

Decompression Engine Brake Modeling and Design for Diesel Engine Application

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ABSTRACT

The role of the engine brake is to convert a power-producing engine into a power-absorbing retarding mechanism. Modern heavy-duty vehicles are usually equipped with a compression braking mechanism that augments their braking capability and reduces the wear of the conventional friction brakes. This work presents an engine brake mechanism modeling and design based on decompression effect, obtained by exhaust valve opening during the end of the intake cycle. Besides that, during the system operation the emissions are drastically reduced, even eliminated, since there is no fuelling, contributing to pollution level reductions. In this sense, this work describes a development of such engine brake system for a 4 and a 6 cylinder diesel engines.

The engine brake performance was predicted by the development of 1D engine models. The 1D engine models are able to simulate the valve train, including the valves operation, brake flap actuation, hydraulic actuator behavior, and also the major engine breathing characteristics: gas flow rate, turbocharger efficiency, temperatures and pressures along the intake/exhaust system, etc. The gas distribution along the exhaust system can be predicted and its effects on the brake system performance evaluated. With these first assumptions, the first prototype is constructed and the simulation results are compared to the test bench acquired data.

INTRODUCTION

Conventional brake systems are designed to operate in very short periods of time. The continuous use of this kind of system leads to excessive wear on its components and also causes dangerous overheating of braking plates affecting

driving confidence and security, mainly in trucks. Another issue related to conventional system is fuel consumption, as the combustion process is kept activated during the brake operation. For long uses, such as extended downhill routes, the amount of fuel burned and the brake components deterioration are considerable. In order to preserve the conventional brake system and avoid unnecessary fuel burning, the engine can be converted from a power-source to a power-absorbing retarding mechanism. Under usual operating conditions, there are several sources of power dissipation, such as the pumping work (intake and exhaust strokes), rubbing friction and accessories driving. The sum of these factors is the total friction power generated, which aids on engine braking [1,2]. However, to substitute the use of conventional brake systems, much more braking power is necessary, requiring the development of auxiliary braking systems.

The engine brake converts a power producing engine in a power absorbing retarding mechanism and the valvetrain is a significant part of this mechanism. All the engine brakes have one common characteristic: the retarding horsepower increases as engine speeds increases. The major issues that affect the brake performance are compression ratio, engine displacement, valve opening timing and the possible limits on turbo back pressure and injector temperature.

This work presents the development of an engine brake with an exhaust valve opening system since its early design stages, with 1D flow calculation, going through FEA and multibody simulation and also presenting the first test results of a prototype. [Figure 1](#) presents the methodology adopted.

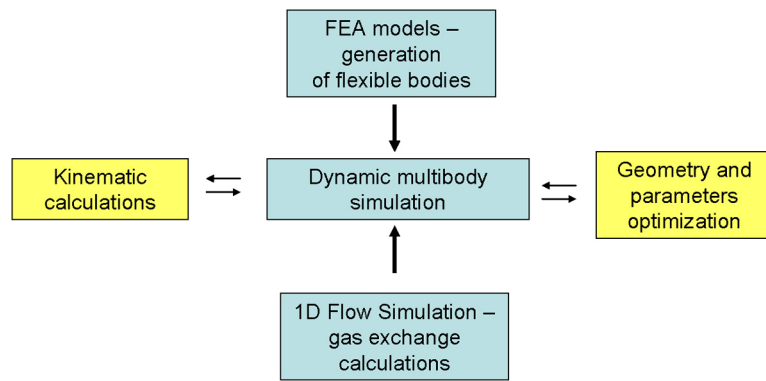


Figure 1. Engine Brake model simulations

EXHAUST BRAKE SYSTEM

Among several kinds of brake systems, the exhaust gases restriction is widely used, being the concept behind this alternative based on power absorbing during the exhaust stroke.

Some of MWM International Acteon engines are equipped with the called Exhaust Brake System. In this system, the exhaust gases are trapped in the engine by the use of an end flow flap installed after the turbine exit, at the beginning of the exhaust pipes. When the system is activated, the flap is closed, avoiding exhausting. The exhaust manifold is then fulfilled with pressurized gases, creating a retarding performance when the exhaust valve opens and the piston is moving upwards forcing the intake air out of the cylinder.

The exhaust valve springs must be well designed in order to avoid valve bouncing due to high back pressure on the exhaust runners. Continuous valve bouncing during brake activation may result in valve seat damages and in some cases, valve breakage. A compromise relation between valve spring preload, flap leakage and maximum pressure on exhaust pipes shall be achieved.

In the Exhaust Brake System, the ignition and fuel injection are turned off, reducing the amount of power generated on the cycle. However, during the compression stroke, the pressurized air exerts a force on the piston, returning most part of the power to the crankshaft.

EXHAUST VALVE OPENING BRAKE SYSTEM

Keeping the exhaust valve opened during the compression stroke is an alternative to improve the performance of the Exhaust Brake System. The energy stored in compressed air during the compression stroke is dissipated in the exhaust system, even with the high back pressure caused by the exhaust flap, due to the exhaust valve opening. During the expansion stroke, there is also a significant increase in

braking efficiency as the piston downwards movement generates a decompression on the cylinder chamber. The ideal exhaust valve lift to achieve this operation condition is shown in [Figure 2](#).

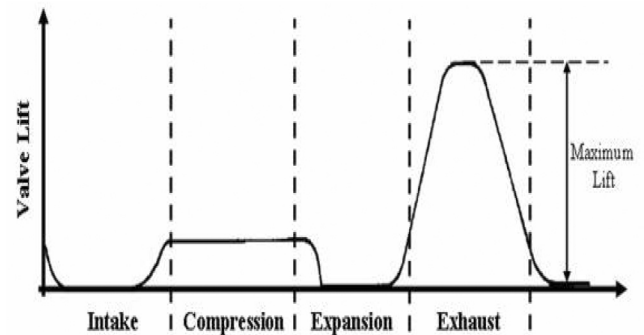


Figure 2. Ideal Exhaust Valve Opening during Compression Stroke [3]

In order to achieve this braking behavior, it was developed by STEYR NUTZFAHRZEUGE AG a braking system based on the positional locking of the exhaust valve, advantaging the fact that, when the exhaust flap is shut, the high back pressure on exhaust runners causes valve bouncing. This is a simple and non-expensive solution that can achieve improvements up to 65% of braking power when compared to the use of only an exhaust flap. [3]

[Figure 3](#) shows details of the mechanism developed to lock the valve open after bouncing. When the exhaust flap is closed, the impulse wave of adjacent cylinders and the back pressure in the exhaust pipe causes high back pressure values on the exhaust runner of a specific cylinder. When the valve bounces, moving down, it is followed by the hydraulic actuator inside the rocker arm, driven by its internal spring. The volume in the rocker arm chamber left by the actuator movement is then fulfilled with oil, which comes through the oil grooves along the spherical check valve at the upper side of the actuator. When the exhaust valve starts closing, the oil pressure inside the actuator chamber gets higher than the pressure in the line. The check valve closes and the space with the oil above the actuator is fixed and the piston locks its position.

The exhaust valve will remain opened, with its lift is limited by the stroke defined by the actuator lateral rail, which is locked by a pin. The high pressure of the oil at this moment forces the rocker arm upwards but the rocker arm movement is avoided by the bracket above it.

When normal valve lift occurs, the rocker arm moves down, opening the oil groove that connects the oil chamber to the top of the rocker arm. The actuator returns to its original position as the oil leaves through oil groove. This sequence of

events occurs every 720° of the crankshaft, while the brake flap is closed.

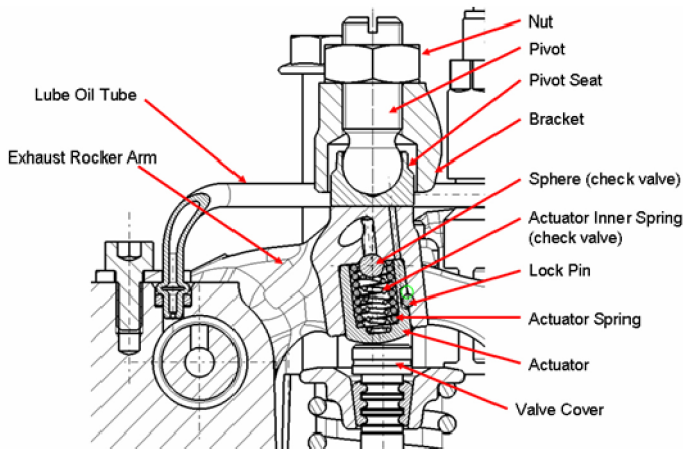


Figure 3. Exhaust Valve Brake System Scheme

The exhaust valve lift obtained with the system mentioned above is shown in [Figure 4](#). During the expansion, as the exhaust valve remains opened, the residual gases in the cylinder slight reduces the brake power efficiency due to decompression.

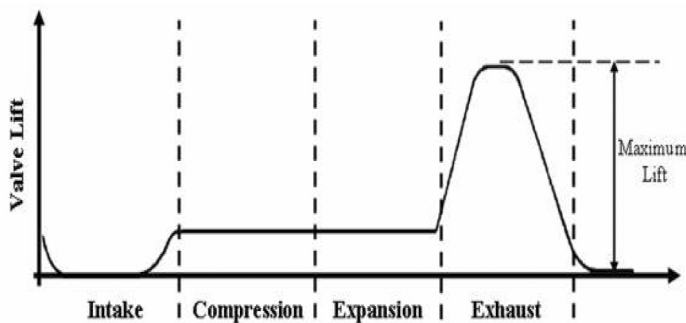


Figure 4. Exhaust Valve Opening during with Exhaust Valve Brake System actuation [3]

This system allows the valve bouncing but, as it is locking the valve opened, the detrimental consequences of this effect, like seat wear and valve breakage, are avoided. Also, although the application engine has two exhaust valves and the hydraulic actuator is installed in only one of them, it is quite not possible to occur valve bouncing in the uncovered side. The bouncing is caused by the force balance between cylinder gas pressure, exhaust runner pressure (back pressure) and valve spring pre-load. As the cylinder and back pressure magnitudes are very similar for both valves, the actuator spring has a strong influence on the system confidence. The actuator spring preload is about 10% to 20% of the valve spring preload, reducing the seating force on this position, and guarantying that this valve opens with lower gas resultant

force. As the pressure on runners decreases, the other valve position remains closed.

[Figure 5](#) shows a comparison between each braking system for the Acteon 6.12 V engine. The Exhaust Valve Brake values were estimated based on a benchmarking comparison, considering base engine design parameters that can directly affect the braking efficiency of this system, such as: displacement, compression ratio, boost pressure, pressure before turbine, etc.

ENGINE BRAKE MODEL

The exhaust valve brake concept has been adapted by MWM International Motores (MIM), to be used in the Acteon engine family, based on design and test results of other engine applications. Then, became the need of knowing how suitable would be this engine brake system and how it would match with MIM applications.

The best way to get this kind of feedback to improve the engine brake design is analyzing its components through FEA and 1D flow calculation, which give some outputs like reliability and performance response.

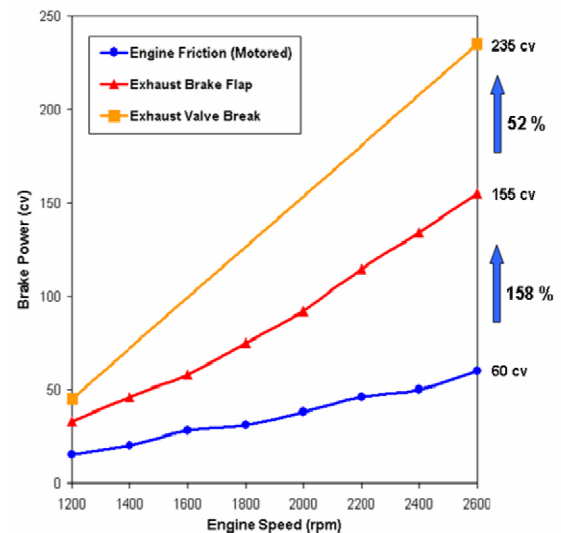


Figure 5. Brake Performance Prediction for a 6 cylinder engine.

Studying the alternatives to predict not only the system operation and its components behavior but also the influence of this operation on the engine performance, GT-Power v6.2 (1D flow simulation software) was chosen instead of conventional CFD analysis.

The advantages of building a model using GT-Power are related to obtaining results that should aid during the design step, avoiding waste time and eliminating costs with engine

tests and prototypes construction. To accomplish these goals, the model must have high robustness and accuracy.

The first approach regarding the model was to correctly simulate the engine brake operation itself, like the valve bouncing when the back pressure wave comes to the runner and the performance of the hydraulic actuator to hold the valve opened. This can be done with a single cylinder model (Fig. 5), where the inputs are only the boundary states just at the engine interfaces which provide results faster than a complete engine model. When a good correlation for this first model is obtained it is ready to be coupled with the other cylinders and the whole engine, where the increasing of brake power under different speeds can be analyzed.

1D MODEL DEVELOPMENT

The valve train components (mechanical parts of the system as valve and actuator) were modeled using linked masses, and a ground template with an imposed movement was used for the rocker arm, as showed in Figure 6. What must be pointed out is that the model elements at this stage have no flexibility avoiding deformation during its operation. So the contact connections between the parts must consider the material properties and also the resultant stiffness of the designed assembly. The flexibility of some parts are then inserted in the multibody model.

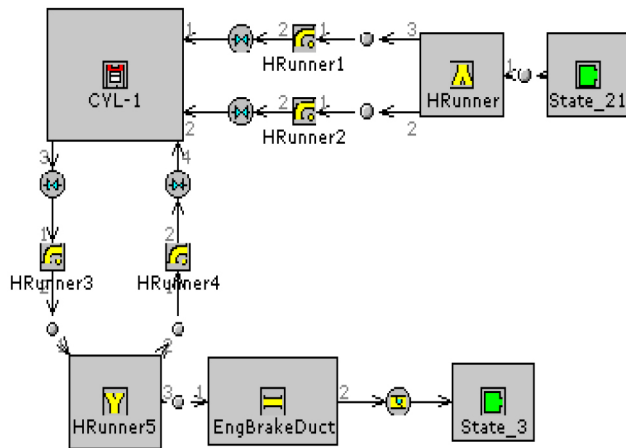


Figure 6a. Single cylinder model

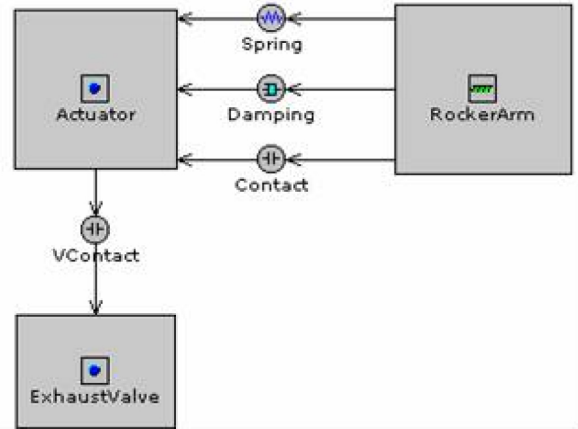


Figure 6b. Engine brake assembly.

Some design features, like spring pre-load, can be inserted into the model, providing mounting characteristics that can be easily changed to evaluate their effect on the system. The contact model allows inserting different multiplicity for each connection side, useful when modeling a hydraulic reduction mechanism or even multiple elements linkage, for instance. The distance in which the stiffness and damping forces will be applied can also be inserted, simulating an oil film between the parts or even a less rigid material. A non linear stiffness coefficient, as shown in Fig. 7, is used to simulate a typical contact between the parts.

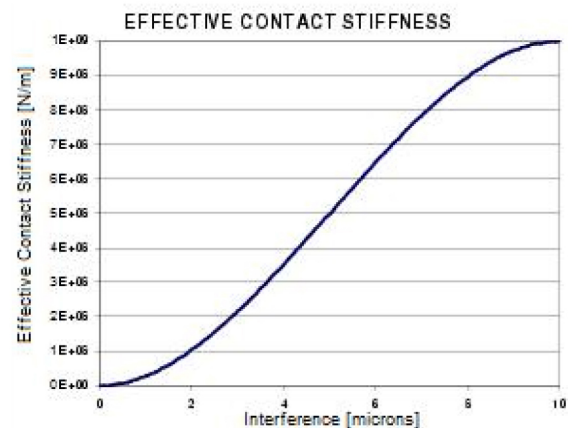


Figure 7. Contact stiffness transition.

Regarding the hydraulic mechanism, a dynamic valve connection was used to allow oil feed get into the actuator and avoid leakage after bouncing. Input parameters like spring installed length and spring stiffness shall be inserted in order to get the same dynamic performance as the real system. The leakage orifice, representing the oil groove, was modeled to directly follow the clearance in the top of rocker arm. Its effective diameter increases between zero and the nominal value (Eq. 1), in order to get the correct effect of fluid leaking given by the rocker arm movement.