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Deterministic model for the simulation of scroll expanders: preliminary results

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Abstract. *The increase of global energy consumption and the worries about global warming has attracted a great deal of interest in Waste Heat Recovery (WHR) technologies. The Organic Rankine Cycle (ORC) is one of the WHR technologies and has been shown to be more suitable in some applications when compared to the traditional Rankine Cycle. Different types of expander could be use in an ORC, scroll expanders are indicated for ORC with low power ratings generally below 10kW (micro and small-scale ORC applications). Previous studies demonstrate the difficulty of performing a numerical simulation of a scroll type expander using Computational Fluid Dynamics (CFD). These simulations are too time-consuming even with current computers, because of their transient nature. In the present work, we analyse scroll expander under ideal operation conditions. One numerical model was implemented using Engineering Equation Solver (EES) that permit verify the influence of losses by sub and over expansion. This model is a premillinary version that, when finished, will have the potential to provide results similar from those obtained using CFD in a much faster way.*

Keywords: *ORC, WHR, EES, scroll expander, deterministic model*

1. INTRODUCTION

The use of scroll geometry is well established for the design of compressors and vacuum pumps. However, the first successfully tests with scroll expander were only performed 24 years ago by Zanelli and Favrat (1994), resulting peaks of overall isentropic efficiencies in the range of 63% to 65% for speeds of rotation varying between 2400 and 3600rpm. Zanelli and Favrat (1994) also proposed the use of isentropic efficiency and filling factor parameters to characterize the performance of the machine.

A geometric model developed for a scroll compressor proposed by Blunier *et al.* (2009) aimed to be coupled with the thermodynamic model by using the standardized VHDL-AMS language allowing to verify the influence of leakages and compressor geometrical parameters on the compressor performance that can be used to improve the compressor's design. Simulations and experiments have shown good agreement.

A thermodynamic modeling of a scroll expander was presented by Peng *et al.* (2016) based on energy and mass balances using Euler explicit method and considering radial and flank leakage, heat transfer between the work fluid, scroll wraps and plates. Volume, pressure, mas flow of working chamber and power are investigated by solving the thermodynamic modeling predicting the output power very well (within 5%).

Mendoza *et al.* (2014) proposed a model to determine the scroll expander performance using semi-empirical parameters such as built-in volume ratio, leakage area and mechanical losses, obtained through experimentation, to calculate the mechanical power, exhaust temperature and supply mass flow rate and the deviations to the experimental values was respectively $\pm 9\%$, ± 4 K and ± 5 Hz.

Lemort (2008) model associates a geometrical description of the machine with a thermodynamic description of the expansion process. The evolution of the fluid in the expander is modeled with each expander chamber considered as a control volume for which the governing equations of conservation mass and energy are established and numerically solver. This model was validated for the two expanders investigated experimentally. The model validation revealed that the performance of the expanders is mainly affected by the supply pressure drop and by the internal leakages.

In this work, a deterministic model is presented, based on a transient solution of the mass and energy conservation equations. The objective of this work is to quantify the efficiency of the expander and analyze the effects of sub and overexpansion on a scroll expander. The preliminary model results present physical consistency. The difference when compared to experiments is high because internal leakages and other losses were not implemented in the model. The behavior related to these effects are going to be implemented later.

2. METODOLOGY

This section discusses the methodology used in the numerical model elaboration. Initially the volume variation of each chamber of the scroll expander was determined as a function of time. Next, energy and mass conservation equations in differential form were developed to calculate the power produced by each chamber, and consequently the total power.

2.1 Volume of the chambers

The volume of the chambers were calculated using a CAD program and the scroll geometry used to construct the prototype tested by FANTI (2017). The physical behavior in each chamber needs to be modeled. The volume of the chambers depend on the angle θ . In the figure 1 it is possible to observe the representation of the evolution of the volumes in each chamber of the expander as a function of the angle θ .

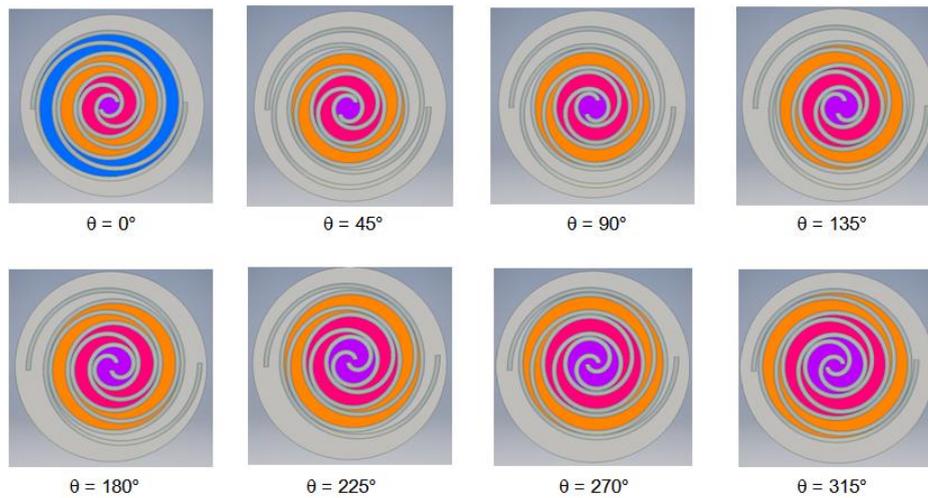


Figure 1. Volume chambers evolution as a function of angle θ .

The initial volumes (a_0) of each chamber, as well as the other polynomial coefficients given by the equation 1 are presented in table 1 and were obtained through a CAD program.

$$V_n(\theta) = a_0 + a_1\theta + a_2\theta^2 \quad (1)$$

Analysing the coefficients in table 1, it is possible to observe that the innermost chamber increases the volume quadratically as a function of the angle θ . Intermediate chambers increase the volume linearly. Finally, the discharge chamber decreases its volume quadratically.

Table 1. Polynomial coefficients for equation 1.

n	a_0	a_1	a_2
1	11,63	0,106980	0,000297167
2	77,03	0,213960	0,0
3	154,05	0,213960	0,0
4	630,59	-0,534900	-0,000297165

The volume derivative in time can be written as a function of the rotation ω of the expander and the angular derivative of the volume as shown in equation 2:

$$\frac{dV}{dt} = \frac{dV}{d\theta} \frac{d\theta}{dt} = \frac{dV}{d\theta} \omega \quad (2)$$

The volume of the chambers as a function of the angle θ is shown in the figure 2. The volumetric expansion rate of the scroll is determined by the minimum volume of chamber 2 (77.02 cm^3) divided by the maximum volume of chamber 3 (231.08 cm^3), resulting in 1:3 volumetric expansion ratio. Chamber 1 does not influence the rate of volumetric expansion and the pressure within the chamber remains close to the supply pressure during a revolution. Only the pressure drop in the inlet section can reduce the pressure at the most central point of the chamber.

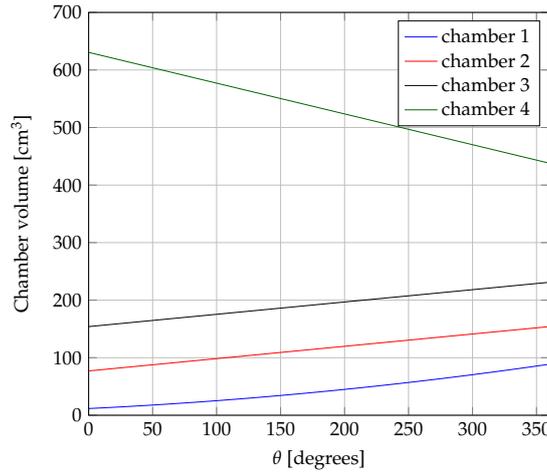


Figure 2. Volume of the chambers as function of the angle θ obtained with CAD program used to develop the expander prototype.

2.2 Thermodynamic Model

The thermodynamic model is based on a transient solution of the mass and energy conservation equations.

These two equations are solved for each chamber formed within the scroll expander modeling the evolution of the fluid expansion. The conservation of energy for a control volume is given by equation 3.

$$\frac{dE}{dt} = \dot{Q} - \dot{W} + \sum_{in} \dot{m} \left(h + \frac{V^2}{2} + gz \right) - \sum_{out} \dot{m} \left(h + \frac{V^2}{2} + gz \right) \quad (3)$$

Where: $\frac{dE}{dt}$ is the rate of total internal energy increase, \dot{Q} is the heat transfer, \dot{W} is the power, \dot{m} is the mass inflow or outflow rate, $h + \frac{V^2}{2} + gz$ is the total enthalpy.

The mass balance for the control volume is:

$$\frac{dM}{dt} = \sum_{in} \dot{m} - \sum_{out} \dot{m} \quad (4)$$

Where: M is the mass inside the scroll chamber, \dot{m} is the mass flow passing through the internal leaks located between the chambers or between the chambers and the suction (or discharge) of the scroll expander.

Energy can exist in various forms and its sum is the total energy of the system. In the case of compressible systems, the total energy of the system during a process is the sum of the changes in the internal, kinetic and potential energies.

Some hypotheses have been used to simplify the solution of the equation 3. Potential and kinetic energy are neglected. The process is considered to be adiabatic. Consequently, the total energy of the control volume E is reduced to its internal energy value U .

$$\frac{dE}{dt} = \frac{dU}{dt} = \frac{d(Mu)}{dt} = M \frac{du}{dt} + u \frac{dM}{dt} \quad (5)$$

2.3 Scroll without internal leak

A model was elaborated considering the case of a scroll expander without internal leaks. Thus, the mass change in time inside each chamber is considered zero and under this hypothetical condition the equations take on simpler forms and the problem becomes the determination of pressure variation as a function of time in each chamber.

$$\frac{dM}{dt} = 0 \quad (6)$$

$$M \frac{du}{dt} = -P \frac{dV}{dt} \quad (7)$$

In order to perform the simulation of the system, initial and final conditions need to be defined, these conditions were determined in such a way that could better represent the experimental tests performed with the prototype built. Thus, the fluid considered for the simulation was compressed air. The inlet pressure of the expander is known by the supply

air pressure. The model considers the expansion of the scroll to atmosphere. The suction temperature of the expander is the ambient temperature at the place of the experimental runs and the chamber volumes were calculated using a CAD program.

The pressure variation with time in each chamber was calculated using two properties of the fluid: internal energy and specific volume. The transient solution was performed using EES and Runge-Kutta method.

3. RESULTS

The figure 3, shows the power produced by the expander as a function of the pressure ratio. The power was calculated through the analytical model using the EES Engineering Equation Solver software. The power calculated using the model is very close to and inferior to the isentropic power (expander without losses). In fact, the two results are the same for the pressure ratio $r_p = r_v^k$, related to the volume ratio of the machine.

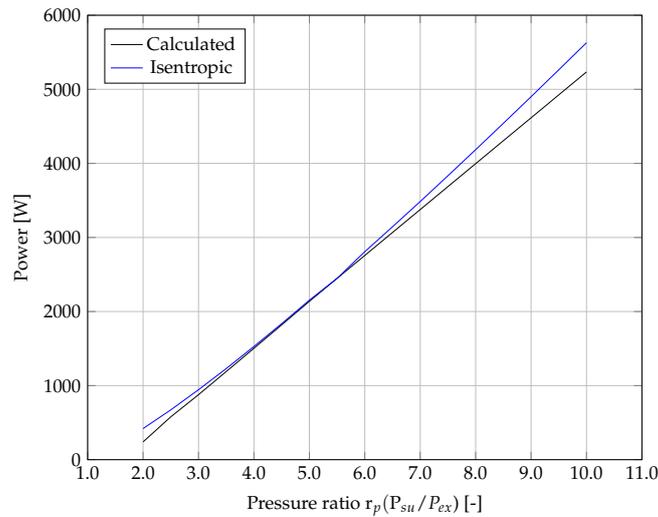


Figure 3. Expander power as a function of pressure ratio in the scroll expander at 2600 rpm

The figure 4, shows the power predicted by the model for different rotation speeds of the expander. The behavior is in agreement with previous work from literature and experiments conducted by Fanti *et al.* (2016) and Romão (2017).

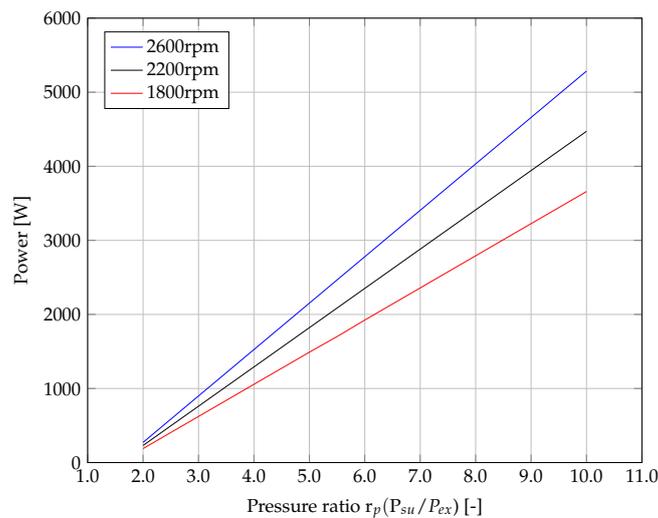


Figure 4. Expander power as a function of pressure ratio in the scroll expander for different rotation speeds

The isentropic efficiency is showed on the figure 5, as a function of pressure ratio. The highest value is obtained with $r_p=4.65$. This result was expected since no other losses are included in the model, only sub and overexpansion losses.

Finally, figure 6, shows the power predicted with the model compared to experiments with the prototype. The high disagreement is caused by the losses that were not included in the model.

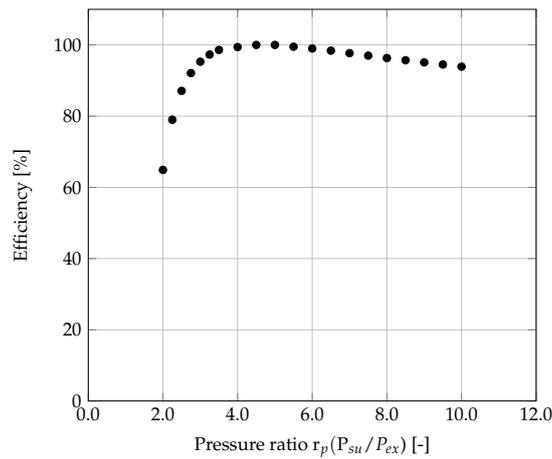


Figure 5. Evolution of the efficiency in percentage with the pressure ratio

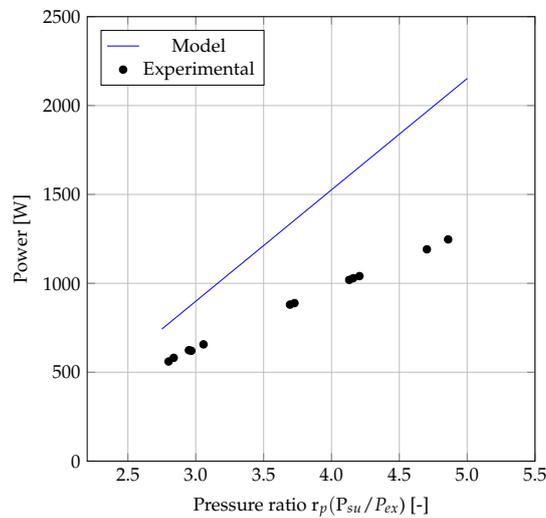


Figure 6. Comparison between calculated power through the model and experimental values obtained through the prototype

4. CONCLUSION

The model allowed to determine the isentropic efficiency and the total power over time and as a function of the pressure ratio of the scroll expander. The results have shown that the total power predicted by the model is lower than the isentropic power when the pressure ratio is different than an optimum value. The result have evidenced the occurrence of under and over-expansion.

However, the model still does not take into account the power losses caused by the following factors: internal leakage, mechanical losses and heat transfer with the external medium that impact the efficiency of the scroll expander. These effects will be included into the model in order to improve its accuracy.

5. ACKNOWLEDGEMENTS

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