On the Performance of Ejector Refrigeration Systems

ABDELOUAHID DAHMANI [1], ZINE AIDOUN [2], NICOLAS GALANIS [1]
[1] Génie Mécanique, Université de Sherbrooke
Sherbrooke (Qc) J1K2R1, CANADA
[2] CanmetENERGY, Natural Resources Canada
1615 Lionel Boulet, C.P. 4800, Varennes (Qc) J3X 1S6, CANADA
Abdelouahid.Dahmani@USherbrooke.ca, zaidoun@nrcan.gc.ca, Nicolas.galanis@usherbrooke.ca

Abstract: The design characteristics of an ejector refrigeration system using R134a with fixed cooling capacity and fixed inlet temperatures of the external fluids at the inlet of the generator, the condenser and the evaporator are presented for different pressures of the R134a in the generator and different values of the pinch in the heat exchangers. It is shown that the COP increases while the total conductance of the heat exchangers, the mass flow rate of the R134a and the area of the mixing chamber decrease as this pressure increases and the pinch decreases.

Keywords: R-134a; Computer simulation; Finite size thermodynamics; COP.

1 Introduction

Cooling in industrial processes, air-conditioning of buildings and refrigeration of perishable products are common practises throughout the world. In industrialised countries the energy consumption of such installations, used to create and maintain relatively low temperatures, represents an appreciable part of the corresponding total. Thus, in Canada approximately 10% of the total annual energy consumption is used for such operations.

The systems used to achieve heat removal from a low temperature reservoir are driven either mechanically (e.g. compression of vapour refrigerants) or thermally (e.g. absorption systems or ejector driven systems). The former constitute the vast majority of industrial, commercial and residential installations but the latter are attracting a lot of interest since they can be activated by low temperature renewable energy sources (e.g. solar) or industrial waste heat. Absorption systems use a combination of two fluids and have been commercially available for several decades but are quite complex due to the heat and mass transfer processes in the absorber and desorber. Furthermore, the fluid combinations used in absorption systems are very limited in number. On the other hand, ejector systems use a single fluid and thus offer great flexibility and promise for the replacement of environmentally unacceptable refrigerants by benign ones. However, except for one very recent automotive application, ejector systems are not available commercially.

Early semi-analytical studies using ideal gas relations to model the operation of ejector refrigerators do not reflect the complexities due to real fluid properties [1]. Thus, recent numerical studies [2, 3] use tabulated thermophysical properties for the design of ejectors and the prediction of the performance of ejector refrigerators. This approach is quite successful as shown by Eames et al. [4] who tested such a system using low pressure steam (2 to 3.6 bars, or 120-140 °C, at the boiler) and found that the experimental data was approximately 85% of the theoretical values. Their experiments also showed that choking of the secondary flow plays an important role in the system performance. Maximum COP was obtained when the ejector was operated at its critical flow condition. Other interesting experimental results have been reported by Aphornratana & Eames [5] who showed the benefit of using an ejector with a primary nozzle that can be moved axially in the mixing chamber, by Huang & Chang [6] who tested 15 ejectors using R141b as the working fluid and by Selvaraju & Mani [7] who studied experimentally the influence of generator, evaporator and condenser temperatures on the performance of an ejector refrigeration system using R134a as the working fluid. They reported that for a given ejector geometry and fixed condenser and evaporating temperatures, there exists an optimum temperature of the primary vapour which maximises the entrainment ratio and the COP.

The present paper presents the results of a systematic parametric study of an ejector refrigeration system of fixed cooling capacity using R134a as the working fluid and fixed temperatures of the external fluids at the inlet of the vapour generator, the condenser and the evaporator. The methodology is based on classical as well as finite size thermodynamics and shows the effects of two independent design parameters (the pressure of the working fluid in the vapour generator and the pinch in the three heat exchangers) on several geometrical characteristics (throat area of the ejector, area of the mixing chamber, thermal conductance of the heat exchangers) and operating conditions (working fluid pressure in the heat exchangers, coefficient of performance, etc.).

2 System description and modelling

Figure 1 shows a schematic representation of the system under consideration while Figure 2 shows a temperature-entropy diagram of the corresponding processes. The superheated vapour at 1 is condensed by rejecting heat to a fluid stream whose temperature increases from $T_{c,in}$ to $T_{c,out}$. At 2, the exit from the condenser, the working fluid is assumed to be saturated liquid. Part of it (the secondary or entrained fluid) is throttled to a low pressure and evaporated by receiving heat from a second fluid stream whose temperature decreases from $T_{e,in}$ to $T_{e,out}$. At 6, the exit from the evaporator, the working fluid is assumed to be saturated vapour. Another part of the working fluid at 2 (the primary or driving fluid) is pumped to a high pressure and evaporated from 3 to 4 in the vapour generator by receiving heat from a third fluid stream whose temperature decreases from $T_{s,in}$ to $T_{s,out}$. The high pressure vapour at 4 expands in a converging-diverging nozzle to a very low pressure P_7 and aspirates the saturated vapour from 6. These two low pressure streams of working fluid mix in a constant area chamber emerging at state 8. Finally this mixture is decelerated to state 1 in a diffuser.

To study the performance of such a system it is first of all necessary to have a reliable method for determining the thermodynamic properties of the working fluid. These are usually available in handbooks or in electronic form (e.g. REFPROP).

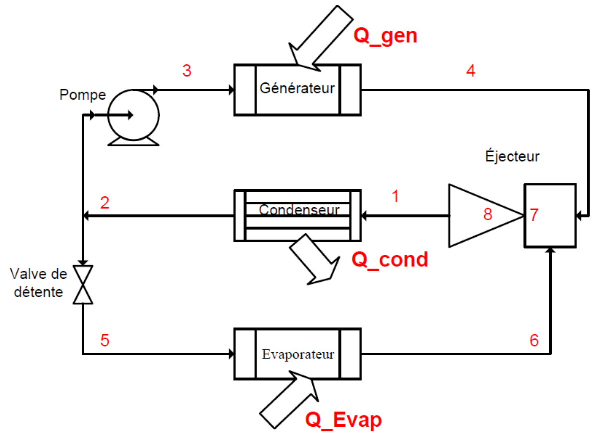

Fig. 1: Schematic representation of the system

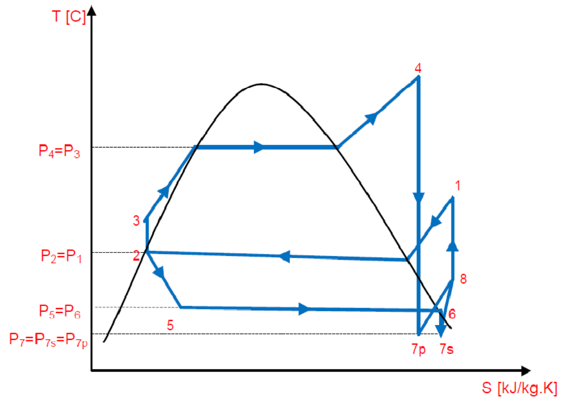

Fig. 2: T-S diagram of the processes

In order to model these processes it is assumed that potential energy as well as friction and heat losses are negligible while acceleration (from 4 to 7p and from 6 to 7s) as well as deceleration (from 8 to 1) and pumping (from 2 to 3) are reversible and adiabatic. Hence the following relations for pressure P and entropy S apply:

$$P_3 = P_{3f} = P_{3g} = P_4 \qquad P_2 = P_{2g} = P_1 \quad P_5 = P_6 \tag{1}$$

$$S_2 = S_3 \quad S_4 = S_{7p} \quad S_6 = S_{7s} \quad S_1 = S_8 \tag{2}$$

Three other relations for the thermodynamic properties result from the previously stated assumptions of saturated conditions at states 2 and 6 and the equality of the temperature at states 5 and 6. Two additional ones are obtained by calculating the pressure of the primary and secondary fluids at the inlet of the mixing chamber (P_{7p} = P_{7s}) so as to minimize the corresponding flow area of the secondary flow (A_{7s}) based on the experimental results of Eames et al. [4].

Kinetic energy is also assumed to be negligible except at states 7p, 7s and 8. The application of the first law of thermodynamics to the evaporator, the two parts of the condenser (desuperheating and condensing) and the three parts of the vapour generator (liquid heating, evaporation and superheating) therefore results in six equations which express the corresponding heat quantities Q_E, Q_C, Q_G or parts thereof. These equations involve the mass flow rates of the primary and secondary working fluid ($\dot{m}_p$ and $\dot{m}_s$ respectively), the specific enthalpy of the working fluid at all states except 7p, 7s and 8 as well as the mass flow rates of the three external fluids ($\dot{m}_G$ at the vapour generator, $\dot{m}_C$ at the condenser and $\dot{m}_E$ at the evaporator) and their corresponding temperatures indicated in Figure 2.

It is also possible to express the heat transferred between the working fluid and the external streams in each of the previously identified parts of the heat exchangers as the product of a thermal conductance UA and a corresponding logarithmic temperature difference, thus resulting in six additional equations relating internal and external temperatures to these thermal conductances.

The application of the first law to the acceleration and deceleration processes gives

$$h_4 = h_{7p} + 0.5(V_{7p})^2 \qquad h_6 = h_{7s} + 0.5(V_{7s})^2$$
$$h_1 = h_8 + 0.5(V_8)^2 \tag{3}$$

while $h_2 = h_5$ for the throttling process and the expression of the pump's specific work leads to

$$h_3 - h_2 = v_2 (P_3 - P_2) \tag{4}$$

The number of variables appearing in these equations is 47. They include two thermodynamic properties of the working fluid at each of the nine states identified in Figures 1 and 2 as well as its velocity at states 7p, 7s and 8. They also include the five mass flow rates ($\dot{m}_p$,$\dot{m}_s$,$\dot{m}_G$, $\dot{m}_C$ and $\dot{m}_E$) as well as the heat quantities transferred between the working fluid and the external streams in each of the previously identified six parts of the heat exchangers and the corresponding six thermal conductances. Finally they include the nine temperatures of the external streams indicated in Figure 2.

Since the number of the previously identified equations is only 35 it is necessary to add more equations and fix the value of some variables. It is therefore assumed that the cooling effect Q_E and the temperatures of the three external streams entering the heat exchangers (i.e. $T_{s,in}$, $T_{c,in}$ and $T_{e,in}$) are known. Furthermore the value of a temperature difference DT which characterizes the heat transfer processes is also assumed known. This parameter relates the external and internal temperatures by means of the following relations:

$$T_4 = T_{s,in} - DT \qquad T_2 = T_{c,in} + DT$$
$$T_6 = T_{e,in} - DT \tag{5}$$

$$T_s'' = T_{3f} + DT/2 \qquad T_c' = T_2 - DT/2$$
$$T_{c,out} = T_5 + DT/2 \tag{6}$$

With these additions the solution depends on DT and one other parameter. In the present study we have chosen to analyse the effects of the working fluid pressure in the vapour generator ($P_3 = P_{3f} = P_{3g} = P_4$) on the performance of the system.

3 Results and discussion

The results presented here are for R134a as the working fluid and water as the external fluid for all three heat exchangers. They were obtained with Q_E = 5 kW, $T_{s,in}$ = 95 °C, $T_{c,in}$ = 20 °C, $T_{e,in}$ = 10 °C and three values of DT (5, 7.5 and 10 °C). The calculations were performed with EES (Engineering Equation Solver) which includes thermodynamic properties for a large number of natural and synthesized fluids.

For these conditions the pressure in the evaporator (P_5=P_6), the pressure of the primary and secondary flows at the inlet of the mixing chamber (P_7), the mass flow rate of the secondary flow ($\dot{m}_s$) and the area of the secondary flow at the inlet of the mixing chamber (A_{7s}) are independent of P_4. Their values depend only on DT and are given in Table 1. We note that, as DT increases, the values of P_5=P_6 and of P_7 decrease. Furthermore, when DT and P_5=P_6 increase the specific enthalpy difference in the evaporator (h_6-h_5) decreases and, since Q_E is fixed, the mass flow rate of the secondary fluid increases. However, the increase of $\dot{m}_s$ is approximately 10% when DT increases by 100%. This increase of the mass flow rate causes a corresponding increase of the area A_{7s}.

Figure 3 shows that the mass flow rate of the primary fluid increases when P_4 decreases and DT increases. It should be noted that the value of P_4 should always be higher than the saturation pressure corresponding to T_2. Furthermore, an upper limit equal to the saturation pressure corresponding to T_4 is also imposed on the value of P_4. By virtue of Eq. 5

both the upper and lower limits of P_4 depend on DT. The results of Fig. 3 indicate that the range of P_4 respecting these limits decreases as DT increases.

DT [°C]	P_6=P_5 [kPa]	P_7 [kPa]	A_{7S} [cm^2]	$\dot{m}_S$ [kg/s]
5	349.9	206.6	6.232	0.02995
7.5	320.5	189.0	6.997	0.03088
10	293.0	172.6	7.878	0.03187

Table 1: Effect of DT on some operational parameters and geometric characteristics

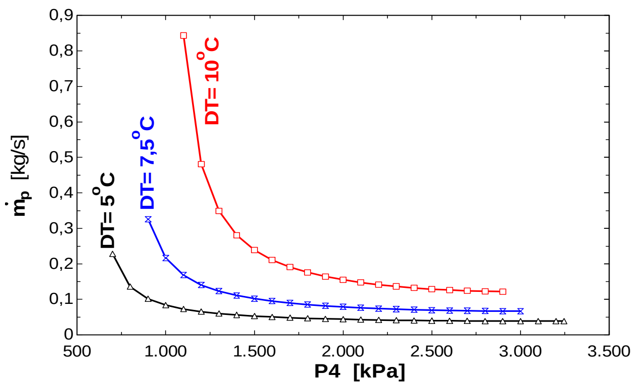

Fig. 3: Effects of P_4 and DT on primary fluid mass flow rate

This behaviour of the primary fluid mass flow rate results in corresponding changes to the heat which must be provided in the generator (Fig. 4), the heat rejected by the condenser (Fig. 5), the mass flow rate of the cooling fluid through the condenser (Fig. 6) and the power required by the pump. On the other hand, the mass flow rate of the external fluid through the evaporator is independent of P_4 while that of the generator varies very little with P_4.

As a result, the coefficient of performance of the cycle

$$\mathrm{COP} = \mathrm{Q_E}/(\mathrm{Q_G} + \mathrm{W_p}) \tag{7}$$

increases when P_4 increases and when DT decreases (Fig. 7). However the increase of the COP due to the decrease of DT comes at a high price because it is accompanied by an increase of the total conductance of the three heat exchangers (sum of the corresponding UA) as illustrated in Fig. 8.

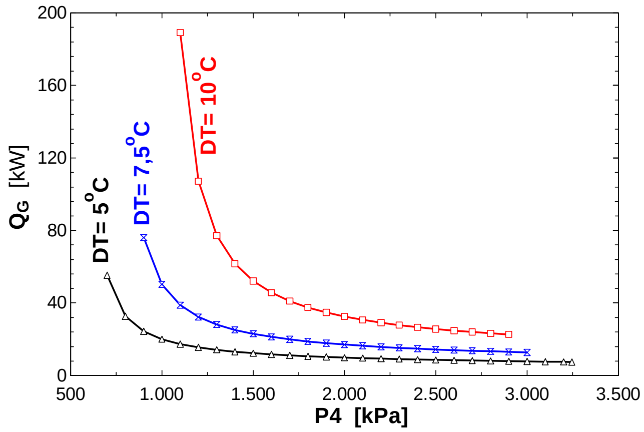

Fig. 4: Effects of P_4 and DT on the heat provided to the generator

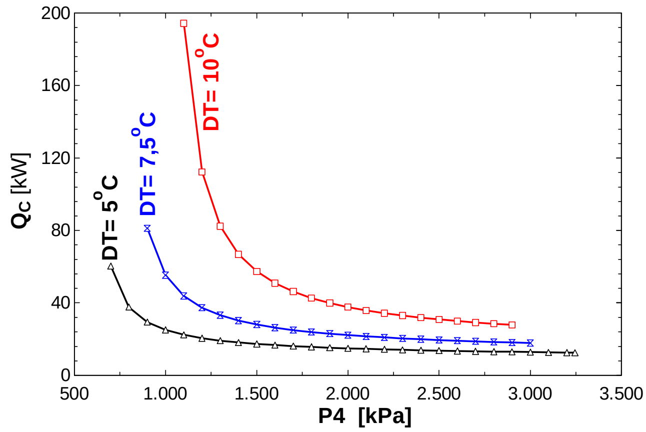

Fig 5: Effects of P_4 and DT on the heat rejected by the condenser

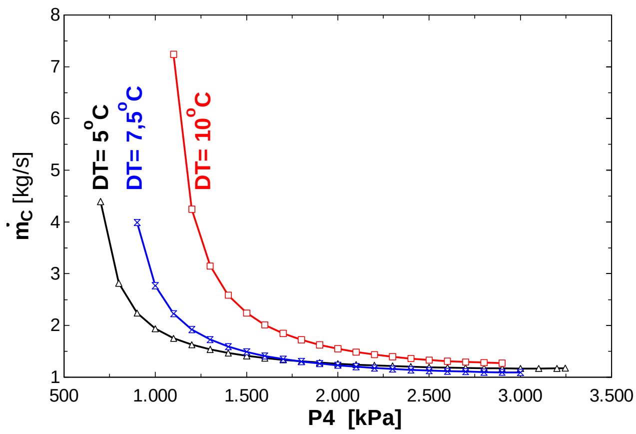

Fig. 6: Effects of P_4 and DT on the mass flow rate of the condenser cooling fluid

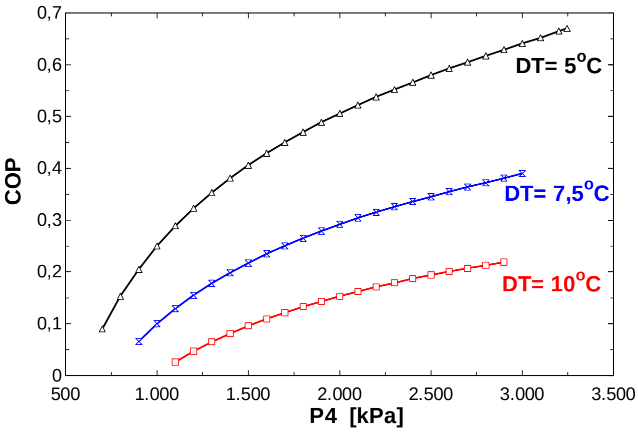

Fig. 7: Effects of P_4 and DT on the coefficient of performance

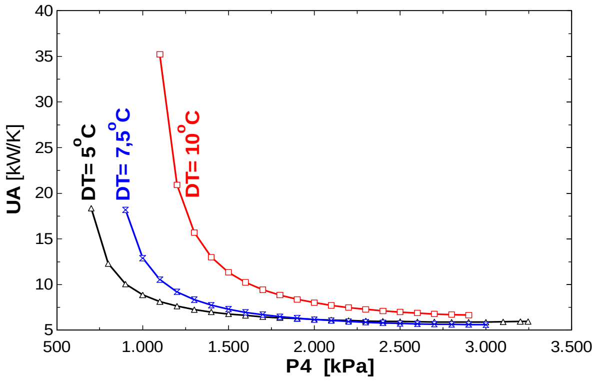

Fig. 8: Effects of P_4 and DT on the total thermal conductance of the system

These results show that the COP is very sensitive to the value of DT. On the other hand, for high evaporator pressures the total thermal conductance of the system does not vary much with DT. Since the operation cost is inversely proportional to the COP and the construction cost increases with UA, the above results suggest that for the conditions under consideration here the designer of the system should aim for a high value of P_4 and a low value of DT. A more detailed cost analysis is however necessary to confirm this conclusion.

4 Conclusion

A model of an ejector refrigeration system using R134a as the working fluid with fixed cooling capacity and fixed inlet temperatures of the external fluids at the inlet of the generator, the condenser and the evaporator was used to evaluate the dimensions of the mixing chamber and the conductance of the heat exchangers. It was shown that the COP increases while the total conductance of the heat exchangers, the mass flow rate of the R134a and the area of the mixing chamber decrease as this pressure increases and the pinch decreases.

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