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Experimental Study of a Low-Temperature Compressor-Ejector Refrigeration System

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ABSTRACT

The paper presents the results of a theoretical and experimental study of the low-temperature compressor-ejector refrigeration machines, which led to an increase in the efficiency of refrigeration equipment by 20-60% under conditions of an extreme rise in the ambient temperatures. Such conditions take the character not sporadic, but rather long, which in practice disables a large fleet of refrigeration equipment and causes significant losses. The proposed schematic solutions allow the ejector to be paired with the compressor, regardless of the mode of operation with a positive effect. The use of this scheme makes it possible to increase the effective refrigerating coefficient simultaneously both by reducing the power consumption and by increasing the cooling capacity.

1. INTRODUCTION

The role of the refrigeration economy in the conditions of the civilizational boom is growing at the highest rates, its contribution now reaches about 15% of the world energy balance and continues to increase. It is necessary to find ways to reduce energy intensity with simultaneous increase of the reliability of refrigeration equipment. While radically new solutions are being sought for creating new types of cold generators, it is necessary to find reserves when carrying out cycles of already operating equipment that has a sufficiently long service life, taking into account skilled and timely maintenance.

The relief of compressor operating conditions at peak times due to the ejector booster stage has found its application, however, the efficiency of this solution was small, because for the operation of the ejector, a part of the working steam compressed by the compressor was used (Alyokhin, et.al., 1991, Badylkes & Danilov, 1961).

Partly the role of this stage was performed by an ejector whose main purpose was to replace the throttling with a more reversible process of expansion in the ejector nozzle with the performance of external work, with a small preload of the steam before sucking into the compressor (Elbel, et.al., 2008).

This gives a stable overall increase in efficiency of 4-6%. More should not be expected for most refrigerants, because the work of expanding the liquid, even taking into account its effervescence in the nozzle, gives insignificant work. More significant results can be obtained when the compressor is operating on CO2 as the isoenthalps in the two-phase region on the T-S diagram of this substance are very flat.

It is more efficient to use the exergy of ballast steam separated after throttling to intermediate pressure. At the same time, losses from throttling are reduced, steam is compressed in front of the compressor, ballast steam is not circulated through the evaporator, which also reduces hydraulic losses in the machine, which represents the very long channels with many turns. Therefore, this method is the most promising in the sphere of application of the ejector booster stage.

2. CALCULATION AND THEORETICAL ANALYSIS OF A NEW SCHEME FOR THE INCLUSION OF A BOOSTER STAGE

Schematic of the compressor-ejector refrigeration machine is shown in Fig. 1. The corresponding cycle in the diagram is shown in Fig. 2.

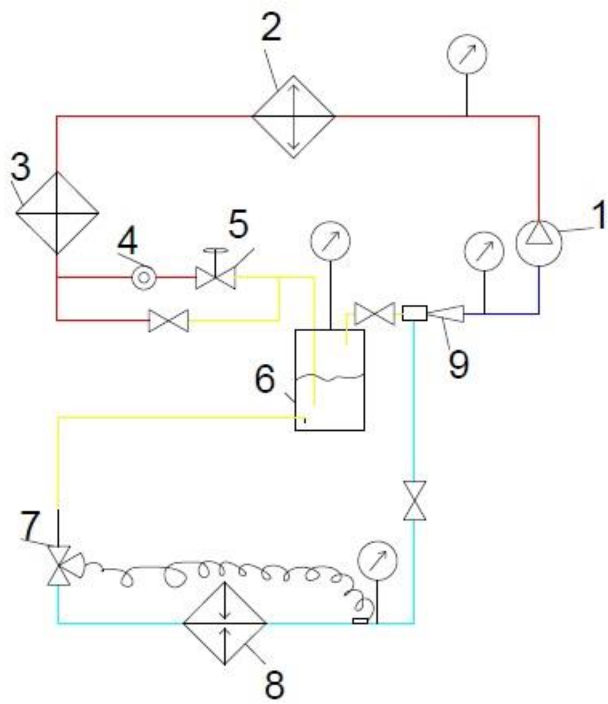

Figure 1: Schematics of an experimental test rig with ejector: 1- compressor; 2- capacitor; 3 - dehumidifier filter; 4 – flow meter; 5 - control valve; 6- liquid separator; 7 - throttling valve; 8 - evaporator; 9 - ejector.

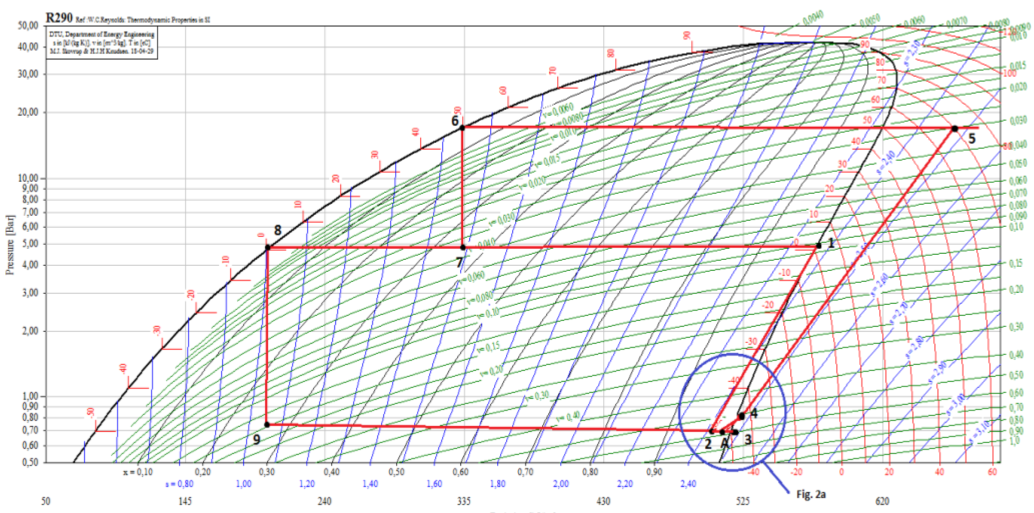

Figure 2: Cycle of vapor-compression machine with booster ejector

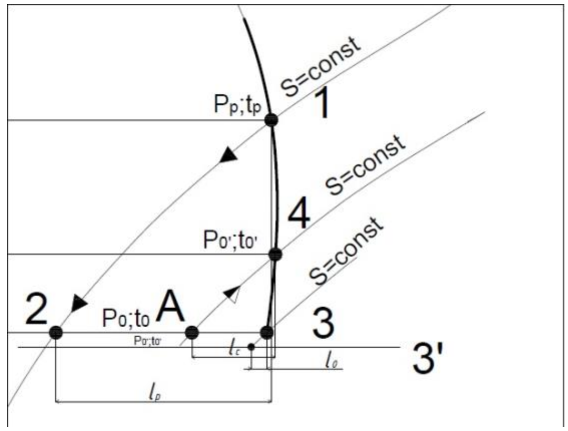

Figure 2a: Processes in the ejector on the h/lgP diagram

In this case, the ejection coefficient is set in advance by the ratio of the steam flow from the evaporator to the flow rate of the ballast steam.

$$U = (G_{eva} + G_m)(1 - x)/G_m \tag{1}$$

$$G_m = (G_{eva} + G_m)\,x \tag{2}$$

$$U = 1/x - 1 \tag{3}$$

The actual range of variation in the ejection coefficient lies in the range 0.7-5.5. The more gently the line of constant enthalpy passes through the beginning of the throttling, the greater the ballast steam consumption, so the pressure beyond the ejector can be increased. In this sense, the CO_2 benefits among other refrigerants, which coincides with the conclusions of S.Elbel and P.Hrnyak. In this case, the specific cooling capacity of the cycle falls, which adversely affects the operation of the compressor. Therefore, there is an optimal ratio of liquid and steam after the first throttling, where the compressor-ejector refrigeration machine produces the best result. Figure 3 shows the dependence of entrainment ratio from mixture dryness.

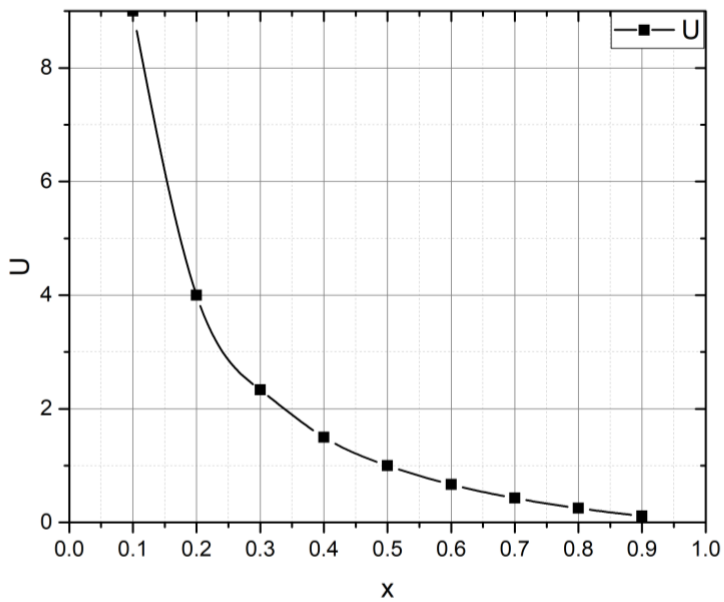

Figure 3: Dependence of U from dryness of vapor-liquid mixture after first throttling.

As it can be seen from the cycle, the main operating parameter is the intermediate pressure to which the condensate expands in the first throttle valve. If the intermediate pressure is high, the specific energy of the working flow will also be high. But the working steam flow will not be sufficient to ensure that the entire volume of the low-pressure steam is compressed to a value to significantly affect the energy and volumetric characteristics of the compressor. The reverse picture is observed if the intermediate pressure is excessively low. Therefore, the task of optimizing the cycle is complicated by the fact that the compressor pressure is affected both by the value of the intermediate pressure after the first throttling and by the value of the vapor compression pressure in the ejector. The ejector stage is calculated from the joint solution of the equation to determine the achievable entrainment ratio, the compressor feed rate, the power consumption and the resulting cooling capacity. In this case, the entrainment ratio is set in advance by the ratio of the steam flow from the evaporator to the ballast steam flow.

The operating modes of the ejector are determined by the expansion of the working flow at supersonic speed as the ratio of the pressure behind the nozzle to the pressure of the entrained flow is less than the critical

$$P_{eva} / P_p < \Pi^* \tag{4}$$

$$\Pi^* = \left(2 / \left(k+1\right)\right)^{k/(k-1)} \tag{5}$$

The value of Π^* lies in the range of 0.5-0.6.

The compression ratio in the booster ejector is low, lies in the range of 1.05-1.15, so the speed of the ejected vapor does not reach the speed of sound in the mixing chamber, i.e. the realization of limiting regimes is not observed at mixing.

To calculate the achievable pressure behind the ejector and the geometry of the flow section of the ejector, the following algorithm was proposed, eq. 6-16 (Sokolov and Zinger, 1989).

$$\frac{\Delta P_{out}}{P_{eva}} = \frac{k_p}{2\left(k_p+1\right)} \frac{1}{\Pi_{p.n.}} \frac{\varphi_1^2 \varphi_2^2 \lambda_{p.n.}^2}{\left[\left(1/\varphi_3 - 0.5\right) v_c / v_p \left(1+U\right)^2 - \left(\varphi_2 \varphi_4 - 0.5\right) v_n / v_p n U^3\right]} \tag{6}$$

$$\frac{f_3}{f_*} = \frac{-b+\sqrt{b^2-4ac}}{2a} \tag{7}$$

$$a = \varphi_1 \varphi_2 q_{p.n.} \tag{8}$$

$$b = -\left\{\varphi_1 \varphi_2 + 2\varepsilon_{p.n.} \left[\left(1/\varphi_3 - 0.5\right) v_c / v_p \left(1+U\right)^2 - \left(\varphi_2 \varphi_4 - 0.5\right) v_n / v_p U^2\right]\right\} \tag{9}$$

$$c = 2\varepsilon_{p.n.} / q_{p.n.} \left(1/\varphi_3 - 0.5\right) v_c / v_p \left(1+U\right)^2 \tag{10}$$

$$\frac{\Delta P_c}{P_n} = \frac{0.5 k_p \varepsilon_{p*} \Pi_{p*} v_n / v_p}{\varphi_4^2 \Pi_{p.n.} \left(f_3 / f_{p*} - 1/q_{p1}\right)^2} U^2 \tag{11}$$

$$\frac{\Delta P_d}{\Delta P_k} = \left(\frac{1+U}{U}\right)^2 \frac{v_c}{v_n} \tag{12}$$

$$\frac{\Delta P_d}{\Delta P_k} = \left(\frac{1+U}{U}\right)^2 \frac{v_c}{v_n} \tag{13}$$

$$v_c / v_p = P_p T_c / \left(P_c / T_p\right) \tag{14}$$

$$v_n / v_p = P_p T_n / \left(P_n / T_p\right) \tag{15}$$

$$T_c = \left(T_p + U T_n\right) / \left(1+U\right) \tag{16}$$

The results of this calculation were used in the calculation of the CFD model of the ejector in the ANSYS program, on the basis of which the drawings of the ejector's geometry profile were produced.

The CFD model of the low-pressure ejector differs in that the high-speed core of the working flow occupies a small space, area ratio is in most cases greater than 20, and the difference in velocities in the flow core and on the boundary layers is much greater than in the high-pressure ejector, Fig. 4,5.

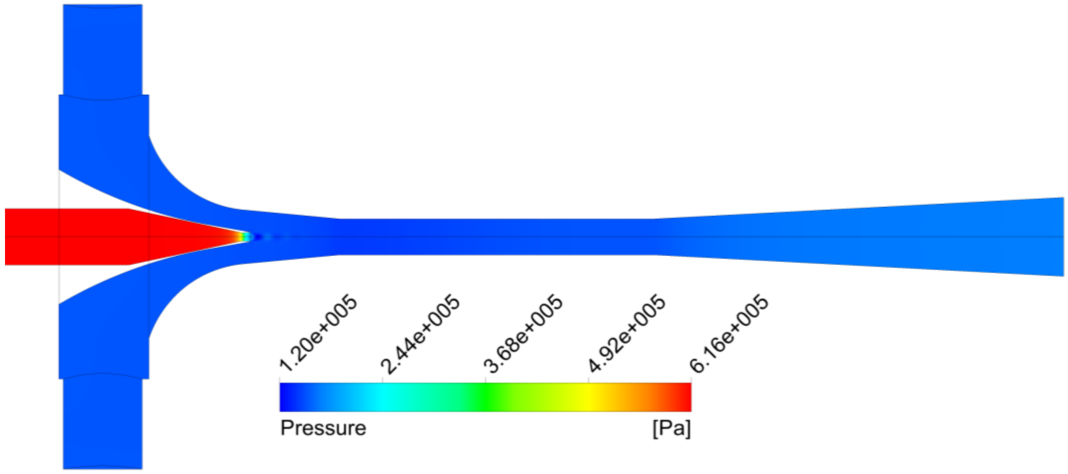

Figure 4: Pressure profile of booster ejector.

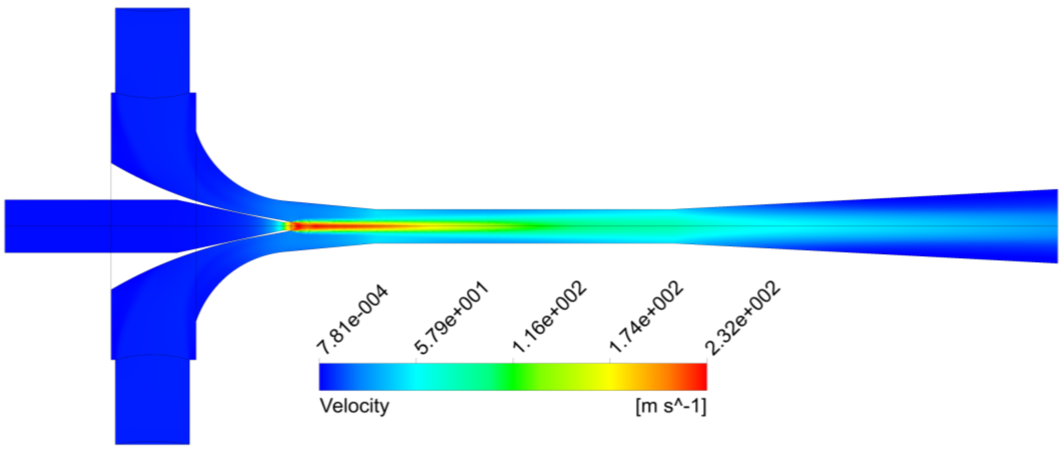

Figure 5: Velocity profile of booster ejector.

The results of calculating the achievable pressure and the effective refrigerating coefficient of the compressor in different modes for R-290, R-134a, R-404 are shown in Table 1. It can be seen from the table that the most important application of the ejector step affected the compressor operating on propane.

3. EXPERIMENTAL STUDY OF A COMPRESSOR-EJECTOR REFRIGERATION SYSTEM

For experimental studies, a test bench was installed on the basis of the hermetic compressor Embraco NE6210E with a cylinder capacity of V_h = 8.77 cm^3. The load on the evaporator was simulated by 5 air heaters of 200 W each, which made it possible to vary the cooling capacity over a wide range of values. The temperature of the outside air was changing by heating the air with a dryer.

Table 1: Entrainment ratios and pressures after first throttling

P_{cond} (MPa)	P_{eva} (MPa)	P_p (MPa)	P_{out} (MPa)	t_{cond} (°C)	t_{eva} (°C)	U
R134a						
1,36	0,15	0,6	0,168	55	-20	3
1,36	0,15	0,33	0,165	55	-20	1,857
1,36	0,15	0,246	0,163	55	-20	1,5
R-507a						
2,64	0,314	0,96	0,346	55	-20	1,5
2,64	0,314	0,85	0,343	55	-20	1,326
2,64	0,314	0,566	0,335	55	-20	1
R-290						
1,9	0,244	0,636	0,271	55	-20	1,857
1,9	0,244	0,474	0,269	55	-20	1,5
1,9	0,244	0,354	0,261	55	-20	1,222

Figure 6 shows schematics of the test rig with control measurement points

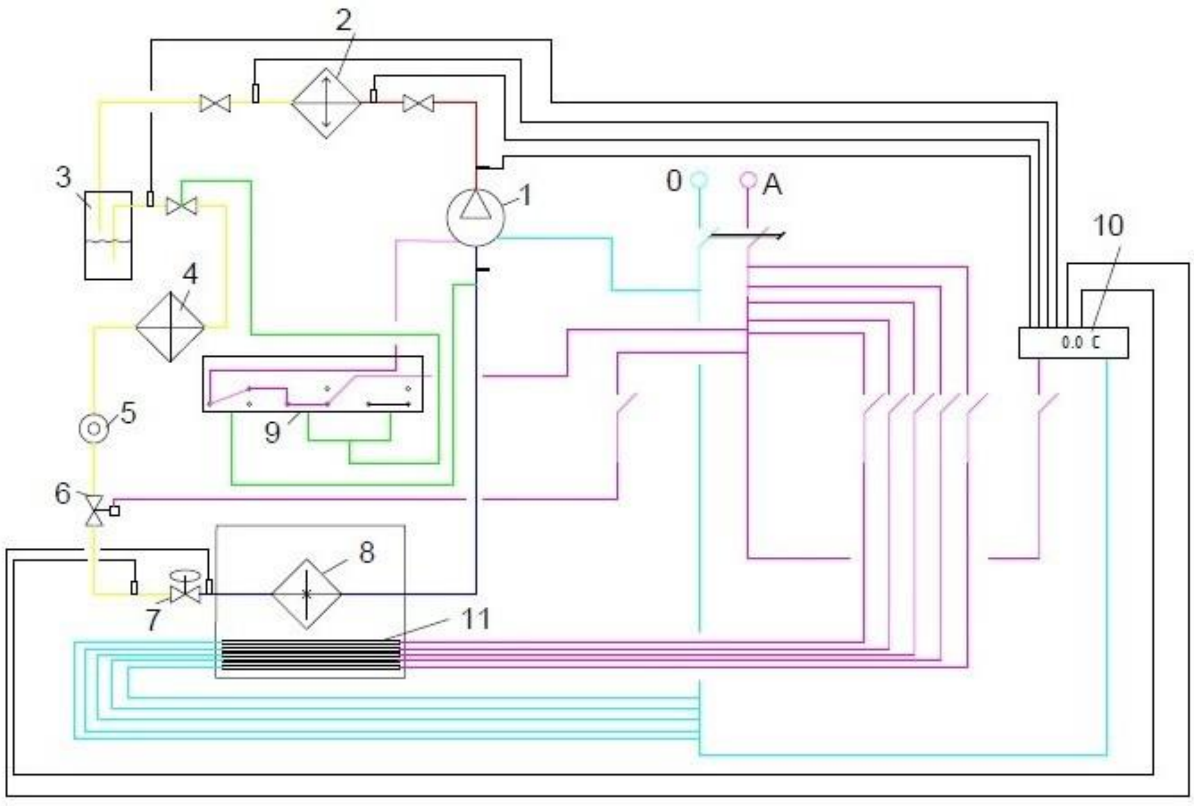

Figure 6: Test rig schematics

In the experimental bench, temperatures at the nodal points of the cycle were measured with the help of calibrated NTC thermistor, pressure was measured using standard manometers. The refrigerant flow was determined by the method of thermal balances, the rotameters and the liquid separator was used as a measuring vessel, and therefore the measurement results were within the permissible error of the measurements. Photo of experimental setup is shown on Fig. 7

Figure 7: Test unit photos.

The first series of experiments allowed to determine the efficiency of the compressor-ejector system, which indicates the optimal values of the intermediate pressure and pressure behind the ejector.

The tests were carried out at 3 evaporation temperatures (-40°C, -30°C and -20°C) and a total condensation temperature of 50°C.

The next stage of the tests was carried out with the ejector turned on. For each mode, a design ejector was installed and readings were taken at various intermediate pressures, which were set with the help of a throttling valve. At each point, the cooling capacity and the electric power consumed by the compressor were determined. The obtained data were plotted with the calculated curves of the effective cooling coefficient versus the compressor suction pressure ratio to evaporation pressure. Measurements were made at the onset of steady-state conditions, when the pulsations of the suction pressure ceased.

When the intermediate pressure was higher than the calculated pressure, the flow rate of the working steam through the nozzle decreased, the pressure behind the ejector was below the design pressure, moreover, there was a separation of the flow from the walls, both in the nozzle and in the diffuser, which requires reducing the cone angle to 1.20 on side.

When the pressure was lowered after the first throttling, the steam flow rate through the nozzle exceeded the calculated value, so part of the ballast vapor was retained in the liquid separator and had to be throttled in the second throttle valve, which also led to a decrease in the refrigerating coefficient and the pressure behind the ejector. The highest values were achieved at the calculated parameters of the first throttling. In this case, the maximum achievable pressure increase in the ejector was observed and, accordingly, the maximum effective cooling coefficient.

Tests have shown that the maximum increase in the refrigerating coefficient is observed at the highest compression ratios in the compressor, which is explained by a sharp drop in the compressor's volumetric characteristics and its productivity. At the same time, the power of the compressor decreases not very fast. The discrepancies between the calculated and experimental values lie within the permissible errors, do not exceed 8-10% in the field of design parameters and 15-25% in the non-calculated regions where the ejector is unstable. Since the evaporation temperature is a stable parameter under real operating conditions, it makes sense to install 3 ejectors in the system: one for the maximum condensation temperature, the other for the average summer maximum, and the third for the off-season. In winter, the installation can work without the ejector, although the option of installing the 4th ejector on the average winter minimum is also considered. Even if the increase in the refrigeration ratio is 10-15%, in general, the electricity consumption for obtaining the cold will be reduced to such a level that it is possible to release more than one large power plant from the load.

6. CONCLUSIONS

1. The use of a new compressor-ejector schematics in refrigeration technology leads to a reduction in electricity consumption by 10-40% and a corresponding reduction in greenhouse gas emissions.
2. Experimental studies of a low-pressure ejector have shown its reliability, efficiency and stable operation under design conditions and in the vicinity of a change in the intermediate pressure of 10-25%.
3. The CFD model of a low-pressure ejector has higher impact losses than high-pressure one due to higher differences in the rates of mixing flows.
4. The mixing process in low-pressure ejectors is more irreversible, so a higher growth of the entropy of the mixture is observed compared to high-pressure ejectors.

NOMENCLATURE

G_m	motive vapour mass flow	kg/s
G_{eva}	evaporator mass flow	kg/s
U	entrainment ratio	(–)
x	dryness degree of vapour-liquid mixture after first throttling	(–)
P	pressure	MPa
T	temperature	°C
k	adiabatic index	(–)
$\varphi_1,\varphi_2,\varphi_3,\varphi_4$	velocity coefficients	(–)
Π	relative pressure	(–)
λ	relative velocity	(–)
v	specific volume	m^3/kg
ε	relative density	(–)

Subscript

cond	condensation
eva,n	evaporation
p	pressure after first throttling

out	ejector outlet pressure
c	compression pressure
d	diffuser
*	critical parameter

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