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Closed-Loop Combustion Control for a 6-Cylinder Port-Injected Natural-gas Engine

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ABSTRACT

High EGR rates combined with turbocharging has been identified as a promising way to increase the maximum load and efficiency of heavy duty spark ignition engines. With stoichiometric conditions a three way catalyst can be used which means that regulated emissions can be kept at very low levels. Obtaining reliable spark ignition is difficult however with high pressure and dilution. There will be a limit to the amount of EGR that can be tolerated for each operating point. Open loop operation based on steady state maps is difficult since there is substantial dynamics both from the turbocharger and from the wall heat interaction. The proposed approach applies standard closed loop lambda control for controlling the overall air/fuel ratio for a heavy duty 6-cylinder port injected natural gas engine. A closed loop load control is also applied for keeping the load at a constant level when using EGR. Furthermore, cylinder pressure based dilution limit control is applied on the EGR in order to keep the coefficient of variation at the desired level of 5%. This way confirms that the EGR ratio is kept at its maximum stable level all times. Pumping losses decrease due to the further opening of the throttle, thereby the gas exchange efficiency improves and since the regulator keeps track of the changes the engine all the time operates in a stable region. Our findings show that excellent steady-state performance can be achieved using closed loop combustion control for keeping the EGR level at the highest level while the stability level is still good enough.

INTRODUCTION

Recently, environmental improvement (CO2, NOx and ozone reduction) and energy issues have become more and more important as worldwide concerns. Natural gas consisting of highly methane (90%) is a good alternative fuel to improve these problems because of its plentiful availability and clean burning characteristics. Heavy duty spark ignited (SI) natural gas engines could be operated with two approaches viz. lean operation and stoichiometric operation. Recent work at the department of energy sciences at Lund University has showed better result by stoichiometric operation [1] since stoichiometric operation with a three way catalyst results in very low emissions while keeping efficiency in a reasonable level. Stoichiometric operation with high EGR rates combined with turbocharging has been identified as a promising way to get the following advantages:

- Better fuel economy than pure stoichiometric operation since [2]:
  - **Reduced throttling losses (at low/part loads):** The addition of inert exhaust gas into the intake system means that for a given power output, the throttle plate must be opened further, resulting in increased inlet manifold pressure and reduced throttling losses.
  - **Reduced heat rejection:** Lowered peak combustion temperatures not only reduce NOx formation, it also reduces the loss of thermal energy to combustion chamber surfaces, leaving more available for conversion to mechanical work during the expansion stroke.
  - **Reduced chemical dissociation:** The lower peak temperatures result in more of the released energy remaining as sensible energy near TDC, rather than being bound up (early in the expansion stroke) in the dissociation of combustion products. This effect is relatively minor compared to the first two.
- Lower emissions than lean burn operation because of using 3-way catalyst
- Decreasing knock tendency

These advantages results in increasing the maximum load and efficiency of heavy duty spark ignition engines.

Many researchers have been dedicated to extend the limit of dilution burn operation in order to improve fuel efficiency, as well as reducing exhaust gas emission from the spark ignition engine. The dilution limit is imposed by increased cyclic variation of the combustion intensity that reduces the drivability and the effect is usually quantified through the coefficient of variation (COV) of the indicated mean effective pressure (IMEP) analysis [2-4]. So there will be a limit to the amount of EGR that can be tolerated for each operating point. As the unburned mixture in a spark ignited engine is diluted with EGR (or even excess air), the flame development period, the duration of burning phase and the cyclic variation in combustion process will increase and if the combustion passes the dilution limit then the operation becomes rough and unstable. It is found that drivability problems tend to occur when COVimep values exceed about 5%; therefore any values above this represents combustion variability that is unacceptably high.

The factors that have been found to influence cycle to cycle variation are:

1. Variation in gas motion in the cylinder during combustion

2. The variation in the amounts of EGR (internal and external), fuel and air each cycle

3. The variation in mixture composition within the cylinder each cycle. This effect is especially pronounced near the spark plug due to the variations in mixing between air, fuel and internal and external EGR.

Since the engine is operating in steady-state mode and with highest amount of EGR, the variations are mostly because of the third reason.

The objective of this work is to develop a tool for mapping the best positions of the throttle and EGR valve in different loads and speeds where the engine has the lowest pumping losses and COV of IMEP is still below 5%. For developing this tool three different regulators have been designed. A standard closed loop lambda control for controlling the overall air/fuel ratio and a standard closed loop load control were developed. Furthermore, cylinder pressure based lean limit control is applied in order to keep the COVimep at the desired level of 5%.

EXPERIMENTAL SETUP & CONTROL METHOD

This section will cover the information about the experimental engine and the modifications, the engine control system, measurements system, gas data, developed regulators and the explanations of the performed experiment.

THE ENGINE & MODIFICATIONS

The experimental engine was originally a diesel engine from Volvo which is converted to a natural gas engine see table 1 for specification. The engine is equipped with short route cooled EGR system and also turbocharger with wastegate.

Number of Cylinder	6
Displacement	9,4 Liter
Bore	120 mm
Stroke	138 mm
Compression ratio	10,5 :1
Fuel	Natural gas

Table 1: Specification of the engine

Following modifications were performed on the engine:

- **Multi-Port injection System:** Originally the engine has single point injection, with four injectors at the fuel injector assembly. The gas pressure is approximately 10 bar. The test bench engine is supplied with natural gas at 4.6 bar, so the port injection system is equipped with 12 injectors (2 per cylinder) to be able to cover the whole load range, see Figure 1.

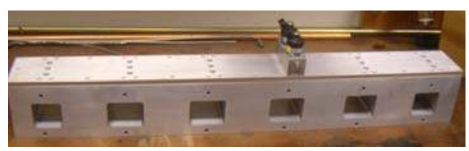

Figure 1: Multi-Port Fuel Injection

- **Mouthpieces**: in order to prevent cross breathing of natural gas between cylinders, six mouthpieces were designed to pass the gas flow in the same direction as the cylinders, see figure 2.

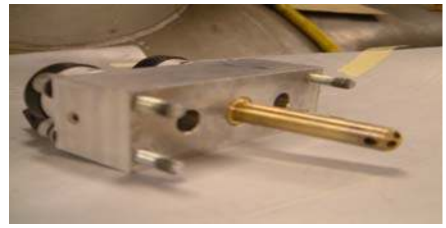

Figure 2: Injector Mouthpiece

ENGINE CONTROL SYSTEM

A master PC based on GNU/Linux operating system is used as a control system. This communicates with three cylinder-control-modules (CCM) for cylinder-individual control of ignition and fuel injection via CAN communication, see figure 3. Crank and cam information are used to synchronize the CCMs with the crank rotation.

Flexible controller implementation is achieved using Simulink and C-code is generated using the automatic code generation tool of Real Time Workshop. The C-code is then compiled to an executable program which communicates with the main control program. The controllers used for this experiment are lambda, load and EGR controller which determine the offset amount of fuel, air and EGR. The controllers can be activated from Graphical User Interface (GUI).

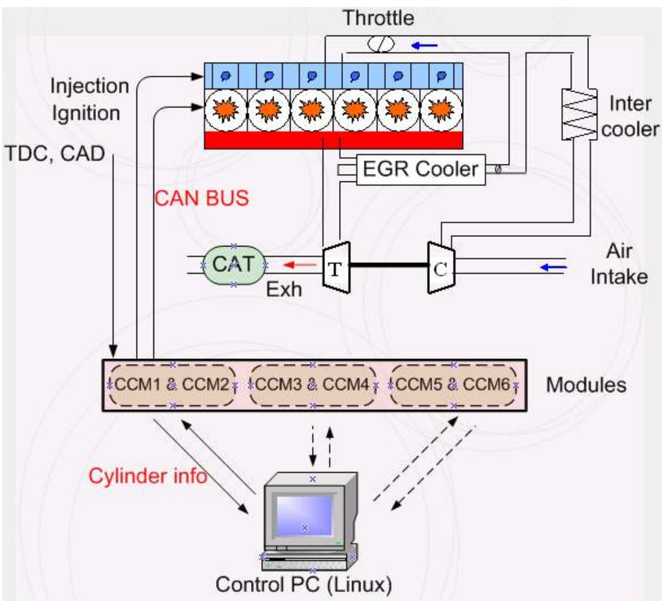

Figure 3: The Engine and its control system

MEASUREMENT SYSTEMS

Each cylinder head is equipped with a piezo electric pressure transducer, Kistler 7061B to monitor cylinders pressures for heat release calculations. Cylinder pressure data is sampled by a Microstar 5400A data acquisition processor. EGR was calculated by measuring CO2 at inlet and exhaust. Emissions (HC, CO, NO, NO2, NOx, CO2, O2) are measured before and after catalyst. Also, temperatures at inlet/exhaust, pressures at inlet/exhaust, fuel and air flow, lambda, torque and engine speed have been measured.

GAS DATA

The composition of the natural gas, which varies slightly over time, is shown in Table 2. The lower heating value is 48,4 MJ/kg.

Composition	%	Structure
Methane	89,84	CH_4
Ethane	5,82	C_2H_6
Propane	2,33	C_3H_8
I-Butane	0,38	C_4H_{10}
N-Butane	0,52	C_4H_{10}
I-Pentane	0,11	C_5H_{12}
N-Pentane	0,07	C_5H_{12}
Hexane	0,05	C_6H_{14}
Nitrogen	0,27	N_2
CO2	0,6	CO_2

Table 2: The natural gas composition

CONTROL METHOD

As mentioned before the objective of this work is to develop a tool for mapping the best positions of the throttle and EGR valve in different loads and speeds where the engine has the lowest pumping losses and COVimep is still below 5%. For developing this tool different regulators were needed. Three different regulators were designed for controlling overall air / fuel ratio, load and EGR level, see Figure 4. Bumpless transfer and Anti-Windup algorithm were applied during the design of the regulators.

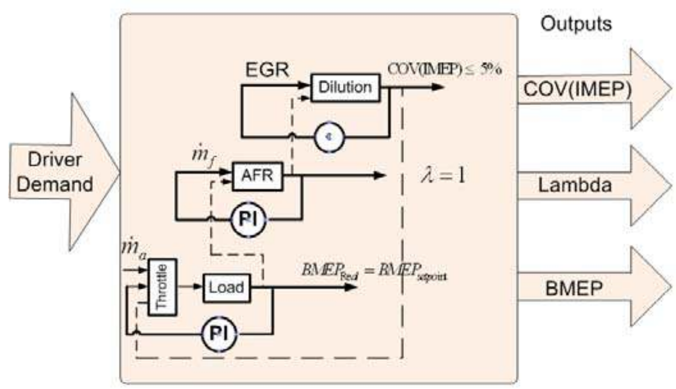

Figure 4: Closed-Loop Combustion Control

<u>CLOSED LOOP LAMBDA CONTROL</u>

Closed loop lambda control evaluates the signals from the broadband lambda sensor. The sensor measures the oxygen content in the exhaust gas, and thus provides information about the mixture composition. The closed-loop lambda control strategy uses the injected fuel quantity as the manipulated variable and can compensate for the lambda error. A Proportional Integral (PI) control strategy is used for controlling lambda. The error signal was based on the differences between the measured lambda and a desired setpoint lambda and, a fuel offset was generated from that.

CLOSED LOOP LOAD CONTROL

The engine is connected to an electric dynamometer, and the torque is measured with a load cell. BMEP is calculated from the measured torque according to the following formula [5].

$$Bmep = \frac{2\pi n_T T}{V_D} \quad (1)$$

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

T = Torque

V_D = Engines Volume

Closed loop load control evaluates the signals from the load cell. The error signal was based on the differences between the measured BMEP and a desired setpoint BMEP and, a throttle offset was generated from that. The throttle was adjusted by the regulator to keep the measured BMEP at the same level as the desired BMEP.

CLOSED LOOP EGR CONTROL

One important measure of cyclic variability, derived from pressure data, is COV_{imep}. It is standard deviation of imep divided by the mean imep.

$$COV_{imep} = \frac{\sigma_{IMEP}}{IMEP} \times 100 \quad (2)$$

EGR closed loop control evaluates the calculated COVimep to control the EGR valve. The error signal was based on the differences between the calculated COVimep and a set setpoint COVimep for 5%. EGR valve opens more as long as the COVimep is less than 5%, and if COVimep exceeds 5% the regulator starts to close the EGR valve. So the regulator attempts always to keep the EGR valve in a position such that COVimep is around 5%.

NEW METHOD FOR COV_{IMEP} CALCULATION

It was desired to calculate and update COVimep continually and smoothly over a fixed number of cycles (100 cycles in this paper). It also was desired to calculate COV_{imep} in a way that transient running the engine dose not affects the COV_{imep} too much. For smoothing the data set and in order to have more realistic values it was desired to put less weight on the latest values. The definition of Low-Pass Filter was helpful to calculate a filtered set of COVimep. A new variable is called $IMEP_{filtered}$ and is calculated according to the following equation:

$$IMEP_{filtered} = (IMEP_{net} \times 0.3) + (IMEP_{filtered} \times 0.7) \quad (3)$$

$$COV_{imep} = \left(\frac{\sqrt{\frac{\sum (IMEP_{net} - IMEP_{filtered})^2}{N}}}{IMEP_{filtered}} \right) \times 100 \quad (4)$$

N = Number of cycles

In order to make a comparison between the developed method and the common method (calculating COV_{imep} based on mean value of a number of cycles), a simulation was performed in Simulink environment and the results are demonstrated in figures 5 and 6. The comparison was performed in two stages, first under steady state condition where the IMEP data were not varied. In the second test a transient condition were provided in order to see how different COV_{imep} calculations react in this situation.

Figure 5 demonstrates comparison between COVimep calculations with filtered based and means based methods under steady state conditions. It can be seen that both methods work well under the steady state conditions and COVimep calculations are smooth and trustable in the both cases.

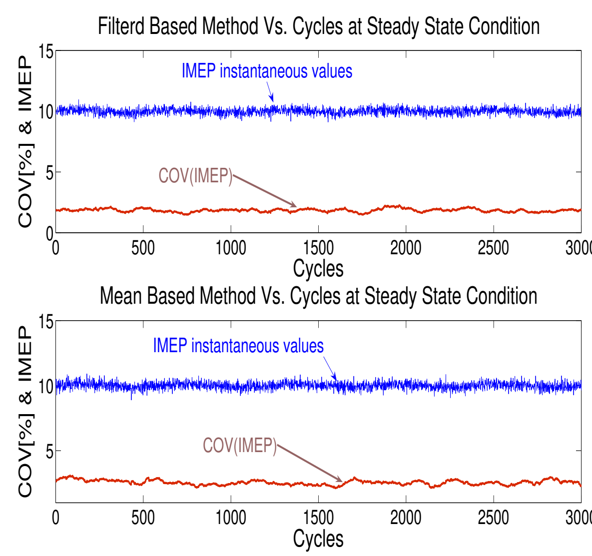

Figure 5: Comparison of COV_{imep} calculations with different methods under steady state conditions

Figure 6 demonstrates comparison between COV_{imep} calculations with filtered based and means based methods under simulated transient conditions. As it is obvious in figure 6 the filter based method can easily catch the transient and the value of COV_{imep} won't fluctuate extreme because the latest value in COV_{imep} calculation is weighted lower than other values. In the other hand COV_{imep} in the mean based method cannot be used under transient (see figure 6 the lower plot) because of really high fluctuation which is coming from the calculation of the mean values. It can be concluded that the mean based COV_{imep} calculation cannot be used under transient running but the filtered based method can be used under the both, steady state and transient conditions.

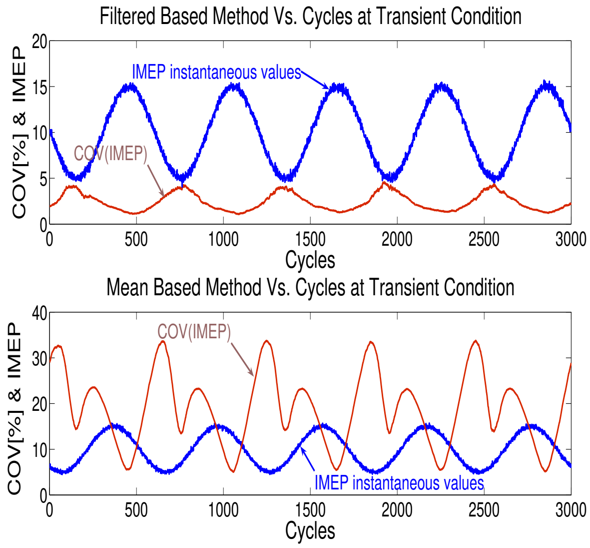

Figure 6: Comparison of COV_{imep} calculations with different methods under transient conditions

EXPERIMENT

The engine was tested for a variety of speed / loads at steady state condition. In order to evaluate the results, it was decided to run the engine at three different loads (2.5, 4 and 5.5 bar[1]) and at three different speed levels (800, 1000, and 1200). Table 3 lists those points and the running strategies. Lower loads have been chosen because pumping losses are more crucial in this region.

To provide a basis for a fair comparison, testing was conducted in two stages, first without adding EGR and without any regulator. In the second stage by using closed loop load and lambda control the load and lambda were kept constant, the amount of EGR was increased by the regulator up to the highest possible EGR while keeping COVimep < 5%. As it is already explained by increasing EGR rate the COVimep will increase. The highest limit of COVimep is predefined to 5% and by "*the highest possible EGR rate*" means the amount of EGR before the level of COVimep passes 5%.

[1] Loads are written in form of BMEP (Break Mean Effective Pressure)

Speed	BMEP	Strategies
800	2,5	NO EGR / NO regulator
800	2,5	With regulator
800	4	NO EGR / NO regulator
800	4	With regulator
800	5,5	NO EGR / NO regulator
800	5,5	With regulator
1000	2,5	NO EGR / NO regulator
1000	2,5	With regulator
1000	4	NO EGR / NO regulator
1000	4	With regulator
1000	5,5	NO EGR / NO regulator
1000	5,5	With regulator
1200	2,5	NO EGR / NO regulator
1200	2,5	With regulator
1200	4	NO EGR / NO regulator
1200	4	With regulator
1200	5,5	NO EGR / NO regulator
1200	5,5	With regulator

Table 3: Engine test operating conditions

INJECTION AND IGNITION TIMING

Injection timing was fixed for all cases but ignition timing varies in order to get MBT (Maximum Break Torque) for each case. A common rule says that 50% of the fuel is burned at about 10 CAD after top dead center (ATDC), resulting in MBT ignition [2]. The results of the comparison between these points are presented in the next section. Running the engine under transient operating conditions will be the follow on program of this project. The amount of calibration and modification should be done before running under the transient operating conditions.

EXPERIMENTAL RESULTS

This section will cover the results of the engine testing. The testing program was performed at a variety of speed / loads at steady state condition. The tests performed in two stages for each speed and load conditions. The first was the regular running of the engine, where no EGR was added and no lambda or load regulator were activated. In the second stage the regulators (Lambda, Load and EGR closed loop control) were activated.

The operating points are evaluated in terms of Brake Efficiency, pumping losses, fuel consumption and stability. Engine runs with stoichiometric operation with 3-way catalyst and emissions were measured after the catalyst but, since the changes in emissions were not significant, those are not presented in the paper.

EFFICIENCIES

Brake Efficiency is a product of different efficiencies as follow:

$$\eta_b = \eta_{GI} \times \eta_{GE} \times \eta_m \qquad (5)$$

Where

η_b = Brake efficiency

η_{GI} = Gross Indicated efficiency

η_{GE} = Gas-Exchange efficiency

η_m = Mechanical efficiency

Figures 7 to 9 shows all these different efficiencies for the two named cases with different speeds.

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

$$\eta_{GI} = \left(\frac{IMEP_{gross}}{\left(\frac{m_f Q_{LHV}}{V_D} \right)} \right) \qquad (6)$$

Where

m_f = fuel mass per cycle

Q_{LHV} = lower heating value of the fuel

V_D = displaced volume

Figures 7-9 show slightly higher gross indicated efficiency in the cases with regulator. By increasing EGR the specific heat ratio will be slightly lower and combustion duration will be longer but it can be compensated somewhat by advancing the ignition timing. The combustion efficiency increases however since the exhaust gas has a second chance to be combusted. The net result is a slight increase in gross indicated efficiency. As it was discussed before, by using the regulators the EGR valve will be open more resulting in increasing pressure after the throttle and thereby the throttle will be opened more in order to keep the same amount of load. Thus the gas exchange efficiency increases with EGR.

Gas-Exchange efficiency is a measure to evaluate the pumping losses in the engine.

$$\eta_{GE} = \left(\frac{IMEP_{net}}{IMEP_{gross}} \right) \qquad (7)$$

$$IMEP_{net} = \frac{\int_{-360}^{360} p\,dv}{V_D} \qquad (8)$$

$$IMEP_{gross} = \frac{\int_{-180}^{180} p\,dv}{V_D} \qquad (9)$$

Figures 7-9 show that gas exchange efficiency increases as the load increases due to the more open throttle resulting in higher inlet pressure. Figures 7 to 9 also show that the gas exchange efficiency is higher in the case with regulator since using EGR lets the throttle open even further to keep the load at same level. Table 4 shows how the inlet pressure increases with EGR.

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

$$\eta_m = \frac{BMEP}{IMEP_{net}} \qquad (10)$$

The differences in mechanical efficiency between running without EGR or with EGR are almost negligible, see Figure 7-9.

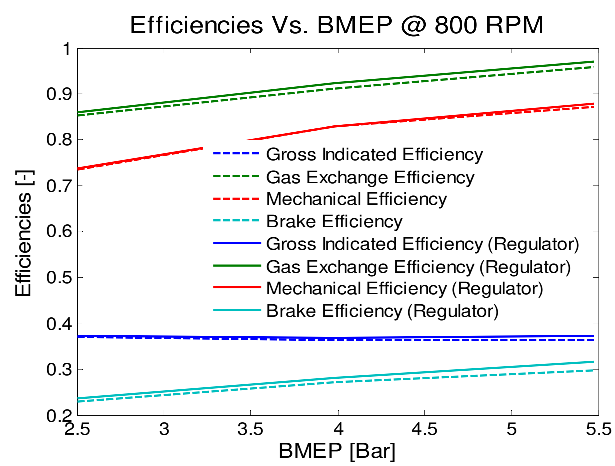

Figure 7: Efficiencies vs. BMEP @ 800 RPM

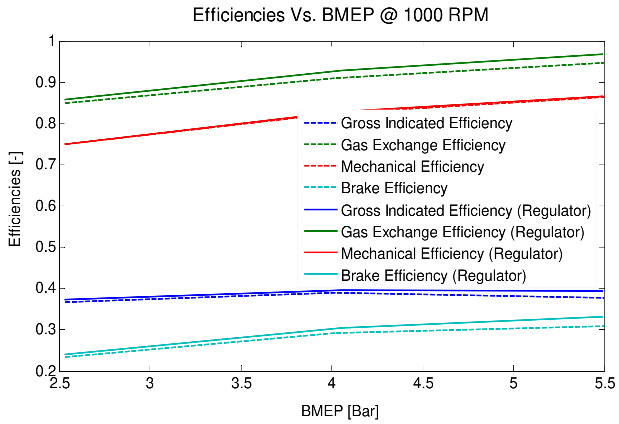

Figure 8: Efficiencies vs. BMEP @ 1000 RPM

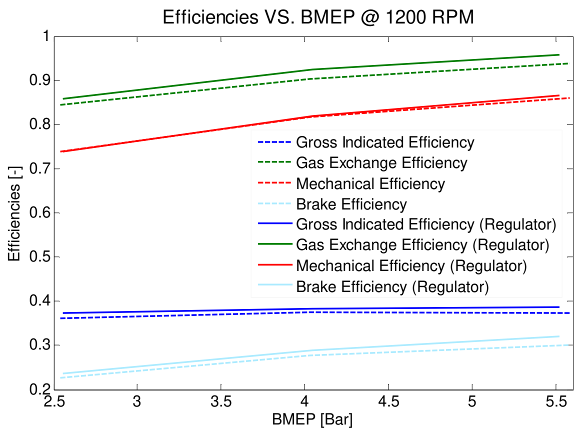

Figure 9: Efficiencies vs. BMEP @ 1200 RPM

Speed	BMEP	Strategies	Inlet P
800	2,5	NO EGR	0,48
800	2,5	With regulator	0,51
800	4	NO EGR	0,61
800	4	With regulator	0,71
800	5,5	NO EGR	0,75
800	5,5	With regulator	0,92
1000	2,5	NO EGR	0,47
1000	2,5	With regulator	0,51
1000	4	NO EGR	0.62
1000	4	With regulator	0,73
1000	5,5	NO EGR	0,75
1000	5,5	With regulator	0,89
1200	2,5	NO EGR	0,49
1200	2,5	With regulator	0,54
1200	4	NO EGR	0,62
1200	4	With regulator	0,75
1200	5,5	NO EGR	0,77
1200	5,5	With regulator	0,91

Table 4: Inlet pressures for different cases

Figure 10-12 show the stable region (the region where COVimep is lower than 5%) for different loads and speeds. X-axis shows BMEP in bar, Y-axis shows the rate of EGR in percentage and the colored region shows the level of COVimep. As load and speed increases, more EGR can be tolerated in the engine because of lower residual fraction and higher turbulence level. These figures also verify the effect of EGR on increasing of COVimep. The maximum EGR rate in different load and speed while the engine runs in a stable condition can be read from the figures.

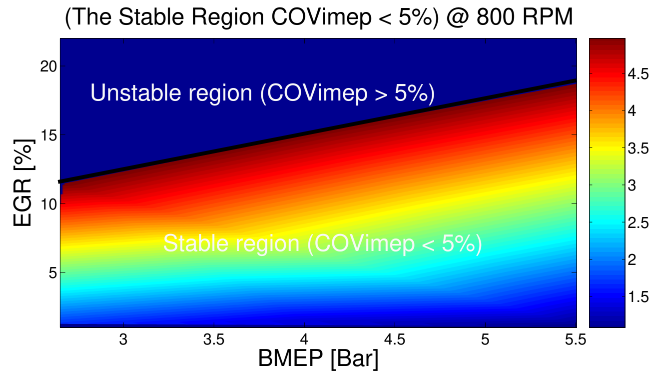

Figure 10: Stable region @ 800 RPM

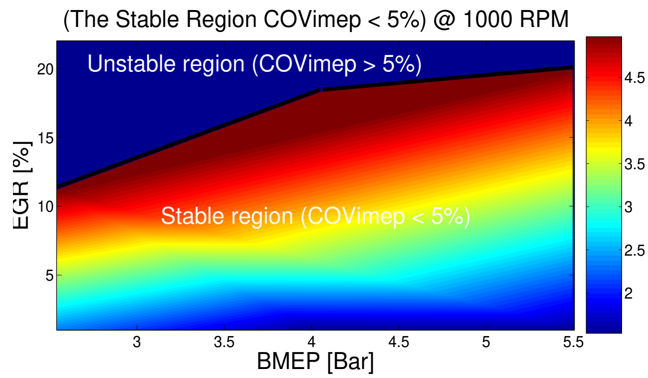

Figure 11: Stable region @ 1000 RPM

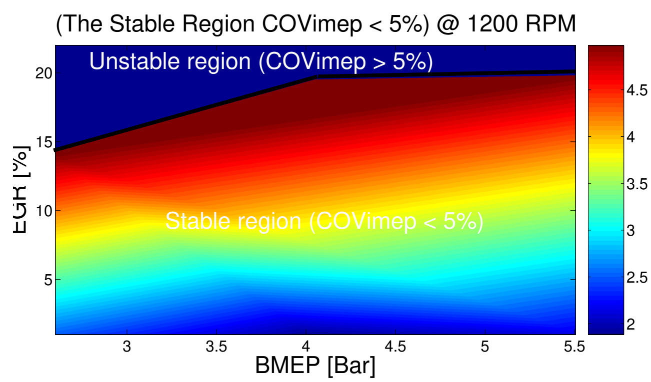

Figure 12: Stable region @ 1200 RPM

Figure 13-15 show the percentage of COVimep for different tested loads and speed. X-axis shows the EGR valve position in percentage and Y-axis shows the throttle position (bigger value means more open valve). The figures show that by increasing EGR, COVimep is increasing but, by opening more the throttle it decreases. It was also shown in figures 7-9 that, using the regulator which results in the highest possible amount of EGR gives the best Brake Efficiency. So the best Brake Efficiency can be found at the points where

COVimep is equal with 5% i.e. the highest possible opening of EGR valve for a corresponding throttle position, see figures 13-15, the dashed line.

The position values from EGR valve and throttle can be used in a map and use the map for running the engine at those points. It should be also mentioned that, it takes a short time for regulator to find the best positions for the first time but when the points are ready then a map can be made from those points and run the engine directly on those points i.e. the regulator's start values can be set by these points.

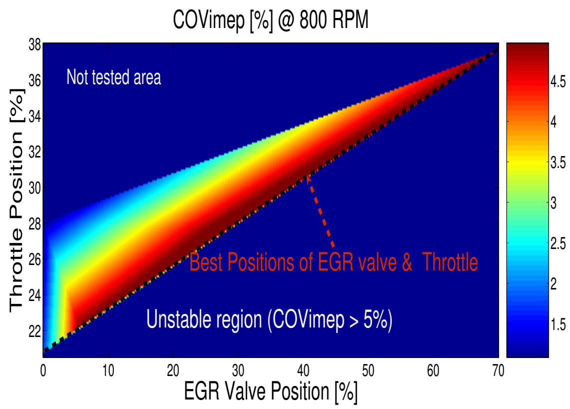

Figure 13: Best EGR valve & throttle position @ 800 RPM

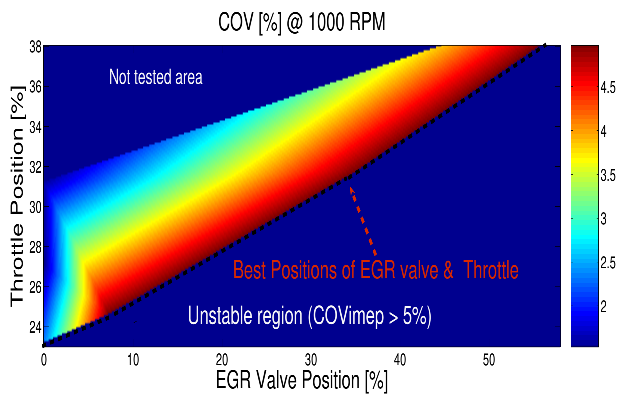

Figure 14: Best EGR valve & throttle position @ 1000 RPM

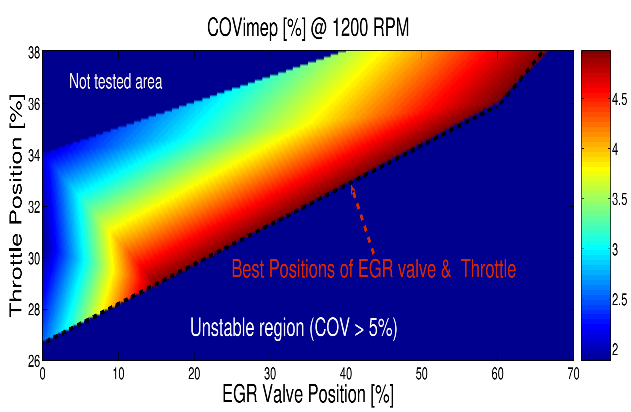

Figure 15: Best EGR valve & throttle position @ 1200 RPM

Pumping losses can be calculated and presented by means of mean effective pressure as follow:

$$Pmep = IMEP_{gross} - IMEP_{net} \qquad (11)$$

Figures 16-18 show Pmep for different loads and speeds. X-axis shows the EGR valve position in percentage and Y-axis shows the throttle position in percentage. The triangle shows the stable region for three loads (2.5, 4, and 5.5). When EGR valve opens more the pressure after throttle increases and the regulator opens the throttle more in order to keeps the engine at the same level of load which results in decreasing pumping losses.

The tests were performed at three loads viz. 2.5, 4 and 5.5 bar once without EGR and once with the regulators. The loads are shown in black lines in figures 16-18. Each black line shows the measuring of two points when the EGR valve is closed and when the EGR valve opens by regulator. Black lines in Figure 16-18 show that Pmep is decreasing in all the tests with regulator due to the more open throttle resulting in higher inlet pressure.

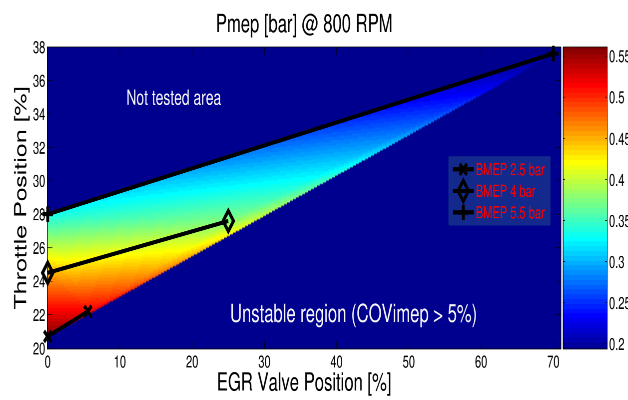

Figure 16: Pmep [bar] in stable region @ 800 RPM

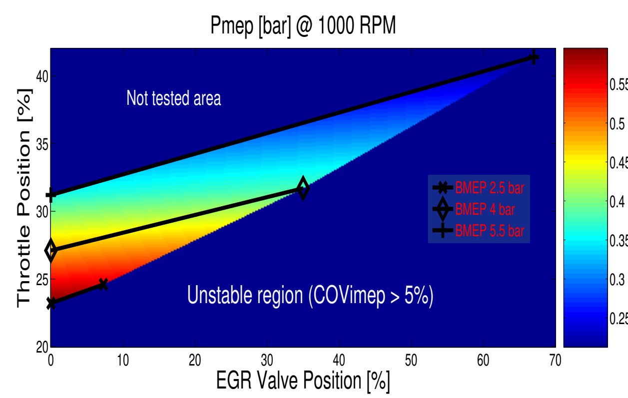

Figure 17: Pmep [bar] in stable region @ 1000 RPM

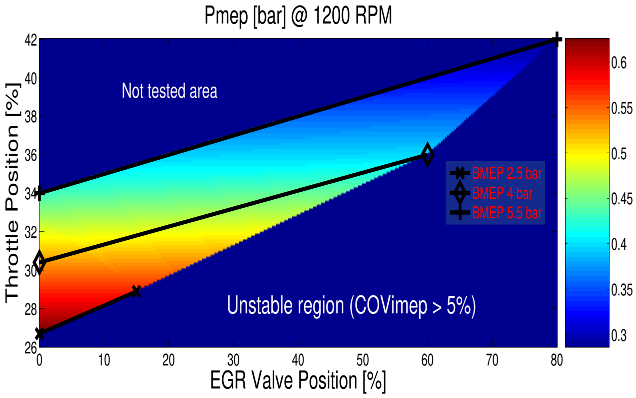

Figure 18: Pmep [bar] in stable region @ 1200 RPM

Figures 19-21 show specific fuel consumption for different loads and speeds. X-axis shows the throttle position in percentage and Y-axis shows EGR valve position in percentage. The triangle shows the stable region for three loads (2.5, 4, and 5.5). As throttle and EGR valve opens more the fuel consumptions decreases because of lower pumping losses and thereby better efficiency. It can also point out that by increasing EGR, fuel consumption increases (if throttle won't be opened more) due to more heat losses.

The loads are shown in black lines in figures 19-21. Black lines in Figure 19-21 show that fuel consumption is decreasing in all the tests with regulator due to the more open throttle resulting in less pumping losses and thereby better efficiency.

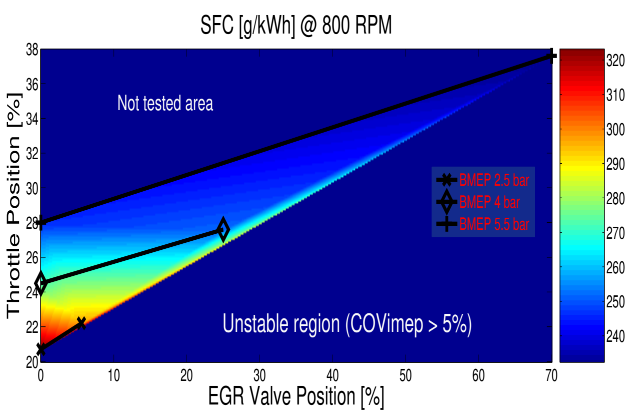

Figure 19: Specific fuel consumption [g/kWh] in stable region @ 800 RPM

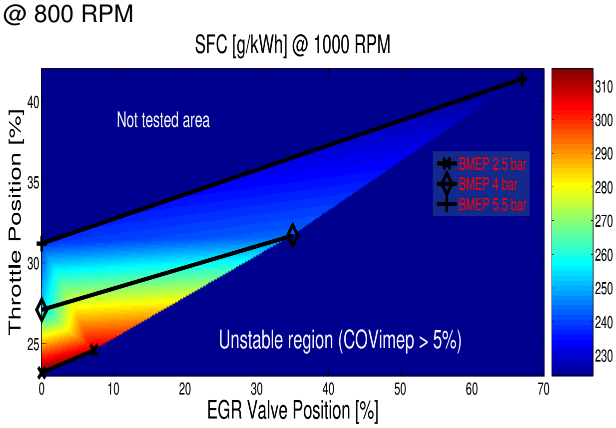

Figure 20: Specific fuel consumption [g/kWh] in stable region @ 1000 RPM

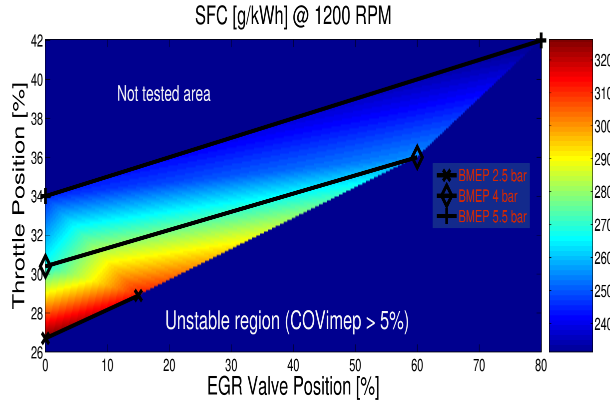

Figure 21: Specific fuel consumption [g/kWh] in stable region @ 1200 RPM

CONCLUSION

The results obtained from this study are as follow:

1. Controlling Lambda, Load and EGR was found to work well. The controller made it possible to have the maximum amount of EGR in the cylinder while keeping the COVimep less than 5% and load at a constant value.

2. The results verified 1.5-2.5 % improvement in Brake Efficiency by using the controller.

3. The controller is used as a tool for mapping the best positions of the throttle and EGR valve in terms of efficiency.

4. The mean based COV_{imep} calculation cannot be used under transient running but the filtered based method can be used under the both, steady state and transient conditions.

All the tests were performed at steady state and the next step is to make more modifications and test the controller at transient conditions. Automatic ignition timing controller using CA50 as feedback is one of the modifications which will be performed in the follow on program in order to be able to test engine under the transient conditions.

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