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# Exergy analysis of operation of two-phase ejector in compression refrigeration systems

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**Abstract** Paper deals with theoretical analysis of possible efficiency increase of compression refrigeration cycles by means of application of a two-phase ejector. Application of the two phase ejector in subcritical refrigeration system as a booster compressor is discussed in the paper. Results of exergy analysis of the system operating with various working fluids for various operating conditions have been shown. Analysis showed possible exergy efficiency increase of refrigeration compression cycle.

**Keywords:** Two-phase ejector; Compression refrigeration system; COP; Exergy

## Nomenclature

| | | |
|---|---|---|
| $b$ | – | specific exergy, J/kg |
| $\dot{B}$ | – | exergy rate, W |
| $h$ | – | specific enthalpy, J/kg |
| $h_{fg}$ | – | specific enthalpy of vaporisation, J/kg |
| $\dot{m}$ | – | mass flow rate, kg/s |
| $p$ | – | pressure, Pa |
| $P$ | – | driving power, W |

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| | | |
|---|---|---|
| $\dot{Q}$ | – | heat flux, W |
| $\dot{Q}_k$ | – | condenser thermal capacity, W |
| $\dot{Q}_o$ | – | refrigeration capacity, W |
| $s$ | – | specific entropy, J/(kg K) |
| $s_{fg}$ | – | specific entropy of evaporation, $s_{fg} = s'' - s'$, J/(kg K) |
| $t$ | – | temperature, °C |
| $T$ | – | temperature, K |
| $x$ | – | two-phase quality |

**Greek symbols**

| | | |
|---|---|---|
| $\eta$ | – | efficiency |
| $\eta_b$ | – | exergy efficiency |

**Subscripts**

| | | |
|---|---|---|
| $a$ | – | ambient |
| $c$ | – | compression cycle |
| $d$ | – | throttling valve |
| $e$ | – | ejector-compression cycle |
| $h$ | – | isenthalpic process |
| $i$ | – | internal |
| $k$ | – | condensation |
| $m$ | – | mechanical |
| $o$ | – | evaporation |
| $p$ | – | compressor |
| $s$ | – | isentropic process |
| $t$ | – | theoretical |
| $Q$ | – | heat transfer |
| $Zo$ | – | external heat source |
| $'$ | – | saturated liquid |
| $''$ | – | saturated vapour |

# 1 Introduction

High differences between condensation and evaporation pressures in refrigeration systems, incured by the throttling losses in the expansion device, can be considered as very high. This causes the low efficiency of these systems. Throttling loss may be effectively reduced by means of application of process that is close to the isentropic expansion. Best possibility for this purpose is the application of the simple expansion devices instead of throttling valves. One of the solutions is the application of two-phase ejector. In ejector the liquid expands in the motive nozzle and therefore use of the momentum of expanded fluid to compress the secondary fluid is possible. For this purpose the two-phase ejectors may be applied.

Investigations concerning the modeling and testing of systems operating under subcritical and supercritical heat rejection conditions with the application of a two-phase ejector are carried out by a limited number of research teams, although the number of publications in this area is steadily increasing. Domanski [1] demonstrated theoretical possibilities for application of the two-phase ejector as a booster compressor for various fluids. Similar analysis was presented by Harrell and Kornhauser [2] for the exemplary case of refrigeration system. Butrymowicz proposed the approach for modeling of the combined ejector-compression cycles for prediction of the capacity of the system [3,4]. In this case the separate analysis is made for the cycle and ejector. This approach allows to use more complicated model of ejector or to apply experimental ejector performance data for determination of the operating conditions of the compression-ejection system. Based on this approach the performance of the two-phase ejector was studied experimentally [4,5]. Dudar and Butrymowicz [6,7] applied this approach for the case of the system operating with hydrocarbons as refrigerants.

Li and Groll [8] presented numerical analysis of the two-phase ejector operation and verified the results obtained experimentally for the specified case of the geometry of the two-phase ejector applied, Liu and Groll [9]. The authors proposed theoretical model of supercritical two-phase ejector as first step compressor in supercritical carbon dioxide ($CO_2$) refrigeration cycle. In discussed compression-ejector cycles the ejector operates between evaporation and interstage pressure, then the mechanical compressor operates between interstage pressure and condensation pressure. Similar analysis and experiments are carried out by Elbel and Hrnjak [10–12]. They analysed supercritical $CO_2$ refrigeration cycle with the two-phase ejector and performed experimental tests. They applied additional interstage heat exchanger to subcool the liquid refrigerant before it feeds the motive nozzle of the ejector. They reached the coefficient of performance (COP) at an average of about 10–11% in comparison with the classical system with throttling valve. Another study with supercritical $CO_2$ refrigeration cycle with ejector as first step compressor was made by Deng *et al.* [13]. They compared the ejector expansion cycle performance with internal heat exchanger system and with a conventional vapour compression system. Also the improvement of the COP of the cycle by means of two-phase ejector was demonstrated experimentally by Disawas and Wongwises [14].

Angielczyk *et al.* [15] investigated operation of the motive nozzle for this type of ejector for the case of carbon dioxide showing good accuracy between experimental and theoretical results. Recently Banasiak and Hafner [16] formulated a complete 1D model of two-phase ejector for the case of carbon dioxide. This model was evaluated on the basis of the experimental data. A lumped parameter model of two-phase ejector based on the systematic experimental data for isobutane was proposed by Dudar [17].

The purpose of the present paper is to formulate exergy analysis of operation of two-phase ejector as a booster compressor in compression systems for various working fluids that shows the possible cycle efficiency enhancement. The schematic diagram of the ejector-compression refrigeration system is presented in Fig. 1 and the thermodynamic cycle of this device in a simple form is shown in Fig. 2.

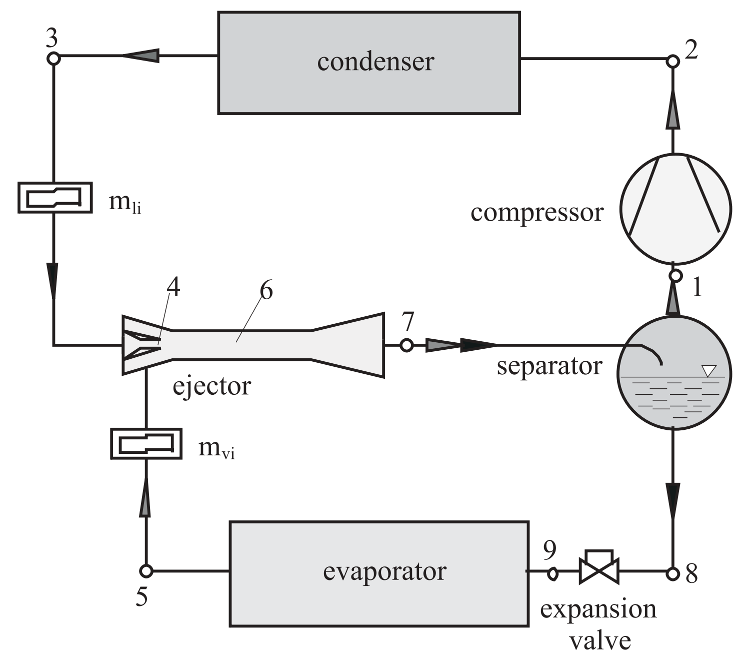


Figure 1. Compression refrigeration system with ejector as a booster compressor.

The ejector is located between evaporator and liquid separator. Liquid refrigerant (3) (Fig. 1) as the motive fluid expands in the motive nozzle and partly evaporates. In isentropic nozzle the expansion takes place along isentropic line 3-4s (see Fig. 2), while the throttling process between pressure $p_k$ and pressure $p_o$ is represented by the line 3-4d. The real motive nozzle expansion is represented by line 3-4. Motive fluid of high velocity learing the nozzle (4) causes suction of vapour from the evaporator (5) as well as

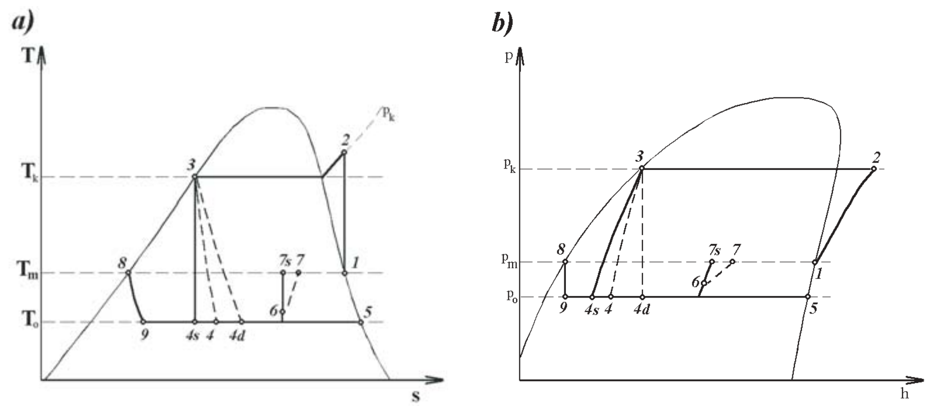


Figure 2. Thermodynamic cycles of efector-compression system: a) at $T$-$s$ coordinates, b) at $p$-$h$ coordinates.

compression in the mixing chamber (6) due to the momentum transfer. The additional compression of the two-phase mixture takes place in the diffuser producing the interstage pressure in the system (7). Compressed two-phase mixture is separated in the separator so the compressor sucks the vapour (1) while the liquid (8) flows to the evaporator through the throttling valve. In this case the compressor is fed by vapour at the interstage pressure. Therefore the compressor operates with lower pressure difference than in classic compression refrigeration systems. This lower pressure differences is related directly to the possible increase of the system efficiency.

## 2 Exergy analysis of the ejector-compression refrigeration system

The exergy losses can be considered as a basic parameter of thermodynamic imperfection of the system. In this analysis the exergy losses in particular components of the system are calculated and every component such as mechanical compressor, heat exchangers and throttling valve are treated as a separate open systems [18]. Thermodynamic processes occurring in the specific components of the refrigeration system are often analysed in pressure – specific enthalpy diagrams. However, in the case of the exergy analysis the cycle in the exergy – specific enthalpy diagram will be more useful, see Fig. 3. The exergy analysis of the ejector-compression refrigeration system was proposed.

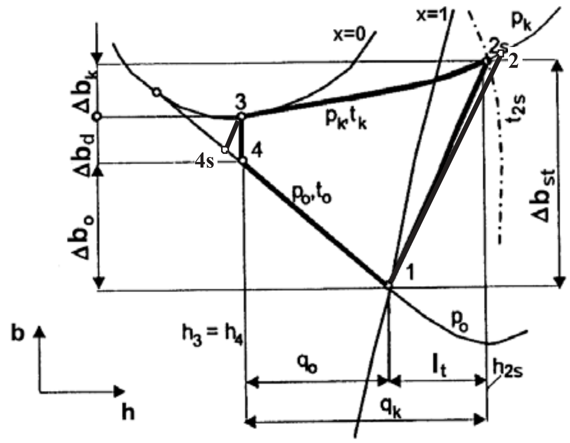


Figure 3. Refrigeration compression cycle in exergy – specific enthalpy ($b$-$h$) coordinates [18].

Theoretical compression of vapour is represented by line 1-2s in Fig. 3. Line 1-2 represents the compression in real operating conditions. The specific exergy at compressor inlet can be calculated as

$$b_{1c} = h_2 - h_a - T_a\left[s''(p_o) - s_a\right] , \tag{1}$$

while the specific exergy of isentropically compressed vapour is

$$b_{2sc} = h_{2s} - h_a - T_a(s_{2s} - s_a) . \tag{2}$$

For the actual process of compression the exergy of compressed vapour can be calculated as

$$b_{2c} = h_2 - h_a - T_a(s_2 - s_a) . \tag{3}$$

The mechanical compressor consumes the effective power $P_o$. However, for the working fluid only part of the effective power is delivered, namely the indicative power $P_i$. The difference between effective power and indicative power is the rate of the external exergy losses:

$$\Delta\dot{B}_{cop} = (1 - \eta_{mc})\, P_o , \tag{4}$$

where $\eta_{mc}$ is the mechanical compression efficiency that covers all of the motive energy losses including the efficiency of the motor of the compressor. The rate of the internal exergy losses is

$$\Delta\dot{B}_{cip} = \eta_{mc}\,(1 - \eta_{ic})\, P_o . \tag{5}$$

Process of vaporization in the evaporator is represented by line 4-1. It was assumed that temperature difference between evaporation and cooled space in the evaporator $\Delta T_{HT} = 5$ K. The average temperature of the cooling medium is $T_c$ and the temperature of evaporation of the refrigerant is $T_o$. The specific exergy at the outlet of the throttling valve is calculated as follows:

$$b_{4c} = h_3 - h_a - T_a\left(s_{4hc} - s_a\right) \ . \tag{6}$$

Quality and specific entropy of the vapour at state 4 are respectively

$$x_{4hc} = \frac{h_3 - h'\left(p_o\right)}{h_{fg}\left(p_o\right)} \ , \tag{7}$$

and

$$s_{4hc} = s'\left(p_o\right) + x_{4hc} s_{fg}\left(p_o\right) \ . \tag{8}$$

The rate of the exergy change $\Delta \dot{B}_{Zo}$ is equal to the increase of the rate of exergy of the external heat source. Therefore $\Delta \dot{B}_{Zo}$ can be calculated as follows:

$$\Delta \dot{B}_{c_{Zo}} = -\left(1 - \frac{T_a}{T_a + \Delta T_{HT}}\right)\dot{Q}_o \ . \tag{9}$$

The rate of exergy change during heat transfer in the evaporator is

$$\Delta \dot{B}_{cQo} = \dot{m}_o\left(b_{4c} - b_{1c}\right) - \Delta \dot{B}_{cZo} \ . \tag{10}$$

In the condenser heat is transferred from the refrigerant to the ambient. This process is represented in Fig. 3 by line $2-3$. Specific exergy at the condenser outlet can be calculated as

$$b_{3c} = h_3 - h_a - T_a(s_3 - s_a) \ . \tag{11}$$

The rate of exergy change in the condenser is

$$\Delta \dot{B}_{cQk} = m_o\left(b_{2sc} - b_{3c}\right) \ . \tag{12}$$

The heat capacity of condenser can be calculated from energy balance of the system

$$\dot{Q}_{kc} = P_c \eta_{mc} + \dot{Q}_o \ . \tag{13}$$

The exergy losses during condensation are a result of irreversibility of the heat transfer processes. If the heat transfer is reversible then the rate of the exergy loss will be $\Delta \dot{B}_{oQk} = 0$. Therefore $b_{2s} = b_3$, which means that the process of condensation occurs at ambient temperature $T_{ot.}$ It should be noted that the exergy losses depend also on the thermodynamic state of refrigerant at the heat exchanger inlet [18].

The rate of internal exergy losses that occur in the throttling valve can be calculated as

$$\Delta \dot{B}_{cid} = \dot{m}_o \left( b_{3c} - b_{4c} \right) \ . \tag{14}$$

With the help of Eq. (9), the exergy efficiency may be calculated as follows:

$$\eta_{bc} = \frac{\Delta \dot{B}_{cZo}}{P_c} \ , \tag{15}$$

since rate of the exergy change of cooled space may be thought as useful exergetic effect of the cycle treated as closed system, while effective driving power is equal to the consumption of the driving exergy.

## 3 Exergy analysis of the ejector-compression refrigeration cycle

In exergy analysis for ejector-compression system the additional components, i.e., the ejector and the vapour-liquid separator should be taken into consideration. In presented analysis it was assumed that refrigeration system operates according to Fig. 1, therefore the nomenclature for the characteristic points of the system correspond to this scheme.

The specific exergy of vapour at compressor inlet is

$$b_{1e} = h_2 - h_a - T_a \left[ s'' \left( p_m \right) - s_a \right] \ , \tag{16}$$

while the specific exergy of the isentropically compressed vapour is

$$b_{2se} = h_{2s} - h_a - T_a (s_{2s} - s_a) \ . \tag{17}$$

For the actual compression process the specific exergy of compressed vapour can be calculated as

$$b_{2e} = h_2 - h_a - T_a (s_2 - s_a) \ , \tag{18}$$

and the specific exergy at point 3 is

$$b_{3e} = h_3 - h_a - T_a (s_3 - s_a) \ . \tag{19}$$

The specific exergy at the motive nozzle outlet after isentropic expansion is

$$b_{4se} = h_{4s} - h_a - T_a(s_3 - s_a) \ , \tag{20}$$

while after the actual expansion it yields:

$$b_{4e} = h_4 - h_a - T_a(s_4 - s_a) \ . \tag{21}$$

Specific exergy at the inlet of the suction chamber of the ejector is equal to specific exergy at the outlet of the evaporator

$$b_{5e} = h''(p_o) - h_a - T_a\left[s''(p_o) - s_a\right] \ . \tag{22}$$

In the mixing chamber (point 6) and at the ejector outlet (points 7 and $7s$, where $s$ is for isentropic condition) respectively, the specific exergy can be calculated as

$$b_{6e} = h_6 - h_a - T_a(s_6 - s_a) \ . \tag{23}$$

$$b_{7se} = h_{7s} - h_a - T_a\left(s_6 - s_a\right) \ , \tag{24}$$

$$b_{7e} = h_7 - h_a - T_a(s_7 - s_a) \ . \tag{25}$$

Now, the throttling valve can by analysed. The specific exergy at inlet to the throttling valve is

$$b_{8e} = h'(p_m) - h_a - T_a\left[s'(p_m) - s_a\right] , \tag{26}$$

and at the outlet of the throttling valve it yields:

$$b_{9e} = h_9 - h_a - T_a(s_9 - s_a) \ . \tag{27}$$

Once the specific exergy in characteristic points of the system are found then the exergy losses for the particular components can be calculated. The external and internal exergy losses for the compressor are, respectively

$$\Delta B_{eop} = (1 - \eta_{mc})P_e \ , \tag{28}$$

$$\Delta B_{eip} = \eta_{mc}\left(1 - \eta_{ic}\right)P_e \ . \tag{29}$$

Rate of the exergy losses in the evaporator is given by

$$\Delta\dot{B}_{eZo} = -\left(1 - \frac{T_a}{T_o + \Delta T_{HT}}\right)\dot{Q}_o \ . \tag{30}$$

The rate of the exergy losses in the evaporator is

$$\Delta \dot{B}_{eQo} = \left[ \frac{T_a}{T_o} - \frac{T_a}{T_o + \Delta T_{HT}} \right] \dot{Q}_o \ . \tag{31}$$

The rate of the exergy losses in the condenser is

$$\Delta \dot{B}_{eQk} = \dot{m}_e \left( b_{2s} - b_3 \right) \ , \tag{32}$$

and the rate of the exergy losses in the throttling valve can be calculated as

$$\Delta \dot{B}_{eid} = \dot{m}_{oe} (b_8 - b_9) \ . \tag{33}$$

Further, rate of the exergy losses in the ejector can be calculated as

$$\Delta \dot{B}_{eiE} = \dot{m}_e b_3 + \dot{m}_{oe} b_5 - (\dot{m}_e + \dot{m}_{oe}) b_7 \ , \tag{34}$$

and the rate of the exergy losses in the vapour-liquid separator

$$\Delta \dot{B}_{eiS} = (\dot{m}_e + \dot{m}_{oe}) b_7 - \dot{m}_e b_1 - \dot{m}_{oe} b_8 \ . \tag{35}$$

Finally, the exergy efficiency of the ejector-compression cycle is given by

$$\eta_{be} = \frac{\Delta \dot{B}_{eZo}}{P_e} \ . \tag{36}$$

## 4 Analysis of ejector-compression cycle for various refrigerants

Analysis of the exergy losses that occur in ejector-compression system is presented for several working fluids. The calculations were made with application of NIST Standard Database for thermodynamic properties of analysed working fluids [19]. Results of calculation for two chosen cases are shown. The motive nozzle efficiency and compression produced by the ejector and mechanical compressor for given temperature of condensation $t_k = +35$ °C were calculated. The results are presented in Fig. 4.

The relative exergy losses presented in Fig. 4 demonstrate that the condenser, the evaporator, and the compressor may be thought of as the sources of the largest share of the exergy losses of the system for most of the analysed fluids. With decreasing of the evaporation temperature the exergy losses produced by the compressor are increasing which is particularly

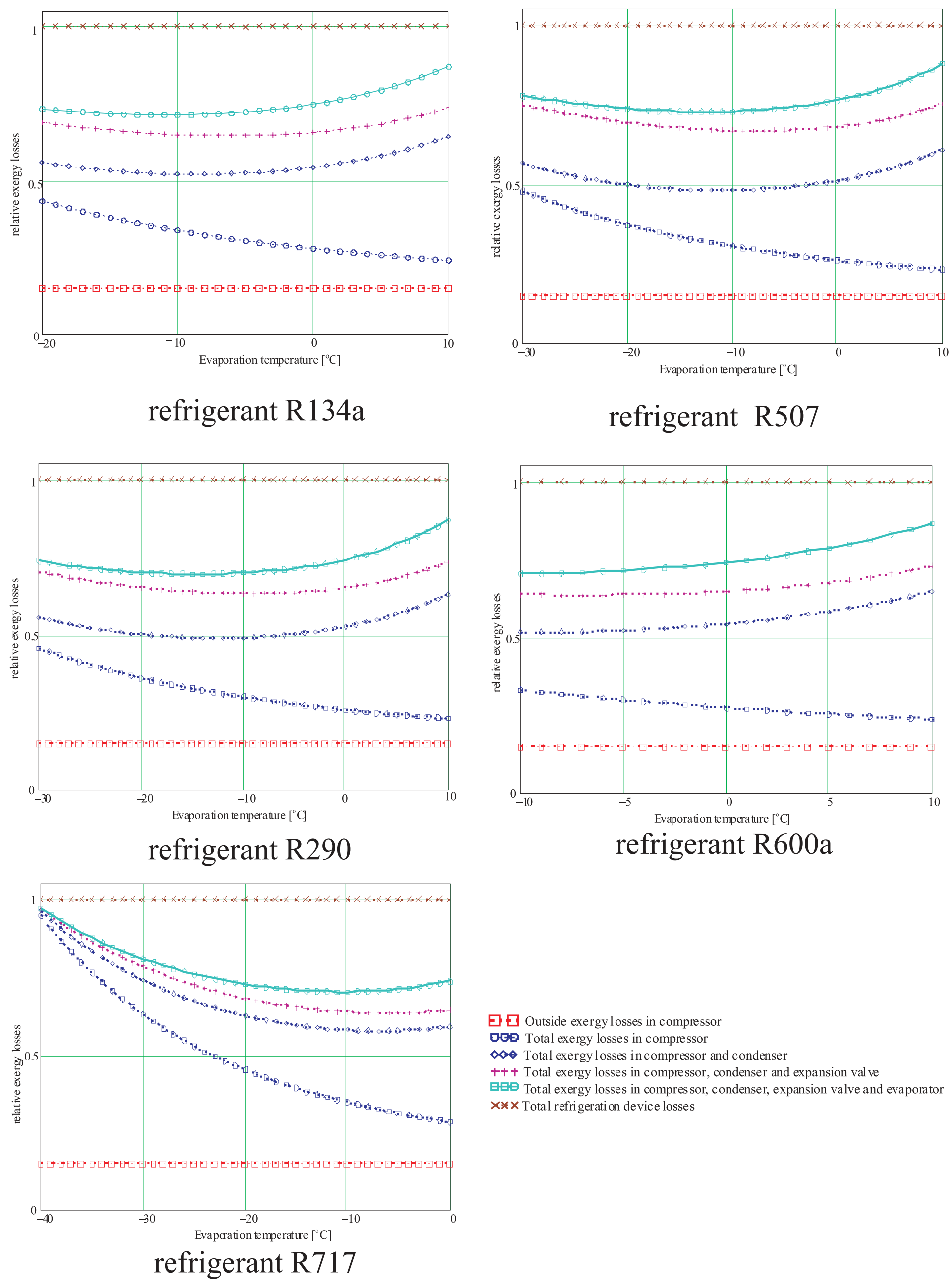


Figure 4. Relative exergy losses of compression refrigeration cycle vs. temperature of evaporation at condensation temperature $t_k = 35$ °C for various working fluids.

demonstrated for the case of ammonia R717. When temperature of evaporation $t_o > 0^o$C then the largest share of the exergy losses is produced by the condenser and the lowest share of the exergy losses is produced by the throttling valve. In the case of relation of the exergy losses as a function of condensation temperature the opposite trend can be observed. Increase of condensation temperature results in the increase of exergy losses produced by mechanical compressor. For most of the analysed refrigerants, the exergy losses that occur in mechanical compressor exceed 50% of the total losses for the same reference temperature. The smallest share of the exergy losses is produced by the evaporator.

As it is seen in Fig. 5 that the benefit of the application of the two-phase ejector is reduction of the exergy losses produced by mechanical compressor. For example, for the case of ammonia (R717) the relative exergy losses produced by mechanical compressor are reduced from 95% of total exergy loses to approximately 80%. For the other analysed fluids the similar trend could be observed. However, reduction of the relative exergy losses is much lower than for ammonia.

The smallest part of the exergy losses in the ejector-compression cycle is produced by the throttling valve. For all of the analysed fluids, the exergy losses occurring in the throttling valve are close to zero. This is the important benefit due to application of the ejector. With increase of the evaporation temperature the exergy losses produced by the condenser in reference to the total exergy losses also increase with the decrease of losses produced by the evaporator. The exergy losses produced by the two-phase ejector do not exceeds few percents.

As it can be seen from the results presented in Fig. 6, the application of the two-phase ejector reduces the exergy losses produced by the compressor and also minimises the losses in the throttling valve. Thus application of two-phase ejector may be thought as profitable. The increase of the exergy efficiency of the ejector-compression system in comparison with classic compression systems could be observed especially for lower temperatures of evaporation. For higher temperatures, i.e., $t_o > +10\,^\circ$C the benefit due to application of two-phase ejector in increasing of the exergy efficiency is not significant. With increase of condensation temperature the exergy efficiency decreases. However, the ejector-compression system is still more profitable than classic compression refrigeration system.

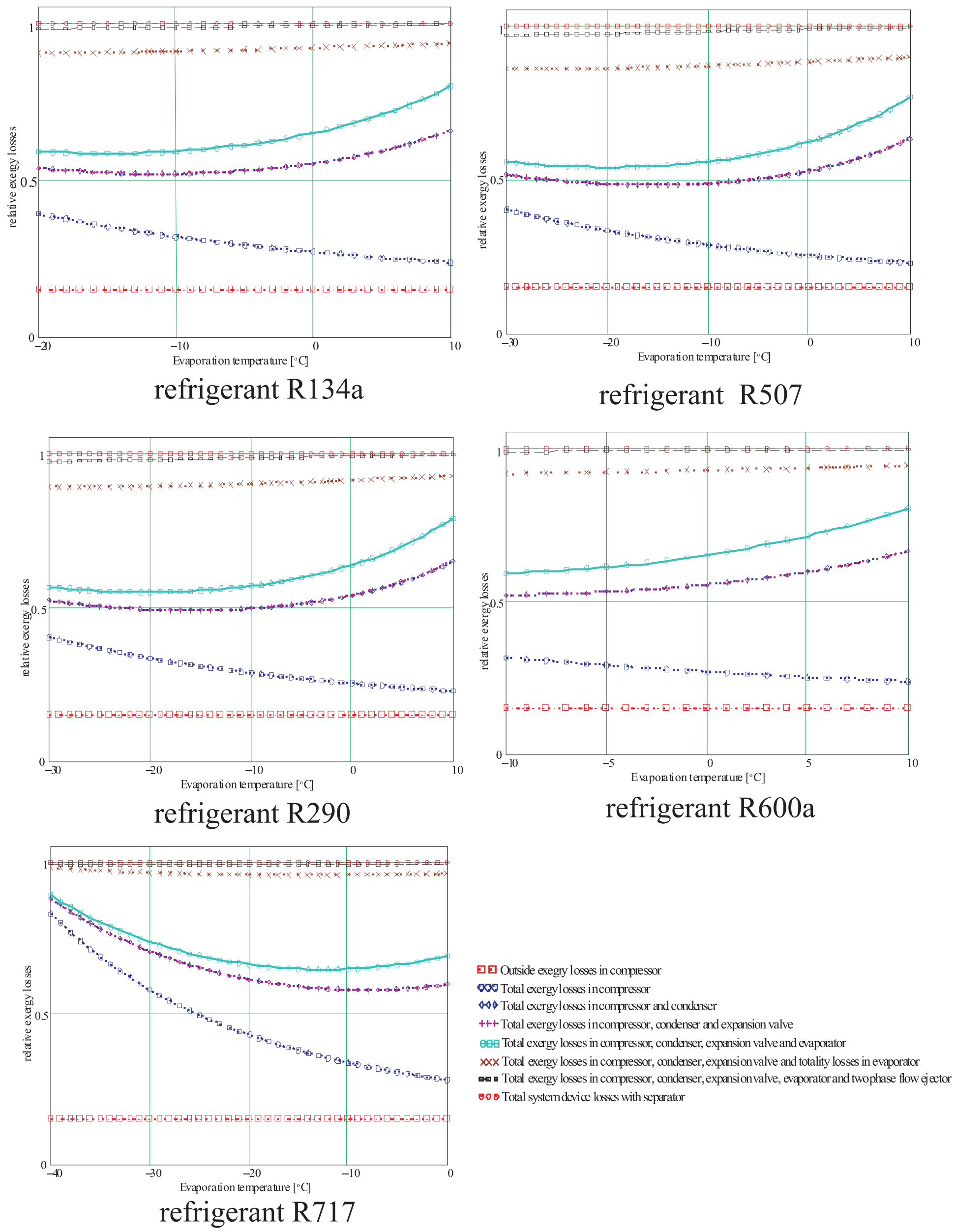


Figure 5. Relative exergy losses of ejector-compression refrigeration cycle vs. temperature of evaporation at condensation temperature $t_k = 35$ °C for various working fluids.

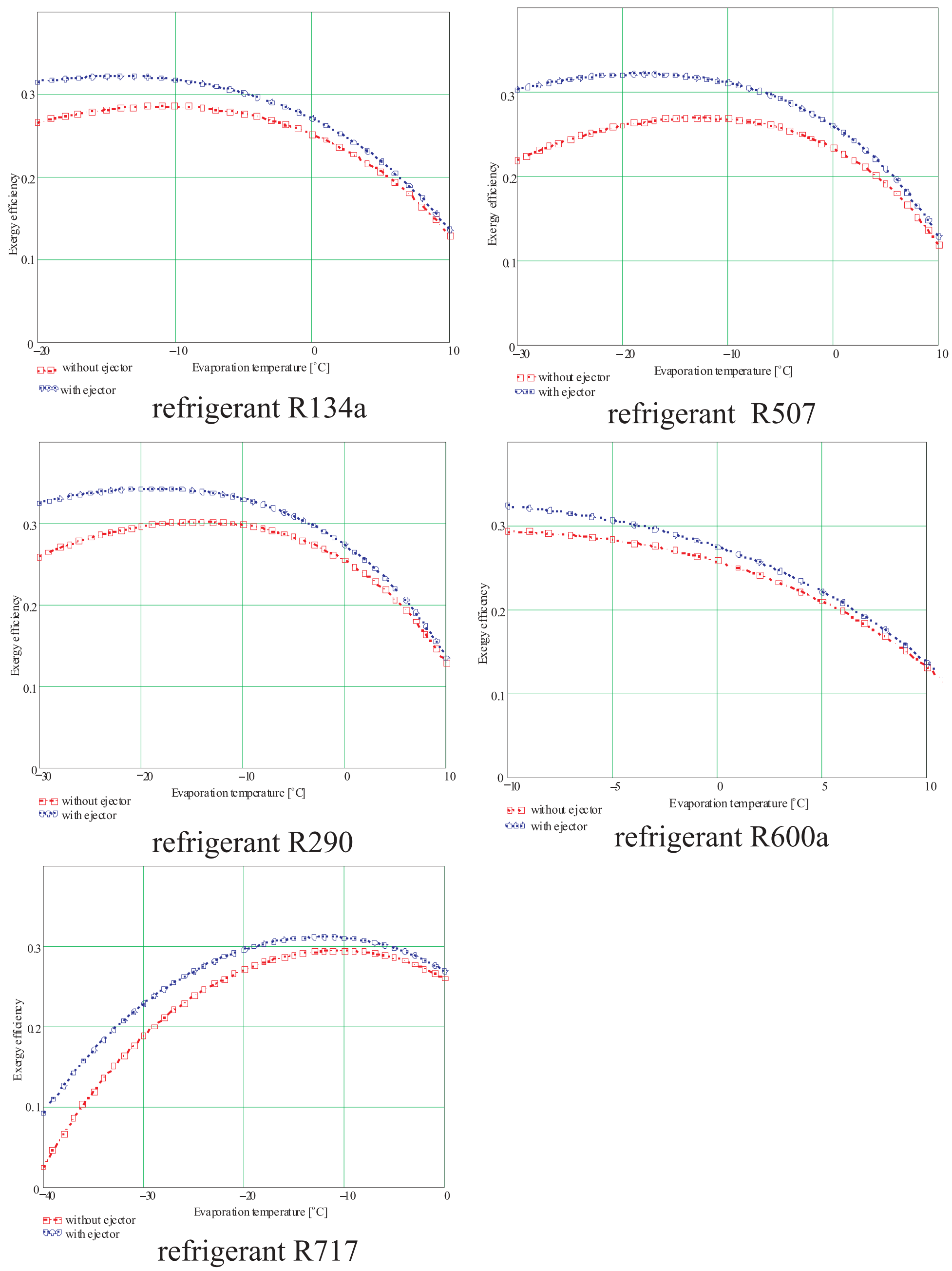


Figure 6. Exergy efficiency vs. temperature of evaporation of ejector-compression and compression refrigeration cycles at condensation temperature $t_k = 35$ °C for various working fluids.

## 5 Summary

On the basis of presented results it could be concluded that application of the two-phase ejector may be thought as attractive method for improvement of the efficiency of the compression refrigeration system. Moreover, it was demonstrated that even for low-pressure refrigerants, e.g. isobutane, the improvement of the efficiency of the compression refrigeration cycle by means of the two-phase ejector is profitable due to almost complete reduction of the exergy losses occurring in the throttling valve and diminishing the losses produced in the compressor.

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