

EGR Effects on Boosted SI Engine Operation and Knock Integral Correlation

Bjoern Hoepke, Stefan Jannsen, Emmanuel Kasseris and Wai K. Cheng
Massachusetts Institute of Technology

ABSTRACT

The effects of cooled exhaust gas recirculation (EGR) on a boosted direct-injection (DI) spark ignition (SI) engine operating at stoichiometric equivalence ratio, gross indicated mean effective pressure of 14-18 bar, and speed of 1500-2500 rpm, are studied under constant fuel condition at each operating point. In the presence of EGR, burn durations are longer and combustion is more retard. At the same combustion phasing, the indicated specific fuel consumption improves because of a decrease in heat loss and an increase in the specific heat ratio. The knock limited spark advance increases substantially with EGR. This increase is due partly to a slower combustion which is equivalent to a spark retard, as manifested by a retarded value of the 50% burn point (CA50), and due partly to a slower ignition chemistry of the diluted charge, as manifested by the knock limited spark advance to beyond the value offered by the retarded CA50. The latter part adds 0.5 ms to the spark advance per 10% increase in EGR rate. A knock correlation based on the Livengood-Wu integral method is developed to include the effect of EGR on the ignition delay. A 10% point increment of the EGR rate corresponds to approximately a 3% increase in the effective fuel octane number.

CITATION: Hoepke, B., Jannsen, S., Kasseris, E. and Cheng, W., "EGR Effects on Boosted SI Engine Operation and Knock Integral Correlation," *SAE Int. J. Engines* 5(2):2012, doi:10.4271/2012-01-0707.

INTRODUCTION

Turbocharged downsized Spark Ignition (SI) engines offer substantial fuel economy improvements over naturally aspirated SI engines at the same engine output [1, 2]. Because of the smaller displacement, the downsized engines operate at a much higher mean effective pressure (MEP) at the same torque output; whence the losses (pumping, mechanical friction and heat transfer) relative to the fuel energy input are reduced. The reduced air breathing capacity of the smaller displacement is compensated for by turbocharging when high output is required from these engines. Because of the high MEP values, knock presents a severe constrain to the high load limit of the engine operating domain. Charge dilution is an effective strategy to alleviate this constrain. Cooled exhaust gas recirculation (EGR) is the preferred means [3,4,5,6,7,8,9,10]. Dilutions with excess air [11] and water vapor [12] have also been attempted. The cooled EGR suppresses knock so that combustion phasing could be advanced to more optimal setting; pumping loss and burned gas temperature are also reduced. The latter effect reduces heat and dissociating losses, and increases the specific heat ratio. All these factors contribute to improved specific fuel consumption.

In this paper, the cooled EGR effects on the operation of a boosted direct injection (DI) SI engine at intermediate load are assessed. The objective is to deconstruct the effects into various physical factors through a combination of data analysis and engine simulation studies. Then a practical knock correlation that includes the EGR effect in the form of the Livengood-Wu integral [13] is developed.

COOLED EGR EFFECT ON KNOCK

In the following discussion, the same fuel and air trapped mass is considered (at fixed λ) so as to produce the same level of engine output. Thus when EGR is introduced, the manifold absolute pressure (MAP) has to increase to compensate for the displacement of air by the EGR gas. Then dilution by EGR has the following effects:

CHANGE OF BURN RATE

Dilution decreases the laminar flame speed and leads to slower combustion. This fact has to be balanced against the increase in turbulence in the engine, hence an increase in the burn rate, due to a higher Reynolds number and presence of higher turbulent kinetic energy (both due to higher charge density); the latter renders a longer decay of the intake motion generated turbulence. Furthermore, a slower

combustion has a similar effect as retarding the combustion phasing. If the spark timing is advanced to maintain the same phasing, however, the change of burn rate would have little effect on the engine knocking behavior. So a slower burn rate per se, does not significantly improve the engine knock resistance. It may even make the engine more prone to knock because the end gas is not consumed as quickly as in a fast burn system.

CHANGE OF IGNITION CHEMISTRY

The end gas temperature is low (typically less than 900K); dissociation of the EGR species should be negligible. Hence the major EGR gases for stoichiometric combustion should function just as inert gases. At the same fuel and air concentrations, the additional presence of diluents increases the extent of quenching reactions [14, 15], thereby leads to an increase in ignition delay. This increase is the dominant mechanism behind the effectiveness of EGR as a means for knock control.

CHANGE OF END GAS TEMPERATURE

The fuel has a high specific heat ($c_{p,f}$) which substantially contributes to the specific heat (c_p) of the charge. Since the specific heat ratio γ is $1/(1-R/c_p)$, higher value of c_p leads to lower value of γ . Adding EGR to the charge decreases the mass fraction of the fuel in the mixture because of dilution. Although the specific heat of the EGR gas is higher than that of air, the reduction of the fuel mass fraction overpowers, and leads to a net decrease of the charge c_p ; thus EGR produces an **increase** in the value of γ . The effect is shown in Fig. 1 (from Ref. [3]). When EGR is changed from 0 to 36%, the value of $(\gamma - 1)$ increases by approximately 2.5%.

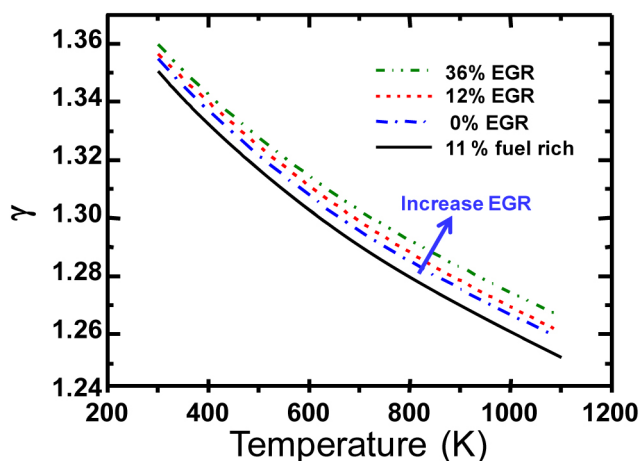


Figure 1. Specific heat ratio γ for fuel rich unburned mixture and for mixtures at different levels of EGR; PRF95 fuel; from Ref. [3].

The increase in the compression temperature (T_2) may be estimated from adiabatic compression (symbols are defined at the end of the paper):

$$\frac{dT_2}{T_2} = \ell n(\varepsilon) d(\gamma - 1) \quad (1)$$

For a compression ratio ε of 10, a 2.5% increase in $(\gamma - 1)$ would result in a 6% increase in the compression temperature.

The unburned gas temperature T_u of the end gas as a result of compression by the heat release process, however, behaves differently. The behavior may be illustrated by a simple ideal gas model for the engine combustion process. To a good approximation, the end gas is compressed adiabatically by the prevailing pressure; whence

$$\frac{\dot{T}_u}{T_u} = \frac{(\gamma - 1)}{\gamma} \frac{\dot{P}}{P} \quad (2)$$

The pressure rise rate may be calculated from the 1st law of thermodynamics:

$$\frac{\dot{P}V + P\dot{V}}{(\gamma - 1)} = \dot{Q} - P\dot{V} \quad (3)$$

Combining (2) and (3)

$$\frac{\dot{T}_u}{T_u} = \frac{(\gamma - 1)}{\gamma} \left[\frac{(\gamma - 1) \dot{Q}}{PV} - \frac{\dot{V}}{V} \right] \quad (4)$$

For the same trapped fuel, the heat release rate $\dot{Q}$ is approximately the same. With dilution, however, the pressure P is higher because of the higher charge mass. Thus according to eq. (4), compared with the no EGR case, the increase of temperature due to compression of the end gas by the burned gas will be smaller when EGR is introduced.

The above effect is illustrated in Fig. 2, which shows the cycle simulation results from EGR variation; see figure caption for operating condition. The unburned gas temperature T_u at spark increases with increase of EGR. However, as combustion progresses, the temperature curves cross over so that T_u **decreases** with increase in EGR. The decrease of the peak end gas temperature is shown in Fig. 3 as a function of EGR. The results are not sensitive to the engine load. The decrease is 5° K at 10% EGR; 12° K at 20% EGR; and 20° K at 30% EGR.

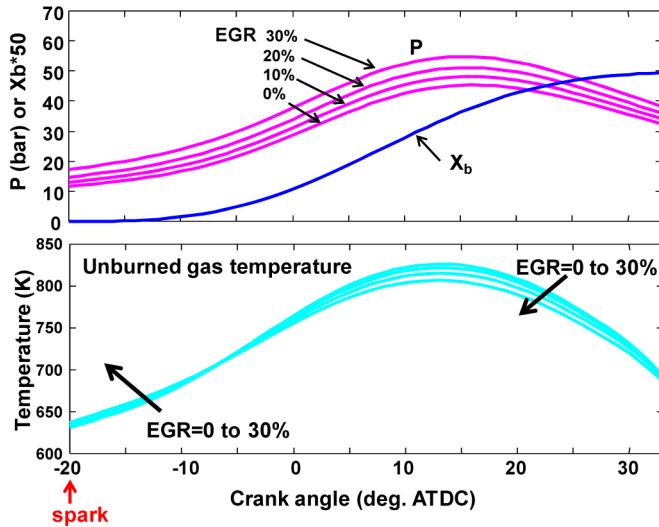


Figure 2. Cycle simulation results for a 0.5 L per cylinder displacement SI engine; 1500 rpm; $\varepsilon = 9.2$. EGR from 0 to 30% with constant fuel mass (34.8 mg) and $\lambda = 1$. GIMEP = 10.5 bar at EGR = 0%.

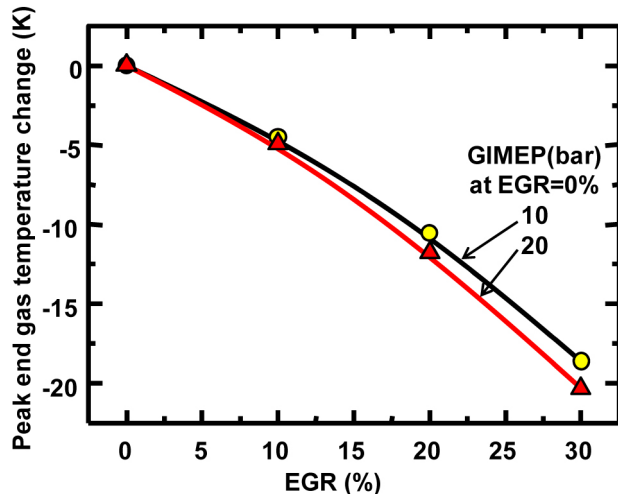


Figure 3. Change in maximum end gas temperature as a function of EGR; constant fuel mass. The two curves are for GIMEP = 10 and 20 bar at EGR = 0%

EXPERIMENT

ENGINE SET UP

The engine used was a 2.0 L 4-cylinder production turbo-charge engine (GM LNF engine) with direct injection and cam phasing control. The engine specifications are shown in Table 1. The turbocharger was a BorgWarner K04 with a twin-scroll turbine design and with an integrated wastegate for load control.

Table 1. Engine Specification of the GM LNF Direct Injection SI engine; the valve timing is a function of speed and load according to the production engine calibration. Values given in the table are for 1500 rpm, 14 bar GIMEP for reference.

Cylinders	I-4
Displaced volume	2.0L
Bore / Stroke	86 mm / 86 mm
Compression ratio	9.2
Inlet Valve Open	31° bTDC gas exchange
Inlet Valve Close	19° aBDC compression
Exhaust Valve Open	18° bBDC expansion
Exhaust Valve Close	22° aTDC gas exchange

A low pressure water-cooled EGR loop was added to the production engine which did not utilize EGR. A schematic of the engine system is shown in Fig. 4. The amount of EGR was controlled manually by a EGR valve. An existing CO₂ meter with a 25% full scale was deemed not sensitive enough to measure the dilution level at low EGR level. A turbine flow meter (opted for because of the lower cost than a sensitive CO₂ meter) was then used to determine the EGR flow rate, from which the EGR mass fraction (x_{EGR}) in the charge was calculated. Comparison of the x_{EGR} determined in this manner and the value calculated from a cycle simulation program (GT-Power) by using the engine operating parameters and by matching the pressure trace, however, showed that the flow meter had significantly over-measured the flow at low flow rates. Therefore only the x_{EGR} values from the matching cycle-simulation program are reported here. A plausible cause for the inaccuracy of the turbine meter is due to the impact of condensed water on the turbine blade at low flow rate. A CO₂ meter with the appropriate sensitivity has been acquired for future work.

The EGR temperature was controlled manually by regulating the coolant water flow in the EGR cooler. The range was from 50° C (at the lowest EGR flow rate with the coolant turned off) to 140° C (at the highest EGR flow rate with the coolant fully on). The dew point of the stoichiometric exhaust stream (with 13% water) is at 52° C; therefore at low EGR rate, there could be water condensation. However, the condensed water is not drained so that it will evaporate when it is mixed with the fresh air and then goes through the compressor, intercooler and air heater. The EGR flow and fresh air were compressed by the compressor, cooled by an intercooler, and then re-heated to a fixed intake temperature (40° C). Depending on the turbocharger operating point, the air flow controlled either by the throttle or, with the throttle fully opened, by the wastegate.

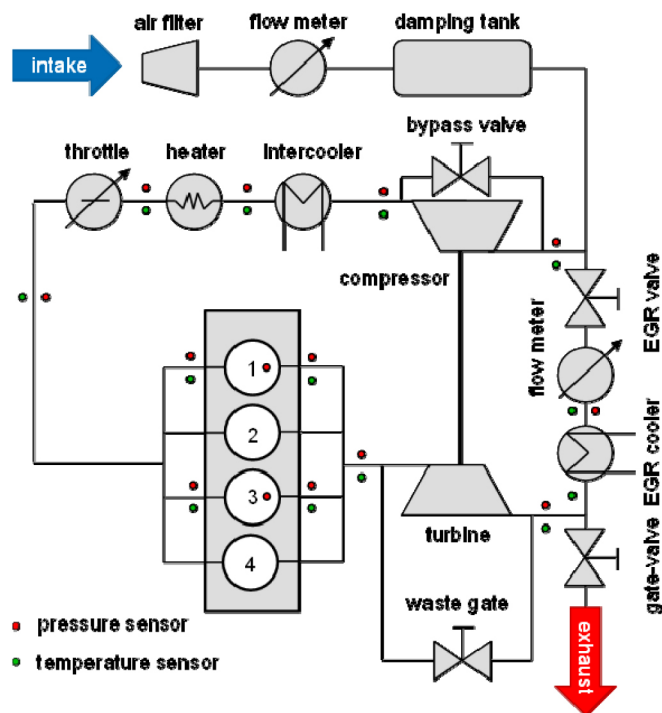


Figure 4. Schematic of the engine and the low pressure EGR loop.

The fuel used was a certification gasoline (EPA Tier II EEE) with 97 RON and 88.5 RON. To obtain optimum timing information without the knock constrain, a high octane fuel produced by mixing the certification gasoline with 30% toluene (by volume) was used.

RANGE OF OPERATION

The operating region of interest is in the low speed and high load range. Data have been taken from at nominal GIMEP values of 14, 16, and 18 bar respectively as shown in Fig. 5. The procedure of the experiment is, at different EGR levels, to meter in the same amount of injected fuel (by adjusting the air intake via the boost pressure, since λ is maintained at 1 with a feedback loop). Therefore the GIMEP values are nominal ones since they change with the amount of EGR. The change in GIMEP, however, is small; the values change by less than 3% from the nominal values for all the test points in Fig. 5. Furthermore, when the spark timing is changed, the turbocharger would deliver a different amount of boost; thus the amount of fuel (at $\lambda = 1$) also change. The wastegate and throttle position have not been readjusted to compensate for this small fuel difference. The fuel change is also small: less than $\pm 1.5\%$. To take into account of all these variations, the specific fuel consumption is used as the metric of performance instead of the MEP values.

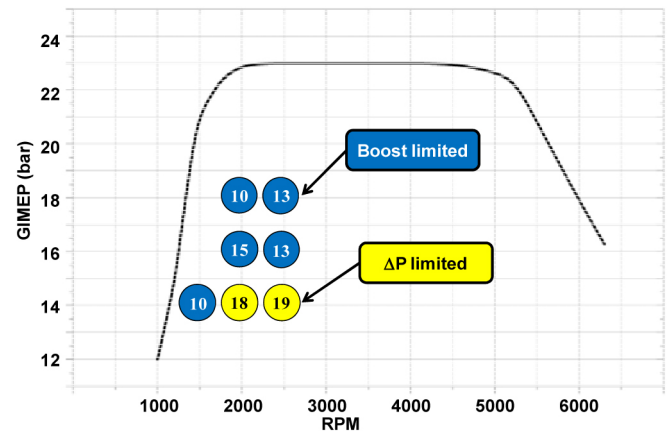


Figure 5. Operating points for the EGR data. The value on each circle indicates the maximum EGR rate that the engine could operate at that load and speed point.

The maximum EGR rates at which the engine could operate are also shown in Fig. 5. At the high loads, the operation was boost limited; i.e. the turbocharger could not deliver enough boost (with the wastegate fully closed) to compensate for the air displaced by the EGR. At the lower loads, the EGR flow was limited by the pressure differential between the exhaust and the intake (with the EGR valve fully open). Although the EGR rate at the lower loads could be increased by throttling the exhaust, the turbine back pressure has been kept at values consistent with the catalyst pressure drop.

DATA PROCESSING

Pressure data were taken at 100 kHz for over 100 cycles for knock analysis. Data were also taken for 50 cycles at 1° crank angle (CA) resolution for heat release analysis which was done by the Rassweiler and Withrow method [16]. For the knock analysis using the 100 KHz data, the individual cycle pressure signal has been processed by a 2.3kHz high pass filter, and the knock intensity (KI) is defined as the maximum amplitude of the high-passed pressure signal; see Fig. 6. An individual cycle is considered as a knocking cycle if its KI is larger than a threshold value of 1 bar. This KI value has been conservatively set; however, KI increases drastically with spark advance so that the knock-limited-spark-advance (KLSA) value is not sensitive to the KI threshold value. For the n cycles of recorded knock data, the probability of knock occurrence p_K is:

$$p_K = \frac{\text{Number of knocking cycles}}{n} \quad (5)$$

When p_K exceeds 10%, the operating point is considered to be knocking. The KLSA value is obtained by advancing the spark in steps of 0.75° CA until the engine knocks according to the above criterion. To avoid severe knocking