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INNOVATIVE TECHNICAL DESIGN OF AIR HEAT PUMPS OPERATING IN THE CONDITIONS OF THE NORTHERN CLIMATE

СОЛТҮСТІК КЛИМАТТА ЖҰМЫС ІСТЕЙТІН АУА ЖЫЛУ СОРАПТАРЫНЫҢ ИННОВАЦИЯЛЫҚ ТЕХНИКАЛЫҚ ЖОБАСЫ

ИННОВАЦИОННАЯ ТЕХНИЧЕСКАЯ КОНСТРУКЦИЯ ВОЗДУШНЫХ ТЕПЛОВЫХ НАСОСОВ, РАБОТАЮЩИХ В УСЛОВИЯХ СЕВЕРНОГО КЛИМАТА

Abstract. *This paper describes the calculations and investigations to adapt an ejector cycle for air-source heat pump designed for operation in Northern climate, where outdoor temperature may drop significantly. The ejector was installed on compressor intake to increase the suction pressure and to lower the compression ratio. The objective of this research was to permit a single-stage compression heat pump to work at outdoor temperatures below –18C, thus avoiding more complicated two-step compression at such conditions.*
Keywords: *ejector, cycle, temperature, climate, air-source.*

Аңдатпа. *Бұл мақалада сыртқы ауа температурасы айтарлықтай мәндерге дейін төмендеуі мүмкін солтүстік климаттық жағдайларда жұмыс істеуге арналған ауа жылу сорғысының эжекторлық циклін бейімдеу бойынша есептеулер мен зерттеулер сипатталған. Сору қысымын арттыру және қысу коэффициентін азайту үшін компрессордың кірісіне эжектор орнатылды. Бұл зерттеудің мақсаты бір сатылы қысу жылу сорғысының -18 °С төмен сыртқы температурада жұмыс істеуіне мүмкіндік беру, осылайша мұндай жағдайларда қиынырақ екі сатылы қысуды болдыртпау болды.*
Түйін сөздер: *инжектор, цикл, температура, климат, ауа көзі.*

Аннотация. *В данной статье описаны расчеты и исследования по адаптации эжекторного цикла воздушного теплового насоса, предназначенного для работы в северных климатических условиях, где температура наружного воздуха может опускаться до значительных значений. Эжектор был установлен на входе компрессора для увеличения давления всасывания и снижения степени сжатия. Цель этого исследования состояла в том, чтобы позволить одноступенчатому компрессионному тепловому насосу работать при температуре наружного воздуха ниже -18 °С, избегая, таким образом, более сложной двухступенчатой компрессии в таких условиях.*
Ключевые слова: *инжектор, цикл, температура, климат, воздушный источник.*

Introduction. With rising costs of electricity and energy carriers there is a market opportunity for new energy-efficient designs for items that consume large amounts of electricity. One area that requires more efficient designs is the HVAC (heating, ventilation, air-conditioning) industry and its associated operating components. It is estimated by US Department of Energy that 40 % of all electricity usage in commercial buildings is related to HVAC systems [8]. This 40 % provides an attractive target for reducing future energy needs. Improved methods of heating, refrigeration, and air-conditioning that can be implemented cost effectively could provide a significant breakthrough in green energy efficiency.

Specifically, this paper deals with application of ejectors in reverse Rankine cycles. There is an extensive body of previous knowledge and a large literature base regarding ejector applications in refrigeration dating back to 1930's [9]. In spite of this, the ejector use in refrigeration systems was always considered as controversial at best, mainly because a lot of research has been conducted for many years with only theoretical results and without visible, commercial products.However, just recently it was disclosed by najor automobile manufacturer, that they developed an ejector-based, air conditioning (A/C) system to be installed in commercial cars and trucks [12]. Further, our own investigations demonstrated another example of effective use of ejectors in A/C systems: we were able to demonstrate the ejector-based air conditioning system for residential dwellings, which was able to considerably lower the ambient temperature, while using low-grade heat (solar, waste, etc.) as a main source of energy and further, to completely eliminate the compressor from the cycle[6]. It is encouraging that two very different approaches using different working fluids and heat sources have demonstrated the practical use of ejectors to provide economical cooling.

The research on ejector application is consistent with present directions in HVAC industry, which are: 1) improving the efficiency, 2) reducing the refrigerant charge and 3) reducing the footprint (size) of the refrigeration units. We believe that ejector, when considered as a "technology platform" is capable to significantly contribute to all three above goals and therefore, it will attract more attention and research funding in the future.

The purpose of the presented research was to adapt an existing model of air-source heat pump to very low outside temperatures. Indeed, several US northern states, Canada and Central Asia experience such drop in winter temperatures, especially at night. Our study showed that in the State of Minnesota there are 500 hours per year when the temperature drops below –18C (0F). Air-source heat pumps are used widely in those areas for space heating, mainly due to their ease of installation, low initial costs and favorable electric rate schedule. Nevertheless, their operation presents numerous problems and difficulties at such low ambient temperatures, specifically [2], [3], [7]:

1. Insufficient heat output due to the fact that required heat output is the highest when heat pump capacity is lowest (at low outdoor temperatures)

2. High compressor discharge temperature caused by the low suction pressure and high pressure ratio. This leads to heating systems in which heat pump works at medium outdoor temperatures, while at low temperatures the electric resistance heating is turned on. The combined efficiency of those systems is quite low.

3. The coefficient of performance (COP) decreases rapidly for high pressure ratios (which takes place at low outdoor temperatures)

4. If the heat pump is designed for low outdoor temperatures, it will have a capacity that is too large at medium temperatures. Therefore, the heat pump needs to cycle on and off at higher outdoor temperatures, which leads to lower efficiency compared to steady-state performance and also a lower comfort level for occupants.

There are many possibilities to address the above problems and Bertsch et al [2] lists and compares eight such possible solutions. These concepts include two-stage compression systems with either intercooling, economizing or in a cascade arrangement, systems which inject two-phase refrigerant into the compressor to decrease the compressor discharge temperature and systems which use high oil flow rates to cool the compressor. However, in a practical applications, those systems might be cost-prohibitive for a residential customer and further, are difficult to operate, mainly due to problems related to proper oil circulation.

Materials and methods of research. Refrigeration Cycle with Ejector Installed on the Suction of the Compressor. We investigated the feasibility of another original solution: to install a two-phase ejector on the compressor suction for an existing heat pump in order to lower its evaporation temperature. The final objective was to permit a single-stage compression heat pump to work at outdoor temperatures below – 18C. The ejector was installed as the first step of compression, increasing the suction pressure. Such ejector application can be considered from two points of view: the first, that it increases the suction pressure of the compressor, lowering the required pressure ratio and decreasing the compression work. The second position treats the ejector as a device in which pressure drops without the throttling process. It is widely recognized that the highest energy loss in refrigeration cycle is caused by the throttling, during which the kinetic energy is converted to friction and eventually lost. Therefore, the most effective way of reducing this loss is the recovery of the kinetic energy of the fluid, produced as an effect of pressure drop. One method to achieve it would be a thermodynamic process with isentropic expansion rather than isenthalpic throttling. Such process is provided by a two-phase ejector installed in the liquid line between the evaporator and liquid separator as shown in Figure 1 below.

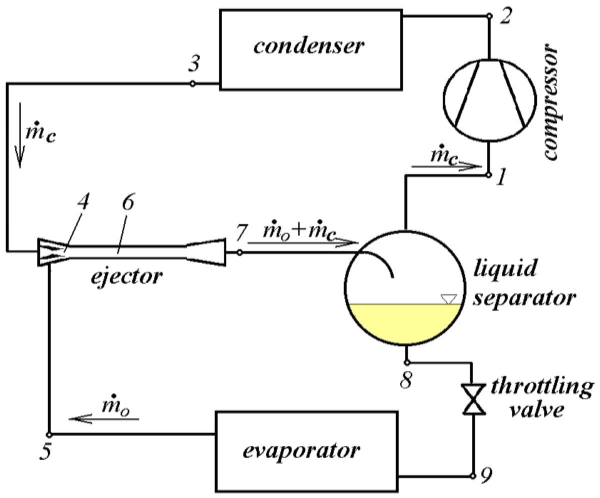

Figure 1. Schematics of ejector refrigeration cycle under study [5]

Liquid refrigerant is the motive fluid in this ejector. Liquid refrigerant expands in the motive nozzle and partly evaporates. The two-phase flow consisting of droplets in the vapor phase flows out of the nozzle. This two-phase mixture, still as a motive fluid, causes suction of the vapor from the evaporator. The mixture of these streams flows from the diffuser of the ejector to the liquid separator. Then the liquid phase flows to the evaporator via the throttling valve and the vapor phase is sucked by the compressor. The physical phenomena occurring in the motive nozzle, suction chamber, mixing chamber and the diffuser of discussed ejector are extremely complicated and the knowledge about the maximum performance and energy effectiveness of such ejector is still very limited. The T-s and p-h graphs for refrigeration cycle of Figure. 1 is shown in Figure 2. It is important to note that mass flow rates of the refrigerant are different for different processes of the cycle. The ideal liquid expansion in the motive nozzle proceeds at constant entropy – the process 3-4s. In the case of the throttling process during which pressure drops from condensation pressure pk to evaporation pressure po, this process is isenthalpic 3-4d. The actual process occurring in the motive nozzle of the ejector proceeds along the curve 3-4. The pressure drop in the suction line from the evaporator to the suction chamber of the ejector can be omitted as negligible, so p5□po. The refrigerant is partly compressed in the mixing chamber due to mixing shock, so the pressure increases from evaporation pressure po to the pressure at the outlet of the mixing chamber p6. The ideal compression of the two-phase flow in the diffuser proceeds at constant entropy 6-7s. The actual compression proceeds along the line 6-7. The suction pressure of the compressor is the inter-stage pressure pm and is higher then evaporation pressure po, so the specific work of the compression ltis lower in comparison with pure vapor compression refrigeration device.

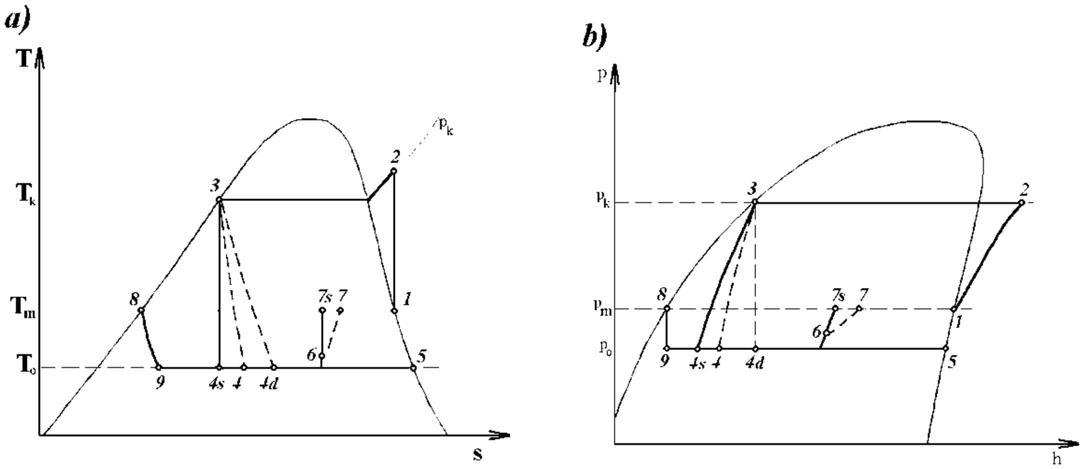

Figure 2. Theoretical cycle of compressor-ejector refrigeration device shown in Figure 1: (a) in coordinates T-s, b) in coordinates p-h [5]

The most comprehensive state-of-the-art and history of this development is presented by Butrymowicz [10] and Elbel et al. [11]. While many researchers showed that from theoretical point of vision it is possible to achieve more than 20 % increase of the COP for most of the refrigerants HCFC and HFC in comparison with classic vapor compression refrigeration devices, the past experimental investigations show that increase only as much as 4-5 % of COP was achieved so far for refrigerant R-12 [13].

It appears that the most important reason for such moderate result is a low efficiency of the two-phase ejector. The processes of the momentum transfer in the suction and mixing chambers between the motive liquid and sucked vapor phase is extremely difficult for modeling and both theoretical and experimental investigations of these processes are still in the early stage. These authors however have undertaken the research mainly for extending the temperature range of the heat pumps rather than improving their efficiency.

Results and discussion. Investigations of Existing Air-Source Heat Pump for Northern Climate. For the purpose of our evaluation, a specific design of a single-stage compression air-source heat pump operating with refrigerant R410A was selected, the one that was already proven to operate at minimum temperatures down to about –20C. The subject heat pump was equipped with a scroll compressor and an injection of refrigerant to the compressor suction at low temperatures.

The schematics of the original design is shown below in Figure 3. [5] in black, while the ejector system additions (which will be discussed later in this paper) shown in red.

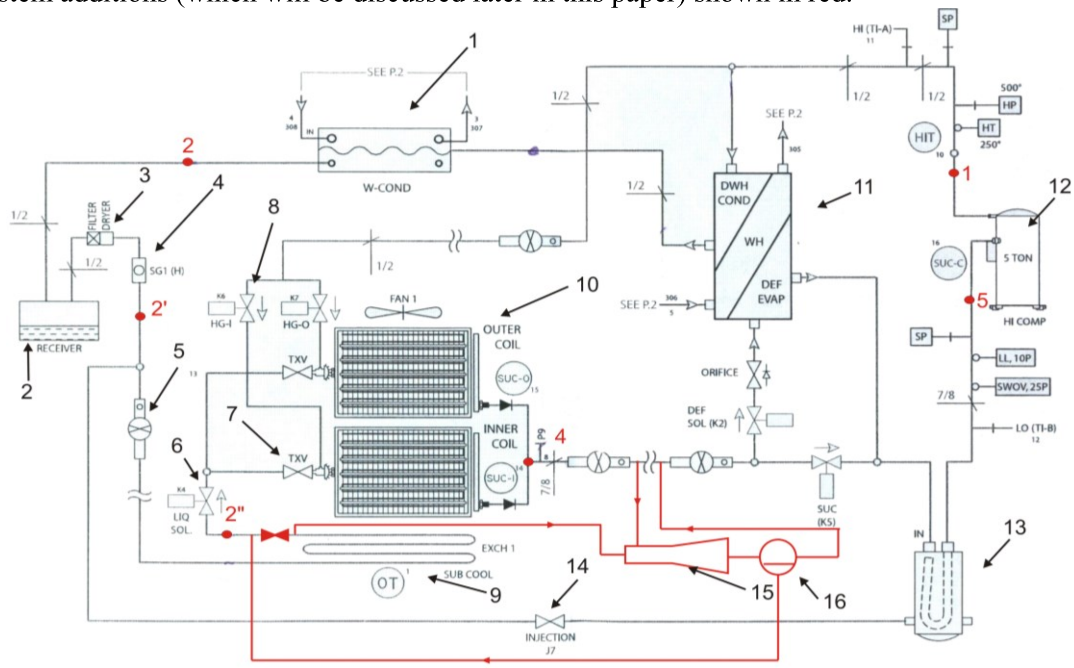

Figure 3. Schematics of air-source heat pump for Northern Climate: 1 – condenser; 2 – liquid receiver; 3 – filter-dryer; 4 – sight glass; 5 – ball valve; 6 – electromagnetic valve; 7 – thermostatic expansion valve; 8 – shut-off valve; 9 – pre-cooler; 10 – evaporator; 11 – regenerative heat exchanger; 12 – compressor; 13 – accumulator; 14 – injection valve. Additional components for ejector modification: 15 – two-phase ejector; 16 – separator

The compressed refrigerant, flows from the compressor (12) to regenerative heat exchanger (11) where it undergoes initial cooling. The heat removed from the refrigerant is then used to superheat the vapor leaving the evaporator in order to decrease the compressor work. The temperature achieved after the compressor (at the level of 80-100C) allows additionally to use exchanger 11 as a desuperheater for the purpose of heating the domestic hot water. Initially cooled refrigerant condenses in a condenser (1) and flows to the liquid receiver (2). From the receiver, liquid refrigerant flows through filter-dryer to the pre-cooler where it is subcooled. Thermostatic expansion valve lowers the pressure of liquid refrigerant to the evaporation pressure and the refrigerant is then evaporated in the evaporator (10). The vapor refrigerant is then superheated in the heat exchanger (11) utlizing the heat recovered during an initial cooling of a compressed refrigerant. The superheated refrigerant is then sucked by a compressor and compressed. The temperature and pressure sensors installed in critical location of a cycle control the refrigerant condition and parameters of a cycle. If the compressed vapor temperature immediately after the compressor (marked by point 1 in Fig. 3.) exceeds 104.5C, the injection valve (14) activates and injects the liquid refrigerant from point 2' to the compressor suction line via accumulator (13). This increases the suction pressure and lowers the compression ratio.

In order to evaluate the operation of this heat pump and to establish the thermodynamic cycle (for example p-h diagram), multiple tests conforming to ASHRAE standards [1] were conducted at the manufacturer's location. In order to achieve low ambient temperatures, the outdoor heat exchanger was placed in a special refrigerated chamber, capable of reaching temperatures down to –30C. The outdoor coil (heat exchanger) is shown in Figure 4 [3], [5].

The results of measurements are presented in Table 1 below for points 1 through 5 as marked in Figure 3. As shown, this heat pump was capable to operate at the outdoor temperature up to –20C.

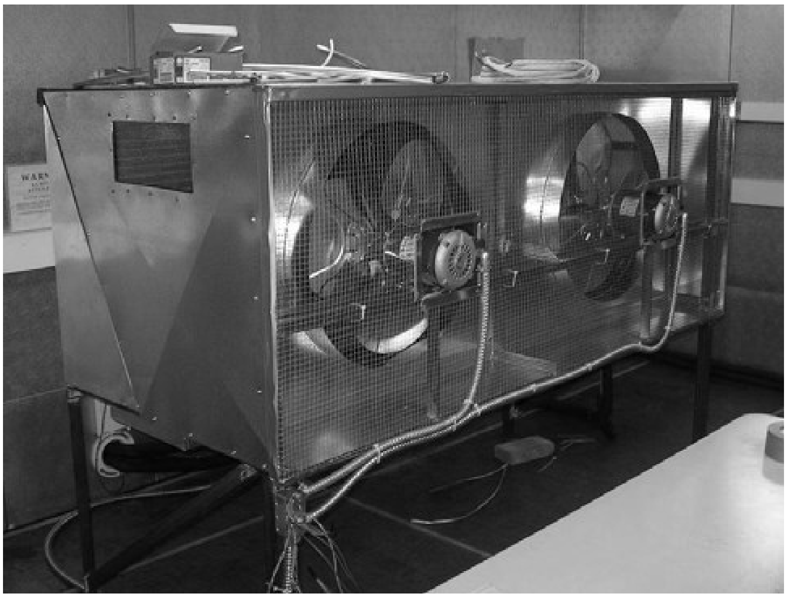

Figure 4. Outdoor heat exchanger for tested air-source heat pump

Table 1. Measurement results for points 1 through 5

Time	p_1 [bar]	T_1 [°C]	p_2 [bar]	T_2 [°C]	$T_{2'}$ [°C]	$T_{2''}$ [°C]	T_4 [°C]	T_5 [°C]
03:53	22.0	105.0	19.62	35.4	32.4	-13.0	-17.0	-14.5
04:04		105.5	19.13	35.0	32.2	-13.5	-17.5	-16.0
04:08		105.5	19.06	35.0	32.2	-14.5	-19.2	-18.5
04:01		105.5	19.00	34.4	31.7	-16.0	-19.2	-19.0
04:12		105.5	19.00	34.4	31.7	-16.0	-19.2	-17.0
04:15		105.0	19.06	34.5	31.7	-16.0	-18.5	-19.0
04:17		105.0	19.06	34.4	31.7	-15.5	-20.3	-13.9
04:02			19.06	32.8	31.7			-16.5

Table 2 below shows pressures, temperatures and specific enthalpy for the refrigerant in all major points of the thermodynamic cycle. In order to determine some of these points, we used a customary/industrial rules for refrigerant temperature vs. evaporator and condenser temperature [5]. For example, to figure the evaporation temperature, it was assumed that the temperature of the refrigerant is 4C to 7C lower than the ambient temperature. The parameters of the refrigerant in Point 3 were determined from the temperature in Point 4. For this temperature, the saturation temperature/pressure was determined from charts. The isenthalpic expansion in an expansion valve was assumed from the pressure of p2 = 19.65 bar to p3=p4=3.24 bar.

The obtained points were then used to draw a thermodynamic cycle for this heat pump and it is shown in Figure 5.

Table 2. Data for major points of refrigeration cycle

Point	p [bar]	t [oC]	entalphy [kJ/kg]
1	22	106	511.58
2	19.65	35.4	432.12
2'	19.65	32.4	427.59
2"	19.65	-13	180.89
3	3.24	-25.5	179.15
4	3.24	-17	419.59
5	3.24	-14.5	421.83

Based on the measurements and the p-h graphs of Figure 5. the remaining data were calculated as follows:

1) The total heat removed by both condensers, i.e. heat exchangers (1) and (11) as in Figure 1. equals to $Q_k = 8.20$ kW.

2) The evaporator capacity is $q_0 = h_4 - h_3 = 240.44$ kJ/kg,

3) The condenser capacity is $q_k = h_1 - h_{2''} = 330.69$ kJ/kg,

4) Compression work: $l_t = h_1 - h_4 = 92$ kJ/kg.

5) And finally, the required refrigerant flow in a cycle:

$$m = \frac{\dot{Q}_k}{q_k} = 0.025 \text{ kg/s} \approx 1.5 \text{ kg/min}.$$

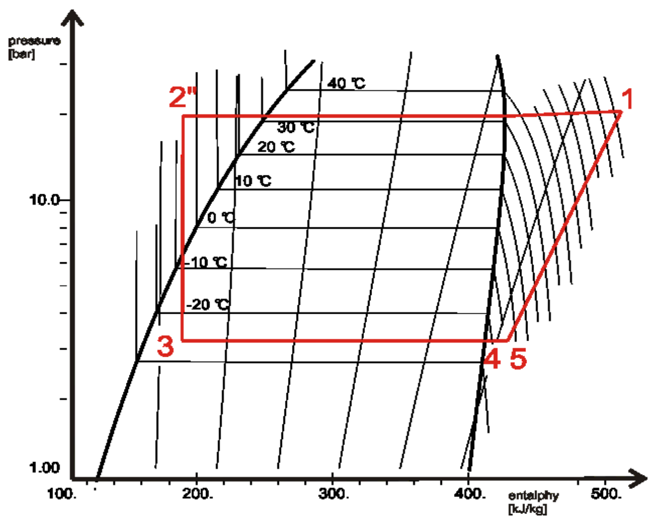

Figure 5. Outdoor heat exchanger for tested air-source heat pump

Conceptual Design of Ejector Cycle and Investigations. This paragraph shows a possible improvement to the existing air-source heat pump by application of a two-phase ejector. The investigations were performed of various working modes of an ejector that would allow the heat pump to work under lower outdoor temperature; down to –30C. Therefore, the main objective was to avoid the second step compression rather than any improvement in COP. The original concept was for an ejector to initially compress the refrigerant from saturation pressure at –30C to –20C while the compressor will take the rest of a compression. As shown in the above paragraph, the present design is capable of such compresion ratio for a compressor.

The proposed modification of a heat pump system was shown previously in Figure 3[5]. The additional curcuitry for ejector is marked with red color. Two additional components are required per Figure 3: an ejector and a separator tank. The two-phase ejector in this application is intended to lower the evaporation pressure without changing the pressure ratio of a compressor. This idea is illustrated in Figure 6.

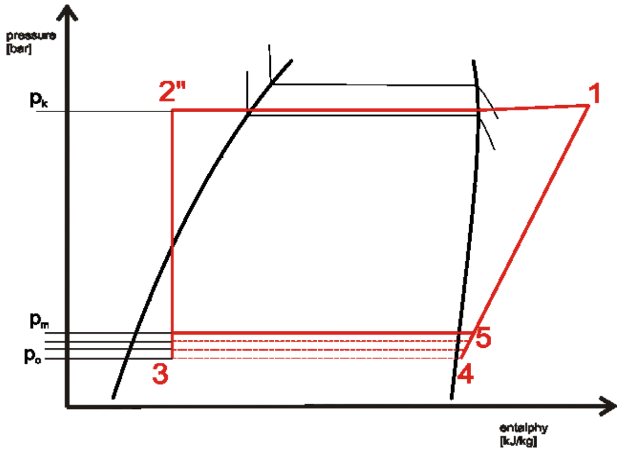

Figure 6. The concept of lowering the evaporation pressure by applying two-phase ejector. The ejector initially raises the pressure from Point 4 to Point 5 and the compression ratio at the compressor remains unchanged (Point 5 to Point 1)

The original heat pump works properly at the minimum suction pressure of 3.24 bar, which corresponds to approx. –20C outdoor temperature. In order to allow the operation at –30C, the saturation pressure has to be about 2.2 bar, which means that the ejector should be capable of compressing the vapor refrigerant by approx 1 bar.

In order to perform calculations required to determine the technical feasibility of a proposed concept, the theoretical model was assumed as shown in Figure 7.

Per Figure 7 our known parameters are 1) a condition of a refrigerant leaving the condenser (Point 2”) and 2) a refrigerant condition on the suction side of a compressor (Point 5). By assuming a certain evaporation pressure, we are at the same time setting values of an enthalpy in Point A and Point 4 as an saturation enthalpy for a liquid h_A and a vapor h_4. Writing the energy balance equation, we obtain:

$$\dot{m}h_{2''} + \dot{m}_0 h_4 = \dot{m}h_5 + \dot{m}_0 h_A \tag{1}$$

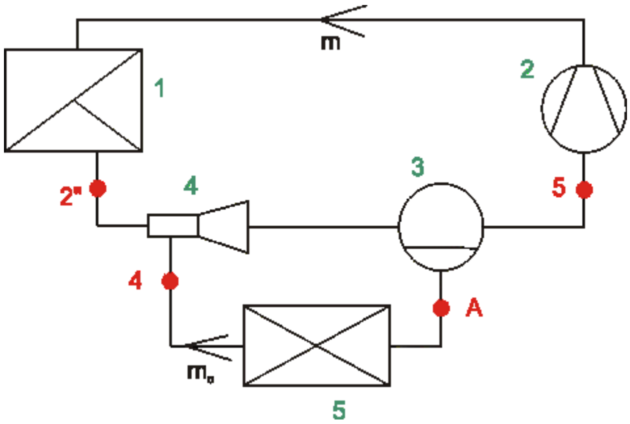

Figure 7. Theoretical model for ejector calculations:
1 – condenser, 2 – compressor, 3 – separator tank, 4 – ejector, 5 – evaporator

The mass entrainment ratio is defined as:

$$U = m_0 / m \tag{2}$$

inserting into Eq. 1 gives:

$$U(h_4 - h_A) = h_5 - h_{2''} \tag{3}$$

and finally:

$$U = \frac{h_5 - h_{2''}}{h_4 - h_A}. \tag{4}$$

Further, the ejector pressure ratio is defined as:

$$\Pi = \frac{p_m - p_0}{p_k - p_0} \tag{5}$$

Table 3 lists temperatures, pressures and enthalpies in main points of a cycle for different values of an evaporation pressure p_0. As discussed above, we need to lower this pressure to approx. 2.2 bar, however for the purpose of obtaining full characteristics, calculations were carried over much broader range of this pressure.

Table 3. Characteristic points of a cycle with two-phase ejector

$p_k = p_{2''}$ [bar]	$h_k = h_{2''}$ [kJ/kg]	$p_m = p_5$ [bar]	p_o [bar]	Π [–]	h_5 [kJ/kg]	h_A [kJ/kg]	h_4 [kJ/kg]	U [–]
22	180.89	3.24	3.1	0.007	421.8	180.89	419.59	1.00
			3	0.013		159.86	410.99	0.95
			2.6	0.033		154.91	409.41	0.94
			2.2	0.053		149.37	407.57	0.92
			1.8	0.071		143.02	405.39	0.91
			1.4	0.089		135.51	402.71	0.89
			1	0.107		126.12	399.21	0.87
			0.6	0.123		113.14	394.14	0.85
			0.3	0.136		97.576	387.77	0.83

The results are shown in a graphical form in Figure 8 as a cycle characteristics presenting the required ejector pressure ratio as a function of its mass entrainment ratio:

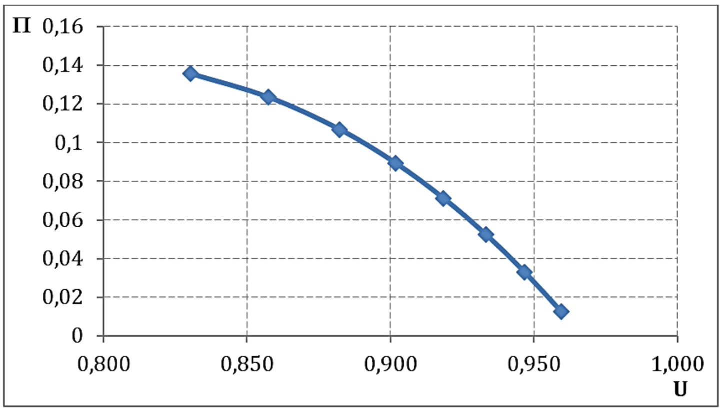

Figure 8. Thermodynamical characteristics of an investigated cycle with two-phase ejector

Based on author's past experience with similar ejectors, the required mass entrainment ratios could be in the upper limit of ejector capability but such a range was achievable in previous investigations. Therefore, as the next step, we investigated few modifications to this thermodynamic cycle that could bring the entrainment ratios to lower, thus more achievable levels. The first such possibility was the evaporator operation in the range of wet vapor. Those calculations were performed for two different vapor qualities: x1 = 0.25 i x2 = 0.85. In those cases, the evaporation process is characterized by the difference in enthalpy, specifically for quality x = 0.25 and for x = 0.85. Thermodynamic cycles for those two cases are illustrated in Figure 9.

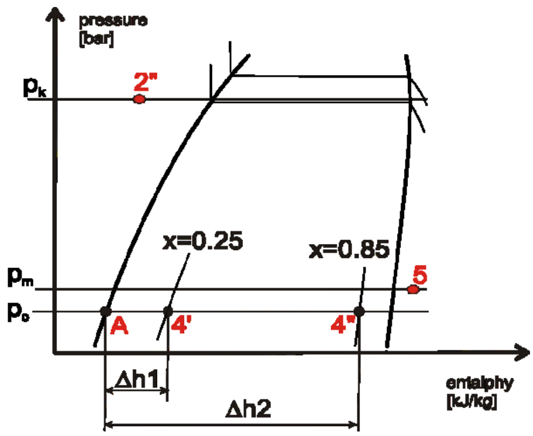

Figure 9. Cycle characteristic points when evaporator operates in the wet vapor range

Similar calculations as for Table 3 above were then conducted in order to determine the characteristics of the cycle with an ejector. Results are listed in Table 4 for x = 0.25 and in Table 5 for x = 0.85. The calculated values of mass entrainment ratio indicate clearly that this kind of evaporator operation is undesired because such cycle requires much higher values of mass entrainment ratios, which will be unachievable in practice.

Table 4. Characteristic points of a cycle with vapor quality x = 0.25

$p_k = p_{2"}$ [bar]	$h_k = h_{2"}$ [kJ/kg]	$p_m = p_5$ [bar]	p_o [bar]	Π [–]	h_5 [kJ/kg]	h_A [kJ/kg]	h_4 [kJ/kg]	U [–]
22	180.89	3.24	3.1	0.0074	421.83	161.02	223.62	3.849
			3	0.0126		159.86	222.66	3.837
			2.6	0.0330		154.91	218.55	3.786
			2.2	0.0525		149.37	213.94	3.731
			1.8	0.0713		143.02	208.63	3.672
			1.4	0.0893		135.51	202.32	3.606
			1	0.1067		126.12	194.4	3.529
			0.6	0.1234		113.14	183.4	3.429
			0.3	0.1355		97.576	170.14	3.320

Table 5. Characteristic points of a cycle with vapor quality x = 0.85

$p_k = p_{2"}$ [bar]	$h_k = h_{2"}$ [kJ/kg]	$p_m = p_5$ [bar]	p_o [bar]	Π [–]	h_5 [kJ/kg]	h_A [kJ/kg]	h_4 [kJ/kg]	U [–]
22	180.89	3.24	3.1	0.0074	421.83	161.02	373.81	1.132
			3	0.0126		159.86	373.33	1.129
			2.6	0.0330		154.91	371.24	1.114
			2.2	0.0525		149.37	368.85	1.098
			1.8	0.0713		143.02	366.05	1.080
			1.4	0.0893		135.51	362.64	1.061
			1	0.1067		126.12	358.25	1.038
			0.6	0.1234		113.14	352	1.009
			0.3	0.1355		97.576	344.25	0.977

Another cycle modification was studied, in which the evaporator superheats the vapor (assumed by 5K). This is illustrated in Figure 10 with cycle characteristic points of a cycle listed in Table 6. It appears that such modification could theoretically lower the required mass entrainment ratio due to increase in enthalpy difference.

Table 6. Characteristic points of a cycle with vapor superheat

$p_k = p_{2"}$ [bar]	$h_k = h_{2"}$ [kJ/kg]	$p_m = p_5$ [bar]	p_o [bar]	Π [–]	h_5 [kJ/kg]	h_A [kJ/kg]	h_4 [kJ/kg]	U [–]
22	180.89	3.24	3.1	0.0074	421.83	161.02	415.97	0.945
			3	0.0126		159.86	415.58	0.942
			2.6	0.0330		154.91	413.91	0.930
			2.2	0.0525		149.37	411.97	0.918
			1.8	0.0713		143.02	409.68	0.904
			1.4	0.0893		135.51	406.86	0.888
			1	0.1067		126.12	403.21	0.870
			0.6	0.1234		113.14	397.95	0.846
			0.3	0.1355		97.576	391.36	0.820

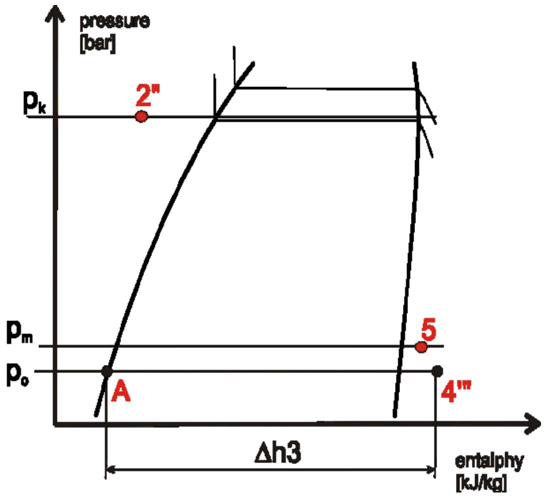

Figure 10. Cycle characteristic points when evaporator superheats the vapor

Indeed, comparing Tables 3 and Table 6 it can be concluded that vapor superheat offers the possibility to slightly lower the required values of mass entrainment ratio. Those values might be achievable in practice and therefore further laboratory experiments could justified. The author's research was however concluded at this point due to unavailability of further funding and the strong possibility that practical application and/or commercialization of those solutions may interfere with existing intellectual property.

Conclusions. Based on conducted investigations and theoretical calculations, a two-phase ejector may offer an attractive option for air-source heat pumps designed for Northern climate, to improve their performance and to broaden the range of outdoor temperatures at which such heat pumps operate. The ejector can be considered as the first-stage compression, however achieved without spending any mechanical energy. This feasibility study showed that the ejector was capable to extend the temperature range of typical air-source heat pump by approximately 10C comparing to the conventional design and when operating with R410A refrigerant. The major advantage will be to avoid more complicated and expensive two-stage compression by mechanical compressor. This research has determined that only several modes of operation may allow using an ejector, namely the cases when evaporator either saturates or superheats the refrigerant vapor. On the other hand, the wet vapor mode of evaporator operation require high mass entrainment ratios, that might be unachievable for ejector to operate.

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