Preliminary studies on the braking efficiency of an engine working in combined Jake brake and exhaust brake regime

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Abstract. The improvements brought lately to heavy duty vehicles led to a decrease of their native retarding capacity. This difficulty is commonly compensated by the auxiliary braking systems such as the Jake brake or exhaust brake. The purpose of this study is to evaluate the behaviour of an auxiliary brake system consisting from a combination of an exhaust brake and a Jake brake. For this purpose, a simulation model was developed to evaluate the retarding power of a single-cylinder engine that works at the same time in Jake brake and exhaust brake mode respectively. The present article deals with five study cases, at constant engine speed, for a better comparison of the braking torques obtained for each operation mode of the I.C. engine. Simulations performed on the studied engine in Jake brake mode, provide an increase in braking efficiency, compared to the brake efficiency when it is used in exhaust brake mode. Obtained results show a significant increase in braking capacity for the case of a single cylinder engine, when the two auxiliary braking systems are simultaneously used. The proposed model can be used to optimize braking performance to the any reciprocating engines that presents an engine brake or an exhaust brake system.

1. Introduction
As a result of the continuous improvements brought by the automotive industry such as lowering of aerodynamic resistance of the vehicle body, reduction of energy losses at the engine level and those at the tire – road interface, led to a decrease in the native braking capacity. This shortcoming is more pronounced in the case of heavy vehicles, where, besides the classic braking system, it is necessary to use auxiliary braking systems. The main auxiliary braking systems implemented on such vehicles are at the engine level, exhaust manifold, or propeller shaft.

The present paper aims to present a study of the Jake brake and the exhaust shut-off, auxiliary braking systems. The action of the Jake brake system is more efficient than the exhaust brake according to [1], but has the drawback of the high noises resulting from the release of compression, for which reason the system was banned in some countries, [2]. The Jake brake involves suppressing the injection and releasing of the compression from the cylinder in the immediate vicinity of the injection time. This braking system leads to consuming a significant amount of the vehicle’s kinetic energy by removing the spring effect of the gas in the power stroke. The exhaust shutter action reduces the gas flow cross section in the exhaust manifold. This leads to the transformation of the exhaust stroke into a compression stroke with a lower compression ratio and to the occurrence of an increased exhaust gas pressure in the outlet of the exhaust valve and the cylinder. In this way, a resistant torque is obtained at
the motor shaft. This effect may be used to obtain a faster deceleration when combining the two abovementioned auxiliary braking systems.

Research on Jake-type braking is carried out by the authors in previous papers [3, 4, 5], but they do not address the influence of the exhaust brake. The purpose of this study is to combine the effects of the two types of braking in order to obtain new information on how to optimize braking performance in heavy vehicles.

2. Model setup
In order to carry out the proposed study, simulations were made on the model of a single-cylinder diesel engine. The single-cylinder model was previously developed by the authors using a computing program under the Mathcad environment. The calculation program developed and used in this paper offers a good approximation, being validated in [6]. It is composed of a series of sub-programs that evaluate the thermo-gas-dynamic parameters characteristic to the processes that succeed each other during the motor cycle. The model proposed for this study does not include the assessment of the combustion process, because the fuel injection is suppressed during the engine brake mode. Thus, the combustion process is replaced by a gas exchange process, which describes the operation of the engine in braking mode.

The block diagram illustrated in figure 1 describes the logical flow of the computing program developed in Mathcad.

![Figure 1. Logical flow diagram for the computing model.](image)

As shown by figure 1, the computation code starts at the beginning of compression stage, from the moment of closing the intake valve, at the \( \alpha_{ISA} \) value of the rotation angle of the crankshaft, and ends in the same point.
Simulations conducted in the present work were based on the constructive and functional parameters of the Lombardini 6LD400 single-cylinder diesel engine. Table 1 presents these parameters according to the manufacturer's catalogue.

Table 1. Construction dimensions for the Lombardini 6LD400 single cylinder diesel engine.

| Parameter                              | Value |
|----------------------------------------|-------|
| Cylinder diameter                      | 86 mm |
| Stroke                                 | 68 mm |
| Engine displacement                    | 395 cm³ |
| Maximum power speed                    | 3600 rpm |
| Engine power                           | 8 hp  |
| Compression ratio                      | 18    |
| Exhaust Valve Seat Diameter            | 31 mm |
| Intake Valve Seat Diameter             | 35 mm |
| Angle of valve seat machining          | 45 degree |
| Rod length                             | 112 mm |

The calculation program specifies the parameters of the environment or the initial temperatures of the cylinder components by means of adopted values. Thus, the values of absolute pressure, temperature and chemical composition of the environment are defined at the entrance of the intake manifold. Several iterative and conditioning loops are used to describe the flow through the manifold in order to determine the flow regime, the amount of substance and the thermo-gas-dynamic parameters. In the case of flow through the exhaust manifold, the calculation program takes into account that if during the evacuation process a reverse flow occurs (manifold to cylinder), then the temperature of that gas lot will have a value determined in the iterative process from the value immediately preceding the change of flow direction.

To perform this study, it was necessary to implement in the calculation code some parameters that characterize the operation of the shutter flap. The evacuation gallery is characterized by a series of thermodynamic and constructive parameters such as $T_{colev}$, $D_{colev}$, $L_{colev}$, etc. In order to implement the evacuation shutter in the exhaust manifold, two distinct values of the $D_{colev}$ parameter were imposed. The values of this parameter are specified in Table 2 and determine the cross-section of the exhaust manifold for the case when the shutter is completely closed or open, respectively.

In the computational program, a series of differential equations are used to construct fixed pitch iterations. The main equations used are: [7]: the differential temperature equation (1), the gas state equation (2), the quantities describing the flow of gases through the pipes equation (3), respectively the gas velocity equations (4) and (5).

\[
\frac{dT_{cil}}{d\alpha} = \frac{-dL_{cil}}{d\alpha} - \frac{dQ_{cil}}{d\alpha} \\
\frac{d\alpha}{d\alpha} = \frac{v_{cil}}{C_{cil}}
\]

\[
P_{cil} = R \frac{V_{cil} T_{cil}}{V_{cil}} 10^4
\]

where:
- $\alpha$ - integration step
- $P_0$ - initial manifold pressure
- $P_{cil}$ - cylinder pressure
- $P_{colev}$ - pressure along exhaust route
- $dL_{cil}$ - performing speed of mechanical work
- $dQ_{cil}$ - heat exchange speed
\( v_{cil} \) - kilo moles of fluid inside the cylinder  
\( C_{cil} \) - specific heat for gases inside the cylinder  
\( T_{cil} \) - gas temperature inside the cylinder  
\( R \) - constant of gases  
\( V_{cil} \) - momentary cylinder volume  
\( W_{gr} \) - gas velocity inside conduits  
\( M_{ev} \) - molecular mass of the exhaust gases  
\( T_{col_{ev}} \) - gas temperatures inside conduits  
\( k_{ev} \) - exhaust adiabatic coefficients  
\( m_{cil} \) - molecular mass of the working fluid  
\( A_{sv_{ev}} \) - area of the passing cross through the exhaust valve gate  
\( W_{cge} \) - gas velocity through the cross-section provided by the exhaust valve  
\( \xi_{ev} \) - mean resistance coefficients for the intake and exhaust routes respectively

The computation program yields thermo-gas-dynamic parameter values at the end of each process, in matrix form. These data are automatically used as input data for the next process. By using Mathcad's Stack and Sub-matrix functions at the end of the program, a matrix is generated that contains the values determined for the parameters of interest after each step.

Increased importance should be given to the exhaust valve actuator when the engine is running in braking mode. The developed calculation program allows the introduction of the valve motion law by specifying the opening or closing angles of the valves and imposing lifting heights by using a cubic interpolation function. Figure 2 shows the intake and evacuation valve lifting laws adopted for this study based on results previously obtained in [8]. From figure 2, it can be noticed that the exhaust valve makes an additional opening at the end of the compression stroke, allowing the gases to escape from the cylinder. This is the classic mode of engine operation during Jake brake functioning. The advance of the exhaust valve, or the closing delay when compared to the TDCF, needed to achieve the brake event was considered at 12 degrees of crankshaft rotation in this study.

![Figure 2. Valve lift evolution.](image)

By customizing the input data, such as the exhaust manifold cross-section, or the valve lift laws, several particular engine operating cases can be investigated. This paper addresses a number of three such study cases. As shown in table 2, all simulations were performed at the same speed, while the lifting law for the exhaust valve and the position of the exhaust valve flap determine the three study
cases. The study cases highlight the evolution of the braking torque developed at the engine shaft for the operation of the engine with suppressed injection: fully closed exhaust shutter mode, Jake brake mode and the combination of the two auxiliary systems. For the last case study, the shutter flap is completely closed.

### Table 2. Study cases definitions.

| Parameter            | Case I Jake-Brake | Case II Exhaust-Brake | Case III Jake-Brake & Exhaust-Brake |
|----------------------|-------------------|-----------------------|-------------------------------------|
| Engine speed [rpm]   | 3600              | 3600                  | 3600                                |
| Exhaust pipe diameter [mm] | 50              | 5                     | 5                                   |

3. Results

The evolution of cylinder pressure is directly influenced by the exhaust valve lifting law and the position of the shutter valve. The more the exhaust gas flow rate decreases, an increase in the pressure in the exhaust valve gate will result in a reverse flow. Figure 3 shows the cylinder pressure evolution for the three case studies obtained from Mathcad simulations.

![Figure 3. Evolution of the in cylinder pressure.](image)

It can be noticed that in the case of Jake brake regime, the pressure evolution shows a sudden decrease as soon as the exhaust valve is opened after the TDCF. Also the cylinder pressure value reaches a maximum in this mode of operation at 46.083 bars.

In the case of the use of the exhaust shut-off braking regime, the cylinder pressure has a normal evolution during the power stroke, but due to the small area of the exhaust manifold during the forced evacuation process, the cylinder pressure records up to 5.2 bars. It is highlighted that the evacuation process is transformed into a process of compressing the gases into the exhaust pipe. Due to this increased pressure, the shaft torque will register a significant drop, which will ultimately lead to a faster deceleration of the vehicle.

In the case of simultaneous operation of the Jake brake and the shut-off valve at maximum capacity, we notice a decrease in the maximum pressure reached within the cylinder. The pressure peak is in this case lowered by 0.1 bars by comparison compared to the action of the exhaust shutter.
The brake system. This is beneficial as it leads to a reduced mechanical stress of the engine mechanism. It is also noted that both the pressure characteristics of the cylinder are retained, both that of the Jake brake during the deceleration process and that of the exhaust shutter brake operation during the evacuation process.

Based on the data yielded by the calculus program developed under Mathcad, a dynamic analysis was conducted for the investigated engine, using specific relations from [9]. The engine torque diagram for the simple action of the exhaust braking system is shown in figure 4.

For the evaluation of the engine torque, it was necessary to determine the gas pressure forces, the inertia forces generated by the translational motion of masses within the motor mechanism and to determine the value of the tangent force component in the rod/crankshaft articulation, as this is the only force that produces torque at the engine shaft. This calculation was made by assuming that the values of the friction forces in the engine mechanism and the forces introduced by the auxiliary systems attached to the engine would be null.

In the case of the engine running in braking mode, it is of interest that the average value of the instantaneous torque per cycle and its amplitude of variation are minimal. This desiderate ensures that a maximum amount of kinetic energy is extracted, as well as the reduction of the mechanical stresses of the motor mechanism.

Thus, in the first case study, for which the injection is suppressed, and the shutter flap is completely closed, the value of the instantaneous engine momentum obtained on the shaft is -13.971 Nm. It can be seen that although the minimum and maximum values of the compression and expansion times are comparable, the decisive influence is that of the torque evolution during the exhaust stroke.

The momentum evolution for the second study case in which the shutter is open is shown in figure 5. The shaft-resistant torque is given this time as a result of opening the exhaust valve in the immediate vicinity of the TDCF.
In the case presented in figure 5, a decrease in the maximum instantaneous torque is observed by comparison to the case of the exhaust brake. Also, the resistive torque registers an increase during the compression stroke, which is directly influenced by the cylinder pressure, which increases due to better filling of the cylinder with fresh gases. It can be observed that the use of this braking system leads to an increase in the braking capacity of the vehicle as a result of reaching a value of \(-16.937\) Nm of the instantaneous torque in this case.

**Figure 5.** Evolution of the engine torque for the case of Jake brake.

**Figure 6.** Evolution of the engine torque for the case of combined effect of exhaust brake and Jake brake.
Figure 6 shows the evolution of instantaneous torque at the shaft corresponding to the combined action of the two auxiliary braking systems. The simulated data shows in this case a decrease of both minimum and the maximum values compared to the case of the Jake engine brake operation. However, due to the evolution of cylinder pressure, which maintains both operating characteristics of the two independent auxiliary systems for this case, a higher average shaft torque value, of -23.661 Nm is obtained. Using the engine power calculus relation, the obtained value of the engine torque, corresponding to the considered engine speed, represents 11.962 hp.

The obtained results highlight the fact that the combination of the two auxiliary systems leads to an important increase in the braking capacity of the vehicle. An overview of the shaft momentum evolution is presented in figure 7.

![Figure 7](image_url)

**Figure 7.** Comparison of the evolutions of the engine torque for the three cases.

An important safety criterion during service that the crankshaft must meet in terms of strength is that of fatigue safety. Due to the fact that the crankshaft is subjected to dynamic stresses, with continuous variations of directions and values of torques applied to the shaft by the rod, it is desirable to ensure that the maximum amplitude per cycle is as low as possible.

From the simulated data, represented graphically in figure 7, it can be observed that the instantaneous moment amplitude per cycle is minimal in the case of combined action of the two braking systems, which leads to a value of 343.689Nm.

4. Conclusions

Although high in-cylinder pressure values lead to an important resistant engine torque, the maximum values of this thermodynamic parameter do not provide enough information about the vehicle's deceleration capacity.

Modeling results illustrated in the present paper confirm the information of [1], according to which the effectiveness of the Jake brake systems is higher than the one corresponding to braking systems that use the exhaust shutter.
The combined action of Jake brake and exhaust shutter auxiliary braking systems was found to yield a significant increase of the mean resistant torque, i.e. an increase of the braking efficiency. The mean resistant torque at the shaft reached in the case of the modelled single cylinder engine a rather important value, of -23.661Nm, which, for the considered engine speed corresponds to a braking power of approximately -12hp.

Combination of the two auxiliary braking systems illustrated by the present work showed that the generated braking power is higher than the nominal power generated by the engine during combustion.

5. References
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