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Model of H2O/LiBr absorption machine using falling films

Jean-Francois Cap[1], Benoit Stutz and Fredy Huaylla (jean-francois.cap@univ-savoie.fr)

Laboratoire LOCIE UMR 5271, Université de Savoie CNRS, Polytech Annecy-Chambéry, Campus scientifique Savoie Technolac. 73376 Le Bourget Du Lac, France

Abstract

The study concerns the development of a 10kW solar powered Lithium Bromide absorption machine devoted to air conditioning of efficient buildings. This paper describes a numerical model of a new concept of absorption machine working with falling films exchangers. In addition to offer both a mass and heat exchange on a plate, this technology provides a cheaper, compact and sturdy machine. Thus the evolution of temperatures, concentrations and flow rates are studied along the plate. The effect of the temperature of heat-transfer fluids evolutions, the flow rates variations of falling film and the length of plate are investigated.

1. Introduction

The interest about absorption machine in terms of reduction of the global electricity consumption in cooling systems has been known for quite a long time (Carré, 1857) but it continues to be massive and expensive for building applications [1]. These machines are fully adapted for warm climates where the need of cooling is significant. This process offers a good alternative against standard refrigeration compression machines. The absorption machine works like a standard refrigerant system by substitution of mechanical compression by chemical compression. The cold water is produced by evaporation of the refrigerant fluid (H2O). Then the vapor is absorbed with an absorber by a solution of lithium bromide at low pressure (10mb). Afterward, the weak absorbent solution is heated to release the refrigerant of the solution in a high-pressure chamber. The strong solution is re-injected in the absorber and the steam produced is condensed at high pressure to be evaporated again at low pressure [2]. These machines are particularly interesting if the hot source necessary to desorb the weak solution is free, unexploited (heat waste) or renewable (solar panel). Thus, the absorption machine although less effective than mechanical systems, can reduce the global electricity consumption [3]. The thermal coefficient of performance (COPth) of absorption machines is approximately between 0.5 and 0.75.

For our application the refrigerant is water and the absorbent is lithium bromide [4]. The hot sources derived from solar thermal collectors, the intermediate source used to cool the condenser and absorber comes from a spiral ground heat exchangers. The machine is composed of four exchangers ensuring the function of absorption, evaporation, desorption and condensation. These exchangers are formed with vertical grooved flat plates and falling films to ensure the mass and heat transfer (figure 1). With this technology, we can obtain a simple geometry and a great compactness due to the possibility to cumulate a large number of plates unlike standard machines with shell and tubes exchangers. Vertical grooves along the plate ensure a uniform distribution of the working fluid at low Reynolds numbers.

We have a nodal model to describe the heat and mass transfer along the plate [5]. An iterative proceeding provides the stationary functioning point of the machine. In this paper, we are interested in the evolution of concentrations and temperatures, as well as the operating conditions on the performance of the machine. Impacts of the exchanger geometry and the recirculation rate on the machine are discussed.

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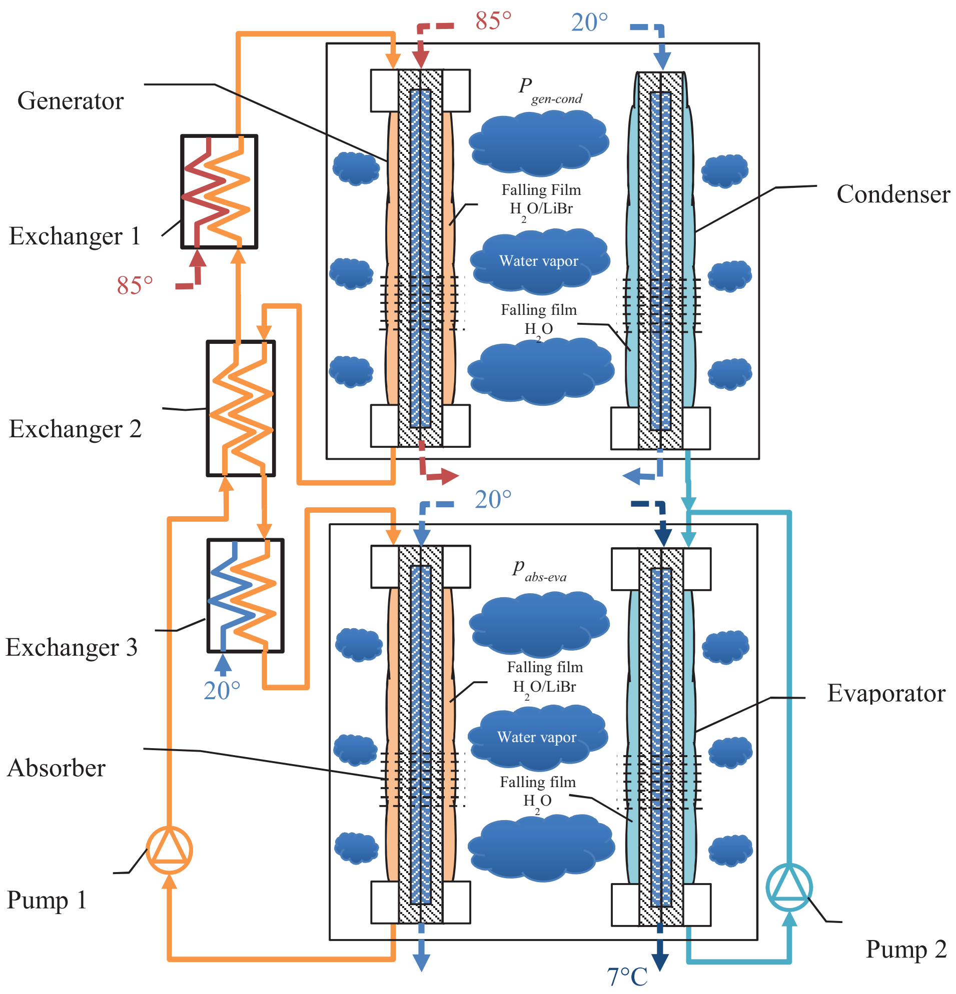

Fig. 1: Outline schematic machine diagram

2. Model description

Two previous studies have led to a sturdy and reliable numerical architecture to describe the absorption machine using falling films [5] [6]. This model is developed with Matlab software and includes three stages as shown in figure 2.

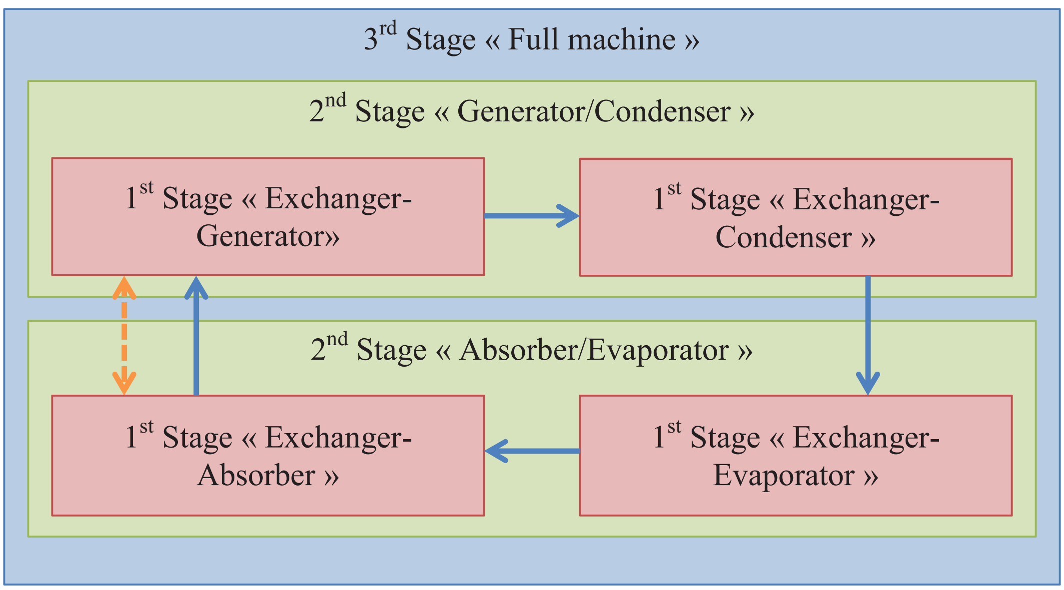

Fig. 2: Model architecture

The first stage simulates the flat plate exchangers with falling films. We can obtain the concentration, temperature and flow rate evolution along the exchanger due to some thermodynamic principles [6].

The second stage involves the exchangers evolving at the same surrounding pressure. The solution is determined by a mass balance between the absorbing mass and the evaporating mass for the absorber/evaporator and between the generating mass and condensing mass for the generator/condenser.

$$\dot{m}_e = \dot{m}_a \quad ; \quad \dot{m}_d = \dot{m}_c \qquad \text{(eq. 1)}$$

The third stage shows the full machine which connects all elements between them and incorporates intermediate heat exchangers between the absorber and the generator. These exchangers improve considerably the performance of the machine, in particular the exchanger 2 heats the weak solution and cools the strong solution. Thus it is easy to obtain the stationary operating regime for given input conditions.

3. Nominal case

To analyze the performance of the machine with the input conditions, it is necessary to choose a nominal case. The chosen values (table 1) are limited by the technology and the surroundings of the machine.

Tab. 1: Nominal case parameters

Parameters	*Values*	Parameters	*Values*	Parameters	*Values*
L	0,3 [m]	T_f	10 [°C]	$\Delta p_{pump\ 1}$	40000 [Pa]
l	1 [m]	T_h	85 [°C]	$\Delta p_{pump\ 2}$	10000 [Pa]
e_{plate}	1.10^{-3} [m]	T_{in}	20 [°C]	*Eff 1*	80 [%]
e_{fluid}	2.10^{-3} [m]	$\dot{m}_l$	1,6 [kg/s]	*Eff 2*	80 [%]
k	16 [W/m.K]	*TR*	0 [%]	*Eff 3*	80 [%]
n	30 [noeud]	η_{Pumps}	50 [%]	$V_{estimated}$	0.32 m^3

Vertical grooves allow a good film distribution on the plate but it restricts the wavy development that steps up transfers [7] [8]. Given the flow rates implemented, the flow regime is laminar along the plate. A study concerning the influence of the flow rate solution (figure 3) is achieved to determine the nominal flow rate, which obtains a COPth of more than 0.5:

$$COP_{therm} = \frac{P_f}{P_h} \quad ; \quad COP_{System} = \frac{P_f}{P_h + P_{pumps}} \qquad \text{(eq. 2)}$$

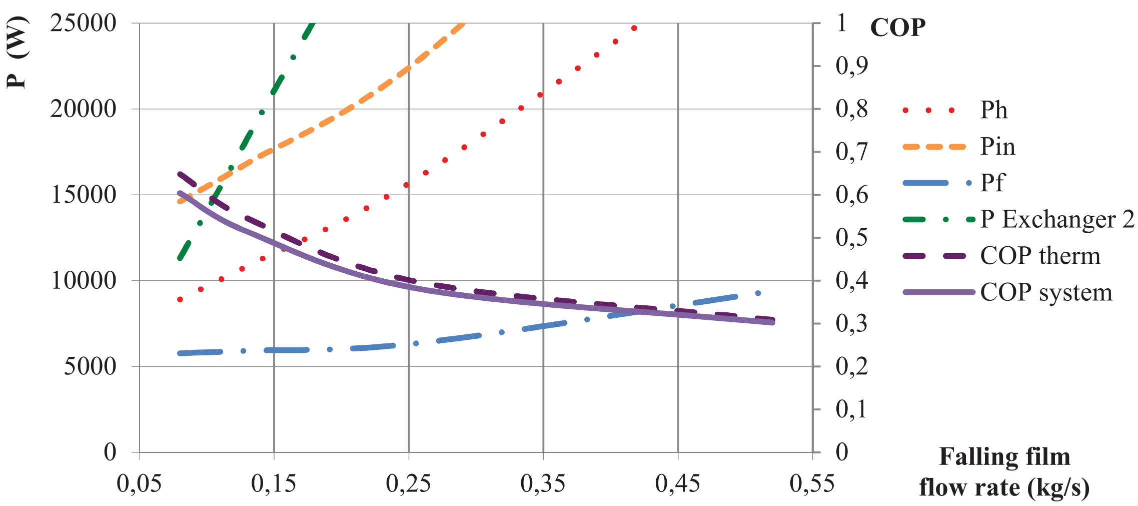

Fig. 3: Powers and COP depending on the flow rate

When the flow rate solution increases, the refrigeration power increases at the same time (9.4 kW) but the COP decreases considerably (0.3). To conserve a reasonable COP, it is necessary to conserve a flow rate lower than 0.15 kg/s. The absorber is the critical part of the system because we have both mass and heat transfer working at low pressure. The diffusion of the water vapor in films and the heat induced by absorption at the interface require considering the boundary condition displacement. Thus, we analyze the temperature and the LiBr mass fraction evolution along the plate of the absorber for nominal conditions and 0.13kg/s flow rate.

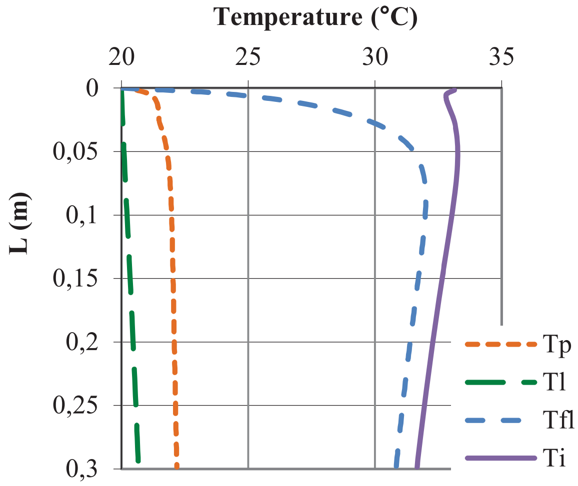

Fig. 4: Temperature depending on height in the absorber

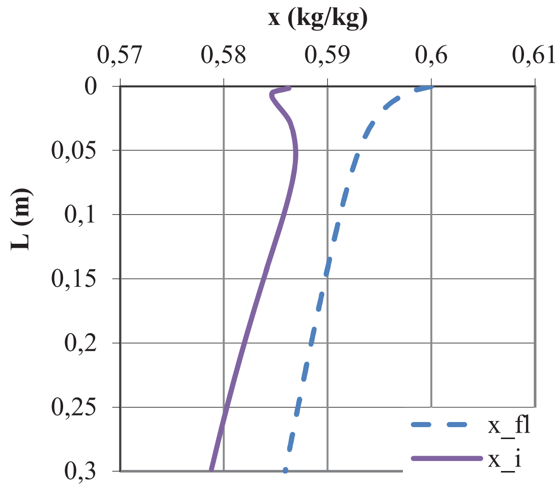

Fig. 5: Mass fraction depending on height in the absorber

The absorption phenomenon is followed by a sudden increase of temperature at the interface (14°C) at the beginning of the solution film (figure 4), then decreases owing to the heat diffusion in the film. The film temperature increases considerably on the first one third of the film before decreasing on account of the cooling by the intermediate source. The temperature increase of the wall and the cooling fluid stayed moderate with this given flow rate.

The interface mass fraction also undergoes a sudden decrease on the entrance area (figure 5) then decreases regularly along the plate.

The nominal case gives the following performances:

Tab. 2: Nominal case results

P_h (Exchanger 1 + Generator) [W]	**10850**
P_{in} (Condenser + Absorber + Exchanger 3) [W]	**16830**
P_f (Evaporator) [W]	**5910**
COP_{therm}	**0,55**
COP_{system}	**0,51**

We have a 5.9kW cooling power for 1m width and 0.3m length (table 2), the estimated volume is $0.32m^3$. A cooling power of 10kW can be obtained by 6 plates (0.3 width, 0.3 length, $\sim 0.48m^3$) with a heating capacity of 19.5 kW.

4. Impact of operating conditions

Now we can observe the influence of operating parameter conditions on performances. With the solar thermal collector, we have a significant temperature fluctuation owing to the weather and sunshine conditions. Moreover, the geothermal source is also inconstant and the variation of intermediate temperature is not negligible. That's why, it is interesting to see the effect of high and medium temperatures on the machine.

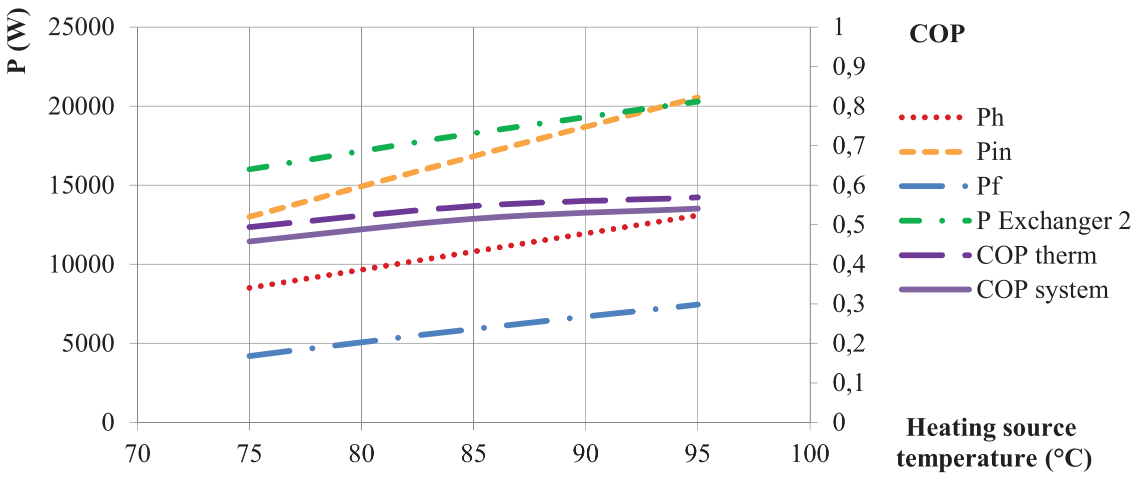

Fig. 6: Influence of heating source temperatures depending on powers

A high temperature (95°C) in the heat source involves a general growth of powers and COP. We are in the favorable case because the cooling needs are as much more important as the climate is hot (figure 6). To satisfy a cooling capacity of 10kW with a temperature source of 95°C, 5 plates (0.3 width, 0.3 length, ~0.42m^3) are necessary with a heating power of 19.5 kW.

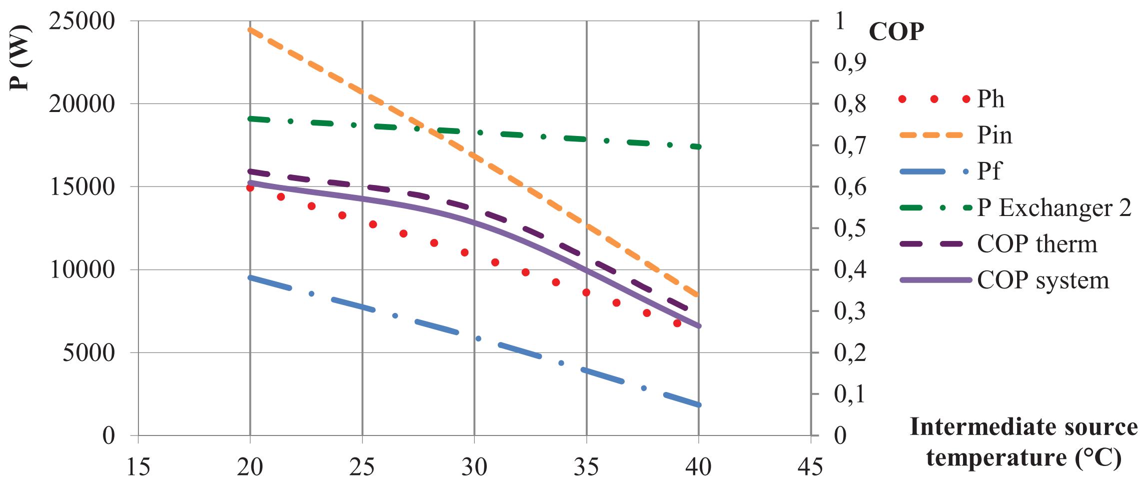

Fig. 7: Influence of intermediate source temperature depending on powers

We can notice the reverse phenomenon for the intermediate temperature. When we have high temperature (40°C) in spiral ground heat exchangers, the cooling capacity and the COP decrease strongly (figure 7). These two effects are logical, the difference in temperature between the hot and intermediate sources impacts directly on the capacity of the machine. Therefore the higher the difference in temperature, the higher the machine powers.

5. Optimisation

A long plate increases the capacity and the COP of the machine (figure 8). A 50cm long plate delivers a cooling capacity 1.7 times more impacting than a 25cm long plate. An optimal length plate has yet to be found regarding to the taken volume and cooling capacity. A 10kW cooling power can be ensured by 4 plates (0.3 width, 0.5 length, ~0.48m3) with a heating power of 16.5kW.

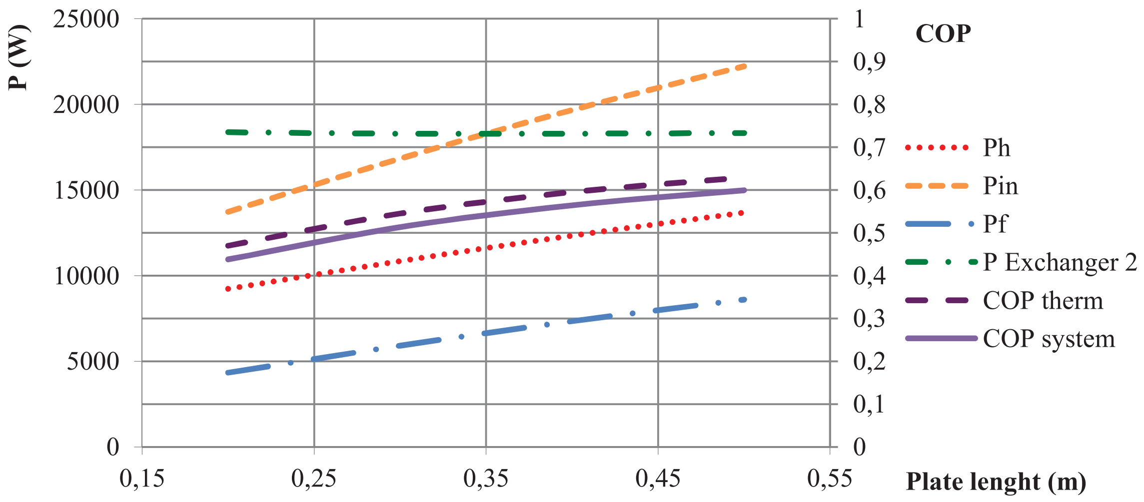

Fig. 8: Influence of length plate depending on powers

The recirculation rate is defined by the following figure:

$$TR = \frac{\dot{m}_{recirculation}}{\dot{m}_{out\_exchangeur}} \qquad \text{(eq. 3)}$$

With : 0 < TR < 1

TR = 0 => No recirculation

TR = 1 => Total recirculation

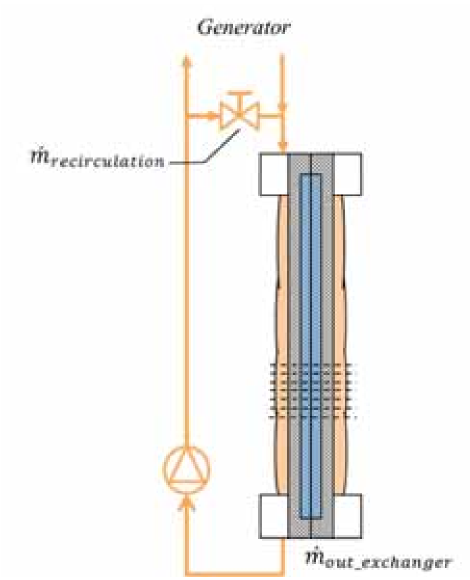

Fig. 9: Recirculation

A recirculation of the solution is implemented only for the absorber and generator. This method is used to recirculate a portion of the weak solution in the absorber and the strong solution in the generator. Also, it reduces the flow rates solutions between absorber and generator and helps to reduce the volume of the intermediate exchanger. A recirculation is implemented in several standard machines and allows the increase of the difference in concentration between the weak and strong solution but in this paper the definition of recirculation rate is a bit different (eq.3). We can notice the important influence of the recirculation rate on the COP of the machine in figure 10. Indeed with a recirculation rate of 0.8, we obtain a thermal COP of 0.8 and a system COP of 0.72. To satisfy a cooling capacity of 10kW with nominal conditions and a circulation rate of 0.8, 6 plates (0.3 width,0.3 length, ~0.48m^3) are necessary with a heating power of 13.5kW.

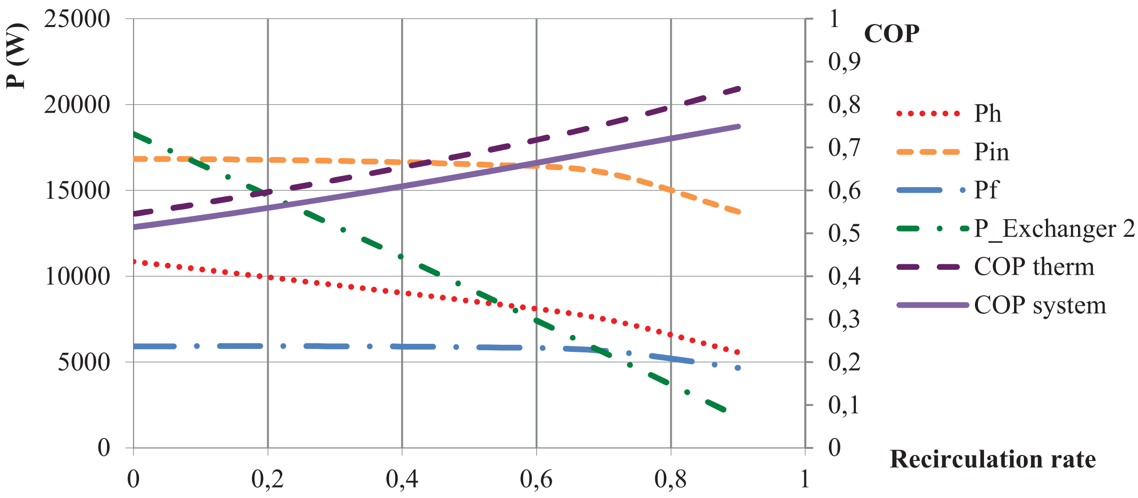

Fig. 10: Influence in recirculation rate depending on powers

In chapter 3 the increase of the flow rate solution intensified the cooling capacity (figure 3) when previously we showed that the increase of recirculation rate enhanced the COP (figure 10). That's why an optimal case with a four times more important flow rate associated with a recirculation rate of 0.9 has been studied (Table 3). Thus, to satisfy a refrigeration power of 10kW, 4 plates (0.3 width and 0.3 length) are necessary with a heating power equal to 14.5kW. The final volume of the machine in this case is about 0.35m3.

Tab. 3: Optimum case results

P_h (Exchanger 1 + Generator) [W]	**12015**
P_{in} (Condenser + Absorber + Exchanger 3) [W]	**20763**
P_f (Evaporator) [W]	**9171**
COP_{therm}	**0,75**
COP_{system}	**0,7**

6. Conclusion

To conclude, it is important to have efficient thermal solar collectors and geothermal systems. Their influences on the temperature are significant and impact directly on the performance of the machine. A study on the performance of the machine with the evolution of the temperature during one year would be interesting.

With the research of the optimal case, we observed that a 42 cm high plate obtains a COP of more than 0.6 with the nominal condition. This parameter is very important to make the most compact machine. However, we have another solution with the introduction of the recirculation rate and this would allow achieving COPs of more than 0.7. The optimum case advertises a COP of 0,75 with a cooling capacity of 9kW and a small volume.

Of course these results are numerical and the technical problems to realize the machine are many. But it is relevant to know the optimization axes and their degree of influence on the machine. The model must still be matched with the experiment.

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7. References

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Table 4: Use of units and symbols

Quantity	Symbol	Unit
Plate thickness	e_{plate}	m
Channel thickness heat-transfer fluid	e_{fluid}	m
Exchanger efficiency	*Eff*	
Thermal conductivity	k	W m^{-1} K^{-1}
Plate length	L	m
Plate width	l	m
Mass flow rate	$\dot{m}$	kg s^{-1}
Mesh node number	n	
Power	P	W
Pressure	p	Pa
Temperature	T	°C
Recirculation rate	*TR*	
Mass fraction	x	kg kg^{-1}
Pressure drop	Δp	Pa
Efficiency coefficient	η	%

Subscript	Symbol
Absorption	a
Condensation	c
Desorption	d
Evaporation	e
Refrigeration	f
Falling film	fl
Heating	h
Interface	i
Intermediaite	in
Heat-transfer fluid	l
Plate	p