CHARACTERISATION OF AN EXPANDER FOR SMALL SCALE ABSORPTION POWER AND COOLING SYSTEMS ACTIVATED BY SOLAR ENERGY

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Abstract

In an absorption cycle the energy in the form of ammonia at high pressure at the outlet of the generator could be used to produce mechanical energy. In this way solar absorption cooling systems could be converted into combined power and cooling systems driven by solar thermal energy at medium temperature. The main required operating conditions for a suitable expander for this application are used in this paper for its design and simulation. The numerical modelling and simulation of the expander is presented as a function of their main input parameters in a second stage the simulation is extended to include all the subsystems: solar thermal collectors and the absorption cooling and power system including the expansion unit modelled according to the results obtained with the expander simulation and validation. Finally, the design of a test rig for the experimental characterisation of this expander is also presented.

1. Introduction and objectives

1.1. Solar Power and cooling system

Today, in the air conditioning sector the dominant cooling systems are electrically driven compression chillers. To reduce the primary energy consumption of chillers, thermal cooling systems are an attractive alternative technology if primary energy neutral heat from solar energy or waste heat is used to produce cooling. It is estimated that the current number of solar cooling systems in Europe is quite low, about 250 with an installed cooling capacity of 12 MW [1]. Due to their favourable climatic conditions Spain is one of the European countries where the development of solar cooling is more attractive and so far concentrates 27.5 % of all the solar cooling installations in Europe only behind Germany (39.1 %) and before Greece (8.7 %) [1].

If solar thermal energy is used in summer for building air conditioning the seasonal cooling loads coincide with the highest solar radiation availability. However, it is desirable the exploitation of the available thermal solar energy throughout the year for power, heating and cooling because it could increase the primary energy saving of a solar thermal installation and the economics of the investment. In this case when the heating or cooling demand is low and the thermal storage is not necessary, the excess of solar thermal energy could be converted into power instead of convert it into heating or cooling.

1.2. Basic principle of the scroll expander

One of the most important components in combined power and cooling systems is the expander. Several types of expander could be used such as turbines, reciprocating expander, screw, scroll, etc. For the case of small-scale power and cooling systems the scroll expander has been selected among all the displacement type machines for its reduced number of moving parts, reliability, wide output power range, and good availability [2].

The first studies about generation of mechanical power using modified scroll compressors came from the work of Moore [3]. The scroll expander or compressor consists of two spiral-shaped elements, one stationary and other rotatory with orbital motion around the center axis. Both spirals are identical and are assembled with a difference of 180 degrees, the expansion process occurs through different chambers ranging from the centre outwards, as is shown in the Figure 1. The expansion process occurs when the volume of the chambers increases along the process and according to the angle of rotation.

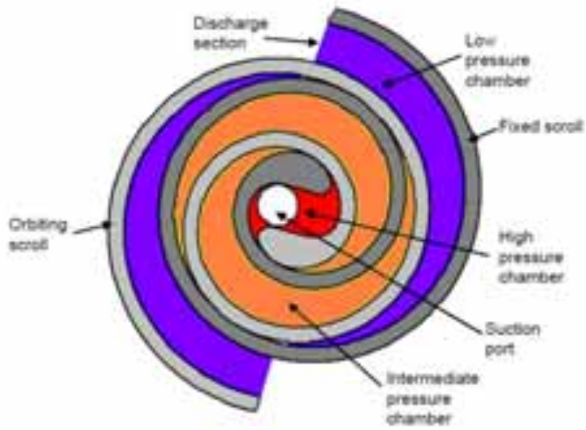


Figure 1. Schematic cutaway view of a scroll expander.

The scroll expander is in the group of positive displacement rotary machines, and is a better option than turbo machines for energy recovery applications at low temperatures. The main advantages of an expander scroll base don the following operation characteristics [4]:

- The performance of the thermal machines depends on the rotation speed around its periphery. The optimal operation conditions in the scroll expander are in the range 1 - 10 m/s and the turbo machinery is close to 300 m/s. [4]. This produce the necessity of fluids with high velocity and it is not available in the combined power and cooling cycle.
- The volumetric machines are more resistant to gas condensation than any turbo machine because the condensation produce corrosion in the blades [4].
- The pressure ratio of a single-stage turbo machine is close to 1.5, while the positive displacement rotary machines have higher expansion ratios [5].

1.3 Objectives

The main objective of this paper is the modelling of a scroll expander suitable for small-scale combined power and cooling (flats). Later this model is implemented in a complete cycle model to determine the global performance of a power and cooling cycle. Finally, it is presented a test bench under construction for the characterisation of the expander.

2. Modelling of the expander

The expander model is based on the setting of geometric parameters ($\eta_{vol}, \eta_{mec}, \eta_{is}$) given the input parameters ($\dot{m}_{in}, P_{in}, T_{in}, P_{out}$), to obtain as main results the outlet temperature and the output power using the corresponding properties thermo physics is used. This model is developed in the Engineering equation Solver programming environment (EES) [6].

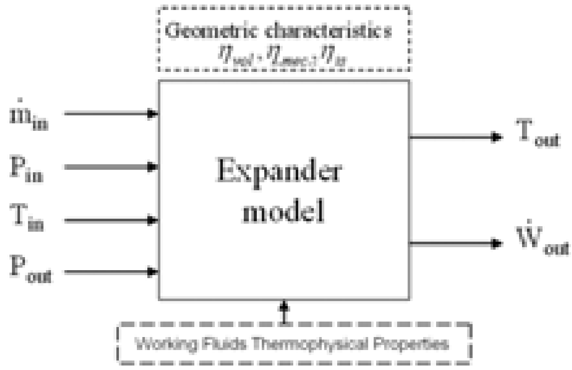


Figure 2. Mathematical model of scroll expander

Thus the overall operation of the expander is a function among other parameters of three specific geometric parameters of the expander and thermo physical properties of the working fluid. The expander model is build following these assumptions:

- The behaviour of the working fluid is treated as ideal throughout the expansion process [7].
- The heat exchanged between the gas and the walls is ignored.
- It is considered that at the inlet and outlet sections of the expander the fluid has not variations in temperature or pressure. The change of potential and kinetic energy at the inlet and outlet is neglected.

2.1. Mass and energy balances

Taking a control volume around the expander and the mass and energy balances are the following assuming ideal gas behaviour.

$$\dot{Q}_{In} - \dot{Q}_{Out} - W_{loss} - W_{out} = 0 \quad (1)$$

$$\dot{m} = \frac{V_{real} N \rho}{60} \quad (2)$$

$$\gamma = \frac{Cp}{Cv} \quad (3)$$

$$\dot{Q}_{In} = \dot{m}_{In}\left(\frac{\gamma}{\gamma-1}\right) RT_{In} \quad (4)$$

$$\dot{Q}_{Out} = \dot{m}_{In}\left(\frac{\gamma}{\gamma-1}\right) RT_{out} \quad (5)$$

$$W_{th} = \dot{m}_{In}\left(h_{is} - h_{In}\right) \quad (6)$$

$$T_{out} = T_{in} \frac{P_{out}}{P_{In}}^{\left(\frac{\gamma-1}{\gamma}\right)} \quad (7)$$

$$W_{out} = \dot{m}_{In}\left(\frac{\gamma}{\gamma-1}\right) RT_{In}\left(\left(\frac{P_{Out}}{P_{In}}\right)^{\left(\frac{\gamma-1}{\gamma}\right)} - 1\right) \quad (8)$$

2.2 Efficiency parameters

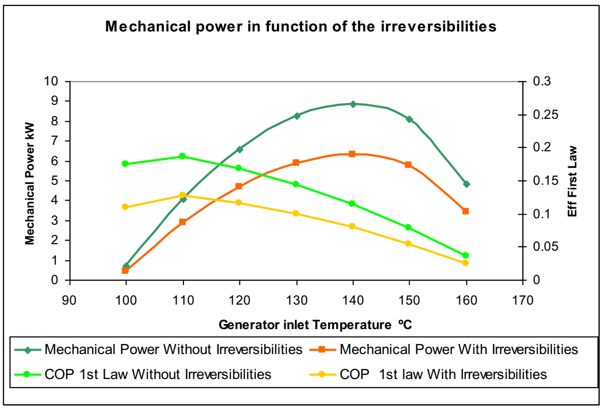
$$\eta_{ie} = \frac{W_{out}}{W_{th}} \quad (9) \qquad \eta_{is} = \frac{\eta_{ie}}{\eta_{mec}} \quad (10)$$

$$\eta_{vol} = \frac{V_{real}}{V_{th}} \quad (11) \qquad \eta_{mec} = \frac{\dot{W}_{out} - \dot{W}_{loss}}{\dot{W}_{out}} \quad (12)$$

The indicated isentropic efficiency is defined as Ec. (9), where W_{net} is the work of the ideal expansion process. There is no resistance loss and the over and deficient expansions do not occur, although they exist in the practical process. W_{net} is calculated by Ec. (6). Taking into account the mechanical losses, the isentropic efficiency η_{is} can be found as Ec. (10), the process of expansion depend of γ, this is shown in the Ec. (3) and the value is typical for each fluid.

3. Results

3.1 Simulation of the expander



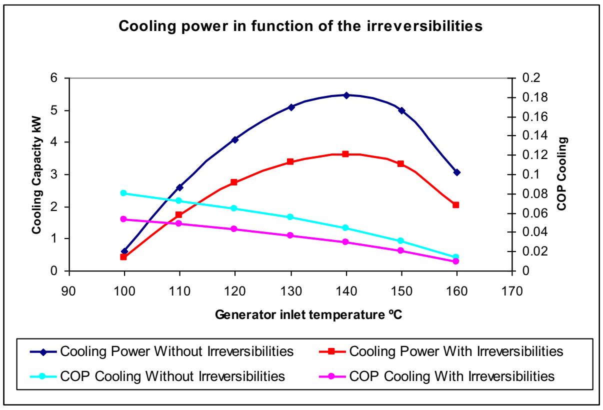


Figure 3. Variation of cooling and me chanical as a function of irreversibilities.

Figure 3 shows the influence of the irreversibilities (isentropic, volumetric and mechanical efficiency) in the simulation of the expansion process, using a mechanical efficiency $\eta_{mec} = 0.9$, volumetric efficiency $\eta_{vol} = 0.9$, and isentropic efficiency $\eta_{is} = 0.8$, as a function of the expander inlet temperature that will correspond to the generator outlet temperature in a combined power and cooling cycle.

3.2 Simulation of a combined power and cooling cycle

A simulation of the cycle proposed by Goswami (Figure 4) [8] and the operating parameters proposed by Montero et al[9], is realized with the expander, with the simulated cycle is finding the necessary heating for activate the cycle. The cycle simulation is build following these assumptions:

- The vapour fraction at the inlet of reheater is 1.
- The vapour fraction at the absorber outlet is 0.
- The concentration of ammonia at the inlet of expander is 0.999.

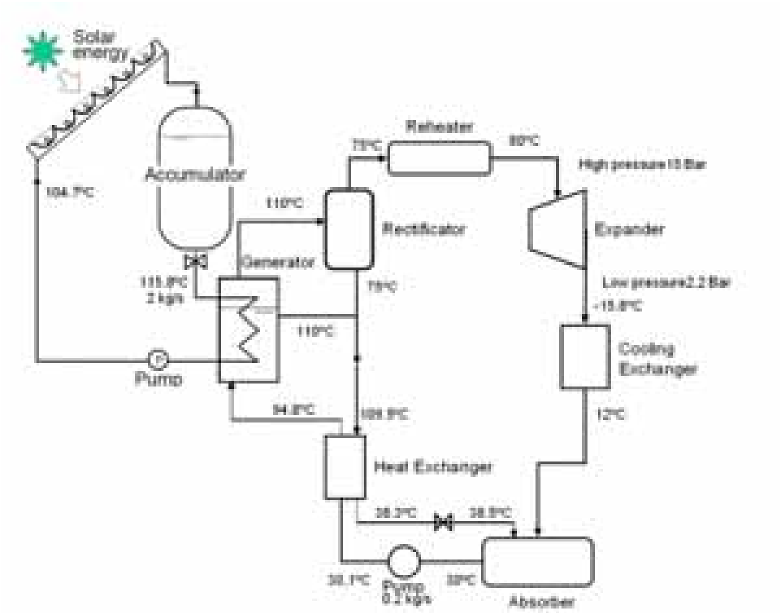


Figure 4. Combined power and cooling cycle proposed by Goswami [8].

Table 1 shows the parameters and results of the simulations of cycle with the expander to study. The principal results are that the heat necessary to drive the cycle is near to 36 kW, and COP First Law of 12.7 % and 2.3 % COP cooling.

Expander Performance	Expander Volume	53.9	cm3/rev
	Rotation Speed	2265	rpm
	High Pressure	15	bar
	Low Pressure	2.2	bar
	Isentropic Eff.	0.8	-
	Volumetric Eff.	0.9	-
	Mechanic Eff.	0.9	-
Power and Cooling Cycle	Generator	35.76	kW
	Rectificator	6.79	kW
	Reheater	0.24	kW
	Expander	3.15	kW
	Cooling Exchanger	1.72	kW
	Absorber	27.75	kW
	Pump	0.29	kW
	Heat Exchanger Solutions	64.63	kW
Final results	COP Cooling	0.048	-
	COP First Law	0.127	-
	Expander Net Power	2.85	kW
	Cooling Capacity	1.73	kW

Table 1. Parameters and results for the simulations cycle proposed by Goswami [8].

The cycle first law or thermal efficiency is typically defined as the useful energy output divided by the total energy input given by Ec. (13). The net work includes both the expander output and pump input.

This definition for first law efficiency can be deceiving, as the available energy in refrigeration is less than that in work. Addition of work and cooling output in a single cycle is a new idea. Finding and justifying an appropriate definition for first – and second law efficiencies including both work and cooling is a new concept and is being explored as motivated cycle.

$$Eff_{first\_law} = \frac{W_{net} + Q_{cool}}{Q_{gen} + Q_{rh}} \quad (13) \qquad COPcool = \frac{Q_{cool}}{Q_{gen}} \quad (14)$$

With the previous information is picked a solar installation, taking care the relations cost/efficiency, in the next table is proposed by [10]. Shown the possible solar technologies for several operations temperatures and the best options it is the ETC-CPC collector at 120ºC.

Solar technology Uptake	Cost	Energy Efficiency			Cost/efficiency		
		70ºC	90ºC	150ºC	70ºC	90ºC	150ºC
FPC	+++	++++	+++	-	++++	+++	-
FPC-IRC	++++	++++	++	-	++++	++	-
FPC-CPC	+++	++++	+++	-	+++	+++	-
FPC-Air	+++	+++	+	-	+++	+	-
ETC	++	++++	++++	+++	+	++	+++
ETC-CPC	+	++++	++++	+++	+	++	+++
+ Poor	++ Inadequate		+++ Good			++++ Excellent	

Table 2. Parameter of CPC solar collector proposed by Lopez [10].

4. Test Bench

The main objective of the test bench is the characterization of the expander to experimentally obtain the variables that affect the different geometric parameters, using the adequate measuring devices. We will develop two test bench, one for air and another one for ammonia.

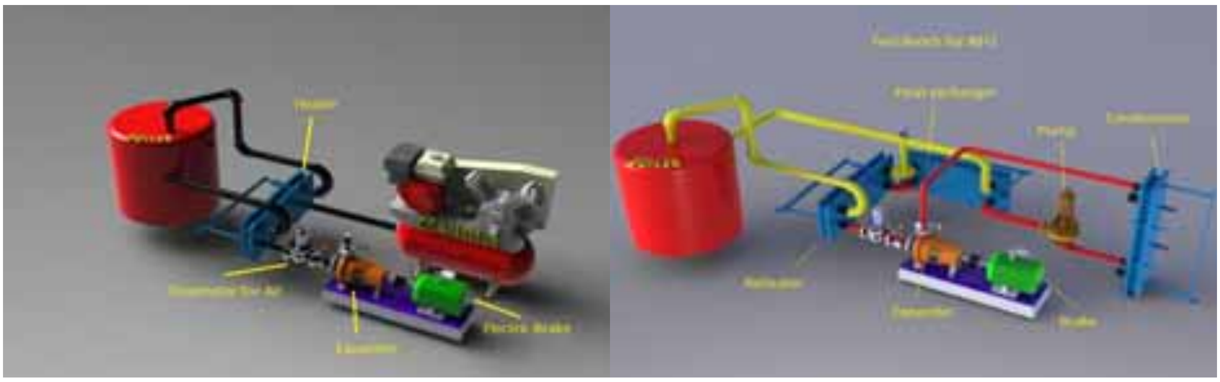


Figure 5. (a) Test Bench with Air **(b)** Test Bench with Ammonia

The tests benches shown in Figure 5 are responsible for the conditioning and measuring of the working fluids at the expander, inlet measuring the temperature, flow, pressure and torque.

The air test bench Figure 5 (a) consists of a compressor that delivers air at high pressure and dry, then passes through a heat exchanger to heat the air at temperature desired and then goes through the expander. This test bench is an open cycle where air that leaves the expander is not reused.

The ammonia test bench Figure 5 (b) is a closed cycle working with ammonia, it is very similar to a Rankine cycle. The ammonia is evaporated to the desired pressure and then reheated to bring it to the desired temperature, passes through expander and then condenses and flows back to the evaporator.

Figure 6. Scroll expander to be characterized in the test bench.

The figure 6, shown the scroll expander to be studied, this expander can work with a flow rate of 53.9 cm^3 per revolution. It is maximum temperature working is 145 ºC and the maximum pressure is 25 bar.

5. Conclusions

It has been built a mathematical simulation model which shows the system performance as a function of the geometric parameters of the expander. It is concluded that the volumetric efficiency and isentropic efficiency of the expander have a great influence in the cycle performance. Thus an experimental evaluation is needed.

It is proposed the experimental characterization of a scroll expander coupled to a combine cycle power and cooling power activated with solar energy, also the design of two test benches for their characterization and possible solar installation for activation of the combined cycle power and cooling.

Nomenclature

			Subscripts	
$\dot{m}$	Mass flow rate	(kg/s)	exp	expander
$\dot{Q}$	Heat flux	(kW)	ie	Indicated isentropic
T	Temperature	(K)	is	isentropic
η	Efficiency	(-)	vol	Volumetric
γ	Polytrophic coefficient	(-)	mec	Mechanical
R	Universal Constant Gas	kJ/kg K	in	Inlet
$\dot{V}$	Volumen flow rate	(m3/min)	out	Outlet
$\dot{W}$	Mechanical Power	(kW)	loss	Loss
P	Pressure	(bar)	ideal	Ideal
N	Revolutions	Rev/min	amb	Ambient
ρ	Density	kg/cm^3	cool	Cooling
h	Enthalpy	kJ/kg	th	Theoretical
rp	Pressure relation	(-)	gen	Generator
IRC	Integrate Roof Collector	(-)	rh	Reheater
ETC	Evacuated tube collector	(-)		
FPC	Flat Plate Colector	(-)		
CPC	Compound Parabolic Collector	(-)		

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