APPLICATION OF TWO-PHASE EJECTOR AS SECOND STAGE COMPRESSOR IN REFRIGERATION CYCLES

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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.

Keywords: *refrigeration cycle, ejector, two- phase flow, compression*

COMPRESSION – EJECTION REFRIGERATION SYSTEM

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 Figure. 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 [1,6].

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.

The compression-ejection cycles in *p-h* and *T-s* diagrams are presented in Figure. 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.

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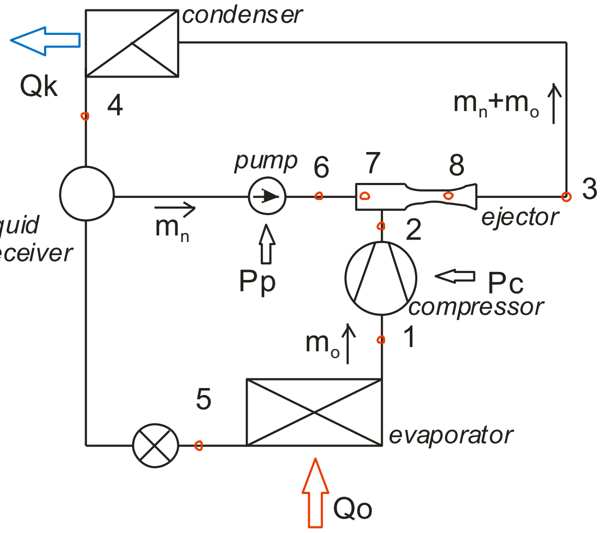

Figure. 1. Schematic of refrigeration compression system with two-phase ejector as a second stage compressor.

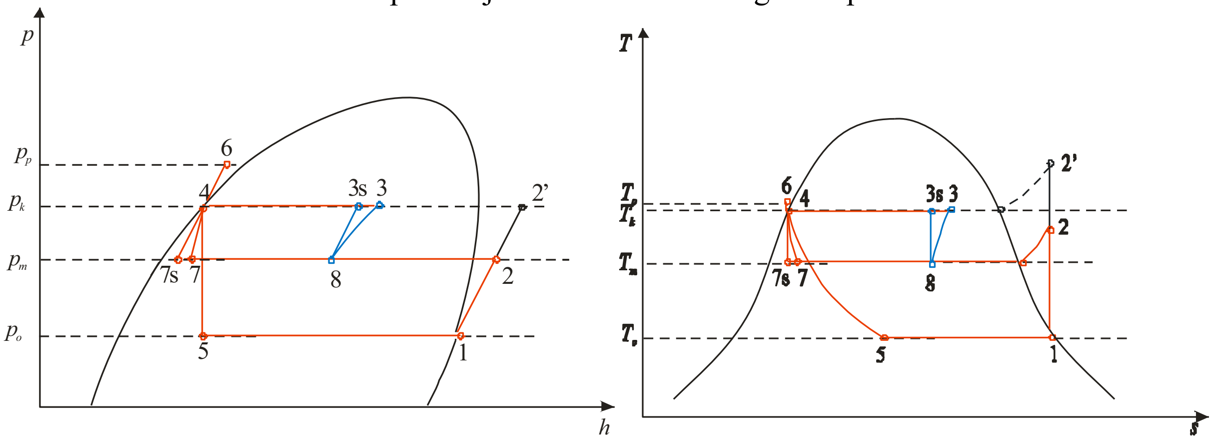

Figure. 2. Compression-ejection cycle of the system presented in Figure. 1.

Important feature of the system is two additional pressures, i.e. inter-stage pressure p_m and corresponding inter-stage saturation temperature T_m, as well as pump motive pressure p_p and liquid phase temperature T_p. 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 Figure. 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. Isentropic process of liquid compression was assumed in Figure. 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.

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 Figure. 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 Figure. 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.

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.

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 Figure. 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.

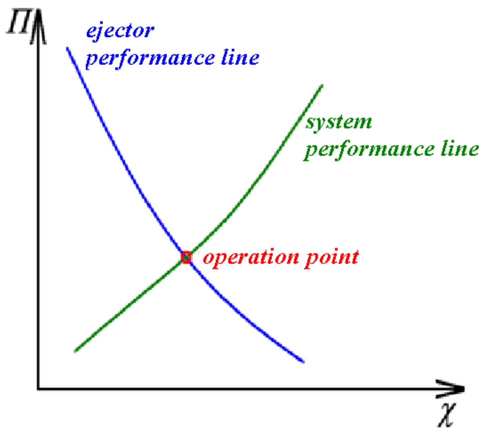

Figure. 3. Idea of the performance line approach: Π is the dimensionless compression ratio, see eq. (21); χ is the volumetric entrainment ratio.

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 paper. The following cycle points can be established unambiguously on the basis of the above parameters (see Figure. 2): 1, 4, 5, and 2'. If so, then the mass flow rate through the evaporator is also known as (see Figure. 1):

$$\dot{m}_o = \frac{\dot{Q}_o}{h_1 - h_5}. \tag{1}$$

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 t_p 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, 7, 8, 3s and 3. Therefore:

$$h_2 = h_2(p_m);\ h_6 = h_6(p_p);\ h_{7s} = h_2(p_m, p_p);\ h_7 = h_2(p_m, p_p);\ h_8 = h_8(p_m, p_p, \dot{m}_n);\ h_3 = h_3(p_m, p_p, \dot{m}_n).$$

These points may be calculated only the performance of the ejector is known. The specific theoretical power consumption by the mechanical compressor:

$$P_{tc} = \dot{m}_o (h_2 - h_1), \tag{2}$$

as well as theoretical power consumption by the mechanical pump:

$$P_{tp} = \frac{\dot{m}_n}{\rho_4}(p_p - p_k). \tag{3}$$

The actual power consumptions may be calculated as follows:

$$P_c = \frac{1}{\eta_{ci}\eta_{cm}} P_{tc}, \tag{4}$$

and

$$P_p = \frac{P_{tp}}{\eta_p} = \frac{1}{\eta_p}\frac{\dot{m}_n}{\rho_4}(p_p - p_k). \tag{5}$$

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

$$COP = \frac{\dot{Q}_o}{P_p + P_c}. \tag{6}$$

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 Figure. 1) is possible increase of the *COP* defined by eq. (6). It may be assumed that in the case of classic compression system the following system efficiency can be defined:

$$COP_c = \frac{\dot{Q}_o}{P_{cc}}, \tag{7}$$

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

$$P_{cc} = \frac{1}{\eta_{ci}\eta_{cm}} \dot{m}_o \left(h_{2'} - h_1\right), \tag{8}$$

The following condition for increase of the system efficiency can be established:

$$P_{cc} - P_c > P_p, \tag{9}$$

so the following relationship can be found:

$$\frac{1}{\eta_{ci}\eta_{cm}} \dot{m}_o \left(h_{2'} - h_2\right) > \frac{1}{\eta_p} \frac{\dot{m}_n}{\rho_4} \left(p_p - p_k\right). \tag{10}$$

Now it is possible to introduce the entrainment ratio:

$$U = \frac{\dot{m}_o}{\dot{m}_n}, \tag{11}$$

the efficiency modulus:

$$E = \frac{\eta_{ci}\eta_{cm}}{\eta_p}, \tag{12}$$

as well as thermodynamic dimensionless work potential:

$$W = \frac{p_p - p_k}{\rho_4 \left(h_{2'} - h_2\right)} = W\left(U, p_m, p_p\right). \tag{13}$$

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

$$\frac{1}{E} \frac{U}{W\left(U, p_m, p_p\right)} > 1. \tag{14}$$

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:

$$U_s > W\left(U_s, p_m, p_p\right), \tag{15}$$

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

$$\pi_p = \frac{p_p}{p_m} \quad ; \quad \pi_k = \frac{p_k}{p_m} \quad ; \quad \Pi = \frac{p_k - p_m}{p_p - p_m} = \frac{\pi_k - 1}{\pi_p - 1} < 1, \tag{16}$$

Therefore:

$$W = \frac{p_p - p_k}{\rho_4 \left(h_{2'} - h_2\right)} = \frac{p_k}{\rho_4}\left(\pi_p - 1\right) \frac{1}{h_{2'}\left(1 - \frac{h_2}{h_{2'}}\right)} = W\left(U, \pi_p, \pi_k\right). \tag{17}$$

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. (17) without any specific information on the ejector performance or the system performance.

REFRIGERATION SYSTEM AND EJECTOR PERFORMANCE

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

$$U = f_s(\Pi). \tag{18}$$

Physically equation (18) describes system demands for the ejector performance based on energy balance for the ejector:

$$\dot{m}_n h_6 + \dot{m}_o h_2 = (\dot{m}_n + \dot{m}_o) h_3, \tag{19}$$

Therefore from eq. (19) mass entrainment may be calculated:

$$U = \frac{h_3(U) - h_6(p_p)}{h_2(p_m) - h_3(U)} = f_s(\Pi, U). \tag{20}$$

Assuming initially the entrainment ratio U, then enthalpy at outlet from ejector can be found from eq. (20). Energy balance equation between inlet and outlet of the motive nozzle can be written as:

$$h_6 = h_7 + \frac{1}{2} w_7^2. \tag{21}$$

Energy equation for mixing chamber may be written as follows:

$$h_7 + \frac{1}{2} w_7^2 + U h_2 = (1+U)\left(h_8 + \frac{1}{2} w_8^2\right) = h_6 + U h_2. \tag{21}$$

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:

$$w_8 = \frac{w_7}{1+U}. \tag{22}$$

Combining eqs. (21) and (22):

$$h_6 + U h_2 = (1+U)\left[h_8 + \frac{1}{2}\frac{w_7^2}{(1+U)^2}\right]. \tag{23}$$

From eq. (23) it is possible to calculate the specific enthalpy at the point 8, so:

$$h_8 = \frac{h_6 + U h_2}{1+U} - \frac{1}{2}\frac{w_7^2}{(1+U)^2}. \tag{24}$$

If the compression process will be isentropic, then the specific enthalpy at the point 3 is equal to:

$$x_8 = \frac{h_8 - h'_m}{h''_m - h'_m}, \tag{25}$$

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:

$$x_3 = \frac{s'_m + x_8(s''_m - s'_m) - s'_k}{s''_k - s'_k}, \tag{26}$$

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

$$h_3 = h'_k + x_3(h''_k - h'_k), \tag{27}$$

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. (20) and (27) make possible finding relation defined by eq. (18). 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.

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

$$U = f_e(\Pi). \tag{28}$$

Physically equation (28) describes ejector possibilities based on given ejector geometry and operation parameters. Butrymowicz [2,3,4] 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. It should be noted that there are some simple equations describing roughly two-phase ejectors performance, e.g. Sokolov and Zinger [5] proposed the following equation:

$$\frac{1}{\Pi} = 2\frac{\varphi_n}{\varphi_s} b\left[\varphi_n\varphi_m\varphi_s - \left(1 - \frac{\varphi_m}{2\varphi_n}\right) b(1+\chi)^2\right]. \tag{29}$$

Here: φ_s - velocity coefficient for the suction nozzle; φ_m - velocity coefficient for the mixing chamber. Basing on eq. (29) it is also possible to find out relationship (28). However, it should be noticed that eq. (29) may be treated as a very rough estimation of the water-air ejector.

CALCULATION RESULTS FOR ISENTROPIC CYCLE

The most important results are given by eq. (20) for the case of isentropic processes. The relationship between dimensionless work *W* and pumping compression ratio are given in Figure. 4 and Figure. 5 for the two cases of refrigerants: R-507 (and R-404A), and R-22, respectively.

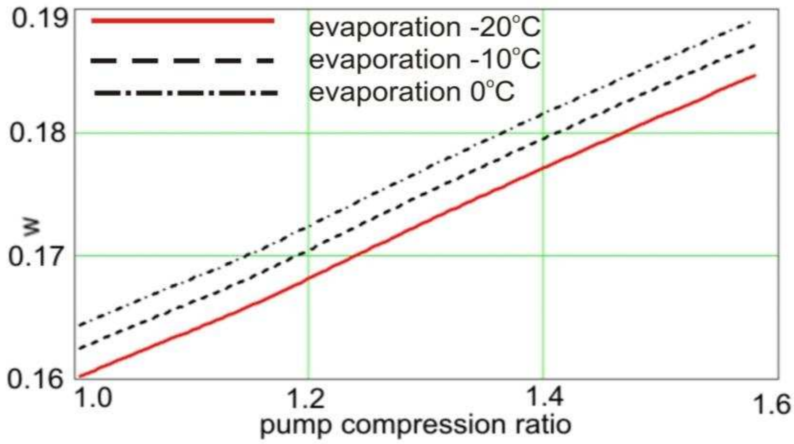

Figure. 4. Calculation results of the dimensionless work potential *W* versus pump compression ratio π_p for refrigerant R-507 (and R-404A); condensation temperature t_k = +40 °C; *E* = 1.

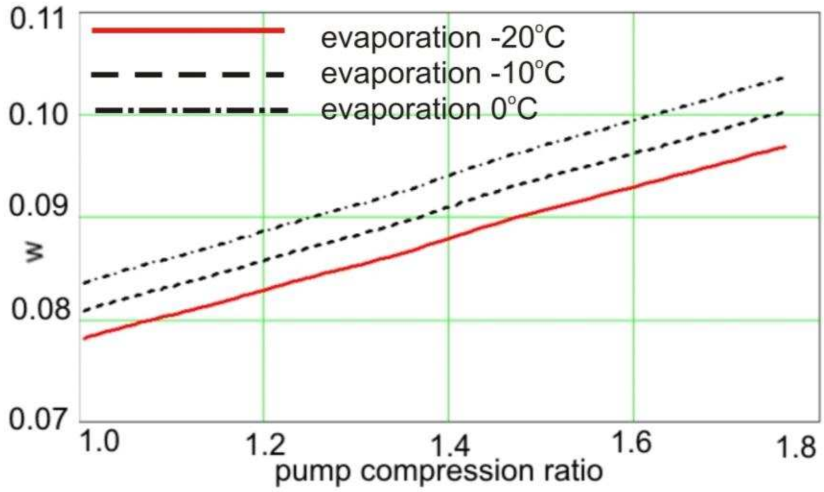

Figure. 5. Calculation results of the dimensionless work potential *W* versus pump compression ratio π_p for refrigerant R-22; condensation temperature t_k = +40 °C; *E* = 1.

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. For refrigerant R-22 this condition should be fulfilled more easier than for refrigerant R-507 (and R-404A as well). The volumetric entrainment ratio required by eq. (15) are $\chi < 5$ in most cases. That means it should be possible to prepare the appropriate geometry of the ejector to ensure the required range of the entrainment ratio.

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.

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