EFFECT OF EXHAUST VALVE TIMING AND BOOST PRESSURE ON ENGINE BRAKE PERFORMANCE

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Abstract: *In recent years, due to the research in the field of road vehicles, there have been developed more powerful engines, in particular by reducing the energy loss in the friction forces, respectively the forces of air resistance. Due to these improvements, the braking ability of the vehicle is reduced, requiring the implementation of the auxiliary braking system, particularly for heavy vehicles. This paper discusses the diesel engine braking through the concept of compression release. Our study takes into the consideration the influence of air supply pressure of the cylinder, respectively the distribution influence of braking phase on braking capacity.*

Keywords: *release of compression, engine brake, heavy vehicles*

1. Introduction

The auxiliary braking systems, which include the compression release brake system, have been developed to improve the ability of deceleration, reduction of fuel consumption, or speed up the rate of travel.

Until now, many companies have developed various auxiliary braking systems as hydraulic or electromagnetic retarders, butterfly valve for exhaust manifold or engine brake systems. After the analysis presented by Maxwell in Ref. [1], the engine brake compression release system enjoys the highest popularity providing a braking capacity of approximately 85% of rated power developed by the engine and a good time response.

Although the compression release engine brake was first presented by Cummins in Ref. [2], researchers like Moklegaard in Ref. [3] Barbieri in Ref. [4] and Lee Ref. [5] uses this concept in studies about improving of traffic safety, increase the braking capacity and fuel consumption reduction.

The braking ability of the engine must provide an improvement with the supply pressure increases, because of the effort to compress a greater amount of working fluid. The gas discharge valve used in this case will be opened at the end of the compression stroke, near the dead-center. This opening takes place to remove the effect of the gas spring. In this way, the kinetic energy taken from the axle will be not returned in the expansion stroke, but released as heat.

The closing or opening angle of the valve which produces braking event is not rigorously established, thus we analyze these two parameters that influence the torque on the crankshaft.

We assume that the transition between combustion mode and the motor brake are done by opening the exhaust valve and with complete suppression of the injection. In these conditions, the working fluid will be represented only by the air and the motor cycle processes will be composed in this order: intake, compression, engine brake, expansion, exhaust.

2. The mathematical model of gas exchange process

Based on the equations from Ref. [6], we developed a MathCAD model to compute the thermo-gas dynamic parameters of motor cycle for two operating modes mentioned (combustion mode and the motor brake).

The numerical model is an iterative one, with fixed pitch. The main equations that describe the braking process are the following.

In order to calculate the pressure in the cylinder, P_{cil} and the volume, V_{cil} there has been used the Eq. (1) or Eq. (2).

$$V_{cil} = V_c \cdot \left[1 + \frac{1}{2}\cdot(\varepsilon - 1)\cdot\left(\lambda_r + 1 - \cos\left(\frac{\pi}{180}\cdot\alpha\right) - \sqrt{\lambda_r^2 - \sin\left(\frac{\pi}{180}\cdot\alpha\right)^2}\right)\right] \quad (1)$$

$$P_{cil} = R\frac{\nu_{cil}T_{cil}}{V_{cil}}10^4 \quad (2)$$

where ν_{cil} - is the number of kilomole in the cylinder[kmol], R - gas constant, ε - compression ratio, λ_r - the ratio between the radius of the crank and connecting rod length, V_c - the minimum volume of the combustion chamber[cm^3], α - the angle of crankshaft rotation[deg].

The temperature of gas in the cylinder is determined by adding the value of the derivative at above calculated value. The computation of temperature derivative as a function of α angle is done using Eq. (3).

$$\frac{dT_{cil}}{d\alpha} = \frac{1}{\nu_{cil}\cdot C_{vcil}}\cdot\left[\begin{array}{l}\left(i_{gac} - u_{cil}\right)\cdot\dfrac{d\nu_{gac}}{d\alpha} + \left(i_{gec} - u_{cil}\right)\cdot\dfrac{d\nu_{gec}}{d\alpha} - \\ \left(i_{cil} - u_{cil}\right)\cdot\left(\dfrac{d\nu_{cge}}{d\alpha} + \dfrac{d\nu_{cga}}{d\alpha}\right) - \dfrac{dL_m}{d\alpha} - \dfrac{dQ_r}{d\alpha}\end{array}\right] \quad (3)$$

where $i_{gec}, i_{gac}, i_{cil}$ - represents the enthalpies of the exhaust manifold gas[kJ/kmol], intake manifold gas, gas cylinder, u_{cil} - the internal energy of the gas cylinder[kJ/kmol], $\nu_{gac}, \nu_{gec}, \nu_{cga}, \nu_{cge}$ - the quantity of substances that pass the intake or exhaust manifolds cylinder or vice versa[kmol], L_m - the work done [kJ], Q_r - heat exchanged with the walls [kJ], C_{vcil} - the instantaneous specific heat of the gas cylinder[kJ/kmol].

First of all, the MathCAD code will determine all these parameters, after that will compute the temperature in the cylinderat the end of each iteration.

The equations for calculating the gas quantities depend on the direction and flow conditions. Since the opening of the brake valve of engine brake process is maximum around the cylinder, and the process is relatively short, we consider that is important to mention only the equation for calculating the gas derivative exchanged between the cylinder and the exhaust manifold Eq. (4).

$$d\nu_{cge} \leftarrow \left| \begin{array}{l} 2\cdot10^{-4}\cdot\mu_{se}\cdot A_{se}\cdot W_{cge}\cdot\dfrac{P_{cil}}{n\cdot T_{cil}}\cdot\left(\dfrac{P_{col.ev}}{P_{cil}}\right)^{\frac{1}{k_{ev}}} \; if \rightarrow P_{col.ev} > P_{critic} \\ 2\cdot10^{-4}\cdot\mu_{se}\cdot A_{se}\cdot W_{cge}\cdot\dfrac{P_{cil}}{n\cdot T_{cil}}\cdot\left(\dfrac{2}{k_{ev}+1}\right)^{\frac{1}{k-1}} \; if \rightarrow P_{col.ev} \le P_{critic} \end{array}\right. \quad (4)$$

in which: μ_{se} - flow rate of artificial exhaust, A_{se} - cross-sectional area provided by the opening of the exhaust valve[cm^2], W_{cge} - the flow rate of gas from the cylinder into the exhaust manifold[m/s], n - the speed of the engine [rpm], $k_{\cdot ev}$ - the exhaust adiabatic exponent, $P_{col.ev}$ - the pressure in the manifold exhaust [bar], P_{critic} -critical pressure [bar].

There are two important gas dynamic parameters to characterize the braking process. These parameters are the speed of gas flow through the intake port and the flow rates of the gases from the cylinder into the exhaust manifold or vice versa. The Eq. (5) represents the MathCAD implementation to compute these parameters for the case of direct flow.

$$W_{cge} \leftarrow \left| \begin{array}{l} \sqrt{16640\cdot\dfrac{k_{ev}}{k_{ev}+1}\cdot\dfrac{T_{cil}}{M_{cil}}} \quad if \rightarrow P_{col.ev} < P_{critic} \\ \sqrt{16640\cdot\dfrac{T_{cil}}{\left(\dfrac{k_{ev}-1}{k_{ev}}\right)\cdot M_{cil}\cdot\left[1-\left(\dfrac{Pcol_{ev}}{P_{cil}}\right)^{\frac{k_{ev}-1}{k_{ev}}}\right]}} \quad if \rightarrow P_{col.ev} \le P_{cil} \\ cilcol \leftarrow 0 \quad otherwise \end{array}\right. \quad (5)$$

Defining the logical variable $cilcol$ we can achieve the switch to flow backwards (gallery-cylinder). The M_{cil} variable from Eq.(5) is the molar mass[g/mol] of the working fluid (air).

To investigate the brake capacity, we will determine the value of instantaneous torque using Eq. (6), also taken from Ref. [6].

$$TQ = \left(Fgaz + Ft_j\right)\cdot\frac{\sin(\alpha+\beta)}{\cos(\beta)}\cdot\frac{S}{2} \quad (6)$$

in which TQ - is the instantaneous torque[Nm], F_{gaz}, Ft_j - gas pressure forces and inertia forces[N], β - the angular displacement of the rod[rad].

The gas pressure forces are calculated taking into account that the gas pressure in the crankcase is equal to the normal atmospheric pressure. With this assumption, the formula for calculating the gas pressure force is Eq. (7).

$$F_{gaz} = \left(P_{cil} - P_0\right) \cdot \frac{\pi \cdot D^2}{4} \cdot 10^{-6} \qquad (7)$$

in which: D - the cylinder bore [cm], P_0 - atmospheric pressure [bar].

To determine the inertia force for the components with translation movement we used the Table 10.6 from Ref. [6], which provides the range of values of rod crank masses according to the cylinder bore. The calculation of the forces is given in Eq. (8).

$$Ft_j = -\frac{mt_j \cdot S \cdot \pi^2 \cdot n^2}{18} \cdot 10^{-5} \cdot \left[\cos(\alpha) + \lambda_b \cdot \cos(2\alpha)\right] \qquad (8)$$

In Eq. (8), S represents the piston stroke [cm] and mt_j is the mass sum of piston and connecting rod [kg].

Based on the cylinder pressure matrix corresponding to the compression, braking and expansion processes, we obtained the pressure and inertia force values.

In this study, we didn't consider the influence of the motor mechanism friction forces or the auxiliary forces related to the engine subsystems.

3. The influence of braking phase angles on torque

To determine the instantaneous torque, respectively the braking process quantities, we considered as input for MathCAD code the constructive and functional characteristics of Lombardini6LD400 engine.

In Fig.1 we plotted the instantaneous torque for normal operation.

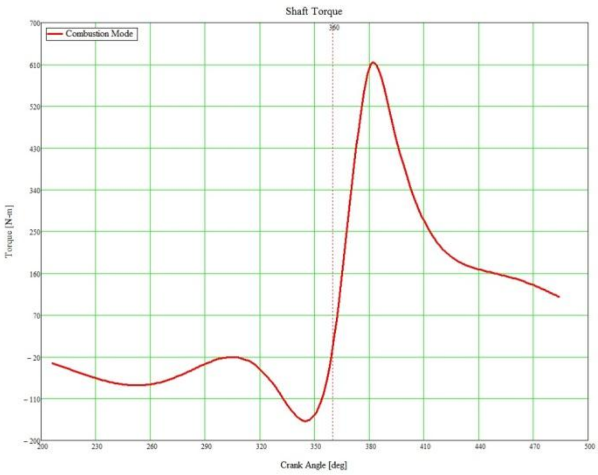

Figure1: *The instantaneous torque for combustion mode.*

We can observe from that over the cycle, the average shaft torque is positive. The maximum, minimum or average values are listed in Table 1.

We intend to determine the instantaneous torque for different values of the opening angle of the brake valve. Because the angle of opening of the brake valve must take place closer to PMI, wesetas initial valuefor opening angle 355 crank angle degrees. The opening angle values of the exhaust valve in the braking process are selected symmetrically to this value.

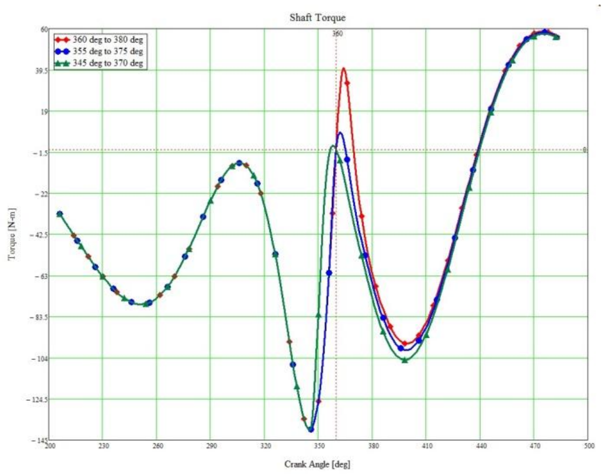

Figure2: *The evolution of torque for different angles to open or close the exhaust valve.*

Fig.2 illustrates the variation of torque of crankshaft angle when operating in braking mode.As can be seen in Fig.2, the braking process duration was kept constant, changing only the opening and closing angle value of the exhaust valve.

Although the chart would show that the angles used for the third situation are the most suitable, the data derived from simulations show that the most negative average time is produced for the angles values of the second case. The data relating to torque values are summarized in Table 1.

Table 1

	360 to 380 deg	355 to 375 deg	345 to 370 deg	Combustion mode
Mean (TQ)	-39.8	-41.8	-41.6	78.97
Max (TQ)	40.38	8.51	1.89	614.95
Min (TQ)	-139.9	-139.9	-139.9	-158.75

It should be noted that the maximum values in Table 1, for the engine braking case, are the ones close to the point dead center (PMI).

Since the second interval of opening, respectively closing angles values of the valve are the most advantageous for braking mode, we keep constant these values and we modified the duration of the trial.

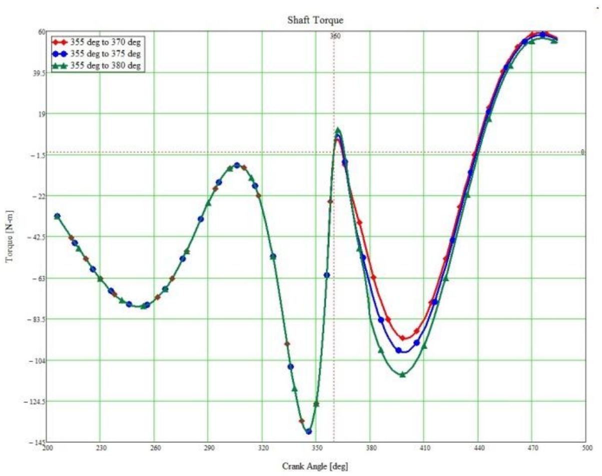

Figure3: *Torque evolution for constant opening angle of exhaust valve.*

The variation of the torque obtained by changing the duration of the braking process, while maintaining the opening value of the exhaust valve at 355 crank angle degrees is represented in Fig.3.

From Fig.3 it is concluded that the braking torque value is even greater as the closing angle of the exhaust valve increases.

Likewise, we keep constant the value of closing angle of the exhaust valve, changing the process time. The values obtained for this case are drawn in Fig.4.

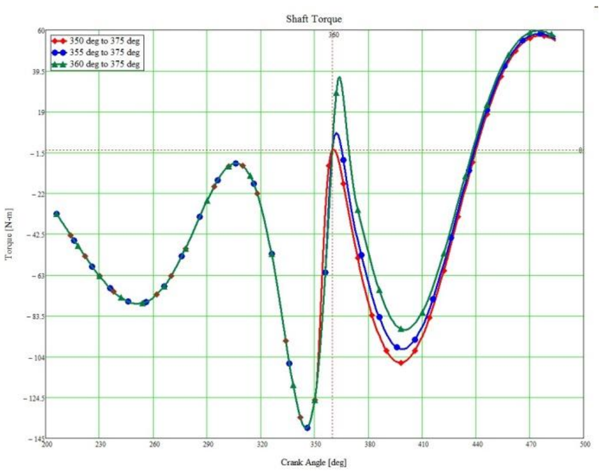

Figure4: *Torque evolution for constant closing angle of exhaust valve.*

In this case, it can be observed that the average value of the torque is more negative as the exhaust valve opening angle decreases.

The data obtained for the torque values corresponding to the Fig.3 and Fig.4, are summarized in Table 2.

Table 2

	Mean (TQ)	Max (TQ)	Min (TQ)
355 deg to 370 deg	-40.095	6.213	-93.42
355 deg to 375 deg	-41.869	8.511	-100.207
355 deg to 380 deg	-44.241	10.661	-111.262
350 deg to 375 deg	-43.417	0.099	-107.017
360 deg to 375 deg	-38.424	36.322	-90.559

From these simulation cases we conclude that the optimum value of the exhaust valve opening angle to produce the braking event is around 355 crank angle degrees. We can also say that a longer period of the braking process leads to more negative values of torque. Obtaining average values more negative for torque means consuming an increasing amounts of kinetic energy from the crankshaft.

When using this system for downhill driving, the energy consumed is actually the kinetic energy of the vehicle movement. In these conditions, by using the engine braking system it can be controlled the rate of travel or maintaining a constant speed when downhill driving.

4. The influence of boost pressure on torque

When operating in braking mode, due to an increase in pressure in the cylinder, the work done to compress the gas is increased. This leads to increase the braking capacity, which is reflected in the amount of torque consumed by the crankshaft.

The boost pressure values, for which the simulations were carried out on the way of change of torque, are chosen arbitrarily to a value of 1.2, and 1.3 bars. The characteristics obtained are compared with those of naturally aspirated engine and are presented in Fig.5.

The graph in Fig.5 was drawn to the opening angle of the exhaust valve,

$\alpha_{DSE} = 355^o$, respectively the closing angle, $\alpha_{ISE} = 375^o$.

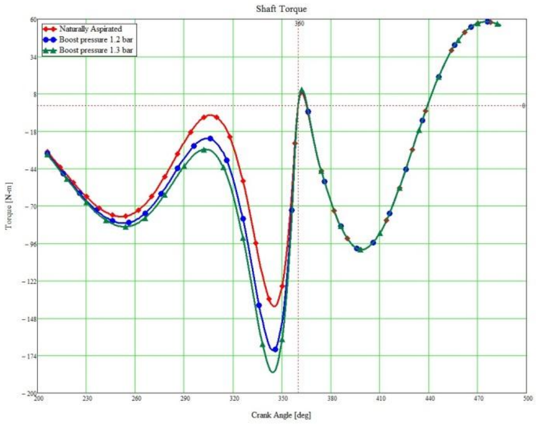

Figure5:*The evolution of torque with the pressure supply.*

We can see that the increasing of supply pressure leads, as we have assumed, to an increased braking performance. The torque values obtained for this case are shown in Table 3.

Table 3

	Mean (TQ)	Max (TQ)	Min (TQ)
naturally aspirated	-41.869	58.223	-139.914
boost pressure 1.2 bar	-48.634	58.192	-170.58
boost pressure 1.3 bar	-51.97	58.176	-185.48

From Table 3 we see that increasing the supply pressure of 0.2 bars, the braking moment increase by approximately 1.15%. It also shows that an extra increase of only 0.1 bars leads to an increase of approximately 1.25% over aspirated engine.

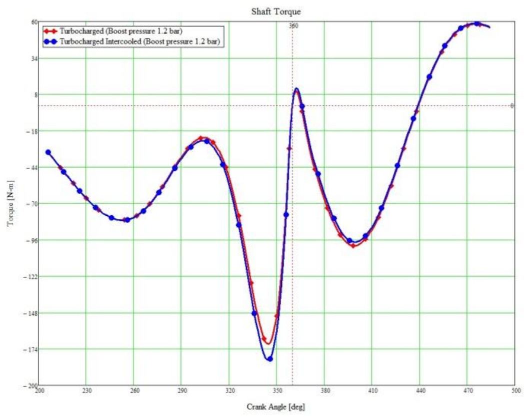

Figure 6:*The evolution of torque with the air fresh cooling*

Another parameter that significantly influences the engine performance is fresh gas temperature. As a result of the compression in the turbocharger, the gases are heated. The cooling of these gases is achieved by guiding them through a radiator called intertwined. Considering a cooling intertwined ratio of 1.2, in Fig.6 we drew the variation graph of torque for 1.2 bar boost pressure.

In Table 4 are presented the boundary values and the mean torque.According to these, the engine torque shows an increase of 1% by using the gas inlet cooler vs. the non-cooled turbocharger.

Table 4

	Mean (TQ)	Max (TQ)	Min (TQ)
Boost pressure 1.2 bar	-48.634	58.192	-170.58
Intercooled Boost pressure 1.2 bar	-49.346	58.624	-182.038

4. Conclusions

The analysis of the opening or closing angle of the exhaust valve, and the influence of the supply pressure on torque, leading to the following conclusions:

- An early opening of the exhaust valve leads to a decrease in energy consumption of the shaft as a result of performing a small mechanical work in the compression stroke;
- The opening of the exhaust valve after PMI, allows a return of significant amount of energy to shaft through the gas spring effect;
- For the engine studied, the optimum exhaust valve opening should be made with an advance of about 5 degrees before PMI;
- The exhaust valve can be kept open until the equalization of pressure in the cylinder to the back pressure in the exhaust manifold;
- Increasing the supply pressure with 0.2 bar leads to an increase in the average torque to approximately 1.15%;
- Using a fresh gas cooling system supplements the braking performance by 1%.

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