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

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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 declaration 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.

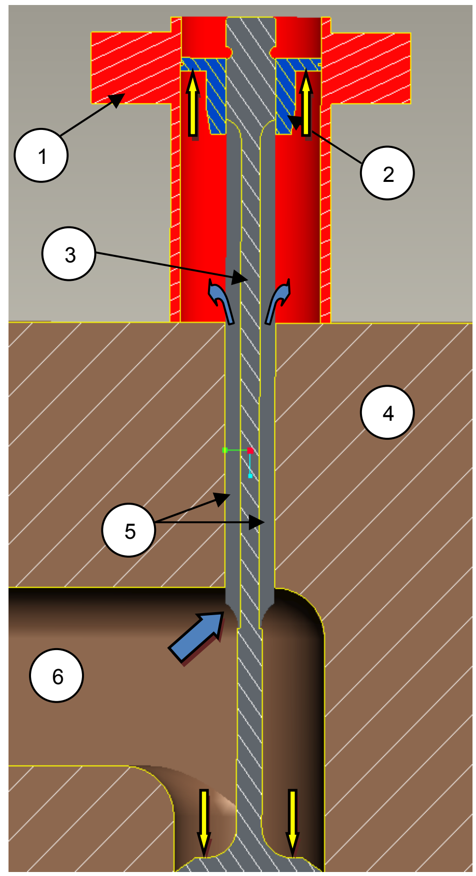

Figure 1 A simple cross section illustration of the pneumatic valve spring arrangement mounted on the cylinder head

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"

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.

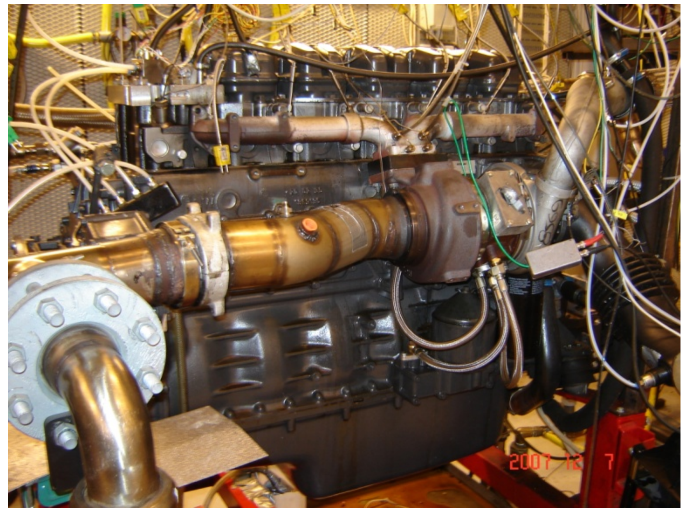

Figure 3 The engine used in this study

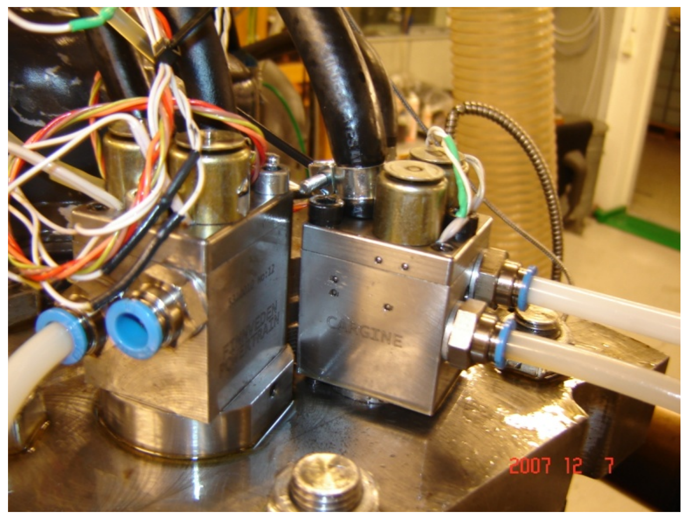

Figure 4 The pneumatic valve actuators mounted on the Scania cylinder head

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 cm3
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:

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

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 introducers 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.

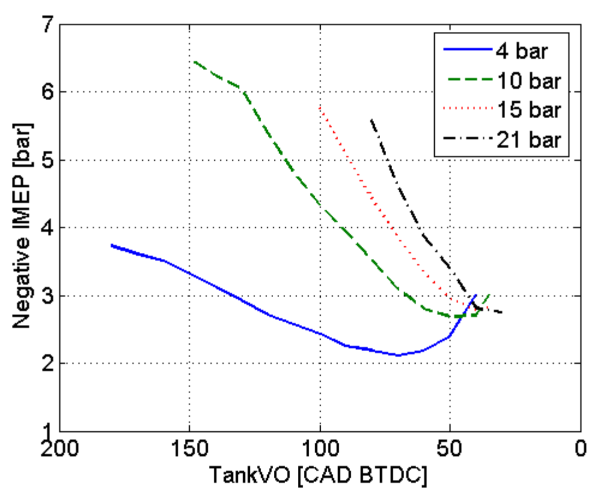

Figure 7 Negative IMEP obtained during steady-state CM as a function of TankVO for the small tank valve setup during optimization of CM at various tank pressures and an engine speed of 600 rpm

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

TankVC will affect the negative IMEP at higher tank pressures because ideally TankVC should be retarded towards TDC with increasing tank pressure. IVO should be set in such way that the pressurized air trapped in the cylinder after TankVC will be expanded to atmospheric pressure. At higher tank pressures, a late TankVC means that the in-cylinder pressure is higher than desired, and this excess pressure pushes the piston as it moves towards BDC and thereby contributes with positive IMEP which decreases the negative IMEP for the whole cycle.

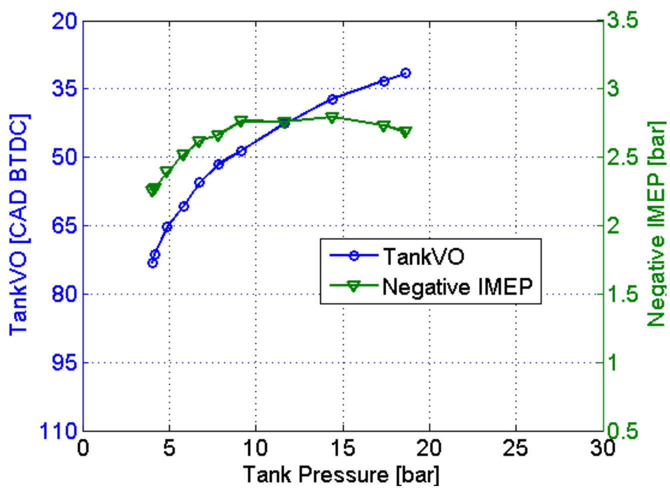

Figure 8 TankVO and negative IMEP as a function of tank pressure for the small tank valve setup at an engine speed of 600 rpm

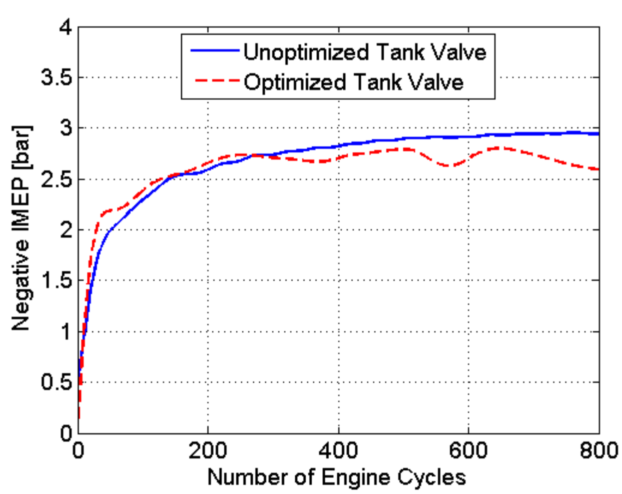

Figure 9 Comparison of negative IMEP for unoptimized and optimized TankVO as a function of engine cycle number during CM operation for the small tank valve setup. End tank pressure is about 21 bars in both cases

CM operation testing has been done both with an unoptimized tank valve, i.e. the feedback control of the tank valve was based on equation (1), and with the optimal TankVO timings seen in Figure 8. The result can be seen in Figure 9 where negative IMEP for both the unoptimized tank valve and the optimized tank valve is shown. Initially, negative IMEP for the unoptimized tank valve is similar to negative IMEP for the optimized tank valve, but after about 300 engine cycles negative IMEP for the optimized tank valve remains reasonably constant while negative IMEP for the unoptimized tank valve continues to increase throughout the rest of the test.

The reason why negative IMEP is decreasing during optimized tank valve operation between 600 and 800 engine cycles is that the optimization is based on the valve timings shown in Figure 8, and thus the same IMEP behavior is expected.

<u>Optimizing the compressor mode, large tank valve</u>

In order to reduce the pressure drop over the tank valve the small valve has been replaced with a valve that has a larger valve head diameter in combination with the previously described pneumatic spring arrangement.

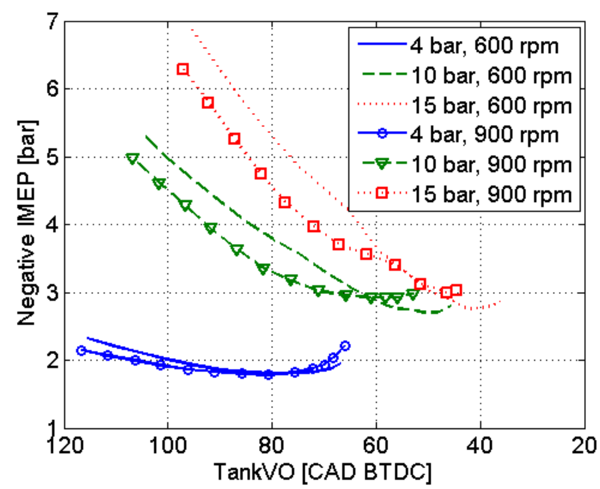

Figure 10 Negative IMEP obtained during steady-state CM as a function of TankVO for the large tank valve setup at various tank pressures for engine speeds of 600 and 900 rpm

Figure 10 shows a TankVO optimization sweep for the large tank valve at various steady-state tank pressures, similar to the TankVO sweep in Figure 7. With the larger valve, the negative IMEP at a tank pressure of 4 bar and at optimal TankVO for both 600 and 900 rpm is lower compared to the negative IMEP in Figure 7 at the same tank pressure. This is because the pressure drop is decreased due to the larger tank valve head diameter. At 10 and 15 bars, there is hardly any difference between negative IMEP for the large tank valve compared to the small tank valve. The only difference is that the optimal TankVO for the large tank valve is advanced a number of CAD compared to TankVO for the small valve. One of the reasons for this behavior is most likely that there is a blowdown of pressurized air into the cylinder through the tank valve. It seems as the tank valve is not completely pressure compensated as expected and thus there exist a net force acting to open the tank valve. Another reason is that there is some pressure losses in the pressurized air supply line between the tank and the pneumatic valve actuator, which means that the pressurized air fed to the valve actuator at a certain time is not the same as the mean tank pressure at the corresponding time and therefore the ability to open the tank valve at optimal timing is lost and has to be advanced. An advance in TankVO compared to optimal timing means that there will be a blowdown of pressurized air into the cylinder and thus negative IMEP will increase. The reason why negative IMEP for the large tank valve is considerably lower than for the small tank valve at 4 bars tank pressure is that at this tank pressure level, the actuator is fed with 6 bars of compressed air from an external source. This means that, at this point there is a surplus of 2 bars feeding the valve actuator and thus the optimal TankVO timing can be achieved.

Another observation that can be made from Figure 10 is that at a certain tank pressure, there is a difference in negative IMEP between the case at 600 rpm and the case at 900 rpm. The reason is that at a higher engine speed, there will be less time for the blowdown process previously described and thereby its effect on the negative IMEP will be less.

Figure 11, Figure 12 and Figure 13 show how optimal TankVO and corresponding negative IMEP varies with increasing tank pressure at various engine speeds for the large tank valve setup. Comparing negative IMEP from Figure 11 with negative IMEP from Figure 8 indicates that the pressure losses over the valve are lower with the large valve than with the small valve. If focus is put on the TankVO, it can clearly be seen that the optimal TankVO for the large tank valve is occurs considerably earlier compared to the optimal TankVO for the small tank valve. The reason for this behavior can once again be explained as inadequate amount of pressurized air supplied to the tank valve actuator and therefore the valve timing has to be advanced a number of CAD away from the real optimum, which contributes to a higher negative IMEP.

It can also be seen that TankVO advances with engine speed at the same tank pressure with the reason being the same as stated previously. As the engine speed increase, the pressure tank charging rate increase faster than the supply rate of compressed air to the actuator due to restrictions in the pressurized air supply line between the pressure tank and the valve actuator. The result is that at a certain time, mean tank pressure will be higher than actuator supply pressure. As engine speed increase, the pressure difference between the tank pressure and the actuator supply pressure will also increase. Therefore TankVO has to be advanced with increasing engine speed.

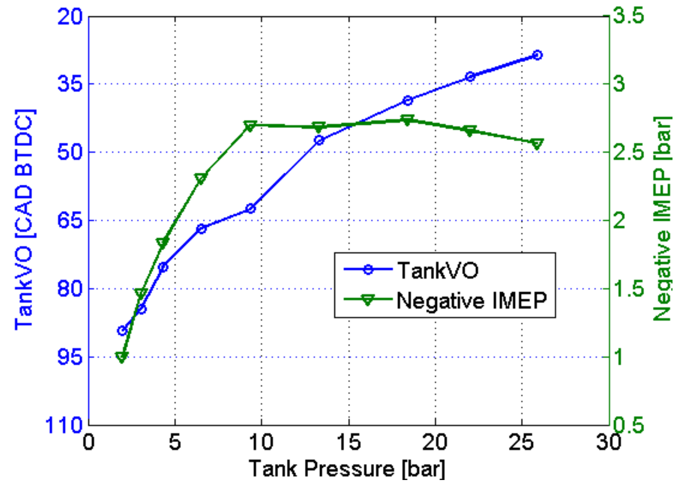

Figure 11 Optimized TankVO timing and corresponding negative IMEP during CM optimization as a function of tank pressure for the large tank valve setup at an engine speed of 600 rpm

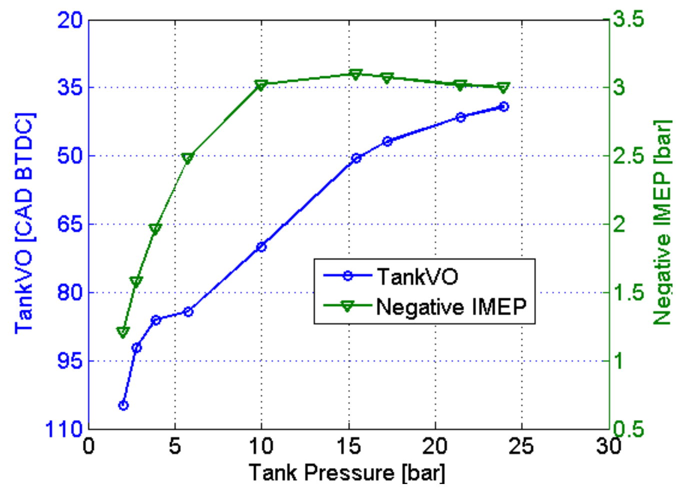

Figure 12 Optimized TankVO timing and corresponding negative IMEP during CM optimization as a function of tank pressure for the large tank valve setup at an engine speed of 900 rpm

Notice that at a tank pressure of 4 bars, negative IMEP at all three engine speeds presented in Figure 11, Figure 12 and Figure 13, are lower than negative IMEP shown in Figure 8. This is once again due to the surplus of 2 bars in the pressurized air supply line feeding the tank valve actuator since the compressed air source is external with an air pressure set to 6 bars. This indicates that in order to achieve optimal TankVO timing, there should always be a surplus of minimum 2 bars in the supply line to compensate for all the pressure losses through it.

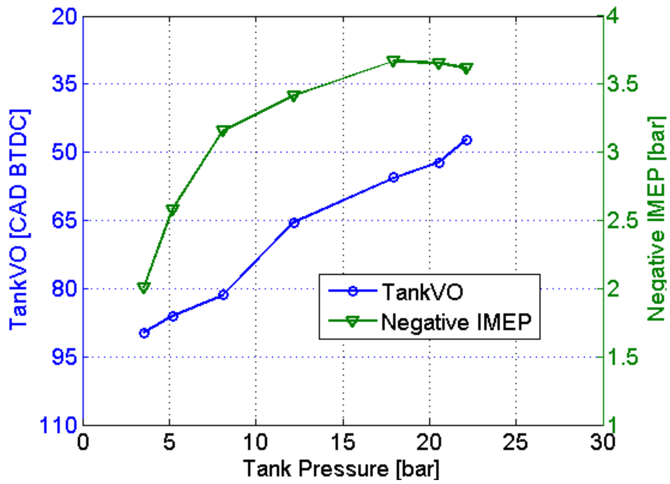

Figure 13 Optimized TankVO timing and corresponding negative IMEP during CM optimization as a function of tank pressure for the large tank valve setup at an engine speed of 1200 rpm

Comparing Figure 11, Figure 12 and Figure 13 with one another indicates that at equal tank pressure, negative IMEP increases with engine speed. This is due to the fact that with increasing engine speed there is less time to vent the cylinder from compressed air and thus there will be an overshoot in cylinder pressure which increases with increasing engine speed. This is illustrated in Figure 14.

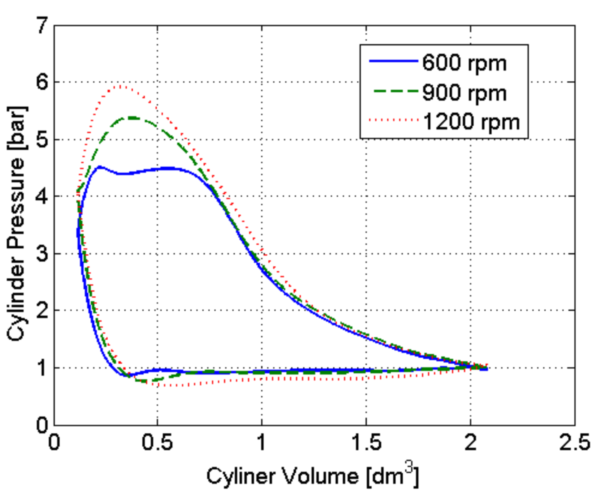

Figure 14 PV-diagram during CM for three different engine speeds at 4 bars of tank pressure

Figure 15 shows negative IMEP during optimal CM at three different engine speeds for the large tank valve setup. Comparing the result displayed in Figure 15 with the result from the optimized test shown in Figure 9 indicates that there is only a difference the first 200 cycles. The results obtained with the large tank valve shows that IMEP is lower than the corresponding results obtained with the small tank valve which verifies that increasing the tank valve diameter decreases the pressure drop over the tank valve. The reason why negative IMEP is almost the same in both cases after 200 cycles is probably once again insufficient pressure in the compressed air supply line feeding the tank valve actuator.

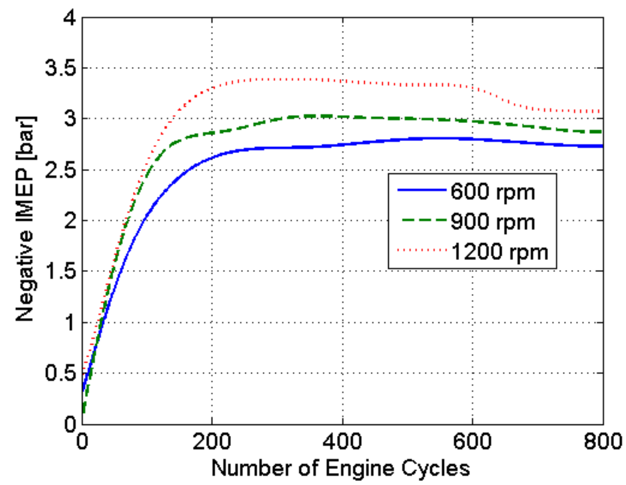

Figure 15 Negative IMEP during optimal CM as a function of engine cycle number at three different engine speeds for the tank valve setup

The corresponding mean tank pressure as a function of engine cycle number at three different engine speeds can be seen in Figure 16. There is a slight difference between the curves. The end pressure at 900 rpm is higher than at both 600 and 1200 rpm. This can probably be explained with tank pressure leakage into the cylinder through the tank valve. This would mean that there exists a net force acting to open the tank valve and in that case the tank valve is not pressure compensated. Since leakage of compressed air through an orifice is constant in time at a constant pressure, the amount of air inducted into the cylinder per cycle due to leakage will decrease with increasing engine speed, and thus the end tank pressure should reach a higher level. At 1200 rpm, this explanation is not enough since the end tank pressure is at the same level as at 600 rpm. Careful observation of the tank pressure curve at 1200 rpm in Figure 16, indicates that it follow the tank pressure curve at 900 rpm quite well up to approximately 600 engine cycles and then it starts to deviate and ends at the same pressure level as at 600 rpm. The reason is probably that, during this specific test run, the tank valve actuator didn’t behave as expected. This can be verified from Figure 15, where negative IMEP at 1200 rpm starts to decrease at about 600 cycles. The reason for this behavior is that, due to inexplicable performance of the tank valve actuator, the duration of maximum valve lift abruptly decreases after 600 cycles, as displayed in Figure 17. A shorter duration means that there will not be enough time to vent the cylinder on pressurized air and therefore the cylinder will still be filled with pressurized air at tank valve closing. This excess of compressed air pushes the piston as it moves towards BDC and thus contributes with positive IMEP, which in turn decreases negative IMEP for the whole cycle.

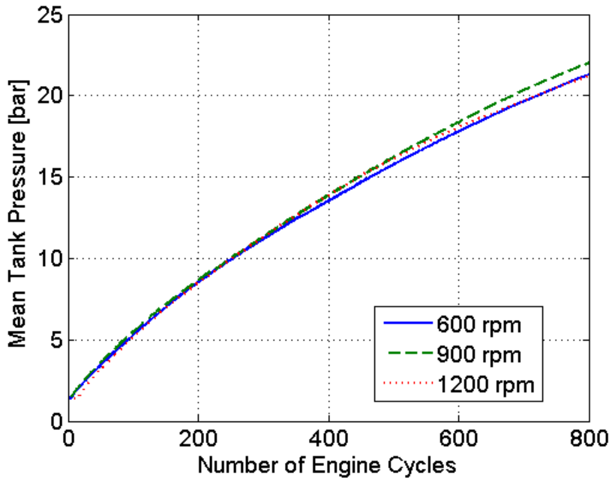

Figure 16 Mean tank pressure as a function of engine cycle number at three different engine speeds

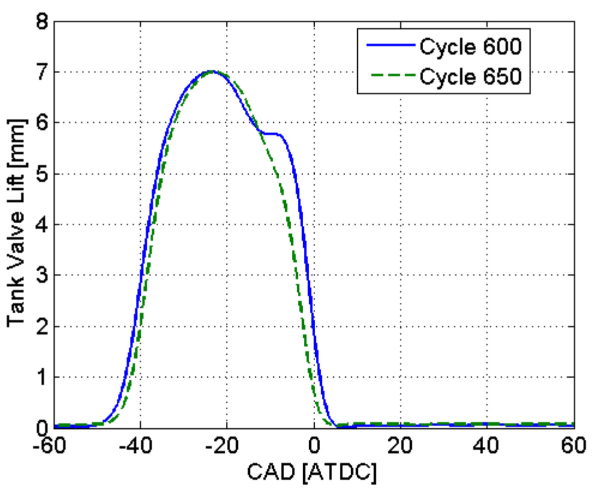

Figure 17 Tank valve lift at two different engine cycles at an engine speed of 1200 rpm

Figure 18 and Figure 19 displays the port temperature immediately after the tank valve and the temperature in the pressure tank, respectively, at three different engine speeds. The results indicate that there is a considerable amount of heat losses through the system. The explanation is that the metal tubing, which connects the cylinder head with the tank, and the pressure tank are not heat insulated in any way. Therefore, a considerable amount of the total thermal energy generated during CM will be transferred to the surroundings and thus the total efficiency of the pneumatic hybrid will be decreased.

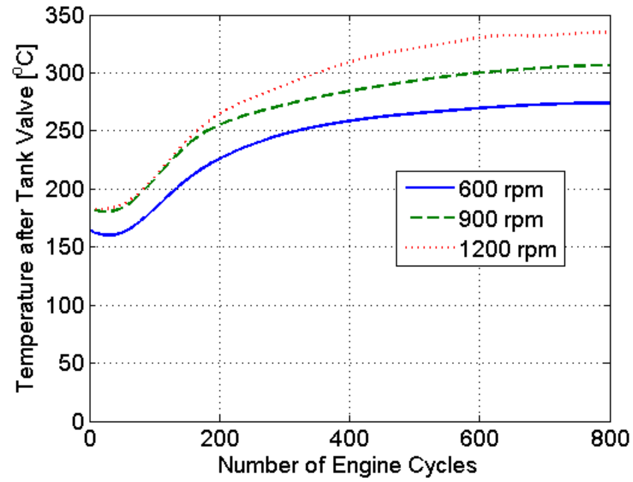

Figure 18 Temperature after tank valve (port temperature) as a function of engine cycle number at three different engine speeds

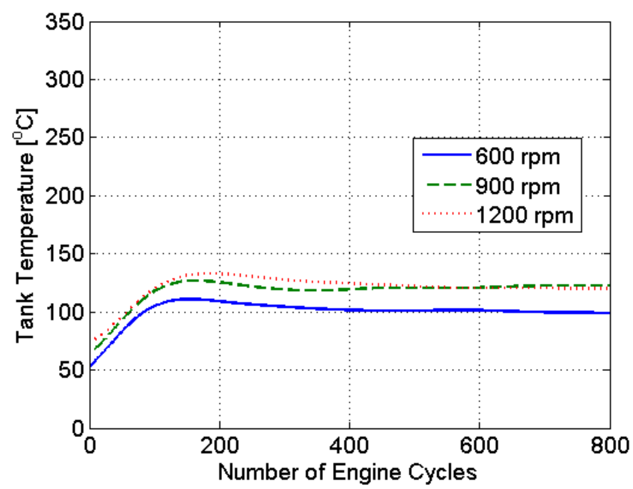

Figure 19 Pressure tank temperature as a function of engine cycle number at three different engine speeds

Figure 20 and Figure 21 illustrates the pressure drop over the small tank valve and the large valve, respectively, obtained at three different engine speeds during CM. The hump-like behavior between 100 and 200 engine cycles at both 900 and 120 rpm in Figure 21 occurs due to the pressurized air source switching system described earlier. As the tank pressure is reaching a pressure level close to the switching pressure level, the tank valve lift height is decreasing, as can be seen in Figure 22. The tank valve lift has decreased by almost 1 mm at engine cycle 150 compared to cycle 100. At cycle 200 the tank valve lift height has nearly returned to a maximum. The reason for this behavior is thought to be bad interaction between the check-valves due to pressure oscillations in the pressurized air supply line. During normal running conditions one of the check-valves will be completely open, but during the transition period, both check-valves will open and close frequently due to the pressure oscillations, which lead to a deficit in pressure and thus the valve lift will decrease.

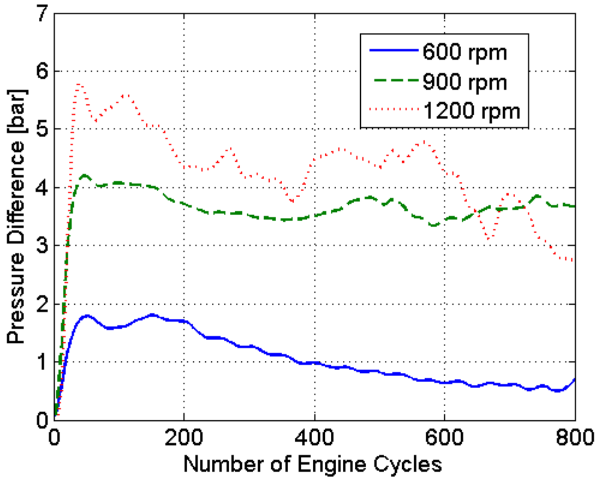

Figure 20 Pressure difference between maximum in-cylinder pressure and maximum pressure after the tank valve (port pressure) as a function of engine cycle number at various speeds. Tank valve with small diameter was used during this test.

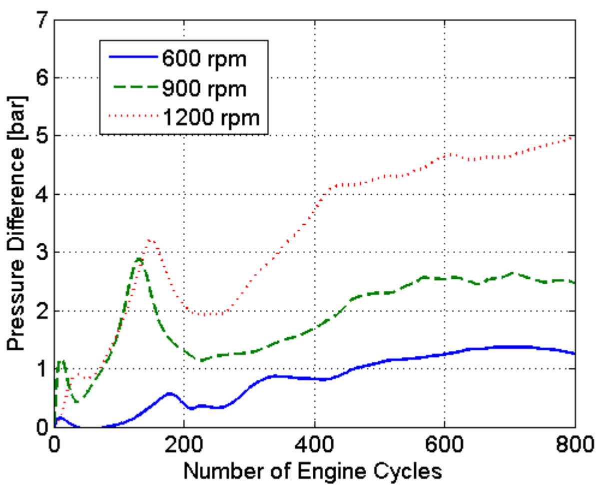

Figure 21 Pressure difference between maximum in-cylinder and maximum pressure after the tank valve (port pressure) as a function of engine cycle number at various engine speeds. Tank valve with large diameter was used during this test.

Another observation made from Figure 20 and Figure 21 is that the curves displayed in Figure 21 are indicating an increasing trend, while the curves in Figure 20 are decreasing or somewhat constant throughout the whole test. The reason is, as stated before, insufficient pressure in the pressurized air supply line feeding the tank valve actuator which forces TankVO to deviate from the optimal timing, and thereby the pressure difference between the pressure after the valve and the cylinder pressure will increase.

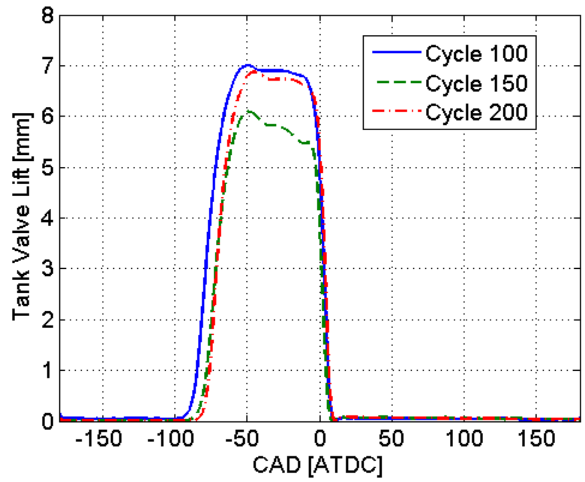

Figure 22 Tank valve lift at three different engine cycles and at an engine speed of 1200 rpm

Figure 23 and Figure 24 show cylinder pressure at a mean tank pressure of 8 bars for the small tank valve setup and the large tank valve setup, respectively. It can clearly be seen that the pressure losses over the small tank valve are considerably higher than they are with the larger tank valve. This is mainly because choking occurs over the small tank valve which limits the air flow and thereby the pressure will increase. Figure 23 and Figure 24 indicates that this overshoot in pressure can be lowered dramatically if the small tank valve diameter is increased.

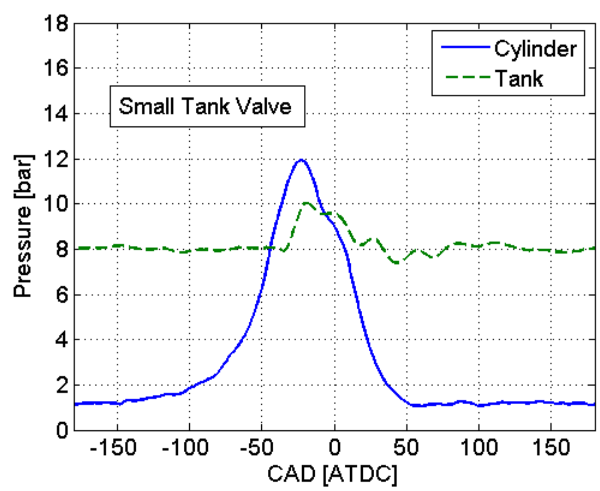

Figure 23 In-cylinder pressure and tank pressure during CM with a small valve diameter at revolution 200 and an engine speed of 600 rpm

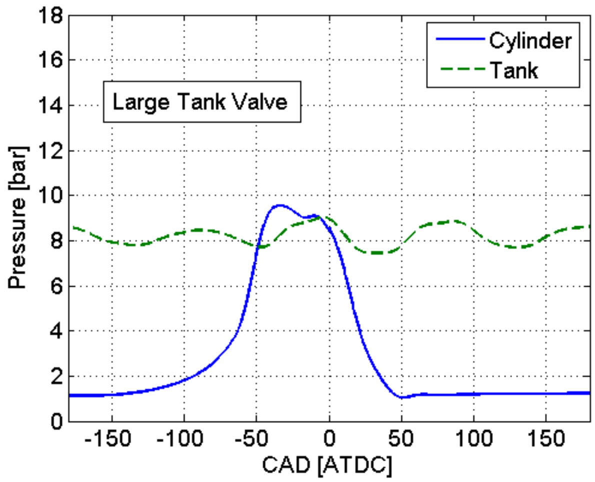

Figure 24 In-cylinder pressure and tank pressure during CM with a large valve diameter at revolution 200 and an engine speed of 600 rpm

The cylinder pressure curves from Figure 23 and Figure 24 can be seen in the form of PV-diagrams in Figure 25. The PV-diagram for the large tank valve setup, bear a great resemblance to an ideal CM PV-diagram, with an almost constant cylinder pressure between cylinder volume 0.5 and 0.2 dm^3. It can be noticed that for the case with the large tank valve, at approximately 0.7 dm^3, the cylinder pressure deviates from the small tank valve pressure trace. The cause is that TankVO is advanced a number of CAD in comparison to optimal tank valve timing due to reasons already stated in this paper, namely insufficient pressure in the compressed air supply line feeding the tank valve actuator.

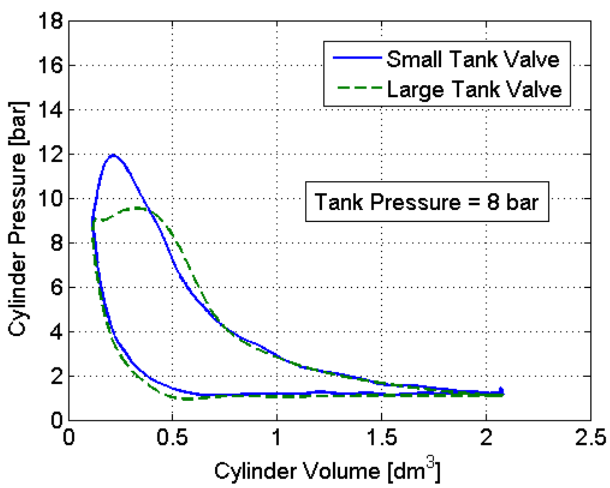

Figure 25 PV-diagrams for two different tank valve diameters at a tank pressure of 8 bars

Figure 26 illustrates how the pressure in the port, immediately after the tank valve varies during one engine cycle for both the small tank valve setup and the large valve setup. The major difference between the curves is the large pressure oscillation during the entire cycle for the test carried out with the large tank valve. The reason for this occurrence is that the metal tubing for the large tank valve setup has a larger diameter than for the small valve setup. The increase in diameter eliminates the possibility of having any choking in the tubing, but instead the pressure oscillations through it increase. If the tank valve opens when the pressure oscillation at the valve is at a local maximum, the cylinder will be filled with some additional air, which in turn leads to a higher negative IMEP.

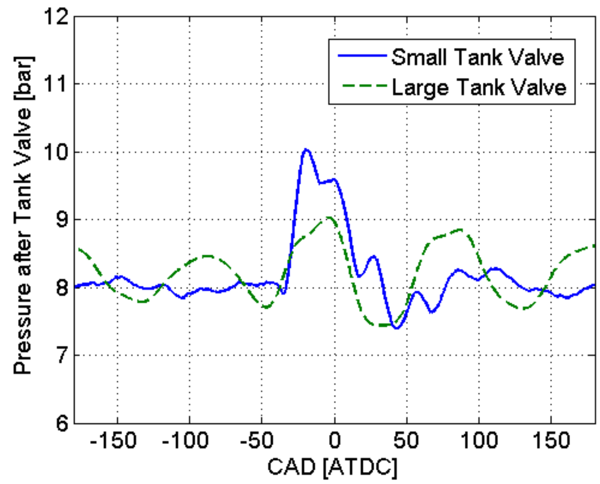

Figure 26 Pressure after the tank valve for two different tank valve diameters

AIR-MOTOR MODE

The AM can, as CM, mainly be executed in three ways – in regards to efficiency, power, or a combination of both. Achieving highest air-motor efficiency is done by a feedback control of both the tank valve and the intake valve. The tank valve should open at TDC and TankVC should be set in such way that the pressurized air is expanded to atmospheric pressure at BDC. The intake valve should open at BDC and IVC should be set in such way that when the piston reaches TDC, the inducted air is compressed to the same level as the tank pressure level.

In order to accelerate the vehicle more rapidly compared to the high-efficiency case, high power is needed. This can be achieved by prolonging the tank valve duration compared to the optimal timing and thereby induct more compressed air which increases the work done on the piston. Highest air-motor power is achieved with TankVC at BDC or shortly before. The inlet valve should be controlled in the same manner as with the high-efficiency method.

In a real application, a combination of the previously explained methods will be utilized. For instance, as long as the driver presses the gas pedal moderately, the AM will be operated at highest efficiency or close to it. As the driver continues to press the pedal towards its end position, the AM operation will drift away from highest efficiency towards maximum air-motor power.

This study focuses mainly on the first method, namely achieving as high AM efficiency as possible.

<u>Optimizing the air-motor mode</u>

An attempt to use the polytropic compression law, in order to achieve a proper valve strategy during AM, has been done. TankVC is controlled in such way that at a certain tank pressure, a proper closing angle is calculated with the help of the polytropic relation. Also the IVC is calculated in a similar way. TankVO and IVO are set to a constant value of 0 CAD ATDC and 180 CAD BTDC, respectively. TankVC and corresponding positive IMEP obtained with this method can be seen in Figure 27. These results are quite poor compared to the results shown by Trajkovic et al [1], where IMEP levels of 4 bars have been shown with constant valve timings. The reason is that, as stated before, the specific-heat ratio depends on the heat losses and setting this ratio to a constant value introduces some errors to the valve control algorithm. Also, the polytropic relation doesn't take the pressure losses over the tank valve into account, and therefore the TankVC will be chosen closer to TDC than what would be optimal.

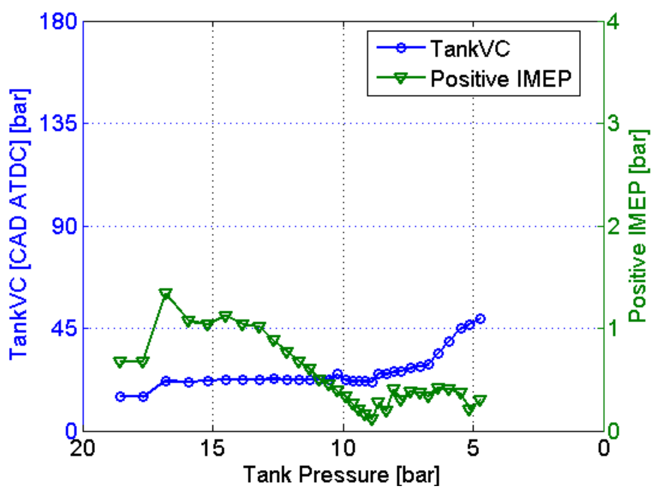

Figure 27 TankVO and corresponding positive IMEP during AM as a function of tank pressure for the large tank valve setup at an engine speed of 600 rpm. TankVO timings are based on the polytropic relation.

In order to optimize the AM, there is a need for a method that can help finding the optimal valve timings. The steady-state method, used for optimizing the CM, cannot be used in order to optimize the AM, since there is no charging of the pressure tank during AM and thereby a steady-state tank pressure cannot be achieved.

A method for finding the optimal IVC has been developed. The idea is to vary IVC and thus the corresponding peak cylinder pressure will also be varied. In that way a map, containing IVC as a function of peak cylinder pressure can be created.

In order to find the optimal TankVC, results from the CM optimization, shown in Figure 11, have been used. During the compression stroke in CM, the atmospheric air in the cylinder is compressed to the same pressure level as the tank pressure level before the tank valve opens. In AM, the procedure should be the opposite. The pressure level in the cylinder is the same as the tank pressure level when the tank valve is supposed to close. The pressurized air is then expanded to the atmospheric pressure level. Thereby the results obtained during CM optimization can, with some modification, be used to control the valve during AM. In order to fit the results from CM to AM, some tuning of the valve timings had to be done.

Figure 28 shows the final results from AM testing where both IVC and TankVC have been optimized with the methods described above. Comparing the results from Figure 28 with the results from Figure 11 indicates that the curves bear a resemblance to one another. But there are some differences, mainly for the TankVC. For instance, at a tank pressure of 20 bars, the results in Figure 28 indicates a TankVC at approximately 33 CAD ATDC, while the results in Figure 11 indicates a TankVO at approximately 38 CAD BTDC. The difference of 6 CAD is due to the fact that IVC during AM is chosen in such way that the peak in-cylinder pressure is lower than the tank pressure when the tank valve opens, which can be seen in Figure 29. Thereby, the deficit in the pressurized air supply line is compensated for.

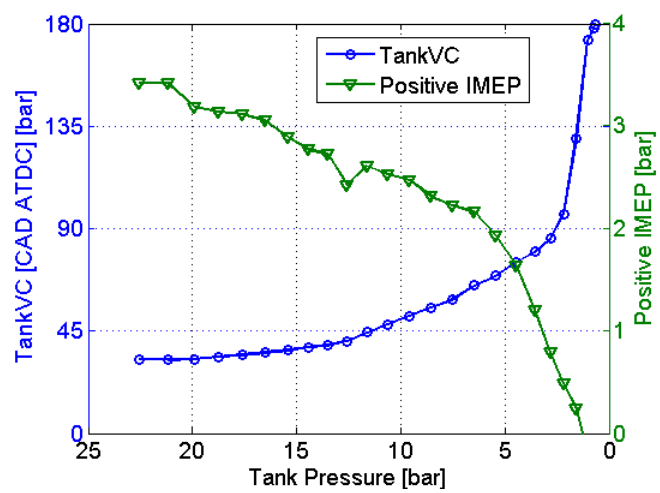

Figure 28 Optimized TankVC and corresponding positive IMEP during AM as a function of tank pressure for the large tank valve setup at an engine speed of 600 rpm

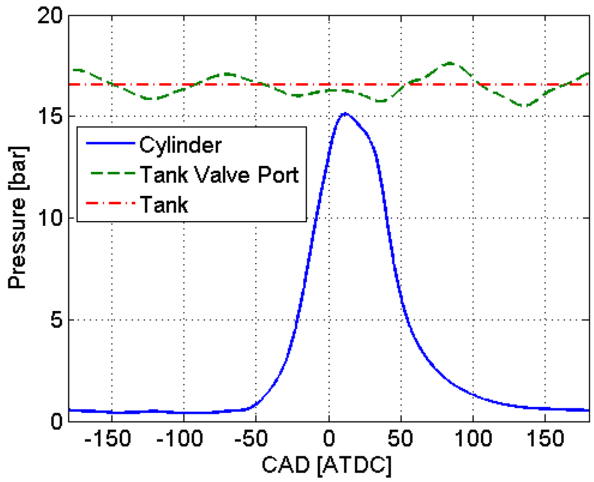

Figure 29 Illustration on the difference between the tank valve port pressure and the cylinder pressure at TankVO during AM operation

Figure 30 shows positive IMEP obtained with two different tank valve setups and valve strategies during AM operation. The small tank valve curve has been obtained with the constant valve timings displayed in Table 4. The large tank valve curve has been obtained with the optimal valve timings shown in Figure 28. The starting tank pressure is about 20 bars in both cases. It can be realized that, an increase in valve head diameter together with optimal valve timings, has a large impact on the AM operation. This will in turn lead to a considerable increase in the AM total efficiency. The reason why IMEP for the large tank valve setup is much higher throughout the whole test compared to the small tank valve setup, is that the pressurized air is used in a much more efficient way. A larger tank valve diameter contributes to less pressure losses over the tank valve and an optimized valve control strategy contributes to a more efficient use of the pressurized air, and together they contribute to a higher positive IMEP.

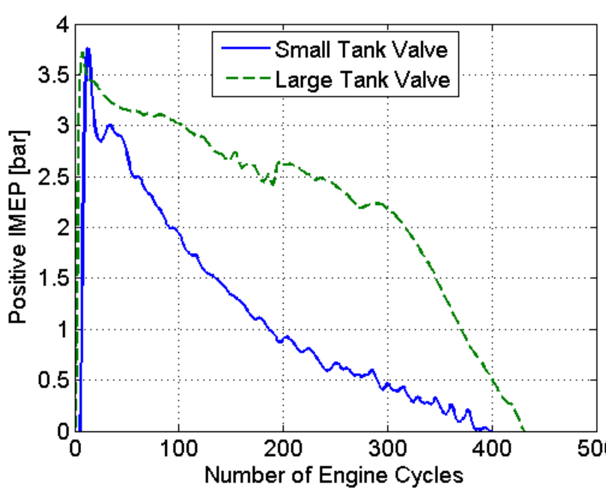

Figure 30 Positive IMEP for two different tank valve setups and valve strategies as a function of engine cycle number at an engine speed of 600 rpm during AM operation

Table 4 Valve timing used for the small tank valve setup during AM operation

IVO [CAD BTDC]	180
IVC [CAD BTDC]	0
TankVO [CAD ATDC]	-5
TankVC [CAD ATDC]	40

Figure 31 shows the tank pressure decrease for the small tank valve setup and large tank valve setup, obtained during the test displayed in Figure 30. With the large tank valve, the tank pressure decreases almost linearly. With the smaller valve, the tank pressure has a higher rate of discharge in the beginning, but after about 150 cycles the discharge rate starts to decrease. The reason for this behavior is that the valve timings with the small tank valve are set to a constant value. This means that in the beginning the tank valve duration is much longer than the optimal tank valve duration, and thereby the cylinder will be filled with more compressed air, thus leading to a higher rate of discharge of the tank pressure for this case. After a while, as the tank pressure decrease, the duration becomes too short and the cylinder is filled with less air than is optimal and thereby the tank pressure discharge will decrease.

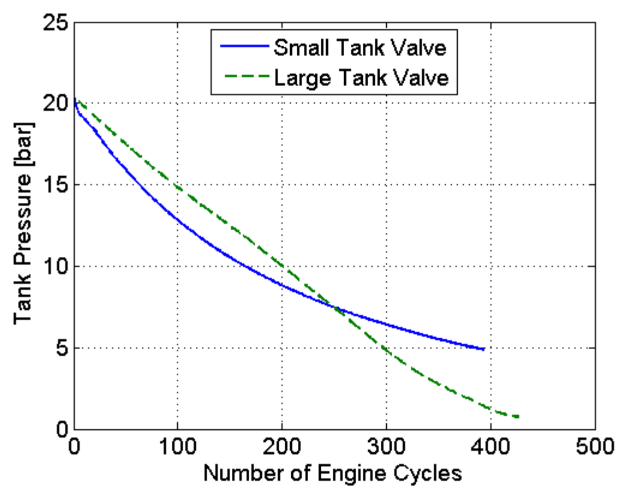

Figure 31 Mean tank pressure for two different valve geometries at an engine speed of 600 rpm

Figure 32 and Figure 33 illustrates PV-diagrams for both tank valve setups at tank pressures 16.5 bars and 6.5 respectively. There are evident differences in peak cylinder pressure between the small tank valve setup and the large tank valve setup in both figures. The reason is that the flow over the small tank valve will become chocked due to a very restricted flow area. With the larger tank valve, the flow area is increased more than three times compared to the small tank valve flow area and therefore the threshold for choked flow has been raised. Another issue with constant tank valve timing, which can be seen in Figure 33, is that one portion of the PV-curve indicates negative IMEP. Too short tank valve duration at a certain tank pressure will lead to an expansion of the inducted pressurized air below atmospheric pressure thus generating vacuum. Generation of vacuum is an energy consuming process, at least in this case. Since the inlet valve opens at BDC, the vacuum is canceled by the induction of fresh air into the cylinder and thereby the vacuum created cannot be used as an upward-acting force on the piston as it moves towards TDC.

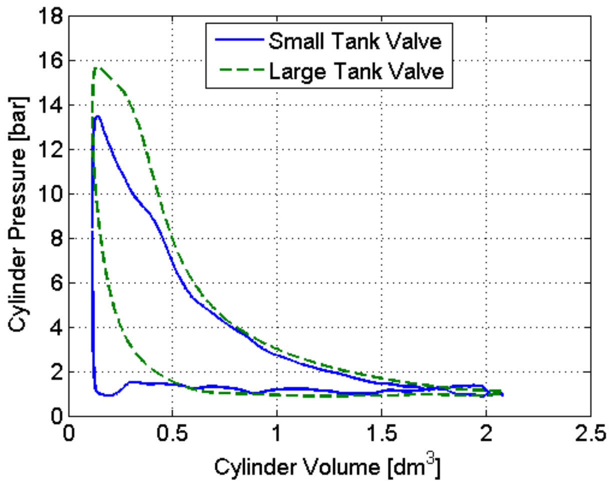

Figure 32 PV-diagram for two different tank valve geometries at a tank pressure of 16.5 bars and an engine speed of 600 rpm

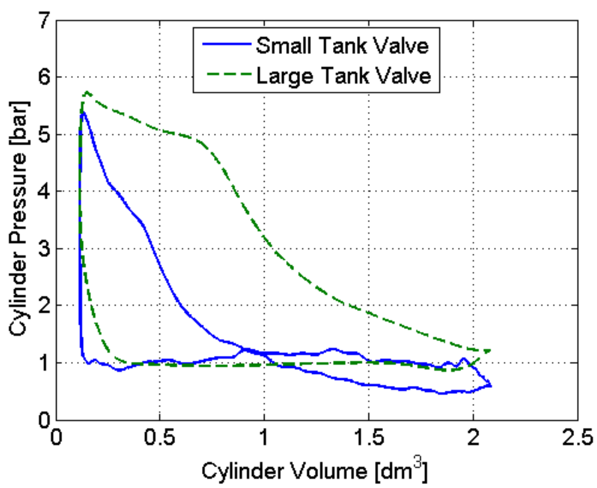

Figure 33 PV-diagram for two different tank valve geometries at a tank pressure of 6.5 bars and an engine speed of 600 rpm

During AM operation, the intake valve is controlled in such way that, the peak in-cylinder pressure should ideally be equal to the tank pressure. The reason is that in this way there will be a smooth filling of the cylinder when the tank valve opens. If the intake valve would close near TDC, the in-cylinder pressure at TDC would be close to atmospheric, and thus the compressed air from the pressure tank would rush into the cylinder as the tank valve opens. Since a rush of compressed air into the cylinder would contribute with a higher air flow velocity compared to a smooth filling of the cylinder, the pressure drop over the vale, which increases with speed, would increase. Figure 34 shows the results obtained both from a test with constant IVC and from a test with open-loop controlled IVC. With the intake valve closing near TDC, positive IMEP reaches a level almost two bars higher than with the open-loop controlled intake valve. The reason behind this behavior is that, with IVC set to TDC, compression of inducted air, which is an energy consuming process, is eliminated.

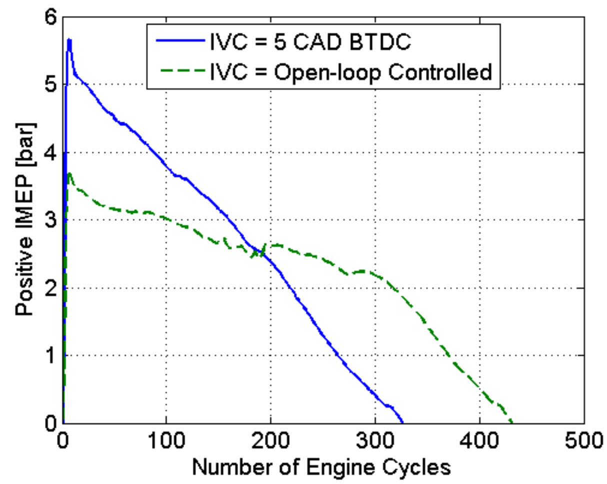

Figure 34 Positive IMEP during AM, solid line represents an AM test run with constant IVC while the dashed line represents an AM test run with IVC open-loop controlled

In Figure 34 the rate of decrease for positive IMEP is higher with constant IVC than with open-loop controlled IVC. This can be explained with the help of Figure 35, which shows the tank pressure as a function of engine cycles from the tests shown before in Figure 34. It can be seen that the rate of discharge is greater for the test with constant IVC, compared to the test with feedback controlled IVC. This implies that, the strategy with constant IVC, which leads to a rush of compressed air into the cylinder, contributes to higher compressed air consumption due to pressure losses over the tank valve.

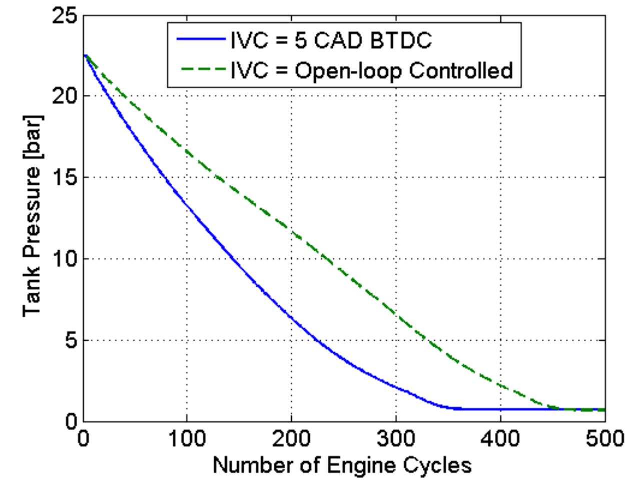

Figure 35 Tank pressure during AM, solid line represents an AM test run with constant IVC while the dashed line represents an AM test run with IVC open-loop controlled

Figure 36 shows torque for both constant IVC and open-loop controlled IVC. The result is quite impressive for a 2-litre engine operating without any combustion. The reason behind the large difference in maximum torque between the curves is that due to that rushing compressed air flow hits the piston with a huge force when the tank valve opens. With feedback controlled IVC, compressed air enters the cylinder smoothly and therefore no shock force will be generated.

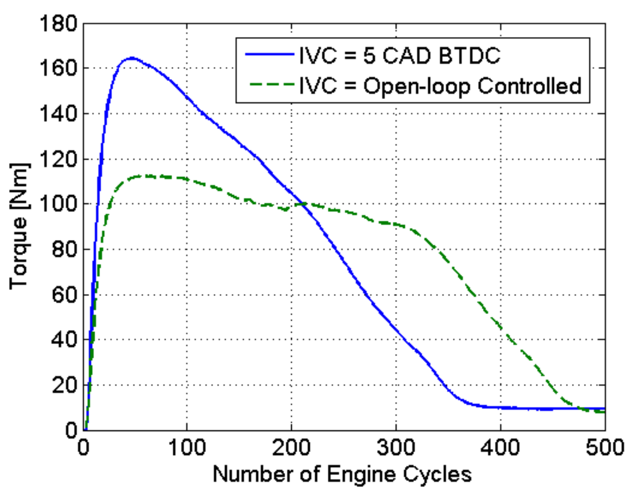

Figure 36 Torque during AM for both constant IVC timing and open-loop controlled IVC at an engine speed of 600 rpm

Figure 37, Figure 38 and Figure 39 illustrates PV-diagrams for both intake valve strategies at tank pressures 16.5 bars and 6.5 respectively. The pressure curve, obtained with constant IVC, has an overshoot at TDC compared to the curve obtained with open-loop controlled IVC. The reason is that as the compressed air starts to rush into the cylinder, a pressure wave is generated which propagates though the cylinder and out again through the open tank valve. A close-up of the PV-diagrams in Figure 37 is illustrated in Figure 38. It can clearly be seen that with constant IVC, the cylinder pressure overshoots at first and then decreases bellow the cylinder pressure obtained with open-loop controlled IVC. This indicates that there is a pressure wave propagating through the cylinder.

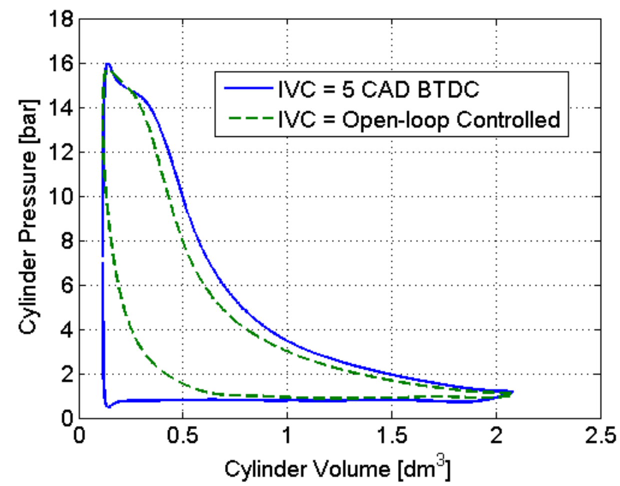

Figure 37 PV-diagram obtained during AM at a tank pressure of 16.5 bar and at an engine speed of 600 rpm, solid line represents an AM test run with constant IVC while the dashed line represents an AM test run with IVC open-loop controlled

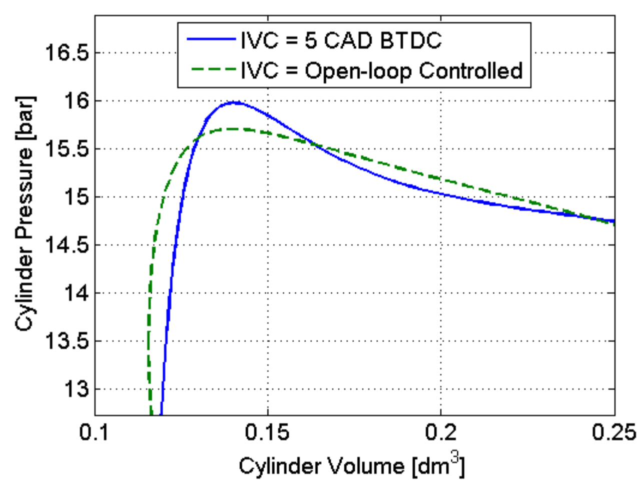

Figure 38 Close-up of the PV-diagrams at a tank pressure of 16.5 bars shown in Figure 37

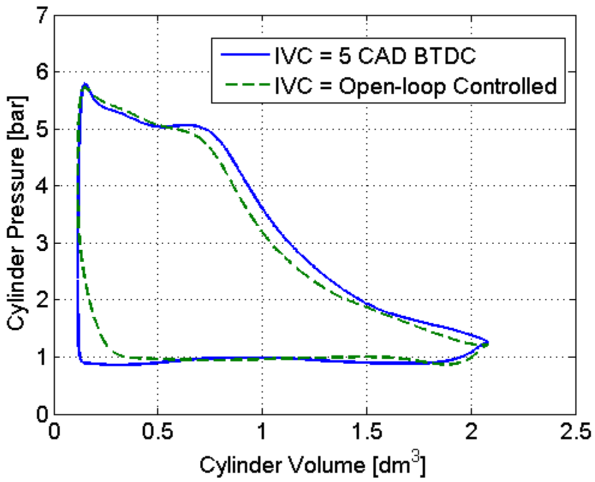

Figure 39 PV-diagram obtained during AM at a tank pressure of 6.5 bar and at an engine speed of 600 rpm, solid line represents an AM test run with constant IVC while the dashed line represents an AM test run with IVC open-loop controlled

REGENERATIVE EFFICIENCY

In order to estimate the potential of the pneumatic hybrid, and the possibility to compare different tests with each other, a regenerative efficiency has to be calculated. The regenerative efficiency is the ratio between the energy recovered during AM and the energy consumed during CM. It can also be defined as the ratio between positive and negative IMEP:

$$\eta_{regen} = \frac{IMEP_{+}}{IMEP_{-}} \qquad (2)$$

Table 5 show the regenerative efficiency with different tank valve setups and valve strategies at three engine speeds. The regenerative efficiency obtained with the small tank valve setup is retrieved from [1]. It has a maximum value of 33% at 900 rpm. The reason why the efficiency is higher at 900 rpm than at 600 rpm, as explained by Trajkovic et al. [1], is that the unoptimized feedback controller by coincidence is better suited for the case at 900 rpm than at 600 rpm. The regenerative efficiency for the large tank valve setup indicates that with optimized tank valve timing, the maximum efficiency occurs at 600 rpm and decreases with increasing engine speed. A tremendous improvement has been achieved while switching from the small tank valve setup to the large tank valve setup. The improvement mainly depends on a larger tank valve head diameter and optimized tank valve timing during AM. A change in inlet valve strategy from constant IVC to open-loop controlled IVC, increases the regenerative efficiency further.

The pneumatic tank valve actuator consumes energy in the form of compressed air from the pressure tank. Since its energy consumption decreases the total energy stored in the pressure tank, it has to be seen as energy losses. These losses have automatically been taken into account in the calculation of regenerative efficiency. This is only valid for the large valve setup, since the pneumatic tank valve used in the small tank valve setup has been feed with compressed air generated from an external source. This means that the regenerative efficiency calculated for the large tank valve setup, is lower than it would be if the pneumatic valve actuator energy losses were excluded from the calculation.

Table 5 Calculated total regenerative efficiency for different tank valve setups and valve strategies at three different engine speeds

	η_{regen}		
Engine speed	600	900	1200
Small tank valve setup	32	33	25
Large Tank valve setup, constant IVC during AM	44	40	37
Large tank valve setup	48	44	40

It should be noticed that the regenerative efficiency, described in this paper is actually an indicated efficiency, apart from the included pneumatic valve actuator energy losses. This means that a real vehicle cannot utilize the energy in the same extent due, among other things, to engine and driveline friction losses, which will lead to a lower regenerative efficiency.

CONCLUSION

A pressure compensated tank valve with a valve head diameter of 28 mm has been tested and evaluated. The pressure compensation has been achieved with an in-house developed pneumatic valve spring arrangement mounted on the cylinder head. The evaluation has shown that there exists some trouble with the pneumatic valve spring concerning tank valve actuation, mostly because the pneumatic valve actuator is driven by compressed air from the pressurize air. This can be resolved by the use of externally generated pressurized air in order to drive the pneumatic tank valve actuator.

Results from tests done with the pressure compensated tank valve have been compared with results from tests where a tank valve with a valve head diameter of 16 mm has been used. The results indicate that the increase in valve diameter reduces the pressure drop over the tank valve, contributing to a higher regenerative efficiency.

Feedback control of tank valve timing based on the polytropic compression law is not suitable for the pneumatic hybrid. Instead, optimal tank valve timings can be obtained from steady-state engine testing.

Optimization of both tank valve timing and inlet valve timing for CM and AM contributes to an increase of the regenerative efficiency.

Test during AM with constant IVC versus open-loop controlled IVC, shows that a much higher torque can be achieved with constant IVC due to the blowdown of compressed air. The drawback is that such AM operation lowers the AM efficiency due to higher pressure losses.

The total regenerative efficiency has been increased from 33% with the small tank valve setup to 48% with the large tank valve setup, primarily due to a larger valve head diameter and optimal valve timing,

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NOMENCLATURE

AM: Air-motor Mode

ATDC: After Top Dead Centre

BDC: Bottom Dead Centre

BMEP: Brake Mean Effective Pressure

BTDC: Before Top Dead Centre

CAD: Crank Angle Degree

CM: Compressor Mode

η_{regen}: Regenerative efficiency [%]

κ: polytropic exponent [-]

ICE: Internal combustion Engine

IMEP: Indicated Mean Effective Pressure

IVC: Inlet Valve Closing

IVO: Inlet Valve Opening

p: In-cylinder pressure [bar]

TankVO: Tank Valve Opening

TankVC: Tank Valve Closing

TDC: Top Dead Centre

RPM: Revolutions Per Minute

V: Cylinder volume [m^3]