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An optimized configuration for the triple-effect absorption cycle

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Abstract

The triple-effect absorption cycles were developed since the end of the eighties for residential (air-conditioning) applications. Among them only one particular architecture is able to use ammonia-water ($\text{NH}_3/\text{H}_2\text{O}$) as mixture. It consists of two single-effect loops, one of them (the lower loop) using for driving its generator the heat rejected from three heat exchangers of the upper loop : condenser, absorber and rectifier. The analysis performed in previous studies showed that a value of 1.5 for the COP may be reached.

The paper presents some new ideas trying to improve these performances. They are concerning both the design of the cycle and the operating parameters of the two loops. The model assumptions are presented and then the results of the calculations are shown in COP's variation diagrams. Finally, a discussion reveals the improvements which were obtained and their consequences for the equipment design.

1. INTRODUCTION

The interest for the use of the absorption cycles as a cooling technology has considerably increased in the last years. This resurgence is caused especially by the requirement of finding new non-ozone-depleting substances to substitute CFCs. This study is focused on the air-conditioning applications, therefore it will refer to the cycles extracting the heat at a level of a few degrees Celsius above zero. Up to now, the triple-effect systems proved to deal the best with the two essential demands: low energy consumption and feasibility. There are many possibilities to configure a cycle of this type, which may be classified in two categories. The first of them includes the variants coming from the extension of the double-effect systems (by adding an identical loop) and the second one refers to the proper triple-effect schemes. Unfortunately, most of these architecture parts do not fit with the thermodynamic properties of the classical mixtures. Only one cycle from the first category has been conceived for the $\text{H}_2\text{O} - \text{LiBr}$ pair (/2/) and another one, belonging to the second group, was developed for the $\text{NH}_3 - \text{H}_2\text{O}$ mixture (/3/, /4/). The last of them leads to a better COP, it also has fewer heat exchangers, so it was chosen to be presented in this paper.

The calculation model used allows to determine the thermodynamic parameters in all the points of the cycle and its energy performances. It represents the theoretical base of a computer program by which an optimization analysis was carried out.

The simplest configuration characterizes a classical $\text{NH}_3 - \text{H}_2\text{O}$ absorption system. It contains the four heat exchangers: the generator G, the condenser C, the evaporator E, the absorber A and the solution pump P. The signification of the temperatures presented in the scheme are:

- T_C - the temperature of the refrigerant leaving the condenser;
- T_E - the bubble temperature of the refrigerant in the evaporator;
- T_A - the temperature of the strong solution at the absorber outlet point;
- T_G - the temperature of the weak solution leaving the generator.

It is assumed that the heat flux received at the generator equals 1 kW. The values of the other heat flux depend specially on the four above mentioned temperatures. In the case of an air-conditioning system producing at the evaporator a flow of water of about 7 °C and using as a cooling agent water at 32 °C, the following values may be estimated: $T_E = 4$ °C; $T_C = T_A = 35$ °C. A 3 °C pinch was considered for each heat transfer process which is reasonable from the viewpoint of the heat exchanger design calculation. Therefore it remains only one degree of liberty, the temperature T_G , which can modify the other heat flux and the COP. The calculations show that the highest value for the COP is reached when T_G is situated between 85 and 90 °C. For the assumed thermodynamic parameters, the next values will be found for the heat fluxes:

$$q_A \approx q_G \approx 1 \quad (1)$$

$$q_C \approx q_E \approx 0,7 \quad (2)$$

If h_p is neglected, (it never exceeds 0,02) the COP for the single stage cycle may be determined as:

$$\text{COP} = \frac{q_E}{q_G} \approx \frac{0,7}{1} = 0,7 \quad (3)$$

It is reminded that the aim of this approach is only to give an idea about the approximate size of the presented parameters and not to perform an exact calculation.

Let us consider now two single effect cycles supplying water at 7 °C as shown in the figure 1. They are not identical from the viewpoint of their operating parameters and, in order to distinguish them, the one situated inside will be called the lower loop and the other, the upper loop. The first of them operates at the same temperature values as the installation described in the above paragraph ($T_{E0} = 4$ °C, $T_{C0} = T_{A0} = 35$ °C and $T_{G0} = 86$ °C). Its parameters will be always designated by the '0' index. The thermodynamic levels characterizing the heat exchangers of the upper loop are different ($T_C \approx T_A \approx 90$ °C, $T_G = 210$ °C), excepting the evaporator ($T_E = 4$ °C). These values make possible the transfer of the heat rejected at the upper loop condenser and absorber to the lower loop generator. It will be assumed, now that a heat flux $q_G = 1$ kW is received by the upper loop generator. Since the operating temperatures differ from the values of the lower loop the other heat fluxes are:

$$q_A \approx q_G = 1 \quad (4)$$

$$q_C \approx q_E \approx 0,4 \quad (5)$$

Therefore, the heat removed to the lower loop generator is:

$$q_{G0} = q_C + q_A \approx 1 + 0,4 = 1,4 \quad (6)$$

If the equations (1) and (2) are used, the other heat quantities will be found:

$$q_{A0} \approx q_{G0} = 1,4 \quad (7)$$

$$q_{C0} \approx q_{E0} \approx 0,7 * 1,4 \approx 1 \quad (8)$$

The total system COP may be calculated as:

$$COP = \frac{q_E + q_{E0}}{q_G} \approx \frac{0,4 + 1}{1} = 1,4 \quad (9)$$

Its value was increased with about 100 %, in comparison with the single-effect installation. The cost of this improvement is a more complicated architecture (the number of heat exchangers doubles) and a higher temperature level of the hot source (210 °C compared to 86 °C). The upper loop realizes a lower COP than the global system:

$$COP_H = \frac{q_E}{q_G} \approx \frac{0,4}{1} = 0,4 \quad (10)$$

Only the basic principles of the triple-effect cycle were described here above because, as it will be seen later, the scheme and the calculation model of a real installation are more complex. However, the remaining conclusion is that the new architecture is much more efficient than the single-effect installation.

2. THE TRIPLE-EFFECT-CYCLE OPERATING MODE

First, an analysis of the already known cycle configuration (/4/) will be carried out, in order to determine its performance limits and the possible improvements. It consists of two similar loops that have an identical architecture, as shown on figure 2. Between the two loops there is an intermediate heat transfer fluid (pressurized water) transporting the heat from the condenser, absorber and reflux heat exchanger of the upper loop to the lower loop generator.

The refrigerant vapor, which has a high ammonia concentration, enters the condenser C at the point 8. In the lower loop the resulting heat is rejected to the cooling agent while in the upper loop it is removed to the intermediate agent.

Then, the liquid refrigerant flows through the expansion valve VR, reducing its pressure and temperature. It enters the evaporator E at the point 10, as a two-phase mixture close to the saturated liquid state. Here, the refrigerant absorbs heat from the water flow circulating in the air-conditioning installation. It is incompletely evaporated and leaves the apparatus as a two-phase mixture close to the saturated vapor state (point 1). The boiling process is carried out in the pre-cooler by using the heat recovered from the liquid refrigerant.

Next, the saturated vapor flow 11 enters the absorber A at the liquid mixture outlet point and begins to be absorbed into the solution. The absorption process ends in the point 3 where the weak absorbent is introduced. In the

first part, the resulting heat is rejected to the cooling agent (for the lower loop) or to the intermediate agent exiting the condenser (for the upper loop).

At the absorber outlet the strong solution (2) exits and then passes through the solution pump where its pressure rises. The point 4 designates the inlet of the strong refrigerant mixture in the exchanger EA, where it will be heated to the point 15. From here, this flow is introduced in the solution heat exchanger ESS where its temperature is still increased using the sensible heat of the weak solution flow.

Finally, the strong mixture will enter the generator at point 6. First it flows over the heat exchanger EEG where it is heated by the refrigerant-weak mixture coming from EBG and by the partial condensation of the vapor stream 19. Consequently the liquid reaches the bubble point at the EBG inlet. Here it will continue to receive only the heat of the weak solution issued from the generator. The solution begins to generate vapor immediately after it enters the device and its temperature continuously rises as the fluid goes down. From the point 22 the liquid mixture is heated by the high temperature external driving source for the upper loop generator, or by the intermediate agent for the lower loop generator. The liquid flow rate and concentration diminish from the entry to the exit point 7. The same variation may be observed for the similar parameters of the generated vapor.

The generator supplies a vapor stream 17 having an insufficient high ammonia concentration, so a rectifier is needed to improve this parameter. The vapor temperature and flow rate are decreased in this device and a reflux liquid stream 18 is also produced. It is sent back to the generator and mixed with the entering strong solution 6. The resulted heat flux is rejected to the cooling agent in the lower loop, or transferred to the intermediate agent leaving the absorber in the upper loop. And so the beginning condenser entry point 8 was found again.

The air-conditioning agent has a parallel circulation through the two evaporators as much as the cooling agent through the lower loop condenser, absorber and rectifier. The intermediate agent turns in a closed circuit, being cooled in the lower loop generator and heated in the upper loop condenser, absorber and rectifier respectively.

3. THE CALCULATION MODEL

A design calculation procedure was conceived in order to determine the operating parameters allowing to reach the highest values for the COP. The thermodynamic properties of the water-ammonia solution were modeled by using computer subroutines based upon Schulz and Ziegler formulas (/5/, /6/). All the hereafter appearing notations are made in keeping with the figure 2. The points referring to the lower loop always have a '0' index. The following input data, which are specific to design calculations, were considered for the beginning:

- the input/output temperatures of the chilled water: $t_{35} / t_{36} = 12 / 7 \text{ } ^\circ\text{C}$;
- the input/output temperatures of the cooling agent: $t_{33} / t_{34} = 32 / 37 \text{ } ^\circ\text{C}$;
- the heat flux to be extracted by the two evaporators: $q_{E, \text{tot}} = 15 \text{ KW}$;

The thermodynamic levels of the hot sources, which must be known to have a determined system, are considered by imposing the temperatures of the weak solution at the generator outlet: t_{07} and t_7 . Their values will be defined after an optimization study in order to maximize the global COP.

The heat transfer equations are taken into account by utilizing the pinch concept. This approach considers that the minimal temperature difference between two fluids exchanging heat specifies the size of the heat transfer area. The following pinch values were used:

- $\delta t_C = \delta t_{C0} = 3 \text{ } ^\circ\text{C}$ for the two condensers;
- $\delta t_E = \delta t_{E0} = 2 \text{ } ^\circ\text{C}$ for the two evaporators;
- $\delta t_{SR} = \delta t_{SR0} = 3 \text{ } ^\circ\text{C}$ for the pre-coolers;
- $\delta t_A = \delta t_{A0} = 3 \text{ } ^\circ\text{C}$ for the heat flux rejected from the solution liquid film to the coolant, in the two absorbers;
- $\delta t_{EEG} = 3 \text{ } ^\circ\text{C}$ and $\delta t_{EEG0} = 4 \text{ } ^\circ\text{C}$ for the heat transfer between the weak-refrigerant solution and the boiling liquid film, in EEG and EEG0 respectively;
- $\delta t_{EBG} = 3 \text{ } ^\circ\text{C}$ and $\delta t_{EBG0} = 4 \text{ } ^\circ\text{C}$ for the analogous processes taking place in EBG and EBG0;
- $\delta t_{EA} = 3 \text{ } ^\circ\text{C}$ for the EA heat exchanger and $\delta t_{EA0} = 4 \text{ } ^\circ\text{C}$ for the analogous lower loop device (these values refer to the heat transfer between the solution liquid film and the coolant);
- $\delta t_{29-07} = \delta t_{30-022} = 4 \text{ } ^\circ\text{C}$ for the heat flux removed from the intermediate agent to the boiling liquid film in the lower loop generator.

If the above mentioned pinch value for the precooler leads to obtain superheated vapor at this device exit, then the condition was replaced by the constraint to have saturated vapor in this point. A $5 \text{ } ^\circ\text{C}$ heating was assumed for the vapor flows passing through the respective absorbers ($\delta t_{13-2} = \delta t_{013-02} = 5 \text{ } ^\circ\text{C}$). The temperatures t_3 and t_{03} were considered as input data instead of the pinch values for ESS and ESL heat exchangers.

The heat transfer equation from the boiling liquid film to the generated vapor was taken into account by imposing the efficiencies $\eta_G = \eta_{G0} = 0,97$ for the devices EBG, EBG0, G and G0. A $5 \text{ } ^\circ\text{C}$ cooling was considered for the vapor streams passing through EEG and EEG0 devices ($\delta t_{19-17} = \delta t_{019-017} = 5 \text{ } ^\circ\text{C}$). In the two rectifiers (R and R0)

the concentration values of the rectified vapor (y_8 and y_{08}) are input data instead of the pinch values for the heat transfer between the reflux liquid film and the coolant. The other heat flux exchanged in these devices, which passes from the vapor to the liquid film, is considered by imposing the rectification efficiencies $\eta_R = \eta_{R0} = 0,95$.

Each mass transfer equation is approached by supposing that the participating liquid and vapor flows leave the exchanger at a saturated state. The only exception is at the absorber, from where the strong solution is supposed to exit in a subcooled state, given by the absorption efficiency value $\eta_A = \eta_{A0} = 0,95$.

The liquid compression in the solution pumps was modeled by introducing the pump efficiencies $\eta_P = \eta_{P0} = 0,5$. The heat and pressure losses were neglected for all the flow processes, so that any loop will have two characteristic pressures.

The COP variations with the efficiencies and pinch temperature differences are easy to predict, therefore they will not be studied. Their presented values will be kept constant in all the future considerations. The other input parameters i.e. the temperatures t_3 , t_{03} and the concentrations y_8 , y_{08} will be the object of an optimization study.

4. THE COP OPTIMIZATION FUNCTION OF THE INPUT DATA

It is reminded that in the paragraph 3 the following input parameters were considered the object of an optimization study: the temperatures t_7 , t_{07} , t_3 , t_{03} and the concentrations y_8 , y_{08} . The study of the t_3 and t_{03} influence upon the COP showed that their optimal values correspond to the saturated state of the weak solution. So that only the effect of the four remaining parameters (t_7 , t_{07} , y_8 , and y_{08}) will be analyzed in this study. The COP variations are shown in figure 3. Only the variables belonging to the same loop were concomitantly modified. As it may be seen, the following values seem to maximize the energetic performances: $t_7 = 86$ °C; $t_{07} = 215$ °C; $y_8 = 0,976$; $y_{08} = 0,996$. A maximal COP of 1,5418 is predicted.

7. CONCLUSIONS

A calculation model and a computer program were developed for the design calculation of the triple-effect absorption cycle using the ammonia-water pair. The pinch-point analysis technique allowed a realistic approach of the complex heat transfer processes. The optimization study revealed the limits of the energetic performances that may be reached by this installation. The presented architecture is feasible and does not require expensive technologies.

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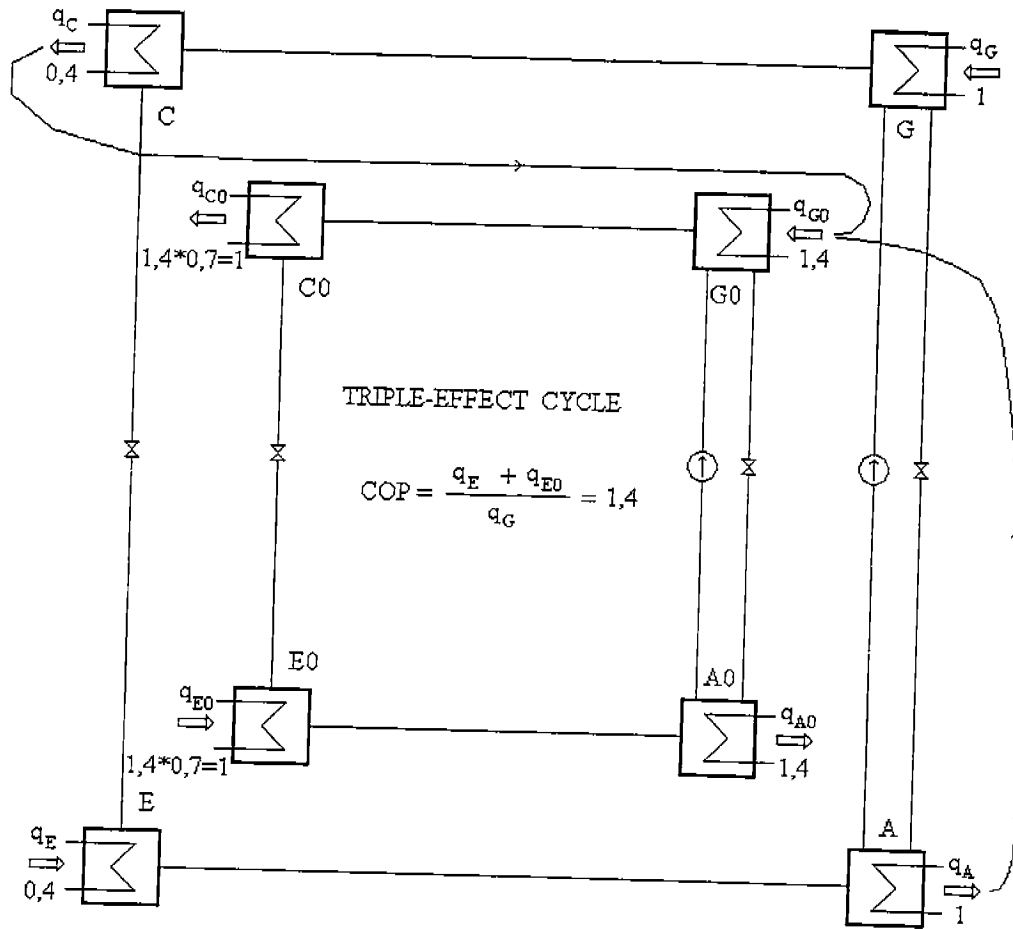


Fig. 1. Triple-effect cycle.

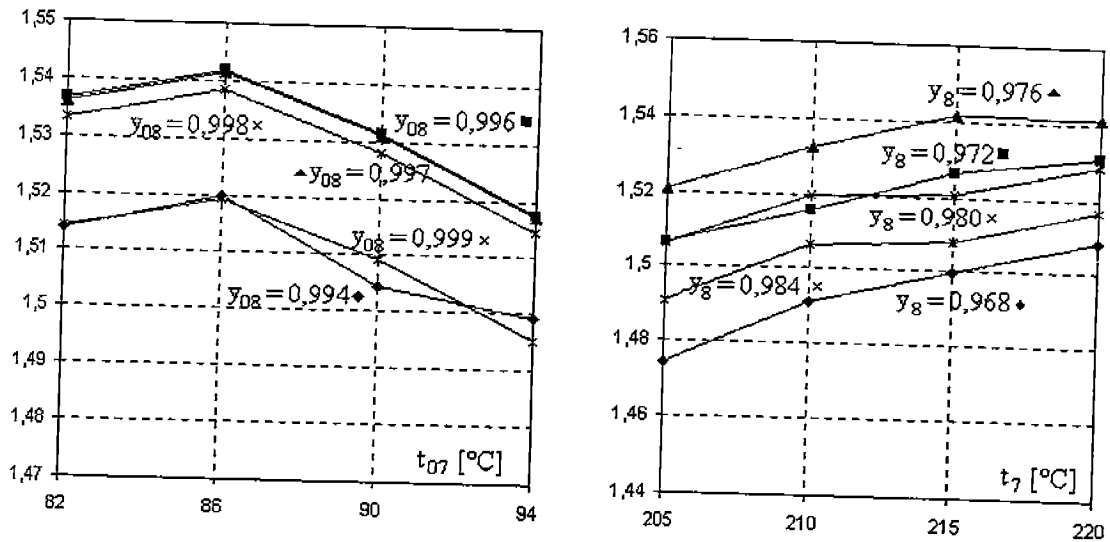


Fig. 3. COP variations with t₀₇, y₀₈, t₇ and y₈.

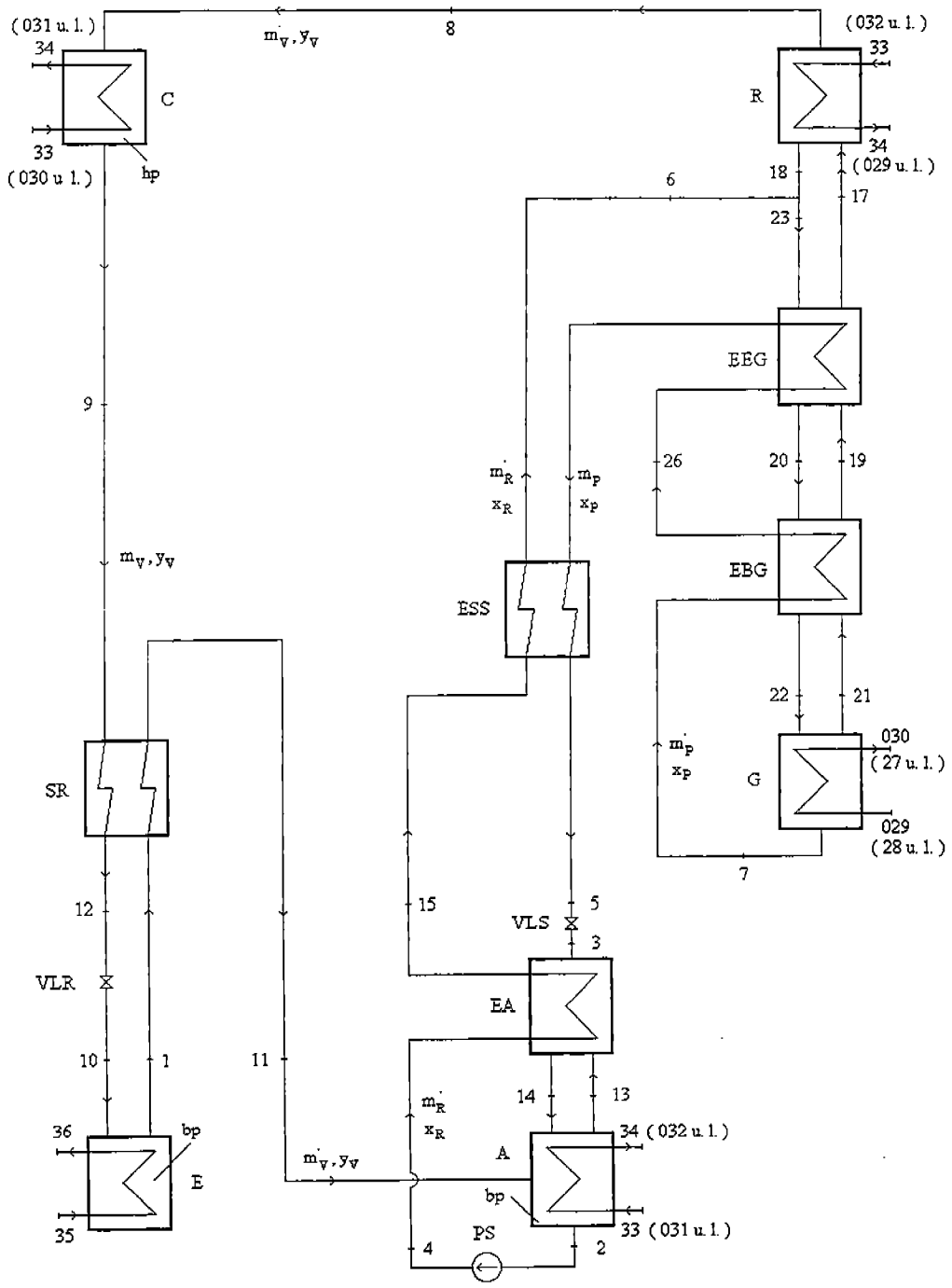


Fig. 2. Single loop scheme.