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Low capacity diffusion absorption refrigeration: Experiments and model assessment

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Abstract— **In this paper, a low capacity commercial absorption-diffusion refrigeration machine using ammonia/water/hydrogen as working fluids, was tested under different operating conditions in order to master the operation of the refrigerator and to study its performances. A series of thirty experiments were realized which consisted of varying the electrical power supplied to the generator from 15W to 63W in order to identify the optimal operating conditions of the machine and to study the various operating modes. All these experiments are all performed in an air-conditioned room at 26°C. It is further noted that the optimal COP found is 0.15 for a cooling capacity of 7W.**

Keywords— **Refrigeration machine; absorption-diffusion; experimental study; performance.**

I. Introduction

The absorption-diffusion refrigeration machine, the subject of our study, was invented by Platen and Munters in 1928 [1] and has been recognized as one of the most encouraging sustainable technologies for cold production. The cycle of the machine (Figure 1) operates at a constant total pressure, and uses ammonia as a refrigerant, water as an absorber and hydrogen or helium as a non-absorbable auxiliary inert gas. This inert gas is necessary to reduce the refrigerant partial pressure in the evaporator to allow the evaporation process to take place in the uniform pressure device.

The main feature of this machine is that it has no moving parts, hence its good reliability. The circulation of the aqueous ammonia solution is driven by a bubble pump and that of gas flows between the absorber and the evaporator by gravity. There are many theoretical and experimental studies on the analysis of the performance of this machine operated with different energy sources and using various mixtures of fluid work in the literature. Mansouri and al [2] has carried out experimental investigations of a commercial diffusion absorption refrigerator (DAR) cycle and developed a detailed steady state simulation model of this cycle using Aspen-Plus. He has found deviations between model predictions and experimental measurements in terms of cooling capacity and coefficient of performance are less than 1%. Experimental tests on a domestic DAR were performed by Ben Jemaa and al [3]. The refrigerator was modelled in dynamic mode for different electric heater powers input. He has used the black-box technique for the prediction of the transient behavior of the commercial DAR. Starace et al. [4] developed a thermodynamic model of the absorption-diffusion cycle without any hypothesis regarding the purity of the refrigerant leaving the rectifier. Using this model, he compared the performances of the machine with that of another thermodynamic model proposed by Zohar et al. [5]. Greater accuracy has been shown in predicting the actual state of the machine. In another study, Starace et al. [6] experimentally validated their model using a prototype of a bubble pump coupled to a home magnetron to reduce the start-up transient of the circuit. In order to validate the model, he varied the thermal power supplied to the heat pump, he tested the operating conditions of the machine in each element of the machine.

In this paper, an experimental study is carried out on a low-power commercial refrigerator with a capacity of 25 liters, powered by an electrical resistance and which operates according to the Platen and Munters cycle. This machine involves the ammonia/water as refrigerant/absorbent and hydrogen, pressure equalizer, as an inert gas. It consists mainly of a generator, a rectifier, a condenser, an evaporator, a gas exchanger, an absorber, a liquid reservoir and a solution exchanger. All these elements are made of steel.

II. EXPERIMENTAL APPARATUS

A. *Description of the cycle*

When the machine is energized, the electrical resistance begins to heat the ammonia-rich solution (1), Figure 1, with a thermal energy supply Q_g in the lower part of the bubble pump. This energy supply is the origin of the evaporation of ammonia, the most volatile compound, and small amounts of

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water. During this evaporation, small bubbles of vapor form and gather together to form large bubbles occupying the entire section of the tube. Having a lower density than the liquid mixture, these big bubbles rise and push the liquid carrying with them a quantity of the liquid solution until the upper part of the generator (2). This happens in a vertical tube of small diameter. The pumped liquid is discharged by gravity into the annular space of the coaxial tube of the generator which is in direct contact with the electrical resistance. This liquid is then heated further, thereby causing the generation of an additional amount of steam.

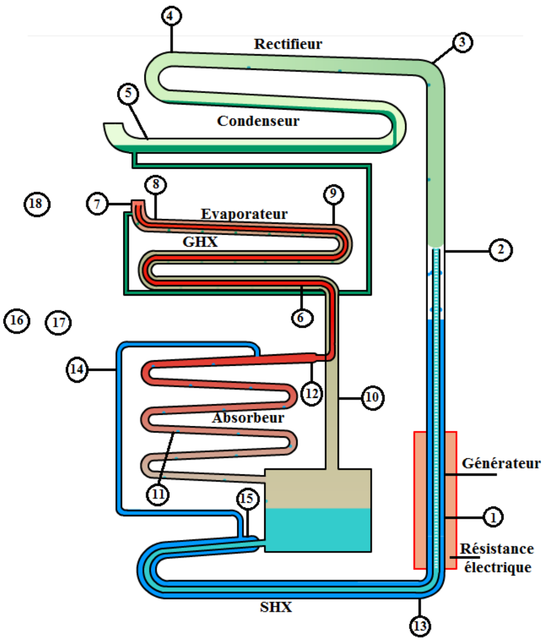

Fig 1 Refrigerant circuit of the refrigerating machine.

The resultant poor solution (13) is gravity-fed to the absorber via a solution exchanger. All of the steam rises to the condenser through the rectifier to condense the water it contains and returns to the annulus of the coaxial tube of the generator to rejoin the ammonia-poor solution to the absorber (14). The ammonia vapor thus purified (4) liquefies in a condenser cooled in ambient air by rejecting a thermal power Qc. The liquid obtained in point (5) travels through the heat-exchanger gas, thermally insulated, where it is cooled before accessing the evaporator in point (7). The gas exchanger and the evaporator consist of two coaxial tubes: the hydrogen-rich gas leaving the absorber flows into the central tube and the binary mixture consisting of the cold inert gas and the refrigerant leaving the evaporator in the annular space. The small diameter tube carrying the liquid refrigerant from the condenser (5) to the evaporator (7) is placed in direct contact with the outer tube of the GHX and the evaporator. The gaseous mixture (12), ammonia / water / hydrogen, rich in hydrogen. By passing through the evaporator, the gaseous mixture (9), ammonia / water / hydrogen, richer in hydrogen is cooled further. The partial pressure of the ammonia / water system in the flow (8) at the inlet of the evaporator is greatly reduced. The evaporation of the liquid ammonia is completed in the GHX and the gaseous mixture (10) rich in ammonia goes to the absorber.

Inside the absorber, the ammonia vapor in the flow (10) is absorbed by the lean solution (14), ammonia / water, returning from the generator. The gaseous mixture thus becomes lighter at point (12) and rises towards the evaporator. The liquid solution becomes richer in ammonia and colder at the outlet of the absorber (15). Then, it goes back to the bubble pump through the solution exchanger. And the cycle starts again.

B. *Measuring devices*

To follow the operation of the refrigerator, we began by equipping it with adequate measuring devices as shown in Figure 2. Eighteen thermocouples type K (Chromel / alumel) were fixed by clamps on the outer surfaces tubes at the desired position and are connected in the other side to a data acquisition system.

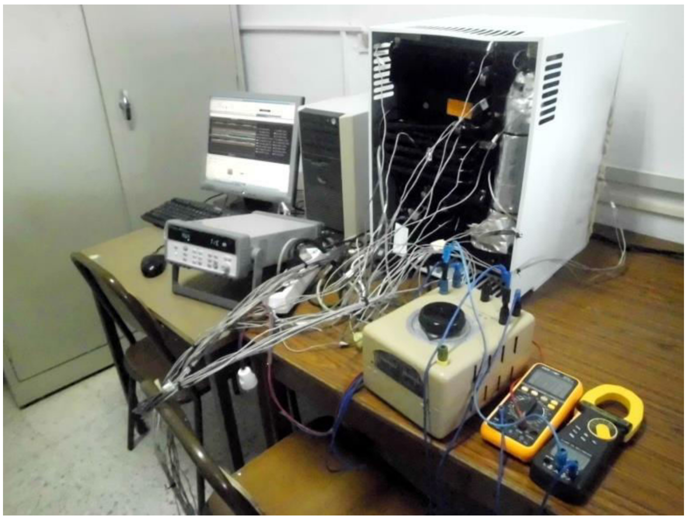

Fig 2 Experimental device and different measurement instruments used.

III. RESULTS

We started this study by identifying the minimum input power required to start the machine and ensure the production of cold. By slightly increasing the input power with each test and by visualizing the evolution of the temperatures of the various points of the machine, it was possible to determine the minimum power necessary to operate the machine. This power is of the order of 22W.

A. *Oscillatory operation of the machine*

This series of tests was started from the minimum starting power of the machine (22W). By slightly amplifying the motive power at each test and by visualizing the evolution of the temperatures of the various points of the machine, it was easily found that all the measured temperatures are remarkably affected by the oscillatory operation of the generator. In FIG. 3 it can be seen that the evolution of the temperatures of the generator and of the evaporator (for the heating powers 22, 24 and 30W) present a stationary

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oscillating regime of the same frequency. The temperature at the generator oscillates between 205 and 210 ° C, that at the evaporator fluctuates between -12 and -20 ° C. These three powers do not allow to have a constant temperature in the evaporator.

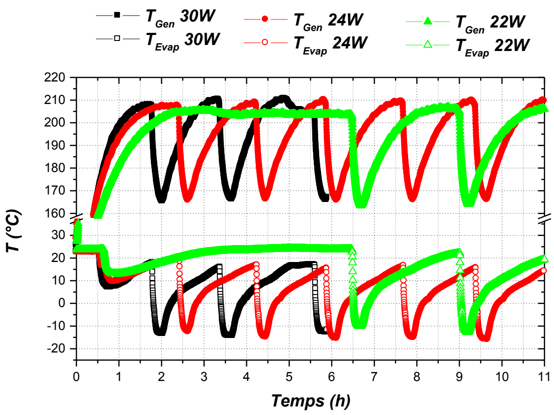

Fig 3 Evolution of generator and evaporator temperatures for heating powers: 22, 24 and 30W.

B. *Stable operation of the machine*

In this part, we continued the series of previous tests by increasing the input power gradually and by visualizing the evolution of the temperatures of the various points of the machine. A total disappearance of all the oscillations is noted in the profiles of the measured temperatures. In particular, the evolution of the temperatures of the generator and the evaporator (for the heating powers 43, 45, 47, 50, 55, 58 and 63W) are shown in FIGS. 4 and 5.

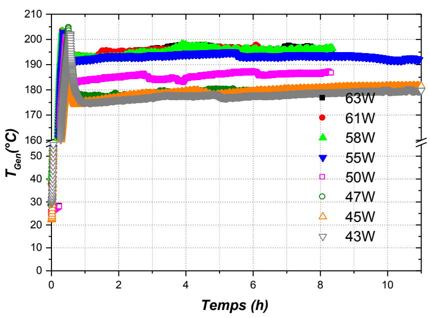

Fig 4 Evolution of generator temperatures for heating powers: 43, 45, 47, 50, 55, 58, 61 and 63W.

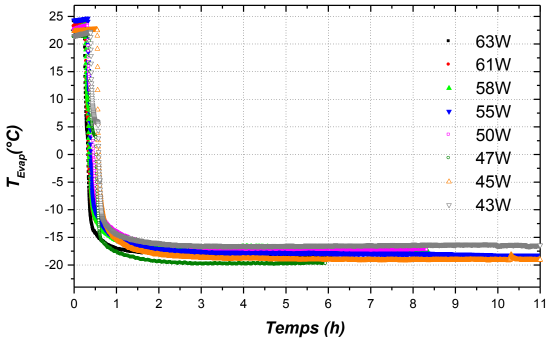

Fig 5 Evolution of evaporator temperatures for heating powers: 43, 45, 47, 50, 55, 58, 61 and 63W.

IV. CHARACTERIZATION OF THE MACHINE

To characterize the thermal exchanges between the machine and the external environment, we have adopted the notion of the overall exchange coefficient, one for the cabin-external environment exchange and the other within the cabin between the contents of the cabin and the evaporator. These exchange coefficients can be deduced from measurements in dynamic or stationary mode.

The energy balance of the cabin is written as:

$$\frac{dU_{cab}}{dt} = \dot{Q}_{ext} - \dot{Q}_{int} \qquad (1)$$

Où U_{cab} is the internal energy of the cabin and respectively the heat flow exchanged between the cabin and the outside environment on one side, and the evaporator and the cabin, on the other:

$$\dot{Q}_{ext} = (UA)_{ext}(T_{amb} - \bar{T}_{cab}) \qquad (2)$$

$$\dot{Q}_{int} = (UA)_{int}(\bar{T}_{cab} - \bar{T}_{evap}) \qquad (3)$$

The energy balance is written then:

$$\frac{dU_{cab}}{dt} = (UA)_{ext}(T_{amb} - \bar{T}_{cab}) - (UA)_{int}(\bar{T}_{cab} - \bar{T}_{evap}) \qquad (4)$$

Using the result of equation (4), we can write:

$$\frac{dU_{cab}}{dt} = C_{p,cab}\frac{dT_{cab}}{dt} = (UA)_{ext}[T_{amb} - T_{cab}] - (UA)_{int}[T_{cab} - \bar{T}_{evap}] - (m\,C_p)_{eau}\frac{dT_{eau}}{dt} \qquad ($$

Where (C_p) is the average mass heat of the cabin, including its contents..

By dividing the two members by $C_{p,cab}$, the average heat mass of the empty cabin, we obtain:

$$\frac{dT_{cab}}{dt} = \frac{(UA)_{ext}}{C_{p,cab}}[T_{amb} - T_{cab}] - \frac{(UA)_{int}}{C_{p,cab}}[T_{cab} - \bar{T}_{evap}] - \frac{(m\,C_p)_{eau}}{C_{p,cab}}\frac{dT_{eau}}{dt} \qquad (6$$

If we put $\alpha = \frac{(UA)_{ext}}{C_{p,cab}}$; $\beta = \frac{(UA)_{int}}{C_{p,cab}}$; $\gamma = \frac{(m\,c_p)_{eau}}{C_{p,cab}}$

$$\frac{dT_{cab}}{dt} = \alpha\,[T_{amb} - T_{cab}] - \beta\,[T_{cab} - \bar{T}_{evap}] - \gamma\,\frac{dT_{eau}}{dt} \qquad (7)$$

In the particular case of the empty cabin and therefore $m_{eau} = 0$. Equation (7) becomes :

$$\frac{dT_{cab}}{dt} = \alpha[T_{amb} - T_{cab}] - \beta[T_{cab} - \bar{T}_{evap}] \qquad (8)$$

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Based on the temperature dynamic measurements T_{amb}, T_{eau}, T_{cab} et $\overline{T}_{evap}$, equation (9) could be solved for 20 experiments (empty cabin or with different volumes of water in the cabin). A nonlinear regression procedure is performed using the MATHEMATICA ® software to determine the model parameters α, β and γ and to predict the temperature of the water and the cabin.

The model of the water temperature is of the form: : $T(t) = T_0\,(1 + a\,t\,e^{-b\,t} + c\,e^{-d\,t} + + e\,e^{-f\,t})$

Where a, b, c, d, e and f are smoothing parameters identified for each temperature (water and cabin) and for each experiment (cabin empty or not).

The experimental values recorded for the cabin temperature as well as the regression model used are shown in Figure 6 for an empty cabin. We note that the model of regeneration used faithfully represents the experimental data.

Table 1 also shows the found values of the parameters α, β and γ as well as their average values for the experiments carried out.

The knowledge of the mean value of γ allows us to determine $C_{p,cab}$. Once this last parameter has been identified, we can directly deducet $(UA)_{ext}$ et $(UA)_{int}$ respectively from the average values of α and β.

$C_{p,cab} = 1.91$ kJ/K

$(UA)_{ext} = 0.53$ W/K

$(UA)_{int} = 0.23$ W/K

1.2. Machine performance study

The performance of the refrigerator studied is shown in Figures 7 and 8. As previously described, it is noted that a minimum power required for the start of the refrigerating machine is in the order of 36W producing a cooling capacity of 5W.

As the heat supplied to the generator is progressively increased, the flow rates of the pumped solution and the steam increase too. Increasing the flow rate of the vapor and the refrigerant flow allows to have an increasingly large cooling capacity.

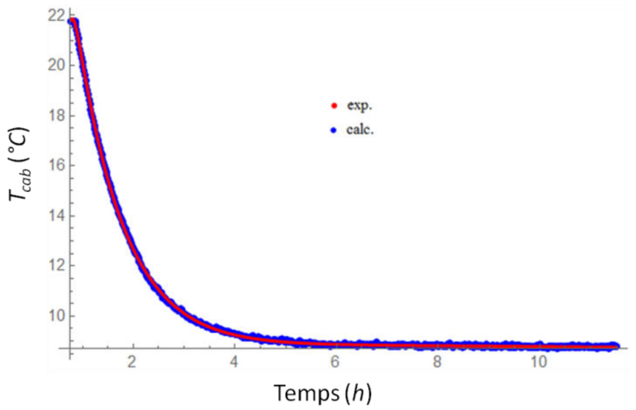

Fig 6 Comparison of the cabin temperature profile (Veau = 0l) with the regression model found : T0= 8.40592, a=0.0192634, b= 0.145793, c= -12968, d= 14.3739, e= 1.06511 et f = 1.06511

TABLE I
Values of parameters α, β and γ

n°	α (h^{-1})	β (h^{-1})	γ (kg^{-1})
1	0,954881	0,4022393	0,7888909
2	0,916906	0,3917	0,7888909
3	0,916906	0,3917	0,981186
4	0,903291	0,3847	0,981186
5	1,11416	0,496746	0,955986
6	1,22393	0,568542	0,955986
7	1,22393	0,568542	0,955986
8	1,05858	0,460507	
9	1,05858	0,460507	
10	0,945599	0,442812	
11	0,9099	0,402664	
12	0,945599	0,442812	
13	0,9099	0,402664	
14	1,06664	0,486158	
15	1,06664	0,486158	
16	0,916664	0,355459	
17	0,886484	0,399174	
18	0,983791	0,44676	
19	0,983791	0,44676	
20	0,902639	0,349796	
21	1,0579	0,410276	
22	1,0579	0,410276	
23	1,0579	0,410276	
The mean	**1,0027179**	**0,4355317**	**0,9154445**

A larger flow rate of the pumped solution allows to absorb a larger amount of steam. This explains the growth of the cooling capacity and consequently the coefficient of performance. It is also recalled that the increase of the motor temperature leads to a pumped solution poorer in ammonia. A pumped solution, at the inlet of the absorber, increasingly poor with a flow more and more increasing leads to an efficiency of the absorber more and more increasing and a partial pressure of the refrigerant lower.

Beyond 43W, the cooling capacity stabilizes and the COP begins to decrease. This is explained by the fact that the heat exchange at the evaporator is limited by the temperature along the evaporator and also by the overall internal exchange coefficient $UA)_{int}$ between the evaporator and the inside of the refrigerator.

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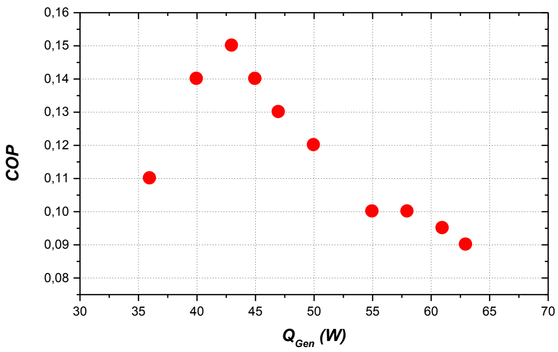

Fig 7 Variation of the experimental COP according to the power supplied to the generator.

The amount of ammonia that can evaporate inside the evaporator is then limited. The additional amount produced by the increase in power will not be evaporated and will leave the evaporator in the liquid state. The cooling capacity produced becomes constant since the amount of ammonia evaporated remains the same. Also, an additional heating at the generator is not necessary beyond these conditions because it will only further heat the ammonia vapor which will subsequently be cooled in the rectifier and then in the condenser, which leads only a loss of energy and a decrease in the COP.

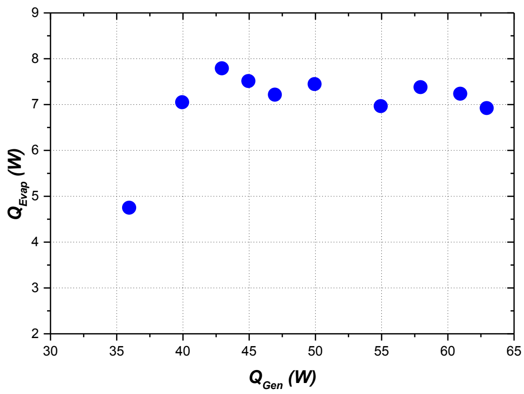

Fig 8 Variation of the cooling capacity according to the power supplied to the generator.

V. Conclusions

In this paper, we conducted an experimental study of a low-power commercial refrigerator with a capacity of 25 liters, powered by an electrical power. We have studied the performance of the machine for various operating conditions which allowed us to completely characterize this machine. In dynamic mode, the values of $(UA)_{cab}$ and $(UA)_{evap}$ found are respectively 0.636 WK^{-1} and 0.276 WK^{-1}. These values allowed us to evaluate the cooling capacity at 7.5W and the COP of the machine in the order of 0.15.

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