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Thermodynamic Simulation of a Hybrid Pneumatic-Combustion Engine Concept

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Abstract

Although internal combustion engines display high overall maximum global efficiencies, this potential cannot be fully exploited in automotive applications: in real conditions, the average engine load (and thus efficiency) is quite low and the kinetic energy during a braking phase is lost. This work presents a new hybrid pneumatic-combustion engine and the associated thermodynamic cycles, which is able to store energy in the form of compressed air. This energy can be issued from a braking phase or from a combustion phase at low power. The potential energy from the air tank can then be restored to start the engine, or charge the engine at full load. The regenerative breaking and the suppression of the idling phases could provide an improvement in terms of fuel economy as high as 15% or more if combined with engine downsizing.

Key words: internal combustion engine, pneumatic motor, thermodynamic cycles, pneumatic pump

1. Introduction

To be able to meet future challenges for internal combustion engines in terms of pollutant emissions and greenhouse effect gases emissions, the global efficiency over a real world driving cycle needs to be improved. Today, internal combustion engines display high overall maximum global efficiencies, topping at about 45% for turbocharged direct injection (DI) diesel engines. But this great potential cannot be fully exploited:

- The first problem is that the peak efficiencies can only be reached at full load. In real life operation, for example city traffic, the situation is even worse: full load is never reached, the engine is mainly idling or operating at very low load, and thus at very low efficiency.
- The second problem with internal combustion engines is that the thermodynamic cycle cannot be reversed: i.e. you cannot produce air and fuel from combustion products by providing reverse torque to the crankshaft. As a result, the kinetic energy of a vehicle cannot be recovered into chemical energy in negative torque i.e. braking situations: the kinetic energy must be dissipated into heat through the braking system.

One solution to this problem is to use a hybrid powertrain combining a conventional combustion engine for easy energy storage and high overall conversion efficiency, and a reversible energy converter and storage subsystem. Usually the second energy converter is an electric motor and the energy storage subsystem is a battery. The downside of this configuration is that the electric motor, generator and battery are heavy and expensive.

In this work, we propose a new hybrid engine concept, which combines an internal combustion engine with a pneumatic energy converter and storage subsystem. To lower the cost and weight and to stay with a well-known technology, the internal combustion engine itself has been used as a pneumatic pump or motor when there is no combustion.

2. The Concept

The concept is based on a conventional internal combustion engine where one additional valve called the charging valve is connecting the combustion chamber of each cylinder to an air

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tank (Schechter, 1999). The timing of the opening and closing of this valve is driven by an additional camshaft. The additional valve could also be electrically actuated (*Figure 1*).

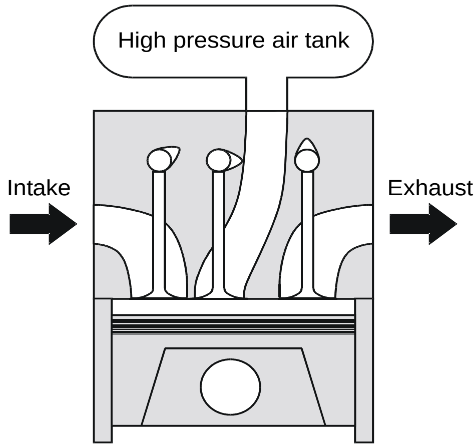

Figure 1: Each combustion chamber is connected to a high-pressure air tank through an additional valve and pipe

With this concept, in addition to the conventional internal combustion engine operation, several new operating modes must be considered (Higelin et al., 2001):

2.1 Pneumatic motor operation

The hybrid pneumatic-combustion engine can produce torque from 0 rpm, operating as a pneumatic motor powered by the high-pressure air tank. As long as the air tank has sufficient air pressure available, this potential energy can be converted to mechanical energy on the crankshaft at any time, just by opening the charging valve.

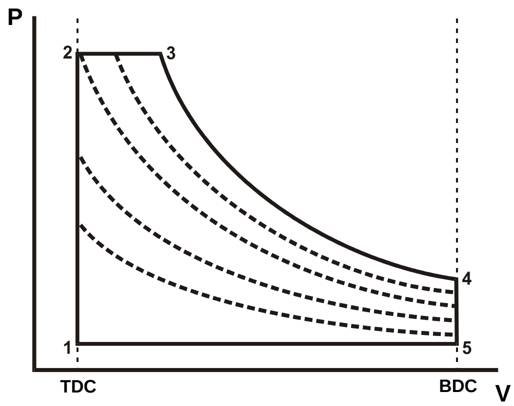

Figure 2. Idealized P-V diagram of pneumatic motor operation for several air charges

The pneumatic motor cycle consists of three steps: cylinder charging, expansion and exhaust. The full cycle is completed within one crankshaft revolution: it is a two-stroke cycle. *Figure 2* displays the idealized P-V diagram:

Cylinder charging: The charging valve is opened from 1 to 3. The air tank discharges into the cylinder between 1 and 2, where the cylinder reaches the tank pressure. From 2 to 3 the discharge is continued at a constant pressure (assuming the tank volume is much larger than the cylinder volume). The work of the pneumatic motor cycle can be continuously varied by adapting the timing of the charging valve closure (*Figure 2*, dashed lines). For low load operation, points 2 and 3 are superimposed and the corresponding pressure might be lower than the tank pressure.

Expansion: From 3 to 4 the charging valve is closed and the air charge expands. The expansion stroke (2 to 4) produces the work of the cycle. To obtain a full expansion of the air charge, the timing of the charging valve closure must be optimized in order to superimpose points 4 and 5.

Exhaust: The exhaust valve opens at point 4. From 4 to 5 the cylinder discharges into the exhaust pipe and the potential energy of the air at point 4 is lost. From 5 to 1 the exhaust stroke expels the remaining air into the exhaust pipe at atmospheric pressure.

The two stroke pneumatic motor operating mode leads to high torque output as well as low friction losses. During pneumatic motor operation, no fuel is injected into the intake ports.

This operating mode can be used to completely suppress the idling operation of conventional engines by shutting down the engine at each stop. After a stop, the engine can be rapidly brought to normal speed and start using the high level of torque provided by the pneumatic motor operating mode. It is well known that engine operation without idling phases can substantially improve fuel economy in typical city driving.

Due to the high level of torque available during pneumatic motor operating, an alternative starting mode could be to start the engine with clutch (or automatic transmission) engaged. So, the engine operates in pneumatic mode for a few cycles and switches to combustion mode as soon as the engine speed is high enough. With this strategy, the vehicle starting is brought from a two-step process (engine starting and engine speed going up to minimum operating speed, then transmission engaging) to a one-step process. In this mode, the vehicle starting process can be as smooth as with an electric motor if the number of cylinders is high enough.

The pneumatic motor operating mode can also be used for low power in-city cruising with zero pollutant emissions. However, to get a sufficient operating range, the size of the air tank must be appropriately dimensioned.

2.2 Pneumatic pump operation

One of the largest weaknesses of internal combustion engines is that the thermodynamic cycle cannot be reversed: i.e. you cannot produce air and fuel from combustion products by providing reverse torque to the crankshaft. As a result, the kinetic energy of a vehicle using a conventional internal combustion engine cannot be recovered into chemical energy during negative torque, i.e. braking situations: the kinetic energy must then be dissipated into heat through the friction of the braking system.

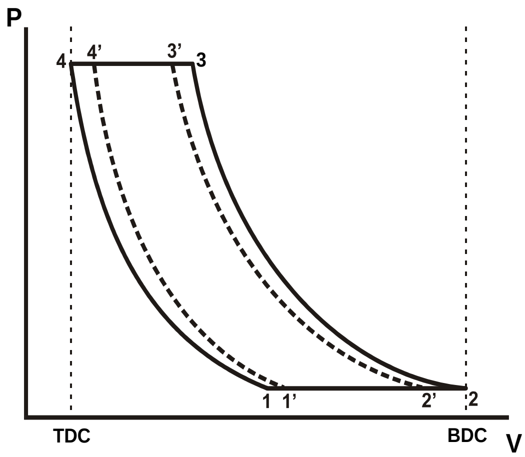

Figure 3. Idealized P-V diagram of pneumatic pump operation for two intake and two charging valve timings

The hybrid pneumatic-combustion engine can produce negative torque, i.e. consume torque from the driveline during a braking phase. The kinetic energy of the vehicle is converted to potential energy in the form of pressurized air pumped into the air tank.

The pneumatic pump cycle consists of four steps: intake, compression, air tank charging and expansion. Again, the full cycle is completed within one crankshaft revolution: it is a two-stroke cycle. *Figure 3* displays the idealized P-V diagram:

<u>Intake:</u> The intake valve opens at point 1. From 1 to 2 the cylinder is filled with fresh air from the intake port. The intake valve is closed at point 2 if the maximum air mass is needed. To reduce the air mass swept by the cycle, and conversely the cycle braking work, the intake valve closure can be retarded to 2' (*Figure 3*). In this case, part of the air mass inside the cylinder at point 2 is swept back into the intake manifold between points 2 and 2'.

<u>Compression:</u> From 2 to 3 all the valves are closed and the air charge is compressed up to the tank pressure. If the braking torque with full air mass is not high enough, the timing of the charging valve opening can be advanced to obtain a cycle with a larger area (*Figure 4*, points 1-2-5-6-4-1). The same effect can be obtained by retarding the opening of the charging valve, at the expense of a much higher maximum cylinder pressure (*Figure 4*, points 1-2-7-8-4-1)

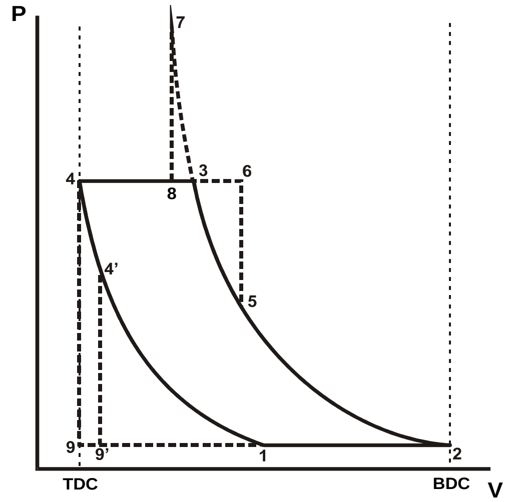

Figure 4. Idealized P-V diagram of pneumatic pump operation for three timings of the charging valve opening

<u>Air tank charging:</u> From 3 to 4 the charging valve is open. The end of the compression stroke is used to expel compressed air from the cylinder to the air tank. The air tank charging performed is at nearly constant pressure, assuming the volume of the air tank is much larger than the displacement of the engine.

<u>Expansion:</u> From 4 to 1, all the valves are closed. The compressed air that could not be pushed into the air tank expands to atmospheric pressure when the intake valve opens (point 1). To lower the braking torque, the closure of the charging valve can be retarded to 4' (*Figure 3*). In this case, part of the air mass inside the air tank at point 4 is swept back into the cylinder between points 4 and 4'. To enhance the braking torque, the expansion stroke can be completely or partially removed (*Figure 4*). In this case, the exhaust valve is opened at point 4 (or 4' for a partial expansion) and closed at point 1. The cycle then becomes 1-2-3-4-9-1 or 1-2-3-4-4'-9'-1 (see *Figure 4*).

The pneumatic pump operating mode can be used for regenerative braking. The braking torque as well as the mass of compressed air stored into the air tank can be continuously adjusted by varying the intake, charging and exhaust valve timings.

2.3 Supercharged engine operation

During strong transient accelerations or short-term high power output, the compressed air contents of the high-pressure tank can be used to supercharge the internal combustion engine. The supercharging pressure can be quite high and is only limited by the compressed air tank pressure.

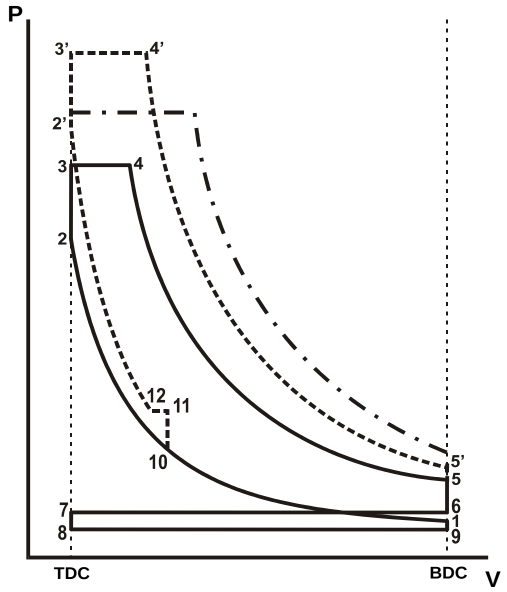

Figure 5. Idealized P-V diagram of engine supercharging with air from the high-pressure tank

The positive effect of this supercharging capability is that the engine design can be optimized for the maximum power meeting 80 to 90% of the driving situations without supercharging. The remaining 10 to 20% high power output operating conditions are met by transient supercharging with the compressed air from the high-pressure tank. Thus, the engine can be downsized, allowing a higher mean operating efficiency over the whole range of operating conditions.

The pneumatically supercharged cycle consists of several steps: intake, compression 1, supercharging, compression 2, combustion, expansion and exhaust. *Figure 5* displays the idealized P-V diagram:

Exhaust and intake: The pneumatic supercharged cycle is identical to a conventional four-stroke engine cycle during the exhaust (*Figure 5*, points 5-6-7) and intake (*Figure 5*, points 8-9-1) process.

Combustion and expansion: The combustion and expansion phases of the supercharged cycle (*Figure 5*, points 2'-3'-4'-5') display the same shape as the non-supercharged cycle (*Figure 5*, points 2-3-4-5). The higher level of air mass inside the cylinder during the supercharged combustion and expansion phase (*Figure 5*, dashed line) comparatively to the atmospheric cycle (*Figure 5*, solid line) provides, as expected, a higher IMEP, but also a higher level of maximum cylinder pressure. This higher pressure can be overcome by retarded combustion timing, at the expense of the global cycle efficiency (*Figure 5*, dash-dotted line).

Compression: The main difference between a conventional internal combustion engine cycle and the pneumatically supercharged cycle resides in the compression stroke. This compression stroke is composed of three phases:

- After the closing of the intake valve, a conventional compression phase starts, bringing the in-cylinder air to a pressure below the tank pressure (*Figure 5*, points 1-10).
- Then, the charging valve opens, allowing additional air to flow from the high-pressure tank into the combustion chamber (*Figure 5*, points 10-11-12).
- At the closing of the charging valve, a second compression stroke is performed (*Figure 5* points 12-2').

This strategy offers three main differences compared to a conventional supercharged cycle:

- By using the conventional intake and exhaust stroke, the tank only has to provide an amount of air in excess of the cylinder charge at atmospheric pressure. Consequently, the lifetime of the high pressure tank air charge can be greatly enhanced (2 to 4 times).
- Unlike a conventional supercharged cycle, where the complete air mass is in the cylinder when the intake valve closes, the pneumatically supercharged cycle can provide a higher high-pressure cycle IMEP due to the dual stage compression stroke. Indeed, the first part of the compression stroke is performed with a lower air mass, i.e. lower pressure, providing additional work to the cycle. To maximize IMEP, the supercharging process should be performed as late as possible. The optimal strategy would be to close the charging valve when cylinder pressure reaches tank pressure, and to determine the timing of the charging valve opening for the desired additional air mass.
- On the downside, the exhaust pressure of the hybrid engine being higher than the intake pressure, the low-pressure cycle IMEP is negative. A conventionally supercharged engine benefits from a positive low-pressure cycle.

2.4 Undercharged engine operation

The energy recovered during the braking phases is usually not enough to provide a full air tank charging. It is generally not possible to get the same state of charge (SOC) of the high-pressure air tank at the end of a test cycle than at the beginning. Therefore, to fully benefit from the hybrid pneumatic-combustion engine concept, an operating mode allowing the air tank charging while in combustion mode must be added.

The undercharged engine operation mode consists of taking away part of the fluid enclosed in the combustion chamber during the compression to charge the high-pressure air tank.

The undercharged engine cycle consists of several steps: intake, compression 1, charging, compression 2, combustion, expansion and exhaust. *Figure 6* displays the idealized P-V diagram:

Exhaust and intake: The undercharged engine cycle is identical to a conventional four-stroke engine cycle during the exhaust (*Figure 6*, points 5-6-7) and intake (*Figure 6*, points 8-9-1) phase.

Combustion and expansion: The combustion and expansion phases of the undercharged cycle (*Figure 6*, points 2'-3'-4'-5') display the same shape as a conventional four-stroke engine cycle (*Figure 6*, points 2-3-4-5). The lower level of air mass inside the cylinder during the undercharged combustion and expansion phase (*Figure 6*, dashed line) comparatively to the atmospheric cycle (*Figure 6*, solid line) provides, as expected, a lower IMEP, but also a lower level of maximum cylinder pressure. To optimize the global cycle efficiency, the start of combustion can be advanced to reach a higher peak cylinder pressure.

Compression: The main difference between a conventional internal combustion engine cycle and the undercharged cycle resides in the compression stroke. This compression stroke is composed of three phases:

- After the closing of the intake valve, a conventional compression phase starts, bringing the in-cylinder air to the same pressure as the tank (*Figure 6*, points 1-10).
- Then, the charging valve opens, allowing air from the combustion chamber to flow to the high-pressure tank during part of the compression stroke (*Figure 6*, points 10-12).
- At the closing of the charging valve, a second compression stroke is performed (*Figure 6*, points 12-2').

This strategy offers two main differences compared to a conventional low-load operating mode:

- The first compression being performed with the full air charge, the undercharged cycle is always performed with a wide-open throttle. In this case, points 7 and 8 as well as 1, 6 and 9 (*Figure 6*) are superimposed, providing zero pumping losses during the low-pressure cycle. The global IMEP is increased, as well as the global efficiency.
- On the downside, during the first compression stroke (*Figure 6*, points 1-10), work is drawn from the crankshaft to compress the air mass that will be pumped into the high-pressure tank. Thus, the high-pressure cycle IMEP is lower than with a conventional low-load cycle.

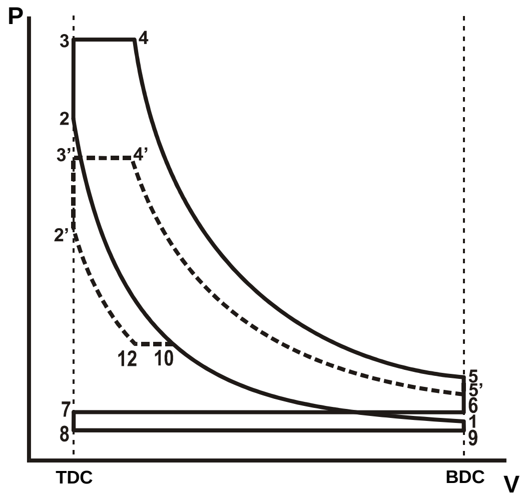

Figure 6. Idealized P-V diagram of undercharged engine operation

3. Hybrid Engine Model

To be able to compute the global efficiency over a test cycle, the efficiency and IMEP of each operating mode must be expressed as a function of tank pressure, as well as intake, exhaust and charging valve timing:

3.1 Pneumatic motor

The transformation from 1 to 2 being at constant volume without heat exchange (*Figure 2*), application of the first law yields:

$$m_2 \cdot u_2 - m_1 \cdot u_1 = (m_2 - m_1) \cdot h_t \tag{1}$$

$$T_2 = \frac{p_t}{\frac{p_t - p_e}{\gamma \cdot T_t} + \frac{p_e}{T_e}} \tag{2}$$

$$m_2 - m_1 = \frac{V_m}{r} \cdot \left(\frac{p_t}{T_2} - \frac{p_e}{T_r} \right) \tag{3}$$

The process from 2 to 3 being isobaric at air tank pressure (*Figure 2*), it comes:

$$m_3 \cdot u_3 - m_2 \cdot u_2 = = (m_3 - m_2) \cdot h_t - p_t \cdot (V_3 - V_m) \quad (4)$$

$$m_3 = \frac{p_t \cdot V_3}{r \cdot T_t} - m_2 \cdot \left(\frac{T_2}{\gamma \cdot T_t} - 1 \right) - \frac{p_t \cdot V_m}{c_p \cdot T_t} \quad (5)$$

$$T_3 = \frac{p_t \cdot V_3}{m_3 \cdot r} \quad (6)$$

$$W_{23} = -p_t \cdot (V_3 - V_m) \quad (7)$$

Since the expansion process from 3 to 4 is considered adiabatic and reversible (*Figure 2*):

$$T_4 = T_3 \cdot \left(\frac{V_3}{V_M} \right)^{\gamma - 1} \quad (8)$$

$$W_{34} = m_3 \cdot (u_4 - u_3) \quad (9)$$

The exhaust stroke being isobaric (*Figure 2*), the produced mechanical work can be expressed:

$$W_{51} = -p_e \cdot (V_m - V_M) \quad (10)$$

The specific work of the pneumatic motor operating mode can be written as the quotient of work produced (7, 9, 10) and mass provided by the tank (3, 5):

$$\xi = \frac{m_3 \cdot (u_4 - u_3) - p_t \cdot (V_3 - V_m) - p_e \cdot (V_m - V_M)}{m_3 - m_1} \quad (11)$$

3.2 Pneumatic pump

The intake stroke being isobaric and isothermal at intake pressure and temperature (*Figure 3*):

$$m_2 - m_1 = \frac{p_i \cdot (V_2 - V_1)}{r \cdot T_i} \quad (12)$$

$$V_1 = V_m \cdot \left(\frac{p_t}{p_i} \right)^{\frac{1}{\gamma}} \quad (13)$$

$$W_{12} = -p_i \cdot (V_2 - V_1) \quad (14)$$

The compression process from 2 to 3 is considered adiabatic and reversible (*Figure 3*):

$$T_3 = T_i \cdot \left(\frac{p_i}{p_t} \right)^{\frac{1-\gamma}{\gamma}} \quad (15)$$

$$W_{23} = m_2 \cdot (u_3 - u_2) \quad (16)$$

The tank charging process from 3 to 4 is considered isobaric and isothermal without heat exchange (*Figure 3*):

$$T_4 = T_3 \quad (17)$$

$$W_{34} = -p_t \cdot (V_m - V_3) \quad (18)$$

The expansion process from 4 to 1 is considered adiabatic and reversible (*Figure 3*):

$$W_{41} = m_1 \cdot (u_1 - u_4) \quad (19)$$

The specific mass of the pneumatic pump operating mode can be written as the quotient of mass pumped into the tank (12) and mechanical work consumed (14, 16, 18, 19):

$$\zeta = \frac{m_2 - m_1}{\left(\begin{array}{l} -p_i \cdot (V_2 - V_1) - p_t \cdot (V_m - V_3) \\ + m_1 \cdot (u_1 - u_4) + m_2 \cdot (u_3 - u_2) \end{array} \right)} \quad (20)$$

3.3 Pneumatic supercharging

The compression from 1 to 10 is considered isentropic without heat exchange (*Figure 5*), hence:

$$T_{10} = T_1 \cdot \left(\frac{V_M}{V_{10}} \right)^{\gamma - 1} \quad (21)$$

$$p_{10} = p_1 \cdot \left(\frac{V_M}{V_{10}} \right)^{\gamma} \quad (22)$$

$$W_{110} = m_1 \cdot (u_{10} - u_1) \quad (23)$$

The flow through the charging valve is modeled by a constant volume transformation from 10 to 11 followed by a constant pressure transformation from 11 to 12 (*Figure 5*):

$$T_{11} = \frac{p_t}{\frac{p_t - p_{10}}{\gamma \cdot T_t} + \frac{p_{10}}{T_{10}}} \quad (24)$$

$$m_{11} - m_1 = \frac{V_{10}}{r} \cdot \left(\frac{p_t}{T_{11}} - \frac{p_{10}}{T_{10}} \right) \quad (25)$$

$$m_{12} \cdot u_{12} - m_{11} \cdot u_{11} = = (m_{12} - m_{11}) \cdot h_t - p_t \cdot (V_{12} - V_{10}) \quad (26)$$

$$m_{12} = \frac{p_t \cdot V_{12}}{r \cdot T_t} - - m_{11} \cdot \left(\frac{T_{11}}{\gamma \cdot T_t} - 1 \right) - \frac{p_t \cdot V_{10}}{c_p \cdot T_t} \quad (27)$$

$$T_{12} = \frac{p_t \cdot V_{12}}{m_{12} \cdot r} \quad (28)$$

$$W_{1112} = -p_t \cdot (V_{12} - V_{10}) \quad (29)$$

Again, the second stage of the compression process is considered isentropic without heat exchange (*Figure 5*), it comes:

$$T_{2'} = T_{12} \cdot \left(\frac{V_{12}}{V_m} \right)^{\gamma-1} \quad (30)$$

$$p_{2'} = p_t \cdot \left(\frac{V_{12}}{V_m} \right)^{\gamma} \quad (31)$$

$$W_{122'} = m_{12} \cdot \left(u_{2'} - u_{12} \right) \quad (32)$$

The combustion process is modeled by heat transferred to the working gas:

$$Q = \eta_c \cdot m_f \cdot Q_{LHV} \quad (33)$$

The first part (from 2’ to 3’) of the combustion process is modeled by a constant volume process, the second part (3’ to 4’) is modeled by a constant pressure transformation (*Figure 5*):

$$Q_v = \varepsilon_v \cdot Q \quad (34)$$

$$Q_p = \varepsilon_p \cdot Q = (1 - \varepsilon_v) \cdot Q \quad (35)$$

$$T_{3'} = T_{2'} + \frac{Q_v}{m_{12} \cdot c_v} \quad , \quad p_{3'} = \frac{m_{12} \cdot r \cdot T_{3'}}{V_m} \quad (36)$$

$$T_{4'} = T_{3'} + \frac{Q_p}{m_{12} \cdot c_p} \quad , \quad V_{4'} = \frac{m_{12} \cdot r \cdot T_{4'}}{p_{3'}} \quad (37)$$

$$W_{3'4'} = -p_{3'} \cdot \left(V_{4'} - V_m \right) \quad (38)$$

The expansion process from 4’ to 5’ is considered adiabatic and reversible:

$$T_{5'} = T_{4'} \cdot \left(\frac{V_{4'}}{V_M} \right)^{\gamma-1} \quad (39)$$

$$W_{4'5'} = m_{12} \cdot \left(u_{5'} - u_{4'} \right) \quad (40)$$

The pumping work of the low pressure cycle can be expressed as:

$$W_p = (p_e - p_i) \cdot (V_M - V_m) \quad (41)$$

The supercharged hybrid pneumatic engine efficiency can be written as the quotient of work produced (23, 29, 32, 38, 40, 41) and thermal energy provided by the combustion (28):

$$\eta_s = \frac{\begin{pmatrix} m_1 \cdot (u_{10} - u_1) - p_t \cdot (V_{12} - V_{10}) \\ + m_{12} \cdot (u_{2'} - u_{12} + u_{5'} - u_{4'}) \\ - p_{3'} \cdot (V_{4'} - V_m) + (p_e - p_i) \cdot (V_M - V_m) \end{pmatrix}}{m_f \cdot Q_{LHV}} \quad (42)$$

3.4 Pneumatic undercharging

As can be seen on *Figure 6*, the undercharged cycle only differs from the supercharged cycle in that point 10 and point 11 are superimposed at tank pressure, hence:

$$V_{10} = V_M \cdot \left(\frac{p_1}{p_t} \right)^{\frac{1}{\gamma}} \quad (43)$$

$$T_{11} = T_{10} \ , \ p_{11} = p_t \ , \ m_{11} = m_1 \quad (44)$$

In this case, the undercharged hybrid pneumatic engine efficiency has the same expression as the supercharged efficiency:

$$\eta_u = \frac{\begin{pmatrix} m_1 \cdot (u_{10} - u_1) - p_t \cdot (V_{12} - V_{10}) \\ + m_{12} \cdot (u_{2'} - u_{12} + u_{5'} - u_{4'}) \\ - p_{3'} \cdot (V_{4'} - V_m) + (p_e - p_i) \cdot (V_M - V_m) \end{pmatrix}}{m_f \cdot Q_{LHV}} \quad (45)$$

3.5 Conventional combustion

The conventional combustion cycle has a single compression stroke at constant mass (*Figure 5*):

$$p_2 = p_1 \cdot \left(\frac{V_M}{V_m} \right)^{\gamma} \quad (46)$$

$$T_2 = T_1 \cdot \left(\frac{V_M}{V_m} \right)^{\gamma-1} \quad (47)$$

$$W_{12} = m_1 \cdot (u_2 - u_1) \quad (48)$$

The conventional combustion cycle efficiency can be written as:

$$\eta_c = \frac{\begin{pmatrix} m_1 \cdot (u_2 - u_1 + u_{5'} - u_4) \\ - p_3 \cdot (V_4 - V_m) + (p_e - p_i) \cdot (V_M - V_m) \end{pmatrix}}{m_f \cdot Q_{LHV}} \quad (49)$$

3.6 Engine friction

To compute the effective torque produced by the engine from the indicated work, the friction losses of the engine must be evaluated:

$$W_f = W_c - W_e \quad (50)$$

For this study a simple model is enough to perform the comparison between conventional and pneumatic hybrid engine operation. From the many friction models available, the Bidan model has been chosen for its simplicity (Chamaillard et al., 2001). This black box model estimates friction losses in a global way. The only variable parameter is the engine rotational speed (N):

$$W_f = (V_M - V_m) \cdot \left(a + b \cdot N + c \cdot N^2 \right) \quad (51)$$

Parameters a, b, c have been determined from engine measurements.

3.7. High pressure tank

The air storage tank is considered adiabatic and of constant volume. Therefore, the mass and energy balance can be written:

$$m_t = \sum \Delta m_t \tag{52}$$

Where Δm_t represents the variation of mass inside the tank for one engine cycle and is equal to the variation of mass inside the cylinder during the tank charging and engine charging processes (Equations 3, 12 and 25).

To determine the tank temperature, two distinct situations must be considered:

In the case of tank charging, the variation of tank temperature during a single cycle can be written:

$$\Delta T_t = \frac{\gamma \cdot T_c - T_t}{m_t} \Delta m_t \tag{53}$$

In the case of tank discharging:

$$\Delta T_t = \frac{(\gamma - 1) \cdot T_t}{m_t} \Delta m_t \tag{54}$$

The state of charge of the high pressure air tank (m_t, T_t, p_t) is computed from equations 52, 53, 54 for each engine cycle.

4. Driving Cycle Simulation

Unlike the efficiency of conventional engines, the global efficiency of the hybrid pneumatic-combustion engine cannot simply be computed from the thermodynamic cycles and the thermo-physical properties of the working fluid.

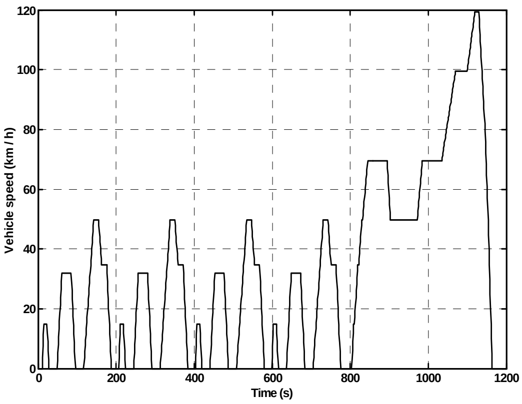

Figure 7. New European Driving Cycle (NEDC)

During the hybrid engine operation over a test cycle, operating modes must be switched to achieve best global efficiency as well as tank refilling. The choice of the current operating mode is performed at any time by the engine controller. Thus, the global efficiency of the engine over the test cycle not only depends on the thermodynamic cycles and the thermo-physical properties of the fluid, but also on the test cycle itself and the engine control strategies.

The driving cycle consists of a vehicle speed versus time graph that must be followed by the driver. The New European Driving Cycle (NEDC) (European Directive, 1990) has been used for this study (*Figure 7*).

4.1 Vehicle and driveline model

To be able to follow the driving cycle, a dynamic vehicle model has to be used. It can be expressed by:

$$\dot{x} = \int \frac{F_w - F_r - F_a - F_b}{m_v} dt \tag{55}$$

Where F_w represents the driving force at the wheels, F_r the rolling drag force, F_a the aerodynamic drag force and F_b the braking force:

$$F_w = \frac{k_d \cdot \eta_d}{4 \cdot \pi \cdot r_w} \cdot W_e \tag{56}$$

$$F_r = m_v \cdot g \cdot k_r \tag{57}$$

$$F_a = \frac{\rho \cdot A_f \cdot C_x \cdot \dot{x}^2}{2} \tag{58}$$

$$F_b \geq 0 \tag{59}$$

The braking force only exists to close the force balance in case of strong deceleration, when engine braking is not sufficient.

In the case of a conventional engine simulation, the engine cannot produce torque at 0 rpm. Therefore, a clutch model must be added:

$$\omega_e > \omega_d \tag{60}$$

$$P_s = \frac{W_e \cdot |\omega_e - \omega_d|}{4 \cdot \pi} \tag{61}$$

Where ω_e and ω_d respectively represent the engine and driveline speeds, P_s the engine power dissipated into heat during clutch slipping.

4.2 Operating mode selection rules

During the conventional internal combustion engine driving cycle simulation, the following set of rules has been adopted:

- If engine speed drops below 1000 rpm, downshift one gear. If engine speed goes beyond 2500 rpm, shift one gear up.
- If engine speed drops below 1000 rpm in first gear, the clutch is slipping.

During the hybrid pneumatic-combustion engine driving cycle simulation, the following set of rules has been used:

- When the vehicle starts, use pneumatic motor mode as long as the tank is not empty. Then switch to undercharge mode to fill up the tank to 80% of its capacity.
- If the tank is at 80% of its capacity and speed is greater than 40 km/h, use conventional combustion mode.
- In braking situations, switch to pneumatic pump mode.
- If full load conventional combustion mode torque is not high enough, switch to supercharged mode.

The global average efficiency can then be computed from the total fuel consumption and the total work produced by the powertrain.

5. Simulation results

5.1 Model parameters

The engine simulation parameters can be found in TABLE I. The friction parameters of the engine have been fitted to a real two-liter engine (Chamaillard et al., 2001). The air tank

TABLE I. ENGINE SIMULATION PARAMETERS

$V_M - V_m$	2000 cm^3
V_M / V_m	10
r	288 J/K·kg
γ	1.4
p_i	0.1 MPa
p_e	0.11 MPa
T_i	300 K
Q_{LHV}	42500 kJ/kg
η_c	0.8
ε_p	0.5
ε_v	0.5
a	0.776
b	0.189
c	0.0209

TABLE II. AIR TANK PARAMETERS AND INITIAL VALUES

Vt	50 dm^3
p_t	2 MPa
T_t	300 K

TABLE III. VEHICLE SIMULATION PARAMETERS

m_v	800 kg
k_r	0.00863
k_d	13.45 7.57 5.01 3.77 2.84
r_w	0.282 m
η_d	0.8
ρ_a	1.293 kg/m^3
A_f	2.06 m^2
C_x	0.312

volume as well as its initial state of charge can be found in TABLE II and the vehicle simulation parameters can be found in TABLE III. The vehicle data (aerodynamic parameters, gear ratios, wheel size) have been taken from a real, small size car.

5.2 Conventional operation results

On *Figure 8* it can be seen that the idle phases account for almost one-third of the total driving cycle time (373.8 seconds). The energy dissipated during this time could be saved if the engine was stopped.

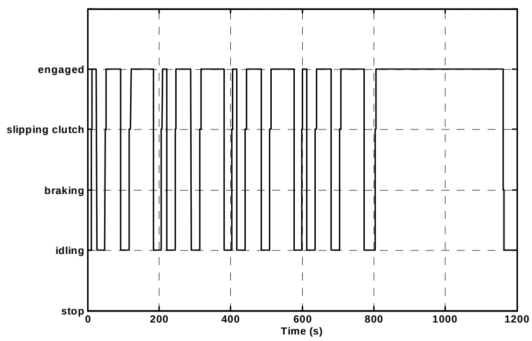

Figure 8. Conventional engine operating modes

Each start comprises a phase with a slipping clutch. Again, this energy could be saved if the internal combustion engine could provide torque at very low speed. But this amount of energy is very low compared to the total energy consumed (TABLE IV).

TABLE IV. ENERGY BALANCE OF CONVENTIONAL ENGINE OPERATION

Total energy consumed	26.4 MJ
Total mechanical energy provided	2.10 MJ
Total braking energy	0.92 kJ
Total clutch slipping energy	4.19 kJ
Global driving cycle efficiency	7.96%

The slowdown phases being relatively smooth on the NEDC driving cycle, most of the slowdown can be performed with engine braking alone. The brakes are activated only once during the final slowdown of the extra-urban part of the cycle (*Figure 8*). So, it seems that the pneumatic pump mode would not be really useful on this kind of driving cycle (TABLE IV).

5.3 Hybrid operation results

The hybrid engine operation displays a great improvement in global driving cycle efficiency compared to the conventional engine operation (TABLE IV). The fuel economy is improved by more than 15% (TABLE V).

TABLE V. ENERGY BALANCE OF HYBRID ENGINE OPERATION

Total energy consumed	22.4 MJ
Total mechanical energy provided	2.10 MJ
Braking energy recovered	0.46 kJ
Global driving cycle efficiency	9.38%

On *Figure 9*, it can be seen that with the engine chosen for this study, supercharged mode is never necessary. To further enhance the global efficiency, the engine should be downsized to a one-liter displacement. The friction losses could then be reduced and the thermodynamic efficiency of the cycle could be further improved.

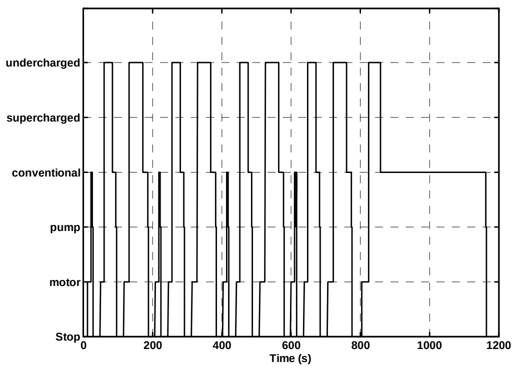

Figure 9. Hybrid engine operating modes

The amount of energy recovered during the slowdown phases is very small. This can be attributed to the smooth slowdown of the driving cycle: most of the time the friction losses of the engine are sufficient to slowdown the vehicle considered. A heavier vehicle should allow more energy to be recovered.

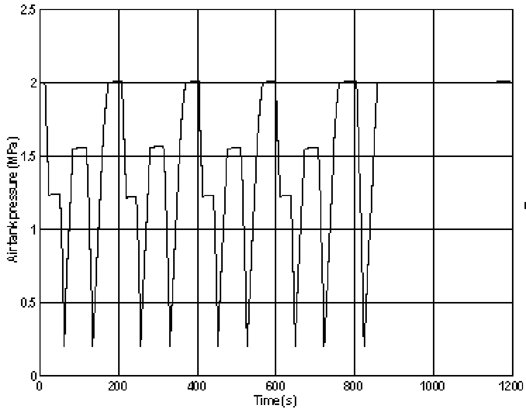

Figure 10: Air tank pressure

On *Figure 10*, it must be noted that the final tank pressure is the same as the initial pressure. Thus, the energy balance is not biased by the air tank contribution. Further optimization could be achieved by selecting a more optimal tank volume as well as minimum and maximum tank pressure thresholds.

On *Figure 11*, the main differences between conventional and hybrid operation are apparent. At each stop, the hybrid engine displays a plateau in the fuel mass consumed where the conventional engine is idling. When the hybrid engine is operating in combustion mode, the fuel consumption slope is steeper than in conventional mode. Indeed, the hybrid engine needs to refill the air tank each time it switches to combustion (undercharging) mode.

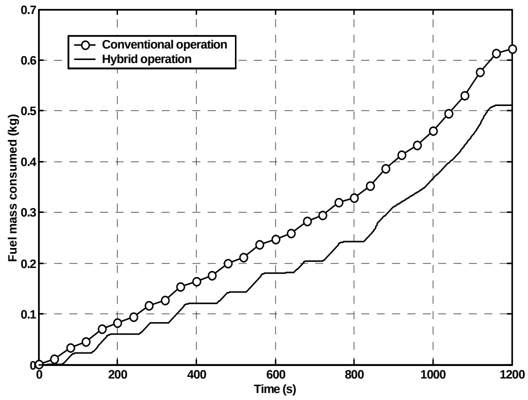

Figure 11. Fuel mass consumed

6. Conclusion

A new kind of hybrid pneumatic-combustion engine has been modeled using simple thermodynamic cycles. To be able to compare this engine to a conventional internal combustion engine, the simulation had to take into account the driving cycle, vehicle configuration as well as the mode switching strategy.

The simulation of a transient driving cycle shows a great improvement of the global efficiency with the new hybrid concept. The global fuel consumption could be reduced by around 15% without any specific optimization.

Regenerative braking is only applicable for quite harsh braking situations. In most usual driving situations, the engine internal friction is sufficient to dissipate the kinetic energy during a deceleration phase.

Further improvement could be achieved by using a downsized engine configuration. Tank volume, tank pressure thresholds and mode switching strategy optimization should also lead to significant reduction of fuel consumption.

Nomenclature

Symbols

A_f : frontal area
a, b, c : friction coefficients
c : specific heat
C_x : drag coefficient
F : force
g : acceleration of gravity
h : enthalpy
k : gear ratio
m : mass
N : engine speed
P : power
p : pressure
Q : heat
r : gas constant, radius
T : temperature, torque
u : internal energy
V : volume
W : work
x : distance

Greek symbols

γ : specific heat ratio
ε : heat fraction
η : efficiency
ξ : pneumatic motor efficiency
ζ : pneumatic pump efficiency
ω : rotational speed

Subscripts

a : aerodynamic
b : brake
c : combustion, cycle
d : driveline
e : exhaust, engine
f : fuel, friction
i : intake
LHV : lower heating value
M : maximum
m : minimum
p : at constant pressure
r : rolling
s : slipping
t : tank
v : at constant volume, vehicle
w : wheel

Acronyms

DI: Direct Injection
TDC: Top Dead Center
BDC: Bottom Dead Center
IMEP: Indicated Mean Effective Pressure
NEDC: New European Driving Cycle
SOC: State Of Charge

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