. i.e. the valve speed is increased. Since the IV is open during a shorter time, the turbocharger has a greater demand. The right amount of charged air shall be supplied at a shorter time.

Which are disadvantage with the Miller Cycle?

As a criterion to accomplish a sufficient efficiency with the Miller Cycle concept, a high intake charge pressure is needed. A turbocharged engine inherently has “turbo lag” during power

increase from the time when the engine is operated under the naturally aspirated condition until the turbocharger has become fully functional to increase the intake charge pressure. Having a poor load response is typical with an engine that produces high thermal efficiency because on these engines the intake charge pressure is generally high. [6]

A negative effect when applying the LIVC is the intake charge air temperature is increased because of the back flow. The backflow occurs when the pressure generated by the piston is higher than that of the turbocharger. This implicates that the piston forces some inlet air back to the manifold. The reason of concern is that an increase of temperature of the inlet air occurs and these increases the risk of combustion knocks.

The Miller Cycle is a possible frontier of the internal combustion engine industry. The greatest obstacle for the cycle to overcome is the additional costs it incurs from the supercharger. In a consumer-dominated world, the problem must be dealt with in order to make the Miller Cycle engine truly competitive.

6.2.4 Conclusions

The Miller Cycle has the compression process shortened by either closing the intake valve early or late. With the Miller Cycle it's possible to reduce the temperature of the cylinder charge. The immediate advantage is the thermodynamically favorable increase of the compression ratio and an earlier firing point and/or extension of engine operation at high BMEP's to smaller methane numbers. The expansion process is unaltered. The performance characteristics of the Miller Cycle show that there are both advantages and drawbacks. The advantages are improvement of efficiency and reduction of emissions; the ,drawbacks of the Miller Cycle are that due to creating a volumetric efficiency loss due to the early or late intake valve closing, very high turbocharger speeds are reached and knock sensitivity seems to be worse because of higher temperatures before ignition by the Miller Cycle.

7. Results

7.1 Modifications

The measured data consisted of several cases. From these cases only three data profiles were chosen. These types of tests were made in order to tune the model to the real engine: EGR, Lambda and Speed sweep.

This time the pressure and temperature in the Intercooler was set to measure data out from the compressor in the real engine.

The compressor was then put back onto the model, and forced to turn at the measured rotational speed. To get the proper mass flow through the compressor, the compressor's mass flow multiplier was tuned, which made the compressor larger. The next step was of course to do the same thing for the turbine.

The temperature before the turbine was used turned out to be very high. To solve this, the exhaust valves and collector were split up. In the real engine the exhaust collector is cooled by the air while the exhaust ports are cooled by water. In the model, the heat transfer coefficient was changed for the ports to be cooled by the same water as in the cylinder.

Now, the only thing that had to be done before the entire engine was put together with the turbocharger was to compare the power between the turbine and compressor. The power should be the same for both, but if that was not the case the efficiency multiplier could be turned-on either the turbine or compressor to get the same power. Thereafter, the expectation is that when every part is put together for an entire engine, the engine should show nearly the same data as in the reality, or the difference should be just a tuning of the EGR valves.

7.2 Results compared to measured data

During the whole verifying process, we have different tests of both real engine and simulated model. The three cases used in this test were EGR, Lambda and Speed sweep.

7.2.1 EGR Sweep

The first type of test is the EGR sweep. The engine is operated at 1300 rpm, with different EGR valve openings, but the Lambda is a constant in this test. The throttle opening in this test is 40°.

This EGR measured data case is shown below in Table 3.

Engine Speed	1300	1300	1300	1300	1300	1300
Throttle Angle	40	40	40	40	40	40
EGR valve opening %	0	5	10	30	50	90

Table 3: the EGR map

Figure 1 below shows the EGR flow for both the real and modeled engine. In the first four cases there is almost no difference in EGR flow, but in the two lasts cases it differs considerably. The problem with the last case is that the EGR valves are half or fully open, but in spite of this, the EGR flow is inefficient.

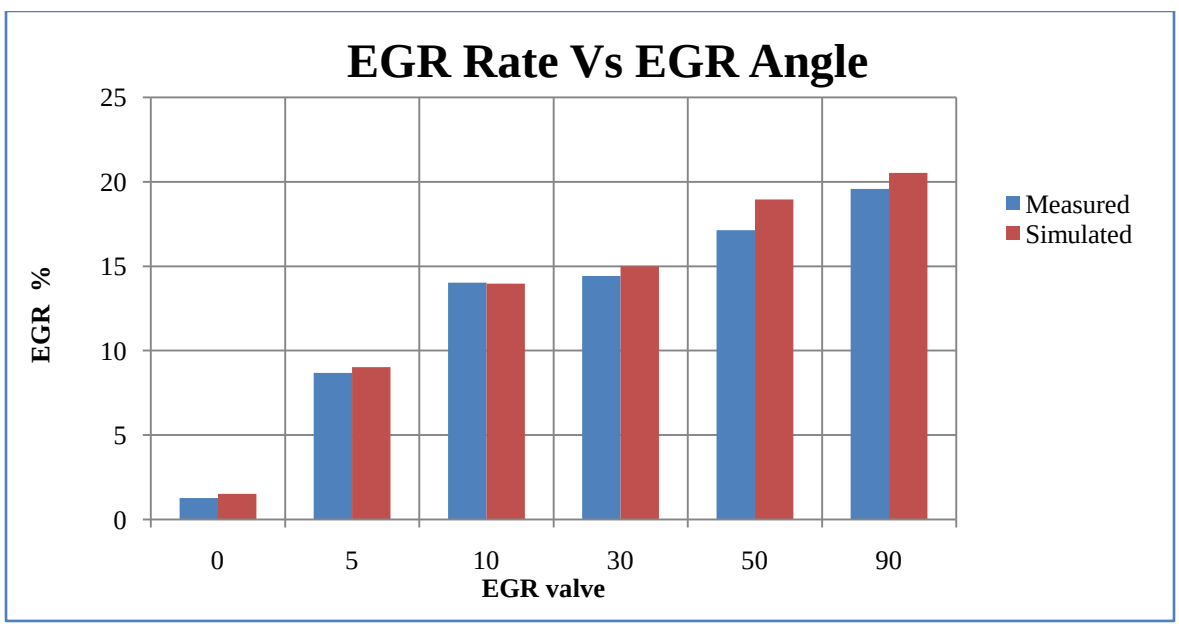


Fig 12: EGR Rate versus EGR Angle

7.2.2 Lambda Sweep

The second type of test is the Lambda sweep. The goal of developing a low emission heavy-duty SI engine CNG can be achieved by following basic approaches of the Stoichiometric engine with TWC (three-way catalyst).The engine is like the EGR sweep operated at 1300 rpm and with a different Lambda between 0.9 and 1.5, but without EGR in this test. The throttle opening in this test is 40°.

The Lambda measured data case is shown below in Table 4.

Engine Speed	**1300**	**1300**	**1300**	**1300**	**1300**	**1300**	**1300**
Throttle Angle	**40**	**40**	**40**	**40**	**40**	**40**	**40**
Lambda	**0,9**	**1**	**1,1**	**1,2**	**1,3**	**1,4**	**1,5**

Table 4: the Lambda map

In order to validate the working conditions, a comparison of the mass fuel flow and the mass air flow has been done between the real engine and the simulation model.

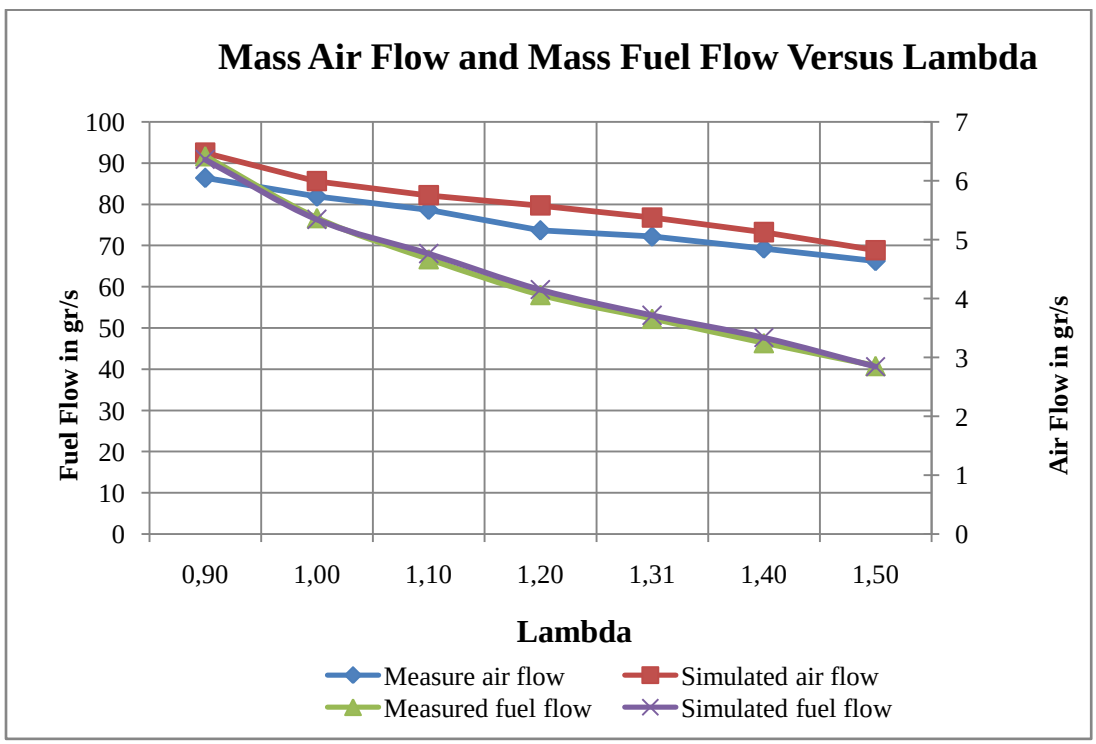


Figure 13: Mass Air Flow and Mass Fuel Flow versus Lambda

7.2.3 Speed Sweep

The last type of test was a Speed sweep. The goal of this test was to experiment with higher loads and increased speed. The engine is operated at a constant Lambda and without EGR in low speed, but we have EGR in last three cases.

This speed measured data case is shown below in Table 5.

Engine Speed	800	1000	1100	1200	1300	1500	1600
Throttle Angle	90	90	90	90	90	90	90
EGR valve opening %	0	0	0	0	5	17	100
Lambda	1	1	1	1	1	1	1

Table 5: The Speed map

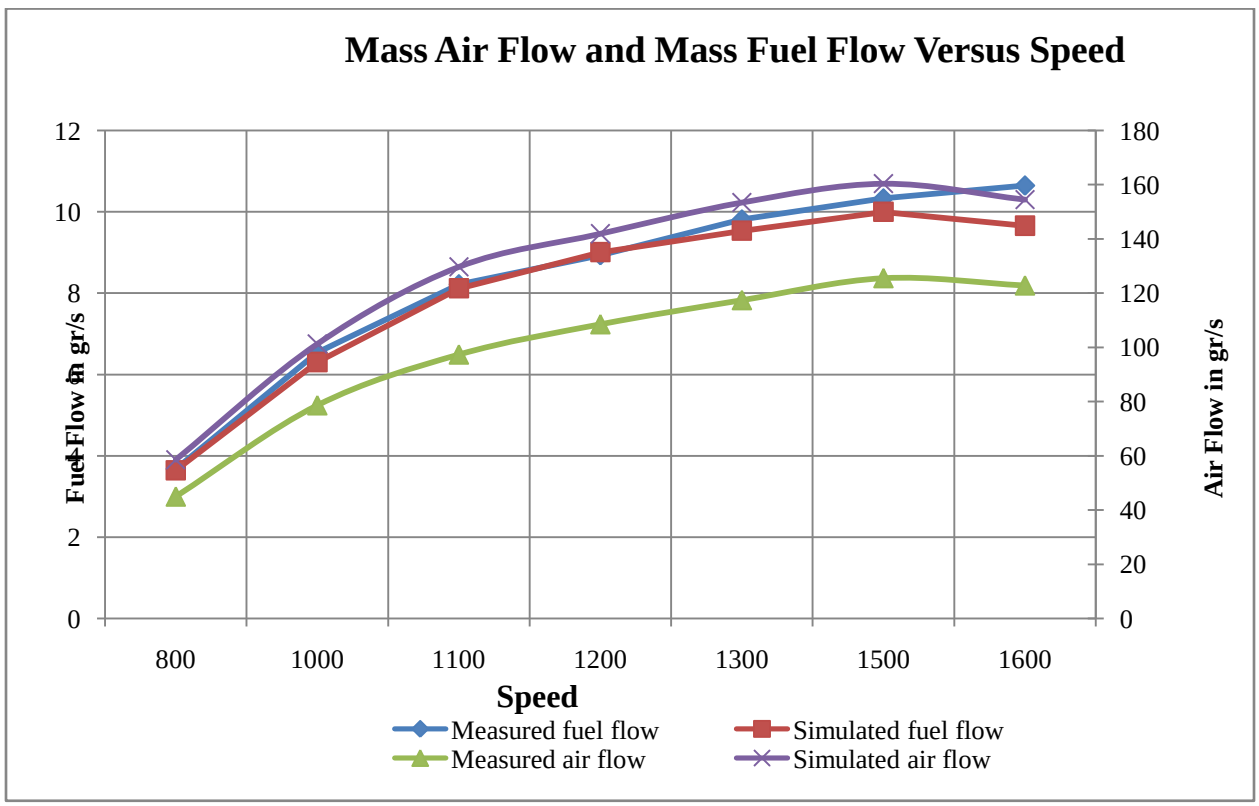


Figure 14: Mass Air Flow and Mass Fuel Flow versus speed

7.3 Results of the Miller Cycle

One of the ways to improve thermodynamic efficiency of the Spark Ignition Engine is by the optimization of valve timing and lift and compression ratios. The SI engine and the Miller Cycle engine are proven concepts for efficiency improvement of such engines.

In the Otto Cycle with a low expansion ratio, the BMEP is limited by the EGR ratio which is used to control the engine knocking and decrease the exhaust temperature. But in the Miller Cycle, with a high expansion ratio and low compression ratio, it is limited by the boost pressure. The combination of the Miller Cycle and EGR system demonstrates promising performance, substantially improving the BMEP under Stoichiometric conditions compared to conventional engines.

The objective of this thesis was to evaluate and investigate higher performance potential with late closing of the intake valves of a turbo charged six cylinder spark ignition engine. As well, the evaluations aim was to identify which of these brands used the Miller Cycle and what they gained with it. The evaluation was carried out by means of one- dimensional engine simulations. The use of LIVC provides a lower effective compression ratio [5], which reduces the risk for knock.

The same base engine was run on the test bench under the SI engine and Miller Cycle conditions, enabling direct thermodynamic comparisons under a wide variety of conditions of speed and load.

For this experimental study, the LIVC and EIVC versions of the Miller Cycle were selected. Schematic valve timing diagrams of the LIVC and EIVC are shown in Fig 15 and Fig 16. Three versions of the LIVC and two versions of the EIVC Miller Cycle were designed and tested; the detail parameters are presented in the Section Experimental plan.

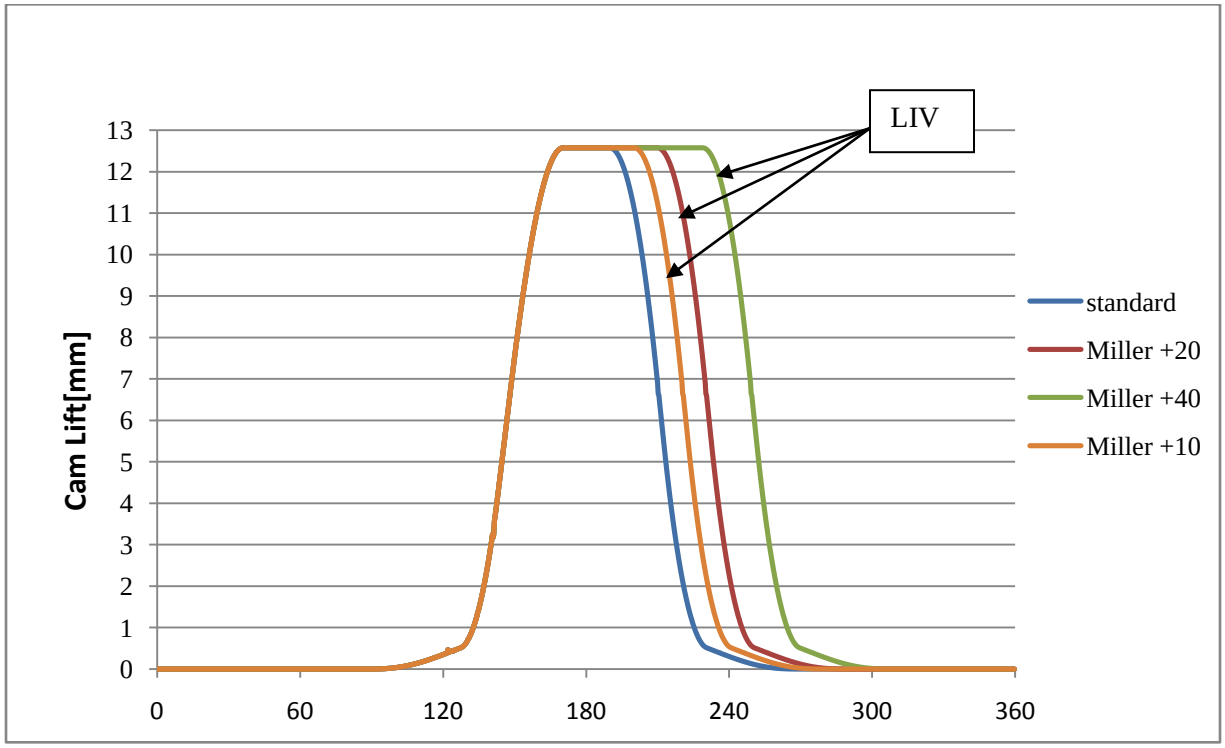


FIGURE 15: Schematic of valve timing of LIVC Miller Cycle

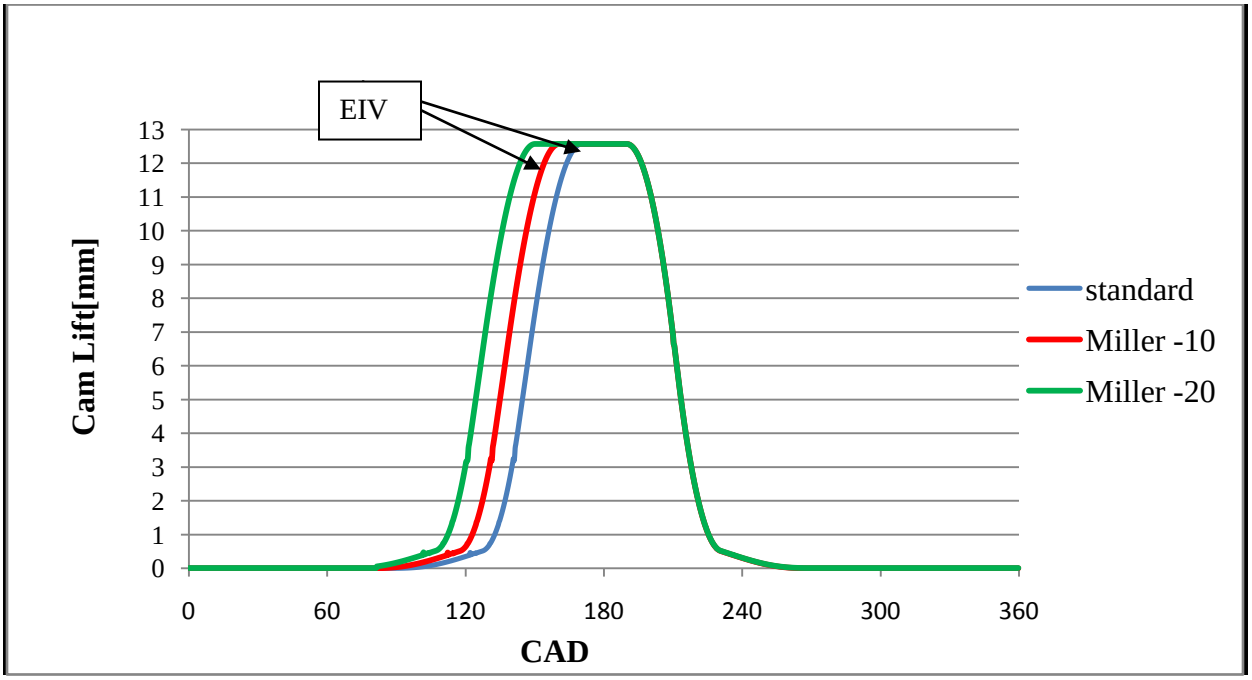


FIGURE16: Schematic of valve timing of EIVC Miller Cycle

7.4 Experimental plan

A test plan was designed to carry out the engine tests on the original Otto Cycle and two types of Miller Cycles. For comparison, the intake throttle was fixed at the maximum open position for all the tests. There were no changes for the other engine system, except for the intake valve timing. The running range of the engine was from 800 rpm to 1600 rpm.

Three versions of LIVC and two versions of EIVC of the Miller Cycle were designed and tested as follows:

1. LIVC Miller Cycle: the intake valve closed 10°, 20° and 40° later than that of the original SI engine.

2. EIVC Miller Cycle: intake valve closed 10° and 20° earlier than that of the original SI engine.

The whole experimental plan was realized in two stages: (i) running engine on standard SI Cycle; and (ii) running engine on the two types of Miller Cycle. Each test was repeated several times to make sure the data were reliable.

7.5 Tests results

The test results of the engines torque, Brake Specific Fuel Consumption (BSFC), gas exchange efficiency, mechanical efficiency, gross indicated efficiency, and brake efficiency for the standard engine and the two types of Miller Cycle are shown in Figure 17 to 24.

Figure 17; the figure shows *bsfc* versus engine speed for the standard engine and Miller engines. It can be seen that there is an improvement just by the use of a variable valve timing system to control the load via LIVC. In both cases of the LIVC and EIVC Miller Cycles, in LIVC the minimum load presented in Figure 18 is close to the minimum possible due to referred problems of ignition during open intake valve. But in the EIVC Miller Cycle for engine speeds control, it is possible to achieve much lower speed, and for higher speeds it’s better for LIVC and for lower speeds it’s EIVC possible.

The engine, either working as ELVC or LIVC Miller Cycle, has better efficiency. This is mainly caused by reduced pumping losses. In the case of LIVC the air/fuel mixture in inducted into the cylinder and after the BDC it is blown-back again to the intake manifold. This effect is more intense as longer IVC delays are used.

7.5.1 Brake specific fuel consumption

Brake specific fuel consumption (BSFC) is a measure of an engine's efficiency. It is a rate of fuel consumption divided by the rate of power production. The BSFC has to be the inverse from the net indicated efficiency. Then the maximum values for net indicated efficiency correspond to the minimum values for BSFC.

One comment about the high consumption points is if the combustion efficiency is not really good, it means that fuel escapes combustion, this implies that more fuel consumption is necessary to reach the load required.

BSFC is calculated from the measurement of fuel consumption divided by the rate of power according to the following formula [12].

$$\text{BSFC} = \frac{\dot{m}_f}{P_B} \qquad \text{(Eq 4)}$$

Where

BSFC = Brake specific fuel conception

$\dot{m}_f$ = fuel consumption

P = power

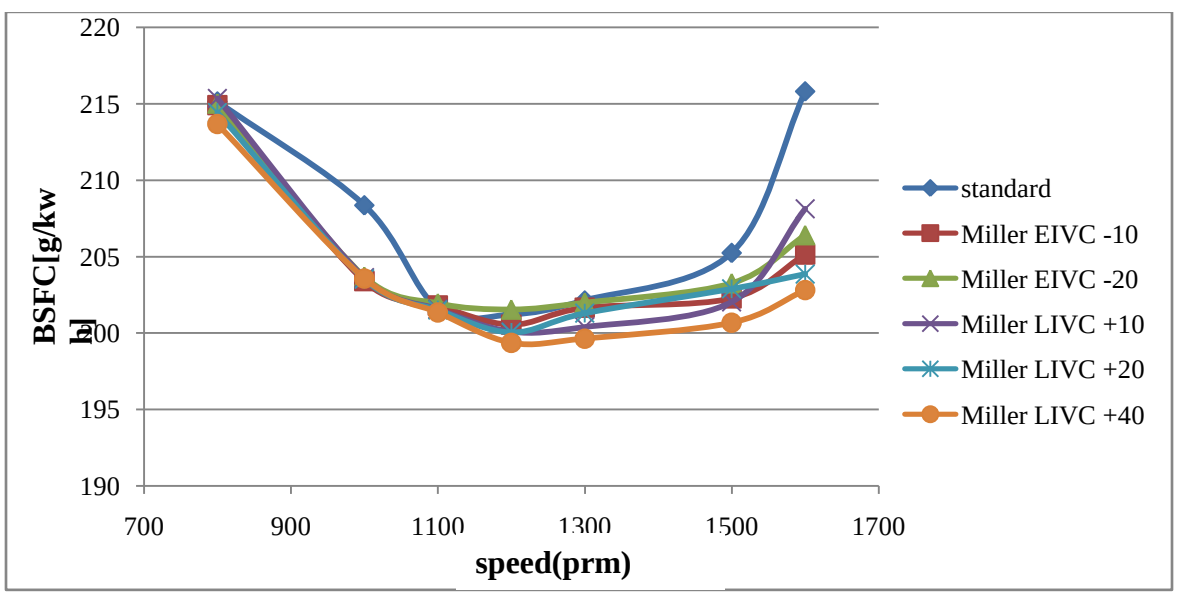


Figure 17: Brake Specific Fuel Consumption vs. Miller Cycle

By decreasing the pumping losses there is a potential to decrease the BSFC. The simulations that were compared have a brake specific fuel consumption ranging from 216g/kWh, for the standard engine configuration, to 213 g/kWh for the best LIVC +40 model in the same case.(See Figure 17).

7.5.2 Brake Mean Effective Pressure

An engine, power output can be put in relation to its size, making it possible to swiftly compare engines, regardless of their respective size. The most common term for this is brake mean effective pressure (BMEP). BMEP is the engine's actual power output divided by the engine's speed and displacement. BMEP is calculated from the measured torque according to the following formula [12].

$$\mathrm{BMEP} = \frac{P}{V_D \frac{N}{n_T}}$$

P = Power [W(Nm/s)] (Eq 5)

V_D = Engine Volume [m^3]

n_T = Stroke factor (2 for 4-stroke engines)

The factors that limit BMEP are mainly engine knocking, thermal loading such as exhaust temperature, and boost pressure. A lower compression ratio helps prevent engine knocking and a higher expansion ratio reduces the exhaust temperature in the Miller Cycle; also, the EGR system improves the knock limit by reducing the exhaust temperature. [16]

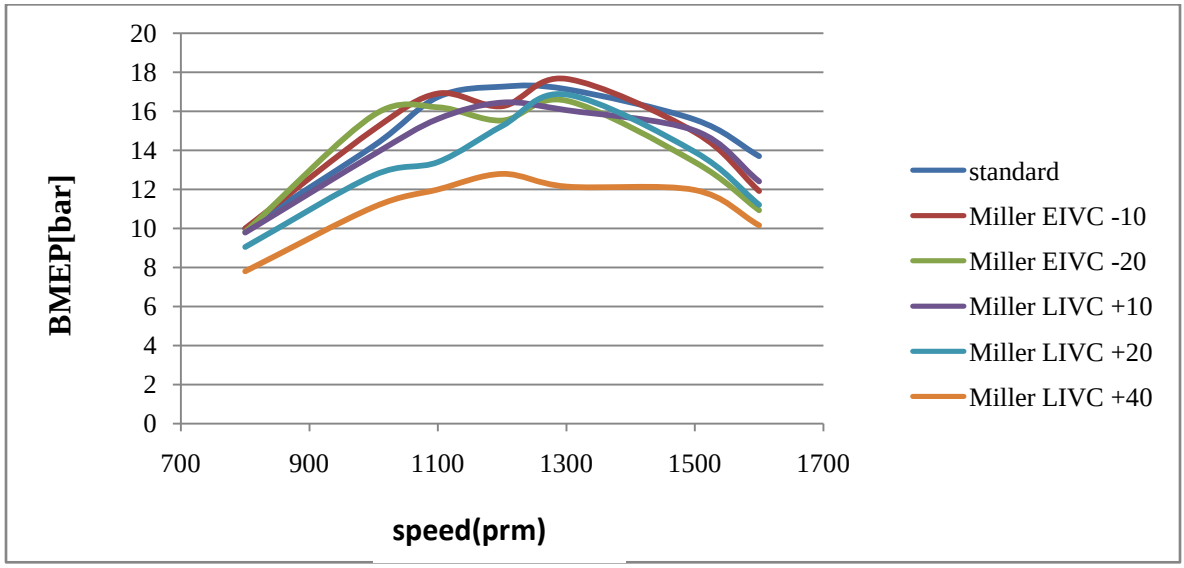


Figure 18: Brake Mean Effective Pressure vs. Miller Cycle before improvement

During the Miller Cycle experiment, the mechanical turbocharger shaft efficiency was used as a control variable to increase BMEP. This is because when intake valve timing was closed early or late to reduce BMEP, mechanical turbocharger shaft efficiency was also decreased. But after the mechanical efficiency of the turbocharger was increased, the efficiency of the Miller Cycle was increased.

As Figure 18 shows, it is demonstrated that the Miller Cycle has no inherent efficiency advantage because of the shortened compression stroke. However, the pre-ignition or engine knock problem of the Otto Cycle can be suppressed by the cooling of the air-fuel mixture before combustion, with early or late close of the intake valve. Therefore, the Miller Cycle innovation is ideal when used with supercharging.

Comparing the Miller Cycle with the Otto engine is an improvement in terms of torque and BMEP, and can be seen in Figure 18. This improvement is caused by the performance of the first LIVC camshaft, which has better volumetric efficiency at these speeds.

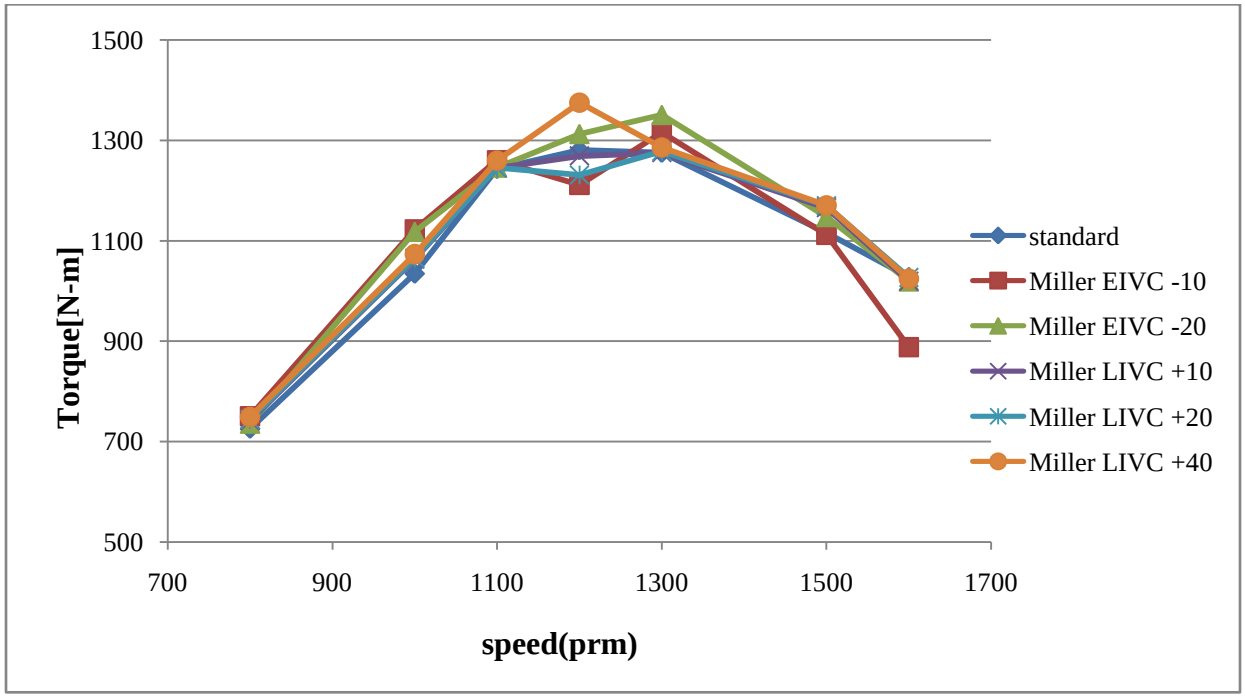


Figure 19: Effective Torque vs. Miller Cycle

The volumetric efficiency for the LIVC is somewhat lower than for the standard valve configuration. This is because some of the inducted charge is pushed back by the piston into the intake manifold due to the late closing of the intake valves. The boost pressure must therefore be somewhat higher than for the standard valve configuration. The combination of a lower intake port temperature and a higher boost pressure gives approximately the same knock sensitivity for the LIVC as for the standard valve configuration. The decreased compression ratio of the LIVC valve configuration seems in this particular case not to have any effect on the knock sensitivity compared to the standard case. A possible explanation for this is that the compression of residual before IVO raises the cylinder temperature and pressure to a higher level than for the standard case at the beginning of the intake stroke. When the intake valves of the LIVC valve configuration closes, both concepts have the same temperature and pressure in the cylinder. The maximum improvement happens for the maximum loads of the Miller LIVC +40 at 16 bar BMEP and for engine speeds from 1100 (rpm) to 1300 (rpm) (See Figure 20).

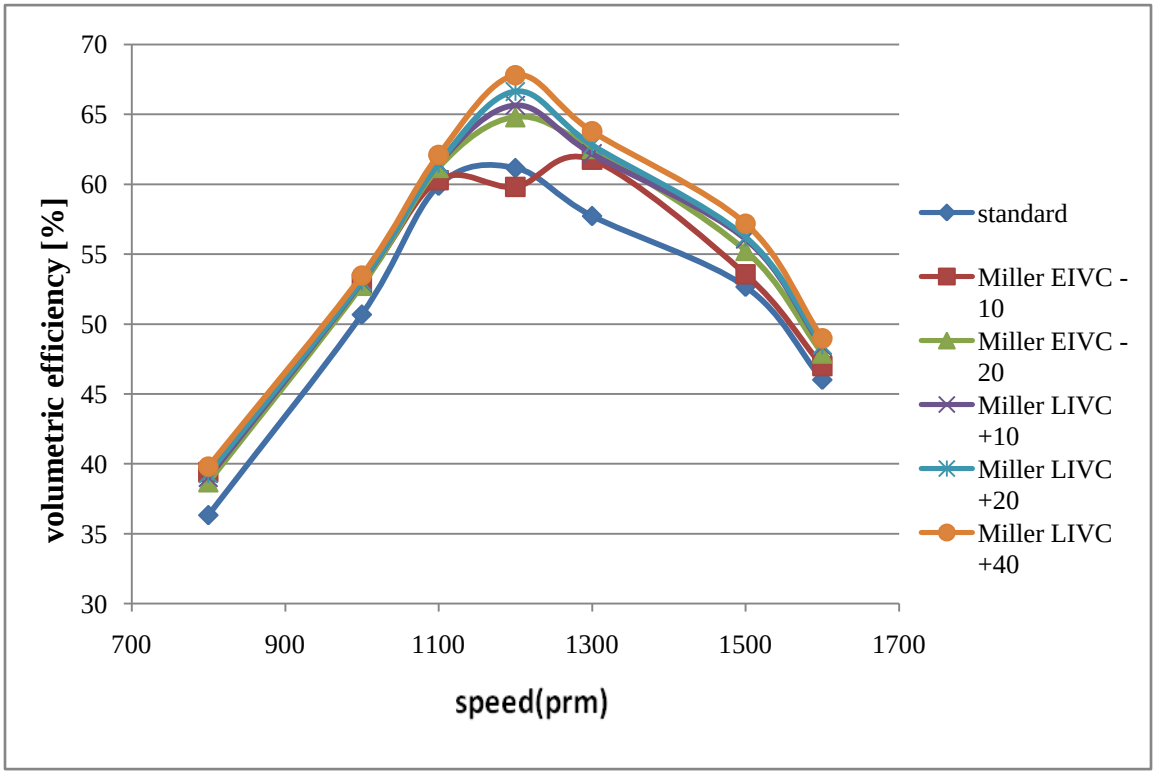


Figure 20: Volumetric Efficiency vs. Miller Cycle

7.5.3 $IMEP_{gross}$ and $IMEP_{net}$

The Indicated Mean Effective Pressure (IMEP) is the energy transformed into work (Wc) on the piston divided by the displacement.

$$IMEP = \frac{\oint P\, dv}{V_d} \approx \frac{\sum P_k \quad \Delta V_k}{V_d} \qquad \text{Eq. (6)}$$

IMEP is further divided into two difference definitions: IMEPnet and IMEPgross. IMEPnet is calculated over the entire engine cycle while IMEPgross is calculated only over the compression and expansion stroke [9].

$$IMEPnet = \frac{W_c}{V_d} = \frac{\oint P_{entired\ cycle} \quad dv}{V_d} \qquad \text{Eq. (7)}$$

$$IMEPgross = \frac{\oint P_{closed\ cycle} \quad dv}{V_d} \qquad \text{Eq. (8)}$$

7.6 Engine Efficiency

Higher efficiencies are expected with all engines and low emission. This is a result of working with low combustion temperatures, a short combustion period and throttled operation.

The natural gas engine used in this master thesis work had a sufficient compression ratio (10.5:1); this means it has high efficiency from a thermodynamic point of view although it increases the heat losses since it also increases pressure and temperature.

7.6.1 Brake Efficiency

Adding up pumping losses, mechanical losses, thermal losses and losses during the combustion, it is possible to know the real engine efficiency, the brake thermal efficiency.

The mechanical losses are generally due to the friction work produced by the movements of the different parts of the engine.

Brake efficiency is a product of different efficiencies as follows: [11]

$$\eta_B = \eta_{GI} \times \eta_{GE} \times \eta_m \qquad \text{Eq. (9)}$$

Where

H_B = Brake Efficiency

H_{GI} = Gross Indicated Efficiency

H_{GI} = Gas- Exchange Efficiency

H_m = Mechanical Efficiency

Figures 21 to 24 show all cases with different sweeps:

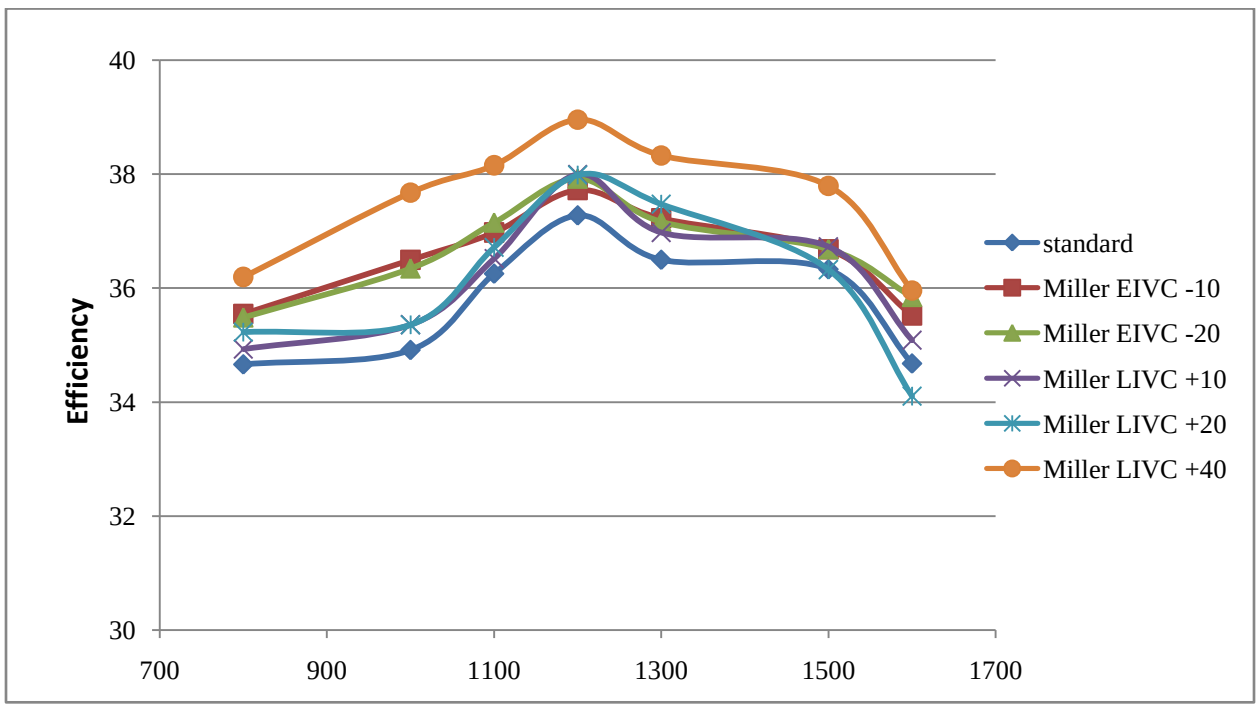


Figure 21: Brake Efficiency

7.6.2 Gross Indicated Efficiency

Gross Indicated Efficiency is the product of thermodynamic and combustion efficiencies and it can be calculated as follows:

$$\eta_{GI} = \left(\frac{IMEP_{gross}}{\left(\frac{m_f Q_{LHV}}{V_D} \right)} \right) \qquad \text{Eq. (10)}$$

Where

M_f = fuel mass per cycle

Q_{LHV} = lower heating value of the fuel

V_D = displaced volume

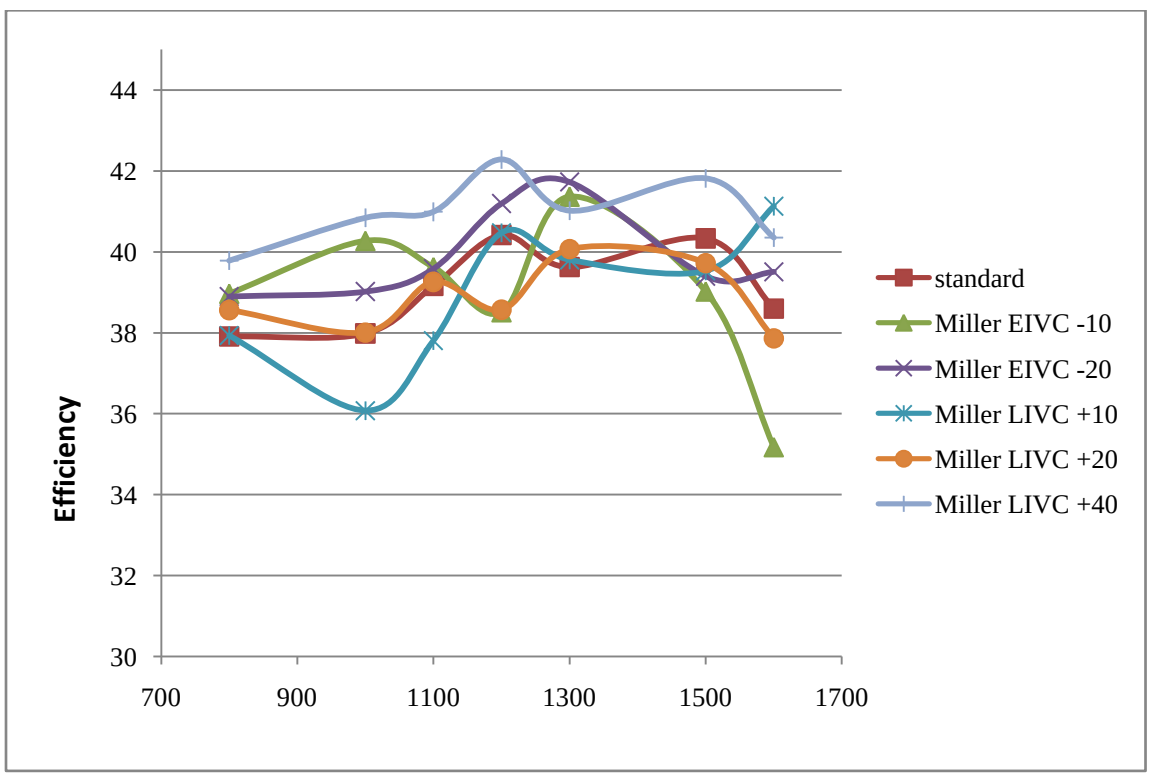


Figure 22: Gross indicate Efficiency

7.6.3 Gas- Exchange Efficiency

Gas- Exchange Efficiency is measured to evaluate the pumping losses in the engine.

$$\eta_{GE} = \left(\frac{IMEP_{net}}{IMEP_{gross}}\right) \qquad \text{Eq. (11)}$$

Gas-exchange efficiency shows the engine losses due to the pumping work. But the gas-exchange efficiency is more important and shows more useful information when the engine runs with the turbo charging.

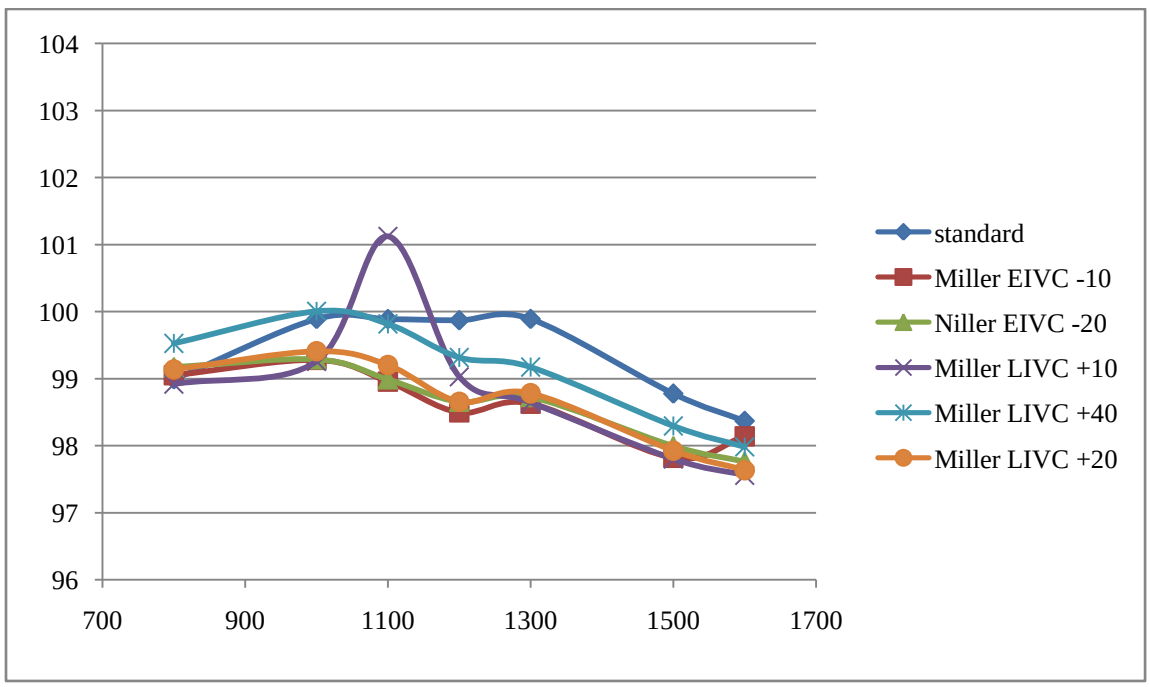


Figure 23: Gas exchange Efficiency

7.6.4 Mechanical Efficiency

Mechanical Efficiency is a measure to evaluate the mechanical losses, comprising in particular friction losses and, drive losses in oil, water and fuel supply pumps. The definition of mechanical efficiency is the relationship between work and the indicated work:

$$\eta_m = \frac{\mathrm{BMEP}}{\mathrm{IMEP_{net}}} \qquad \text{Eq. (12)}$$

This efficiency increases as load increases, since the friction losses are almost independent of the load, especially in the case of speed sweep. The differences in mechanical efficiency between running without EGR or with EGR are almost negligible.

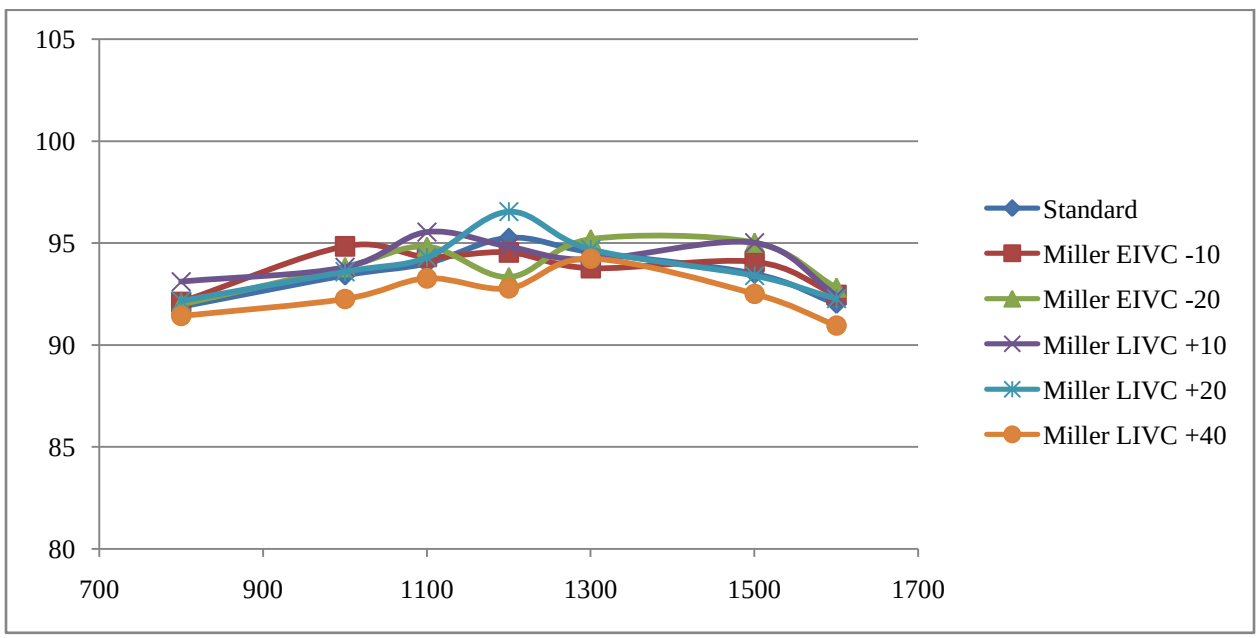


Figure 24: Mechanical Efficiency

8. Summary and Conclusions

The goal of this thesis work was to determine the durability and verify the accuracy of the engine simulation. Several simple models have been used to test the simulation.

GT-Power is a one dimensional simulation code based on thermodynamic principles. There is no consideration of flow phenomena like turbulence, etc. There is not yet any emission control except the value for accumulated NOx.

The Miller Cycle has the compression process shortened by either closing the intake valve early or late. With the Miller Cycle it is possible to reduce temperatures of the cylinder charge. The immediate advantage is the thermodynamically favorable increase of the compression ratio and an earlier firing point and/or extension of engine operation at high BMEP's to smaller methane numbers. The expansion process is unaltered. The performance characteristics of the Miller Cycle show that there are both advantages and drawbacks. The advantages are improvement of efficiency and reduction of emission. Drawbacks of the Miller Cycle are creating a volumetric efficiency loss due to the early or late intake valve closing, and very high turbocharger speeds are reached and knock sensitivity seems to be worse because of higher temperature before ignition by the Miller Cycle. The Miller Cycle can be an addition to the internal combustion engine industry. The greatest obstacle for the cycle to overcome is the additional costs it incurs from the supercharger. In a consumer-dominated world, the problem must be dealt with in order to make the Miller cycle engine truly competitive. [13].

The combination of the Miller Cycle (with LIVC) and the EGR system demonstrates promising performance, which substantially improves the BMEP under Stoichiometric conditions when compared with conventional engines.

In the Otto Cycle with a low expansion ratio, the BMEP is limited by the EGR ratio which is used to control the engine knocking and decrease the exhaust temperature. But in the Miller Cycle with a high expansion ratio and low compression ratio, it is limited by the boost pressure. The combination of the Miller Cycle and EGR system demonstrates promising performance, substantially improving the BMEP under Stoichiometric conditions compared by conventional engines. [16]

The aim has been to evaluate the Miller Cycle when converting the 6- cylinder natural gas engine. The two different Miller concepts, LIVE and EIVE, have been examined. The simulation was performed in GT-Power. The main issue was to examine the differences that occurred when changing the EIVC and LIVC. Concerning the LIVC, the results gained were satisfactory without changing the camshaft knock profiles, also use the better turbocharger. Concerning EIVC, the results were unsatisfactory due to a too high demand on the boost pressure.

The simulation result shall be seen as a recommendation when the correct valve timing is situated. The modified EIVC showed better results in the lower load and for higher load the LIVC is adapt.

The greatest obstacle for the cycle to overcome is the additional costs it incurs from the supercharger. In a consumer-dominated world, the problem must be dealt with in order to make the Miller Cycle engine truly competitive.

Recommendations on continuing the competitor analysis concerning the Miller Cycle is to completely focus on natural gas engines. Concerning the simulation, a next step would be to simulate a whole engine model. This would demand a detailed pre-study of the engine geometry so the computer model would be as close to reality as possible. Finally in the GT-Power, there is not yet any emission control except the value for accumulated NOx.

9. Future Improvement

In the simulations that were made in this study, the duration of the valves was constant, both on the intake and exhaust valves. There may be future improvements of the BSFC and the pumping losses by applying different duration on the valves.

With a combination of the Miller Cycle and electrically controlled valves, the improvement in the fuel consumption and gas exchange work can be of substantial size. The throttle losses at part load can be avoided completely by controlling the load with intake valves only, at least down to very low loads.

By using other material in the turbine, the sensitivity for high exhaust temperatures will decrease, and the possibilities to run on a Stoichiometric air/fuel ratio at high load and speed will increase. In the future it will probably be possible to use a turbine with variable nozzle, and to improve efficiency on a wider range of speeds and still maintain a sufficient boost pressure at low speeds, and thereby an increase in the compression ratio can be possible.

Downsizing can be possible when a combination of the variable turbine and electrical valves are combined with the Miller Cycle. If the torque can be increased for an engine, would be wise to decrease the size of the engine and still have the same torque as the standard engine. By downsizing the friction and mechanical losses will decrease.

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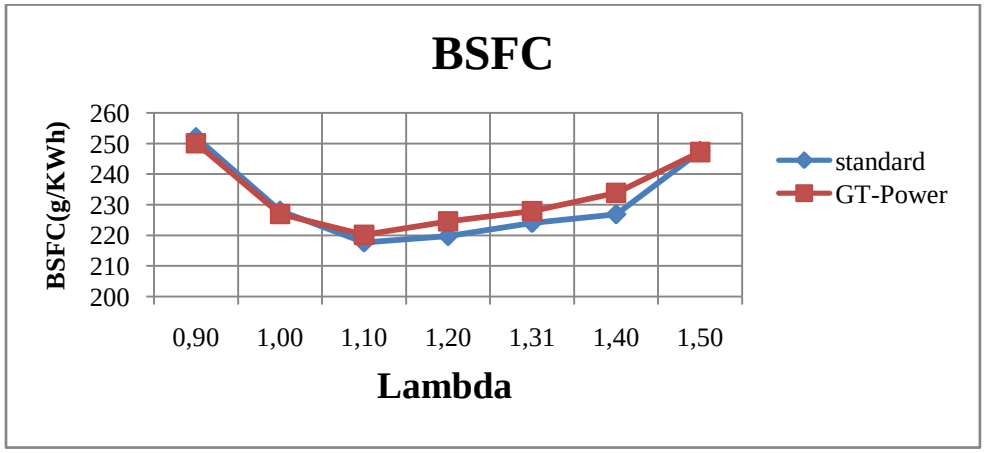
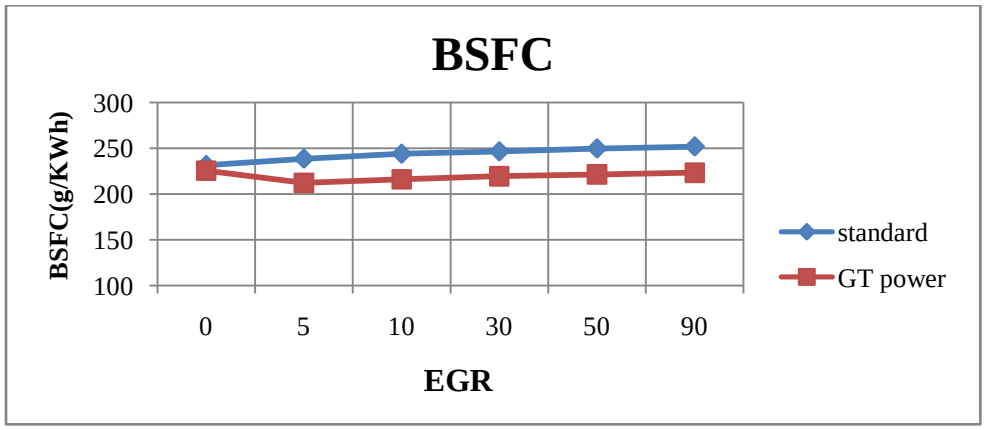
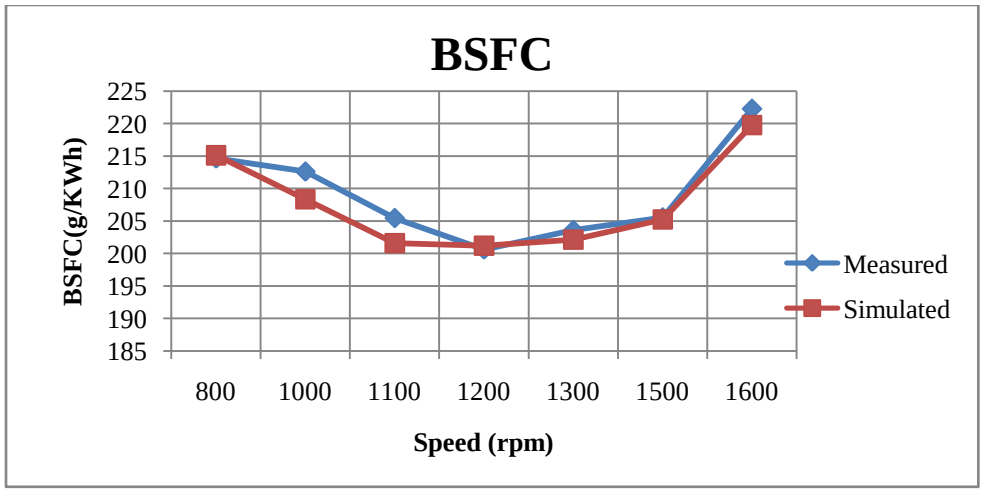
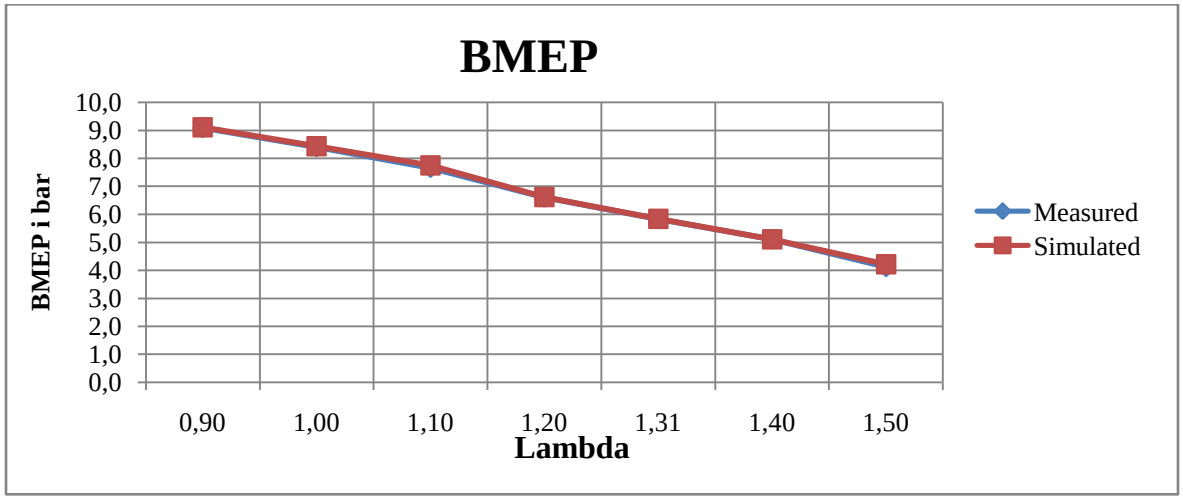
Appendix: Comparison between standard engine and GT-Power model

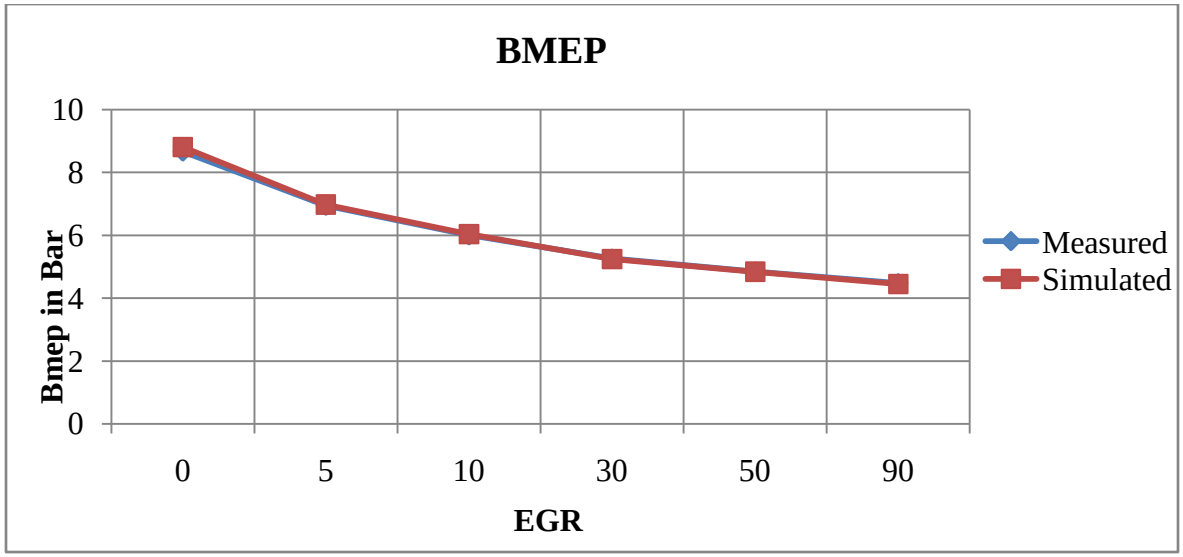
This appendix reviews a comparison between standard engine test data and the GT-Power.

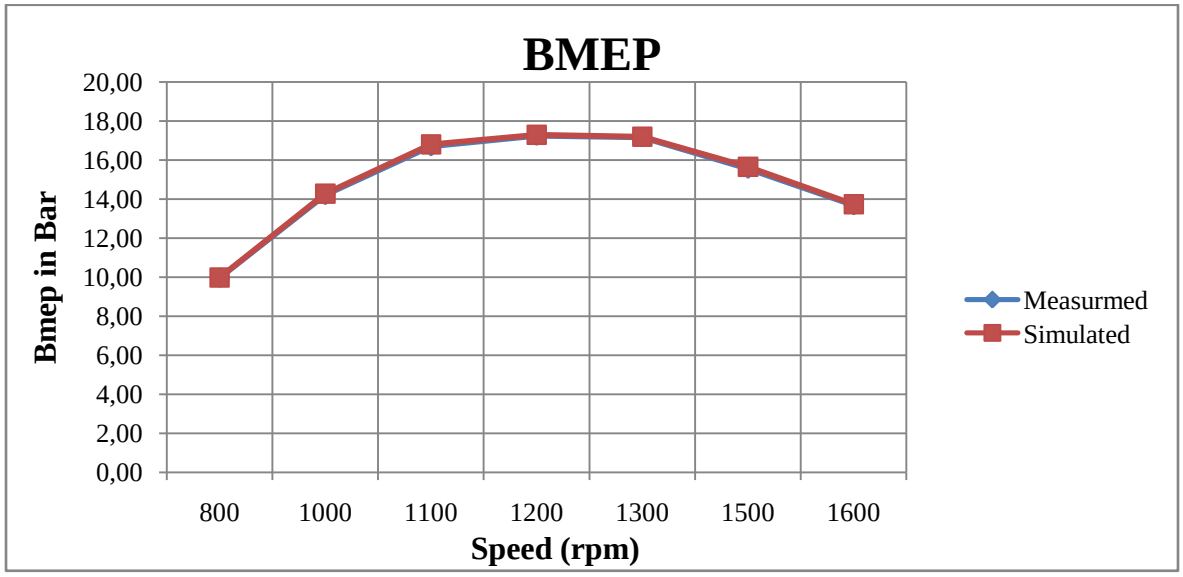


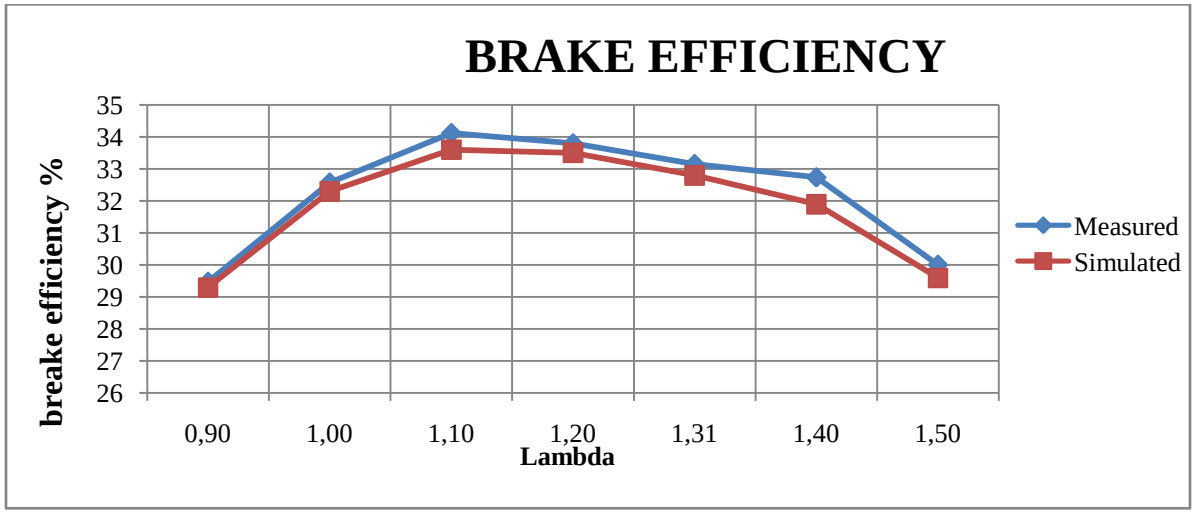


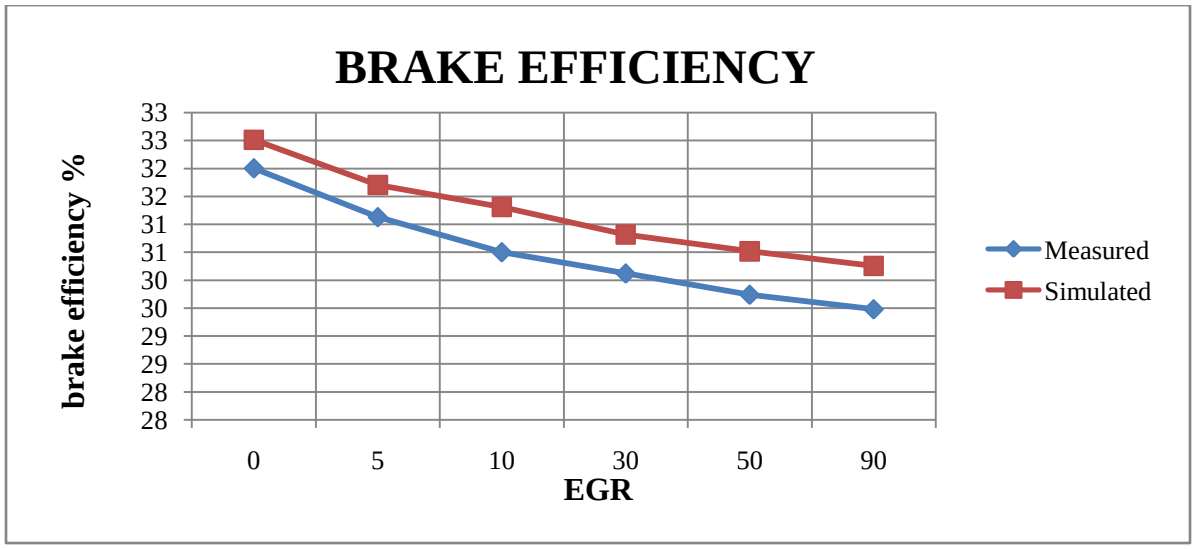


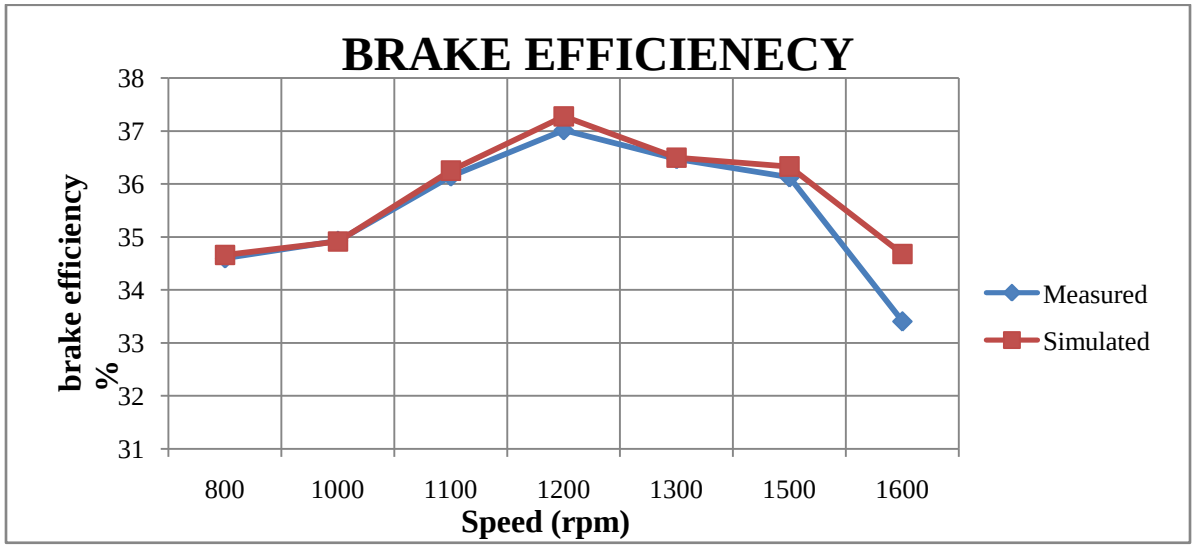

BMEP
BMEP i bar
10,0
9,0
8,0
7,0
6,0
5,0
4,0
3,0
2,0
1,0
0,0
0,90
1,00
1,10
1,20
1,31
1,40
1,50
Lambda
Measured
Simulated

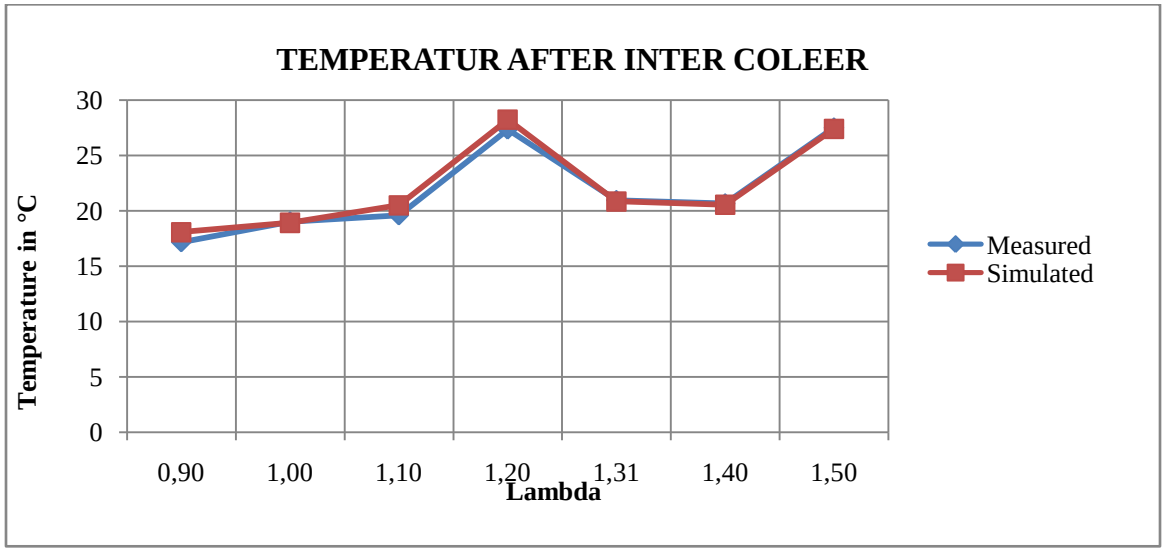

BMEP
Bmep in Bar
10
8
6
4
2
0
0
5
10
30
50
90
EGR
Measured
Simulated

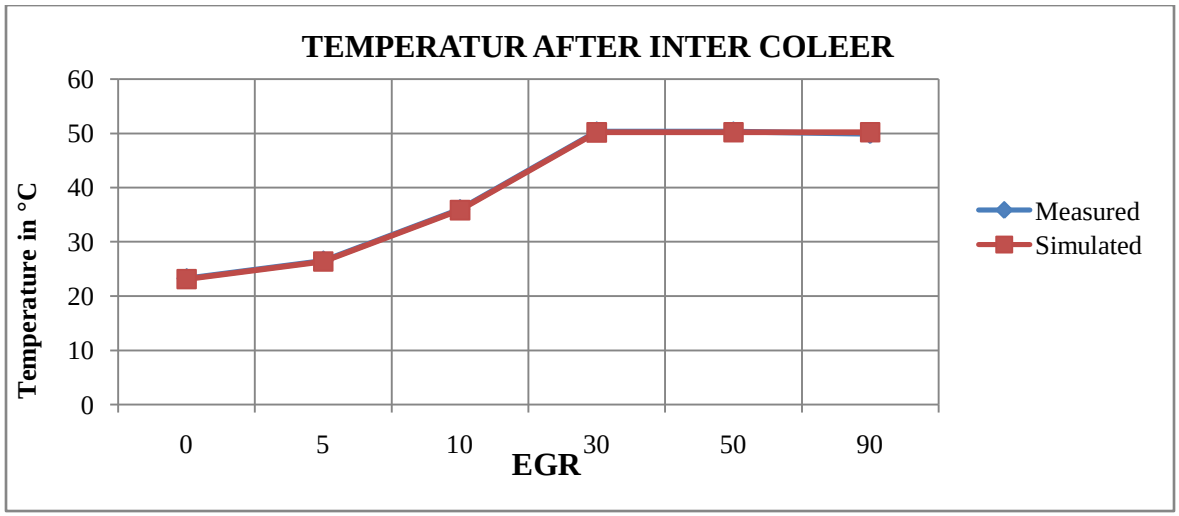

BMEP
Bmep in Bar
20,00
18,00
16,00
14,00
12,00
10,00
8,00
6,00
4,00
2,00
0,00
800
1000
1100
1200
1300
1500
1600
Speed (rpm)
Measurmed
Simulated

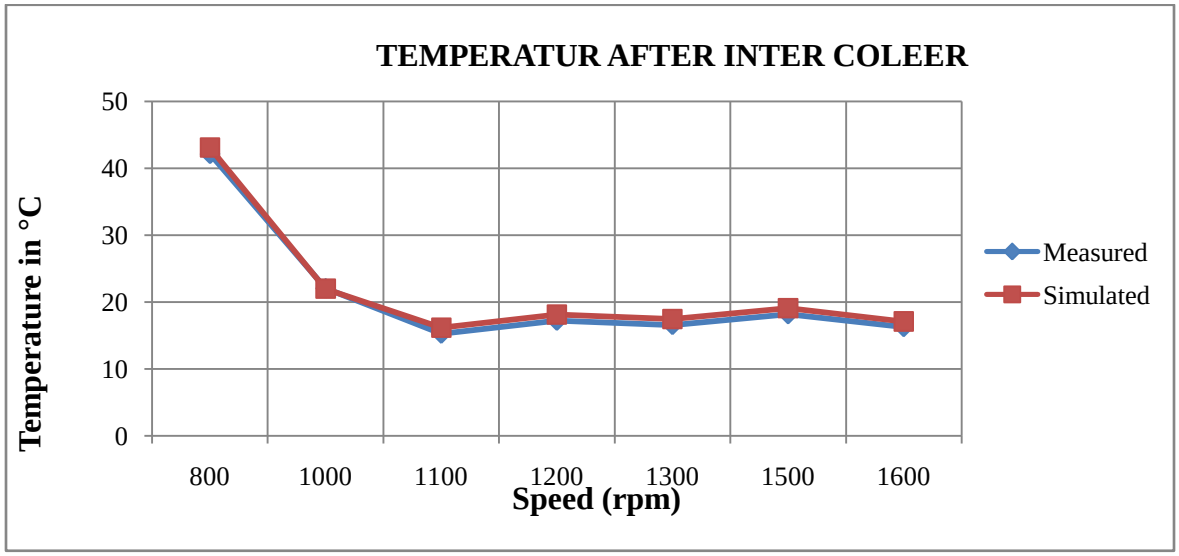

BRAKE EFFICIENCY
breake efficiency %
35
34
33
32
31
30
29
28
27
26
0,90
1,00
1,10
1,20
1,31
1,40
1,50
Lambda
Measured
Simulated

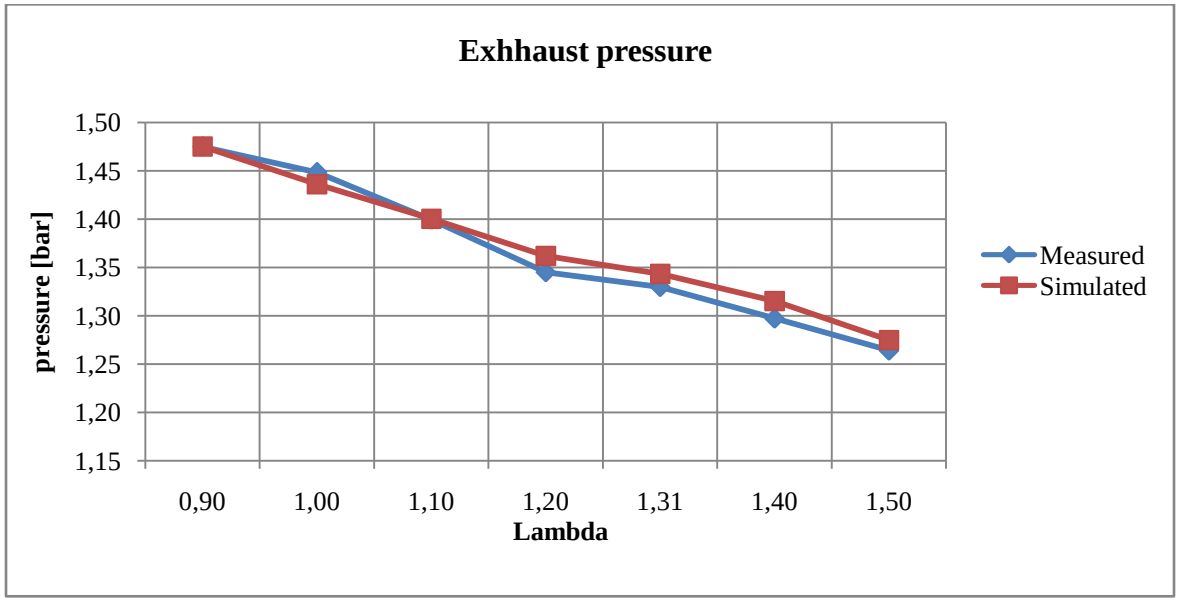

BRAKE EFFICIENCY
brake efficiency %
33
33
32
32
31
31
30
30
29
29
28
28
0
5
10
30
50
90
EGR
Measured
Simulated

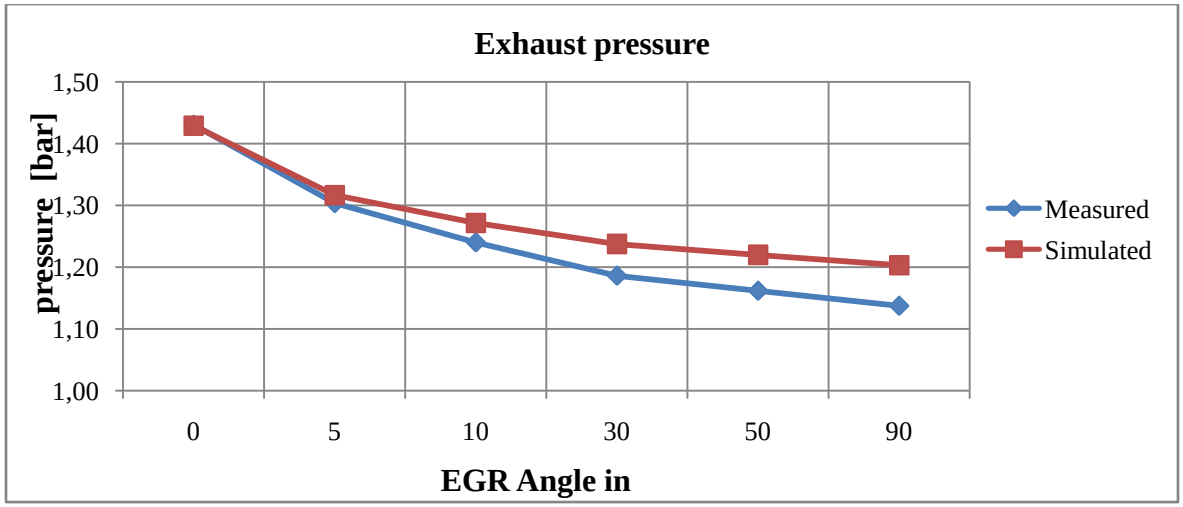

BRAKE EFFICIENECY
brake efficiency %
38
37
36
35
34
33
32
31
800
1000
1100
1200
1300
1500
1600
Speed (rpm)
Measured
Simulated

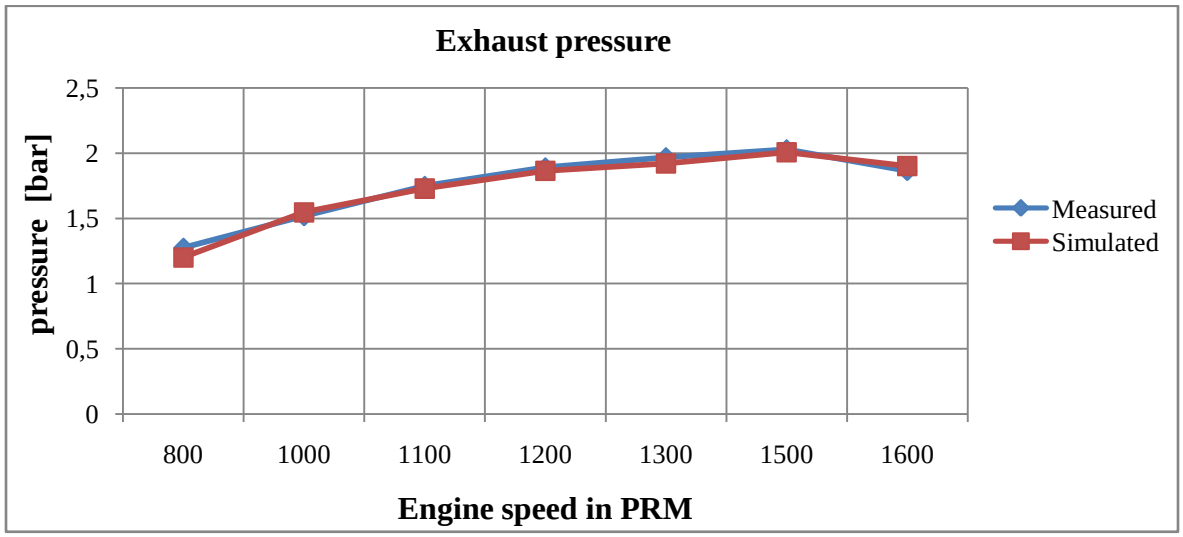

TEMPERATUR AFTER INTER COLEER
Temperature in °C
30
25
20
15
10
5
0
0,90
1,00
1,10
1,20
1,31
1,40
1,50
Lambda
Measured
Simulated


TEMPERATUR AFTER INTER COLEER
Temperature in °C
60
50
40
30
20
10
0
0
5
10
30
50
90
EGR
Measured
Simulated


TEMPERATUR AFTER INTER COLEER
Temperature in °C
50
40
30
20
10
0
800
1000
1100
1200
1300
1500
1600
Speed (rpm)
Measured
Simulated


Exhhaust pressure
pressure [bar]
1,50
1,45
1,40
1,35
1,30
1,25
1,20
1,15
0,90
1,00
1,10
1,20
1,31
1,40
1,50
Lambda
Measured
Simulated


Exhaust pressure
pressure [bar]
1,50
1,40
1,30
1,20
1,10
1,00
0
5
10
30
50
90
EGR Angle in
Measured
Simulated


Exhaust pressure
pressure [bar]
2,5
2
1,5
1
0,5
0
800
1000
1100
1200
1300
1500
1600
Engine speed in PRM
Measured
Simulated