Cooled EGR for a Turbo Charged SI Engine to Reduce Knocking and Fuel Consumption

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Abstract

Cooled exhaust gas recirculation is emerging as a promising technology to address the increasing demand for fuel economy without compromising performance in turbocharged spark injection engines. The objectives of this study are to quantify the increase in knock resistance and to decrease the enrichment and emission at high load. For this purpose four stroke turbo charged Spark Ignition engine (EF7-TC) including its different systems such as inlet and exhaust manifold, exhaust pipe and engine geometry are modeled using GT Power Software. As predicted, using cooled EGR at high load enabled operation at lambda near to one with the same serial engine performances, which offers substantial advantages Such As BSFC reduction (up to 14%), and emission reduction (CO, NOx).

Keywords: *Knock, emission, fuel consumption, GT power*

1. Introduction

One of the major goals for manufacturers of turbocharged spark ignited (SI) engines is to increase knock resistance at high load. In order to move out of knocking combustion, the timing of the spark ignition is retarded, which results in the limitation of maximum pressure and temperature in the combustion chamber. This is an effective method to eliminate the knock. The negative effects of retarding the spark timing are, however, multiple [1]:

The thermodynamic efficiency of the engine decreases.

The exhaust gas temperature increases.

The thermal stress of the exhaust components, such as the exhaust valve, turbocharger and the catalyst increases, due to the higher temperature.

In order to reduce the exhaust gas temperature, rich mixtures are normally used. As a result, fuel consumption increases and the emission of CO and HC increase. So far this has not been considered as a problem due to the relatively short time the engine is running at full power. However, new legislation is expected, which will regulate high load emissions. Consequently, fuel enrichment during full load has to be reduced in order to meet the new legislation. An alternative and efficient solution that is emerging as a promising technology to limit knocking through dilution is cooled exhaust gas recirculation (EGR).

In engine revolution, EGR technique was firstly adopted in diesel engines in order to limit thermal NOX formation rate by reducing combustion chamber temperature thanks to the dilution of fresh charge with a certain amount of exhaust gases recycled at engine intake. Some studies also suggest that appropriate EGR ratio can reduce combustion noise by lowering the pressure rise rate and the pressure high frequency oscillation magnitude [2, 3]. But with the growing energy and environmental issues, various countries in the world have developed more stringent regulations on vehicle emission and fuel economy to drive gasoline engine to be developed towards downsizing direction. Turbo-charged spark ignition engines are becoming increasingly popular in the world market due to their compactness and high power density. However, due to the high power density of turbo-charged engine, knock combustion and high exhaust gas temperatures constitute make problems at high loads [4]. As a result, knock control is becoming increasingly important. Using EGR as diluents allows the use of an overall stoichiometric composition of the charge, so that it insures a good conversion of all emissions. Therefore, using cooled EGR instead of excessive fuel to inhibit knock and to reduce emission is an effective measure in gasoline engine [1, 5, and 6].

In some studies EGR has been used for reducing knock. Diana et al conducted tests with a naturally aspirated SI engine with the goal to raise the compression ratio [7]. With an addition of 11% EGR, they were able to run the engine with a compression ratio of 13.5 with spark timing set to achieve maximum brake torque. Unfortunately, introducing EGR to a naturally aspirated engine decreases the power output, since EGR replaces air. In a supercharged engine, however, EGR can be added without reducing the amount of oxidizer for the combustion. Brüstle and Hemmerlein [8] who investigated various methods to reduce full load fuel consumption, including EGR and lean A/F-ratios, also found that EGR indeed could be used for knock suppression. Grandin and Ångström [1] examined the potential to suppress knock and to reduce the exhaust gas temperature by using either lean mixtures or mixtures diluted with cooled EGR, at high load conditions, in a turbo charged spark ignited engine. They stated both HC and CO were reduced when an EGR was used instead of a rich mixture. Potteau and Leroux [5] evaluated the potential of several architectures for their capacity to achieve exhaust gas recirculation over the widest operating range. They selected and tested two architectures, one with high pressure EGR and the other with low pressure EGR, on a 2L turbocharged gasoline engine. They claimed using cooled EGR at high load enabled operation at lambda one with the same serial engine performances, which offers substantial advantages such as BSFC reduction and exhaust gas temperature reduction.

2. EF7-TC Engine Model in GT-POWER Software

2.1. Engine technical specifications

EF7-TC Engine model in GT-POWER software is a 0D model which different kinds of its input consist of those related to engine numeral, boundary and initial conditions and some of them are related to performance conditions of the engine.

Engine numerals consist of intake and exhaust manifold maps, combustion chamber map, piston map, crankshaft map of EF7-TC engine that was obtained from IPCO Company and also engine performance condition was replaced on the basis of IPCO Company laboratory information. Engine specifications are in Table 1. Certain boundary conditions on the basis of gathered information have been replaced.

2.2. Combustion Model

In this study two combustion models were investigated in order to predict burn rate: 1) Non-predictive combustion model, 2) Predictive combustion model.

A non-predictive combustion model simply imposes a burn rate as a function of crank angle. This prescribed burn rate will be followed regardless to the conditions in the cylinder, assuming that there is sufficient fuel available in the cylinder to support the burn rate. Therefore, the burn rate will not be affected by factors such as EGR or injection timing. This may be appropriate as long as the intended use of the model is to study a variable which has little effect on the burn rate.

Table 1: Engine specifications

Number of cylinders	4
Bore	78.6mm
Stroke	85mm
Conrad	134.5mm
Compression ratio	9.5
Max. power	110 kW in 5500 rpm
Max. torque	215 N.m (2000 rpm-4800 rpm)

However, a non-predictive model may not be a good choice when the intended use of the model is to study a variable that has a direct and significant effect on the burn rate such as EGR. In that case, a predictive combustion model is a more appropriate choice so the burn rate will respond appropriately to the change in the variable of interest. In theory, predictive combustion models are an appropriate choice for all simulations. As mentioned before, EGR affects burn rate, so predictive combustion model must be used.

Predictive combustion model predicts the burn rate for homogeneous charge, spark-ignition engines and is based on the work in references 9, 10, and 11. This prediction takes into account the cylinder's geometry, spark-timing, air motion, and fuel properties. The mass entrainment rate into the flame front and the burn rate are governed by the following three equations:

$$\frac{dM_e}{dt} = \rho_u A_e (S_T + S_l) \tag{1}$$

$$\frac{dM_b}{dt} = \frac{(M_e - M_b)}{\tau} \tag{2}$$

$$\tau = \frac{\lambda}{S_L} \tag{3}$$

These equations state that the unburned mixture of fuel and air is entrained into the flame front through the flame area at a rate proportional to the sum of the turbulent and laminar flame speeds. The burn rate is proportional to the amount of unburned mixture behind the flame front, (Me-Mb), divided by a time constant, τ. The time constant is calculated by dividing the Taylor micro scale, λ, by the laminar flame speed.

2.3. Knock model

Empirical induction-time correlations are usually used for predicting knock. Induction-time correlations are derived by matching an Arrhenius function to measure date on induction or auto ignition times, for given fuel-air mixtures, over the relevant mixture pressure and temperature. According to above it is assumed that auto ignition occurs when [9, 10]:

$$\int_{t=0}^{ti} \frac{dt}{\tau} = 1 \tag{4}$$

A number of empirical relations for induction time for individual hydrocarbons and blended fuels have been developed from fundamental or engine studies of auto ignition. These relations have the form of [11, 12]:

$$\tau_{reac} = AP^{-n} \exp(\frac{B}{T}) \tag{5}$$

2.4. NOx calculation:

The NOx calculation is based on the extended Zeldovich mechanism. k1, k2, and k3 are the rate constants that are used to calculate the reaction rates of the three equations below, respectively [12].

$$O + N_2 = NO + N \tag{6}$$

$$N + O_2 = NO + O \tag{7}$$

$$N + OH = NO + H \tag{8}$$

$$K_1 = F_1 * 7.6 * 10^{10} * e^{\frac{-38000*A_1}{Tb}} \tag{9}$$

$$K_2 = F_2 * 6.4 * 10^{6} * e^{\frac{-3150*A_2}{Tb}} \tag{10}$$

$$K_3 = F_3 * 4.1 * 10^{10} \tag{11}$$

Where:

F1 = N2 Oxidation Rate Multiplier
F2 = N Oxidation Rate Multiplier
F3 = OH Reduction Rate Multiplier
A1 = N2 Oxidation Activation Temperature Multiplier
A2 = N Oxidation Activation Temperature Multiplier
Tb = Burned Sub-zone Temperature (K)

2.5. CO calculation

The CO calculation is based on the following mechanism and was developed for homogenous combustion [12].

$$\mathrm{CO + OH = CO_2 + H} \tag{12}$$

$$K_1 = A * 6.76 * 10^7 * e^{\frac{T}{B*1102}} \tag{13}$$

where:
A = Pre-exponent Multiplier
B = Activation Temperature Multiplier

2.6. Calibration and Validation Model with Experimental result

Predictive combustion models attempt to model the important physics in the combustion process in order to predict the combustion burn rate. This implies that the model should automatically adjust changing conditions (engine speed, EGR rate, etc.) with no change in the model inputs. Realistically, predictive combustion models include assumptions and simplifications, and therefore will require some calibration of the physical constants to best match the specific combustion system. The goal of predictive combustion model calibration process is to find the single set of model constants that will provide the best possible match to a wide variety of operating points.

In this model there are three multipliers for calibration as follows:

1- Flame Kernel Growth Multiplier: Multiplier used to scale the calculated value of the growth rate of the flame kernel. This variable influences the ignition delay. Larger numbers shorten the delay, advancing the transition from laminar combustion to turbulent combustion.

2- Turbulent Flame Speed Multiplier: Multiplier used to scale the calculated turbulent flame speed. This variable influences the overall duration of combustion. Larger numbers increase the speed of combustion.

3- Taylor Length Scale Multiplier: Multiplier used to scale the calculated value of the "Taylor microscale" of turbulence. The "Taylor microscale" modifies the time constant of combustion of fuel/air mixture entrained into the flame zone by changing the thickness of the plume.

Default values of these multipliers in the software are 1. To calibrate this model, first at one operation condition multipliers were found in order to calculate burn rate so that it would become similar to experimental burn rate. For this reason multipliers at full load and 5000 rpm were calibrated according to experimental result. Figures 1 and 2 shows experimental calculated burn rate and cylinder pressure after calibrating multipliers. After calibration of multipliers, this model can predict burn rate at other loads and speeds. Figures 3 and 4 show the ability of this model to predict burn rate at full load and 3000 rpm. As shown in this figure, this model has very good ability to predict burn rate.

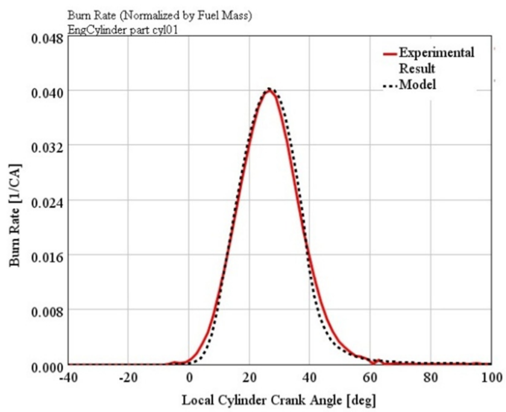

Fig1. Comparison of calculated burn rate with experimental date at 5000 rpm and full load after calibrate multipliers

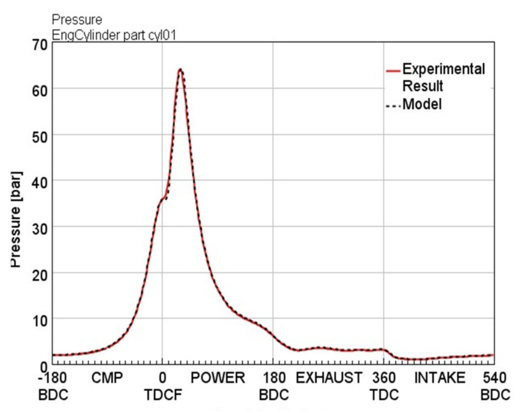

Fig2. Comparison of calculated cylinder pressure with experimental date at 5000 rpm and full load after calibrate multipliers

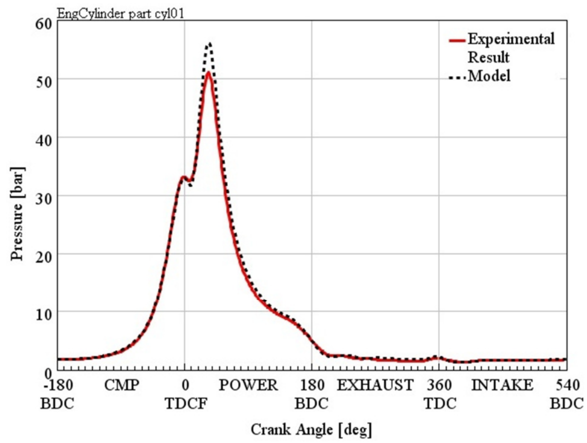

Fig3. Investigate performance of predictive combustion model in predicting burn rate at 3000 rpm and full load

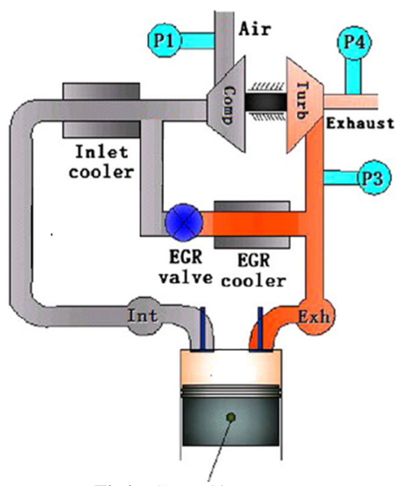

Fig4. EGR architecture

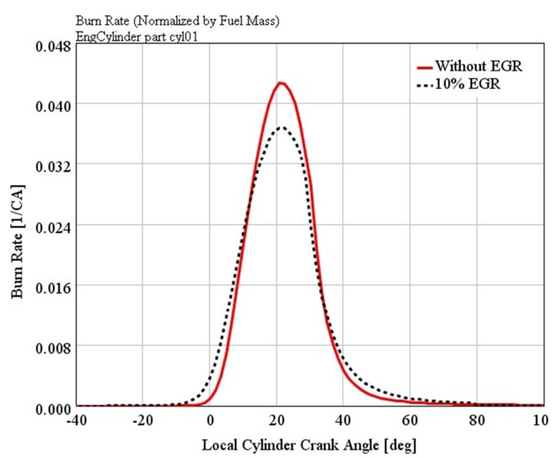

Fig5. Comparison of cylinder pressure with and without EGR at 3000 part loa

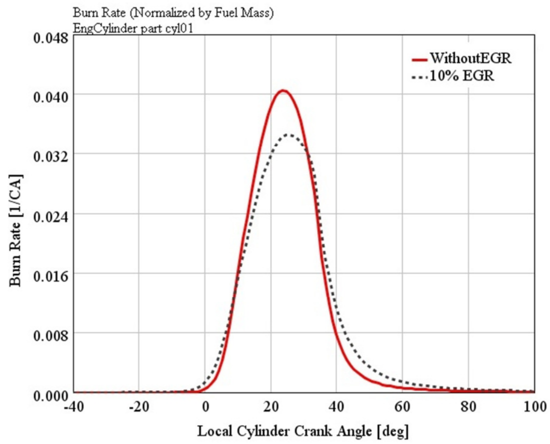

Fig6. Comparison of cylinder pressure with and without EGR at 5000 full load

3. Result and Discussion:

In this section, the potential to reduce fuel consumption and emission by using mixtures diluted with cooled EGR, at high load conditions, in a turbo charged spark ignited engine is examined. In this case the gases are extracted before the turbine comes from cylinders 1&4. The EGR gases are introduced upstream of the charge air cooler. The EGR analyzed architectures are depicted in Figure 4.

3.1. Combustion duration

Increased dilution leads to a lower laminar flame speed, which prolongs both the transition to a fully developed turbulent flame, after the spark has ignited the charge, and the main combustion duration. Rhodes and Keck [13] showed that the laminar flame speed decreased as much as 50 % of its undiluted stoichiometric value with a 15 % EGR dilution. The decreased laminar flame speed, due to increased dilution, results in prolonged combustion duration. In Figures 5 and 6, the combustion duration for 0-90 % burnt charge with different EGR is displayed. As the EGR rate is increased, the combustion duration is increased too.

Due to the prolonged combustion of the diluted charges, maximum cylinder pressures are kept low. A reduction of the knock intensity can therefore be anticipated due to a reduction of the combustion related compression of the end-gas, which results in lower endgas temperature.

3.2. Knock

The end-gas temperature and consequently the propensity to autoignite, is expected to be reduced with cooled EGR, due to two effects. Firstly, due to the prolonged combustion, maximum pressures are reduced and consequently, the compression of the end-gas as a result of combustion is decreased. Secondly, the specific heat capacity of the charge is increased, with EGR in comparison to an undiluted mixture. This results in decreasing of end gas temperature. Because of the two mentioned reasons, the intense of knock decreases by using EGR. So using EGR caused the spark timing to advance and then decreased enrichment.

3.2. Fuel Consumption:

Using EGR decreased the enrichment at full load. In the high load cases with a rich mixture, fuel consumption is very high. Naturally consumption is reduced when the fuel enrichment is replaced by EGR. This is shown in Figure 7, where the fuel consumption of a case with 10 % EGR is compared with a case with fuel enrichment.

3.3. NOx Emission

Exhaust gases of an engine can contain more than 2000 ppm of oxides of nitrogen. Released NOx reacts in the atmosphere to form ozone and is one of the major causes of photochemical smog. In this case the most important engine variables that affect NO emissions are the fuel/air equivalence, the burned gas fraction and spark timing. The burned gas fraction depends on the amount of diluents such as EGR. In this case decreasing the enrichment and advancing spark timing increases the NOx emission. On the other hand using EGR caused NOx emission to decrease. As shown in Figure 8 NOx emission decreased. So EGR has more effect than other two parameters in this case.

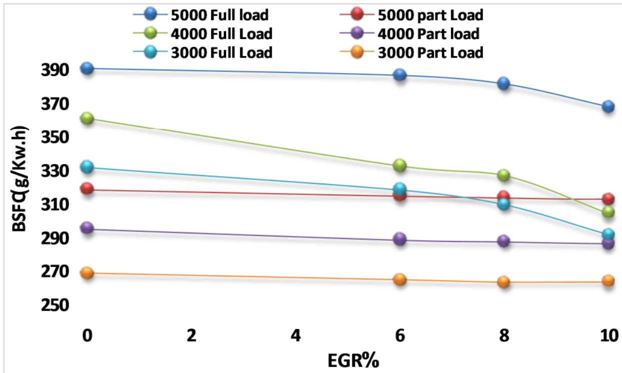

Fig7. Comparison of fuel consumption at high load

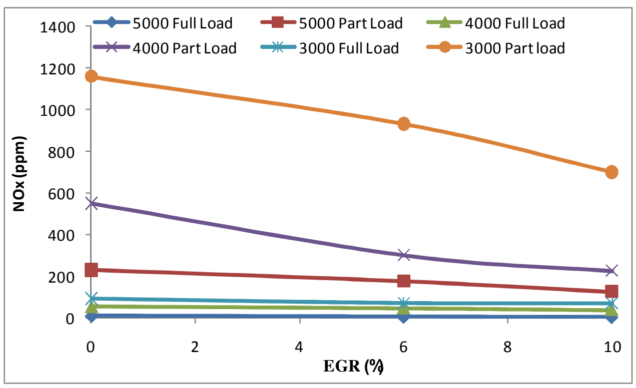

Fig8. Comparison of NOx emission at high load

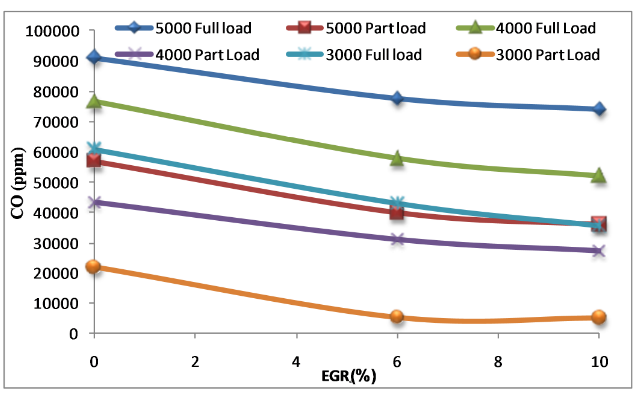

Fig9. Comparison of CO emission at high load

3.3. CO Emission

Carbon monoxide, a colorless, poisonous gas, is generated in an engine when it is operated with a fuel-rich equivalence ratio. When there is not enough oxygen to convert all carbon to CO2, some fuel does not get burned and some carbon ends up as CO. Maximum CO is generated when an engine becomes rich. As it is shown in figure 9, CO emission decreased when EGR was used due to decreasing the enrichment at full load.

4. Conclusion:

It has been shown in this paper that cooled EGR can be used at high power in a turbo charged engine in the replacement of a rich charge. As predicted, using cooled EGR at high load enabled operation at lambda near to one with the same serial engine performances, which offers substantial advantages such as BSFC reduction (up to 14%). Emissions of NOX and CO can be substantially reduced with cooled EGR in comparison with fuel enrichment.

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