PNEUMATIC HYBRIDIZATION OF DIESEL ENGINE IN A HYBRID WIND-DIESEL INSTALLATION WITH COMPRESSED AIR ENERGY STORAGE

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ABSTRACT
In this paper, we are studying an innovative solution to reduce fuel consumption for electricity production in remote areas. Historically, in these areas electricity is provided by Diesel generators. The cost of energy is therefore very high not only because of inherent cost of technology but also because of transportation costs. On the other hand, use of fossil fuels for electricity generation is a significant source of Greenhouse Gas emissions. The use of hybrid systems that combine renewable sources and diesel generators, allows reducing fuel consumption, improves the operation cost and creates therefore environmental benefits. Adding a storage element to the hybrid system above increases fuel saving as it allows an increased penetration of the wind power in the overall system capacity. In a previous work, we demonstrated that Compressed Air Energy Storage (CAES) is a serious solution thanks to its low cost, high power density and reliability. Pneumatic Hybridization of the Diesel engine consists to introduce the CAES through the admission valve. We have proven that we can improve the combustion efficiency and therefore the fuel consumption by optimizing Air/Fuel ratio thanks to the CAES assistance. As a continuation of these previous analyses, we studied the effect of controlling the pressure at the inlet and the exhaust of the Diesel engine on the thermodynamic cycle and evaluated the potential on fuel consumption reduction.

Keywords: Pneumatic Hybridization, Wind Energy, Diesel Engine, Wind-Diesel Hybrid System, Compressed Air Energy Storage.

1. INTRODUCTION

Most of the remote and isolated communities or technical installations (communication relays, meteorological systems, tourist facilities, farms, etc.) that are disconnected from national electric distribution grids rely on diesel engines to generate electricity (Ibrahim 2007). Diesel generated electricity is more expensive in itself than large electric production plants (gas, hydro, nuclear, wind), even prior to taking into account the transport and environmental costs associated with this type of energy.

In Canada, approximately 200,000 people live in more than 300 remote communities (Yukon, Northwest Territories, Nunavut, etc.) that use diesel generated electricity, which is responsible for the emission of 1.2 million tons of greenhouse gases annually (Ibrahim 2007). In Quebec alone, there are over 14,000 subscribers scattered in about forty communities that disconnected from the main grid. Each community constitutes an autonomous network that uses diesel generators.

In Quebec, the total production of diesel power generating units is approximately 300 GWh per year. The operation of these diesel generators is extremely expensive due to the high oil price and transportation costs. Indeed, as the fuel has to be delivered to remote locations, some of them only reachable during the summer by boat, the cost of electricity in 2007 reached more than 50 cent/kWh in some communities, while in the rest of the province the electricity tariff was approximately 6 cent/kWh (Ibrahim 2007). The resulting deficit, which is shared among all Quebecers, is far from being negligible. In 2004, the autonomous networks accounted for 144 MW of installed power, and the consumption was established at 300 GWh. Hydro-Quebec, the provincial utility, estimated at approximately CAD $133 million the annual loss resulting from the difference between the diesel electricity production cost and the uniform electricity tariff (Ibrahim 2007).

Moreover, diesel electricity production is considered ineffective, presents significant environmental risks (spilling), pollutes local air and contributes to global warming. Overall, we estimate at 140,000 tons the annual GHG emission resulting from the use of diesel generators for the autonomous networks in Quebec. This is equivalent to the GHG emitted by 35,000 cars during one year.

While they require relatively little investment, diesel power generating units are generally expensive to operate and maintain, particularly when they function regularly with a partial load (Ibrahim 2007). The use of diesel power generators under weak operating factors accelerates wear and increases fuel consumption (Ibrahim 2007).

The use of Hybrid Wind-Diesel Generators solves a part of the problems listed above, but its efficiency is limited by wind intermittency. Highly fluctuating wind energy requires the Diesel engine running on idle most of the time and dissipates a part of the wind power in order to control the balance between the energy production and the energy demand. Energy storage is the key to solve this issue, because it allows storing otherwise wasted energy and recover it later, when the Wind power is not sufficient to cover the demand. The use of storage allows an increased penetration of the wind power in the overall system capacity. The high penetration of wind energy means that the installed wind power exceeds the maximum load and, during periods of strong winds, diesel generators may be stopped while surplus of wind energy is stored. The overall energy provided by the Diesel generator is significantly decreased with beneficial economic and environmental results. In a previous work, we demonstrated that Compressed Air Storage (CAES) is a serious solution thanks to its low cost, high power density and reliability. Pneumatic Hybridization of the Diesel engine consists to introduce the CAES in the combustion chamber. We have proven that we can improve the combustion efficiency and therefore the fuel consumption through optimisation of Air/Fuel ratio. As a continuation of these previous analyses, we studied the effect of controlling the pressure at the inlet and the exhaust of the Diesel engine on the thermodynamic cycle and evaluated the potential on fuel consumption reduction

2. IMPROVING INDUSTRIAL DIESEL ENGINES EFFICIENCY

Due to their superior efficiency when compared to other thermal engines especially Spark Ignition engines, Diesel Engines are widely used for applications where high power concentration is required. For this reason, a great deal of research is conducted worldwide to reduce their pollutant emissions and to further improve their efficiency. Taking into account the green house effect that has been recognized the last years caused by increased CO2 emissions, it seems that they offer a reasonable solution for the minimization of this problem. The efficiency of the direct injection diesel engine has been considerably improved during the last decade mainly due to turbo-charging and downsizing technique.

Downsizing Diesel engines consists to reduce its displacement while providing the same maximum power. This is possible thanks to turbo-charging that increases the air filling and thus the specific power (i.e: maximum power per unit displacement) of the engine. Smaller engines have better efficiency than bigger engines, when working under the same load, thanks to better mechanical efficiency and reduced heat losses.

3. PNEUMATIC HYBRIDIZATION OF DIESEL ENGINES

Since the late 1990s, pneumatic hybridization of Internal Combustion Engines has been discussed, but mainly focusing for automotive application. The key idea is to use the engine also as a pump to recuperate the vehicle's kinetic energy during braking phases, and as an expansion motor for propulsion. This can be realized by connecting an air pressure tank to all cylinders via electronically controlled charge valves (Dönitz 2008) as shown in figure 1 – right (ibrabim 2010).

In addition to the recuperation capability, the idea offers the possibility of a rapid start/stop and permits to shift the engine's operating point to high efficiency zones. The latter can be achieved by operating half of the cylinders conventionally while the other half of the cylinders work in the *pump mode*. Previous concepts assumed two-stroke pump and pneumatic motor modes which is possible only if all valves of the cylinders can be variably actuated (Higelin 2002).

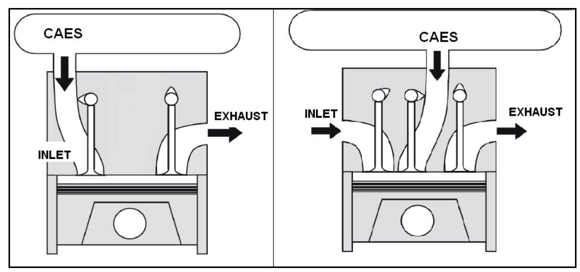

Figure 1: Pneumatic Hybrid ICE Engine through inlet valve (left), and through additional valve (right).

In our current study, we have focused on a more simple technique of Hybridizing Diesel engine. This technique consists in introducing the compressed air through the initial intake valve as shown in figure 1 – left (Ibrahim 2010). The main reason for this choice is to suggest a practical adaptation of existing Diesel power plants installations.

The compressed air is supposed to be previously generated by independent electric air compressor whose energy is provided by wind turbines, during periods of high Wind Power Penetration Rate (WPPR>1), as shown in figure 2 (Ibrahim 2010).

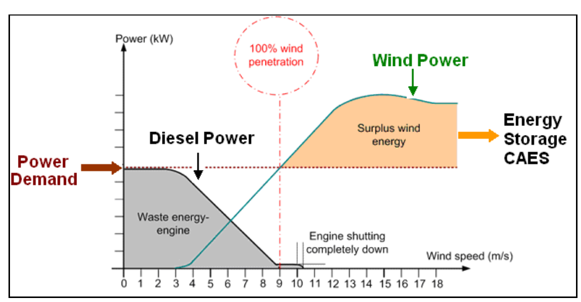

Figure 2: Variation of wind and diesel power with wind speed for a high-penetration Wind Diesel system

4. MATHEMATICAL MODEL

A zero dimensional mathematical model of the thermodynamic Diesel cycle has been developed to and used to conduct our parametric study. The mathematical models used are found in literature review (Heywood 1988; Lopes Correia da Silva 2007; Higelin 2002; Ibrahim 2007). We did not conduct any tests to calibrate and fit our model . We have used parameters that are found in literature review (Guibert 2002). The purpose actually of our study was to give orientations to our conception. In the future, a one dimensional modelling will be conducted to simulate the whole system in steady state mode and in transient mode.

Figure 3 shows the system considered to calculate the evolution of the thermodynamic characteristics (pressure, temperature, specific heat constant, etc…).

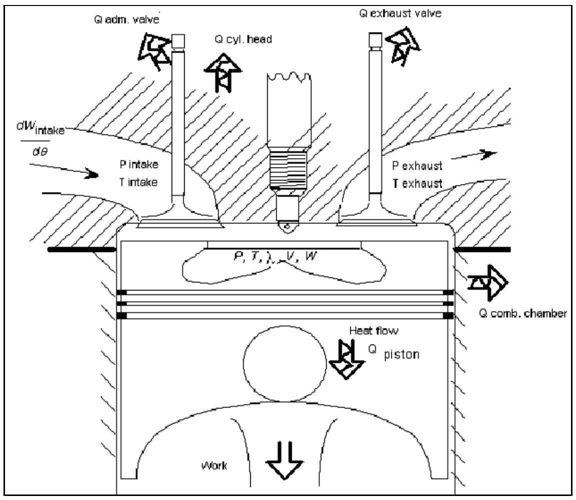

Figure 3: Direct injection Diesel engine simplified model

The hypotheses considered are the following:

1. The thermodynamic equilibrium is established for each step of calculation
2. The mass of gas is a homogeneous mixture of perfect gas. Its thermodynamic characteristics (enthalpy, intern energy, molar mass, etc…) are calculated by interpolation in JANAF thermo-chemical tables (third edition, 1985)
3. The mass transfers occur only through the admission and exhaust valves. The blow-by flow is neglected.
4. The admission back-flow is calculated and taken in consideration. The exhaust flow transferred back to the admission is readmitted in the cylinder at the exhaust temperature.
5. The fuel is injected in the combustion chamber at a constant temperature and it is immediately burned following Wiebe law.
6. The heat transfers occur through the five boundary limits (cylinder head, piston, cylinder wall, exhaust valve and admission valve) that are at a constant and uniform temperature.

4.1. Main equations

Main equations are issued from the mass and heat conservation equations as well as the ideal gas equations review (Heywood 1988).

$$\frac{dW}{dt}=\frac{dW_{intake}}{dt}+\frac{dW_{exhaust}}{dt}+\frac{dW_{inj}}{dt} \quad (1)$$

$$\frac{dW_{air}}{d\theta}=\frac{\frac{dW_{intake}}{d\theta}}{1+\frac{\lambda_{intake}}{\lambda_{st}}}+\frac{\frac{dW_{exhaust}}{d\theta}}{1+\frac{\lambda}{\lambda_{st}}} \quad (2)$$

$$\frac{d\lambda}{d\theta}=\frac{\frac{dW}{d\theta}-\left(1+\frac{\lambda}{\lambda_{st}}\right)\times\frac{dW_{air}}{d\theta}}{W_{air}}\times\lambda_{st} \quad (3)$$

$$\frac{dT}{d\theta}=\frac{A_1-\frac{\frac{\partial u}{\partial P}}{A_2}\times\left(\frac{\frac{dW}{d\theta}}{W}+\frac{\frac{\partial r}{\partial\lambda}}{r}\times\frac{d\lambda}{d\theta}-\frac{\frac{dV}{d\theta}}{V}\right)-\frac{\partial u}{\partial\lambda}\times\frac{d\lambda}{d\theta}}{\frac{\partial u}{\partial T}+\frac{\partial u}{\partial P}\times\frac{A_2}{A_3}} \quad (4)$$

$$\frac{dP}{d\theta}=\frac{\frac{\frac{dW}{d\theta}}{W}+\frac{\frac{\partial r}{\partial\lambda}}{r}\times\frac{d\lambda}{d\theta}-\frac{\frac{dV}{d\theta}}{V}+A_3\times\frac{dT}{d\theta}}{A_2} \quad (5)$$

Where *A1*, *A2* and *A3* are calculated with equations 6, 7 and 8 (Heywood 1988)..

$$A_1=\frac{\left(\begin{array}{l}-P\times\frac{dV}{d\theta}+\frac{dQ}{d\theta}+h_{intake}\times\frac{dW_{intake}}{d\theta}+\cdots\\ \qquad\cdots h_{ech}\times\frac{dW_{exhaust}}{d\theta}+h_{inj}\times\frac{dW_{inj}}{d\theta}-u\times\frac{dW}{d\theta}\end{array}\right)}{W} \quad (6)$$

$$A_2=\frac{1}{P}-\frac{\frac{\partial r}{\partial P}}{r} \quad (7)$$

$$A_3=\frac{1}{T}+\frac{\frac{\partial r}{\partial T}}{r} \quad (8)$$

4.2. Kinematic model

The volume delimited by the piston, the cylinder wall and the cylinder head can be calculated function of the angular position of the crankshaft θ as shown in equations 9 and 10.

$$V(\theta)=\frac{\pi D^2}{8}\times L\left(1+\frac{R}{L}-\cos\theta-\sqrt{\left(\frac{R}{L}\right)^2-\sin^2\theta}+\frac{2}{\varepsilon-1}\right) \quad (9)$$

$$\frac{dV(\theta)}{d\theta}=\frac{\pi D^2}{8}\times L\times sin\,\theta\times\left(\frac{cos\,\theta}{\sqrt{\left(\frac{R}{L}\right)^2-sin^2\theta}}+1\right) \tag{10}$$

The mean piston velocity is calculated as shown in equation 11.

$$VMP=2\times L\times\frac{N}{60} \tag{11}$$

4.3. Intake and exhaust flow models

The first principle of thermodynamics leads to the Baré Saint-Venant model of flow through the admission intake valve as shown in equation 12.

$$\frac{dW}{dt}=A_{intake}\times P_{intake}\times\ldots$$
$$\ldots\sqrt{\frac{2.\gamma_{intake}}{(\gamma_{intake}-1)\times r_{intake}\times T_{intake}}\times\left[X^{\frac{2}{\gamma_{intake}}}-X^{\frac{\gamma_{intake}+1}{\gamma_{intake}}}\right]} \tag{12}$$

Where X is the maximum found between two values as shows equation 13.

$$X=max\left(\frac{P}{P_{intake}};\left(\frac{2}{\gamma_{intake}+1}\right)^{\frac{\gamma_{intake}}{\gamma_{intake}-1}}\right) \tag{13}$$

4.4. Backflow model

Backflow is an undesirable phenomenon that reduces the amount of the amount of fresh air admitted in the engine. It occurs during the intake phase and during the exhaust phase.

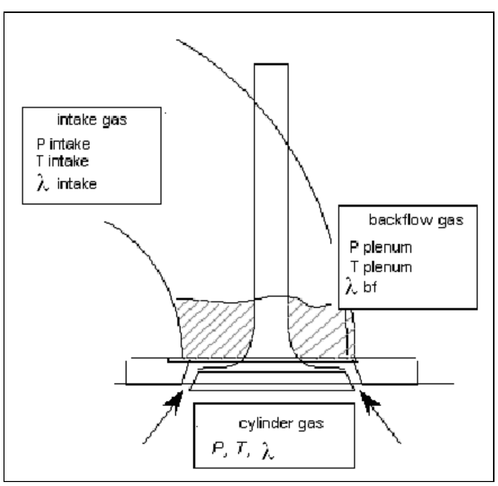

Figure 4: intake backflow phenomena illustration

As shown in figure 4 (case of admission backflow), the exhaust gas that backflow to the admission through intake valve are readmitted in the cylinder in the next cycle. The hypothesis taken for modelling the backflow are the following:

1. the exhaust gas is a homogeneous mixture
2. the temperature of the backflow gas is constant and is equal to the exhaust gas temperature
3. the pressure of the exhaust gas is equal to the intake pressure

The mass conservation laws applied to the admission can be written as shown in equation 14:

$$dW_{bfl}=-dW(\theta) \tag{14}$$

If *P>Pintake*, gas is backflow and *dW(θ)* is calculated with equation 12 after inverting in and out.

If *P<Pintake,* backflow gas are readmitted and *dW(θ)* is calculated with equation 12 in its standard form, until W_{bfl} equals to zero.

4.5. Combustion model

The accurate description of the combustion process in the Diesel engines requires the knowledge of different phenomena that occur starting with the fuel injection and ending with its transformation into combustion products. This knowledge needs the study of the aerodynamic motion in the combustion chamber, the penetration and the vaporization of the fuel as well as the self ignition delay and the combustion itself. Those phenomena cannot be modelled without a three dimensional study.

In our study, we have decided to simplify the modelling work in order to demonstrate a first order potential of our technical solution. A global approach of the combustion process has been modelled considering four phases:

1. The phase of the delay of self ignition: the injected fuel does not start burning before this delay ends
2. The phase of the pre-mixed combustion, where the fuel injected during the delay of self ignition is burned.
3. The phase of the diffusion combustion: while the injection is still ongoing and the combustion chamber is already hot, the combustion happens almost instantly and its speed is limited by the injection flow rate.
4. The final combustion phase: After the fuel injection ends, the remaining unburned fuel (relatively small quantity) burns slowly depending on the diffusion phenomena.

The global combustion model can be simplified by the combination of two Wiebe laws (Wiebe 1970). The first one describes the *pre-mixed combustion* and the second one describes the *diffusion combustion.* Each Wiebe law allowing the calculation of the burned fuel ratio is written as shown in equation 15.

$$xb = \frac{1-exp\left(-w\times\left(\frac{\theta-\theta_{phase}}{\Delta\theta_{phase}}\right)^{m+1}\right)}{1-exp(-w)} \tag{15}$$

Where m and w are constants verifying the equations 16 and 17.

$$w = \frac{m}{m+1}\times\frac{1}{\left(\frac{\theta_{HLC}-\theta_{phase}}{\Delta\theta_{HLC}}\right)^{m+1}} \tag{16}$$

$$HLC = \frac{m}{\left(\frac{\theta_{HLC}-\theta_{phase}}{\Delta\theta_{HLC}}\right)}\times\frac{\exp\left(-\frac{m}{m+1}\right)}{1-\exp(-w)}\times\frac{1}{\Delta\theta_{phase}} \tag{17}$$

The contants above should be calibrated in order to fit with the studied engine performance. As mentioned previously, no test was conducted to fit those constants with a specific engine. All the parameters were taken from literature review (Guibert 2002).

4.6. Heat transfers model

Heat transfers through boundary limits are described with equation 18.

$$\frac{dQ}{d\theta} = \sum_{j=boundary}\varphi.A_j.(CT1\cdot(Tj-Tsc)+CT2\cdot(Tsc-T)) \tag{18}$$

Where the thermal exchange coefficient φ is calculated with Woschni (Woschni 1967) model illustrated in equation 19.

$$\varphi = 12.986\times10^{-3}\times D^{-0,2}\times P^{0.8}\times T^{-0,53}\times ...$$
$$...\left(2.28\times VMP+0.00324\times\frac{Vs\times T0}{V0\times P0}\times(P-Psc)\right)^{0.8} \tag{19}$$

5. RESULTS AND DISCUSSIONS

5.1. Parametric study conducted

In order to demonstrate the potential of the pneumatic hybridization of the Diesel engine, we have conducted a parametric study varying simultaneously the intake and exhaust pressure. The intake temperature is considered constant and equals to 293 K.

Unfortunately, increasing intake pressure is constrained by stress levels in critical mechanical components. These maximum stress levels limit the maximum cylinder pressure which can be tolerated under continuous operation, though the thermal loading of critical components can become limiting too. As boost pressure is raised, unless engine design and operating conditions are changed, maximum pressures and thermal loadings will increase almost in proportion (Heywood 1988).

Reliability criteria of most common Diesel engines specify 200 bars and 1100 K as limits not to be exceeded. Those limits were taken in consideration in our parametric study.

5.2. Operating at low loads (BMEP = 5 bars)

Figure 5 shows the variation of the specific fuel consumption of the Diesel engine function of the intake and exhaust pressure for a fixed mechanical output equivalent to a BMEP of 5 bars. Natural aspirated engines have an inlet and exhaust pressure close to 1 bar, thus the specific consumption is about 250 g/kWh. Turbocharged Diesel engines has higher intake pressure but also higher exhaust pressure as well. At low loads, exhaust pressure are slightly higher than intake pressure because the exhaust gas enthalpy is relatively low. Previous work (Ibrahim 2007) has showed that intake pressure and exhaust pressure at BMEP of 5 bars for the same engine, are respectively 1.7 bar and 1.9 bar.

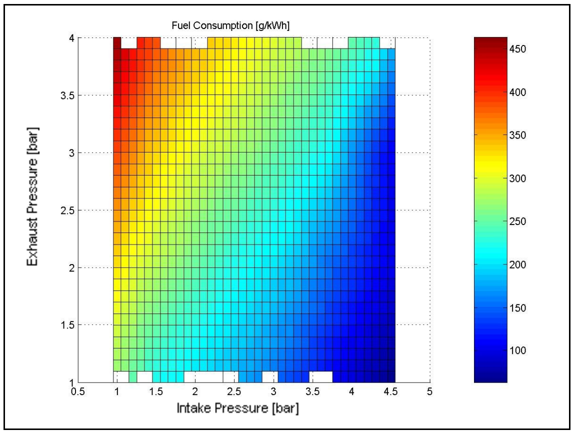

Figure 5: Fuel consumption at BMEP of 5 bars, function of intake and exhaust pressures

As we can see in figure 12, the maximum cylinder pressure increases directly and significantly with intake pressure and it is relatively independent from exhaust pressure. The maximum allowed pressure is reached at an intake pressure of 4.5 bars.

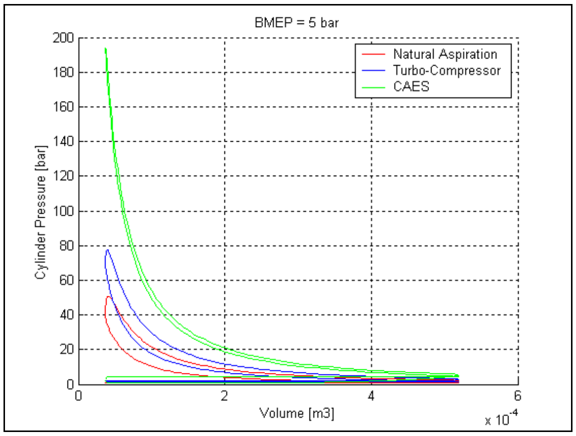

Figure 6: P-V Diagram of Thermodynamic Diesel cycle for Natural Aspirated engine (red), Turbocharged

engine (blue) and pneumatic hybrid engine (green), at BMEP of 5 bars

According to figure 5, the specific consumption for the turbocharged engine working will be close to 275 g/kWh. Now if the CAES is used to increase the inlet pressure until the peak cylinder pressure reaches its maximum allowed and the turbine is removed, the optimal operating point will be for an inlet pressure of 4.5 bars and an exhaust pressure of 1 bar. The specific consumption will be 75 g/kWh; witch is an improvement of about 70% comparing to natural aspirated engines and turbocharged engines. The reasons of this apparently huge improvement are explained as follows.

Figure 6 shows the thermodynamic Diesel cycle for the three operating types above. The three cycles develop the same area. We can observe that the more the intake pressure is, the higher is the peak pressure and the thinner is the shape high pressure cycle.

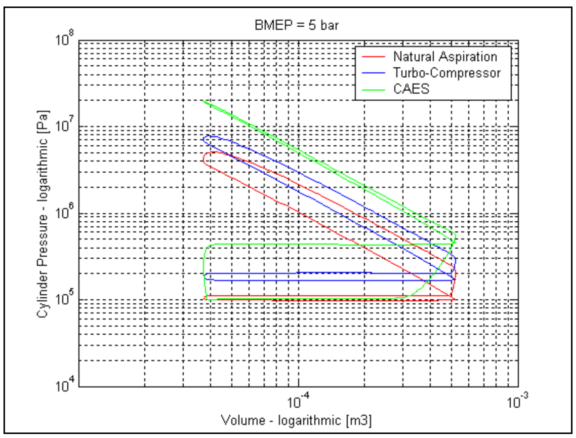

Figure 7: Log (P)-Log (V) Diagram of Thermodynamic Diesel cycle for Natural Aspirated engine (red), Turbocharged engine (blue) and pneumatic hybrid engine (green), at BMEP of 5 bars

Figure 7 is the same as figure 6 but drawn with a logarithmic scale. We can notice the difference in the low pressure cycle. For both Natural aspirated and Turbocharged operation, the low pressure cycle does not contribute significantly in the BMEP; at the opposite, for operation with CAES, the positive gap between inlet pressure and exhaust pressure generates work that contributes significantly in BMEP, as shown also in figure 9. This work is purely pneumatic and does not result from any combustion.

Figure 9 shows the contribution of the pneumatic *green* power in the total power generated by the engine. We can observe that for the highest allowed intake pressure and lowest exhaust pressure, the contribution is positive and can reach 40% witch means 40% of direct fuel saving. On the other hand, increasing intake pressure and reducing exhaust pressure make this contribution negative and thus increases the fuel consumption.

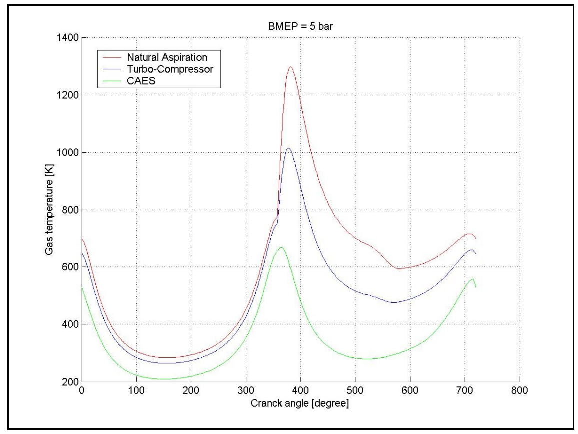

Figure 8: Instant cylinder gas temperature for Natural Aspirated engine (red), Turbocharged engine (blue) and pneumatic hybrid engine (green), at BMEP of 5 bars

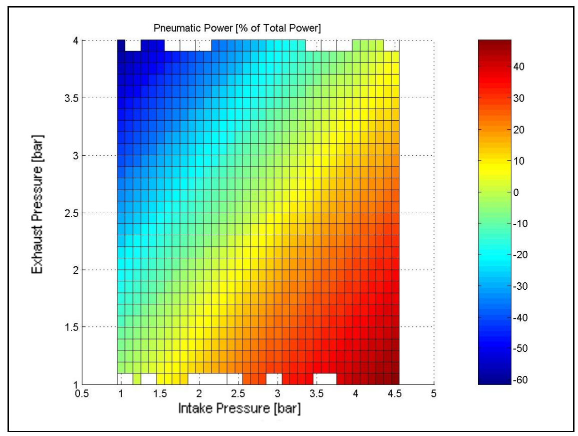

Figure 9: Contribution of the pneumatic power in the total power at BMEP of 5 bars, function of intake and exhaust pressure

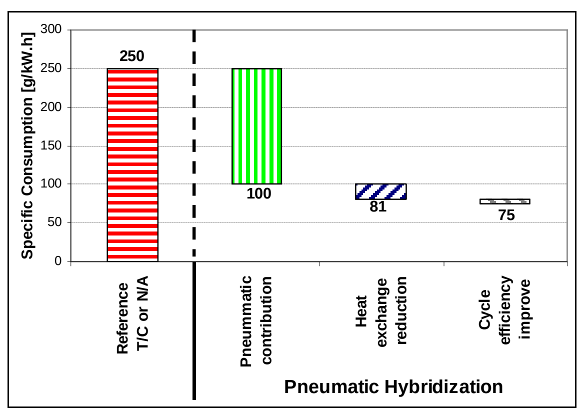

Figure 10: Fuel reduction contribution at BMEP of 5 bars, for Inlet pressure and exhaust pressure of 5 bars and 1 bar respectively

As shown in figure 10, the improvement in fuel consumption is not only the due to pneumatic power contribution, but also to two other phenomena:

1. The first phenomenon is the significant reduction of the heat exchange thanks to two reasons:
   (a) less fuel burned because less high pressure BMEP is needed to provide the target BMEP
   (b) higher air density and therefore high calorific capacity

   The heat exchange is responsible of about 25% according to figure 11 (Heywood 1988; Wanhua 2009) of the total loss in the engine. The gas temperature decrease from a Turbocharged engine to CAES assisted engine, reaches up to 300 K as shows figure 8. The thermal loss reduction is therefore around 30%.
2. The second phenomenon responsible of the remaining improvement is the increase of the combustion velocity thanks to the increase of air density and therefore the *cycle efficiency*.

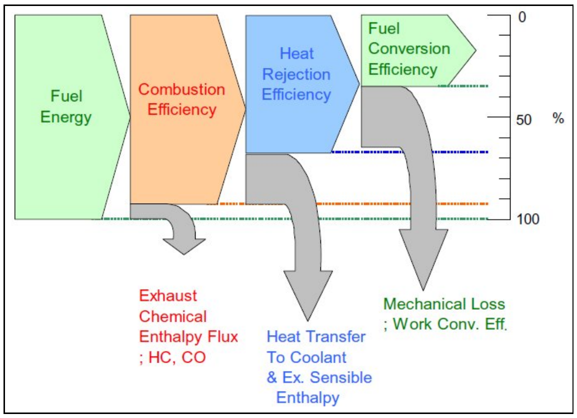

Figure 11: Typical fuel energy flow and efficiencies and losses in internal combustion engines (Heywood 1988; Wanhua 2009)

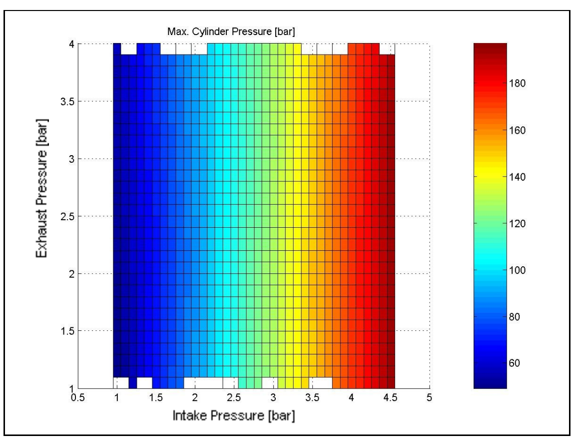

Figure 12: Peak cylinder pressure at BMEP of 5 bars, function of intake and exhaust pressures

The main limitation for fuel economy is the maximum allowed pressure in the combustion chamber. Increasing the admission pressure causes a much higher increase of the maximal pressure in the cylinder. For the Diesel engine specified above, 200 bars in the combustion chamber is reached when the admission pressure is about 5 bars at low engine load and 3.5 bars at high load.

5.3. Operating at medium loads (BMEP = 10 bars)

Operating at a BMEP of 10 bars is considered a medium load for turbocharged engine but is a full load for Natural Aspirated engine.

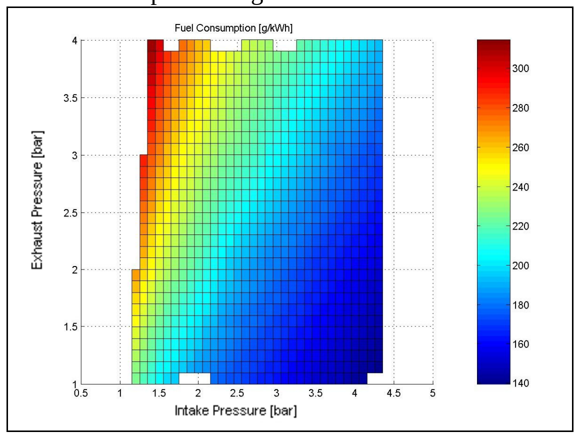

Figure 13: Fuel consumption at BMEP of 10 bars, function of intake and exhaust pressures

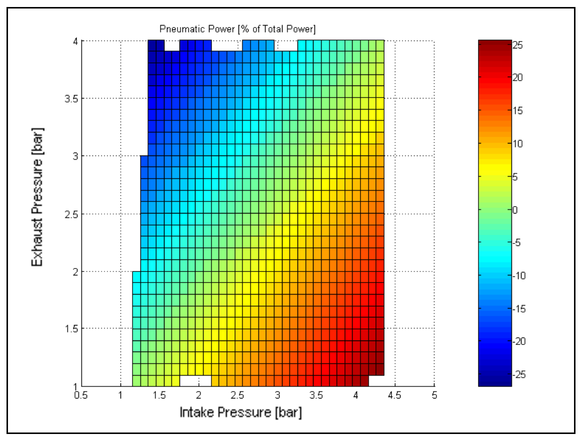

Figure 14: Contribution of the pneumatic power in the total power at BMEP of 10 bars, function of intake and exhaust pressure

For both turbocharged engine where the intake pressure and the exhaust pressure are both around 2.5 bars (Ibrahim 2007) and Natural aspirated engine where the intake pressure and the exhaust pressure are both around 1 bar, the fuel consumption is around 210 g/kWh, as shows figure 13.

Working under CAES assistance, at the limit of stress acceptance, intake pressure is 4.4 bars and exhaust pressure is 1 bar, therefore the fuel consumption is 140 g/kWh as shows figure 13, witch corresponds to an improvement of 33%. The pneumatic power contribution in the total power reaches 20% as shown in figure 14.

As shown in figure 15, heat loss reduction improve contribute significantly in the global fuel economy, thanks to a great temperature drop of the gas in the cylinder chamber, that reaches up to 200 K.

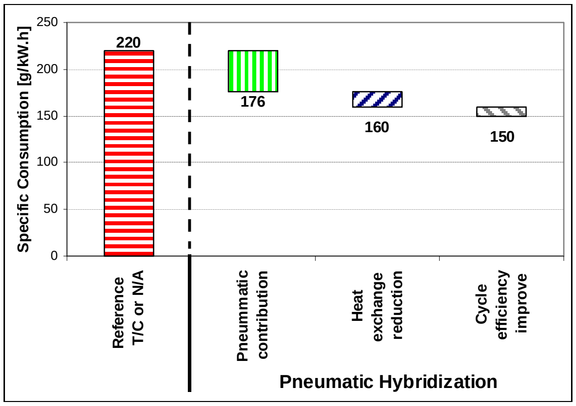

Figure 15: Fuel reduction contribution at BMEP of 10 bars, for Inlet pressure and exhaust pressure of 4 bars and 1 bar respectively

5.4. Operating at high loads (BMEP = 15 bars)

Natural Aspirated Engines cannot provide a torque equivalent to BMEP of 15 bars, while Turbocharged engines can do that. For turbocharged engine where the intake pressure and the exhaust pressure are both around 3 bars (Ibrahim 2007), the fuel consumption is around 210 g/kWh, as shows figure 16.

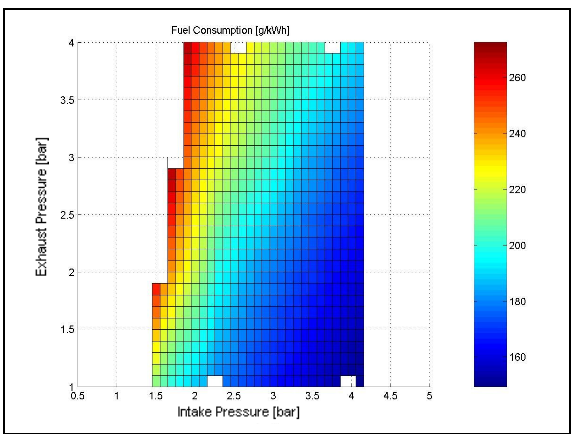

Figure 16: Fuel consumption at BMEP of 15 bars, function of intake and exhaust pressures

Working under CAES assistance, at the limit of stress acceptance, intake pressure is 4 bars and exhaust pressure is 1 bar, therefore the fuel consumption is 150 g/kWh as shows figure 15, witch corresponds to an improvement of 28%. The pneumatic power contribution in the total power reaches 15% as shown in figure 17.

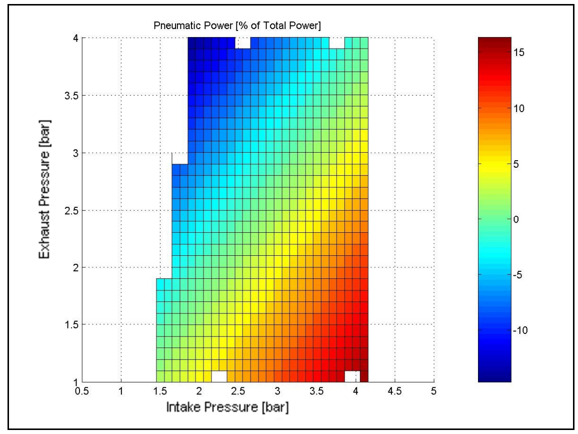

Figure 17: Contribution of the pneumatic power in the total power at BMEP of 15 bars, function of intake and exhaust pressure

Figure 18 shows the contribution of different phenomenon in fuel economy under CAES assistance at the maximum intake pressure allowed and the minimum exhaust pressure.

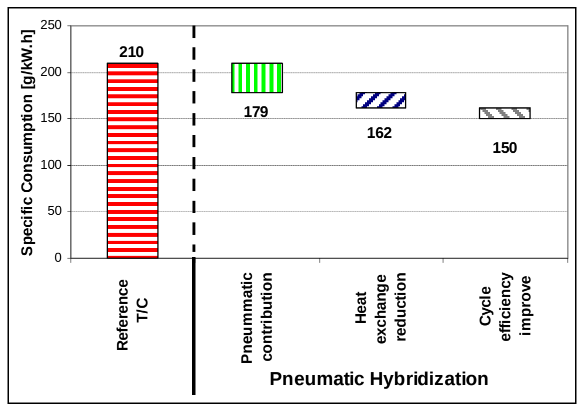

Figure 18: Fuel reduction contribution at BMEP of 15 bars, for Inlet pressure and exhaust pressure of 4 bars and 1 bar respectively

6. IMPACT OF PNEUMATIC HIBRIDIZATION ON POLLUTANT EMISSIONS

As mentioned in the previous paragraph, working under CAES assistance reduces significantly the temperature of gas in the combustion chamber. Previous work indicates that the lowered combustion temperature in diesel engines is capable of reducing soot and nitrogen oxides (NOx) simultaneously (Zheng 2006). CAES has therefore the same effect as Exhaust gas recirculation used in most Diesel applications in order to reduce NOx emissions. Figure 19 shows the impact of local temperature and equivalence ratio in the combustion chamber on the soot and NOx formation.

Operating with a high admission pressure and low exhaust pressure moves the local conditions in the combustion chamber to the right bottom side of Pischinger Diagram and therefore minimizes the chances of NOx and soot formation and maximizes the chances of soot oxidation. Therefore the pneumatic hybridization of Diesel engines is expected to have a positive effect on pollutant emissions as well as greenhouse gas emissions.

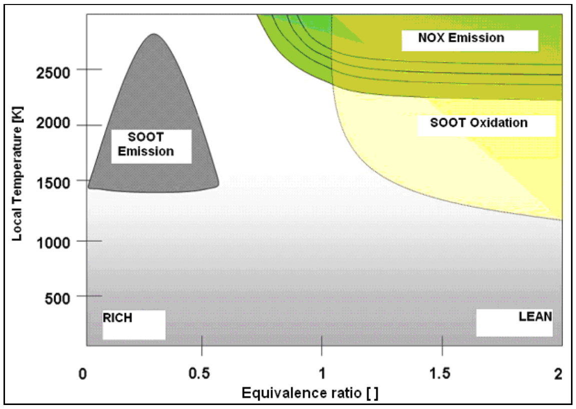

Figure 19: Pischinger Diagram (Pischinger 1988)

7. CONCLUSION

Increasing up to 4.5 bars the pressure at intake and decreasing down to 1 bar the pressure at exhaust of the Diesel engine results in increasing the power produced by the engine for the same fuel quantity and therefore improving the Diesel efficiency considering that the Compressed Air is free of charge. This improvement can reach up to 70% for the low loads (BMEP = 5 bar) and up to 40% for the high loads (BMEP = 15 bar), for a Diesel engine whose compression ratio is 14 and maximum allowed cylinder pressure is 200 bars.

This improvement is a combination of three phenomena:

1. The Low Pressure BMEP usually slightly negative for Turbo-Compressed engines and Natural Aspirated Engines becomes positive and therefore generates pure pneumatic power. The ratio of this pneumatic power to the total power can reach up to 40% for low loads and 10% for high loads.
2. The increase of intake air density reduces significantly the temperature in the cylinder and therefore reduces the thermal loss with the cylinder borders. This thermal loss is usually responsible of about 25% of the total loss in the engine. The gas temperature reduction due to increasing of air pressure can reach up to 200 K for low loads, and 350 K for high loads; the thermal loss reduction is therefore around 30% for low loads and 60% for high loads.
3. The increase of the combustion velocity thanks to the increase of air density and therefore the *cycle efficiency* is responsible of the remaining improvements.

In addition to its potential in fuel economy and greenhouse gas emission reduction, Diesel pneumatic hybridization is expected to have positive effect on pollutants emissions such as nitrogen oxides and soot.

NOMENCLATURE AND ABREVIATIONS

A	Effective area [m^2]
Aj	Boundary areas involved in heat transfer [m^2]
Cp	Specific heat capacity at constant pressure [J/kg/K]
CT1	Multiplicative coefficient
CT2	Multiplicative coefficient
Cv	Specific heat capacity at constant volume [J/kg/K]
D	Bore [m]
h	Enthalpy [J/kg]
HLC	Maximum heat release rate [1/deg]
L	Stroke [m]
N	Engine angular speed [RPM]
P	Pressure [Pa]
Psc	Gas pressure in the chamber if there was no combustion [Pa]
P0	Pressure of combustion chamber at the intake valve close [Pa]
Q	Heat transfer [j]
R	Manivelle radius [m]
T	Absolute temperature [K]
Tj	Mean temperatures of the boundary areas involved in heat transfer [K]
Tsc	Gas temperature in the chamber if there was no combustion [K]
T0	Temperature of combustion chamber at the intake valve close [K]
u	Internal specific energy [J/kg]
V	Volume [m^3]
VMP	Mean piston velocity [m/s]
Vs	Combustion chamber volume at Bottom Dead Centre [m3]
V0	Volume of combustion chamber at the intake valve close [Pa]
W	mass of gas [kg]
WPPR	Wind power penetration rate [%]
xb	Proportion of burned fuel [ ]
$_{air}$	Refers to fresh air conditions
$_{bfl}$	Refers to Backflow conditions
$_{exhaust}$	Refers to exhaust conditions
$_{inj}$	Refers to injected fuel conditions
$_{intake}$	Refers to intake conditions
$_{phase}$	Refers to the combustion phase start
$_{stoch}$	Refers to stoichiometry conditions
ε	Compression ratio [ ]
Θ	Crankshaft angle [deg], where 360° refers to combustion phase Top Dead Center
Φ	Thermal exchange coefficient [J/m^2K]
λ	Equivalence ratio
r	Perfect gas constant [J/kg/K]
γ	Specific or molar volume [ ]

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