THEORICAL STUDY OF A VOLUMETRIC HOT AIR JOULE CYCLE ENGINE

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ABSTRACT

Currently, micro combined heat and power (CHP) technologies are internal combustion engine (ICE), Stirling engine, organic Rankine cycle (ORC) and fuel cell. Ericsson engines show the same flexibility in term of fuel as ORC and Stirling engines with a potentially better electrical efficiency. The Ericsson engine is a volumetric hot air Joule cycle engine. It consists of a reciprocating compressor and a reciprocating expander and a heat exchanger to provide external heat. In order to study the Ericsson engine, a model of a piston machine is built. The model associates a geometric model of the machine and a thermodynamic model of processes that the fluid undergoes. This model can be used to simulate both the compressor and the expander of the Ericson engine. By coupling these two components, the whole system can be simulated. This detailed piston engine model computes instantaneous thermodynamic properties of the working fluid inside the cylinder. It is first used to simulate the compressor and the expander separately and to show the influence of the pressure drop and heat transfer. Then the whole model is used to optimize the design of the engine, to evaluate the performance and to highlight the sources of losses. Finally, the proposed Ericsson engine based micro-CHP system shows good performance; this shows that the Ericsson engine deserves the same interest as the Stirling engine and ORC.

Keywords: micro-cogeneration, CHP, Ericsson engine, Joule engine, reciprocating expander

INTRODUCTION

In the current energy and environmental context (depletion of fossil fuels and climate change), the interest in CHP is growing. Special attention is currently paid to micro-CHP (<50kWe) because it is not widely implemented and interest for domestic applications is growing. Moreover, CHP is within one of the key issues in the energy sector of the European Union [1].

A quick literature review [1..6] shows that the main technologies available for micro CHP at the present time are the internal combustion engine (ICE), the Stirling engine and fuel cells. The micro-gas turbine and the ORC (Organic Rankine Cycle) are still in development. Among these presently-available technologies, only the Stirling engine and ORC are external combustion process, allowing the use of renewable energy such as biomass or solar energy. Both technologies show electrical efficiency between 10-20%.

In this context, the Ericsson engine regains interest. It allows for the conversion of sustainable energy sources such as solar energy or biomass into electricity with relatively high efficiency. Moreover, it shows several advantages compared to the Stirling engine, such as the fact that heat exchangers are not clearance volumes like in Stirling engines [7,8].

The goal of this paper is to present a detailed simulation model of reciprocating expander and compressor used to simulate an Ericsson engine. In the literature, reciprocating expanders are modeled in different ways. A. Touré and M. Creyx [8,9] propose both a model without losses and using a perfect gas. Bonnet et al. [7] uses isentropic process and adds pressure drops using ideal gases. Y. Glavatskaya [10] proposes a semi empirical model with pressure drops, heat transfer and mechanical losses based on overall parameters. The model proposed in this paper is a detailed model that computes crank angle evolution of the fluid inside the cylinder considering pressure drops and heat transfers. This model could be used for optimizing the geometry of both the compressor and the expander used in the Ericsson engine.

THE ERICSSON ENGINE

The Ericsson engine is an external combustion piston engine. The working fluid is generally air so it is classified as a hot air engine. A schematic representation of an Ericsson engine is shown in Figure 11. It shows that the fluid enters the compression cylinder (noted C) where it is compressed. Then, the fluid exits the compressor to enter the heater (noted H). There, it receives an amount of thermal energy and its temperature rises. Finally, the fluid expands into the expander (E) to produce mechanical energy.

This is the simplest configuration of Ericsson engine. Indeed, improvement of the engine can be made by adding a regenerator that recovers the heat available at the exhaust of the expander to preheat the fluid before it enters the heater (see Figure 11). It is also possible to introduce a heat exchanger to cool the fluid that leaves the expander before driving it to the compressor, so that the engine works in a closed loop.

Contrary to what its name suggests, the best theoretical cycle to describe an Ericsson engine is the Brayton cycle and not the Ericsson cycle. Indeed, the Ericsson cycle is made up of two isobaric and two isothermal processes. However, in an Ericsson engine, the heat transfers between the heat source/sink and the working fluid take place in dedicated heat exchangers and not in the cylinders. Then, the Brayton cycle, consisting of two isobaric and two isentropic processes, used to describe gas turbines, is more suitable. In this sense, the Ericsson engine is a gas turbine where turbomachines are replaced by volumetric engine [7].

For this type of engine, the pressure ratio is defined as the ratio between the pressure in the heater and the supply/exhaust pressure. And the temperature ratio is defined as the ratio between the temperature at the exhaust of the heater and the temperature at the supply of the engine.

MODEL

The proposed model of Ericsson engine is based on a piston machine model. This piston machine model is used to simulate both the compressor and the expander. The goal of the model is to compute instantaneous states of the fluid in the cylinder. To do this, it is assumed that:

- The state of the fluid is homogeneous inside the cylinder
- The temperature of the cylinder walls is uniform

GEOMETRIC MODEL

The goal of the geometric model is to express the cylinder volume in terms of the crank angle θ (Figure 1). Considering Figure 1, the volume can be expressed as:

$$V = V_D + \pi.\frac{D^2}{4}.x$$
$$x = B + R - \sqrt{B^2 - R^2.\sin^2(\theta)} - R.\cos(\theta) \quad (1)$$

Where V_D is the clearance volume, D is the bore diameter, B is the length of the connecting rod and R the length of the crank.

The ratio between the dead and the total volume is called the clearance factor ($CF = V_D/(V_S + V_D)$). And the link between the geometry of the compressor and the expander is characterized by the volume ratio. This is the ratio between the swept volume of the expander and that of the compressor $\phi = V_{S,E}/V_{S,C}$.

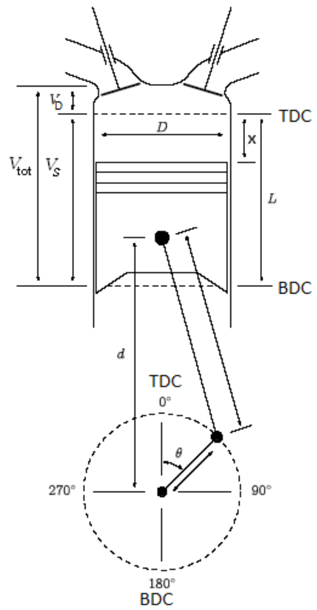

Figure 1: Sketch of the set cylinder, piston, connecting rod and crankshaft

CONSERVATION EQUATIONS

The outline of the volume defined above forms an open thermodynamic system. By applying the first law, it can be demonstrated [11] that the rate of change of the fluid temperature inside the cylinder can be expressed as in Eq (2). In this equation, $\omega = d\theta/dt$.

$$\frac{dT}{d\theta} = \frac{1}{m.c_v}.\left[-T.\left(\frac{\partial P}{\partial T}\right)_v.\left(\frac{dV}{d\theta} - v.\frac{dm}{d\theta}\right) - h.\frac{dm}{d\theta} + \frac{\dot{Q}}{\omega} + \sum\frac{\dot{m}_{in}}{\omega}.h_{in} - \sum\frac{\dot{m}_{out}}{\omega}.h_{out}\right] \quad (2)$$

The principle of mass conservation applied to the system give the rate of change of the mass in the cylinder:

$$\frac{dm}{d\theta} = (\dot{m}_{in} - \dot{m}_{out}).\frac{1}{\omega} \quad (3)$$

GAS STATE EQUATION

By solving the differential equations (2) and (3) and combining with an equation of state of the working fluid (4), the specific volume and the temperature of the fluid in the cylinder can be computed as a function of the crank angle.

$$P = P(T, v) \tag{4}$$

The working fluid is air and will be considered as an ideal gas, thus its equation of state is given by $P = r.\frac{T}{v}$.

HEAT TRANSFER

Differential equation (2) involves heat flux $\dot{Q}$. This heat flux results from convection at the walls cylinder and can be expressed as:

$$\dot{Q}_w = h_c.S.(T - T_w) \tag{5}$$

Where h_c is convection heat transfer coefficient, S the surface of the walls cylinder and T_w the temperature of these walls.

The convective coefficient h_c can be estimated through several correlations that give the Nusselt number. Here, the Woschni correlation is taken [12]:

$$Nu = 0.035.Re^{0.8} \tag{6}$$

To compute the Reynolds number, Woschni assumes that the characteristic length is the diameter of the cylinder bore and the speed of the gas is given by:

- During gas exchange process:

$$C = 6.18(4.R.\frac{\omega}{2\pi}) \tag{7}$$

- During compression and expansion:

$$C = 2.28(4.R.\frac{\omega}{2\pi}) \tag{8}$$

MASS FLOW RATES

Finally, to solve the problem, the mass flow rates entering and exiting the cylinders must be computed. Assuming that the flows through the valves are isentropic, the mass flow rate is expressed by:

$$\dot{m} = S_v.\frac{P_{up,v}}{\sqrt[2]{r.T_{up,v}}}.R_{p,v}^{\frac{1}{\gamma}}.\sqrt[2]{\frac{2\gamma}{\gamma-1}.(1 - R_{p,v}^{\frac{\gamma-1}{\gamma}})} \tag{9}$$

Where S_v is the flow area, $P_{up,v}$ and $T_{up,v}$ are respectively the pressure and the temperature upstream of the valve, $R_{p,v}$ is the pressure ratio between the upstream and the downstream of the valve and γ is the isentropic exponent.

The flow area depends on the lift of the valve and is computed as in [12]. The valve lift is calculated according to G. Blair [13]. This lift depends of the crank angle and of its maximal amplitude $L_{v,\max}$. For kinematics (cam profile) and dynamics (acceleration of the valve and thus strength) considerations, the maximal amplitude will be assumed proportional to the crank angle during which the valve is open θ_v(see Figure 2) :

$$L_{v,max} = \frac{L_{v,max,ref}}{\theta_{v,ref}}.\theta_v \tag{10}$$

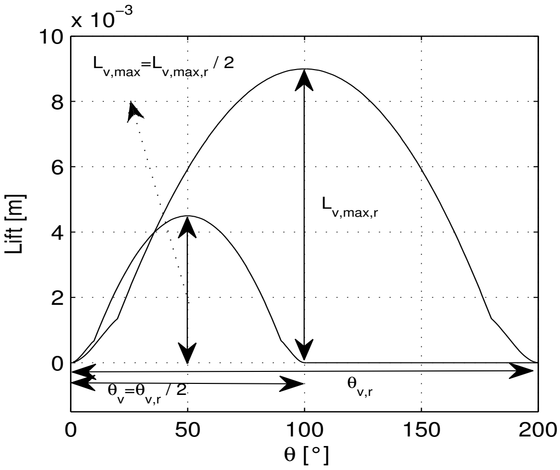

Figure 2: Valve lift

TIMING OF THE VALVE

For efficient operation, the valves must open and close at the right time. For the compressor, the admission take place during the downstroke but must start only when the pressure in the cylinder has reached the inlet pressure and stop at the bottom dead center when the volume is maximal. The exhaust takes place during the upstroke, start when the pressure in the cylinder has reached the outlet pressure end stop at the top dead center. For the expander, the admission starts at the top dead center and must stop so that the pressure at the bottom dead center in the cylinder has reached the outlet pressure in order to avoid under expansion losses. The exhaust start at the bottom dead center and the end can stop at the top dead center or shortly before so that the pressure rises in the cylinder before the admission. It can be shown [8] that, in an Ericsson engine, the performance are better when the exhaust valve closes so that the pressure in the cylinder reaches the inlet pressure.

It must be noted that, for the compressor, the activation of the valves can be done automatically with reed valves, as is commonly done in reciprocating compressors. But for the expander, the valves must be controlled externally, for example by a camshaft.

Figure 5 shows crank angle evolution of pressure in the cylinder for the compressor and expander. On this plot, the angle when inlet and exhaust valve open and close (IVO, IVC, EVO and EVC) are shown. Among these angles, the θ_{IVO} and the θ_{EVO} of the compressor and the θ_{IVC} and the θ_{EVC} of the expander depends on the pressure ratio. The angle during which a valve is open (θ_v) is the difference between its angle of closing and opening. The dependence between the θ_v and the pressure ratio for the ideal case (without losses, see below) is shown on Figure 3. These dependencies will influence the pressure drop since the maximal lift is proportional to θ_v. More θ_v is low, more the pressure drop is high.

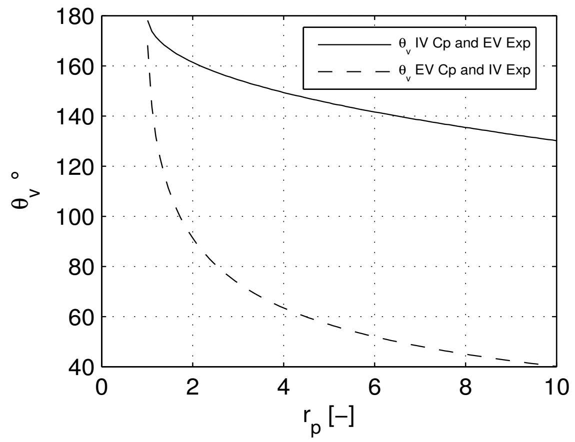

Figure 3: Opening angle in term of pressure ratio

COUPLING

In an Ericsson engine, the compressor and the expander are coupled both mechanically (shafts are connected) and through the working fluid. This implies that the rotational speed and the mass flows rate are the same for the compressor and the expander: $\omega_C = \omega_E$ and $\dot{m} = \dot{m}_C = \dot{m}_E$.

As A. Touré shows [8], the consequence is that for a given geometry (ϕ and CF), the pressure ratio depends only on the temperature ratio and conversely.

POWER AND EFFICIENCY

As it is assumed that the pressure is homogeneous in the cylinder, the mechanical work done by the piston in one rotation is:

$$W = -\oint P.dV \quad (11)$$

And the power is:

$$\dot{W} = \frac{\omega}{2\pi}.W \quad (12)$$

In order to take into account the friction losses, a mechanical efficiency is set to compute the net power:

$$\dot{W}_{net} = \left(\dot{W}_E - \dot{W}_C\right).\eta_m \quad (13)$$

And the efficiency of the cycle is this net power divided by the thermal energy provided by the heater:

$$\eta = \frac{\dot{W}_{net}}{\dot{Q}_H} \quad (14)$$

RECOVERED HEAT

In the engine describe above, there are three sources of heat losses. The first one is the heat flux through the wall cylinder. The second one is the mechanical friction that dissipates heat in the engine. And the third one is the rejection of hot gases at the exhaust of the engine.

The two first heat fluxes are evacuated by the cooling loop of the engine and at the ambience. Assuming that the half [10] of this energy is evacuated in the cooling water, the thermal power of the engine cooling loop is:

$$\dot{Q}_{cool} = \dot{Q}_C + \dot{Q}_E + \frac{\dot{W}_{net}}{\eta_m}.(1-\eta_m).0.5 \quad (15)$$

Where $\dot{Q}_C$ and $\dot{Q}_E$ are the compressor and expander fluxes expressed by Eq. (5). These fluxes are computed by setting the wall temperature. This temperature cannot exceed a maximum value (~180 °C) to avoid deterioration of the oil [12].

If the hot gases flux ($\dot{Q}_{hg}$) is recovered by a heat exchanger, it can be computed by setting a heat exchanger's efficiency ϵ_{rec}.

Then, the thermal efficiency of the CHP system is expressed as:

$$\eta_{th} = \frac{\dot{Q}_{cool}+\dot{Q}_{hg}}{\dot{Q}_H} = \frac{\dot{Q}_{th}}{\dot{Q}_H} \quad (16)$$

SIMULATION OF COMPRESSOR AND EXPANDER SEPARATLY

INPUTS

To study the compressor and the expander, some data need to be set:

- Geometrics parameters: The dimensions of the cylinder are set arbitrarily (the order of magnitude corresponds to an automotive engine cylinder) and the values are presented in Table 1. The dimensions of the valves are, according to [12], proportional to the diameters of the bore.
- Supply temperature and pressure: The supply of the compressor corresponds to the ambient thus the supply temperature and pressure are respectively set to 25°C and 1 bar. The supply of the expander corresponds to the exhaust of the heater of the Ericsson engine. The temperature and the pressure at this point are respectively set to 800°C and 6 bar. These values are justified hereafter.

- Exhaust pressure: The exhaust pressure of the compressor corresponds to the inlet pressure of the expander and vice versa.
- Wall temperature: $T_w = 100°C$
- Rotational speed: $\omega = 2\pi.25\ rad/s$

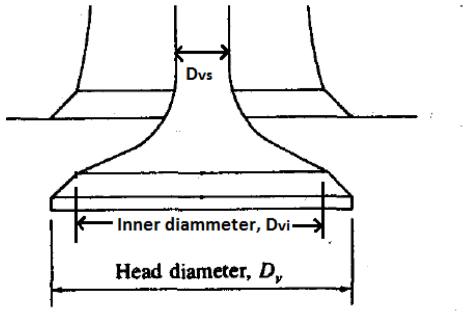

Figure 4: Valves dimensions

Table 1: Geometric parameters

Cylinder		**Valves**	
D	90 mm	D_v	$0.4D$
R	45 mm	D_{vi}	$D_v/1.11$
B	150 mm	D_{vs}	$0.221D_{vi}$
CF	0.05 [-]	$L_{v,max,ref}$	$D_v/4$

RESULTS

Two sources of losses are modeled: the pressure drops and the heat transfer. In order to evaluate the consequences of these sources of losses, three simulations are performed and compared: one without losses (WL) which correspond to an isentropic process, one with the pressure drops (PD) and finally one with the pressure drops and the heat transfer (PD + HT).

Figure 5, Figure 6 and Figure 7 show respectively the crank angle evolution of the cylinder pressure, the crank angle evolution of the cylinder and the PV diagram for both the compressor and the expander. The pressure profile shows that the pressure drops are more important for the compressor. This is explained by the fact that the compressor admits a higher mass flow (more than double, see Table 2) rate through the same flow area. The heat transfer does not induce significant changes on the pressure but well on the temperature, especially for the expander. This induces change in the mass flow rate and exhaust temperature. Indeed, for the expander, the mass flow rate passes from 6.2 g/s to 6.6 g/s and the exhaust temperature from 420 °C to 350 °C when the heat transfers are taken into account.

Finally the isentropic efficiency of the compressor is 0.8 and that of the expander is 0.91. The mass flow rate of the compressor decreases with the losses while that of the expander increases. And the exhaust temperature of the compressor increases while that of the expander decreases.

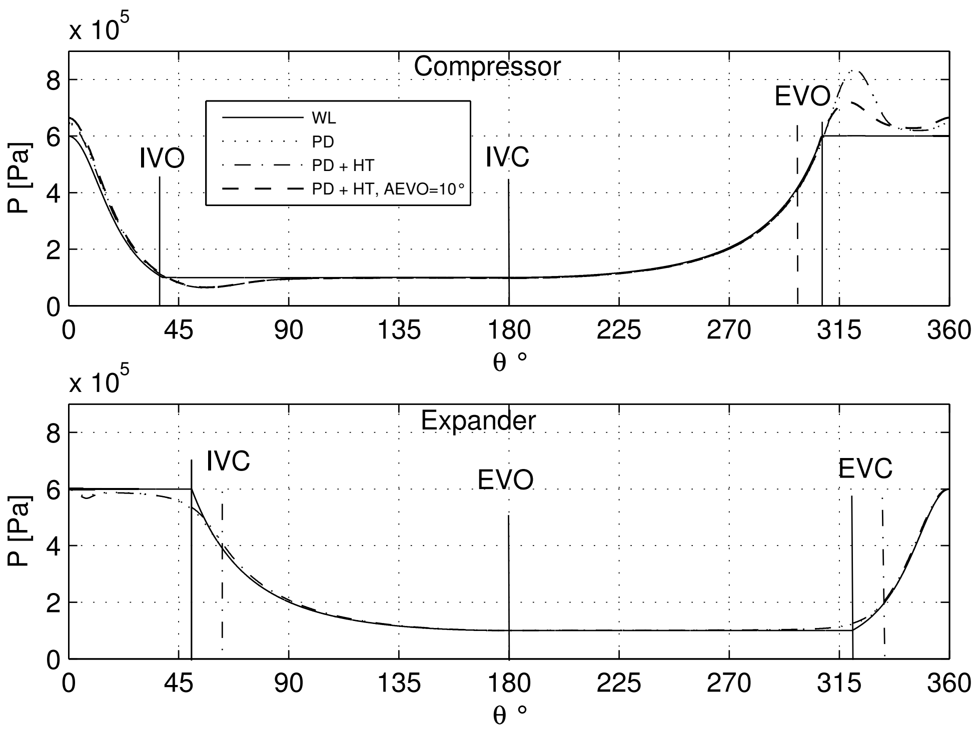

Figure 5: Crank angle evolution of the pressure inside the cylinders

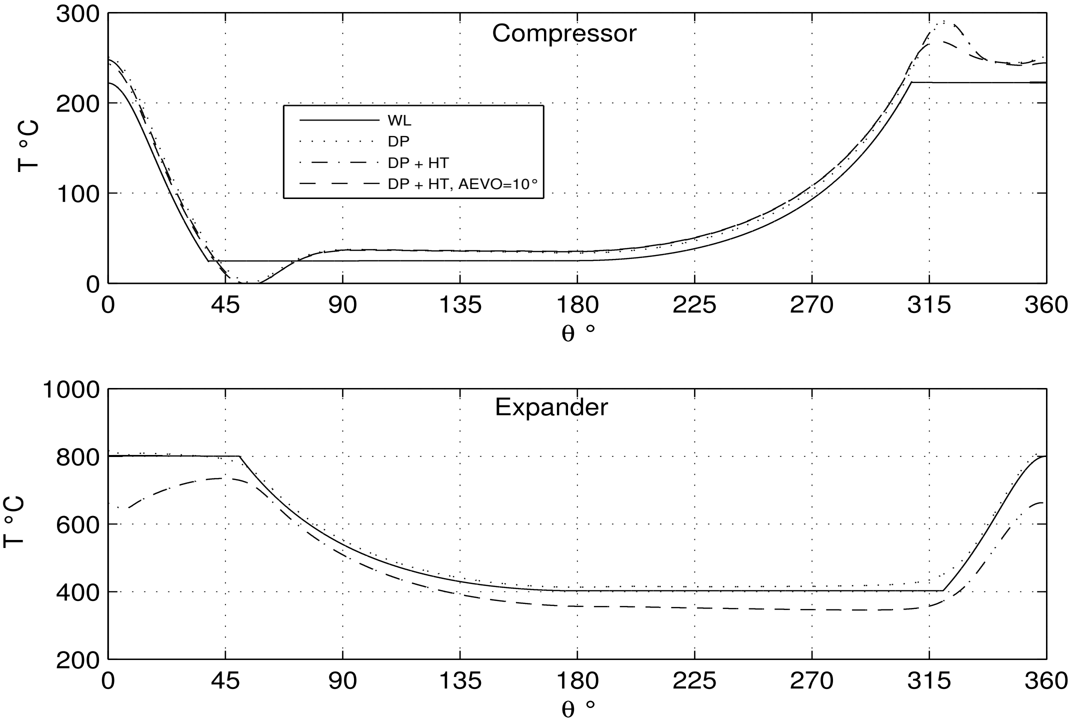

Figure 6: Crank angle evolution of the temperature inside the cylinders

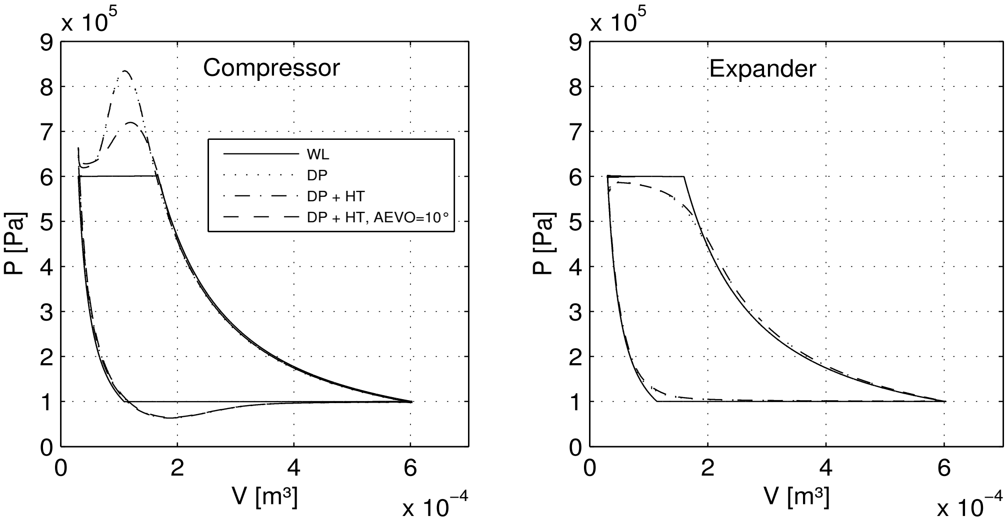

Figure 7: PV diagram

Table 2: Results for separates compressor and expander

	Compressor			**Expander**		
	WL	**PD**	**PD + HT**	**WL**	**PD**	**PD + HT**
$\dot{W}$ [W]	2890	3318	3347	2792	2626	2690
$\dot{M}$ [g/s]	14.43	13.44	13.35	6.3	6.2	6.6
w **[kJ/kg]**	200	246	250	443	423	404
T_{ex} [$°C$]	222	267	268	402	420	350
$\dot{Q}_w$ [W]	0	0	38	0	0	622
ϵ_s [$-$]	1	0.81	0.8	1	0.96	0.91

ADJUSTMENT OF THE TIMING OF THE VALVES

To match the two criteria that condition the timing of the valves of the expander, ie that the pressure in the cylinder has reached the outlet pressure at the end of the downstroke and the inlet pressure at the end of the upstroke, θ_{IVC} and θ_{EVC} had to be adjusted. This adjustment can be seen on Figure 5.

For the compressor, no adjustment had to be made. However, it was found that anticipating slightly the EVO allows for a decrease in the pressure drop and an increase in the isentropic efficiency. The diminution of the pressure drop at the exhaust can be seen in Figure 5 and Figure 7 for an advance of 10°. The isentropic efficiency increases from 0.8 to 0.84 and the mass flow rate decreases from 13.35 g/s to 13.29 g/s.

The same advance can be made on the intake of the expander. But it was found that the increase of the power is offset by the increase of the mass flow rate. This leads to the same specific work and then no increase of the efficiency.

SIMULATIONS

In this section, the model presented above is used to the whole Ericsson engine. However, some parameters can be set by simply considering that, as explained above, the Ericsson engine can be described by Brayton cycle. Indeed, it is possible to demonstrate that [14]:

- For the ideal cycle (isentropic effectiveness of components equal 1), the efficiency of the cycle increase with the pressure ratio and is independent of temperature ratio.
- For the real cycle (isentropic effectiveness of components <1), efficiency (with and without regenerator) and power increase with temperature ratio and there is an optimum pressure ratio that maximize these values.
- The optimum pressure ratio is smaller for cycle with regeneration and the efficiency is higher.

Considering this, the temperature ratio needs to be higher but it is limited by material constraints. It is set to $r_T = 3.6$, which corresponds to an expander supply temperature of $T_{su,E} = 800\ °C$ for a compressor supply temperature of $T_{su,C} = 25\ °C$.

INPUTS

The inputs are the same as the above except:

- The pressure ratio becomes an output that depends of the volume ratio ϕ.
- The volume ratio ϕ is optimized hereafter.
- The heat exchangers efficiencies are set to $\epsilon = 0.8$.

OPTIMISATION OF THE VOLUME RATIO

As explain above, for a given temperature ratio, the pressure ratio depends only on the geometry. Figure 8 shows the evolution of the volume ratio in term of pressure ratio. It shows that the pressure ratio increases as the volume ratio decreases and that the pressure ratio without losses is higher than with losses. This is explained by the fact that, with losses, the mass flow rate of the compressor decreases and the one of the expander increases.

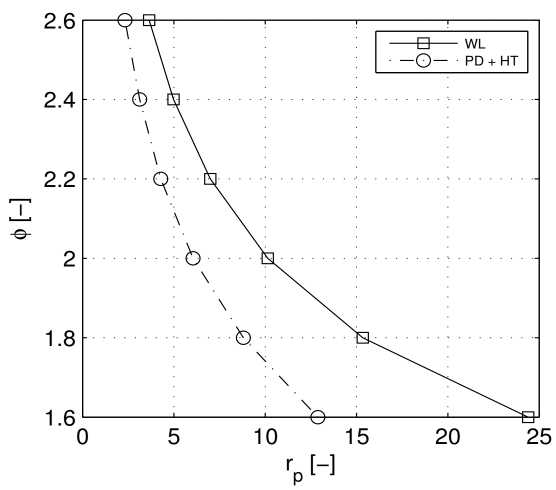

Figure 8: Volume ratio in term of pressure ratio

This variation of the pressure ratio leads to a variation of the efficiency and the power. Figure 9 shows the evolution of the efficiency with and without regenerator for the case without losses, with losses and with an advance of the EVO of the compressor. This graph shows that the losses (pressure drop and heat transfer) affect significantly the efficiency of the Ericson engine, that the advance of the EVO of the compressor induces a slight increase of the efficiency and that there is an optimum around $r_p = 5$ which corresponds to a volume ratio of $\phi = 2.2$.

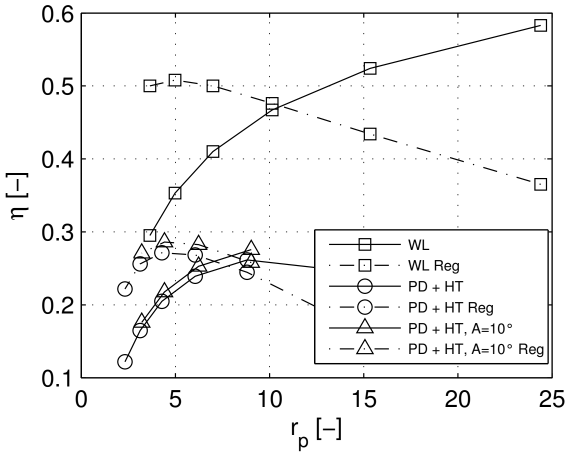

Figure 9: Influence of the volume ratio on the efficiency.

INFLUENCE OF THE HEAT TRANSFER ON THE HEAT REGENERATION

As shown above, the temperature at the exhaust of the expander is lower when there is a heat transfer to the wall. However, this temperature is also the supply temperature of the regenerator. Then, the energy recovered by the regenerator decreases with heat transfer and then the efficiency with regenerator decreases too. This is illustrated in Figure 10 where the ratio between the recovered heat ant the total heat is plotted in terms of pressure ratio. It can be seen that this ratio of heat is lower with heat transfer.

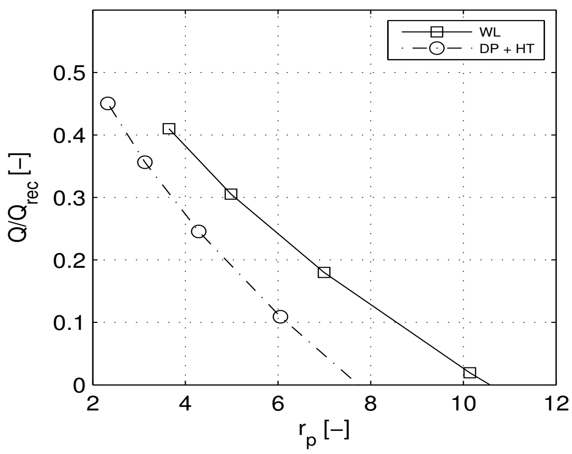

Figure 10: Influence of heat transfer on recovered heat.

RESULTS

The results presented hereunder correspond to the optimum volume ratio, i.e $\phi = 2.2$, and a mechanical efficiency of 85% is assumed. Figure 11 shows the temperatures at different points of the engine. Other results are presented in Table 3. The efficiency of the engine is 24.3% and it produces 1.6 kW of mechanical energy for 3.5 kW of thermal energy. These results show that the Ericsson engine is comparable to ORC and Stirling engines in terms of performance.

The isentropic effectiveness of the compressor and of the expander are high (see Table 3), which is due to the fact that mechanical losses are not taken into account.

Table 3: Results

$\dot{W}_{net}$	1623 W	η	24.3 %
$\dot{Q}_{th}$	3522 W	η_{th}	53 %
$\dot{M}$	14 g/s	$\epsilon_{s,E}$	89 %
r_p	4.4 [-]	$\epsilon_{s,C}$	77 %

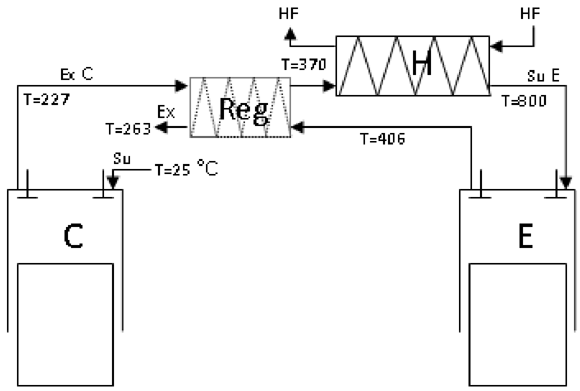

Figure 11: Schematic representation of an Ericsson engine with computed temperatures.

CONCLUSION

A simulation model of an Ericsson heat engine is proposed. This model is based on equations of conservation of mass and energy and kinematic equation of the piston. The model allows to takes into account the geometry of the valves to simulate pressure drop. It includes also heat transfer to evaluate the importance of heat losses.

Simulation of separate compressor and expander was conducted to show the influence of the two modeled sources of losses. One of the main conclusions is that, when losses are taken into account, the mass flow rate of the expander increases and the one of the compressor decreases. This leads to a lower pressure ratio when the two are coupled to form an Ericsson engine. The other conclusion is that for the expander, the heat losses are consistent and need to be taken into account. Indeed, the heat losses induce a decrease of the temperature at the exhaust of the expander and then a decrease of the efficiency with a regenerator.

Then the model was used to study an Ericsson engine. First, the influence of the ratio between the volume of the expander and the compressor was shown. The result is that there is a volume ratio that maximizes the efficiency of the engine. This optimum volume ratio is $\phi = 2.2$. For this geometry, the Ericsson engine base micro-CHP system shows a cycle efficiency of 24% and a thermal efficiency of 53%. However, the model needs to be validated. Nevertheless, it was shown that Ericsson engine deserves the same interest as Stirling engine and organic Rankine cycle.

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