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IMPROVEMENT OF REFRIGERATION CYCLE EFFICIENCY BY MEANS OF TWO-PHASE EJECTOR AS SECOND STAGE COMPRESSOR

M. Bergander[1,3*] D. Butrymowicz,[2] J. Karwacki[2]

[1] University of Hartford, 200 Bloomfield Avenue, West Hartford, CT 06117, USA
[2] Institute of Fluid Flow Machinery Polish Academy of Sciences, Gdansk, 80231 Poland,
[3] AGH University of Science & Technology, Cracow, Poland

ABSTRACT

Paper deals with analysis of application of two-phase ejector as a second stage compressor in refrigeration compression systems. The thermodynamic reason for the efficiency improvement due to application of the ejector is effect of difference in compression work for liquid and vapour phases. The own approach of the calculation of the compression-ejector refrigeration cycle has been proposed in the paper. This approach is based on the set of two performance characteristics: one for the ejector and the second one for the rest of the refrigeration installation. On the basis of the above approach the limiting conditions of operation of the systems have been established for selected refrigerants. It has been showed that possible increase of coefficient of performance of the system strongly depends on the entrainment ratio of the two-phase ejector.

INTRODUCTION

The paper deals with aspects of application of two-phase ejector as a second stage compressor in refrigeration compression systems. The schematic of the investigated system is presented in Fig. 1. Vapour compressed in the mechanical compressor is sucked by the ejector. The motive fluid in the ejector is liquid delivered by a mechanical pump. The presented configuration is totally novel approach for improving of the efficiency of the refrigeration systems by means of two-phase liquid-vapour ejector. This configuration was proposed and patented by Bergander [1].

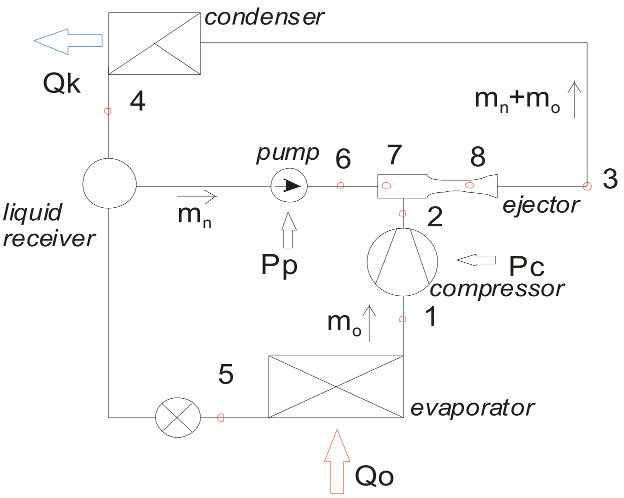

Fig. 1. Schematic of refrigeration compression system with two-phase ejector as a second stage compressor.

The thermodynamic reason for the efficiency improvement is effect of difference in compression work for liquid and vapour phases. Therefore increasing the efficiency of the standard single-stage vapour compression cycle is caused through a reduction of mechanical compression at the expense of harnessing kinetic energy of vapour in the ejector device.

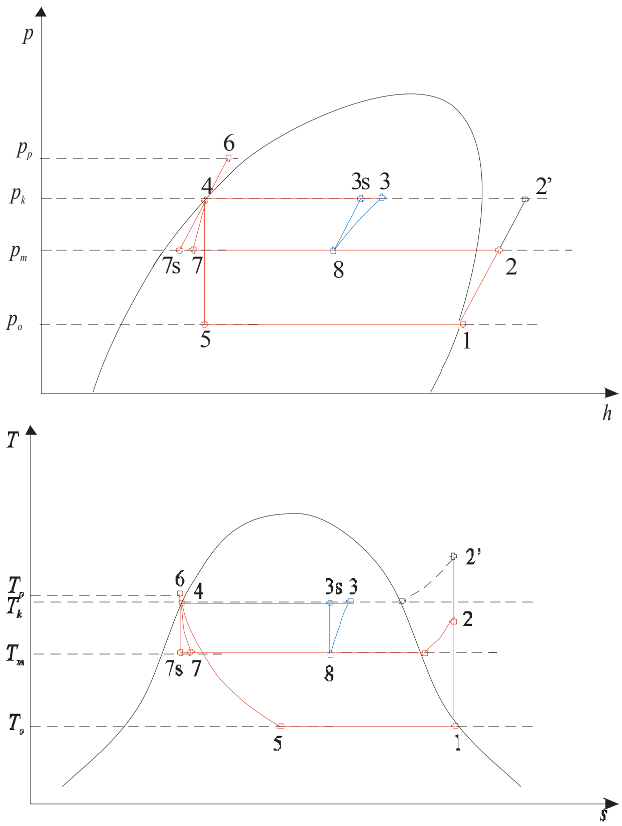

Fig. 2. Compression-ejection cycle of the system presented in Fig. 1.

The compression-ejection cycles in p-h and T-s diagrams are presented in Fig. 2. The suction point 1 is located at the saturation line. However, there is no obstacles to take into consideration also superheating of the sucked vapour in the analysis. The isentropic compression process in the mechanical compressor was assumed in these charts. However, the internal efficiency of the mechanical compressor has been included in the analysis presented in this paper. The only reason the isentropic process was chosen is the problem with the appropriate representation of the discharge parameters from real mechanical compressors in refrigeration systems. Moreover assumption of the isentropic compression process in the mechanical compressor is the common approach made in the calculations of the compression refrigeration systems. Important feature of the system is two additional pressures, i.e. inter-stage pressure pm and corresponding inter-stage saturation temperature Tm, as well as pump motive pressure pp and liquid phase temperature Tp. Therefore the discharge of the mechanical compressor in the discussed system is represented by point 2 while in classic one-stage system the discharge will be located at the point 2'.

The next assumption made in Fig. 2 deals with liquid parameters at the outlet of the condenser. No subcooling was taken into account in these charts. Therefore liquid state is represented by point 4 located at the liquid saturation line. Nevertheless the liquid subcooling may be taken into account in the further analysis. Isentropic process of liquid compression was assumed in Fig. 2. If the expansion process of the liquid phase occurs isentropically then the outlet of the ejector motive nozzle is represented by the point 7s. Taken into account efficiency of the motive nozzle – real outlet is located at the point 7. In most practical cases wet vapour may be expected at the outlet of the motive nozzle as the flashing process will occur. The quality of the vapour emanating from the motive nozzle depends on the nozzle efficiency as well as possible liquid subcooling for the given operation pressures in the system. However, since the motive pump pressure pp does not depend on the rest of the operation parameters it may be also assumed that the effect of this pressure on the quality of the fluid at the outlet of the motive nozzle may be also discussed.

A good representation of the operation of the ejector at the thermodynamic charts is by no means a simple task as a combination of compression and mixing processes occur at the mixing chamber simultaneously. Therefore processes occurring inside the mixing chamber of the ejector may be presented by assumed representation process lines. In Fig. 2 common approach was applied: the mixing process and compression process due to momentum exchange were separated. Moreover, the mixing process is assumed as the isobaric process of the discharged vapour from mechanical compressor 2 and fluid expanded in the motive nozzle 7. Therefore the mixing process 7-8-2 was assumed in Fig. 2. Compression process due to momentum and energy transfer between motive fluid and secondary fluid (vapour discharged form mechanical compressor) is represented by compression line 8-3s for the ideal case of isentropic compression and 8-3 for real compression in the mixing chamber and the diffuser. At this stage of research the possible discussion on share of the compression process between mixing chamber (due to momentum and energy transfer between motive and secondary streams) and the diffuser (due to increasing of cross-section area) is not possible, and moreover there is no clear need for it. Location of point 8 as well as point 3 depends on the entrainment ratio of the ejector which is not visible in the thermodynamic charts.

It is important to note that there will be temperature difference between motive fluid temperature Tm and secondary fluid temperature T2. This temperature difference may strongly affect on the operation of the ejector changing the entrainment ratio and compression efficiency. However, the thermodynamic charts do not present any information on the mass rates for the discussed processes and this effect may be analysed during preparation of the model of the ejector prepared on the basis of the relevant experimental tests.

Condensation process is represented by line 3-4. It means that wet vapour enters the condenser. Liquid phase flows to the receiver and then is delivered to pump to motive the ejector and main part of the liquid is delivered to the expansion valve feeding the evaporator. The operation of the evaporator (process line 5-1) should not be affected by reorganisation of the system due to two-phase ejector. The saturated vapour at the evaporator outlet was assumed. However, in the case of feeding of the evaporator by means of the thermostatic expansion valve superheated vapour enters the mechanical compressor. This may be included in the analysis without any obstacles.

It is worth to note that the discussed configuration of the system is in some manner similar to the case of the so-called flash booster double-stage system. Therefore operation of the ejector may be also interpreted as very effective inter-stage cooling. Also, instead of the flashing valve where isenthalpic throttling process occurs the fluid expansion in the motive nozzle is applied in the discussed ejection system. The above two reasons lead to expectation of significant increase in COP of the discussed system in comparison with classic solutions. There is also the second similarity with existing compression systems, i.e. forced induction (supercharging) quasi double-stage system used for the screw compressors. The similar solution developed by Voorheese [4] in 1930s for piston compressors has not been wildly applied in refrigeration. In this case a part of liquid is injected to the compressor during compression process in order to cool vapour and diminish power consumption demand for compression process. The limitation of the benefits of this solution is attributed to the limitation of the injected liquid to the screw compressor. Since in the discussed ejection system there is no such limitation therefore some further increase of COP may be expected on the condition that power consumption for pumping of liquid phase will not overcome the benefits.

COMPRESSION-EJECTION CYCLE ANALYSIS

The general purpose of this paper is to provide with rational methodology based on which it is possible to assess possible improvement of the COP due to application of the second stage compression in two-phase ejector in compression refrigeration system. Moreover, there is a clear need to apply such methodology in order to establish the range of the operation parameters of the refrigeration system in order to establish a preliminary geometry of the two-phase ejector and the pump performance.

The proposed approach is based on the set of two performance characteristics: one for the ejector and the second one for the rest of the refrigeration system, as it was depicted in Fig. 3. The operating parameters of the whole system can be determined then by the intersection of these two characteristic lines. This is the very similar method as in the case of the well known calculation procedure of the pressure rise and liquid flow rate of the pump system. The similar approach has been applied by the authors for the case of two-phase ejector as a booster compressor in refrigeration systems [2,3]. However, in the case of the discussed system the configuration is completely different and this methodology has to be applied taken into consideration the specific system features.

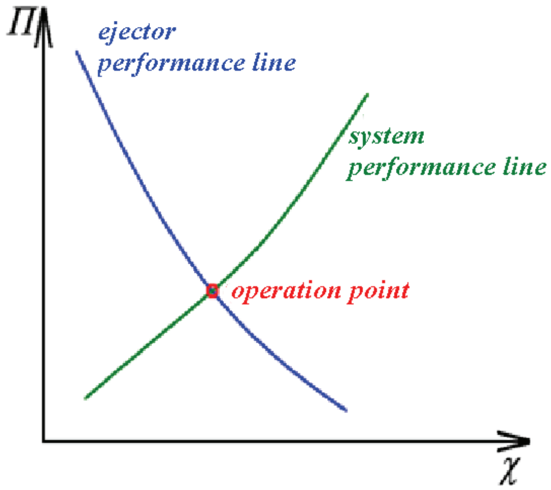

Fig. 3. Idea of the performance line approach: Π is the dimensionless compression ratio, see eq. (21); χ is the volumetric entrainment ratio

It necessary to define the main operation parameters and classify them as fixed for the discussed application and parameters dependent on the ejector operation. The following quantities are assumed as fixed for given application of the refrigeration system:

- evaporation temperature t_o and corresponding evaporation pressure p_o (because of saturated state in the evaporator);
- condensation temperature t_k and corresponding condensation pressure p_k (because of saturated state in the condenser);
- refrigeration capacity $\dot{Q}_o$;
- vapour superheating at the outlet of the evaporator ΔT_o and condensate subcooling at the outlet of the condenser ΔT_k.

The internal efficiency of the compressor η_{ci}; mechanical efficiency of the compressor η_{cm}; and liquid mechanical pump efficiency η_p have to be also treated as known quantities. There are assumed as constant in this report. However, the relationship for η_{ci} as a function of compression (i.e. discharge to suction pressure ratio) may be introduced in the analysis, if needed. The following cycle points can be established unambiguously on the basis of the above parameters (see Fig. 2): 1, 4, 5, and 2'. If so, then specific enthalpies are known for a given refrigerant and the specific refrigeration capacity can be calculated:

$$q_o = h_1 - h_5 . \qquad (1)$$

Therefore the mass flow rate through the evaporator is also known as (see Fig. 1)

$$\dot{m}_o = \frac{\dot{Q}_o}{q_o} . \qquad (2)$$

The following quantities depend on the operation of the ejector and pump, i.e. depend on the performance which is not known:

- inter-stage pressure p_m and corresponding inter-stage temperature t_m (because of saturated state in the mixing chamber and the diffuser of the ejector at the thermodynamic equilibrium conditions);
- pump discharge pressure p_p (discharge temperature tp of liquid depend on the efficiency of the pump and may be calculated on the basis of the discharge pressure);
- motive liquid mass flow rate $\dot{m}_n$ flowing through the pump.

The rest of the characteristic cycle points may be predicted on the basis of the above parameters: 2, 6, 7s and 7, 8, 3s and 3. Therefore:

$$h_2 = h_2\left(p_m\right); h_6 = h_6\left(p_p\right); h_{7s} = h_2\left(p_m, p_p\right);$$
$$h_7 = h_2\left(p_m, p_p\right);\ h_8 = h_8\left(p_m, p_p, \dot{m}_n\right);$$
$$h_3 = h_3\left(p_m, p_p, \dot{m}_n\right); l_{tc} = h_2 - h_1 .$$

These points may be calculated only the performance of the ejector is known. The specific theoretical technical compression work in the mechanical compressor can be then calculated

$$l_{tc} = h_2 - h_1 , \qquad (3)$$

and the specific condenser performance:

$$q_k = h_3 - h_4 \qquad (4)$$

If the above quantities are known, then the performance of the condenser may be calculated:

$$\dot{Q}_k = \left(\dot{m}_o + \dot{m}_n\right) q_k , \qquad (5)$$

also the theoretical power consumption by the mechanical compressor:

$$P_{tc} = \dot{m}_o l_{tc} , \qquad (6)$$

as well as theoretical power consumption by the mechanical pump:

$$P_{tp} = \frac{\dot{m}_n}{\rho_4}\left(p_p - p_k\right). \qquad (7)$$

The actual power consumptions may be calculated as follows:

$$P_c = \frac{1}{\eta_{ci}\eta_{cm}} P_{tc} = \frac{1}{\eta_{ci}\eta_{cm}} \dot{m}_o l_{tc} \qquad (8)$$

and

$$P_p = \frac{P_{tp}}{\eta_p} = \frac{1}{\eta_p}\frac{\dot{m}_n}{\rho_4}\left(p_p - p_k\right). \qquad (9)$$

The coefficient of performance of the whole system is defined as follows:

$$COP = \frac{\dot{Q}_o}{P_p + P_c}\,. \quad (10)$$

Note that all above quantities cannot be calculated without specific knowledge about the operation of the ejector.

ANALYSIS OF EFFICIENCY INCREASE DUE TO EJECTOR APPLICATION

The general reason for application of the ejector in compression system in the discussed configuration (see Fig. 1) is possible increase of the COP defined by eq. (10). It may be assumed that in the case of classic compression system the following system efficiency can be defined:

$$COP_c = \frac{\dot{Q}_o}{P_{cc}}\,, \quad (11)$$

where the power consumption by the compressor in classic compression system is equal to:

$$P_{cc} = \frac{1}{\eta_{ci}\eta_{cm}} P_{tcc} = \frac{1}{\eta_{ci}\eta_{cm}} \dot{m}_o l_{tcc}\,, \quad (12)$$

and the specific theoretical work of vapour compression is calculated as follows:

$$l_{tcc} = h_{2'} - h_1\,. \quad (13)$$

Comparing eq. (10) and (11) the following condition for increase of the system efficiency can be established:

$$P_{cc} - P_c > P_p\,, \quad (14)$$

so on the basis of eq. (3) and (8) the following relationship can be found:

$$\frac{1}{\eta_{ci}\eta_{cm}} \dot{m}_o \left(h_{2'} - h_2\right) > \frac{1}{\eta_p} \frac{\dot{m}_n}{\rho_4} \left(p_p - p_k\right). \quad (15)$$

Now it is possible to introduce the entrainment ratio:

$$U = \frac{\dot{m}_o}{\dot{m}_n}\,, \quad (16)$$

the efficiency modulus:

$$E = \frac{\eta_{ci}\eta_{cm}}{\eta_p}\,, \quad (17)$$

as well as thermodynamic dimensionless work potential:

$$W = \frac{p_p - p_k}{\rho_4 \left(h_{2'} - h_2\right)} = W\left(U, p_m, p_p\right). \quad (18)$$

Therefore the condition for the system efficiency increase is as follows:

$$\frac{1}{E} \frac{U}{W\left(U, p_m, p_p\right)} > 1\,. \quad (19)$$

Note that from direct relationship describing specific enthalpies W does not depend directly on U. However, because compression produced by the ejector depends on U, therefore the dimensionless work potential has to depend on U. If the theoretical case is considered for isentropic processes, then E = 1, and the following condition has to be fulfilled in order to increase the efficiency of the cycle:

$$U_s > W\left(U_s, p_m, p_p\right), \quad (20)$$

where U_s is the mass entrainment ratio for isentropic ejector and compressor, and ideal pump. The following compression ratios may be defined:

$$\pi_p = \frac{p_p}{p_m} \quad ; \quad \pi_k = \frac{p_k}{p_m}; \qquad \Pi = \frac{p_k - p_m}{p_p - p_m} = \frac{\pi_k - 1}{\pi_p - 1} < 1 \quad (21)$$

Therefore:

$$W\left(U, \pi_p, \pi_k\right) = \frac{p_p - p_k}{\rho_4 \left(h_{2'} - h_2\right)} = \frac{p_k}{\rho_4}\left(\pi_p - 1\right)\frac{1}{h_{2'}\left(1 - \dfrac{h_2}{h_{2'}}\right)} \quad (22)$$

It is clear now that W may be interpreted as the ratio of the specific work of isochoric liquid compression between pressures p_p and p_k to the specific work of isentropic vapour compression between points 2 and 2'. It is not possible to draw any conclusion from eq. (20) without any specific information on the ejector performance or the system performance.

REFRIGERATION SYSTEM PERFORMANCE

Graphical representation of this approach is presented in Fig. 3. The performance line of the system may be written as the following relation:

$$U = f_s(\Pi)\,. \quad (23)$$

Physically equation (23) describes system demands for the ejector performance based on energy balance. From energy balance for the ejector the following relation occurs:

$$\dot{m}_n h_6 + \dot{m}_o h_2 = \left(\dot{m}_n + \dot{m}_o\right) h_3\,, \quad (23)$$

so:

$$h_3 = \frac{h_6\left(p_p\right) + U h_2\left(p_m\right)}{1 + U}\,. \quad (24)$$

Therefore from eq. (24) mass entrainment may be calculated:

$$U = \frac{h_3(U) - h_6\left(p_p\right)}{h_2\left(p_m\right) - h_3(U)} = f_s(\Pi, U)\,. \quad (25)$$

In order to apply eq. (25) the value of the specific enthalpy at the outlet of the ejector has to be known. Taking into account the issues concerning operation of installation the processes occurring in the ejector are unimportant. However, there is still important to know the condition of refrigerant at the outlet from the ejector. This is the reason for making following assumption: mixing takes place at constant pressure and processes of expansion and compression proceed isentropically.

Assuming initially the entrainment ratio U, then enthalpy at outlet from ejector can be found from eq. (24). Energy balance equation between inlet and outlet of the motive nozzle can be written as:

$$h_6 = h_7 + \frac{1}{2} w_7^2\,. \quad (26)$$

Energy equation for mixing chamber may be written as follows:

$$h_7 + \frac{1}{2} w_7^2 + U h_2 = (1 + U)\left(h_8 + \frac{1}{2} w_8^2\right) = h_6 + U h_2\,. \quad (27)$$

The mixture velocity in the mixing chamber before compression is calculated on the basis of the momentum equation as well as assumption of the isobaric mixing:

$$\dot{m}_n w_7 = \left(\dot{m}_n + \dot{m}_o\right) w_8\,, \quad (28)$$

therefore:

$$w_8 = \frac{w_7}{1+U}. \tag{29}$$

Combining eq. (27) and (29):

$$h_6 + Uh_2 = (1+U)\left[h_8 + \frac{1}{2}\frac{w_7^2}{(1+U)^2}\right]. \tag{30}$$

From eq. (30) it is possible to calculate the specific enthalpy at the point 8, so:

$$h_8 = \frac{h_6 + Uh_2}{1+U} - \frac{1}{2}\frac{w_7^2}{(1+U)^2}. \tag{31}$$

If the compression process will be isentropic, then:

$$s_3 = s_8 \tag{32}$$

The specific enthalpy at the point 3 may be calculated on the basis of eq. (32) in the following manner:

$$x_8 = \frac{h_8 - h'_m}{h''_m - h'_m}, \tag{33}$$

where h'_m and h''_m are the specific enthalpies of saturated liquid and saturated vapour, respectively, at the inter-stage pressure p_m. Then:

$$s_8 = s'_m + x_8\left(s''_m - s'_m\right), \tag{34}$$

where s'_m and s''_m are the specific entropies of saturated liquid and saturated vapour, respectively, at the inter-stage pressure p_m. From eq. (32) and (34):

$$x_3 = \frac{s_8 - s'_k}{s''_k - s'_k}, \tag{35}$$

where s'_k and s''_k are the specific entropies of saturated liquid and saturated vapour, respectively, at the condensation pressure p_k. Then:

$$h_3 = h'_k + x_3\left(h''_k - h'_k\right), \tag{36}$$

where h'_k and h''_k are the specific enthalpies of saturated liquid and saturated vapour, respectively, at the condensation pressure p_k. Combining eq. (25) and (36) make possible finding relation defined by eq. (23). Note that this is indirect relationship and therefore numerical calculations are necessary to find relation between compression ratio and entrainment ratio demanded by the system.

EJECTOR PERFORMANCE

The performance line of the ejector may be written as the following relation:

$$U = f_e(\Pi). \tag{37}$$

Physically equation (37) describes ejector possibilities based on given ejector geometry and operation parameters. Generally eq. (37) may be developed on the momentum equation describing momentum and energy transfer between both primary and secondary fluids.

Butrymowicz [2] developed the theoretical model of the two-phase ejector. However, this model cannot be directly applied to the discussed case because it is dedicated for the isothermal compression case. Moreover, the correlations for loss coefficients have been prepared for the case of lower operating pressures so there is a clear need to evaluate them under conditions of higher operation pressures and temperatures. It should be noted that there are some simple equations describing roughly two-phase ejectors performance, e.g. Sokolov and Zinger [6] proposed the following equation:

$$\frac{1}{\Pi} = 2\frac{\varphi_n}{\varphi_s} b\left[\varphi_n\varphi_m\varphi_s - \left(1 - \frac{\varphi_m}{2\varphi_n}\right) b(1+\chi)^2\right]. \tag{38}$$

Here: ϕ_s - velocity coefficient for the suction nozzle; ϕ_m - velocity coefficient for the mixing chamber. Basing on eq. (38) it is also possible to find out relationship (37). However, it should be noticed that eq. (38) may be treated as a very rough estimation of the water-air ejector and under conditions of the discussed two-phase ejector application of this equation may lead to substantial errors. No existing models have been validated for the case of high-pressure two-phase ejector operating in the system illustrated in Fig. 1. Therefore in order to appropriate predict the operation point of the system there is a clear need to develop appropriate model of this case of two-phase ejector and validate it by means of the systematic experimental investigations.

CALCULATION RESULTS FOR ISENTROPIC CYCLE

The most important results are given by eq. (20) and eq. (23) for the case of isentropic processes. The relationship between dimensionless work W and pressure ratio are given in Fig. 4 and Fig. 5 for the two cases of refrigerants: R-507 (and R-404A), and propane R-290, respectively. It was assumed that π_p = 1.21 and E = 1.0 for the analysed cases. It is evident from these results that the condition given by eq. (20) should be always fulfilled for both refrigerants only if the mass entrainment ratio will be higher than numerical values of W given in these figures.

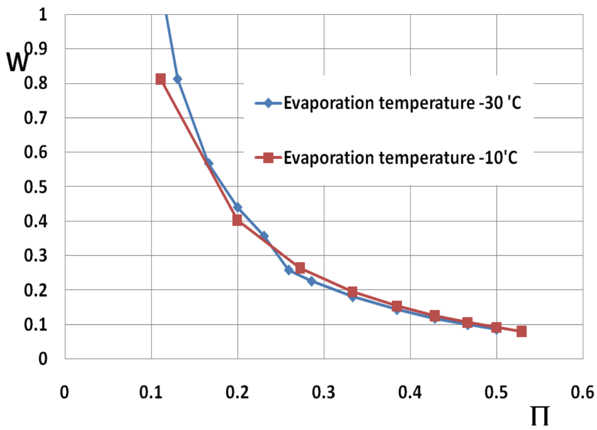

Fig. 4. Calculation results of the dimensionless work potential W versus pump compression ratio Π for refrigerant R-507 (and R-404A); condensation temperature t_k = +40 °C; π_p = 1.21; E = 1.

For both refrigerants the W depends slightly only on the evaporation pressure. Foe propane theoretical W values are significantly lower than for R-507 for the same required compression ratio Π. The volumetric entrainment ratio required by eq. (20) are $\chi < 5$ in most cases for low U conditions. That means it should be possible to prepare the appropriate geometry of the ejector to ensure the required range of the entrainment ratio producing increase of COP.

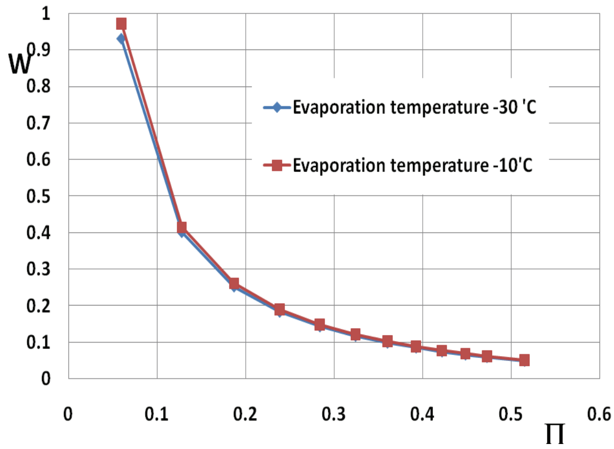

Fig. 5. Calculation results of the dimensionless work potential W versus pump compression ratio Π for propane R-290; condensation temperature t_k = +40 °C; π_p = 1.21; E = 1.

CONCLUSIONS

The own approach of the analysis of the operation of the ejection-compression system has been proposed in this paper. Based on this analysis the limiting entrainment ration of the two-phase ejector has been predicted for given operation conditions. However, the experimental research of the operation of high-pressure two-phase ejector is required in order to operate with reliable ejector performance in the analysis.

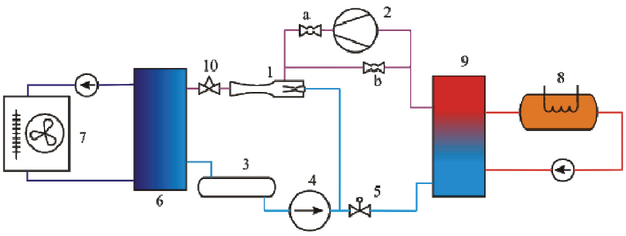

Fig. 6. General layout of the experimental stand: 1 – two-phase ejector , 2 – compressor, 3 – liquid receiver, 4 – refrigerant pump, 5 – expansion valve, 6 – condenser, 7 – fan cooler, 8 – electrical heater, 9 – evaporator, 10 – control valve.

The schematic diagram of the experimental apparatus equipped with the two-phase ejector is shown in Fig. 6. The system consists of three loops: the refrigerant loop and two additional, the cooling liquid loop, and the heating liquid loop. The main parts of refrigeration loop are: tested ejector 1, compressor 2, liquid receiver 3, refrigerant pump 4, condenser 6, evaporator 9. Based on the experiments the loss coefficient and performance analysis will be done for high-pressure two-phase ejector.

NOMENCLATURE

p – pressure	t - temperature
h – enthalpy	m – mass flow
w – velocity	s – entropy
U – entrainment ratio	Π – compression ratio
P – power consumption	η – efficiency
q – specific capacity	ρ - density

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