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Introductory Study of Variable Valve Actuation for Pneumatic Hybridization

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ABSTRACT

Urban traffic involves frequent acceleration and deceleration. During deceleration, the energy previously used to accelerate the vehicle is mainly wasted on heat generated by the friction brakes. If this energy that is wasted in traditional IC engines could be saved, the fuel economy would improve. One solution to this is a pneumatic hybrid using variable valve timing to compress air during deceleration and expand air during acceleration. The compressed air can also be utilized to supercharge the engine in order to get higher load in the first few cycles when accelerating.

A Scania D12 single-cylinder diesel engine has been converted for pneumatic hybrid operation and tested in a laboratory setup. Pneumatic valve actuators have been used to make the pneumatic hybrid possible. The actuators have been mounted on top of the cylinder head of the engine. A pressure tank has been connected to one of the inlet ports and one of the inlet valves has been modified to work as a tank valve. The goal has been to test and evaluate 2 different modes – compression mode (CM) where air is stored in an air tank during deceleration and air-motor mode (AM) where the previously stored pressurized air is used for accelerating the vehicle. This paper also includes an optimization of the CM.

INTRODUCTION

As fuel prices increase, together with more stringent pollution standards, the demand for better fuel economy increases. Today there are several solutions to meet this demand and one of them is electric hybrids. In urban traffic the vehicle has to accelerate and decelerate frequently. In conventional vehicles the energy used for acceleration of the vehicle is wasted in the form of heat generated by the friction brakes during deceleration. This leads to a higher fuel consumption during city driving compared with freeway driving. The idea with

electric hybridization is to reduce the fuel consumption by taking advantage of the, otherwise lost, brake energy. Hybrid operation can also allow the combustion engine to operate at its best operating point 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 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 naturally leads to a higher price for the consumers to pay. One way to keep the costs down is the introduction of pneumatic hybrid. It doesn't need 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 and stores the compressed air into a pressure tank. After a standstill the engine is used as an air-motor that uses the pressurized air from the tank in order to accelerate the vehicle. During a full stop the engine can be shut off. All these features of the pneumatic hybrid contribute to lower fuel consumption. Simulations made by Tai et al. [4] show a so called "roundtrip" efficiency of 36% and fuel economy improvement as high as 64% in city driving. This indicates that the pneumatic hybrid can be a promising alternative to the traditional vehicles of today and a serious contender to the better known electric hybrid. Andersson et al. [3] describes simulations of a dual pressure tank system for heavy vehicles with a regenerative efficiency as high as 55% and promising fuel savings.

PNEUMATIC HYBRID

The main idea with pneumatic hybrid is to use the ICE in order to compress atmospheric air and store it in a pressure tank during vehicle deceleration. 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. It is also possible to completely shut off the engine at for instance a stoplight, which in turn contributes to lower fuel consumption. [1, 4]

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 vehicle 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 will be conducted in a near future in order to find the optimal number of converted cylinders.

PNEUMATIC VARIABLE VALVE ACTUATION

In order to be able to switch between all these modes of engine operation, a variable valve 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 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. [2].

TANK VALVE

In order to run the engine as a pneumatic hybrid, a pressure air tank has to be connected to the cylinder head. Tai et al. [4] describes an intake air switching system in which one inlet valve per cylinder is feed by either fresh intake air or compressed air. 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, that will 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

In this paper two different engine modes have been investigated – compressor mode (CM) and air-motor

mode (AM) and they will be described more thoroughly below.

COMPRESSOR MODE

In CM the engine is used as a 2-stroke compressor in order to decelerate the vehicle. The inlet valve opens a number of CAD after TDC and brings fresh air to the cylinder and closes around BDC. The moving piston compresses the air after BDC and the tank valve opens somewhere between BDC and TDC, depending on how much braking torque is needed and closes around TDC. The compressed air generated during CM is stored in a pressure tank that is connected to the cylinder head. A simple illustration of CM can be seen in Figure 1.

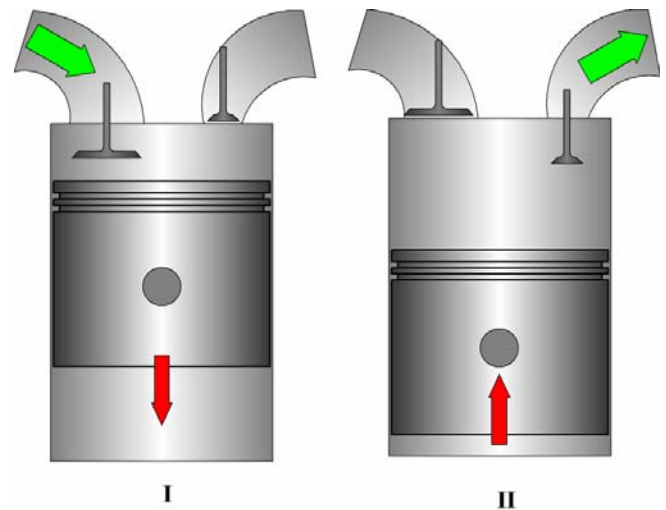


Figure 1 Illustration of CM. I) Intake of fresh air, II) Compression of air and pressure tank charging.

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 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 will close, depending on how much torque the driver demands. Increasing tank valve duration will increase the torque generated by the compressed air. The inlet valve opens around BDC in order to avoid compression of the air in the cylinder. The AM is illustrated in Figure 2.

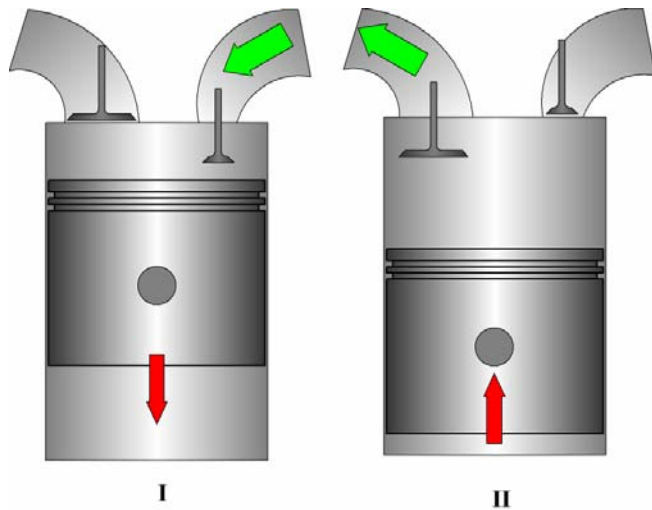


Figure 2 Illustration of AM. I) Intake of compressed air, II) Air venting

EXPERIMENTAL SETUP

The engine used in this study is a single-cylinder Scania D12 diesel engine together with the pneumatic variable valve actuating system described earlier in this paper. The geometric properties of the engine can be seen in Table 1. Figure 3 shows a close-up of the pneumatic valve actuators mounted on top of 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.

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 4. Note that this testing involved a one-tank hybrid system, unlike the two-tank system described in [3]. 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.

Table 1 Engine geometric properties.

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

Table 2 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 2 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	340 N
Maximum valve lift	7 mm

MODIFICATIONS TO TANK VALVE

In order to open the tank valve at high in-cylinder pressures some modifications of the tank valve had to be introduced. The valve diameter had to be decreased from 45 mm to 16 mm. The tank valve spring preloading had to be changed from 100 to 340 N in order to keep the tank valve completely closed for tank pressures up to 25 bars.

MODIFICATIONS TO CYLINDER HEAD

In order to use the modified tank valve the original valve seating had to be exchanged for a smaller seating.

The inlet port related to the tank valve has been connected to the pressure tank with metal tubing resistant to high temperatures and pressures.

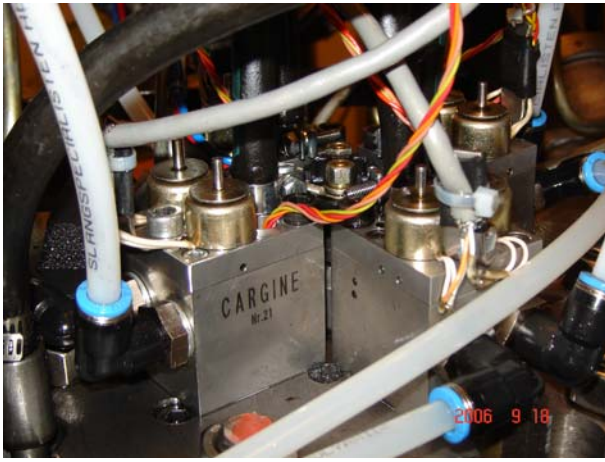


Figure 3 The pneumatic valve actuators mounted on the Scania cylinder head



Figure 4 The pressure tank connected to the cylinder head by metal tubing.

ENGINE EXPERIMENTAL RESULTS

Both CM and AM have been tested in this study. Also results from optimization of CM will be shown. The results will be discussed thoroughly. Notice that all presented pressure values are absolute.

COMPRESSOR MODE

The CM tests can be done in two ways. The first one 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 one 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. This paper focuses more on the first method, i.e. achieving higher compression efficiency.

Table 3 shows the valve strategy used in this part of the experiment. The tank valve opening is feedback controlled and depends on the in-cylinder pressure and the tank pressure. The feedback control is based on the isentropic compression law:

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

p_1 corresponds to the pressure at BDC and p_2 is the pressure at any other point in the cycle. V_1 is the maximal volume in the cylinder and V_2 is the cylinder volume when the cylinder pressure is 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 the correct tank valve timings.

Table 3 Valve strategy in CM

Tank valve opening	Cylinder pressure \approx tank pressure
Tank valve closing	10 CAD ATDC
Inlet valve opening	35 CAD ATDC
Inlet valve closing	180 CAD ATDC

Figure 5 and Figure 6 show the in-cylinder pressure and the tank pressure during one engine revolution in 2-stroke. Normally the maximum in-cylinder pressure occurs at TDC or shortly before, but in Figure 5 the peak of the in-cylinder pressure trace has moved more than 20 CAD away from TDC. The reason for this phenomenon is that the pressurized air in the cylinder escapes into the tank once the tank valve opens.

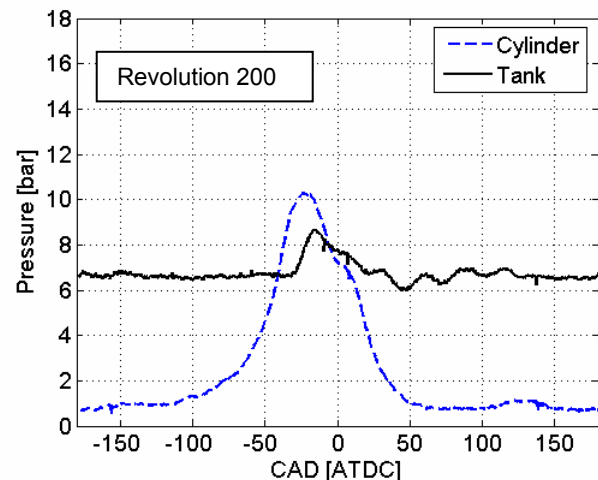


Figure 5 In-cylinder pressure and tank pressure in CM at revolution 200 and an engine speed of 600 rpm

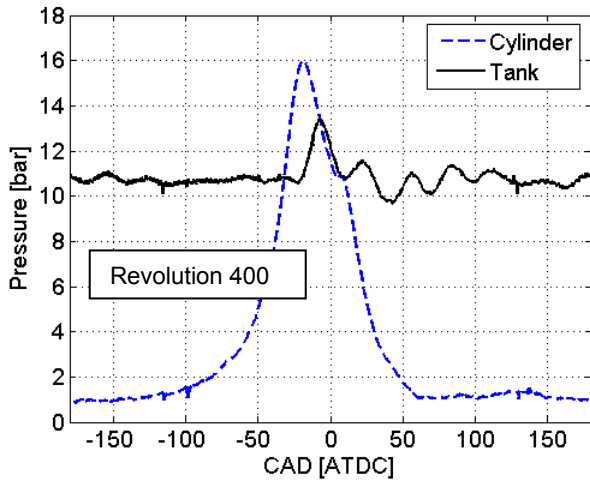


Figure 6 In-cylinder pressure and tank pressure in CM at revolution 400 and an engine speed of 600 rpm

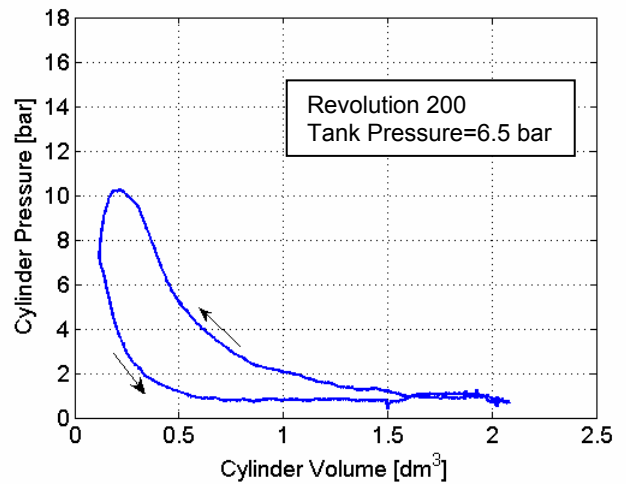


Figure 8 PV-diagram from engine testing at engine revolution number 200 and an engine speed of 600 rpm.

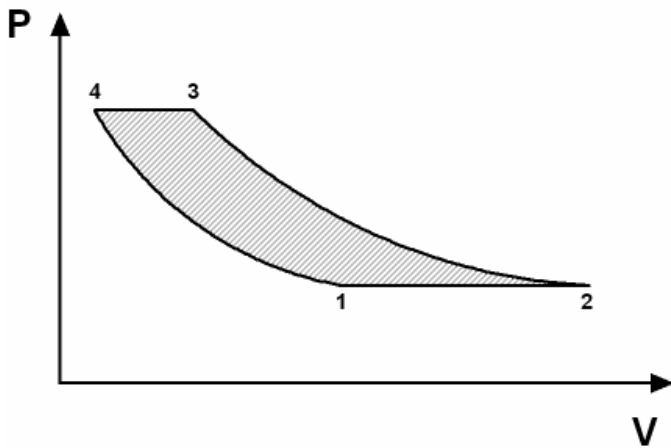


Figure 7 Illustration of an ideal PV-diagram of one CM revolution.

Figure 7 illustrates an ideal PV-diagram of one revolution in CM. At point 1 the inlet valve opens and fresh air enters the cylinder. The inlet valve closes at point 2 and the piston starts to compress the air until equilibrium between the tank and the cylinder pressure exists. At point 3 the tank valve opens and most of the compressed air is transferred to the pressure tank until the tank valve closes at point 4. The step between point 3 and 4 is isobar, which means that the cylinder pressure will remain constant while the tank valve is open. Between point 4 and 1, the remaining compressed air is expanded.

Figure 8 and Figure 9 shows the PV-diagram from real engine testing at two different tank pressures. Comparing Figure 7 with Figure 8 and Figure 9 clearly indicates that there is an absence of the isobar event in the real engine testing. The reason for this is that choking occurs over the tank valve which limits the air

flow and thereby the pressure will increase. This overshoot in pressure can probably be lowered dramatically if the small tank valve diameter is increased.

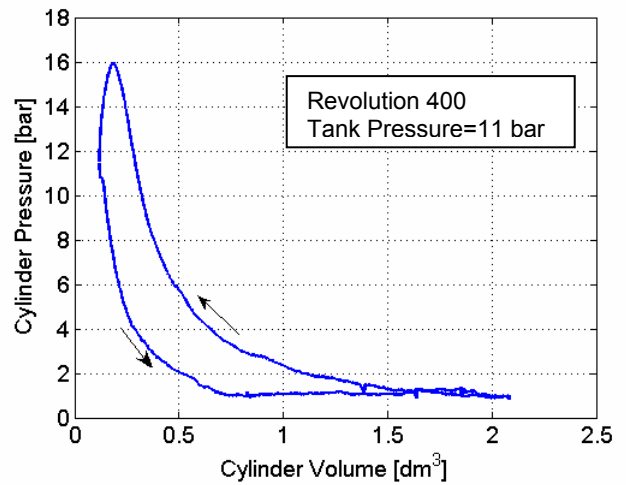


Figure 9 PV-diagram from engine testing at engine revolution number 400 and an engine speed of 600 rpm.

Figure 10 shows how the tank pressure increases with time during CM for three different engine speeds. Figure 11 shows the same data with the timescale changed from seconds to number of engine revolutions.

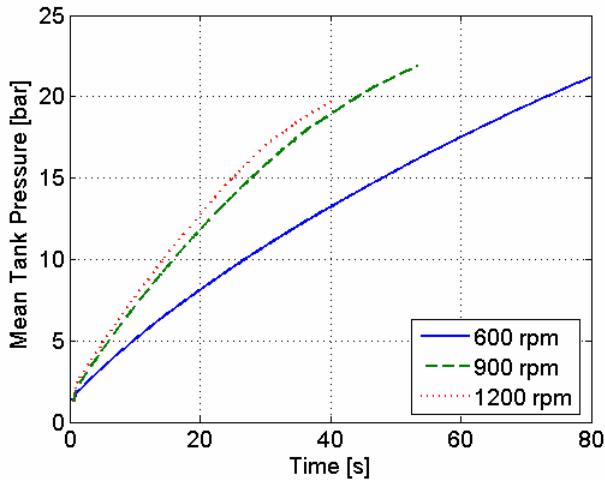


Figure 10 Mean tank pressure as a function of time for three different engine speeds.

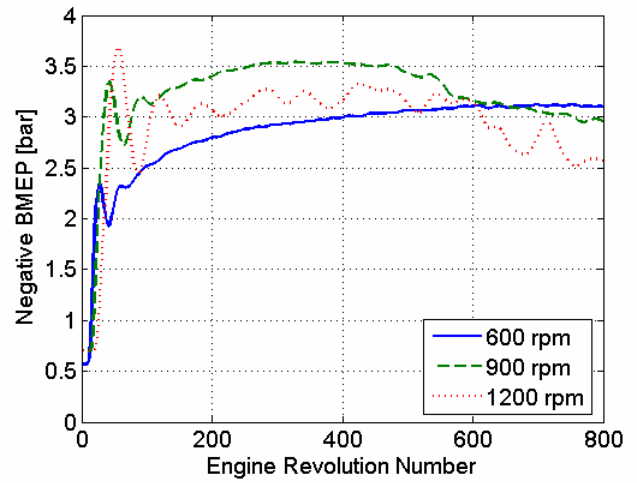


Figure 12 Negative BMEP as a function of engine revolutions for three different engine speeds.

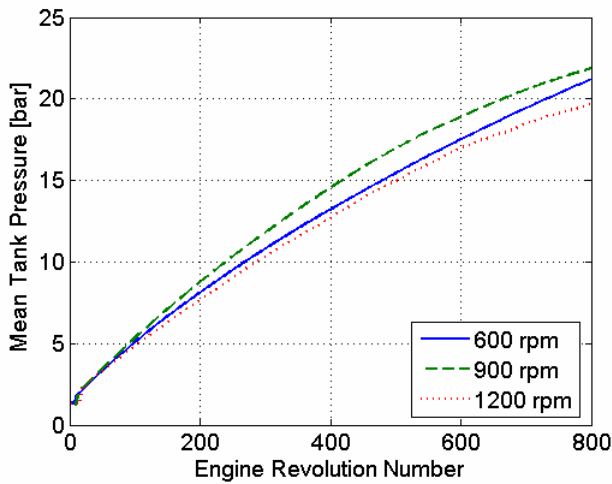


Figure 11 Mean tank pressure as a function of engine revolutions for three different engine speeds.

It is noticeable from Figure 11 that there is a difference between the tank pressures for different engine speeds. The reason for this is probably that the control program by coincidence is better optimized for the case at 900 rpm than at the other two cases.

Figure 12 shows negative BMEP during CM for three different engine speeds. The reason why BMEP is decreasing after about 500 revolutions at 900 and 1200 rpm is that the tank valve closing is not feedback controlled but set to a constant value. This leads to a premature tank valve closing when the tank pressure is high. The cylinder is then still filled with pressurized air that pushes the cylinder and thereby contributes with positive BMEP which decreases BMEP for the whole revolution. This phenomenon is engine speed dependant and it is not visible at 600 rpm. If the tank valve closing would be controlled in the same way as the opening, then the BMEP curve would probably be flat throughout the whole test run.

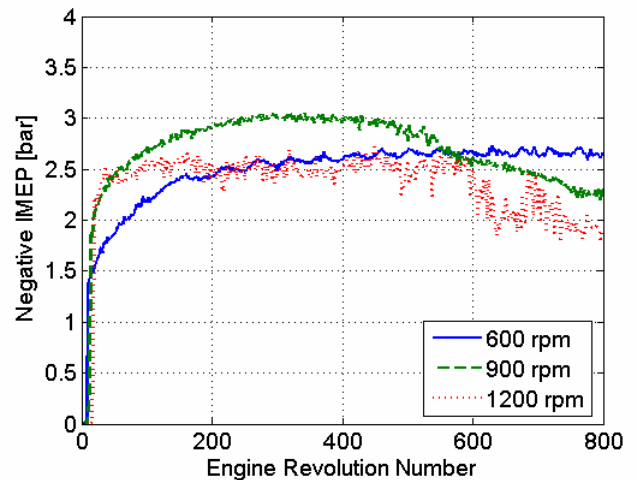


Figure 13 Negative IMEP as a function of engine revolutions for three different engine speeds.

Figure 13 shows IMEP during CM for the three different engine speeds. The same line of argument as for Figure 12 can be used in explaining Figure 13.

BMEP and IMEP in Figure 12 and Figure 13 have been calculated by two different methods. BMEP has been calculated as a function of torque and IMEP as a function of in-cylinder pressure. Figure 12 and Figure 13 can then be used to evaluate both methods. Since the graphs of IMEP and BMEP look almost the same, the methods can be seen as reliable.

Figure 14 shows clearly how the pressure losses over the tank valve increases with increasing engine speed. The pressure losses at 600 rpm are surprisingly low considering that the tank valve has a very small diameter.

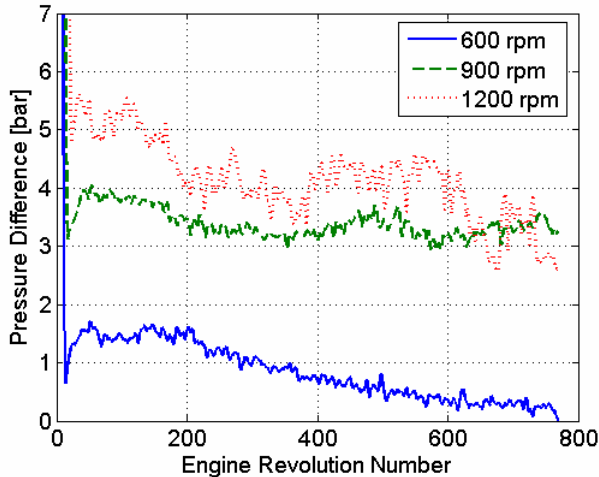


Figure 14 Pressure difference between maximum in-cylinder and tank pressure at various engine speeds.

OPTIMIZING THE COMPRESSOR MODE

Equation (1) described earlier in this paper is an isentropic relation between the pressure and the volume for various values of γ . γ is the specific-heat ratio and depends on the heat losses but in this study γ has been set to a constant value of 1.4. This will introduce some errors to the tank valve control algorithm and in order to avoid this, a method for optimizing CM mode has been tested.

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. Since the amount of air charged into the tank should equal the amount of air released from the tank in order to keep the tank pressure constant, a pressure relief valve is connected to the tank. It is then possible to achieve the desired steady state tank pressure by adjusting the pressure relief valve opening angle. The greater the opening angle the lower the steady state tank pressure.

This paper describes only the optimization of the tank valve opening.

Notice that BMEP and IMEP in the figures in this section are given in 4-stroke scale (doubled).

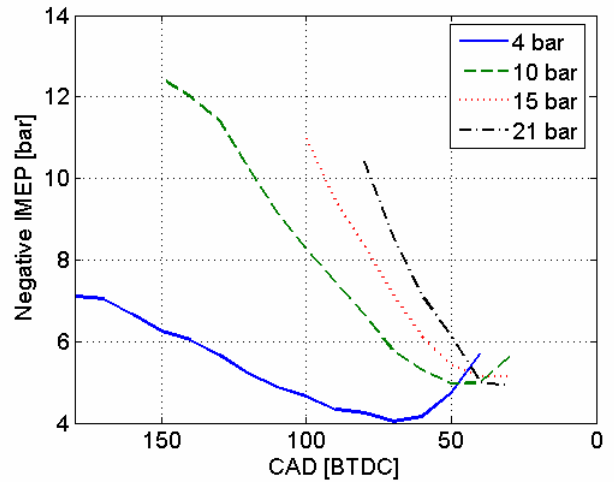


Figure 15 Negative IMEP as a function of valve opening timing during optimization of CM at various tank pressures and an engine speed of 600 rpm.

Figure 15 and Figure 16 show how negative IMEP and BMEP are affected by the tank valve opening timing during optimization of CM. From the figures it can be seen that there is an optimal tank valve opening 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 air at this point than at any other point on the curve at a given tank pressure. If higher braking power is needed, the efficiency has to be sacrificed.

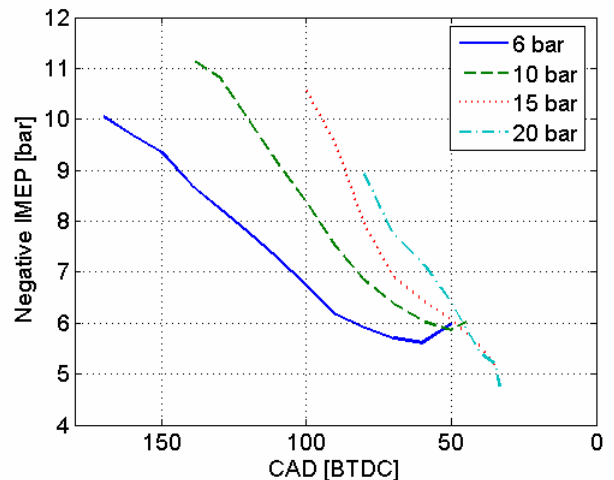


Figure 16 Negative IMEP as a function of valve opening timing during optimization of CM at various tank pressures and an engine speed of 900 rpm.

Figure 17 and Figure 18 show how the pressure difference between the maximum in-cylinder pressure and maximum tank pressure are affected by the tank valve opening during optimization of CM. The higher the differences are the higher the pressure losses are. The smallest differences occur at the same points as the minimum in the 4 previous figures. This verifies that the highest efficiency is achieved in these points.

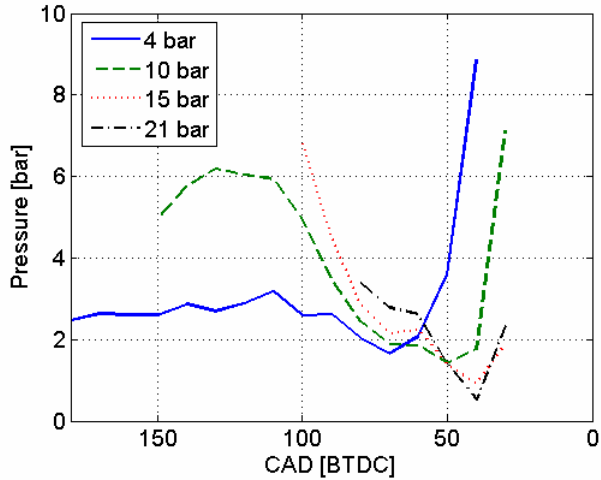


Figure 17 Pressure difference between maximum in-cylinder pressure and maximum tank pressure as a function of valve opening during optimization of CM at various tank pressures and an engine speed of 600 rpm.

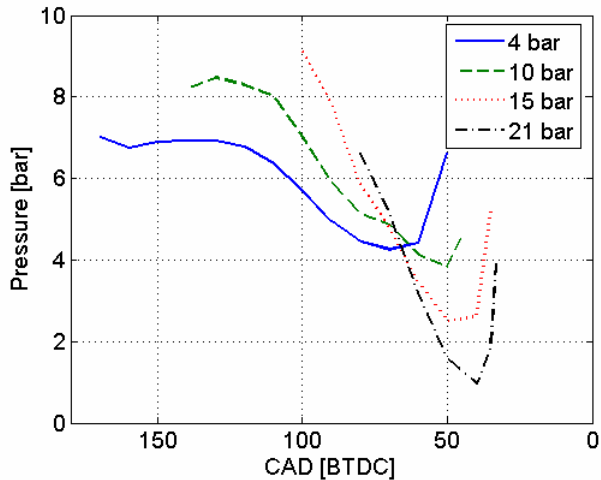


Figure 18 Pressure difference between maximum in-cylinder pressure and maximum tank pressure as a function of valve opening during optimization of CM at various tank pressures and an engine speed of 900 rpm.

AIR-MOTOR MODE

The AM can, as CM, be executed in two ways, either in regard to efficiency or brake power. The latter one has been used in this study. An optimization of AM similar to the one performed for CM will be presented in a future publication.

The tank valve opening is set to 5 CAD BTDC while the closing is varied with various tank pressures in order to get the highest torque. The inlet valve opens at BDC and closes at TDC. Table 4 shows the tank valve closings used in this experiment.

Table 4 Tank valve closing at various tank pressures in AM.

TankVC@22 bar	40 CAD ATDC
TankVC@15 bar	60 CAD ATDC
TankVC@12.5 bar	70 CAD ATDC
TankVC@10 bar	80 CAD ATDC

Figure 19 indicates that in order to achieve as high torque as possible, a valve control strategy involving valve timings similar to the valve timings in Table 4 has to be used.

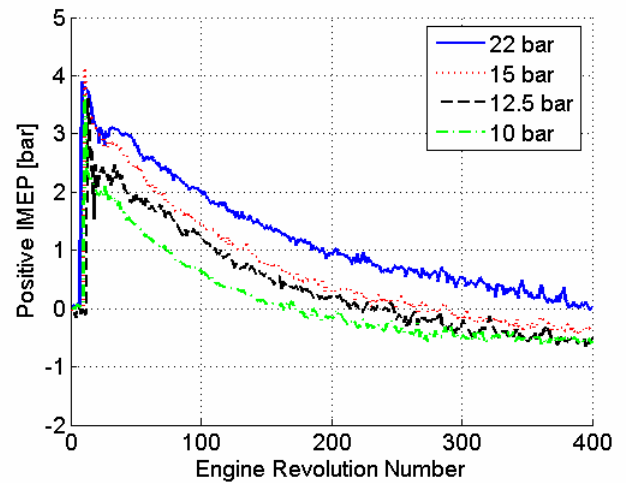


Figure 19 Positive IMEP in AM at an engine speed of 600 rpm.

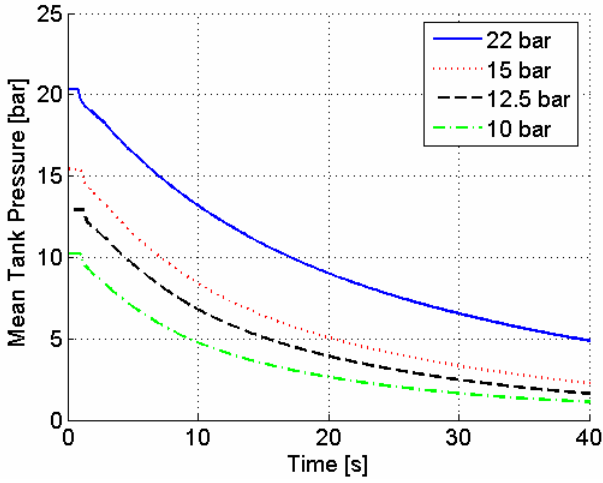


Figure 20 Mean tank pressure during AM at an engine speed of 600 rpm.

It can be seen in Figure 20 that when the pressure is as high as 22 bars, it lasts for over 40 seconds at the current engine speed and valve timings. This time duration can be extended if the tank valve duration is decreased and thereby the AM efficiency would be increased.

Notice that when the tank pressure reaches a value below 4.5 bars, IMEP becomes negative. This is due to wrong tank or inlet valve timings. The amount of pressurized air charged into the cylinder is not enough in order to expand it to atmospheric pressure at BDC. Instead the pressure will decrease below atmospheric pressure and since the inlet valve opens at BDC there will be a blowdown of atmospheric air into the cylinder which leads to negative IMEP. This event is illustrated in Figure 21. In order to avoid this, the inlet valve should not open until the cylinder pressure reaches atmospheric pressure. Another way to avoid negative IMEP in this case is to have longer tank valve duration. In this way the cylinder pressure will reach atmospheric pressure at BDC and the inlet valve can then open at BDC without any backflow of atmospheric air into the cylinder.

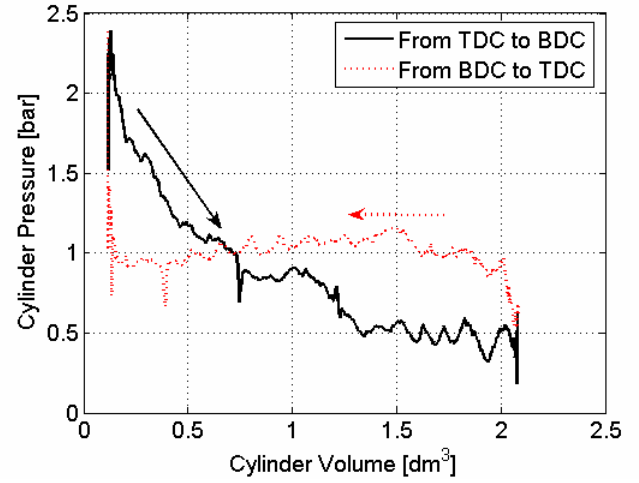


Figure 21 PV-diagram illustrates why IMEP is below 0 bars at some point during AM.

REGENERATIVE EFFICIENCY

In order to estimate the potential of the pneumatic hybrid, the regenerative efficiency has to be calculated. The regenerative efficiency is defined as the ratio between the energy produced during AM and the energy stored during CM. It can also be defined as the ratio between the positive and negative IMEP:

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

It can also be interesting to see the efficiencies for every revolution during CM and AM. They are defined as:

$$\eta_{CM} = \frac{P_{tank,CM}}{P_{engine-}} \quad (3)$$

and

$$\eta_{AM} = \frac{P_{engine+}}{P_{tank,AM}} \quad (4)$$

where $P_{engine-}$ is the engine motoring power during CM and $P_{engine+}$ is the engine brake power during AM.

P_{tank} in (3) and (4) is defined as the change of energy in the tank per time unit:

$$P_{tank,CM} = -P_{tank,AM} = \frac{\Delta E_{stored}}{\Delta t} \quad (5)$$

Figure 22 and Figure 23 show the efficiency in CM and AM, respectively. The reason why all the efficiency curves are going towards zero in Figure 23 is that IMEP

is going towards 0 bars. These really low, near zero, efficiencies are not desirable and it is then better to use the remaining pressurized air to supercharge the engine as described by Schechter et al. [1, 4].

Figure 24 and Figure 25 show the total efficiency in CM and AM, respectively.

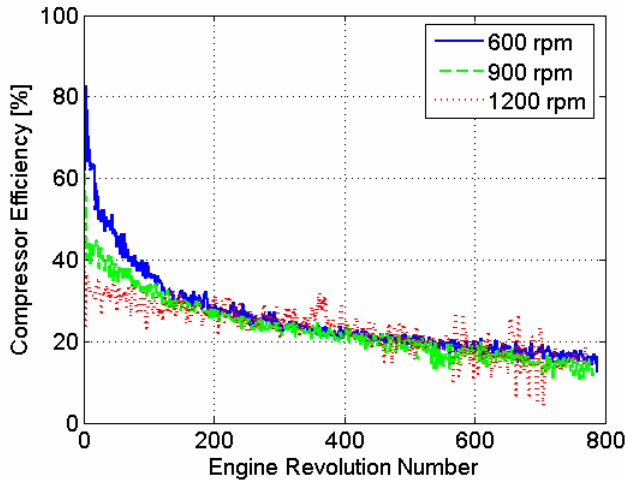


Figure 22 Compressor efficiency calculated for every engine revolution during CM at three different engine speeds.

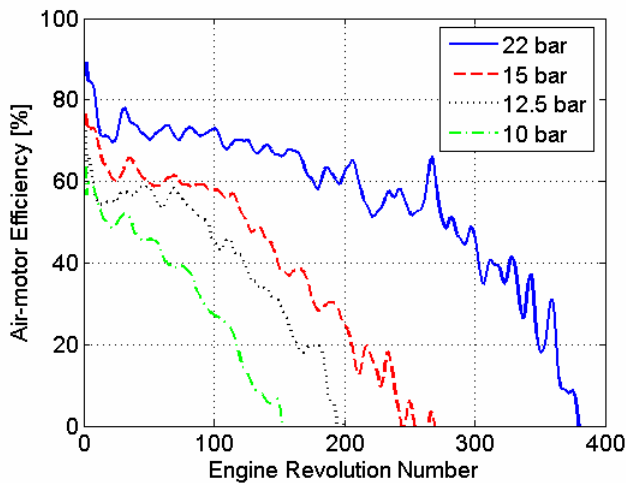


Figure 23 Air-motor efficiency calculated for every engine revolution during CM at various tank pressures and at an engine speed of 600 rpm

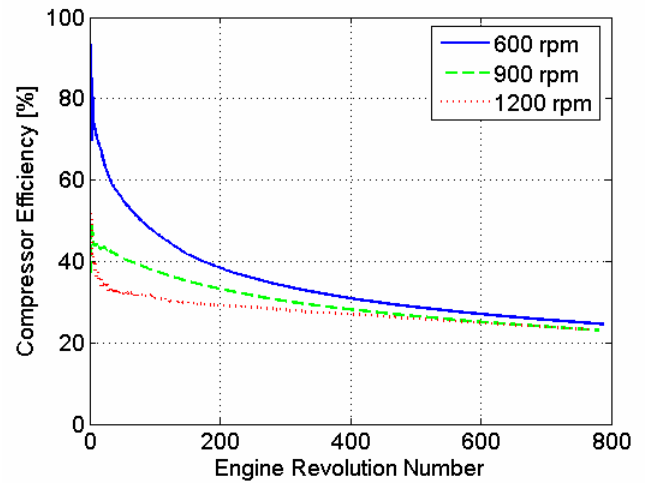


Figure 24 Total compressor efficiency as a function of engine revolutions during CM at three different engine speeds.

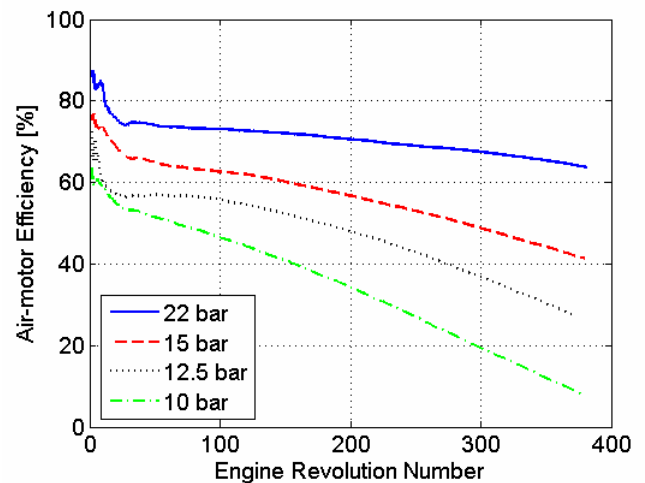


Figure 25 Total air-motor efficiency as a function of engine revolutions during CM at various tank pressures and an engine speed of 600 rpm.

Next step is to calculate the regenerative efficiency. First thing to do is to calculate how much work has been used in order to compress air from 5 bars to 22 bars in the three different CM runs. These numbers have been chosen because the starting tank pressure in AM is about 22 bars and 5 bars corresponds to the remaining air pressure when IMEP is around zero in AM. The total work used during CM is then divided with the total work recovered during AM. The calculated regenerative efficiency can be seen in Table 5 and the results shows that the run at 900 rpm was the most optimized as stated in one of the previous sections.

Table 5 Total calculated regenerative efficiency for three different engine speeds.

	η_{regen} [%]
CM@600rpm	32
CM@900rpm	33
CM@1200rpm	25

FUTURE WORK

- A third engine mode will be tested, namely the supercharge mode also called air-power-assisted mode by Schechter et al. [1, 4]. The idea with this mode is to use the compressed air in order to supercharge the engine and thereby achieve higher loads.
- The tank valve diameter will be increased as an attempt to lower the pressure losses.
- The pressure tank and the connecting tubing will be insulated in order to prevent cooling of the hot compressed air with increased efficiency as a result.
- This paper describes only the optimization of the tank valve opening in CM. Therefore optimization of valve closing and inlet valve timings will be done. All the obtained optimal measurements will then be introduced into a look up table in the control program.
- AM will be optimized in a similar way as CM.
- The engine testing described in this study will be simulated in GT-Power, in order to verify the real-life results. The simulations can be a useful tool in future optimization of the pneumatic hybrid.

CONCLUSION

The results achieved during the study show the potential with the pneumatic hybrid. Compared with simulations made by Tai et al. [4] the maximum regenerative efficiency obtained in this study (33 %) is a little lower, but still promising considering that it is an almost unoptimized system. The regenerative efficiency of 55% described by Anderson et al. [3] seems to be out of reach with the use of only one pressure tank.

The main factors that affect the efficiency negatively are probably the pressure losses over the valve and the heat losses due to poor insulation. The pressure losses can easily be lowered by switching to another tank valve with larger diameter. The heat losses can be lowered significantly by the introduction of some sort of insulation to the tank and the connecting tubing.

The results show that there is a pretty simple way to optimize the valve timings in order to increase the efficiency in CM. They also indicate on how to change the valve timing in order to achieve higher braking torque when needed.

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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
- E_{stored} : Energy stored in the tank
- η_{AM} : Air-motor Mode efficiency [%]
- η_{CM} : Compressor Mode efficiency [%]
- η_{regen} : Regenerative efficiency [%]
- γ : Specific heat ratio [-]
- ICE: Internal combustion Engine
- IMEP: Indicated Mean Effective Pressure
- p: In-cylinder pressure [bar]
- P_{engine} : Engine power [W]
- P_{tank} : Change of energy in the tank [J/s]

t: Time [s]

TDC: Top Dead Centre

RPM: Revolutions Per Minute

V: Cylinder volume [m³]