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I. CARVAJAL, G. POLUPAN, F. SANCHEZ,
P. QUINTO, A. ZACARIAS

Thermal Hydraulics Applied Engineering Laboratory, National Polytechnic Institute of Mexico

HEAT TRANSFER STUDY IN THE ELEMENTS OF AN ABSORPTION REFRIGERATION SYSTEM OPERATING IN PARTIAL HEAT LOADS

В роботі дано опис експериментального обладнання і методики, що використовувалися для отримання даних на абсорційній холодильній системі при різних теплових навантаженнях. Холодильні характеристики були проаналізовані при номінальному тепловому навантаженні (100%), а також при двох знижених навантаженнях (60% і 80%) і двох підвищенних теплових навантаженнях (120% и 140%). Холодильна видатність абсорбційної системи при тепловому навантаженні 120% підвищувалась на 30% по відношенню до номінального навантаження і в подальшому понижувалася більш ніж на 50% при підвищенному тепловому навантаженні в 140%. Найбільш низьке значення коефіцієнта використання було отримане для теплового навантаження 140% і склало тільки 0,04. Для інших досліджених теплових навантажень цей коефіцієнт знаходився в межах від 0,15 до 0,22 при його номінальному значенні 0,19.

В исследовании, представленном в статье, описана экспериментальная установка и методика, использованные для получения экспериментальных данных на абсорбционной охладительной системе при разных тепловых нагрузках. Холодильные характеристики были проанализированы при номинальной тепловой нагрузке (100%), а также двух пониженных нагрузках (60% и 80%) и двух повышенных тепловых нагрузках (120% и 140%). Холодильная производительность абсорбционной установки при тепловой нагрузке в 120% возрастала на 30% по отношению к номинальной нагрузке и далее снижалась более чем на 50% при повышенной тепловой нагрузке в 140%. Наиболее низкое значение коэффициента использования было получено для тепловой нагрузки 140% и составило только 0,04. Для других исследованных тепловых нагрузок этот коэффициент находился в пределах от 0,15 до 0,22 при его номинальной величине 0,19.

The study presented in this paper describes the experimental rig and the methodology used to carry out the experimental research for an absorption refrigeration system operating in partial heat loads. The refrigeration performance, was analysed using a nominal thermal load (100%) and two partial loads of 60% and 80% and two overloads of 120% and 140%. The refrigeration capacity respect the nominal, was increased up to a 30 % for the overload of 120 % and decreased more than 50 % for the partial load of 60 % and overload of 140%. The lower value of the Coefficient of Performance was obtained for the overload of 140% being of only 0,04. For the other partial loads and overloads it was located in the range of 0,15 to 0,22 being its nominal value of 0,19.

A – Surface area;
b – Fin pitch;
C_p – Water heat capacity;
d – Tube diameter;
g – gravity force;
h – Convective coefficient;
k – Thermal conductivity;
L – Fin length;
$\dot{m}_w$ – Water mass flow rate;
P_E – Electrical power;
Q – Heat flow;
R_1, R_2 – Electrical resistances;
T – Temperature;
V_E – Voltage.
α – Thermal diffusivity;
β – Volumetric thermal-expansion coefficient;
ν – Cinematic viscosity;

1. Introduction

Due to the energy conservation and the operating cost reduction, the thermal energy recovery from exhaust gases and combustion products is getting common in several industries. One of the alternatives for this purpose is the so called absorption refrigeration systems. This is because the temperature requirements for that cycle falls into the low-temperature (lower than 230 °C) waste heat recovery range [1], where a significant potential for electrical energy saving exists.

The absorption refrigeration systems are used to rise temperatures up to 9 °C for bromide lithium machines and −5 °C for ammonia machines [2].

Both types of systems can be used in industrial applications.

The absorption refrigeration system operation can be found in the technical literature, such as the ASHRAE Transactions [3]. The main advantages of using absorption refrigeration systems for waste heat recovery can be attributed to [4]:

- Their quiet operation;
- Their ability to produce cooling by using hot gases, process steam, process liquids/solids, and exhaust air;
- Their little service or maintenance requirement;
- Their simplicity;
- Their reliability.

To apply the absorption refrigeration systems for waste heat recovery, it is necessary to know their thermal performance when the systems operate with partial heat loads. These kind of situation occur due to the variation of the hot gases flow characteristics (from incinerators, furnaces, and so on), during the thermal energy supply to the boiler.

In this study the thermal performance of the absorption refrigeration system with partial heat loads to the boiler was carried out experimentally and the results were compared with the nominal heat load of the system.

2. Experimental rig

The experimental rig used for the absorption refrigeration process was an absorption refrigeration system which was taken from a small commercial refrigerator with 90 dm^3 of internal volume and a freezing compartment of 5 dm^3.

Figure 1 shows the schematic diagram of the absorption system. The unit charge consist of a 0.40 kg of ammonia, water and hydrogen. These are at a sufficient pressure to condense ammonia at room temperature. From the Fig. 1 it can be observed that in this type of absorption refrigeration systems there are no moving parts.

Figure 2 shows a photograph of the main components of the experimental rig. These components include the boiler (cover with insulating material), the condenser, the evaporator and the absorber.

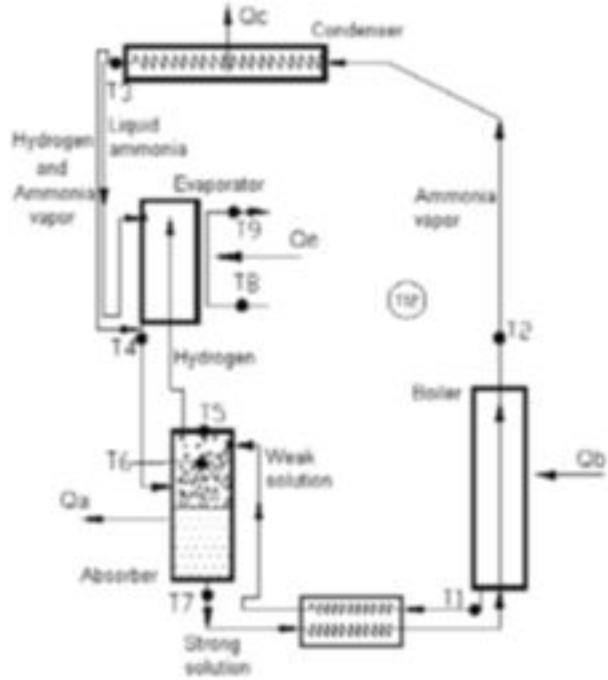

Fig. 1. Schematic diagram of the absorption refrigeration system.

The evaporator is supplied with hydrogen. The hydrogen passes across the surface of ammonia. It lowers the ammonia vapor pressure enough to allow the liquid ammonia to evaporate. The hydrogen circulates continuously between the absorber and the evaporator.

The thermal energy supplied to the boiler was provided from two electrical resistances of 95 .each one, connected in parallel. To supply the electrical current a potentiometer was used. It operates at 127 V with a regulation range of 0-100%.

The experimental system was provided with eight thermocouples, type T (± 0.1°C accuracy), along the system as is illustrated by dots in Fig. 1. The thermocouples were connected to a data acquisition system, using the commercial software Scan Link® 2.0, to record the measured temperatures.

The temperatures along the system were recorded every 5 minutes for an interval of four hours. Three experimental tests were carried out for each load: one nominal, two partial loads and two overloads.

In order to supply the refrigeration process load a cylindrical container was installed outside the evaporator, through that container, water was circulated, transmitting heat to the liquid ammonia, which circulated inside along the evaporator.

Fig. 2. Photograph of the experimental rig.

The water flow rate was measured using a rotameter, which was calibrated for a range of 0 to 1 l/min flow rate. The water flow rates were adjusted by valves at the inlet and the outlet of the cylindrical container. A pump was used to provide sufficient pressure to circulate the water through the cylindrical container. The pump used had a 373 W motor and had a maximum capacity of 11.35 l/min. The volumetric flow rate was maintained the same for each test: 0.25 l/min.

In order to reduce the heat loss from the boiler, fiberglass of thickness of 25.4 mm and an aluminum coating was used to insulate the piping. To avoid the evaporator gain heat from the surroundings, the evaporator piping was insulated with polyurethane.

Based on the Joule's Law and the electrical resistances values, the voltage and its correspondent current for the nominal and partial loads were determined. The Joule's Law is expressed by:

$$V_E = \sqrt{P_E \cdot \frac{1}{\frac{1}{R_1} + \frac{1}{R_2}}} \,. \tag{1}$$

The input data of the heat load of boiler was different for the nominal and each four partial loads (see Table 1).

The measurement errors of the different parameters was estimated to be approximately of 8%. The standard deviation of the experimental data was ± 6%.

Table 1

Heat load of boiler	[W]
60 % of nominal load	75
80 % of nominal load	100
nominal load (100%)	125
120 % of nominal load	150
140 % of nominal load	175

3. Energy balance of the system components

A computational program was developed to calculate the parameters obtained from the energy balance for each component of the system using the experimental data as input data. The equations to calculate the energy balance for each component are:

Boiler

To calculate the heat flow supply to the boiler, the power of the electrical resistances was considered by using the expression [5]:

$$Q_b = hA\left(\overline{T_b} - T_\infty\right), \tag{2}$$

where: $\overline{T_b}$, is the boiler average temperature and T_{out},bis the surrounding temperature; h, is the natural-convection coefficient of a vertical tube (the boiler) [5].

Evaporator

To determine the heat flow in the evaporator the following equation was used [5]:

$$Q_e = \dot{m}_w C_p \left(T_{in} - T_{out}\right) , \tag{3}$$

where: T_{in}, is the water inlet temperature and Tout, is the water outlet temperature of cylindrical container.

Condenser

The condenser has a square-finned tube and a plain tube (see Fig. 2). To calculate the heat flow in the condenser, the total heat rejected in the finned tube and in the plain tube were considered as follows:

$$Q_c = Q_{plaintube} + Q_{finnedtube} \,. \tag{4}$$

For these calculations, the square-finned tube and the plain tube, were analysed separately using the corresponding expressions from [6]:

- square-finned tube

$$Nu_{ft} = \left[\left(\frac{Ra_{ft}^{0.89}}{18} \right)^m + \left(0.62 \;\; Ra_{ft}^{1/4} \right)^m \right]^{1/m} ,$$

where m = 2.7 for air

$$Ra_{ft} = \frac{g\beta(T_w - T_\infty)b^3}{\nu\alpha}\left(\frac{b}{L}\right)$$

$$h_{ft} = \frac{Nu_{ft}k}{b} = \left(\frac{q''_{ft}b}{(T_w - T_\infty)k}\right)$$

$$Q_{finnedtube} = h_{ft} A_{ft} \Delta T$$

- plain tube

$$Nu_{pt} = \left\{ 0.60 + \frac{0.387 \;\; Ra_{pt}^{1/6}}{\left[1 + \left(\frac{0.559}{Pr} \right)^{9/16} \right]^{8/27}} \right\}^2$$

$$Ra_{pt} = \frac{g\beta(T_w - T_\infty)d^3}{\nu\alpha}$$

$$Pr = \frac{\nu}{\alpha}$$

$$h_{pt} = \frac{Nu_{pt}k}{b} = \frac{q''_{pt}d}{(T_w - T_\infty)k}$$

$$Q_{plaintube} = h_{pt} A_{pt} \Delta T$$

T_w – is the wall temperature.

Absorber

The heat flow at the absorber was calculated by a energy balance through the all system using the following expression:

$$Q_{in} = Q_{out} + Q_{loss}, \tag{5}$$

where: Q_{in}, is the inlet system heat flow, Q_{out}, is the outlet system heat flow and Q_{loss}, are the system heat losses to the surroundings.

Therefore, we can write a heat balance,

$$Q_g + Q_e = Q_a + Q_c + Q_{loss} \tag{6}$$

and

$$Q_a = Q_g + Q_e - Q_c - Q_{loss}. \tag{7}$$

The system heat losses to the surroundings Qloss was estimated in [3], to be between 5 and 10%. In this research the estimation of the heat losses were approximately about 8%.

4. Results and discussion

The experimental heat flows for each component, were plotted as a function of the time. The average values of the partial loads investigated were also plotted.

Figures 3, 4, 5, 6 and 7 show the performance of the refrigeration system with five different thermalloads supplied to the boiler that is, 60%, 80%,100%, 120% and 140%. It was observed that for all tests, the refrigeration process reached the steady performance approximately after two hours of continuous operation.

Figure 3 illustrates the performance of eachrefrigeration system component with nominal heat load of 100% (by design or specification) and it can be observed that for the first 15 minutes of operation the heat supplied to the boiler was approximately similar to the heat dissipated by the absorber. At the same time, the amount of heat dissipated by the water in the evaporator is cero. The same occur in the condenser, where the heatdissipated is zero.

The results confirm that the separation and absorption processes of the refrigerant are notefficient at the beginning of the performance of the system.

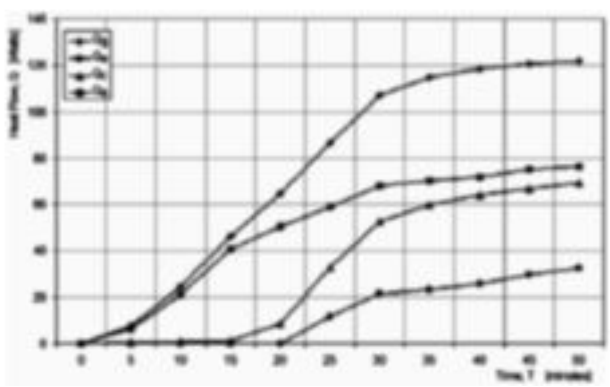

Fig. 3. Heat flow gained/dissipated for the components of the system as a function of the time, with a nominal load (100%).

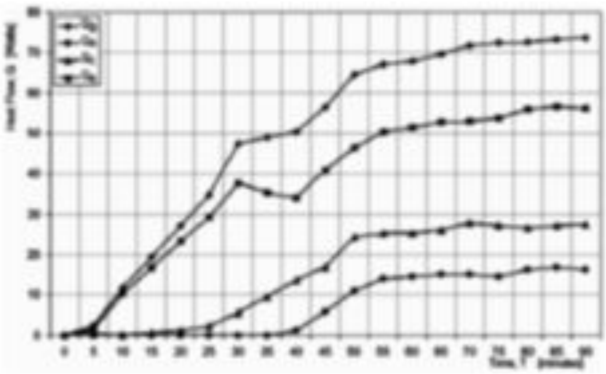

Fig. 4. Heat flow gained/dissipated for the components of the system as a function of the time, with partial load of 60%.

This is, because the heat flow supplied to the boiler is not sufficient. For the same reason, the refrigerant flow is not sufficient to cool the refrigeration load.

After 20 minutes of the system operation it is starting to carry out the refrigeration process.

Figure 4 shows the system operation with a thermal partial load of 60% supplied to the boiler. It can be seen that after 40 minutes of the system operation, the refrigeration process starts.

In the evaporator, the refrigeration rate was about 50% lower in comparison with nominal load test.

In the interval of 30 to 50 minutes, there was instability of the system operation. This was because the refrigerant flow was very low. Therefore, the absorption process was very slow also. As result, the mixed flow flowing to the boiler is also less.

The instability of the process occurred because at the same interval of time, the evaporation process required a greater amount of refrigerant, and for this reason, the mass flow rate passing through the absorber and going toward the boiler was reduced.

Figure 5 shows the 80 % thermal partial load to the boiler. It is possible to observe that the cooling effect is evident 25 minutes after the starting, and the cooling capacity of the evaporator increases in function of the exhausted heat in the condenser. Additionally it is possible to see that the heat dissipated in the absorber was reduced instead of increased, is in spite of the increase of the heat flow supplied to the boiler. All this indicate that the equipment tends to have a nominal behavior where the heat flows in the condenser and in the absorber as well should have almost the same value and at the same average temperature.

Figure 6 shows the system performance with a greater load than the nominal load of 120%, supplied to the boiler.

From figure 6 it can be observed that after 15 minutes of system operation, the refrigeration process starts. It is possible to remark that the refrigeration rate of the evaporator was approximately of 30% higher than the one for the nominal load.

In view that the boiler was supplied heat, the absorber and the condenser dissipated a great amount of heat. In this case there were any instabilities before the evaporator starts cooling the water. Additionally the heat flow in the absorber started to decrease drastically until the case where it was lower than in the condenser. After this radical change, all heat flows tended to be stable.

The 140 % thermal overload to the boiler case is shown in figure 7. The refrigeration process started 15

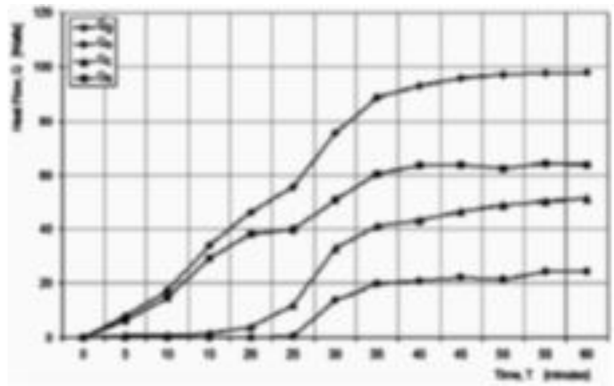

Fig. 5. Heat flow gained/dissipated for the components of the system as a function of the time, with partial load of 80%.

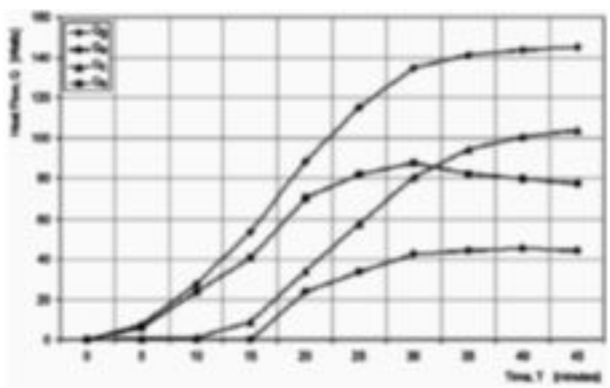

Fig. 6. Heat flow gained/dissipated for the components of the system as a function of the time, with a partial load of 120%.

minutes after the system was turned in. Additionally, the heat flow in the condenser was much higher than in the absorber and the heat flow in the evaporator stopped increasing as in the previous cases, on the contrary it was much lower that in the 60 % load test.

The excess of heat supplied to the boiler produced, as a result, that a great quantity of vapor ammonia were delivered directly to the condenser and this can not transform all vapor into liquid, because is limited by the convective coefficient *h* and the surface area *A*. The liquid transformed part was sent to the evaporator where it performed the environment cooling in a very poor manner. The other quantity of ammonia which was not able to condense was directed to the absorber, where due to the poor absorption by the weak solution for its high temperature, a stagnant zone of the gas was created, that is, the working fluid circulation was restrained in the system. As a result, is not recommended to operate the system with an excess of heat supplied to the boiler of 40 %.

After the analysis of different loads supplied to the boiler, we can conclude that it is very important to know the heat transmission to each element of the system.

The boiler is the element which receives the heat supply and for this reason its operation has a significant influence in the whole system performance.

In figure 8, is shown that the heat transfer behaviour in the boiler for different partial loads is similar. That is due to the use of electrical resistance for the heat supply, the voltage was kept constant for each test, then the variation only occurred while the heating was transient because in steady state the variation was very insignificant during the experimentation.

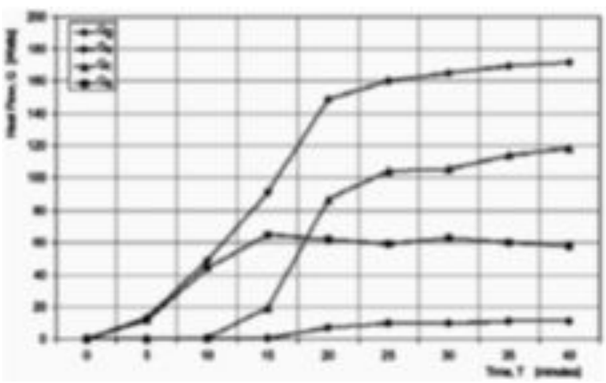

Fig. 7. Heat flow gained/dissipated for the components of the system as function of the time, with part load of 140%.

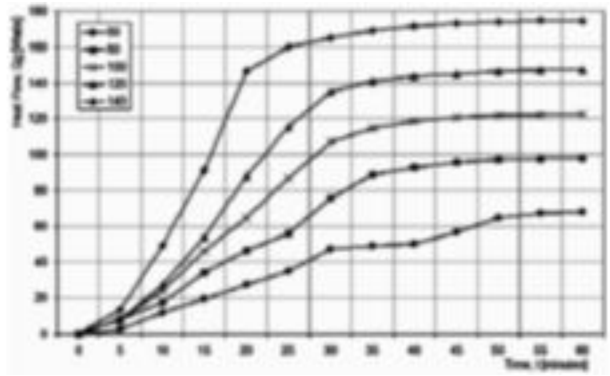

Fig. 8. Heat flow supply to boiler as function of the time, for different part loads.

The condenser has a very important role in the correct operation of the evaporator and in the absorption system in general. Figure 9 shows how the heat rejected to the surroundings was increasing according to the heat flow supplied to the boiler. For this reason, it is clear the important difference we can see in the heat flow figures at overload of 120 % and in the nominal load (100 %). It is more notorious if this difference is compared among the overloads of 120 % and 140 %.

The evaporator is the element determining the cooling system capacity, so that it is very important to analyze the results obtained for this element.

In figure 10, the heat transfer behaviour in the evaporator is shown. We can remark that the heat flow in the evaporator increases since the very moment the refrigeration process starts until the moment when tends to be constant. For the overload of 120 % the evaporator has a refrigeration capacity higher that 30 % than for the nominal load. While for partial load of 60 % and overload of 140 % the refrigeration capacity

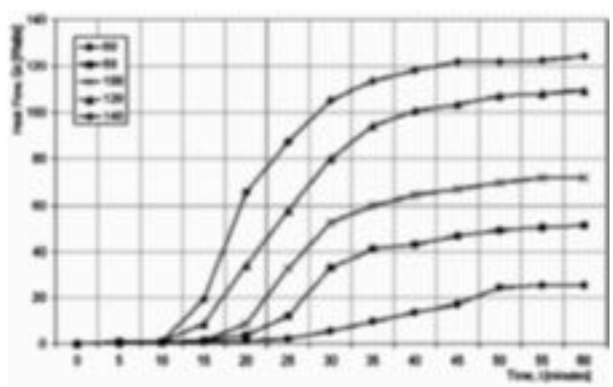

Fig. 9. Heat flow dissipated by the condenser as function of the time, for different part loads.

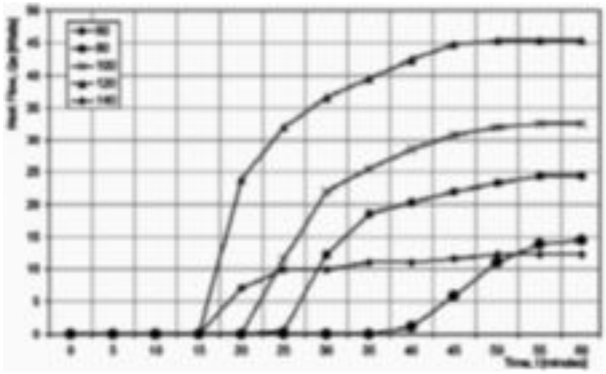

Fig. 10. Evaporator refrigeration capacity as function of the time, for different part loads.

diminishes more than the 50 %, so that it is not recommended to operate the absorption refrigeration system in this two loads.

The behaviour of the absorber has much to do with the efficient stabilization and operation of the system. In figure 11 the heat transfer behaviour in the absorber is shown, which resulted to be the most complex of all here showed.

During the partial loads of 60 %, 80 % and the nominal, the heat dissipation in the absorber has a behaviour practically stable which corresponds with the heat flow supplied to the boiler. On the contrary for the partial load of 120 %, the maximum in heat dissipation is attained 30 minutes after, following a stabilization in a value near to the one obtained for the nominal load.

For the overload of 140 % this maximum is presented 20 minutes after the beginning of the process. However, differently to the anterior case, the heat dissipation decreases drastically until values inferior to the ones presented for the nominal loads and 80 % and is lightly superior to the one presented with the partial load of 60%.

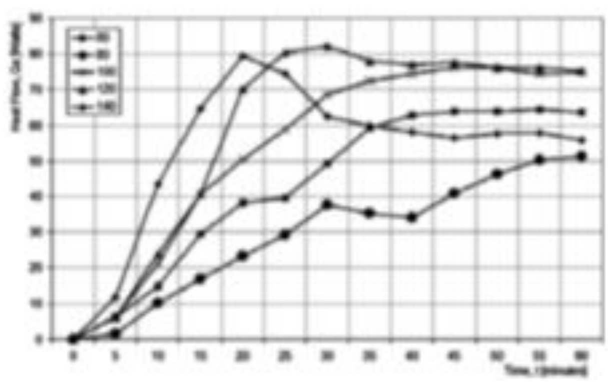

Fig. 11. Heat flow dissipated by the absorber as function of the time, for different part loads.

After the analysis of the refrigeration system behavior operating in partial loads and the heat transfer in its different elements, it is possible to remark that the system operates steadily and with satisfactory values in the refrigeration process in the loads range of 80% to 120 %, being recommended that the system operates the most of the time, up of its nominal load.

To validate completely this information it is necessary to calculate the coefficient of performance.

5. Coefficient of performance

The figure of merit adopted in cooling engineering is the useful effect divided by the input power, defined as the Coefficient of Performance, or COP for short.

For the chiller mode of operation (cooling or refrigeration at the evaporator) of the absorption machine, the COP is [7]:

$$COP = \frac{\textit{cooling rate}}{\textit{thermal power input}} = \frac{\textit{heat transfer at the evaporator}}{\textit{heat input at the generator}}$$

The COP of absorption systems is ostensibly far lower than that of the corresponding mechanical devices. The key advantage of absorption technology lies in the direct utilization of locally-available thermal sources.

In the figure 12, the curves of the coefficient of performance computed for two partial loads, two overloads and the nominal are shown. In this figure it is possible to observe that the COP for the partial load from 60 % to overload of 120 % are in the range of 0.15 to 0.22, being its nominal value of 0.19. It is important to take into account also the beginning of the refrigeration process since for the 120 % overload was only of 15 minutes while for the 60 % partial load was of 35 minutes.

On the other hand, for the overload of 140 % the obtained values are much lower of the ones obtained for the nominal load, in the order of 0.04. All this yields to affirm that it is possible to operate the absorption refrigeration system, studying the range of partial loads

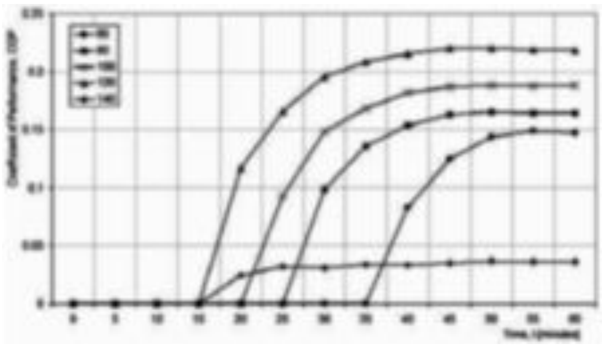

Fig. 12. Coefficient of performance COP of the absorption refrigeration system as a function of the time, for different partial loads.

from 60 % to 120%, being preferable that the most of the time be working over the nominal load which is where the higher COP values were obtained. However the limit of this overload must be considered of 120 % because as we have seen previously instead of having an increment of the COP this decreases drastically.

Conclusions

From the experimental data obtained we can deduce the following conclusions:

For all the overloads and partial loads including the nominal, the system attains its steady operation approximately in two hours after the beginning of the thermal energy supply to the boiler. The refrigeration process starts since the 15 to the 35 minutes, for the loads of 120% to 60% respectively being the nominal value of 20 minutes.

The refrigeration capacity respect the nominal, was increased up to a 30 % for the overload of 120 % and decreased more than 50 % for the partial load of 60 % and overload of 140%.

Some instabilities were detected in the operation of the absorber for all the partial loads and overloads, being the more notorious the ones observed for the overloads of 120 % and 140 %.

The lower value of the Coefficient of Operation was obtained for the overload of 140% being of only 0.04. For the other partial loads and overloads it was located in the range of 0.15 to 0.22 being its nominal value of 0.19.

For all the above exposed, we can affirm that it is possible to operate the absorption refrigeration system studied in the partial loads range of 60 % to overload of 120 % being preferable that the most of the time works above the nominal load where the largest values of COP were obtained. However the limit of this overload must be fixed in 120 %, because if we apply a bigger load as we have seen, instead of having an increment of COP it decreases drastically.

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