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Published in:
International Journal of Powertrains

DOI:
10.1504/IJPT.2013.052657

2013

Link to publication

Citation for published version (APA):
Trajkovic, S., Tunestål, P., & Johansson, B. (2013). A study on compression braking as a means for brake energy recovery for pneumatic hybrid powertrains. *International Journal of Powertrains*, *2*(1), 26-51. https://doi.org/10.1504/IJPT.2013.052657

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A Study on Compression Braking as a Means for Brake Energy Recover for Pneumatic Hybrid Powertrains

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Abstract:

Hybrid powertrains have become a very attractive technology over the last 10 years due to their potential of lowering fuel consumption and emissions. A large portion of all manufacturers worldwide are now focusing on hybrid electric powertrains, which are the most common hybrid powertrains of today. However, in recent years a new technology has emerged, namely the pneumatic hybrid powertrain. Although the concept is still only under research, it has been tested in various laboratories with promising results.

The basic idea with pneumatic hybridization is to take advantage of the otherwise lost energy when braking and convert it to potential energy in the form of compressed air by using the engine as a two-stroke compressor. The compressed air is stored in a pressure tank and when needed it is used to propel the vehicle through expansion in the engine cylinders. Previous studies have shown that more than 40% of the braking energy can be regenerated and used as useful work, leading to a lower fuel consumption compared to a conventional vehicle.

Present paper deals with an in-depth investigation of the parameters influencing the performance of the compressor braking mode. Also, a control strategy has been adopted, that controls the pneumatic hybrid powertrain load during compressor mode at different tank pressures.

The results show that the influence of parameters such as valve head diameter, tank valve opening and closing, and inlet valve opening have a considerable effect on the performance of the compressor mode. Further, the results show that the chosen controller strategy performs well with an ability to control the compressor mode load on an almost cycle-to-cycle basis. The results shown in present study are an important step in the development of the pneumatic hybrid powertrain.

Keywords: hybrid, compressor, pneumatic, VVA

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1 Introduction

The modern world of today relies heavily on combustion engine powered vehicles as a means of transportation. The toxic exhaust gases emitted by these vehicles are a heavy the load on the environment. In an effort to minimize the air pollution manufacturers worldwide are looking for alternative means of transportation with less impact on the environment. There are a couple of solutions under development today, such as hydraulic hybrids, fuel cell hybrids, flywheel hybrids and finally the electric hybrid which is probably the most developed today. Almost all manufacturers are also working on electric vehicles as a "zero emitting" alternative. All these solutions have a couple of things in common. They offer great reductions in fuel consumption and thereby emissions from exhaust gases. However, they are all expansive, due to their complexity, use of exotic materials, extra weight, etc. A way to avoid the complexity and to lower the cost is the introduction of a new type of hybrid vehicle. The concept in question is the Pneumatic Hybrid. In contrast to the other hybrid solutions, the pneumatic hybrid is a relatively simple solution utilizing only the ICE already existent in conventional vehicles as propulsion source. Instead of expansive batteries with a limited life-cycle, the pneumatic hybrid utilizes a relatively cheap pressure tank to store energy.

The pneumatic hybrid operates in a way similar to the electric hybrid. During deceleration of the vehicle, the engine is used as 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. This is also known as the Compressor Mode (CM). 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. This is also known as the Air-Motor Mode (AM). Another feature with the pneumatic hybrid is that it enables stop/start functionality, which means that the engine can be shut off during a full stop, thus eliminating unnecessary fuel consumption and emission of exhaust gases during idle (Schechter, 2000; Tai et al., 2003). The pneumatic hybrid also offers elimination of the "turbo-lag" in turbocharged engines by supercharging the engine with pressurized air (Vasile, et al., 2006; Dönitz, et al., 2009). In present paper focus is on pneumatic hybridization for heavy-duty engine applications, however pneumatic hybridization can also be applied to light-duty engine application (Schechter, 1999; Higelin, et al., 2002).

The purpose of the study described in present paper is to investigate and understand the influence of some parameters on pneumatic hybrid powertrain performance. Previous experimental studies (Trajkovic, et al., 2007, 2008) have shown that different parameters, like for instance tank valve head diameter and valve timings, play an important role in the performance of the pneumatic hybrid powertrain. In present paper, a more in-depth analysis of different parameters and their influence on the performance of the pneumatic hybrid is presented. Due to the extensive amount of information extracted from the above mentioned analysis only the results from compressor mode operation will be presented in present paper.

2 Pneumatic Hybrid

Pneumatic hybridization introduces new operating modes is addition to the conventional internal combustion engine (ICE) operation. Mainly, there are four different modes. Compressor Mode is the mode during which the compressed air is generated. Air-Motor

Mode is the mode during which the compressed air is used in order to propel the vehicle. During Air-Power Assist Mode (APAM) the compressed air is utilized for supercharging purposes. Finally, the stop/start mode (SSM) is the mode during which the engine can be completely shut off as a means of elimination of idle losses. Since the present paper mainly focuses on compressor mode operation, a thorough description of this mode only will be given. More information about the other modes of operation can be found in Tai, et al. (2003) and Trajkovic, et al. (2007).

2.1 Compressor Mode

During compressor mode 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, which is then stored in a pressure tank. The operating cycle of compressor mode can be explained with references to Figure 1. The numbers in bracket refers to the numbers in the PV-diagram.

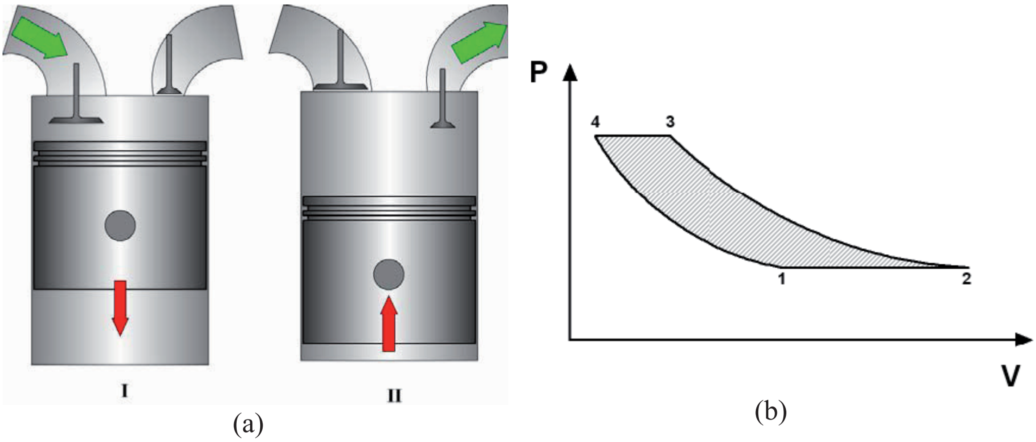


Figure 1 (a) Illustration of Compressor mode, I – intake of fresh air, II – Compression of air and pressure tank charging, (b) Illustration of the idealized PV-diagram of one Compressor Mode cycle

I. *Intake stroke.* During the compressor mode, inlet valve opening (IVO) occurs a number of crank angle degrees (CAD) after top dead centre (TDC) and brings fresh air to the cylinder (1). At the end of the intake stroke, as the piston reaches bottom dead centre (BDC), the inlet valve closes (2).

II. *Compression stroke.* The moving piston starts to compress the air trapped in the cylinder after BDC and the tank valve (the valve which controls the flow of air between the cylinder and the pressure tank) opens somewhere between BDC and TDC (3), depending on how much braking torque is needed. For instance a very early tank valve opening (TankVO) means that there will be a blowdown of pressurized air into the cylinder, and the piston has to work against a much higher pressure, thus a higher braking torque is achieved. The pressure tank is charged with compressed air as long as the tank valve is open. The tank valve closes shortly after TDC (4). At this point the cylinder is filled with compressed air at the same pressure level as the air in the pressure tank. As the piston moves towards BDC, the compressed air expends and the intake valve opens (1) when ambient pressure is reached in the cylinder.

2.2 Variable Valve Actuation

The description of the compressor mode above implies a need for a variable valve actuating (VVA) system. The VVA system used in the study described in present paper is an electro pneumatic valve actuating (EPVA) system. The EPVA 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, i.e. valve timing, duration and lift height can be controlled independently of each other. A thorough description of the EPVA system can be found in the work of Trajkovic, et al. (2006) while Ma, et al. (2006) presents a dynamic model of the EPVA system.

2.3 Tank Valve

In order to run the engine as a pneumatic hybrid powertrain, a pressure tank has to be connected to the engine. There is also a need for a valve dedicated to control the flow of air into and out from the pressure tank. There are numerous suggested solutions to how such a way should be constructed. Tai, et al. (2003) describes an intake air switching system in which one inlet valve per cylinder is feed by either fresh intake air or compressed air form the pressure tank. Andersson, et al. (2005) describes a dual valve system, where one of the intake ports has two valves, one of whom is connected to the pressure tank. Dönitz, et al. (2009), presents a solution where an extra valve is introduced to the cylinder head in addition to the conventional valve train. In the study described in present paper, a different and simpler solution has been chosen where one of the existing inlet valves has been converted to operate as a tank valve. Since the engine used the study has separated 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 due to limited air flow into the cylinder during the intake stroke. Another drawback is that the ability to generate and control swirl for good combustion will be reduced.

3 Load Control during Compressor Mode

An important aspect of the pneumatic hybrid concept is its ability to control the amount of braking torque at a specific time. A desired torque should be achievable whatever the pressure level in the tank is. For this purpose, a control strategy has been adopted, that controls the pneumatic hybrid powertrain load during compressor mode and air-motor mode at different tank pressures. Figure 2 illustrates the closed-loop controller for compressor mode load control. The structure consists of a feed-forward filter and a PID controller. The feed-forward filter contains data about valve timings at different loads acquired from steady-state experiments. It takes the measured load from previous cycle as argument and based on this, it outputs proper steady-state valve timings at current

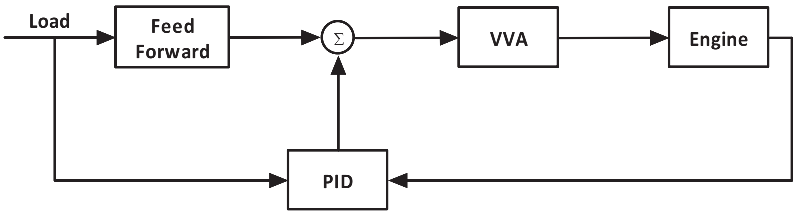


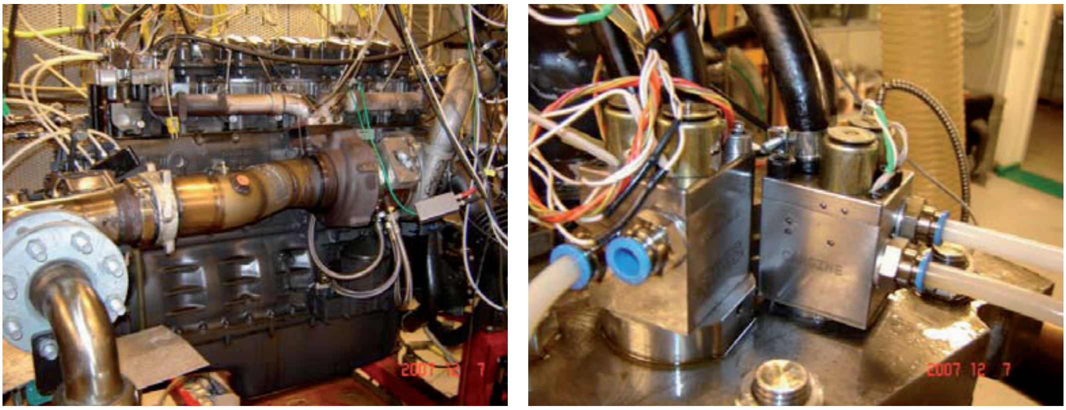
Figure 2 The closed-loop control system for compressor mode load control. Proper valve timings are calculated as the sum of the feed forward term and the output from the PID controller.

load. The task of the PID controller is to minimize the response time during load steps and to eliminate eventual stationary errors. The PID controller used in the study in present paper can by described by the following equation:

$$u(t) = K_p \cdot e(t) + K_i \cdot \int_0^t e(\tau)d\tau + K_d \cdot \frac{d}{dt}e(t)$$

4 Experimental Setup

The engines used in the study presented here were an in-line six cylinder Scania D12 and an in-line six-cylinder Scania D13 Diesel engine. In these setups one of the cylinders was modified for pneumatic hybrid operation, while the remaining cylinders were intact, see Figure 3(a).



(a) (b)

Figure 3 (a) The modified Scania D12 Diesel Engine. The cylinder modified for pneumatic hybrid operation is to the left in the picture. (b) The pneumatic valve actuators mounted on the Scania cylinder head

The standard Scania D12 engine uses a piston with a bowl in its crown. In the pneumatic hybrid project the standard piston has been exchanged for a flat piston in order to increase the piston clearance and thus avoid any valve-to-piston contact when using the pneumatic VVA system. The Scania D13 engine also uses a piston with a bowl in its crown. However, in this setup, the only modification done to the piston was a valve pocket for the tank valve. The reason is that the engine will be used for fired operation later in the pneumatic hybrid project and therefore there is a need for the bowl to achieve satisfying diesel combustion. Some engine specifications can be found in Table 1.

Table 1 Geometric properties of the Scania D12 and D13 diesel engine

	Scania D12	Scania D13
Displaced Volume (x6)	1966 cm3	2100 cm3
Bore	127.5 mm	130 mm
Stroke	154 mm	160 mm
Connecting Rod Length	255 mm	
Number of Valves	4	
Compression Ratio	18:1	17.2
Piston type	Flat	Bowl
Inlet valve diameter	45 mm	
Tank valve diameter	28	
Piston clearance	7.3 mm	0.9 mm

The standard camshaft has been removed from both engines, and the engine valves are instead actuated by the pneumatic VVA system described earlier. Figure 3(b) shows the pneumatic valve actuators mounted on top of the Scania D12 cylinder head. The same setup has been used on the Scania D13 engine. In Table 2, some valve operating parameters are shown. The maximum valve lift height has been limited to 7 mm in order to avoid valve-to-piston contact and thus prevent engine failure. The standard valve springs have been exchanged for less stiff springs in order to reduce the energy required to operate the valves.

Both engines have 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 study presented here because no fuel was injected and thus there was no need for exhaust gas venting.

The pressure tank used in the study presented here was an AGA 50 liter pressure tank normally used for storage of different types of gases at pressures up to 200 bar. It is connected to the cylinder head by metal tubing. The size of the tank in present study was selected based on availability rather than optimality.

Table 2 Valve operating 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	340 N
Maximum valve lift	7 mm

5 Results

The Compressor Mode can be done mainly in three different ways: with emphasis on maximum efficiency, maximum brake torque generation and lastly a combination of both. The first two subsections focus on CM operation with maximum compressor mode efficiency, both with theoretically calculated valve timings and with valve timings retrieved from experimental data. The third subsection focuses mainly on the parameters influencing the compressor mode performance while the subsection deals with closed-loop control of the load during compressor mode.

5.1 Compressor Mode Operation with Theoretically Calculated Valve Timings

One way of running CM operation with focus on maximum efficiency is to use theoretically calculated valve timings. As described earlier in present paper, according to idealized CM operation the TankVO should occur when the pressure inside the cylinder equals the pressure in the tank. This means that there will be no blow down or over-compression of pressurized air. In this part of the study, the TankVO was controlled by an open-loop controller based on the polytropic compression law:

$$p_2 = p_1 \left(\frac{V_1}{V_2}\right)^k$$

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 maximum 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. By comparing V_2 to a precalculated CAD-resolved volume trace, proper valve timings can be determined. The same strategy can also be used to determine TankVC and IVO.

The advantage with this strategy is that the valve timings are determined in a very simple way and it is easy to implement in the control program. However, there are some drawbacks with this strategy. On is that the polytropic exponent, κ, depends on heat losses and setting it to a constant value introduces some errors in the TankVO control algorithm. Another drawback is that the strategy doesn't take the dynamics of the system, such as pressure wave propagation into consideration.

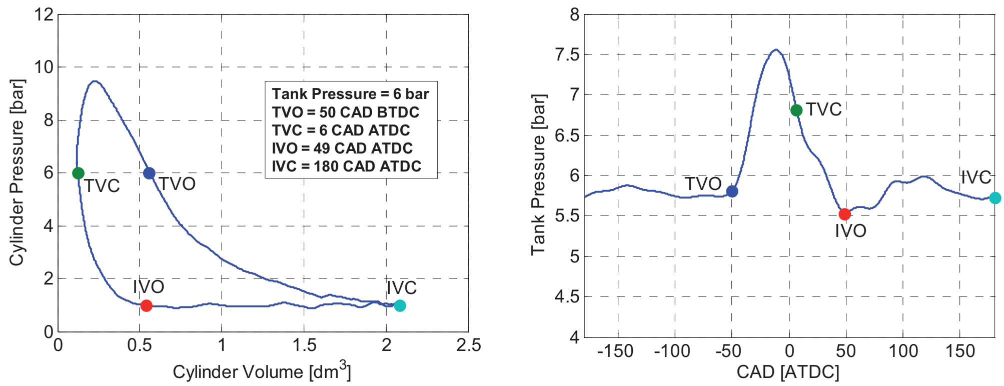


Figure 4 PV-diagram of the cylinder pressure (left) and tank pressure (right) of one CM cycle at a tank pressure of 6 bar and an engine speed of 600 rpm

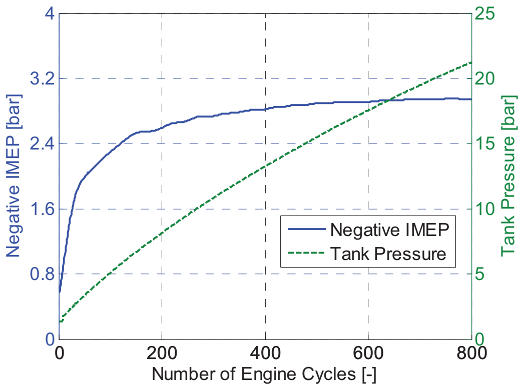


Figure 5 Negative IMEP and mean tank pressure during transient CM operation as a function of engine cycle number at an engine speed of 600 rpm

Figure 4 show a PV diagram of the in-cylinder pressure and tank pressure during one engine cycle. According to the idealized PV-diagram shown in Figure 1(b), the pressure between TVO and TVC should be constant. However, the PV-diagram in Figure 4 shows no such isobaric event during real engine testing. The reason is that the flow over the tank valve becomes very restricted which is also known as choked flow. This will therefore lead to an overshoot in in-cylinder pressure compared to the idealized conditions. This phenomenon highly depends on the size of the tank valve, tank pressure and engine speed, which will be demonstrated later on in present paper.

Figure 5 shows the generated negative indicated mean effective pressure (IMEP) and accumulated tank pressure during 800 consecutive engine cycles. The valve timings are controlled according to the strategy described earlier in current subsection. One thing that can be noted from the figure is that the negative IMEP at first increases rapidly but then it starts to move towards an almost constant value. The reason for this behaviour is the choice of valve timings and can be explained with the help of Figure 6. When the pressure in the tank is low, TankVO occurs early in the cycle. As the tank pressure starts to increase, TankVO moves away from BDC and towards TDC. At low tank pressures the, difference in TankVO for a given pressure difference, is much larger than at a high tank pressure for the same pressure difference. A large change in valve timing will lead to

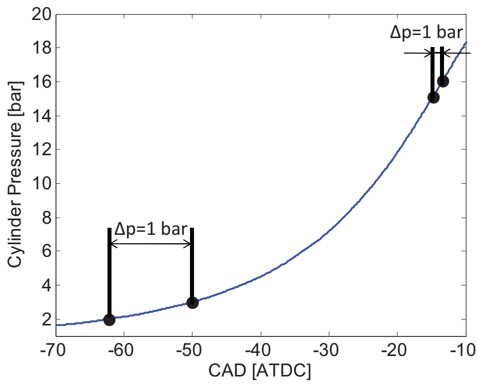


Figure 6 Illustration of how the change in CAD for a change in pressure by 1 bar changes with increasing pressure.

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a large change in IMEP while a small change in TankVO will only have a modest influence on IMEP.

5.2 Optimized Compressor Mode Operation

The strategy with theoretically calculated valve timings have some flaws as described above and this can lead to inaccurate valve timing when taking maximum compressor mode efficiency in consideration. In order to avoid this problem, proper valve timings can be determined experimentally. In present study a method to optimize the compressor mode will be presented. The main idea with this method is to find optimal valve timing at a given tank pressure and, in order to do that, the tank pressure needs to be constant throughout the entire testing interval (steady state). With a pressure relief valve connected to the tank, it is possible to control the pressure level in the system.

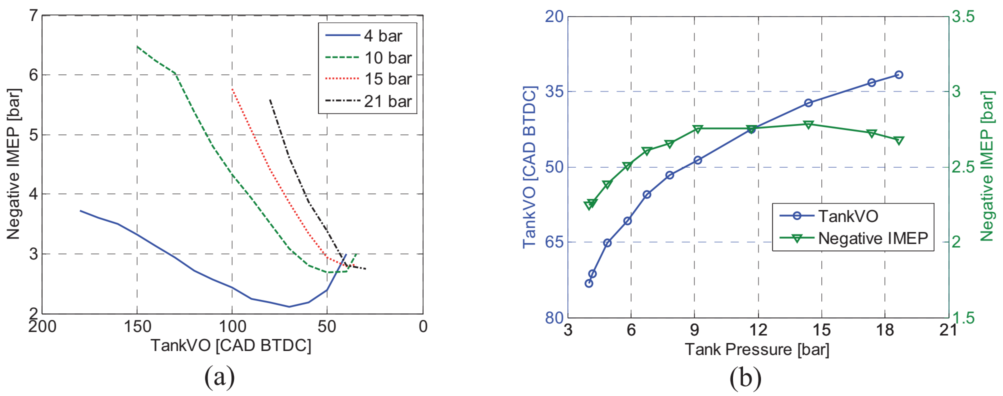


Figure 7 Steady-state optimization of the Compressor Mode at an engine speed of 600 rpm; (a) Negative IMEP obtained during steady-state CM as a function of TankVO for different tank pressures; (b) Optimal TankVO and corresponding negative IMEP for Compressor Mode operation as a function of tank pressure.

Figure 7(a) shows a TankVO optimization sweep at various steady-state tank pressures. It can clearly be seen how negative IMEP is affected by TankVO timing during optimization of CM. Figure 7(a) indicates that there is an optimal TankVO timing for every tank pressure when taking highest compressor mode efficiency in 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 that at any other point on the curve at a given tank pressure. If higher brake torque is needed, the efficiency has to be sacrificed The reason why negative IMEP increases 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 the pressure level 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. Trajkovic et al. (2009) demonstrated that if maximum braking torque is desired, the TankVO should open at BDC. In this way it was possible to achieve over 6000 Nm of braking torque for a 16-liter heavy-duty truck engine.

Figure 7(b) shows the how optimal TankVO and corresponding negative IMEP varies with increasing tank pressure. The reason why the negative IMEP decreases after a tank pressure of approximately 14 bar is most likely due to system dynamics in the form of pressure wave propagation through the system. A more in-depth look on this matter will be presented later in present paper.

In Figure 8, the results from transient compressor mode operation during 800 consecutive engine cycles can be seen. The results contain data from compressor mode operation with both theoretical valve timing and optimized valve timing. Initially, negative IMEP for the theoretical case is similar to the negative IMEP for the optimized case, but after about 300 engine cycles negative IMEP for the optimized case remains reasonably constant while negative IMEP for the theoretical case continues to increase throughout the rest of the experiment. Even though a higher efficiency has been achieved with the optimized case, the final tank pressure after 800 engine cycles is lower by 0.2 bar compared to the theoretical case. This behaviour can once again be explained by the dynamics of the system and a more thorough investigation of this occurrence will be presented later in present paper.

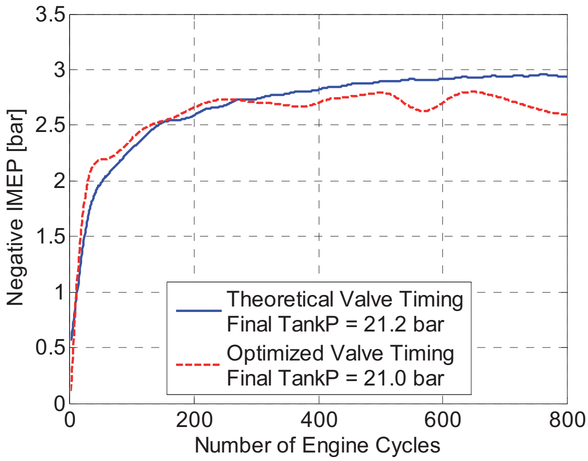


Figure 8 Comparison of negative IMEP for theoretical and optimized valve timings as a function of engine cycle number during CM operation.

5.3 Parametric Study of Compressor Mode

The main goal during compressor mode is always to, at any given situation maximize the charging of the pressure tank. In previous subsection it could be noticed that even though the valve timings were optimized with regards to maximum compressor mode efficiency, the tank pressure decreased compared to the less accurate theoretical approach. This leads to the conclusion that there are some underlying phenomena affecting the performance of the compressor mode. The purpose of current subsection is to investigate different parameters in order to understand their influence on the compressor mode performance. The relevant parameters which will be investigated are the tank valve diameter, tank valve opening and closing, and inlet valve opening. There are some more parameters influencing the performance of compressor mode, but their influence is small compared to the once mentioned above and they are therefore not dealt with here.

5.3.1 Tank Valve Diameter

The importance of proper tank valve diameter high efficiency compressor mode operation has been demonstrated by Trajkovic et al. (2008), where two different tank valves configurations with different valve diameters were tested. A small tank valve diameter will restrict the air flow into and out from the pressure tank with high pressure losses as a result. During compressor mode operation this will show as an overshoot in in-cylinder pressure. By maximizing the tank valve diameter the restriction of the flow can be held to a minimum. Figure 9 show how the in-cylinder pressure at two different engine speeds and a tank pressure of 10 bar. It can clearly be seen how the overshoot in in-cylinder pressure decreases with increasing tank valve diameter. Also the engine speed plays an important role here. At an engine speed of 600 rpm, the overshoot with the 16 mm tank valve is larger compared to the other two valve configurations but still at reasonable levels. However, as the engine speed increases to 1500 rpm, the in-cylinder pressure overshoots by more than 10 bar compared to the tank pressure for the 16 mm tank valve, which implies that the flow into the tank is extremely restricted. In Figure 10

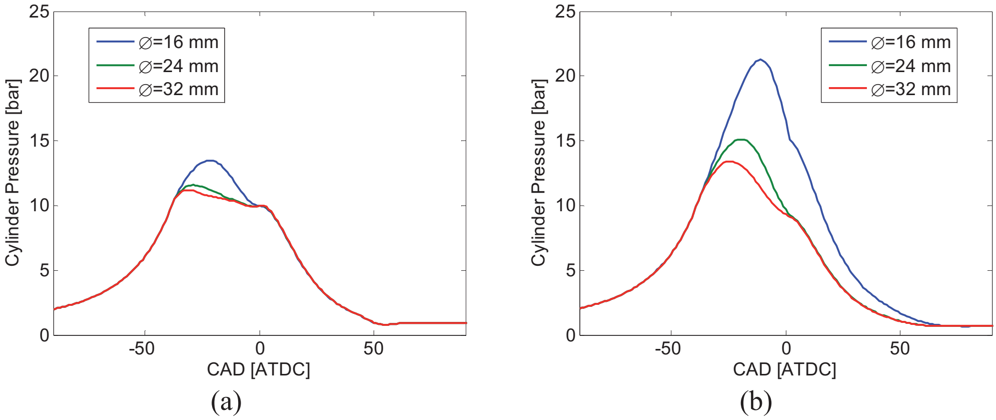


Figure 9 Cylinder pressure for various tank valve diameters at the engine speeds 600 rpm (a) and 1500 rpm (b). Tank pressure = 10 bar.

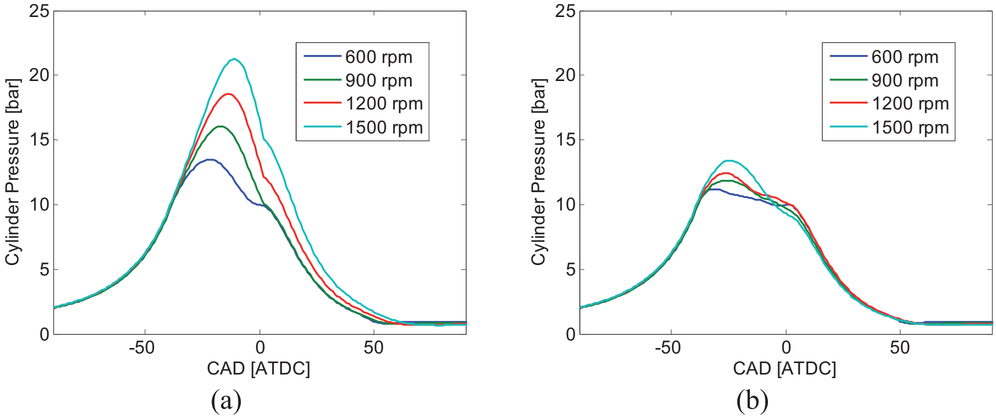


Figure 10 Cylinder pressure with a tank valve diameter of 16 mm (a) and 32 mm (b), respectively, at various engine speeds

it can be more clearly seen how the engine speed affects the overshoot in in-cylinder pressure for a 16 mm tank valve and a 32 mm tank valve, respectively.

A direct measure of the performance of the compressor mode is the mass flow of compressed air into the tank. The higher the mass flow is, the higher the increase in tank pressure will be. In Figure 11 the mass flow of compressed air into the tank for different engine speeds and tank valve diameters can be seen. At low engine speeds, the mass flow is almost constant independently of valve diameter. The reason is that, at low piston speeds, there will be enough time for the compressed air to flow into the tank, even though it is restricted for the smaller tank valve diameters. However at higher engine speeds the available time for venting the cylinder from compressed air decreases and it can clearly be seen that the tank valve diameter of 16 mm limits the mass flow severely compared to the other tank valve configurations. It is also very interesting to see that with the 32 mm tank valve, the mass flow becomes almost constant above 900 rpm, see Figure 11(b).

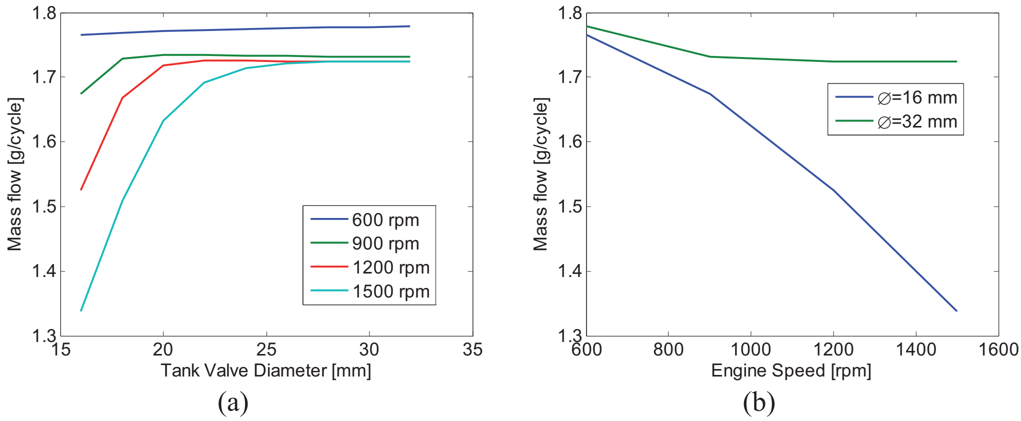


Figure 11 (a) Mass flow of compressed air into the tank as a function of tank valve diameter for various engine speeds, (b) Mass flow of compressed air into the tank as a function of engine speed for two different tank valve diameters. Tank pressure = 10 bar

In The results shown so far in present subsection has all been retrieved during steady-stat Compressor Mode operation. In Figure 12, the results from transient Compressor Mode operation during 800 consecutive engine cycles are shown. The results at 600 rpm are in analogy with what has previously been shown for steady-state operation. In Figure 11(a) the mass flow was almost independent of valve diameter at 600 rpm, and

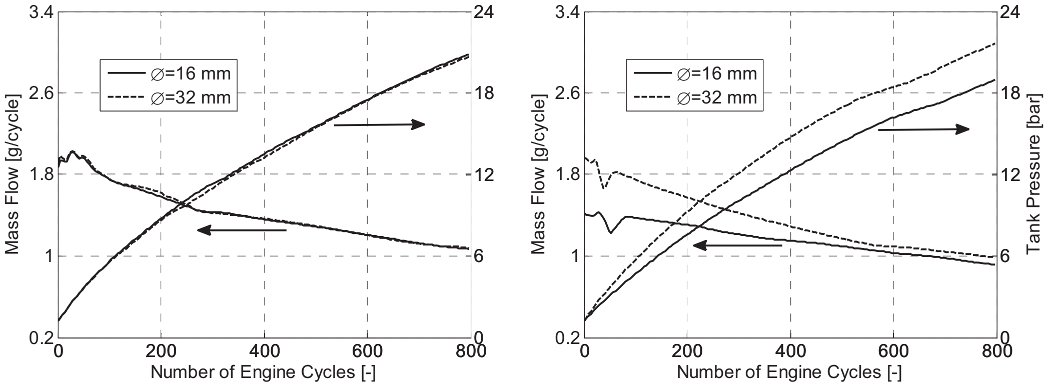
Table 3, IMEP and mass flow for two different valve diameters and two different engine speeds are presented. IMEP represents the energy that the engine consumes in order to compress the air to a specific pressure level and the mass flow represents the energy that goes into the pressure tank. For best performance, the energy consumed should be at a minimum while the energy transferred to the tank should be at a maximum. In other words, the goal is to lower the IMEP and in the same time maximize the mass flow of compressed air into the tank. At 600 rpm, it can be seen that with the 32 mm tank valve IMEP is lower and mass flow is slightly higher compared to the case with the 16 mm tank valve, thus the Compressor Mode performance will be better with the larger valve setup. At 1500 rpm, the difference between the two tank valve configurations becomes much higher, indicating the advantage of using a large tank valve diameter.

The results shown so far in present subsection has all been retrieved during steady-stat Compressor Mode operation. In Figure 12, the results from transient Compressor Mode operation during 800 consecutive engine cycles are shown. The results at 600 rpm are in analogy with what has previously been shown for steady-state operation. In Figure 11(a) the mass flow was almost independent of valve diameter at 600 rpm, and

Table 3 The influence of tank valve diameter and engine speed on IMEP and mass flow during Compressor Mode operation

Engine speed [rpm]	**Ø [mm]**	**IMEP [bar]**	**Mflow [g/cycle]**
600	16	2.49	1.77
	32	2.16	1.78
1500	16	3.25	1.34
	32	2.73	1.72

this behaviour can once again be seen during transient operation in Figure 12(a). At 1500 rpm the mass flow is comparable to what has been shown for steady state operation in Figure 11(a). One thing that requires attention is that even though the mass flow has decreased at1500 rpm compared to at 600 rpm, the tank pressure has increased. The explanation for



(a) (b)

Figure 12 Mass flow and tank pressure as a function of number of engine cycles for two different valve configurations during transient CM operation at an engine speed of 600 rpm (a) and 1200 rpm (b), respectively.

this behaviour is most likely that, at a higher engine speed there will be less time for heat transfer to occur between the hot compressed air and the system it flows through. This means that the temperature in the pressure tank will be higher compared to the case at 600 rpm, with higher pressure in the tank as a result.

5.3.2 Tank Valve Opening

The TankVO is probably the most important parameter influencing the performance of the Compressor Mode operation. It directly determines how much brake torque will be generated at a given tank pressure. Figure 13 shows the results IMEP and mass flow from a TankVO sweep at two engine speeds, 600 and 1200 rpm. As mentioned earlier in section 5.2, optimal TankVO with regards to maximum efficiency will occur where IMEP is at a minimum, in this case at about 40 CAD BTDC. At this optimal point the cylinder pressure will be about equal to the tank pressure. If TankVO is advanced from the optimal point, IMEP will start to increase due to excessive compression of the air in the cylinder. Retarding the TankVO from the optimal point also leads to an increase in IMEP. The reason is that, when TankVO opens earlier than at the optimal point, there will be a blowdown of compressed air into the cylinder due to the fact that the in-cylinder pressure will be below the pressure in the tank. By retarding TankVO, the amount of braking torque generated can be controlled. Highest brake torque will be achieved when TankVO occurs at BDC. Between 180 and 70 CAD BTDC, it can be noticed that IMEP is higher for the case at 600 rpm compared to the case at 1200 rpm. The main reason for this behaviour is that at a higher engine speed, there will be less time for the previously described blowdown process to occur and thereby its effect on IMEP will be less.

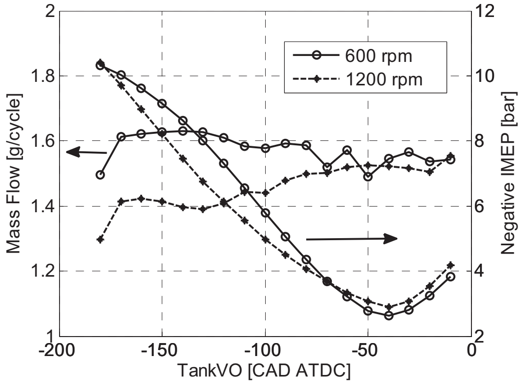
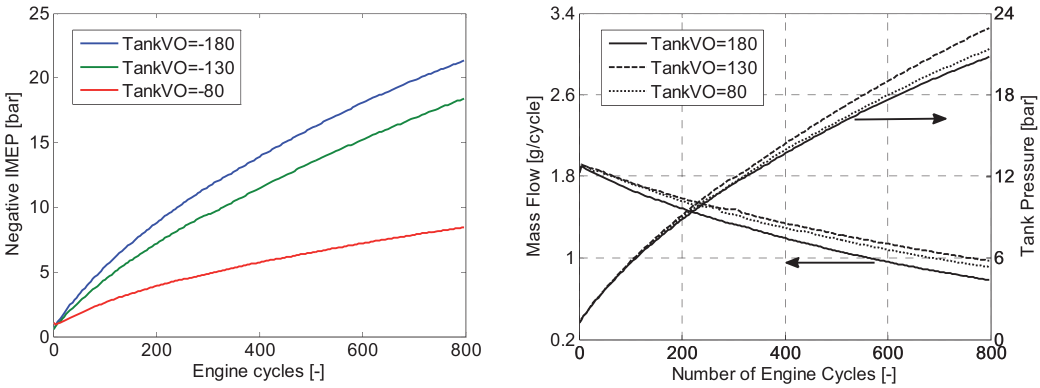


Figure 13 Mass flow and IMEP as a function of TankVO at a tank pressure of 10 bar and two different engine speeds.

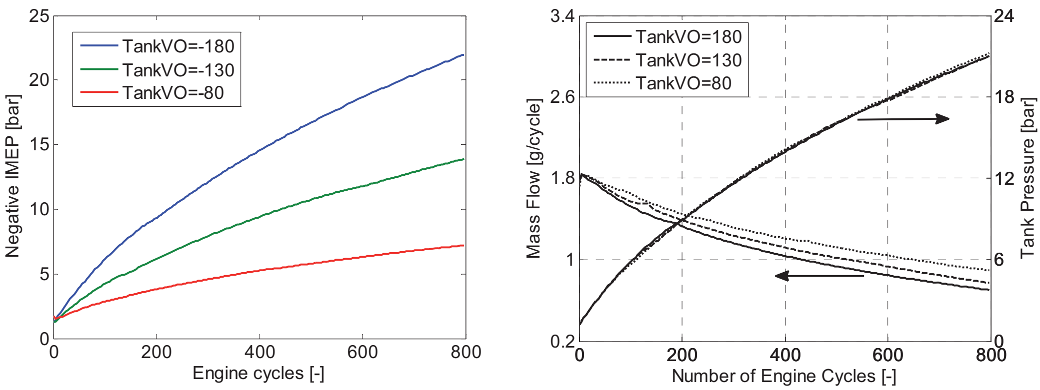
The other variable shown in Figure 13 is the mass flow of compressed air into the tank. It can be observed that the mass flow at the optimum point (40 CAD BTDC) and at an engine speed of 600 rpm is not at a maximum. The maximum mass flow instead occurs with a TankVO at about 130 CAD BTDC. However, at 1200 rpm the maximum air flow occurs in the vicinity of the optimal point. This phenomenon can be explained by pressure wave propagation in the pipeline connecting the tank to the cylinder head. For instance, a pressure wave can propagate back into the cylinder while the tank valve is open which can lead to a less than optimal charging of the tank. This propagation of pressure waves and their effect on the Compressor Mode performance will be discussed in more detail in section 5.3.5.

Figure 14 and Figure 15 display IMEP, mass flow and tank pressure for three different TankVO during transient Compressor Mode operation at two different engine speeds, 600 and 1200 rpm. The results follow the same trend as shown for the steady-state operation shown in Figure 13. In analogy to Figure 13, IMEP increases with retarded TankVO at both engine speeds, see Figure 14(a) and Figure 15(b). At 1200 rpm the mass flow has decreased for early TankVO and consequently the tank pressure has also decreased compared to the case at 600 rpm. For late TankVO (80 CAD BTDC) the mass flow at 600 and 1200 rpm, respectively, is almost the same, and hence similar pressure curves are achieved.



(a) (b)

Figure 14 Various parameters as a function of engine cycles for three different TankVO at an engine speed of 600 rpm, (a) Negative IMEP, (b) Mass flow and tank pressure.



(a) (b)

Figure 15 Various parameters as a function of engine cycles for three different TankVO at an engine speed of 1200 rpm, (a) Negative IMEP, (b) Mass flow and tank pressure.

5.3.3 Tank Valve Closing

TankVC is also a very important parameter influencing the Compressor Mode performance. It determines when the charging of the pressure tank should end. Ideally the TankVC should occur during the expansion stroke at the moment the pressure in the cylinder equals the pressure in the tank. Figure 16 shows the influence of TankVC on both IMEP and mass flow. It can be seen that there is a TankVC where mass flow reaches its maximum. A late closure means that the pressure in the cylinder will be below the tank pressure and therefore a blowdown of pressurized air from the tank into the cylinder will occur and thus results in a lower mean mass flow into the tank. An early closure means that there still is a positive pressure difference between the cylinder and the tank, and therefore a portion of compressed air will remain unused in the cylinder. It can be noticed that the optimal TankVC with regards to mass flow is engine speed dependent, and can be seen in Figure 16. At 600 rpm optimal TankVC occurs at 8 CAD ATDC, while optimal TankVC at 1200 rpm occurs at 2 CAD ATDC. This behaviour can be explained by pressure wave propagation in the pipeline connecting the tank to the cylinder head. At 600 rpm, the local maximum of the pressure wave travelling from the cylinder and into the tank occurs later in the cycle, compared to the case at 1200 rpm. The TankVC should be timed in such way that the positive pressure pulse should enter the tank before the valve closes in order to maximize the charging of the tank.

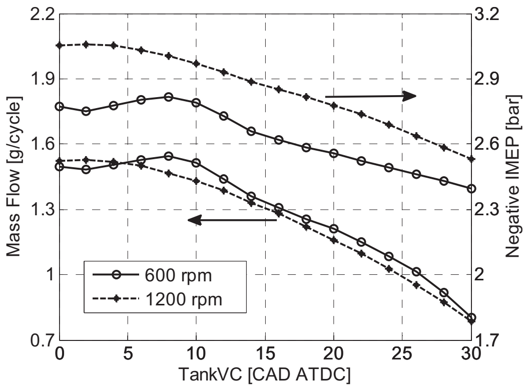


Figure 16 Mass flow and IMEP as a function of TankVC at a tank pressure of 10 bar and two different engine speeds.

The other variable shown in Figure 16 is the IMEP. It can be observed that IMEP decreases with advancing TankVC. The reason is that a late TankVC means there will be a blowdown of pressurized air from the tank into the cylinder. This excess pushes the piston during the expansion stroke and thereby contributes with positive IMEP which decreases the negative IMEP for the whole cycle.

Figure 17 and Figure 18 display IMEP, mass flow and tank pressure for three different TankVO during transient Compressor Mode operation at two different engine speeds, 600 and 1200 rpm. The results follow the same trend as shown for the steady-state operation shown in Figure 16. As expected, IMEP decreases with advanced TankVO at both engine speeds, see Figure 17(a) and Figure 18(a). The same applies to the mass flow. When it comes to tank pressure, it can clearly be seen how it is influenced by TankVC. A late TankVC, decreases the mass flow of compressed air into the tank and thereby the tank pressure. This behaviour indicates the importance of TankVC for optimal tank charging. Another thing that can be noticed is that there is an increase can be seen at 1200 rpm compared to at 600 rpm. Once again this can be attributed to the engine speed dependent heat transfer described in section 5.3.1.

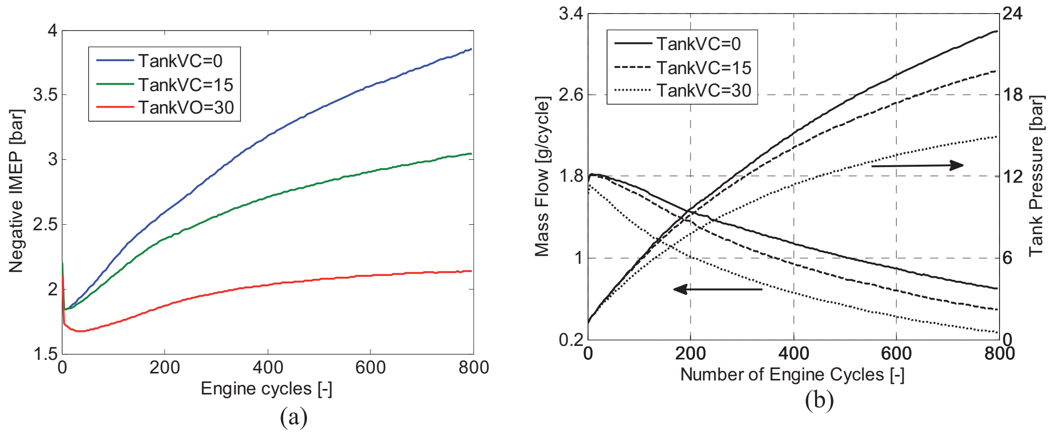


Figure 17 Various parameters as a function of engine cycles for three different TankVC at an engine speed of 600 rpm, (a) Negative IMEP, (b) Mass flow and tank pressure.

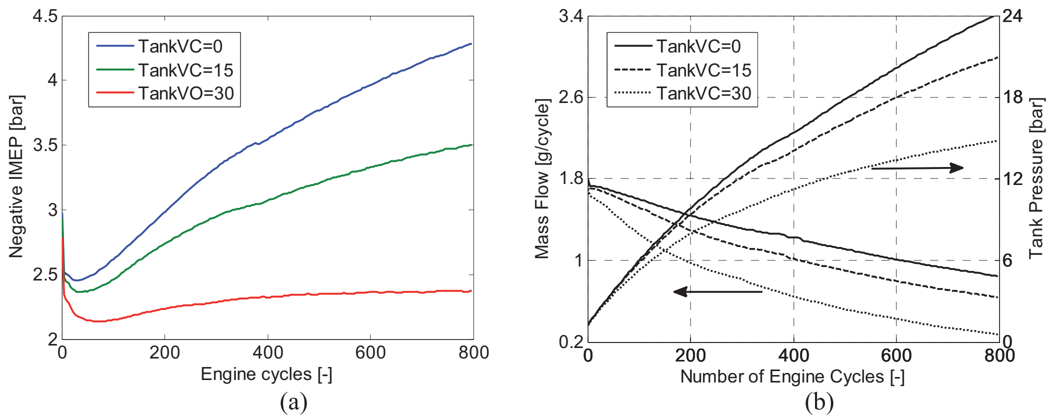


Figure 18 Various parameters as a function of engine cycles for three different TankVC at an engine speed of 1200 rpm, (a) Negative IMEP, (b) Mass flow and tank pressure.

5.3.4 Inlet Valve Opening

The role of the inlet valve during Compressor Mode operation is to admit a fresh charge of air during the end of the expansion stroke. The inlet valve should open when the pressure in the cylinder during the expansion stroke reaches ambient pressure. A too late IVO will lead to a decrease in in-cylinder pressure below ambient pressure and thus vacuum is created which is an energy consuming process. When the IVO finally occurs, the vacuum is cancelled by the induction of fresh air into the cylinder and thereby the vacuum previously created cannot be used as an upward-acting force on the piston as it moves towards TDC. The influence of IVO on mass flow and IMEP during steady-state operation for two engine speeds can be seen in Figure 19. It can be observed that there is an optimal IVO where mass flow reaches a maximum at both engine speeds. There are two different effects occurring when IVO is retarded or advanced, respectively, away from optimal IVO. If IVO occurs too early, the pressure in the cylinder will be above

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ambient pressure, and there will be a blowdown of compressed air into the inlet manifold. This blowdown will start a strong pressure propagation which will travel out from the cylinder leaving a lower pressure in the cylinder. This will result in a lower amount of fresh air trapped in the cylinder for the forthcoming cycle. If IVO occurs too late, there will be less time to induct fresh air and thus the amount of fresh air will be lower for the following cycle. When it comes to IMEP, the same arguments as for mass flow can be applied. An early IVO will lead to a loss of compressed air, which otherwise would have contributed with positive IMEP by pushing the piston towards BDC. A late IVO generates vacuum which contributes with an increase in IMEP.

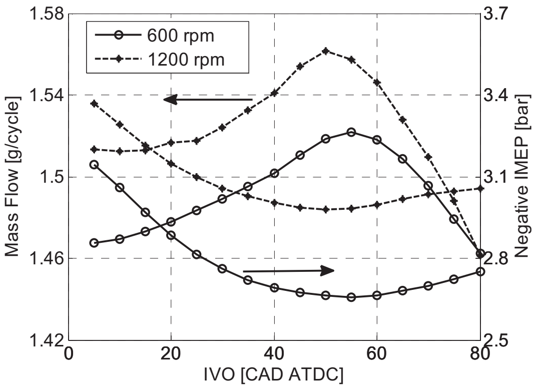

Figure 19 Mass flow as a function of IVO at a tank pressure of 10 bar and two different engine speeds

Figure 20 and Figure 21 display the IMEP, mass flow and tank pressure for three different IVO during transient Compressor Mode operation at two different engine speeds, 600 rpm and 1200 rpm. Figure 20(a) shows something that needs attention. The IMEP at an IVO of 20 CAD ATDC starts at the lowest level amongst the three tested IVO while IMEP at an IVO of 80 starts at the highest level. However, after about 170 engine cycles, the two cases switch places during the remaining part of the experiment. The explanation can be attributed to the fact that proper IVO is a function of tank pressure. At low tank pressures the in-cylinder pressure at TankVC will also be low. The low in-cylinder pressure means that the expansion of the compressed air trapped in the cylinder will reach ambient pressure early during the expansion stroke and in order to avoid vacuum generation the IVO should occur early. As the tank pressure increases, the cylinder pressure will also increase and therefore the IVO should be advanced and close later in the cycle. By setting IVO to a constant value, the above mentioned phenomena will occur. Therefore, with references to Figure 20(a), at low tank pressures the case with IVO at 20 CAD is more appropriate compared to the case with IVO at 80 CAD ATDC where the higher IMEP is caused by vacuum generation. However, as the cylinder pressure increases, the case with the IVO 80 CAD ATDC become more suitable compared to the case with IVO at 20 CAD ATDC where the increase in IMEP is caused by the release of compressed air into the intake manifold. The same arguments can be applied to the mass flow in Figure 20(b) and Figure 21(b).

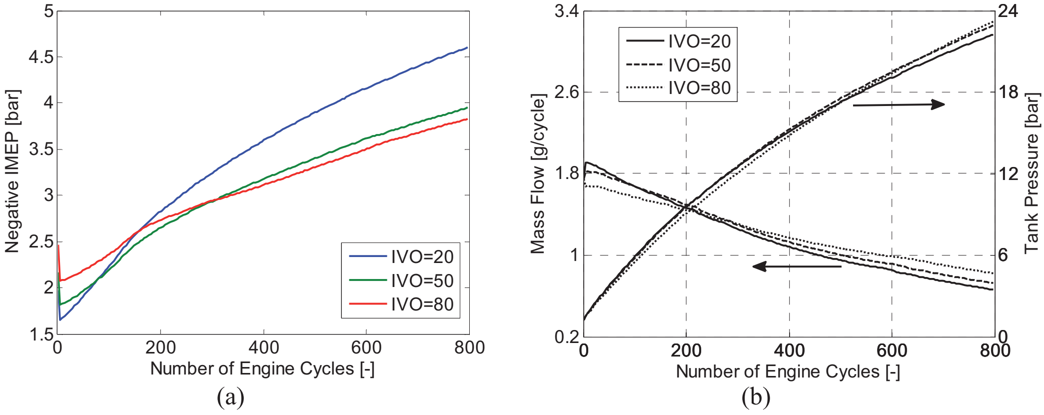

Figure 20 Various parameters as a function of engine cycles for three different TankVC at an engine speed of 600 rpm, (a) Negative IMEP, (b) Mass flow and tank pressure.

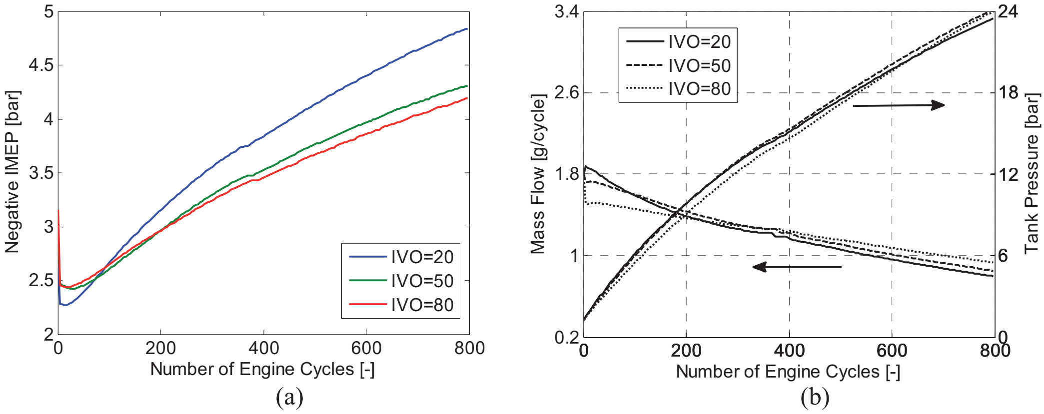

Figure 21 Various parameters as a function of engine cycles for three different TankVC at an engine speed of 600 rpm, (a) Negative IMEP, (b) Mass flow and tank pressure.

5.3.5 The Effects of Pressure Wave Propagation on Compressor Mode operation

It is widely known that the geometry of the intake and exhaust lines has a great influence on the gas wave motion, which can be utilized for supercharging of the engine. Knowledge of pressure wave propagation is also important for the pneumatic hybrid. If the pipeline connecting the pressure tank to the engine is poorly tuned, a pressure wave can propagate back into the cylinder while the tank valve is open which can lead to a less than optimal charging of the tank. Figure 22 shows the mass flow as a function of both IMEP and tank pressure. It can be seen that the isolines show a wave-like behaviour. At low tank pressures the isolines are almost parallel with the y-axis. In other words, the mass flow at low tank pressures is almost independent of the load. However, at higher tank pressures the isolines transforms into wave-like lines. This behaviour can be explained with pressure wave propagation in the pipeline connecting the pressure tank with the cylinder head.

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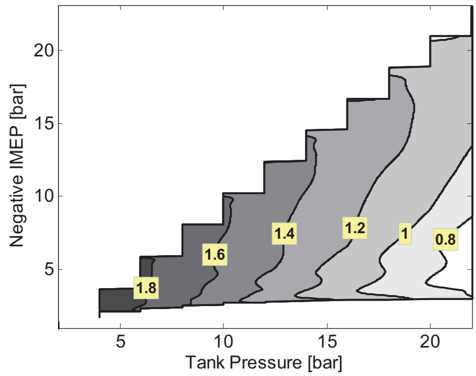
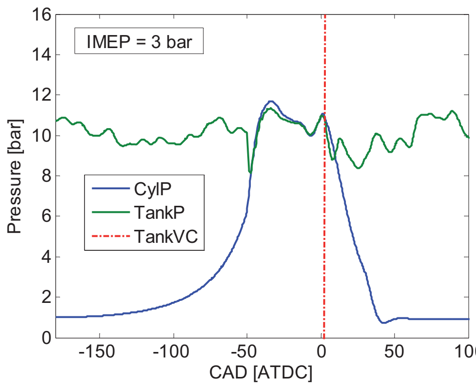


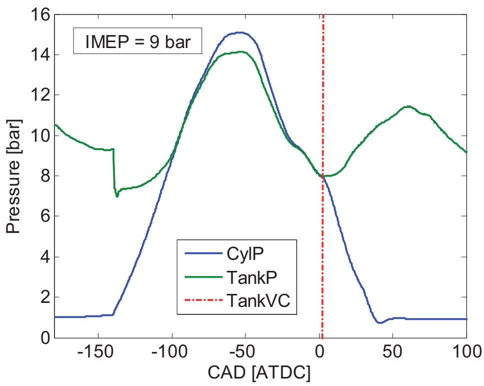
Figure 22 Mass flow rate (g/cycle) in to the tank as a function of both negative IMEP (2-stroke scale) and tank pressure during Compressor Mode

Figure 23 and Figure 24 shows the in-cylinder pressure together with corresponding tank valve port pressure at two different TankVO at an engine speed of 600 rpm and 1200 rpm, respectively. It can be seen that the in-cylinder pressure and tank port pressure at TankVC (indicated by the dotted line) is lower for early TankVO and can be explained by wave movement in the system. When the tank valve closes for the early TankVO case, the pressure in the cylinder is below the average tank pressure of 10 bar, which means that there is a pressure wave travelling into the tank at higher pressure leading to a higher degree of mass flow into the tank. The increase in pressure before TankVC for the late TankVO case means that the pressure wave has returned to the cylinder and is contributing to a mass flow out from the tank and the net result will therefore be a lower mass flow compared to the early TankVO case.

By comparing Figure 23 to Figure 24, it can be seen that there is a difference between both tank port pressure and in-cylinder pressure for the corresponding TankVO. This indicates that the pressure wave propagation not only depends on TankVO, but also on

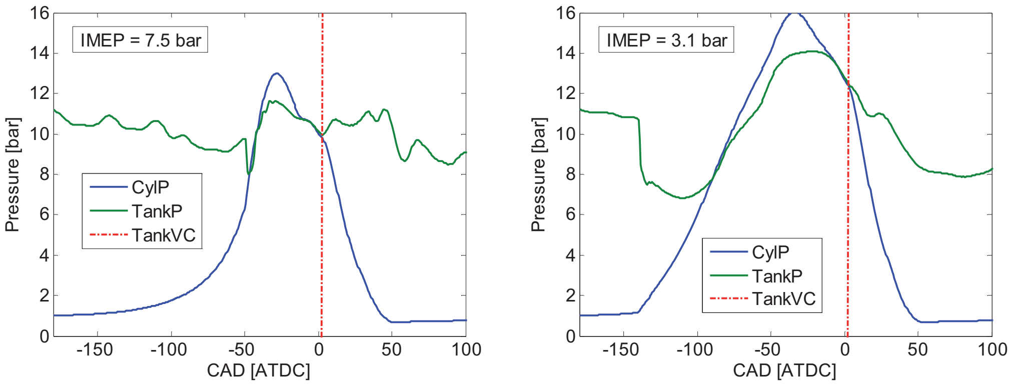


(a)



(b)

Figure 23 Cylinder and tank pressure as a function of CAD at a mean tank pressure of 10 bar and at a TankVO of 50 (a) and 140 (b) CAD BTDC. Engine speed = 600 rpm.



(b)

(a)

Figure 24 Cylinder and tank pressure as a function of CAD at a mean tank pressure of 10 bar and at a TankVO of 50 (a) and 140 (b) CAD BTDC. Engine speed = 600 rpm.

engine speed. In Figure 23(a), the cylinder pressure starts to increase at about 10 CAD before TankVC which indicates that there is pressure wave entering the cylinder. However, in Figure 24(a) the cylinder pressure is decreasing before TankVC, which indicates that a positive pressure pulse is leaving the cylinder. This means that at a TankVO of 50 CAD BTDC, the charging of the tank is more favourable at 1200 rpm. By comparing Figure 24(b) to Figure 23(b) it can be seen that there exist a phase shift in tank port pressure. In Figure 23(b), the tank port pressure before TankVC is decreasing and reaches a minimum at TankVC indicating that the positive pulse has left the cylinder. In Figure 24(b), the tank port pressure has undergone a phase shift. The tank port pressure is decreasing before TankVC but does not reach its minimum, which indicates that the positive pressure pulse has not completely left the cylinder at TankVC resulting in a less than optimal charging of the tank.

Table 4 The influence of TankVO and engine speed on mass flow and IMEP during Compressor Mode operation

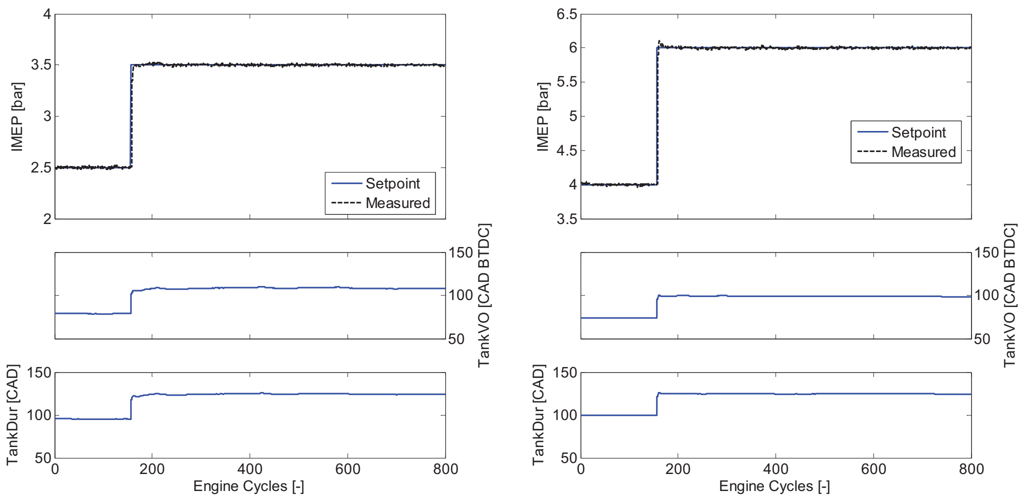
TankVO [CAD BTDC]	n [rpm]	Mflow [g/cycle]	IMEP [bar]
50	600	1.49	2.80
	1200	1.52	3.10
140	600	1.63	8.60
	1200	1.40	7.50

In Table 4, mass flow and IMEP for two different TankVO and engine speeds can be seen. For the case with a TankVO at 50 CAD BTDC, it has been concluded that the tank charging is more favourable at 1200 rpm compared to at 600 rpm. This is proved once again in Table 4, where it can be noticed that the mass flow of compressed air into the tank is higher at 1200 rpm compared to at 600 rpm. For the case with a TankVO at 140

CAD BTDC, it was earlier concluded that the tank charging was more favourable at 600 rpm compared to at 1200 rpm. This is once again proved in Table 4, where it can be seen that the mass flow of compressed air into the tank is higher at 600 rpm compared to 1200 rpm. It should be noted that, at 600 rpm the mass flow at a TankVO of 140 CAD BTDC is higher compared to at a TankVO of 50 CAD BTDC, which is in analogy with the results presented in section 5.3.2.

5.4 Load Control of Compressor Mode

The results in previous sections have shown the influence of different parameters on compressor mode operation at various conditions. With the help of these results it is possible to choose the parameters in such a way that the compressor mode performance is maximized. However, in a real application, the choice of parameters will be entirely determined by the driving conditions and not by what is currently most optimal. During a braking event, the driver should be given the ability to choose the degree of braking power by pressing the brake pedal. Therefore, an important aspect of the pneumatic hybrid concept is its ability to control the amount of braking torque at a specific time. A desired torque should be achievable whatever the pressure level in the tank is. For this purpose, a control strategy has been adopted, that controls the pneumatic hybrid powertrain load during compressor mode at different tank pressures.



(a) (b)

Figure 25 (a) Load step from 2.5 to 3.5 bar at a tank pressure of 5 bar, (b) Load step from 4 to 6 bar at a tank pressure of 10 bar.

Figure 25 illustrates two load steps at different tank pressures with corresponding PID output, TankVO and tank valve duration (TankDur). In Figure 25(a) the load step is between an IMEP of 2.5 and 3.5 bar at a tank pressure of 5 bar, while the load step in Figure 25(b) is between an IMEP of 4 and 6 bar at a tank pressure of 10 bar. It can be seen that the response to the change in set point is very fast, and after the measured IMEP has reached the set point it remains constant during the remaining part of the experiment. Figure 26 shows a close-up of the load step shown in Figure 25(b). It can be observed that there is a response delay from the moment the set point changes to the moment the measured value changes. The response delay is about 2 engine revolutions and occurs mainly due to data handling and communication with the actuators. After the actuator has received the first control signal it takes about 2 additional engine revolutions to reach the desired set point. This delay occurs due to the dynamics of the PI controller. In total, it takes up to 4 engine revolutions from the moment the set point changes and to the moment when the measured variable reaches the set point. By moving the control system to a real-time operating system (RTOS) and better tuning of the PI controller, these delays can most likely be halved.

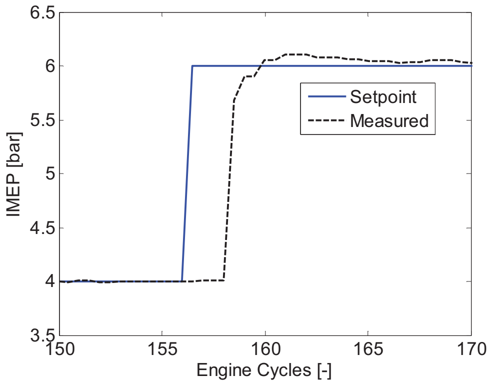


Figure 26 Close-up of the load step shown in Figure 25(b)

In Figure 27, four consecutive load steps are displayed. The load step is changed between 4 and 6 bar with an increment of 1 bar. Once again the controller manages to rapidly bring the measured variable to its new set point. The shorter dwell periods between the load steps seems to have no observable negative effects on the stability of the controller.

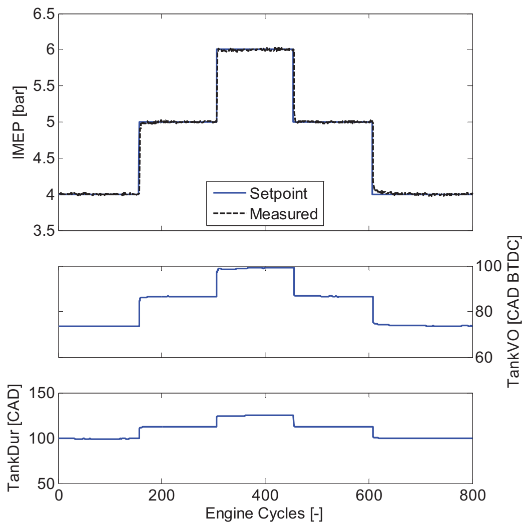


Figure 27 Four consecutive load steps ranging from 4 to 6 bar at a tank pressure of 10 bar.

An important issue that requires attention is that, with references to Figure 22, the maximum achievable negative IMEP at low tank pressures is quite limited. According to Trajkovic et al. (2010), the maximum theoretical IMEP during compressor mode is a function of tank pressure and can be defined as:

$$IMEP_{max} = p_{tank} - p_{im}$$

where p_{tank} is the tank pressure and p_{im} is the manifold pressure. For instance, at a tank pressure of 5 bar the maximum achievable negative IMEP is 4 bar which might not be enough during a hard braking event. The solution is to find a minimum allowable tank pressure that will keep the friction brake usage at a minimum. However, to find a general solution to this problem is quite hard, since the brake power depends heavily on the driving conditions and the drivers braking habits. Setting the minimum allowable tank pressure too high or too low will lead to a decrease in the contribution to fuel consumption reduction by pneumatic hybridization. In Trajkovic et al. (2010), a minimal tank pressure of 8 bar was shown to give the lowest fuel consumption together with a friction brake usage of only 13% during the Braunschweig city driving cycle.

6 Conclusions

In present paper a thorough investigation of the parameters influencing the pneumatic hybrid powertrain performance during compressor mode has been conducted. The results have shown that, theoretically calculated valve timings can give satisfying results during the compressor mode. However, optimal performance can only achieved by optimization of valve timings.

The parametric study of different valve parameters, such as valve head diameter, tank valve opening and closing, and inlet valve opening, has shown that these parameters influence the performance of compressor mode considerably. Therefore, fine-tuning of these parameters is crucial in order to secure optimal operation of the compressor mode. What differentiates theoretical valve timings from valve timings determined from experiments is that there is pressure propagation in the system which will lead to the fact that optimal valve timings with regards to optimal tank charging will be different at different operating conditions. This applies both to TankVO and TankVC, as well as for intake valve opening.

Further, a control strategy for load control has been developed in order to investigate the performance of compressor mode during more realistic driving conditions. The results show that the chosen controller strategy performs well with some minor constraints mainly at low tank pressures. The desired load level of the powertrain can be controlled almost on a cycle-to-cycle basis. The results shown in present study are an important step in the development of the pneumatic hybrid powertrain.

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