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THEORETICAL ANALYSIS ON THE USE OF AN EJECTOR IN VAPOR COMPRESSION REFRIGERATING USING SUBCRITICAL REFRIGERANTS

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Abstract. *This paper presents the results of a theoretical investigation of the use of a two-phase ejector for partial work recovery of the expansion process in a refrigeration cycle operating with subcritical refrigerant R-410A and R-134a. A thermodynamic modeling of the ejector cycle, based on one-dimensional analysis presented by Kornhauser (1990) is presented and solved for proof of its effectiveness by comparing with the results reported by Domanski (1995). The model was solved for ideal operating conditions and for a realistic case using the refrigerants R-410A and R-134a. The results shown increases in COP in the order of up to 33% for R-410A and 26.5% for R-134a in the ideal case and 9% for R-410A and 5.7% for R-134a in the realistic case.*

Keywords: *Refrigeration, Ejector cycle, Performance, COP*

1. INTRODUCTION

The refrigeration is today an essential role in the life of human beings or thermal comfort or for storing products such as foods and medicines. Since the advent of the industrial revolution, the development of refrigeration processes evolved significantly. Today there are several cooling methods, but the environmental policies of modern community require processes that maintain the integrity of the environment by reducing energy consumption and the reduction the emission of gases. Faced with this demand, many researches are being conducted to develop alternatives for maximizing the efficiency of cooling systems.

One of the most irreversible processes in vapor compression refrigeration cycle is the isenthalpic expansion process due to the imposition of a throttling the refrigerant flow from the condenser to reduce its pressure to evaporator pressure. Many studies have been and are being conducted in order to reduce these losses through expansion devices or machines (expanders) with recovery work. A fairly comprehensive review of these devices can be found in Liu and Groll (2008).

One of the possible ways of reducing the irreversibility due to throttling is with an ejector, utilizing the kinetic energy of the flash vapor to increase the suction pressure at the compressor inlet (Domanski, 1995). The ejector is then ideally an expansion isentropic device able to recover the work that is somehow lost in the isenthalpic process of a conventional expansion device.

The basic reason for this statement is the use of the ejector compression effect to re-pressurizing the refrigerant in the suction line to an intermediate pressure, reducing the compressor work. Given that, many researches are developed in search of the ideal model of ejector for this application and to determine the actual increase in cycle performance after applying the ejector as expansion device.

The aim of this study is to analyze one of the models available in the literature and present the cycle performance parameters with ejector for some refrigerants currently used.

2. REFRIGERATION CYCLE WITH TWO-FASE EJECTOR

Elbel e Hrnjak (2008) divides the ejector in four sections, as shown in Fig. 1: the convergent-divergent nozzle, suction chamber, constant area section and the diffuser. The basic principle of operation of an ejector is the conversion of internal energy and pressure of the motive fluid flow stream into kinetic energy. The dimensions of the ejector sections determine their capacity and efficiency.

The compression of the fluid in the ejector is possible from the kinetic energy of the primary fluid that undergoes an adiabatic expansion in the convergent-divergent nozzle. At the exit of primary nozzle, fluid reaches supersonic speed, producing a low-pressure region in suction chamber from the secondary fluid. The secondary fluid is then drawn into the ejector chamber. The primary flow continues to expand in the form of a convergent duct without mixing with the secondary fluid. The secondary fluid is entrained by the primary flow and accelerated due to shear stresses through the mixing chamber. At some point in this mixing chamber, the secondary fluid reaches sonic velocity and undergoes blocking. Munday and Bagster (1977) defined this cross section as an effective area. After the shock, the two flows form a single stream. The flow then decelerating in the ejector diffuser, by compressing up to an intermediate pressure, recovering part of the kinetic energy of the flow and reducing the need of compressor work.

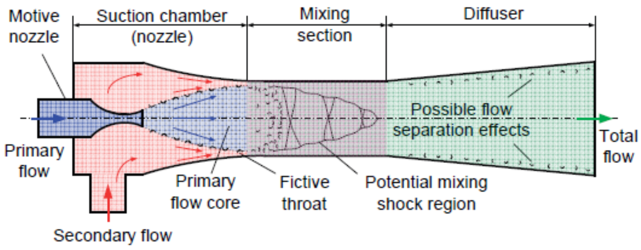

Figure 1. Simplified design of an ejector

2.1 Historical aspect

The existing information in the open literature is that Gay (1931) deposited the first patent for a refrigeration cycle where the ejector is applied for work recovery in expansion process. The cycle proposed by Gay, as shown in Fig. 2, according to Lawrence e Elbel (2013), is considered the standard cycle for refrigeration with expansion ejector and will be called cycle model in this work.

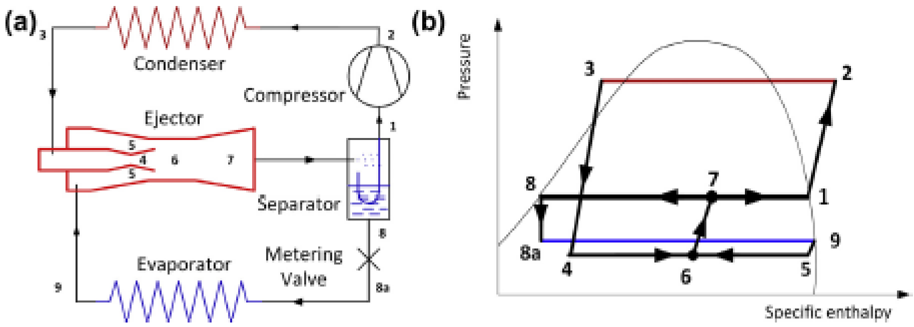

Figure 2. Basic diagram of a standard cycle with ejector (a) and (b) P-h diagram of the cycle proposed by Gay (1931).

There are two advantages in the refrigeration cycle proposed by Gay. First, the cooling capacity is increased since the ideally isentropic inner process results, in the ejector, greater enthalpy difference between evaporator inlet and outlet when compared to a conventional cycle provided with expansion valve. The second advantage is that the coefficient of performance (COP) of the cycle is maximized due to the reduction of the compressor work (Elbel and Hrnjak, 2008). Kornhouser (1990) presented a one-dimensional iterative model working with R-12. The author defines individual efficiencies for each component of the ejector (primary nozzle, secondary nozzle and diffuser). These efficiencies are introduced in the calculation to represent flow losses in the ejector. Kornhauser (1990) introduced the simplification that the losses caused by the shear stresses arising from the mixing of the two flows can be reduced when the flows enter the suction chamber at similar speeds. Many of the current research still follow this line of reasoning. The numerical approximation of Kornhauser (1990) was used for Nehdi et al. (2007) to numerically investigate the performance of a vapor compression cycle equipped with a two-phase ejector instead of an expansion valve. Of all the subcritical fluids investigated in the experiment, the R-141b showed the largest increase in COP of 22% compared to the baseline conventional cycle with expansion valve.

Li and Groll (2004) and Li and Groll (2005) presented results of simulation of a transcritical CO_2 cycle applied to room air conditioning, equipped with a two-phase ejector. The analysis was also based on the Kornhauser (1990) model. The authors reported increases in COP up to 16%. The investigated cycles were not equipped with intermediate heat exchangers. Thereafter, Li (2006) did report an experiment where he found that for temperatures above 49 °C the ejector is unable to improve the efficiency of the transcritical cycle of CO_2, mainly because at high temperatures, the mass flow ratio reduces significantly.

Lawrence and Elbel (2013) conducted an experimental study of substitution of the expansion valve by a two-stage ejector. In this study, the authors proposed the use of COS cycle (Condenser Outlet Split) developed by Oshitani et al. (2005), where the authors tested the performance of the cycle with R-134a and R-1234yf as the working fluids. COS cycle, as shown in Fig. 3, has been proposed by Oshitani et al. (2005) as an alternative to the cycle customarily used to make possible a better control of the cycle stability when using a subcritical refrigerant as the working fluid.

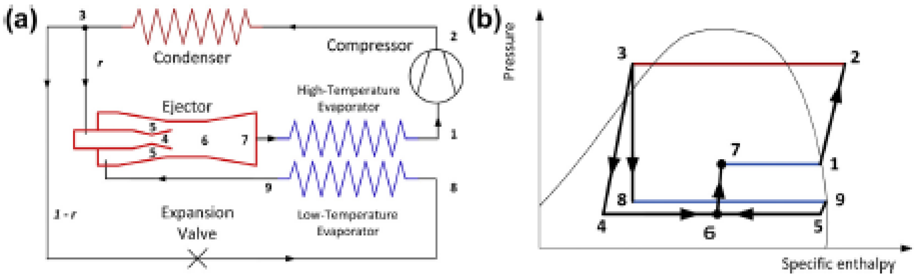

Figure 3. Layout (a) and P-h diagram (b) of the cycle proposed by Oshitani (2005).

In COS cycle, the saturated liquid leaving the condenser is divided into two streams: one stream is sent to the primary nozzle of the ejector and the other is sent to an isenthalpic expansion device, feeding the low temperature evaporator. In the evaporator, the low temperature fluid is expanded and then drawn into the secondary nozzle ejector. In the ejector, the two flows are mixed, compressed and sent to the high temperature evaporator where the mixture is vaporized, entering in the compressor suction line. The authors justified the use of COS cycle by analyzing the impact on the COP of the cycle, if the liquid/vapor separator present inefficient performance. Figure 4, according Lawrence e Elbel (2013), shows the effect on COP of the inefficiency in liquid/vapor for R-134a and R-1234yf refrigerants, were η_{vapor} and η_{liquid} are the efficiencies of the liquid / vapor separator.

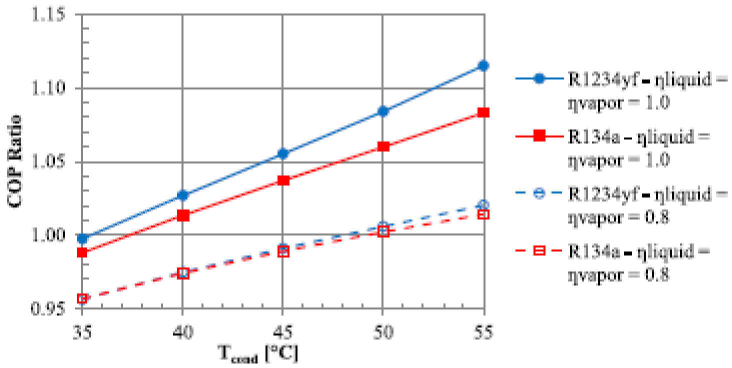

Figure 4. Effect of the inefficiency in liquid/vapor separation for R-134a and R-1234yf in the COP cycle

The authors compare the results of the experiment data obtained constructed of a COS cycle with two expansion devices and a conventional vapor compression cycle. It was reported that R-1234yf had a higher ability to increase the cycle COP compared to R-134a. For the cycle with R-1234yf gain was 6% and R-134a gain was 5% compared to the conventional COP cycle. When compared to COS cycle fluids performance achieved 12% and 8% for R-1234yf and R-134a, respectively.

The isenthalpic expansion process is identified as one of the irreversibility of the CO_2 cycle. Because the ejector present huge potential for improvement in the efficiency of cycles, it is seen as a solution to the low COP found with the use of this refrigerant.

Lucas and Koehler (2012) presented a study of the use of the ejector to expand in a CO_2 cycle. This study was performed at varying pressures and different temperatures of the evaporator outlet to study the influence of fluctuation in the efficiency ratio of input and ejector discharge pressure. The result was an increase in the COP of the cycle 17% to 22%. The authors concluded that the ejector efficiency decreases proportionately with the reduction of the fluid temperature at the evaporator exit and the evaporation pressure reduction, and for the lower evaporating pressure tested was not registered any significant improvement in cycle efficiency.

Liu and Groll (2008) studied the ejector as an expansion device for the CO_2 transcritical cycle and R-410A as the working fluid. The study was carried out in the laboratory with a transcritical CO_2 cycle equipped with micro channels heat exchangers and a single stage semi-hermetic compressor. Initially it was established by testing the cycle COP operating with a conventional expansion valve under standard conditions for the cold inner environment (Ta,i = 26.7 °C and RH = 50%) and the external environment (Ta,e = 35 °C). The COP_b (performance coefficient of the basic cycle) to a discharge pressure of 121.26 bar was 1.105. After the replacement of the expansion device, the authors reported a coefficient of 1.422 for the same operating conditions. This represents an increase of 23% in the system COP. The study concludes that the ejector maximizes the COP and cooling capacity of the system as the temperature of the outside environment increases. The peak of the increase went to condition Ta,i = 26.7 °C and RH = 50% and Ta,e = 37.8 °C where COP_e was 1.639 versus an COP_b 1.185 and $\dot{Q}_E$ = 16.08 kW versus the base cycle $\dot{Q}_E$ = 11.42 kW. The authors used the survey results to establish empirical relationships to forecast performance ejectors. Using these correlations, a theoretical study was made of an ejector for use in a cycle operating with R-410A as the refrigerant. For standard operating conditions (Ta,i = 26.7 °C, RH = 50% and Ta,e = 35 °C) COP and cooling capacity were calculated from 4.786 kW and 23.22, respectively. These results represent an increase in COP 11.1% and 19.8% in cooling capability compared to conventional refrigeration cycle.

2.2 Modeling cycle

In this work we used the model ejector cycle developed by Kornhouser (1990). The basic equations are described below, according to the ejector subdivisions. Like the other models available in the literature, the Kornhauser model is based on the mass conservation, energy and momentum to mixing process equations.

To solve this set of equations was developed a program in the EES (Engineering Equation Solver) software platform (Klein and Alvarado, 2014), mainly for its ease in obtaining the thermodynamic properties of different refrigerants.

According Kornhouser (1990) and Domanski (1995), for a given operating conditions and efficiencies of the nozzle and the diffuser, there is an optimal pressure of the mixture, P_6, below the suction pressure, which optimizes the COP of the ejector cycle. Therefore, P_6 is a model input data, previously optimized. This model also requires the pressure as input data and the specific enthalpy entering the primary nozzle, P_3 and h_3, the pressure and specific enthalpy of the ejector suction inlet, h_9 and P_9, and the ratio between the mass flow rate into the primary nozzle and the total mass ratio, r, defined by Eq. 1:

$$r = \frac{\dot{m}_{mm}}{\dot{m}_{mm} + \dot{m}_{ms}} \tag{1}$$

where $\dot{m}_{mm}$ is the mass flow rate at the primary nozzle and $\dot{m}_{ms}$ is the mass flow rate at ejector suction, defined from the capacity and enthalpy change of the evaporator.

The ejector model also requires estimates of the efficiencies of the primary nozzle, suction and diffuser, η_{mm}, η_{ms} e η_d, respectively. The sub-indices used for the specification of the thermodynamic properties (specific enthalpy h and entropy s) in and out of each device, obtained from the mass balance, energy and momentum, represent the states defined in Fig. 2 (b). In addition, u is the refrigerant velocity and x is the vapor quality.

The enthalpy of the motive fluid at the exit of the primary nozzle, at isentropic condition, is a function of the entropy of the fluid in the primary inlet nozzle (s_3) and the mixing region pressure (P_6). The specific enthalpy of the motive fluid at the exit of primary nozzle is determined by its efficiency, η_{mm}, according to Eq. 2:

$$h_4 = h_3 + \eta_{mm}\left(h_{4,is} - h_3\right) \tag{2}$$

where $h_{4,is}$ is the enthalpy of isentropic expansion process, defined by the Eq. 3:

$$h_{4,is} = f\left(P_6, s_3\right) \tag{3}$$

The motive fluid velocity at the primary nozzle exit is defined by the Eq. 4.

$$u_4 = \sqrt{2\left(h_3 - h_4\right)} \tag{4}$$

Just as for the primary nozzle, equations are obtained for the secondary nozzle, according to Eq. 5, 6 e 7, where η_{ms} is the efficiency of the suction nozzle.

$$h_{5,is} = f\left(P_6, s_9\right) \tag{5}$$

$$h_5 = h_9 + \eta_{ms}\left(h_{5,is} - h_9\right) \tag{6}$$

$$u_5 = \sqrt{2\left(h_9 - h_5\right)} \tag{7}$$

The fluid enthalpy in the mixing region is determined by the Eq. 8, where r is the ratio of the mass flow rate of the primary nozzle and ejector total mass flow rate, defined by the Eq. 1.

$$h_6 = rh_4 + (1-r)h_5 \tag{8}$$

Equation 9 determines the velocity of the fluid mixture.

$$u_6 = r u_4 + (1-r) u_5 \tag{9}$$

At the exit of the diffuser, the specific enthalpy of the refrigerant, considering an isentropic process is defined by Eq.10:

$$h_{7,is} = h_6 + \frac{1}{2} u_6^2 \tag{10}$$

The fluid pressure at the diffuser outlet is a function of the refrigerant enthalpy, considering an isentropic process and the entropy in the mixing region, according the Eq. 11:

$$P_7 = f\left(h_{7,is}, s_6\right) \tag{11}$$

The enthalpy of the fluid in the ejector output is determined by Eq.12.

$$h_7 = h_6 + \frac{1}{2} \eta_d u_6^2 \tag{12}$$

To a steady state operating condition of the ejector cycle, the exit ejector quality needs to be equal to the value of *r*, as shown in Eq. 13. This condition also establishes the iterative procedure of the model.

$$r = x_7 = f(P_7, h_7 \tag{13}$$

The evaporator capacity is determined by Eq. 14:

$$\dot{Q}_E = \dot{m}_{ms}\left(h_9 - h_{8a}\right) \tag{14}$$

The compressor power is calculated for both the ejector cycle and standard cycle by Eq. 15:

$$\dot{W}_{cp} = \dot{m}_{cp}\left(h_2 - h_1\right) \tag{15}$$

where $\dot{m}_{cp}$ is the compressor mass flow rate. For an ejector cycle is equal to $\dot{m}_{mm}$ and h_2 is the enthalpy at the compressor discharge, for a given efficiency, as defined by Eq. 16.

$$\eta_{cp} = \frac{h_{2,is} - h_1}{h_2 - h_1} \tag{16}$$

Equation 17 determined the COP of the cycle equipped with the ejector.

$$COP_E = \frac{(1-r)\left(h_9 - h_{8a}\right)}{r\left(h_2 - h_1\right)} \tag{17}$$

The calculation of the standard cycle was performed using the energy balance for each component, with the trivial considerations. To performance comparison purposes, were used standard cycle COP, COP_S, determined by Eq. 18 and the COP of the Carnot cycle, COP_C, according Eq. 19:

$$COP_S = \frac{\dot{Q}_E}{\dot{W}} \tag{18}$$

where $\dot{Q}_E$ is the evaporator capacity and $\dot{W}$ is the compressor power.

$$COP_C = \frac{T_9}{T_3 - T_9} \tag{19}$$

The quantitative comparison of the performance of the ejector cycle and standard cycle operating under the same temperature conditions was made using the ratio between the ejector cycle COP and the standard cycle COP, *Rz*, as determined by Eq. 20.

$$Rz = \frac{COP_E}{COP_S} \tag{20}$$

For comparison purposes to results obtained with some literature data, was also used the ratio of the ejector cycle COP and the Carnot cycle COP, Rz_C, determined by Eq. 21.

$$Rz_c = \frac{COP_E}{COP_C} \tag{21}$$

For the solution of the model, we used some simplifying assumptions:

a) The pressure drop in the heat exchangers and piping is not considered;
b) Losses for heat transfer in the pipe are not considered;
c) The cycle is equipped with a liquid/vapor separator and the performance of such component is unitary;
d) The frictional losses of the flow in the ejector are corrected by the efficiency coefficients of each volume control;
e) The flow inside the ejector is one-dimensional and occurs in continuous operation;
f) The process is adiabatic;
g) The mixing of the fluid occurs at constant pressure (Munday and Bagster, 1977).

3. RESULTS OF THERMODYNAMIC MODEL

To validate the computational routines, the results published by Domanski (1995) were used. In his work were analyzed the refrigerants R-717, R-32, R-22, R-13, R-290, R-12, R-134a R-11, R-600a, R-141b, R-123 and R-113a operating in an ejector cycle by applying the model proposed by ejector Kornhouser (1990). In this work, Domanski uses the concept of reduced temperature for both the condenser and evaporator, as defined by Eq. 22 and 23, respectively.

$$T_{c,r} = \frac{T_c}{T_{cr}} \tag{22}$$

$$T_{e,r} = \frac{T_e}{T_{cr}} \tag{23}$$

where T_c and T_e are the saturation temperatures in the condenser and evaporator, respectively, and T_{cr} is the critical temperature of the refrigerant. Thereby, all refrigerants are analyzed in the same temperatures that ensures that they all operate at its best application range, even if it means that the absolute saturation temperatures and temperature difference between condenser and evaporator are different. The values of reduced temperatures used by Domanski (1995) were T_c = 0.82, T_e = 0.65. The resulting saturation temperatures for each analyzed refrigerant are shown in Tab. 1. In this table, T_3 is the condensing temperature and the evaporation temperature is T_9.

Table 1. Data input systems

Refrigerant	R-717	R-32	R-22	R-13	R-290	R-12	R-134a	R-11	R-600a	R-141b	R-123	R-113
T3, °C	59.3	15	29.7	-25.5	30.1	42.5	33.8	113.2	61.3	118.3	101.6	126.6
T9, °C	-9.6	-44.8	-33.1	-76.8	-32.8	-22.9	-29.9	33.1	-8.1	37.1	23.9	4.,7

The pressure in the mixing region, P_6, was kept the same for all refrigerants, corresponding to a saturation temperature T_6 below the evaporation temperature, T_9. In the simulations, it was found that the optimal temperature for most of the refrigerants is between 5 and 10 K below the saturation temperature of the evaporator, T_9, which produces the maximum COP, with differences between them to the second decimal place. In the studies by Lawrence (2012), the temperature of the mixture region was also considered constant, corresponding to 5 K below the T_9 condition used in this study.

The solution method performs an iterative process, by varying the initial value of r in increments of 0.001. The new value of r corresponds to mixture quality in the diffuser output, x_7. The calculation continues until the absolute difference between r values, previous and current, is lower than 0.005. This form of calculation allowed obtaining a convergence in the EES for any fluid saturation temperature. The results of calculations for the refrigerants listed above, considering a unitary efficiency of the ejector and compressor and considering superheating and subcooling equal to zero, are presented in Tab. 2 in accordance with the COP of the ejector cycle.

Table 2. Results of thermodynamic model

Refrigerant	**R-717**	**R-32**	**R-22**	**R-13**	**R-290**	**R-12**	**R-134a**	**R-11**	**R-600a**	**R-141b**	**R-123**	**R-113**
COP_E	3.49	3.56	3.70	3.73	3.74	3.74	3.74	3.71	3.66	3.65	3.64	3.56

These results are consistent with those presented by Domanski (1995). Moreover, it can be seen that the COP improvement varies with each refrigerant. The COP of the ejector cycle, COP_E, is heavily dependent on the efficiency of the ejector components. This statement can be proven with the analysis of Fig. 5 where are represented the ratio values between the performance coefficients (R_{ZC}) for a variation of the efficiencies of the ejector components between 0 and 1.

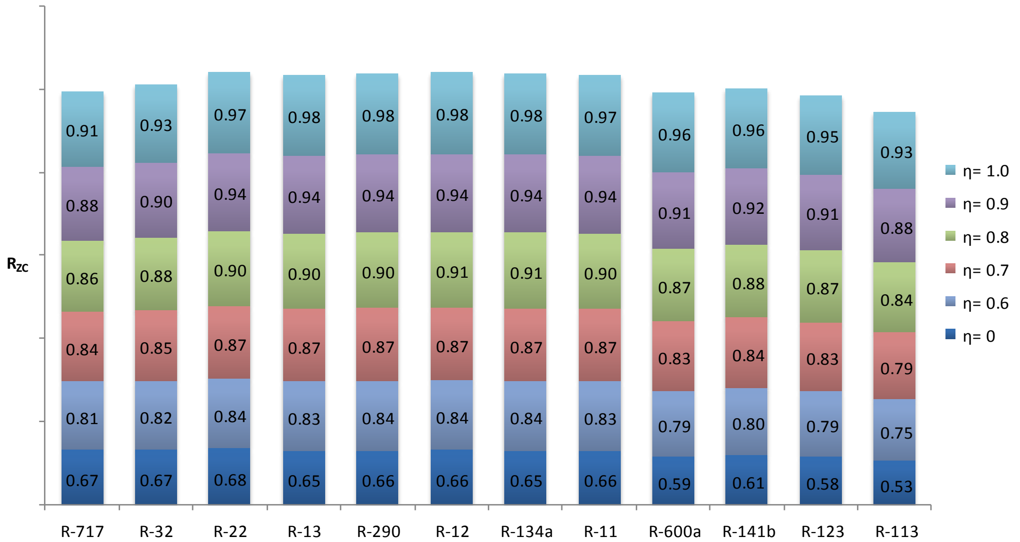

Figure 5. Effect of the ejector efficiency in the relationship between the ejector cycle COP and the Carnot cycle, *Rzc*.

When the efficiencies of the ejector components are equal to zero, the ejector cycle is reduced to the standard cooling cycle. The variation of the ejector cycle performance is more sensitive as the efficiencies of the ejector components increase, which was also discussed by Kornhouser (1990). According to several authors, the efficiencies of the components are in the range 0.7 to 0.8 and is very unlikely to reach values above this range.

3.1 Ejector cycle operating with R-410A and R-134a

The objective of this study is to present an analysis of the impact on the performance coefficient of the ejector cycle, COP_E, operating with two sub-critical refrigerants, R-410A and R-134a. Table 3 shows the parameters used in thermodynamic cycle model.

Table 3. Operating parameters of the ejector cycles.

Data	Ideal case	Real case
T_e [°C]	5	5
T_c [°C]	45	45
Evaporator capacity, $\dot{Q}_E$ [kW]	5	5
Superheating [°C]	0	5
Subcooling [°C]	0	3
η_{mm}	1	0.8
η_{ms}	1	0.8
η_d	1	0.75
η_{cp}	1	0.75

Two different cases were tested. At first, the values of the ejector efficiencies were considered as unitary and the thermodynamic states at condenser and evaporator outlet as liquid and saturated vapor, respectively (ideal case). In the second, were introduced efficiencies in the range estimated in real cycles in addition to considering the superheating at the evaporator exit and subcooling at the condenser exit. The resulting COP_E model is compared to an ideal cycle COP_S operating in the same conditions. The efficiencies adopted to the actual case are recommended by Lawrence and Elbel (2013) and are the result of numerical analysis of flow in ejectors models. In Tab. 4 shows the results of simulations for the refrigerant R-410A.

Table 4. Results by cycle with R-410A

Data	Ideal case	Real case
COP_E	6.735	4.304
COP_S	5.068	3.945
Increment	33%	9.1%
x_7	0.5814	0.5846
$\dot{m}_{mm}$ [kgs^{-1}]	0.03293	0.03184
$\dot{m}_{ms}$ [kgs^{-1}]	0.02454	0.02353
$\dot{m}_{total}$ [kgs^{-1}]	0.05747	0.05537
$\dot{m}_{cp}$ [kgs^{-1}]	0.03293	0.03184

The results show that even for a cycle that simulates the actual conditions, the application of the ejector as an expansion device is able to increase the cycle COP to 9.1%. We can also note that the quality at ejector output remains at similar levels.

Table 5. Results by cycle with R-134a

Data	Ideal case	Real case
COP_E	6.999	4.539
COP_S	5.525	4.295
Increment	26.7%	5.7%
x_7	0.5736	0.5789
$\dot{m}_{mm}$ [kgs^{-1}]	0.03468	0.0343
$\dot{m}_{ms}$ [kgs^{-1}]	0.02681	0.02588
$\dot{m}_{total}$ [kgs^{-1}]	0.06149	0.06018
$\dot{m}_{cp}$ [kgs^{-1}]	0.03468	0.0343

Comparing the increase in the COP in the ideal versus real condition, it is shown the strong dependence of the cycle with respect to the efficiency of the ejector components. Table 5 shows the results of simulations for the refrigerant R-134a, using the same input parameters discussed previously. For R-134a the behavior is similar to R-410A having, however, smaller increments in the gain of the ejector cycle compared to the standard cycle. For the actual case, there was an increase of up to 5.7% in the COP_E compared to COP_S.

In Fig. 6 it can be seen the effect of the condensation temperature (T_3) in COP_E in an ideal condition.

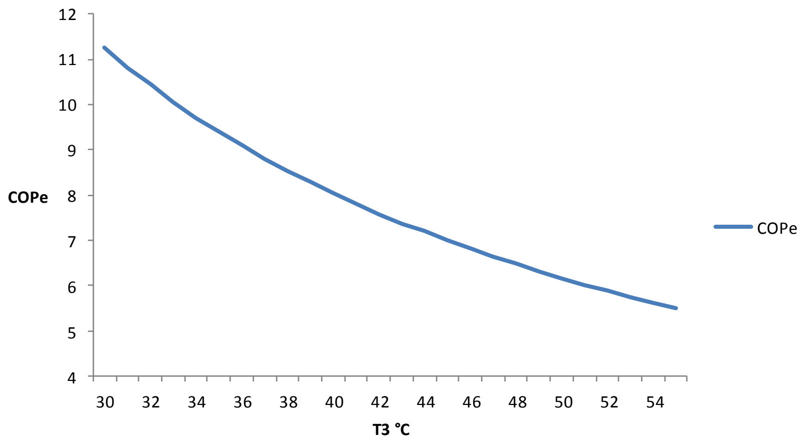

Figure 6. Effect of the condensing temperatures on COP ejector ideal cycle, ranging from 30 °C to 55 °C

As seen in Fig. 6 COP_E undergoes a more abrupt decline in the range of 30 °C to 45 °C tending to a milder reduction after this interval. Figure 7 represents the behavior of the refrigerant mass flow rate in the primary and secondary nozzle ejector as a function of condensing temperature (T_3).

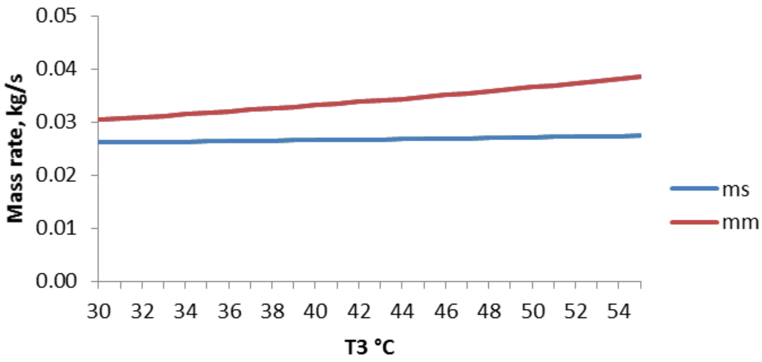

Figure 7. Mass flow rate behavior of primary and secondary refrigerant in the ejector cycle, in ideal condition, for a condensing temperature ranging from 30 °C to 55 °C.

The existence of a suction accumulator in the ejector cycle is justified for the linearity of the mass flow of the secondary fluid even with the increase of the condensing temperature. The mass flow rate of the primary fluid, in turn, is increased due to the increase in fluid pressure at the primary inlet of the ejector nozzle.

4. CONCLUSIONS

This work presents a basic review of the use of ejectors in two-phase cooling emphasizing the recovery of losses during the expansion process. They were summarized the most relevant research and presents the results obtained in some practical and theoretical studies, showing the energy gain opportunity with the implementation of the ejector in the vapor compression cycle.

An ejector cycle simulation model for subcritical fluids was presented. The model was based on dimensional analysis proposed by Kornhauser (1990) and the model solution was developed using the software EES - Engineering Equation Solver. The routine calculations were constructed for and iterative numerical solution in order to determine the optimal operating point cycle. To prove the model and programs, the system was simulated for some of the fluids studied by Domanski (1995) and the results compared. The model showed adequate adherence to the results, and therefore considered reliable in application performance prediction cycles with ejector.

The modeling of the ejector has been resolved for two subcritical refrigerants with large-scale application in refrigeration industry in recent years. The fluids selected were R-410A and R-134a. With the simulation results, it is apparent that there is a high possibility of improvement vapor compression refrigeration cycle with the use of an ejector, reached up increments of up to 9.1% and 5.7% for fluid R -410A and R-134a respectively, in a simulation of a cycle close to the real condition conditions.

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