Bulletin of the *Transilvania* University of Braşov
Series I: Engineering Sciences • Vol. 4 (53) No. 1 - 2011

THE INFLUENCE OF THE INTERNAL EXHAUST GAS RECIRCULATION (EGR) ON THE PUMPING LOSSES

R. COSGAREA[1] M. ALEONTE[1] C. COFARU[1]

Abstract: *The exhaust gas recirculation (EGR) is one of the most used methods to control the start of combustion in the homogeneous charge compression ignition engines. To obtain the homogeneous combustion starting from a gasoline engine the internal EGR strategy has to be used to raise the temperature of the fresh charge. Internal EGR is achieved closing the exhaust valve before the top dead centre to trap a part of the burned gases into the cylinder. This paper presents the results of the simulations made to investigate the effects of the internal EGR on the pumping losses. The simulations were made for three operating where the exhaust valve is closed with different advances before the top dead centre.*

Key words: *homogeneous charge compression ignition engines, internal EGR, pumping losses, variable valve timing.*

1. Introduction

The internal EGR is used at the gasoline HCCI engines to control the start of the combustion [3]. The internal EGR can be obtained in two modes: by closing the exhaust valve with an advance before the top dead centre (TDC) or by reopening the exhaust valve during the intake process. Using the first method the burned gases are trapped inside de cylinder and compressed by the movement of the piston to the TDC [1]. To avoid the repression of the gases into the intake manifold, due to the higher pressure from the cylinder, the intake valve has to be opened with a delay after the TDC. During the beginning of the intake stroke the trapped gases are expanding due to the piston movement from the TDC to the bottom dead centre BDC.

To obtain different EGR rates variable valve timing (VVT) mechanisms have to be used. The cam phasers shown in Figure 1 are used to displace the lifts of the intake and exhaust valves. The disadvantage of this type of mechanisms is the fact that the lift curve can not be modified, it can only be displaced.

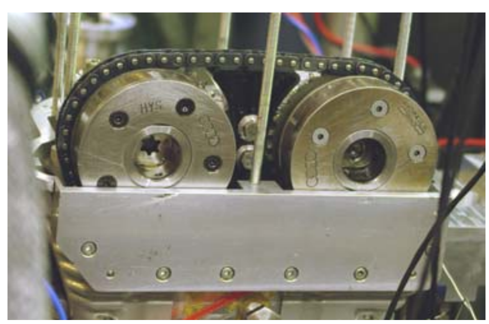

Fig. 1. *VVT with cam phasers*

[1] Centre "High-tech Products for Automotives", *Transilvania* University of Braşov.

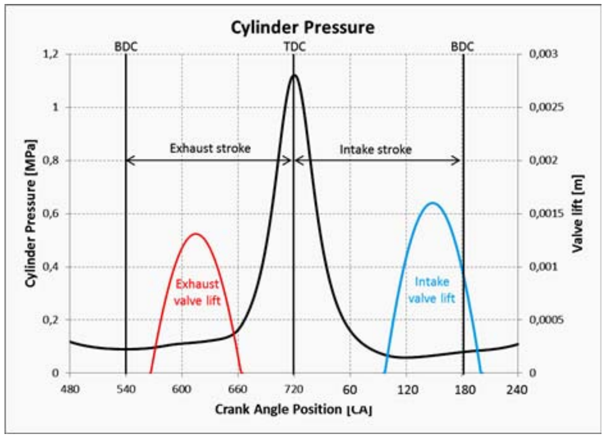

Fig. 2. *The cylinder pressure during the gas exchange process and the valve lifts*

Due to the compression of the burned gases at the end of the exhaust stroke and to the expansion during the beginning of the intake stroke a pressure raise appear on the indicated pressure diagram near the TDC corresponding to the gas exchange process [5], as shown in Figure 2 together with the valve lifts used.

Due to the increase of the pressure during the gas exchange process the pumping diagram is different. To investigate the effects of the increase of the cylinder pressure a simulation was made for three operating points with different EGR rates.

2. The Simulations

The simulation was made using the software AVL BOOST, an advanced virtual engine simulation tool. Three operating points were used with different exhaust valve lifts. The intake valve lift was maintained constant during the measurements. The valve lift strategies can be seen in Figure 3. The parameters used during the gas exchange simulations can be seen in Table 1.

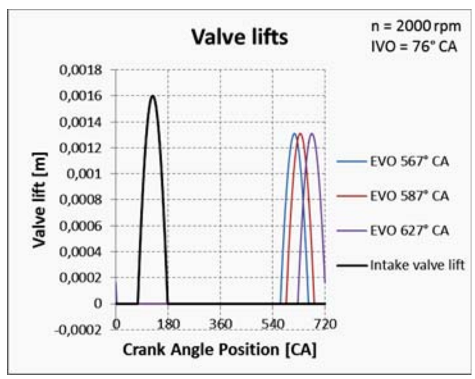

Fig. 3. *The valve lifts*

The engine model is made from 2 system boundaries, 2 plenums and one cylinder. The intake and exhaust pipes are connecting the system boundaries to the cylinder through

Engine model parameters Table 1

Point	Engine speed	Intake valve opening	Intake valve closing	Exhaust valve opening	Exhaust valve closing
[-]	[rpm]	[°CA]	[°CA]	[°CA]	[°CA]
1	2000	76	183	567	668
2	2000	76	183	587	688
3	2000	76	183	627	8

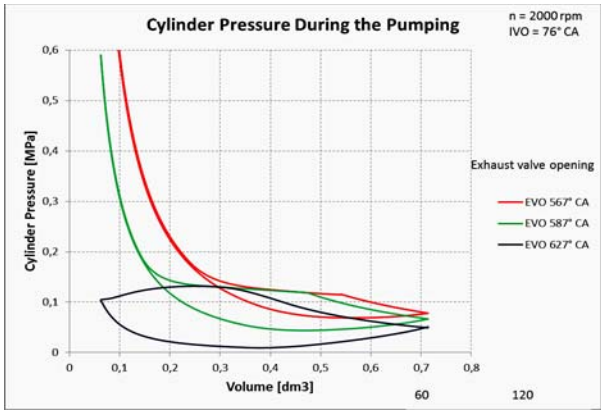

Fig. 4. *The cylinder pressure during the pumping*

the plenums. The engine model was design only to study the influence of the variable valve timing on the pumping losses. The variable valve timing is achieved by displacing the valve lifts (the use of cam phasers is simulated).

The results of the simulations can be seen in Figures 4 and 5. When the exhaust valve is closing with a delay after the TDC (no EGR) the pumping losses are higher.

This can be due to the fact that the exhaust valve is opening later (the valve lift curve is displaced) and less gases are evacuated due to the difference of pressure between the cylinder and the exhaust manifold and more gases are evacuated due to the piston movement from the BDC to the TDC. The main reason of the higher pumping loses when a lower EGR rate is used is the fact that more burned gases are passing through the exhaust valves during the exhaust stroke and more fresh air is passing through the intake valves during the intake stroke and the gasodynamic losses are higher. The admission of the air inside the cylinder starts earlier so the intake pressure drops more when the air is aspirated due to the piston movement from the TDC to the BDC.

Figure 5 shows the cylinder pressure during the gas exchange process depending on the crank angle position. The pressure

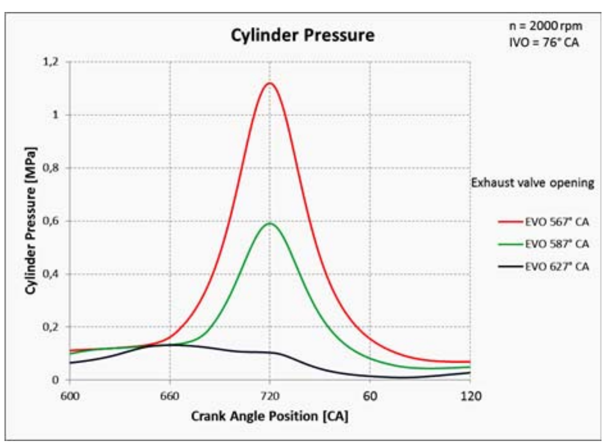

Fig. 5. *The cylinder pressure during the pumping*

during the gas exchange process is rising when higher EGR rates are used because of the higher advance used for the moment when the exhaust valve is closing.

3. The Data Analysis

The pumping work can be calculated using the cylinder pressure diagram during the gas exchange [4].

Figure 6 shows how the pumping work is calculated. It is calculated as the difference between the aria under the exhaust curve (ABC) and the aria under the intake curve (ADC) as shown in equation (1):

$$W_p = S_{ABCD} = S_{A'ABCC'} - S_{A'ADCC'}, \qquad (1)$$

where W_p represents the pumping loses.

The areas needed for the pumping work determination can be calculated using the trapezoidal rule [2], [7].

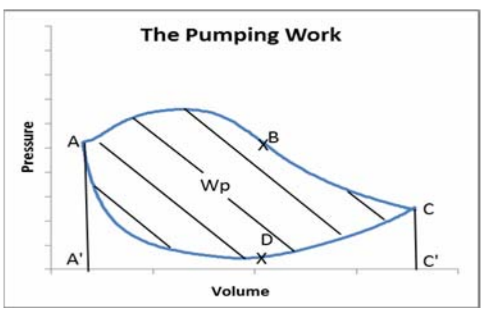

Fig. 6. *The pumping work*

Figure 7 shows how the aria under a curve is calculated using the trapezoidal rule.

The aria under the curve is spited in *n*-1 intervals. The area of the intervals is approximated as the area of a trapeze. If the area is split in more intervals the errors are smaller.

The area of one trapeze is calculated using equation (2):

$$S_i = \frac{p_{i-1} + p_i}{2}(V_i - V_{i-1}), \qquad (2)$$

where S_i is the area of the interval *i*, p_i is the pressure at point *i* and V_i is the volume where the pressure p_i is obtained.

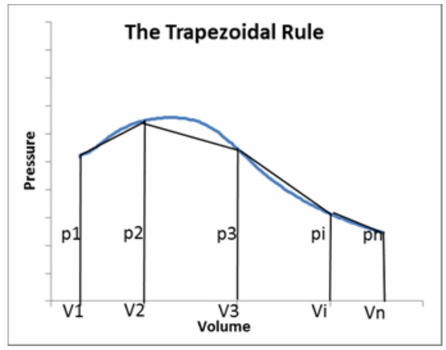

Fig. 7. *The trapezoidal rule*

The area under the curve can be calculated as the sum of all trapezes under the curve using equation (3):

$$S = \sum_{i=1}^{n} S_i . \qquad (3)$$

In Figure 8 the results obtained using the curves resulted from the simulations are presented. The results are showing that the pumping work which is lost during the gas exchange process depends on the internal EGR rate.

The pumping losses during the compression and the expansion of the burned gases consist in: the loses due to the blow-by (the gases from the cylinder are passing into the crankcase) and the losses due to the heat exchange from the burned gases to the cylinder wall.

The pumping losses due to the blow-by depends on the blow-by gap and on the crankcase pressure. The blow-by losses appears only when the cylinder pressure exceeds the crankcase pressure. The effective flow area can be approximated using the following equation [6]:

$$A_{eff} = D \cdot \pi \cdot \delta, \qquad (4)$$

where A_{eff} is the effective flow area, D is the cylinder bore and δ is the blow-by gap of the engine.

The losses due to the heat exchange between the gases and the cylinder walls depends on the temperatures (of the gases and of the walls) and on the time interval when the gases are in contact with the walls. During the compression of the burned gases the temperature is increasing, so the heat exchange is increasing. During the expansion of the burned gases the temperatures are decreasing, so the heat exchange is decreasing.

When the EGR rate is increased the exhaust valve is opened and closed earlier. The trapped gases are compressed during the end of the exhaust stroke which leads to a higher pressure at the beginning of the intake stroke. Due to the higher pressure the intake process is starting later and the pressure drop due to the piston movement from the TDC to the BDC is decreasing. The burned gases are filing a part of the cylinder. Less burned gases are evacuated from the cylinder and less air is aspirated. This leads to much lower gasodynamic losses due to the lower exhaust flow through the exhaust valves and due to the lower intake flow through the intake valves, so the pumping loses are decreasing with the increase of the EGR rate.

In Table 2 the pumping work obtained during the simulations can be observed.

The pumping work Table 2

Point	Exhaust valve opening	Pumping work
[–]	[°CA]	[J]
1	668	15
2	688	27
3	8	48

Figure 8 also shows the dependence of the pumping loses on the exhaust valve opening. The results show that the pumping work W_p has an almost linear increase when the exhaust valve opening is delayed. The pumping losses can be estimated using a first degree polynomial:

$$W_p - W_{p1} = m \cdot (EVO - EVO_1)\,, \qquad (5)$$

where m is the slope of the pumping loss trace and can be calculated using equation (6):

$$m = \frac{W_{p2} - W_{p1}}{EVO_2 - EVO_1}\,, \qquad (6)$$

where W_{p1} and W_{p2} are the pumping losses obtained during the simulation for the exhaust valve openings EVO_1 and EVO_2.

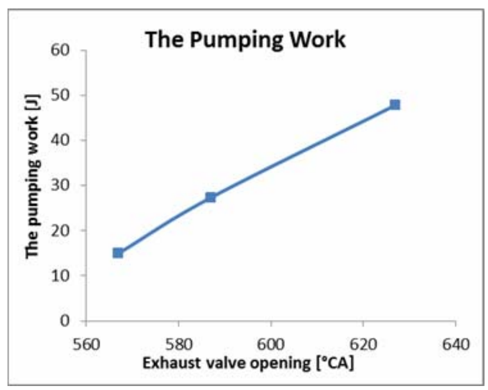

Fig. 8. *The pumping work*

4. Conclusions

The pumping work can be calculated using the cylinder pressure diagram during the gas exchange using the trapezoidal method. The trapezoidal rule can be used to approximate the areas used to calculate the mechanical work of the pumping losses. If the aria is split in a big number of intervals the error is decreasing.

The increase of the EGR rate leads to a lower pumping work due to the lower gasodynamic loses (less burned gases are passing through the exhaust valves at the end of the exhaust stroke and less air is passing through the intake valves during the intake stroke).

The compression and the expansion of the trapped burned gases at the end of the exhaust stroke and the beginning of the intake stroke has a small influence on the pumping losses, due to the heat transfer from the compressed gases (which have a higher temperature) to the cylinder walls (which have a lower temperature) and due to the blow-by.

There is also a leakage of a small part of the gases from the combustion chamber into the crankcase due to the differences of the pressure from the combustion chamber and the crankcase.

These losses during the compression and during the expansion of the burned gases are negligible. On the cylinder pressure diagram it can be seen that the traces during the compression of the trapped gases and during the expansion of the trapped gases are one over the other, so the pumping work on that area is null.

The compression and the expansion of the burned gases during the gas exchange process are similar to the spring effect.

Acknowledgements

This paper is supported by the Sectorial Operational Programme Human Resources Development (SOP HRD), financed from the European Social Fund and by the Romanian Government under the contract number POSDRU/6/1.5/S/6.

The simulations presented in the paper were done using the software AVL BOOST and supported by AVL Graz.

References

1. Baumgarten, C.: *Mixture Formation in Internal Combustion Engine*. Germany. Springer, 2006.
2. Burden, R.L., Faires J.D.: *Numerical Analysis* (*7th ed.*). United States. Brooks/Cole, 2000.
3. Cosgarea, R., Aleonte, M., Săcăreanu, S., Cofaru, C.: *State of the Art of Homogeneous Charge Compression Ignition Engines*. In: Bulletin of the *Transilvania* University of Braşov, (2008) Vol. 1 (50), Series I, p. 11-16.
4. Heywood, J.B.: *Internal Combustion Engine Fundamentals*. United States. McGraw-Hill, Inc., 1988.
5. Zhao, H., Peng, Z., Milliams, J., Ladommatos, N.: *Understanding the Effects of Recycled Burnt Gases on the Controlled Autoignition* (*CAI*) *Combustion in Four-Stroke Gasoline Engines*. SAE paper 2001-01-3607, 2001.
6. *** *AVL Boost Users Guide*. Available at: https://www.avl.com/boost1. Accessed: 02.04.2011.
7. *** *The Trapezoidal Rule*. Available at: http://pages.pacificcoast.net/~cazelais/187/trapezoidal_rule.pdf. Accessed: 27.03.2011.