

Investigation of Different Valve Geometries and Valve Timing Strategies and their Effect on Regenerative Efficiency for a Pneumatic Hybrid with Variable Valve Actuation

Sasa Trajkovic, Per Tunestål and Bengt Johansson
Lund University

Copyright © 2008 SAE International

ABSTRACT

In the study presented in this paper a single-cylinder Scania D12 diesel engine has been converted to work as a pneumatic hybrid. During pneumatic hybrid operation the engine can be used as a 2-stroke compressor for generation of compressed air during vehicle deceleration and during vehicle acceleration the engine can be operated as an air-motor driven by the previously stored pressurized air.

The compressed air is stored in a pressure tank connected to one of the inlet ports. One of the engine inlet valves has been modified to work as a tank valve in order to control the pressurized air flow to and from the pressure tank.

In order to switch between different modes of engine operation there is a need for a VVT system and the engine used in this study is equipped with pneumatic valve actuators that uses compressed air in order to drive the valves and the motion of the valves are controlled by a combination of electronics and hydraulics.

This paper describes the introduction of new tank valve geometry to the system with the intent to increase the pneumatic hybrid regenerative efficiency. The new tank valve has a larger valve head diameter than the previously used setup described in [1], in order to decrease the pressure drop over the tank valve. In order to ensure tank valve operation during high in-cylinder pressures the valve is combined with an in-house developed pneumatic valve spring which makes the tank valve pressure compensated.

A comparison between the old and the new tank valve geometry and their effect on the pneumatic hybrid efficiency has been done. Also, optimization of the valve timings for both CM (Compressor Mode) and AM (Air-motor Mode) has been done in order to achieve further improvements on regenerative efficiency.

INTRODUCTION

As the pollution standards are becoming more stringent going towards zero emissions together with increasing fuel prices, engine developers are looking for alternative engine management to meet the increasing demand for better fuel economy. Of great importance is to improve the fuel economy at part-load, especially during city driving since it involves frequent acceleration and deceleration. In conventional vehicles all the kinetic energy built up during acceleration is lost during deceleration in the form of heat generated by the friction brakes. This leads to a higher fuel consumption during city driving compared with freeway driving where acceleration and deceleration is less frequent.

Today there are several solutions to meet the demand for better fuel economy and one of them is electric hybrids. The idea with electric hybridization is to reduce the fuel consumption by taking advantage of the, otherwise lost, brake energy. Hybrid operation in combination with engine downsizing can also allow the combustion engine to operate at its best operating points in terms of load and speed.

An electric hybrid consists of two power sources, an ICE (Internal Combustion Engine) and an electric motor that can be used separately or combined. During deceleration the electric motor transforms the kinetic energy of the vehicle to electric power which it then stores in the batteries. The energy stored in the batteries will then be used when the vehicle accelerates. The main disadvantage with electric hybrids is that they require an extra propulsion system and large heavy battery. All this costs the manufacturers a lot of money, which is compensated by a higher end-product price.

One way of keeping the extra cost as low as possible compared to a conventional vehicle, is the introduction of pneumatic hybrid. It doesn't require an expensive extra propulsion source and it works in a way similar to the electric hybrid. During deceleration of the vehicle, the

engine is used as a compressor that converts the kinetic energy contained in the vehicle into energy in the form of compressed air which is stored in a pressure tank. After a standstill the engine is used as an air-motor that utilizes the pressurized air from the tank in order to accelerate the vehicle. During a full stop the engine can be shut off.

Tai et al. [2] describes simulations with a so called "round-trip" efficiency of 36% and an improvement of 64% on fuel economy in city driving. Simulations made by Andersson et al. [3] show a regenerative efficiency as high as 55% for a dual pressure tank system for heavy vehicles. Real engine studies made by Trajkovic et al. [1] shows that the pneumatic hybrid has a promising potential in the reality and a regenerative efficiency of 33% has been achieved.

All these features of the pneumatic hybrid contribute to lower fuel consumption and in combination with the simplicity of the system, the pneumatic hybrid can be a promising alternative to the traditional vehicles of today and a serious contender to the better known electric hybrid.

PNEUMATIC HYBRID

Pneumatic hybrid operation introduces new operating modes in addition to conventional ICE operation. The main idea with pneumatic hybrid is to use the ICE in order to compress atmospheric air and store it in a pressure tank when retarding the vehicle. The stored compressed air can then be used either to accelerate the vehicle or to supercharge the engine in order to achieve higher loads when needed. The pneumatic hybrid also makes it possible to completely shut off the engine at idling like for instance at a stoplight, which in turn contributes to lower fuel consumption. [2,4,5,6]

In this study a single-cylinder engine was used. In reality, in for instance a heavy duty truck, one cylinder will not be enough to take full advantage of the pneumatic hybrid. A pneumatic hybrid will most probably utilize multiple cylinders. The number of cylinders that will be converted for pneumatic hybrid operation for a certain vehicle is hard to estimate at this point. It depends on, among other things, the vehicle weight and the maximum braking torque needed. Drive cycle simulations using data from engine experiments obtained in this study will be conducted in a near future in order to find optimal parameters like required number of converted cylinders and optimal tank volume, just to mention a few.

PNEUMATIC VARIABLE VALVE ACTUATION

In order to be able to switch between all pneumatic hybrid operations and conventional ICE operations, a variable valve actuating system is needed. In this study a pneumatic variable valve actuating system has been used. The valve system is designed and manufactured by a Swedish company named Cargine Engineering AB.

The system uses compressed air in order to drive the valves and the motion of the valves are controlled by a combination of electronics and hydraulics. The system is a fully variable valve actuating system, which means that the valve lift, valve timing and valve lift duration can be completely controlled, independently of each other. The pneumatic valve system in question and the control program has been more thoroughly described by Trajkovic et al. [7].

TANK VALVE

In order to run the engine as a pneumatic hybrid, a pressure tank has to be connected to the cylinder head. Tai et al. [2] describes an intake air switching system in which one inlet valve per cylinder is feed by either fresh intake air or compressed air from the pressure tank. Andersson et al. [3] describes a dual valve system where one of the intake ports has two valves, one of whom is connected to the air tank. A third solution would be to add an extra port to the cylinder head, which would be connected to the air tank. Since these three solutions demand significant modifications to a standard engine a simpler solution, where one of the existing inlet valves has been converted to a tank valve, has been chosen for this study. Since the engine used in this study has separated air inlet ports, there will be no interference between the intake system and the compressed air system. The drawback with this solution is that there will be a significant reduction in peak power, and reduced ability to generate and control swirl for good combustion.

MODES OF ENGINE OPERATION

The main pneumatic hybrid vehicle operations are compressor mode (CM) and air-motor mode (AM). Both modes have been tested in the study described in this paper and an attempt to optimize them has been carried out. The optimization of both CM and AM will be presented later on in this paper.

COMPRESSOR MODE

In CM the engine is used as a 2-stroke compressor in order to decelerate the vehicle. The kinetic energy of the moving vehicle is converted to potential energy in the form of compressed air.

During CM the inlet valve opens a number of CAD after TDC and brings fresh air to the cylinder, and closes around BDC. The moving piston starts to compress the air contained in the cylinder after BDC and the tank valve opens somewhere between BDC and TDC, depending on how much braking torque is needed, and closes shortly after TDC. The compressed air generated during CM is stored in a pressure tank that is connected to the cylinder head.

AIR-MOTOR MODE

In AM the engine is used as a 2-stroke air-motor that uses the compressed air from the pressure tank in order

to accelerate the vehicle. The potential energy in the form of compressed air is converted to mechanical energy on the crankshaft which in the end is converted to kinetic energy.

During AM the tank valve opens at TDC or shortly after and the compressed air fills the cylinder to give the torque needed in order to accelerate the vehicle. Somewhere between TDC and BDC the tank valve closes, depending on how much torque the driver demands. Increasing the tank valve duration will increase the torque generated by the compressed air. The inlet valve opens around BDC in order to avoid compression.

PRESSURE COMPENSATED TANK VALVE

In experiments done by Trajkovic et al. [1], one of the intake valves was chosen to work as tank valve. The intake valve had to be modified in order to make it possible to achieve desired TankVO (Tank Valve Opening) at high in-cylinder pressures. The tank valve diameter had to be reduced from the original size of 45 mm, down to 16 mm. Also the tank valve spring preloading had to be changed from 100 N to 340 N in order to keep the tank valve completely closed at tank pressures up to 25 bars. Both modifications lead to some complications. The reduced valve diameter increases the pressure losses over the tank valve and thus the regenerative efficiency will be reduced. The increased spring pre-loading will affect the pneumatic valve actuator energy consumption, but this is only of importance in a real vehicle where the actuators have to be feed with pressurized air from a compressor.

In an attempt to avoid the pressure losses over the tank valve, an in-house designed pneumatic valve spring has been developed and has replaced the conventional tank valve spring. Figure 1 shows a simple cross section illustration of the pneumatic valve spring arrangement mounted on the cylinder head. The pneumatic spring is constructed in such way, that it uses the tank pressure to keep the valve closed. A cylinder (indicated by number 1) is placed on the top of the cylinder head (indicated by number 4), with the tank valve (indicated by number 3) in the center of the cylinder. The cylinder is sealed in the bottom against the cylinder head and on the top the cylinder is sealed with the valve spring retainer (indicated by number 2). The space between the bottom sealing and the tank valve spring retainer is the pneumatic valve spring and it is connected to the tank valve port (indicated by number 6) and thus to the compressed air through 4 passages machined on the tank valve (indicated by number 5). The pressurized air enters the air passages on the valve and is guided up to the pneumatic valve spring, as indicated by the blue arrows. The passages are made in such way, that the pneumatic spring will be connected to the tank valve port at any time and any valve lift. Since the compressed air in the pneumatic spring works on the underside of the tank valve spring retainer and the compressed air in the tank valve port acts on the upside of the tank valve

head, as indicated by the yellow arrows, the net force should be zero, and thus the valve should be pressure compensated. This means that the tank will be kept closed without using any valve spring and the valve diameter can now be increased in order to reduce the pressure drop over the valve. In the study presented in this paper, the tank valve diameter has been increased to 28 mm, which is almost the double of what has been used before. Figure 2 shows a picture of the two tank valves used in this study. Geometrical properties of the pneumatic spring arrangement can be seen in Table 1.

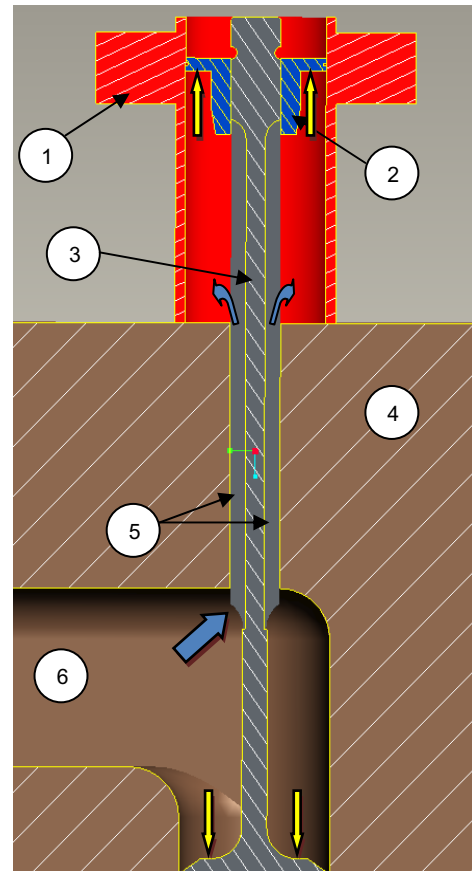


Figure 1 A simple cross section illustration of the pneumatic valve spring arrangement mounted on the cylinder head

When the valve is open the force generated by the pressurized air acting on the upside of the tank valve head is canceled. This means that there now only exist a considerable force on the undersurface of the valve spring retainer trying to close the tank valve. In order to overcome this problem, the tank valve actuator is feed with compressed air from the tank. This means that the pressure at a certain time is the same in the pneumatic valve spring as in the valve actuator. Since the actuator piston has a larger diameter than the tank valve spring retainer, the actuator will always have enough power to open the valve and maintain it open for as long as desired.



Figure 2 Picture illustrating the difference between the “small tank valve” and the “large tank valve”

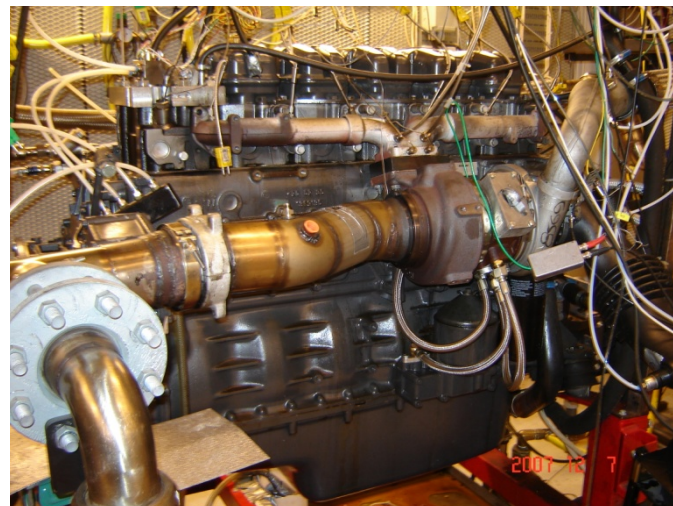


Figure 3 The engine used in this study

Table 1 Geometric properties of the spring

Pneumatic spring cylinder inner diameter	28 mm
Tank valve spring retainer diameter	28 mm
Tank valve poppet diameter	28 mm
Actuator piston diameter	32 mm
Compressed air guiding passage cross-section area	6 mm ²

MODIFICATIONS TO THE PNEUMATIC SPRING

After some initial testing, some issues have been observed with the pneumatic valve spring. The issue of greatest importance is that the tank valve self-opens at certain running conditions during testing. The reason behind this behavior of the valve is high pressure oscillations in the tank valve port which haven't been taken into consideration. When a high pressure pulse arrives to the tank valve, the force acting on the tank valve head will be increased. The tank valve will no longer be pressure compensated and the net force acting on the valve will not be zero, and as a result of this the valve will self-open. In order to eliminate this problem a valve spring with 220 N preloading has been added to the tank valve, which means that there will always be a net force acting to keep the tank valve closed.

EXPERIMENTAL SETUP

The engine used in this study can be seen in Figure 3. It is a 6-cylinder Scania D12 diesel engine converted to a single-cylinder engine. The engine is equipped with the pneumatic variable valve actuating system described earlier in this paper. The geometric properties of the engine can be seen in Table 2. Figure 4 shows a close-up of the pneumatic valve actuators mounted on top of the Scania cylinder head.

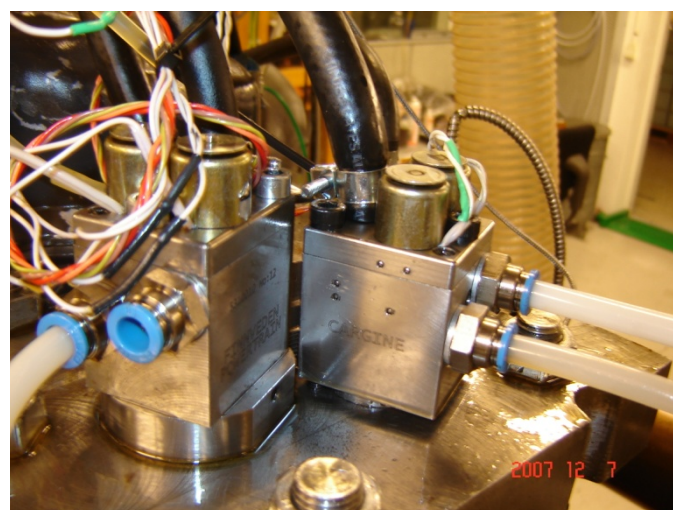


Figure 4 The pneumatic valve actuators mounted on the Scania cylinder head

The engine has two separated inlet ports and therefore they are suitable to use with the pneumatic hybrid since there will be no interference between the intake air and the compressed air. One of the inlet valves was therefore converted to a tank valve.

The exhaust valves were deactivated throughout the whole study because no fuel was injected and thus there was no need for exhaust gas venting.

Table 2 Engine geometric properties

Displaced Volume	1966 cm ³
Bore	127.5 mm
Stroke	154 mm
Connecting Rod Length	255 mm
Number of Valves	4
Compression Ratio	18:1
Piston type	Flat
Inlet valve diameter	45 mm
Tank valve diameter	16 and 28 mm
Piston clearance	7.3 mm

The pressure tank used in this study is an AGA 50 litre pressure tank suitable for pressures up to 200 bars and it is shown in Figure 5. The pressure tank is connected to the cylinder head by metal tubing. In order to eliminate the metal tubing as a potential bottleneck on account of air flow choking, the diameter of the tubing was doubled at the same time as the small tank valve was replaced by the large tank valve. The tank size in the current system is selected based on availability rather than optimality, but in the future the tank volume will be an important parameter for the optimization of the system.

**Figure 5 The pressure tank connected to the cylinder head by metal tubing**

Table 3 shows some valve parameters. The maximum valve lift height in this study is limited to 7 mm in order to avoid valve to piston contact. The valve system can, when unlimited, offer a valve lift height of about 12 mm.

Table 3 Valve parameters

Inlet valve supply pressure	4 bar
Tank valve supply pressure	6 bar
Hydraulic brake pressure	4 bar
Inlet valve spring preloading	100 N
Tank valve spring preloading	220 N
Maximum valve lift	7 mm

Since the pneumatic valve spring arrangement require the actuator to be fed by compressed air from the tank, an extra pressurized air supply line had to be added. A problem with the actuator being fed with tank pressure is that there is a pressure threshold below which the pneumatic valve actuator will not work as expected. This means that the actuator has to be fed with compressed air from an external source. This adds thereby the need of having a pressure source switch. The switching system used in this study is built up by two check-valves which are arranged in such way that the source feeding the valve actuator will always be the source with the highest pressure. For instance if the external source of compressed air is set to 6 bars it will be the main feeding source until the pressure in the pressure tank exceeds 6 bars. A picture of the pressure source switching system is shown in Figure 6.

**Figure 6 Pressurized air switching system built up by two check-valves**

ENGINE EXPERIMENTAL RESULTS

Both CM and AM have been tested in the study described in this paper. Main focus have been on the optimization of CM and AM and comparison between the old and new setup. The results will be discussed thoroughly. Note that all presented pressure values are absolute and IMEP is given in 2-stroke scale.

COMPRESSOR MODE

The CM operation can mainly be done in three ways. The first way is to achieve as high compression efficiency as possible. This is done by the introduction of a feedback control of the tank valve. The tank valve then opens when the in-cylinder pressure is equal to the tank pressure.

The second way is to achieve as much braking torque as possible. The maximum braking torque is achieved when the tank valve opens at or shortly after BDC. This strategy will lead to a blowdown of pressurized air from the pressure tank into the cylinder and thus the cylinder will be charged with air at current tank pressure instead of atmospheric air.

Finally, the third way is actually a combination of the previous two and is the one that will be used in a real application. For instance, if the driver releases the gas pedal, the engine will be operating as a CM at highest efficiency. If the driver then presses the brake pedal, the CM operation will drift away from highest efficiency towards highest braking torque.

This study focuses mainly on the first method, i.e. achieving as high CM efficiency as possible.

The experiments described below are done for both the old setup with a tank valve head diameter of 16 mm and for the new setup with a pressure compensated tank valve with a tank valve head diameter of 28 mm.

From now and further on the tank valve with a valve head diameter of 16 mm will be referred to as "small tank valve" and the pressure compensated valve with a valve head diameter of 28 mm will be referred to as "large tank valve".

Optimizing the compressor mode, small tank valve

Trajkovic et al. [1] focused mainly on a valve strategy based on the polytropic compression law:

$$p_2 = p_1 \left(\frac{V_1}{V_2} \right)^\kappa \quad (1)$$

where p_1 corresponds to the pressure at BDC and p_2 is the pressure at any point in the cycle. V_1 is the maximal volume in the cylinder and V_2 is the cylinder volume at cylinder pressure p_2 . By setting p_2 equal to the tank pressure, the volume at the given pressure can be calculated and from that it is possible to calculate proper tank valve timings. A drawback with this strategy is that the polytropic exponent, κ , depends on the heat losses and setting this ratio to constant value introduces some errors to the tank valve control algorithm. In order to avoid this error, a method for optimizing the CM has been suggested by Trajkovic et al. [1]. The main idea with this method is to find the most optimal valve timing at a given tank pressure and, in order to do that, the tank pressure needs to be constant throughout the whole

testing interval. With the aid of a pressure relief valve it is possible to change the pressure level in the system. The pressure in the tank will become constant when the amount of air charged into the tank is equal to the amount of air released from the tank and by adjusting the pressure relief valve opening angle it is possible to set a desired steady-state tank pressure.

Figure 7 shows a TankVO optimization sweep for the small tank valve at various steady-state tank pressures. TankVC, IVO and IVC were set to a constant value at this optimization. It can clearly be seen how negative IMEP is affected by TankVO timing during optimization of CM. The figure shows that there is an optimal TankVO timing for every tank pressure when taking highest efficiency into consideration, highest efficiency corresponds to the minimum in each curve. This means that it takes less power to compress the inducted air at this point than at any other point on the curve at a given tank pressure. If higher braking torque is needed, the efficiency has to be sacrificed.

The reason why negative IMEP increase at early TankVO is that when the tank valve opens earlier than optimal, there will be a blowdown of compressed air into the cylinder due to the fact that the pressure level in the cylinder is lower than in the pressure tank. At a certain premature TankVO, negative IMEP will dramatically increase with increasing tank pressure, due to a larger pressure level difference between the cylinder and the pressure tank.

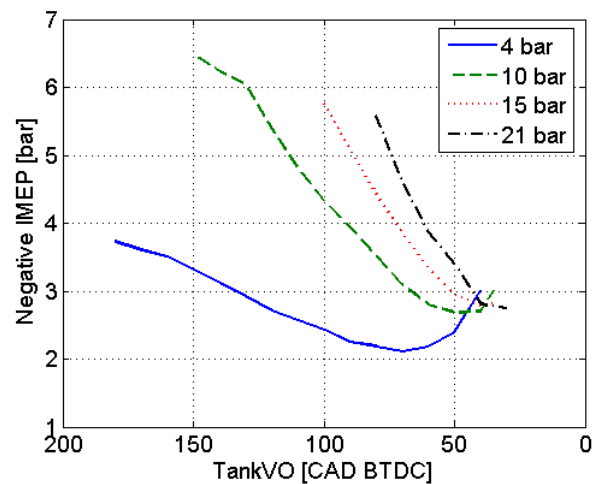


Figure 7 Negative IMEP obtained during steady-state CM as a function of TankVO for the small tank valve setup during optimization of CM at various tank pressures and an engine speed of 600 rpm

Figure 8 shows how optimal TankVO and corresponding negative IMEP varies with increasing tank pressure. The reason why the negative IMEP decrease after a tank pressure of approximately 14 bar, is that the optimization has been done with focus on TankVO while TankVC has been set to a constant value of 10 CAD ATDC. Constant