EVALUATION OF TERMO-GAS DYNAMIC PARAMETERS FOR ENGINE BRAKE PROCESS

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Abstract: *In this work is presented the mathematical model to calculate the thermo-gas dynamic parameters that define the brake engine. This model can be used both for determining the state of the parameters of the working fluid in the cylinder and to specify the evolution of gas dynamic quantities from the intake and exhaust manifolds. Furthermore, the mathematical model can be used to optimize the braking ability. This is achieved by successive calculations of various parameters that define the duration of the braking process and by changing the maximum lift height of the valve that causes the event.*

Keywords: *engine brake, thermo-gas dynamic parameters*

1. Introduction

The engine brake system is an auxiliary braking system usually implemented on freight or passenger cars. This system allows the increase of safe movement speed for these items. Moreover, by using this system, the braking efficiency improves. When driving downhill, this system is used to maintain a constant speed on the way down.

Researchers as Moklegaard and others in Ref.[1] have developed an electronic control module that makes possible the coordination of the control actuator used to achieve the engine braking event with classical braking system. In this way, the wear of the classical braking system components is seriously reduced.

Another advantage of using this system is presented by Lee Cho-Yu and others in Ref.[2]. They propose to use brake system to develop a regenerative pneumatic system. In this way, they obtained a significant reduction in fuel consumption for passenger cars.

Both authors propose a mathematical model for calculating the gas dynamic parameters in the cylinder and manifolds. Because the injection is completely suppressed when using engine brake, there should be considered the calculating of gas state parameters. This is necessary because some parts of the combustion chamber, such as the injector, are not exposed to the cooling of the fuel flow and this can lead to obsolescence.

The mathematical model proposed for calculating the thermo-gas dynamic parameters is developed based on relations from Ref.[3]. Since the engine brake is basically a gas exchange process, it will be treated as an adiabatic transformation. Therefore, this process will take place by opening one valve for gases exchange. The valve that makes possible the braking process will be called brake valve.

The duration of the braking process is defined by the advance at opening, or delay at closing of brake valve. Another important parameter is the maximum lift height of the valve. This parameter is selected based on the minimum length of the combustion chamber, in this paper we used 4 mm for the lift height of the valve. The input angle for crankshaft rotation was defined at 355 degrees. The closing angle of the brake valve is set to value of 375 crank angle degrees.

To determine the values of thermo-gas dynamic parameters with this model, there were adopted as input data, the constructive and functional quantities of Lombardini 6LD400 single-cylinder taken from Ref.[4]. These parameters are summarized in Table 1.

Table 1:*The features of Lombardini 6LD 400engine*.

Bore (D)	8.6 cm
Stroke (S)	6.8 cm
Displacement (V_s)	395 cm^3
Engine speed (n_p)	3600 rpm
Power	5.9 kW/8CP
Compression ratio (ε)	18
Exhaust valve diameter	3.5 cm
Exhaust valve angle (γ_{ev})	45 deg
Connecting rod length (l_{rod})	11.2 cm

2. The evaluation of thermo-gas dynamic parameters

To develop the mathematical model we took into account the following:

- The outlet valve is which that produces the braking event;
- due to the cylinder pressure is superior than the pressure in the exhaust manifold, initially, the flow condition is considered critical;
- initially, the flow isin direct sense (cylinder – exhaust pipe);
- the suppression of gas injection into the cylinder and evolving exhaust manifold is air;
- cylinder is considered perfectly tight, with no loss of substance in the segment;
- the pressure conditions in the exhaust manifold are standard and the initial temperature is 900 K;
- the temperature of the air from cylinder (T_{cil}) and the kilomolnumber of air (ν_{cil}) at the beginning of engine brakingstroke are the same with those at the end of the compression.

The computation for the mathematical model is a fixed pitch iteration model.

To determine the cross-sectional area is necessary to define the brake valve-lift law. This law is implemented in the MathCADas a matrix via „*cspline*” tool. The „*cspline*” function gets an interpolation of lifting height valuesalong the braking process, taking into account the maximum value required for brake valve lift.The instant lift height of exhaust valve is taken from the definition matrix and brought to each step of iteration, using the Eq. (1):

$$h_{\sup ev} = \left(h_{\alpha-360-\beta_d} \cdot ampl_{ev}\right) - joc_{ev} \qquad (1)$$

where β_d – injection advance [deg], joc_{ev} – heat clearance of exhaust valve [mm], $ampl_{ev}$ – amplification ratio of the rocker, and $h_{\sup ev}$ – the opening height of exhaust valve [mm].

The computation of combustion chamber volume defined by the piston movement between two dead-points is based on Eq. (2) taken from Ref.[1].

$$V_{cil} \leftarrow V_c \cdot \left[1+\frac{1}{2}\cdot(\varepsilon-1)\cdot\left(2\cdot\frac{l_{rod}}{S}+1-\cos\left(\frac{\pi}{180}\cdot\alpha\right)-\sqrt{\left(2\cdot\frac{l_{rod}}{S}\right)^2-\sin\left(\frac{\pi}{180}\cdot\alpha\right)^2}\right)\right] \qquad (2)$$

where V_c represent the minimum combustion chamber volume [cm^3]and α the angle of rotation of crankshaft [deg].

The temperature in the cylinder is determined at each iteration step by adding the previous step value of temperature derivative. The calculation is shown in Eq. (3).

$$dT_{cil} \leftarrow \frac{1}{\nu_{cil}\cdot C_{vcil}}\cdot\begin{bmatrix}\left(i_{gec}-u_{cil}\right)\cdot d\nu_{gec}-\left(i_{cil}-u_{cil}\right)\cdot \\ \cdot d\nu_{cge}-dL_m-dQ_r\end{bmatrix}$$
$$T_{cil} \leftarrow T_{cil}+dT_{cil} \qquad (3)$$

where i_{cil} –cylinder gas enthalpy [kJ/kmol], u_{cil} – internal energy of the cylinder gas [kJ/kmol], i_{gec} – gas enthalpy passing from exhaust manifold to cylinder [kJ/kmol], L_m – work done by gas [kJ], Q_r – heat exchanged with the walls [kJ], ν_{cil} – amount of gas in cylinder [kmol], C_{vcil} – specific heat of gas[kJ/kmol].

The evolution of the temperature in the cylinder during the braking process stroke is shown in Fig. 1.

From the chart we can see that after 370 crank angle degrees, the temperature has a slight increase due to the change of flow direction.

The kilomol number of air in the cylinder is also determined by summing the number of kilomol derivative of the previous step value. The calculation formulas used are depicted in Eq. (4).

$$d\nu_{cil} \leftarrow d\nu_{gec}-d\nu_{cge}$$
$$\nu_{cil} \leftarrow \nu_{cil}+d\nu_{cil} \qquad (4)$$

where ν_{cge} – the amount of gas flowing from the cylinder to the exhaust, ν_{gec} – the amount of gas flowing into the cylinder exhaust pipe.

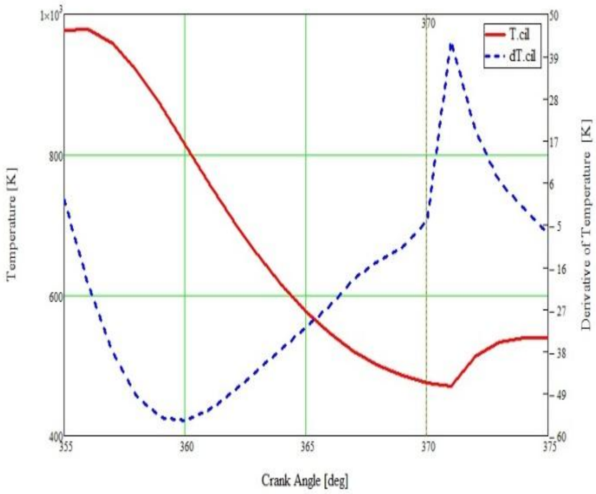

Figure1:*The evolution of the temperature in the cylinder and its derivative value.*

Depending on the direction and flow conditions, we have twoequationsfor each of $d\nu_{gec}$ and $d\nu_{cge}$ quantities, which were computed using the conditional function "if-otherwise" from MathCAD.The calculation formulas for the number of kilomol that passing the exhaust manifold during iteration, is shown in Eq. (5).

$$d\nu_{cge} \leftarrow \left| \begin{array}{l} 2\cdot 10^{-4}\cdot \mu_{se}\cdot A_{se}\cdot W_{cge}\cdot \dfrac{P_{cil}}{n_p\cdot T_{cil}}\cdot\left(\dfrac{P_{ge}}{P_{cil}}\right)^{\frac{1}{k_{ev}}} \ if \rightarrow P_{col.ev} < P_{critic} \\ 2\cdot 10^{-4}\cdot \mu_{se}\cdot A_{se}\cdot W_{cge}\cdot \dfrac{P_{cil}}{n_p\cdot T_{cil}}\cdot\left(\dfrac{2}{k_{ev}+1}\right)^{\frac{1}{k_{ev}-1}} \ if \rightarrow P_{col.ev} \le P_{cil} \\ \quad cilcol \leftarrow 0 \quad otherwise \end{array} \right. \tag{5}$$

where $cilcol$ represent a Boolean value which points the direction flow and P_{critic}–the critic pressure.

The first relation of Eq.(5) is written for subcritical conditions, the second one for the critical conditions, and if none of the conditions of the compliance function is not satisfied means that the flow takes place in reverse.

The evolution of the amount of material in the cylinder and its derivative are depicted in Fig. 2.

Pressure-crank angle and pressure-volume diagrames, are plotted by using the state gas equation, as shown in Fig.3 and Fig.4 respectively.

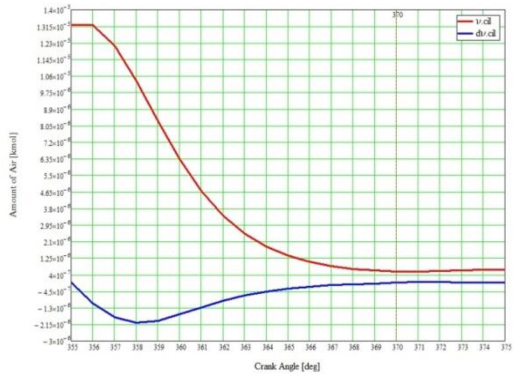

Figure 2:*The evolution of the amount of gas in the cylinder and its derivative*

This graphic also illustratesflow direction changes for the considered lift height and angle of brake valve opening and closing, respectively,corresponding to a crankshaft angle of 370 degrees.

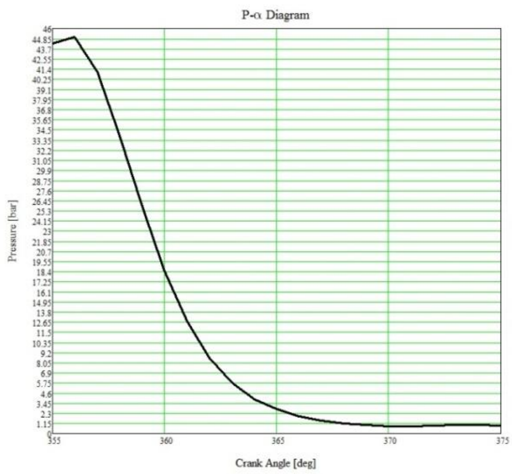

Figure3: *The pressure as function of crankshaft angle*

In P-α diagram it appears that the maximum pressure in the cylinder reaches around 45 bars. The character of the pressure curve is characterized by a high ratio of reduction of the pressure in the first half.

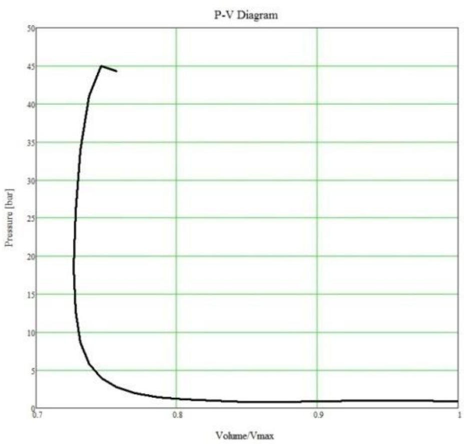

Figure 4:*Pressure-volume diagram.*

The P-V diagram from Fig. 4 had the same trend aspiston compressors diagrams.

The exhaust manifold pressure is calculated based on the flow conditions with the Eq. (6).

$$P_{col.ev}=\begin{cases}\dfrac{P_0+\sqrt{P_0^2+4\cdot\left(P_{cil}\cdot\dfrac{1,275\cdot\mu s_{ev}\cdot As_{ev}}{d_{colev}^{\ 2}}\cdot\left(\dfrac{2}{k_{ev}+1}\right)^{\frac{k_{ev}}{k_{ev}-1}}\cdot W_{cge}\right)^2\cdot\dfrac{0,601\cdot10^{-4}\cdot(1+\xi_{ev})\cdot M_{col.ev}}{T_{cil}\cdot\left(\dfrac{2}{k_{ev}+1}\right)}}}{2}\\[2ex] \dfrac{P_0}{\dfrac{\left(\left(\dfrac{1,275\cdot\mu s_{ev}\cdot As_{ev}}{d_{colev}^{\ 2}}\right)^2+\left(\left(\dfrac{2}{k_{ev}+1}\right)^{\frac{1}{k_{ev}-1}}\right)^2\right)\cdot\left(16640\cdot\dfrac{k_{ev}}{k_{ev}+1}\cdot\dfrac{T_{colev}}{M_{col.ev}}\right)}{1+\left(0,601\cdot10^{-4}\cdot(1+\xi_{ev})\cdot M_{col.ev}\right)\cdot\dfrac{1-\left(\left(\dfrac{1,275\mu s_{ev}\cdot As_{ev}}{d_{colev}}\right)^2+\left(\left(\dfrac{2}{k_{ev}+1}\right)^{\frac{1}{k_{ev}-1}}\right)^2\right)\cdot\dfrac{k_{ev}-1}{k_{ev}+1}}{Tcol_{ev}}}}\end{cases}\qquad(6)$$

where ξ_{ev} – resistance coefficient of the exhaust route, $d_{col\,ev}$ – exhaust manifold diameter [cm], As_{ev} –cross-sectional area of exhaust manifold [cm^2].

As shown in Eq. (6), the calculation of manifold pressure takes into account the resistance of gas dynamic evacuation route induced by the shape, size and the surface roughness of contact through ξ_{ev} coefficient.

In Fig. 5 is presented the exhausted manifold pressure as function of crank angle. We can observe that the maximum exhaust manifold pressure is reached after only 3^{o}RAC, with the approximate value of 4.6 bar.

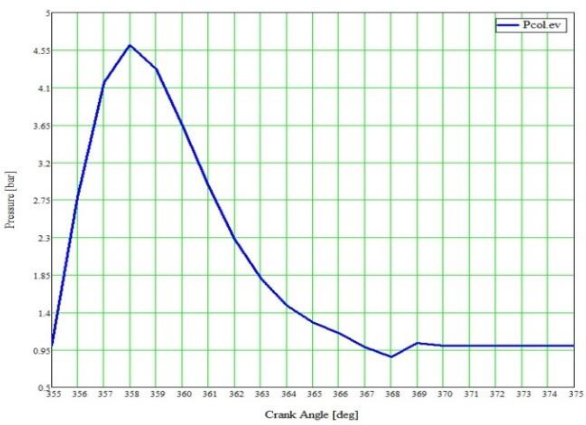

Figure 5: *Variation of exhaust manifold pressure plotted against crankshaft angle.*

The gas flow rates through the gate valve and through the exhaust manifold are determined depending on flow conditions. The equations used to compute these quantities are shown in Eq.(7), Eq.(8) andEq.(9).

$$W_{ge}=\begin{cases}\dfrac{\dfrac{1,275\cdot\mu s_{ev}\cdot As_{ev}}{dcol_{ev}^{\ 2}}\cdot P_{cil}\cdot\left(\dfrac{2}{k_{ev}+1}\right)^{\frac{k_{ev}}{k_{ev}-1}}\cdot W_{cge}}{Pcol_{ev}} & critic\quad condition\\[2ex] \dfrac{1,275\cdot\mu s_{ev}\cdot As_{ev}}{dcol_{ev}^{\ 2}}\cdot W_{cge} & subcritic\quad condition\end{cases}\qquad(7)$$

$$W_{cge}=\begin{cases}\sqrt{16640\cdot\dfrac{k_{ev}}{k_{ev}+1}\cdot\dfrac{T_{cil}}{M_{cil}}} & critic\quad condition\\[2ex] \sqrt{16640\cdot\dfrac{T_{cil}}{\left(\dfrac{k_{ev}-1}{k_{ev}}\right)\cdot M_{cil}}\cdot\left[1-\left(\dfrac{Pcol_{ev}}{P_{cil}}\right)^{\frac{k_{ev}-1}{k_{ev}}}\right]} & subcritic\quad condition\end{cases}\qquad(8)$$

$$W_{gec}=\begin{cases}\sqrt{16640\cdot\dfrac{k_{ev}}{k_{ev}+1}\cdot\dfrac{T_{col.ev}}{M_{col.ev}}} & critic\quad condition\\[2ex] \sqrt{16640\cdot\dfrac{T_{col.ev}}{\left(\dfrac{k_{ev}-1}{k_{ev}}\right)\cdot M_{col.ev}}\cdot\left[1-\left(\dfrac{P_{cil}}{P_{colev}}\right)^{\frac{k_{ev}-1}{k_{ev}}}\right]} & subcritic\quad condition\end{cases}\qquad(9)$$

where $M_{col.ev}$ – molar mass of gas in exhaust manifold [g/kmol], μs_{ev} – flow coefficient, M_{cil} – molar mass of gas in cylinder [g/kmol], k_{ev} – adiabatic exponent, $P_{col\,ev}$ – pressure in exhaust manifold [bar], W_{ge} – gas velocity in the exhaust manifold [m/s], W_{cge} – gas velocity flowing from cylinder to exhaust manifold [m/s]

The graph of flow velocities is shown in Fig. 6.From the graph it is observed that up to 370^{o}RAC there is no flow from the exhaust manifold to cylinder.

One of the most important parameters of this process is the adiabatic exponent. The MathCAD calculation of this parameter is shown in Eq. (10).

$$k_{ev}=\frac{8.314}{cv_{psev}}\qquad(10)$$

The value of this parameter is calculated by imposing an arbitrary initial value. The condition of exiting the”*while*„ loop is that of reaching an imposed error. The form of this condition is shown in Eq.(11).

$$\frac{|Tps_{ev-1}-Tps_{ev}|}{Tps_{ev}}>0,01\qquad(11)$$

where Tps_{ev-1} - gate valve exhaust temperature at the previous iteration step [K] and Tps_{ev} is the temperature in exhaust valve gate [K].

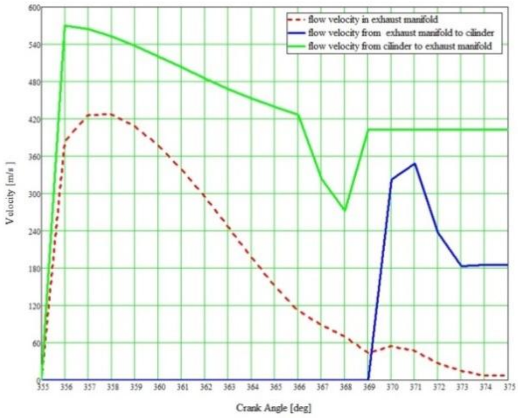

Figure6: *Evolution of air velocity in the exhaust manifold respectively through the valve brake gate.*

When this condition is not satisfied, an exit statement is used to complete the execution of while loop. At this point the last value of adiabatic exponent is stored. Based on the final values obtained at each iteration step, we drawn the chart from Fig. 7.

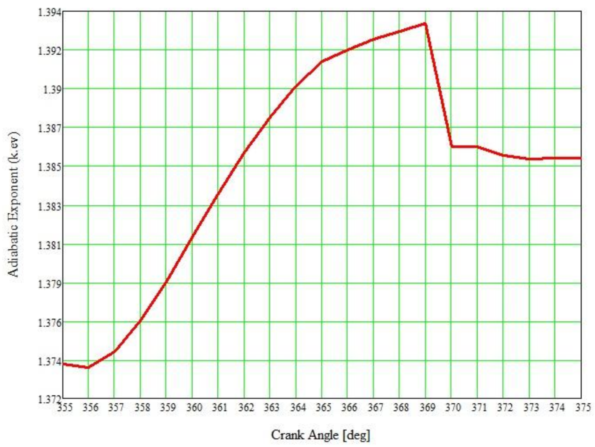

Figure7: *The adiabatic exponent.*

In Fig.7 is observed an increase on the valueof the adiabatic exponent until the change in flow direction occurs, then its value tends to stabilize.

3. Conclusions

The mathematic model developed in MathCAD is easy to use and to modify for different study cases, because the equations which compute the parameters are legible.

The proposed mathematical model can be used to investigate both thermo-gas dynamic parameters of the cylinder and the exhaust manifold.

For the measurements that define the duration of the braking process and the maximum lift height of the brake valve, the flow direction is changing at 370oRAC from pipe to cylinder.

The evolution of the pressure in the cylinder is characteristic to reciprocating compressors. This presents in the first half period a strongly negative gradient.

The temperature in the cylinder shows an increase after changing the flow direction as a result of the high temperature from the exhaust manifold.

The flow rate from the exhaust manifold tends to zero at the end of the braking process.

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