2022

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ABSTRACT

Challenges associated with the mandatory lubrication of the refrigerant gas compressors have led to the development of the oil-less dynamic centrifugal compressors, which among undoubtful operation advantages have a major drawback - limits of the compression lift at single stage design for desirable condensation temperatures range, especially at most unfavorable ambient conditions. The application of two or more compression stages leads to a disproportional increase in material cost and unacceptable hydraulic losses. But even two-stage oil-less compressors will operate at refrigeration mode with 1.15-1.4 times lower to nominal COP if ambient temperature will apparently raise above 30°C (87°F). Such conditions increase the cost of water-or air-cooled condensation by an increased cooling media flow rates or mass-dimensional characteristics of the condensation heat release subsystems.

Internal cooling cycle potential as well as external (waste or renewable) energy sources can be utilized to support compressor operation at higher compression ratios and ambient temperature conditions compared to the designed ones. The internal cycle potential ensures an improvement of the oil-less compressor cooling system's COP without any additional energy input. It only requires equipping the system with simple and inexpensive booster ejector devices. For this scenario, reduction of the compressor power consumption at the same cooling capacity production rate is achieved by two means. The first one refers to a ballast vapor or at least its main fraction to produce additional compression work in the booster ejector or recirculate it via the second stage of a compressor, bypassing the first stage. The other mean is an additional subcooling of the main liquid flow by evaporation of another portion of the working fluid at the intermediate temperature. The evaporated vapor flows to the second suction stage of the compressor or is added to ballast vapor to perform a compression of the main vapor flow from the evaporator. The integral COP of the Booster Ejector Enhanced Compressor Refrigerating system (BEECR) using the oil-less compressor increases by 10-20% and highly depends on operating conditions of the refrigerating system. With an external low-grade heat driven booster ejector subsystem, the overall COP of BEECR increases by additional 5-15% depending on the heat grade, working fluids properties and schematic solution applied.

The article provides an analysis of various vapor-compression refrigerating systems (VCRS) performance enhancement approaches, characteristics of simulated ejectors, technical and economic feasibility of the most effective options for the conditions of Danfoss Turbocor oil-less compressor-based industrial chillers operating at evaporation temperature of -6.5°C (20°F), condensation temperature of 30°C (87°F) to 51°C (124°F), and variable cooling capacity of 192.7 to 220.3 kW (54.8-62.6RT), operating on R134a and R513a refrigerants.

The presented study describes how COP of BEECR depends on operating parameters of the system and selected refrigerant's thermophysical properties.

The study showed that support of the main compressor by enhancing with booster ejector devices on both newly built or retrofitted VCRSs allows decreasing power consumption by a minimum of 15%, which leads to saving of about 2.5% of overall consumed electricity by chillers, improves reliability and durability of the compressor and, as a result, lowers the impact on environment.

1. INTRODUCTION

Artificial cooling became one of the most demanded energy sources in the world. Its production associated with high electric or heat energy intensity and the share of its costs is increasing fast disproportionally. Nowadays, no infrastructure is conceivable without chillers and air-conditioners, including industries, commercial and residential buildings and data centers. The urgent needs to diversify supplies of natural gas caused a rapid growth of LNG production and a high-capacity fleet of gas carriers. Increasing population and its prosperity, especially in low-income communities, results in the quick growth in electricity consumption for cold production.

Most of the cooling capacity production is conducted by the vapor compression refrigerating systems, while emerging oil-less compressors gain its popularity due to absence of lubrication. Such centrifugal turbocharged compressors have a major disadvantage though - low compression lift (typically 4-5.5) comparing to other existing compressors types. This circumstance makes impossible the application of air-cooled condenser for oil-less chillers in hot climates or lower the evaporation temperature that limits its applications. A return to water cooled condensers is barely realistic, since the cooling towers construction is associated with high capital investments and water scarcity also greatly limits its use in most of the cooling demanded areas. So, the best solution of the problem is a development of economic and enhanced compressor systems to operate in an optimal pressure range with minimized power and water consumption.

The proposed ejector-based technology can be used to enhance a compressor system design and efficiency. The first known approaches to use ejectors in vapor-compression cooling systems were dated in a mid of 20th century. Schematics presented on Figure 1(a) were proposed by Badylkes (1961) and it used a portion of the compressed vapor after compressor to improvement the system's energy characteristics operating in extreme ambient conditions. Another presented diagram by Buyadgie (2010) does not require additional energy input and use the internal cycle potential (ballast vapor) to boost the system's efficiency (Figure 1(b)). Various schematics were described and analyzed by Elbel et. al. (2008), Buyadgie et. al. (2011, 2018), Buyadgie (2016), which served to improve efficiency of the cooling cycle by 10-20% and even higher, depending on operating parameters of the refrigerating system.

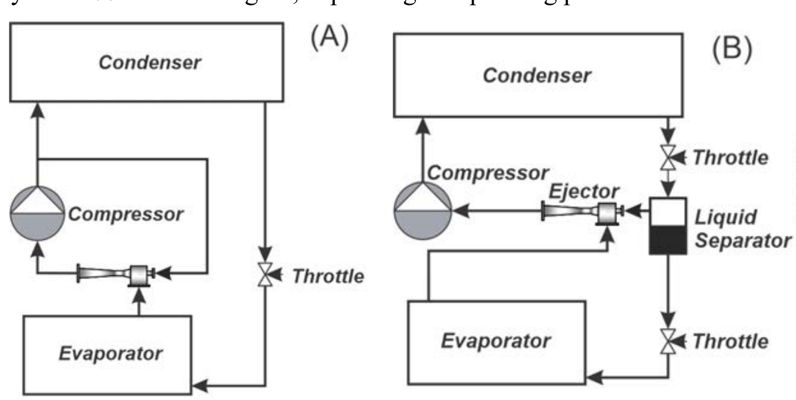

Figure 1: (a) Booster Ejector Enhanced System by Badylkes; (b) Booster Ejector Enhanced System by Buyadgie

It should be noted that, those methods were based on three main principles: to provide a direct or indirect subcooling before throttling, minimize amount of ballast vapor to enter the evaporator; and to utilize exergy of ballast vapor for pre-compression of vapor before the compressor. The described methods showed promising results for a single stage compressor, which were designed for low temperature cold generation (-20°C to -40°C and lower) and high condensation temperatures 35°C-50°C (Buyadgie et.al., 2018). In moderate climate zones such systems could operate at high efficiency during favorably low ambient conditions, while its cooling efficiency dropped significantly during high ambient temperatures periods, which led to disbalancing of freezing or storage regimes. In such scenarios, it was irrational to apply a two-stage compressor at all, since a second stage might not be used most of the time. An introduction of an oil-less centrifugal compressors allowed enhancing a compressor system both by making changes to the internal modernization of the cycle and by installing an auxiliary refrigerating system. Such an auxiliary low-grade heat driven Ejector Refrigerating System (ERS), improves energy efficiency of the main vapor compression cycle by additional compression of vapor after compressor and by additional subcooling of the liquid before its throttling to the evaporator, which allows to reduce energy consumption by decreasing the volume of the ballast vapor entering the evaporator.

2. BASIC EJECTOR ENHANCED SYSTEM WITH TWO-STAGE OIL-LESS CENTRIFUGAL COMPRESSOR.

When design of the oil-less centrifugal compressor refrigerating system includes economizer, a portion of the liquid refrigerant evaporates at intermediate pressure and provides subcooling of the main fluid flow to nearly saturation temperature at an intermediate pressure. Vapor from an economizer is supplied to a second suction stage of the compressor to reduce the compressor's first stage load, due to the decreased fluid flow from the evaporator at the constant cooling capacity generation. A cycle efficiency improvement only from employing the economizer leads to 5-12% efficiency gain as presented on Figure 2.

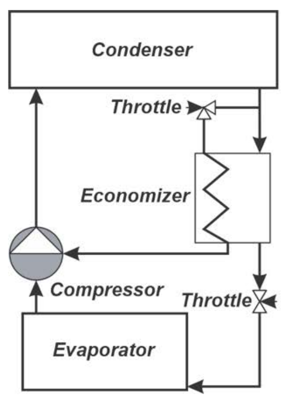

Figure 2: Schematics of vapor compression system with economizer and two stage compressor.

Two-stage compressor allows supplying almost all the ballast vapor directly to the second suction stage significantly avoiding power consumption by the compressor on the first stage. Comparing with a single stage compressor system, all the vapor, including ballast after throttling and vapor from the evaporator, have to be compressed from suction to discharge pressure at no savings. Though an additional work from the ballast vapor expansion after condensate throttling to intermediate pressure can be recovered, it also will require a significant amount of compressor work for its re-compression from evaporation to suction pressure level, which reduces the effect of ballast vapor exergy. In modes, where the difference between evaporation and condensation temperatures is below 60°C, efficiency improvement by a single-phase ejector does not exceed 3-4%. By increasing the operating temperatures range, an efficiency improvement may reach 15-20%, while in a system with a two-stage compressor, a single-phase ejector application allows increasing the system's efficiency up to 12-15% at relatively lower operating temperatures range (Figure 3a).

Promising results of compressor performance enhancement, operating at evaporation temperature t_{eva}=-6.67°C and condensation temperature t_{cond}=51°C, were obtained by using two-phase ejector (Figure 3b). This schematic does not require throttling to an intermediate pressure and allows reducing the subcooling temperature down to 7°C, that eliminates additional throttling losses. Two-phase ejector entrains ballast vapor from the economizer and uses a vapor obtained from motive liquid flow expansion in the ejector's nozzle to supply it to the second suction stage of the oil-less compressor. Thus, only a vapor from the evaporator, which generates cooling capacity, flows through a first suction stage of the compressor and reduces the overall compressor load.

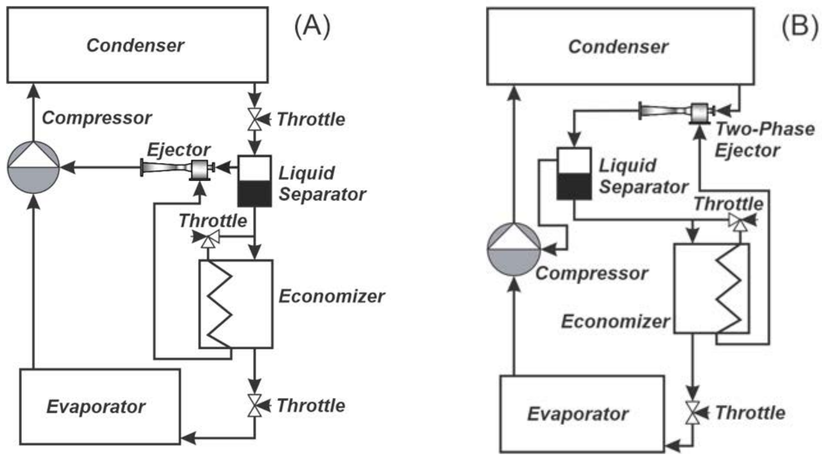

Figure 3: (a) Booster Ejector Enhanced System (option 2), (b) Booster Ejector Enhanced System (option 3)

3. USING EXTERNAL LOW-GRADE HEAT IN BEECR

Another schematic solution driven by external heat source can be compression of discharged vapor to a higher pressure by an additional ejector in ERS. Various modifications of ERS that utilizes exhaust heat from the industrial processes and produces a surplus cooling capacity or increase efficiency of the existing system are available. The presented schematics (on Figure 4) are just few options of those. The feasibility of schematics selection is determined by the payback period for additional capital costs recovery and value of the available low-grade heat. System with a booster ejector after compressor (Figure 4a) protects compressor against increased condensation temperature during peak load periods and guarantees its stable operation throughout the entire operating time, which significantly increases its durability and lowers maintenance costs compared to initially designed conditions.

Second option (Figure 4b) is intended to cover unexpected cooling load increase and expands the capabilities of the BEECR beyond its design parameters while stabilizing the technological standards in extreme and/or emergent situations and increasing the profitability of the industry at no added operating costs. It is especially feasible when out of several modules in use one is forced to be removed for repairs or maintenance, or for any other safety reason.

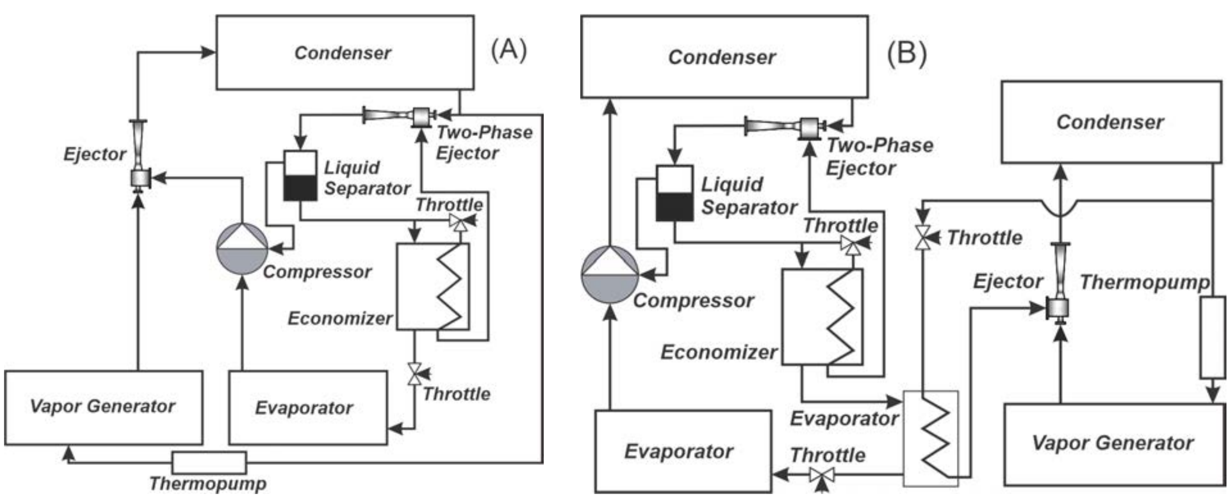

Figure 4: (a) Booster Ejector Enhanced System (option 4), (b) Booster Ejector Enhanced System (option 5).

Third option (Figure 4c) is a combination of option described in Figures 4a and 4b where along with additional fluidical compression after the main mechanical vapor-compressor, another heat driven ejector loop provides additional subcooling to generate extra cooling capacity. This schematic solution is more flexible, since it can operate in a wide range of operating parameters depending on heat source availability. As a results, an additional cooling capacity or reduction in compressor power consumption, or simultaneously both additional cooling capacity and reduction in compressor power consumption can be achieved by applying a low-grade heat.

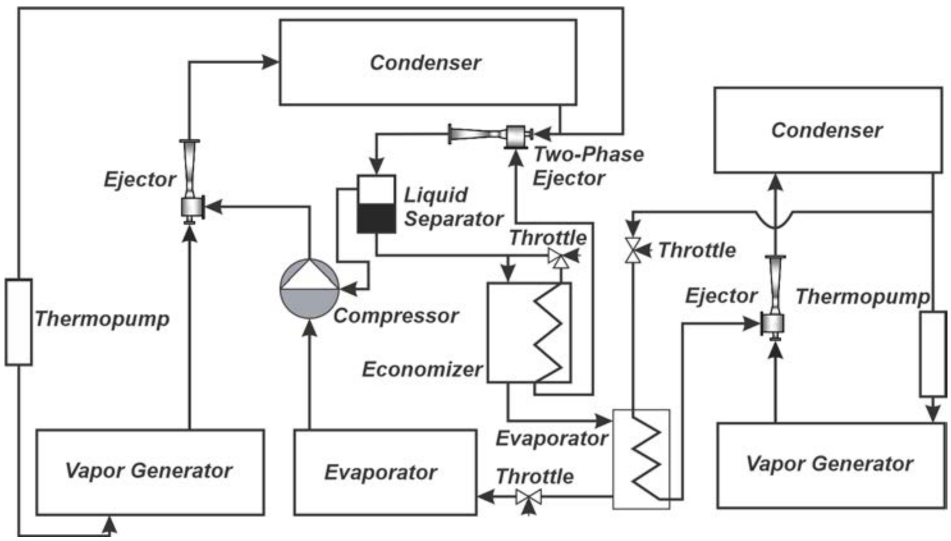

Figure 4: (c) Booster Ejector Enhanced System (option 6).

4. EJECTORS AND EFFICIENCY VALIDATION

4.1 Ejector Calculation

The ejector calculation was performed based on Sokolov and Zinger (1989), Gabuz (1984) and Badylkes et al. (1961) approaches. The optimal operating modes were determined by the character of expansion of the working fluid at the ejector's supersonic area, which is described by the relation of suction and working pressure ratio and critical Π function. The ejector operates in one of the following modes, either as a low lift ejector with high entrainment ratio and low compression:

$$P_s / P_p < \Pi^* \tag{1}$$

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

or as an ejector with low entrainment ratio and high compression ratio:

$$P_s / P_p < \Pi^* \tag{3}$$

To calculate the achievable pressure behind the ejector and the geometry of the ejector's flow profile, the following algorithm was proposed, eq. 4-13 (Sokolov and Zinger, 1989).

$$\frac{\Delta P_{out}}{P_s} = \frac{k_p}{2\left(k_p+1\right)} \frac{1}{\Pi_{p.s.}} \frac{\varphi_1^2 \varphi_2^2 \lambda_{p.s.}^2}{\left[\left(1/\varphi_3 - 0.5\right) v_c / v_p \left(1+U\right)^2 - \left(\varphi_2 \varphi_4 - 0.5\right) v_s / v_p n U^3\right]} \tag{4}$$

$$f_3 / f_* = \left(-b + \sqrt{b^2 - 4ac}\right) / \left(2a\right) \tag{5}$$

$$a = \varphi_1 \varphi_2 q_{p.s.} \tag{6}$$

$$b=-\left\{\varphi_1\varphi_2+2\varepsilon_{p.s.}\left[\left(1/\varphi_3-0.5\right)v_c/v_p\left(1+U\right)^2-\left(\varphi_2\varphi_4-0.5\right)v_s/v_pU^2\right]\right\} \tag{7}$$

$$c=2\varepsilon_{p.s.}/q_{p.s.}\left(1/\varphi_3-0.5\right)v_c/v_p\left(1+U\right)^2 \tag{8}$$

$$\frac{\Delta P_c}{P_s}=\frac{0.5k_p\varepsilon_{p*}\Pi_{p*}v_s/v_p}{\varphi_4^2\Pi_{p.s.}\left(f_3/f_{p*}-1/q_{p1}\right)^2}U^2 \tag{9}$$

$$\Delta P_d/\Delta P_c=\left(\left(1+U\right)/U\right)^2\times v_c/v_n \tag{10}$$

$$v_c/v_p=P_pT_c/\left(P_c/T_p\right) \tag{11}$$

$$v_s/v_p=P_pT_s/\left(P_s/T_p\right) \tag{12}$$

$$T_c=\left(T_p+UT_s\right)/\left(1+U\right) \tag{13}$$

4.2 Compressor

The evaluation of the centrifugal compressor power consumption was proposed by Bykov et.al. (1992). Compression ratio of the compressors is defined by equation 14:

$$\pi_c=P_{out}/P_s \tag{14}$$

The coefficient of the specific volume change in the compressor during isentropic compression defined by equation 15:

$$k_{v,ad}=v_{n,out}/v_{out,ad} \tag{15}$$

A conventional adiabatic index of compression process is given in equation 16:

$$k_v=\lg\pi_c/\lg k_{v,ad} \tag{16}$$

A polytropic efficiency η_{pol} of the compressor's flow part is required to define an adiabatic efficiency η_{ad} of the compressor. The minimal polytropic efficiency of the compressor with various angle of blades and Mach number is up to 70%. The polytropic efficiency of the compressor with an outlet angle on the blades of 32° equals to 81% (Bykov et.al., 1992).

Adiabatic efficiency of the compressor's flow part calculated using equation 17:

$$\eta_{ad}=\left(\pi_c^{1-1/k}-1\right)/\left(\pi_c^{(k-1)/(k\eta_{pol})}-1\right) \tag{17}$$

Specific isentropic compression work is defined by equation 18:

$$l_{ad}=h_{c,ad}-h_s \tag{18}$$

A real compressor work for polytropic compression process is defined by equation (19)

$$l_{comp}=l_{ad}/\eta_{ad} \tag{19}$$

Indicated power of the compressor is calculated using equation 20:

$$W_{comp} = m_s l_{comp} \tag{20}$$

4.3 Efficiency calculation

Two schematics with single and two-phase ejectors, as described on Figure 3, improve the efficiency of the BEECR system using available internal sources of the cycle only. The BEECR utilizes exergy of the refrigerant flow, which expands in the ejector's nozzle and compresses a portion of the ballast vapor flow from the evaporation of a respective liquid portion to sub-cool the main liquid flow before its throttling to the evaporator. In this case, only a minimal amount of a ballast vapor enters the evaporator and increases a specific cooling capacity of the cycle while reducing the compressor work at the first stage by lowering the rate of fluid flow through it.

The COP of the two-stage compressor system with economizer is defined by equation 21:

$$COP_{el}^{VCRS\,w.eco} = \frac{Q_{eva}}{W_{comp}} = \frac{G_{eva}\left(h_{eva,out} - h_{eva,in}\right)}{G_{eva}\left(h_1 - h_{eva}\right) + G_{eva}\left(1+\sigma\right)\left(h_{comp,out} - h_{1,mix}\right)} \tag{21}$$

$$\sigma = G_{eco} / G_{eva} \tag{22}$$

The COP of the booster ejector enhanced system with single-phase ejector (BEECR 2) is defined by equation 23:

$$COP_{el}^{BEECR2} = \frac{Q_{eva}}{W_{comp}} = \frac{G_{eva}\left(h_{eva,out} - h_{eva,in}\right)}{G_{eva}\left(h_1 - h_{eva,out}\right) + G_{eva}\left(1+\sigma\left(1+1/U\right)\right)\left(h_{comp,out} - h_{1,mix}\right)} \tag{23}$$

The COP of the booster ejector enhanced system with two-phase ejector (BEECR 3) is defined by equation 24:

$$COP_{el}^{BEECR3} = \frac{Q_{eva}}{W_{comp}} = \frac{G_{eva}\left(h_{eva,out} - h_{eva,in}\right)}{G_{eva}\left(h_1 - h_{eva,out}\right) + G_{eva}\left(1+\sigma\left(1+1/U\right)x\right)\left(h_{comp,out} - h_{1,mix}\right)} \tag{24}$$

Utilization of a low-grade heat at BEECR provides an additional fluidic compression by the ejector after the conventional mechanical compression and/or operates ERS to support the compressor cycle for additional cooling capacity generation. Both approaches serve to increase the efficiency of the compressor cycle.

In the first case, decrease of a compression ratio of the main compressor is observed with an increase of ambient temperature above the designed conditions. Improving an electrical efficiency of the compressor by decreasing its compression work at the same cooling capacity generation is achievable with booster ejector that performs fluidic compression at the expense of the low-grade heat potential. In the second scenario the low-grade heat driven ERS, operating in parallel with a mechanical compression cycle, provides either additional subcooling before throttling to evaporator to generate greater cooling capacity or provides compression before or after compressor to improve the compressor performance.

Compressor's work for both BEECR 4 and BEECR 5 was defined using equation 24. Heat consumed by BEECR 4, BEECR 5, BEECR 6 schematics are defined by equations 25, 26, and 27 respectively:

$$Q_{gen}^{BEECR4} = \left(G_{eva}\left(1+\sigma\left(1+1/U\right)x\right)\right)/U_{th} * \left(h_{gen,out} - h_{gen,in}\right) \tag{25}$$

$$Q_{gen}^{BEECR5} = G_{eva,th} / U_{th} * \left(h_{gen,out} - h_{gen,in}\right) \tag{26}$$

$$Q_{gen}^{BEECR6} = Q_{gen}^{BEECR4} + Q_{gen}^{BEECR5} \tag{27}$$

4.4 Ejector Modeling

A high-pressure subcooled liquid is supplied to ejectors and boils up during expansion gradually depending on the level of condensate subcooling. The geometry of two-phase nozzle usually differs from the single-phase one. However, the profile of the two-phase ejector's nozzle in the studied case is explained by a sharp change of the speed of sound in the liquid and vapor-liquid flow. First of all, the profile and dimensions of a suction chamber, along with a length of mixing chamber, underwent meaningful changes: length of mixing chamber of two-phase ejector is up to 20 calibers, while a single-phase vapor ejector is of 6-10 calibers only.

The higher compression ratio of a two-phase ejector comparing to a single-phase ejector, driven by a throttled vapor is explained by higher motive flow condensation pressure, and absence of throttling losses after condenser, even taking into account a lower vapor content in the vapor-liquid mixture supplied though the nozzle.

A CFD modeling of ejector's flow part allows verification of the in-house code generated calculation results and optimalization of ejector's flow profile geometry. The initial geometry of ejector's flow part is based on the following parameters: temperature, pressure, density, adiabatic indexes of the working, secondary and outlet flows. The calculated initial parameters are applied to a design modeler, where a primary design geometry is created. Initial modeling and optimization processes are conducted on a coarse mesh. The coarse mesh is built-in by Ansys Meshing tool, the number of elements for a coarse mesh is limited to 600,000 elements. The tetrahedral mesh with prism elements is located near the walls (inflation areas). Due to the high velocities of the working flow at the areas of the diverging part of the nozzle and mixing chamber (where pressure shock loss is observed), the local sizing with smaller element sizes is applied from the nozzle's critical cross section to the center of a cylindrical mixing chamber. Modeling was performed using ANSYS CFX for a steady state process, when heat transfer is set to total energy. The selected turbulence model is SST k-ω, which provides a good solution for a free and laminar flows.
Examples of single phase and two-phase ejector are shown on Figure 5.

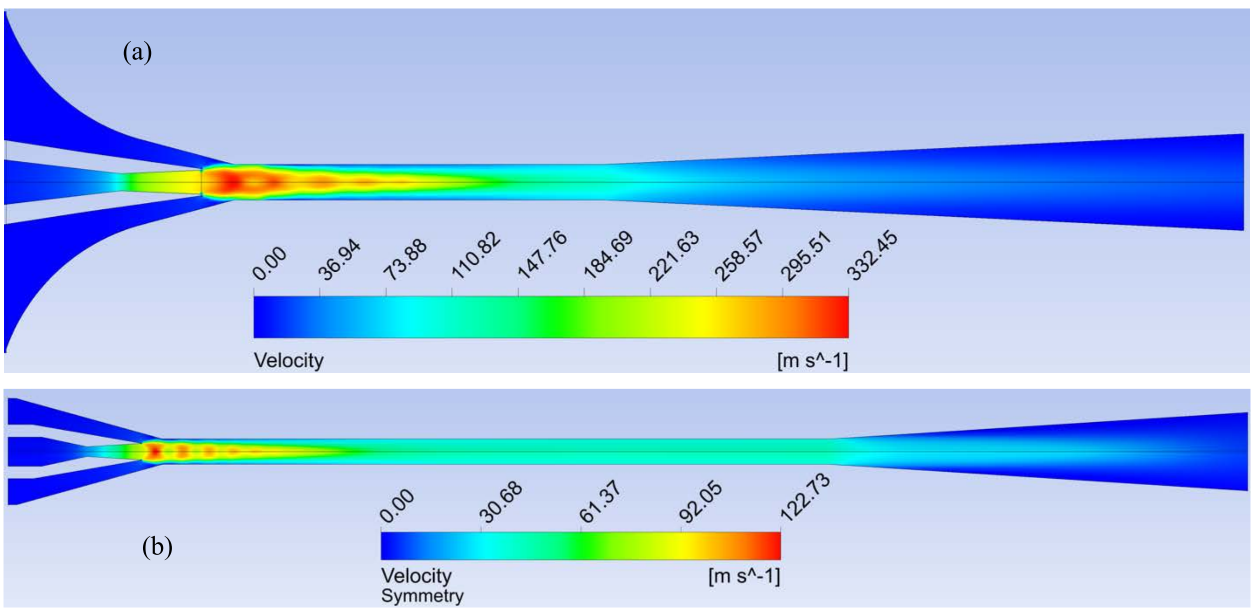

Figure 5: (a) Velocity distribution in a single-phase ejector, (b) Velocity distribution in a two-phase ejector.

5. SYSTEMS EFFICIENCY EVALUATION AND COMPARISON

A comparative analysis based on characteristics of the commercially available oil-less compressor chiller using R134a and R513a, demonstrated the advantages of booster ejector enhancement technology. Calculation results are presented in Table 1, including cooling loads and efficiencies, compressor power consumption, flow rates, condenser loads and generators heat input, ejector entrainment ratios, etc. Table 2 represents results of the calculations for R513a.

All the calculations were performed for the 60 refrigerating tons (RT) or 211kW cooling capacity vapor compression cooling system at -6.5°C of evaporation temperature and maximum available low-grade heat capacity of 300kW at 90°C. Evaporation and condensation parameters selected at maximum possible temperature range that oil-less compressor can operate with. Evaporation temperature of 0°C for BEECR 5 and (-3°C) for BEECR 6 was set. Low evaporation temperature for subcooling in BEECR6 is caused by lower liquid temperature achieved in the economizer. Generation temperature for all heat driven sub-systems was set as 85°C. Condensation temperature for the thermally driven subcooling loop (BEECR 5 and BEECR 6) was set to 35°C. For the BEECR 4, operating parameters corresponds to system for BEECR 3. For the BEECR 5, operating parameters and mass flow rates correspond to the parameters of BEECR 3, with additional subcooling down to 2°C before throttling to the evaporator. A preliminary layout of the BEECR 5, excluding thermally driven loop components, is represented on Figure 6.

As per Table 1 and Table 2, efficiencies of the system do not significantly depend on fluid, however the much lower environmental impact of R513a makes it a highly perspective fluid for the next 15-20 years compared to nearly phased out R134a.

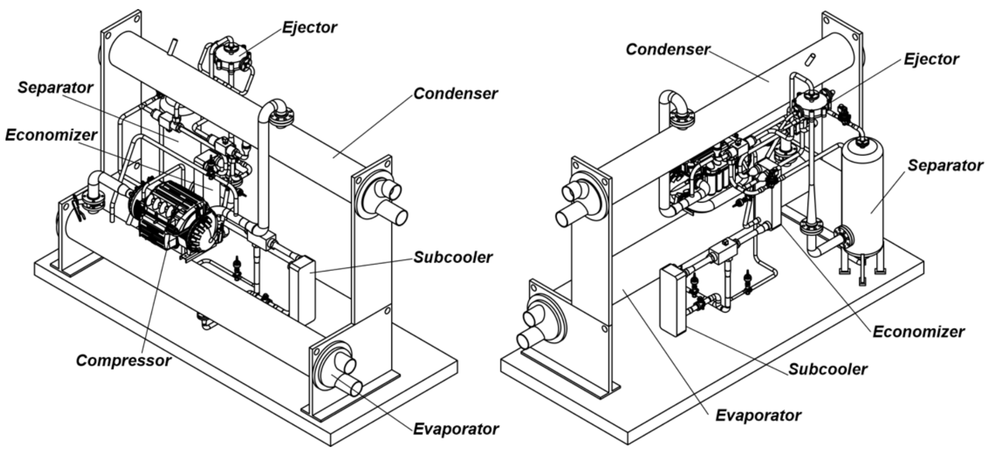

Figure 6: Schematic diagram of the BEECR 5 with a two-phase ejector.

Table 1: Comparison of the studied oil-less vapor compression system and booster ejector enhanced technology variants, operating on R134a

Parameters	VCRS w/o economizer	VCRS w. economizer	BEECR2	BEECR3	BEECR4	BEECR5	BEECR6
Evaporation T, °C	-6.5	-6.5	-6.5	-6.5	-6.5	-6.5	-6.5
Condensation T, °C	51	51	51	51	51	51	51
Condenser Outlet T, °C	46	46	46	46	46	46	46
Economizer/Subcooler Outlet T, °C	-	23/-	19/-	7/-	4/-	7/2	4/-1
Condenser Mass Flow Rate, kg/s	1.48	1.53	1.58	1.56	3.15	1.56/ 0.25**	3.15/ 0.28**
Evaporator Mass Flow Rate, kg/s	1.48	1.28	1.25	1.13	1.11	1.13	1.1098
Entrainment Ratio	-	-	1.16	0.064	0.06/1*	0.064/ 0.26**	0.064/1*/ 0.23**
Condensation Heat, kW	276.5	294.11	293.2	289.4	526.7	339.4	581.85
Cooling Capacity, kW	192.7	211	211	211	211	218.5	218.5
Compressor Power Consumption, kW	83.8	83.11	82.2	78.4	72.5	78.4	72.5
Heat Consumption, kW	-	-	-	-	243.2	35	283.35
COP_{el}	2.3	2.54	2.57	2.69	2.91	2.79	3.01

Efficiency Gain, %		+10.4%	+11.6%	+17.0%	+26.6%	+21.2%	+31.0%

* Entrainment ratio of heat driven ejector as per BEECR (option 4) or ** as per BEECR (option 5). VCRS w/o economizer represents a baseline for comparison.

Table 2: Comparison of the studied oil-less vapor compression system and booster ejector enhanced technology variants, operating on R513a

Parameters	VCRS w/o economizer	VCRS w. economizer	BEECR2	BEECR3	BEECR4	BEECR5	BEECR6
Evaporation T, °C	-6.5	-6.5	-6.5	-6.5	-6.5	-6.5	-6.5
Condensation T, °C	51	51	51	51	51	51	51
Condenser Outlet T, °C	46	46	46	46	46	46	46
Economizer/Subcooler Outlet T, °C	-	22.4/-	19.2/-	7.4/-	4.2/-	7.4/2	4.2/-1.5
Condenser Mass Flow Rate, kg/s	1.78	1.90	1.90	1.87	4.08	1.87/ 0.33**	4.08/ 0.31**
Evaporator Mass Flow Rate, kg/s	1.78	1.48	1.40	1.30	1.26	1.30	1.26
Entrainment Ratio	-	-	1.30	0.07	0.07/ 1.2*	0.07/ 0.28**	0.09/1.2*/ 0.24*
Condensation Heat, kW	278.3	296.2	294.1	290.6	529.8	350.2	597.2
Cooling Capacity, kW	192.7	211	211	211	211	220.28	220.28
Compressor Power Consumption, kW	85.64	85.18	83.14	79.6	73.5	79.6	73.5
Heat Consumption, kW	-	-	-	-	245.3	41	294.1
COP_{el}	2.25	2.48	2.54	2.65	2.87	2.77	3.0
Efficiency Gain,%	-	+10.1%	+12.8%	+17.8%	+27.6%	+23.0%	+33.2%

* Entrainment ratio of heat driven ejector as per BEECR (option 4) or ** as per BEECR (option 5). VCRS w/o economizer represents a baseline for comparison.

6. CONCLUSIONS

The performed analysis of the variety of options to enhancement the oil-less compressor chiller with booster ejectors was presented in this paper. Integration of the BEECR into mechanical compressor refrigerating systems (either newly built or retrofitted) may become an important tool for energy efficiency increase, lowering mechanical work consumption, lead to a cost savings and reduction of carbon footprint. Provided increasing contribution of cooling production on a global energy scale, the proposed modernization of refrigerating systems will reduce energy consumption to power compressors by 10-15% that can save more than 2.5% of total energy consumption by the industry and reduce CO_2 and other emissions correspondingly.

The considered systems increase durability of the compressors by 1.6-2 times, which also contributes to saving of energy resources and reducing emissions for its productions, maintenance and replacement. At the same time, almost no additional costs are required for BEECR servicing, similarly, to limited costs, associated with the waste heat utilization.

After successfully testing several prototypes, presented in the past studies, the next steps will be a full scale demonstration of the technology at the selected food manufacturing facilities, followed by data collection and analysis of the pilot performance of the BEECR system in the real environment.

NOMENCLATURE

COP	coefficient of performance	(-)
G	mass flow rate	(kg/s)
k	adiabatic index	(-)

P	pressure	(kPa)
t	temperature	(°C)
T	temperature	(K)
U	entrainment ratio	(kg/kg)
W	compressor power	(kW)
$\varphi_1,\varphi_2,\varphi_3,\varphi_4$	velocity coefficients	(-)
Π	relative pressure function	(-)
π	compression ratio	(-)
λ	relative velocity function	(-)
v	specific volume	(m^3/kg)
ε	relative density function	(-)
η	efficiency	(-)
x	dryness degree of vapor-liquid mixture	(kg/kg)

Subscript

1	parameter after compression at stage 1
ad	adiabatic
c	compressed
cond	condensation
comp	compressor
d	diffuser
eva	evaporation
el	electrical
gen	vapor generation
m	motive flow
mix	mixed flow parameter
out	outlet parameter
p	motive flow
p1	parameter of the motive flow at the outlet of ejectors nozzle
p.s.	parameter of the motive flow at the inlet of ejectors mixing chamber
pol	polytropic
red	reduced
s	suction flow
th	thermally driven

Supercript

VCRS w. (w/o) eco	vapor compression refrigerating systems with (without) economizer
BEECR 2	booster ejector enhanced compressor refrigerating system, option 2
BEECR 3	booster ejector enhanced compressor refrigerating system, option 3
BEECR 4	booster ejector enhanced compressor refrigerating system, option 4
BEECR 5	booster ejector enhanced compressor refrigerating system, option 5
*	critical parameter

REFERENCES

Badylkes, I.S, Danilov R.L. (1961). *Cooling systems using steam-jet devices as a booster compressors*. Moscow: Energia.

Buyadgie, Dmytro; Sechenyh, Vitaliy; and Nichenko, Sergii. (2010). "Booster Vapor-Compression Refrigerating Systems". *Proceedings of the International Refrigeration and Air Conditioning Conference.* Paper 1062.

Buyadgie, O. (2016). *The Thermotransformation Processes Efficiency Enhancement in The Self-Contained Ejector Refrigerating Machines*, Ph.D. Thesis, The Odessa National Academy of Food Technologies, V.S. Martynovsky Institute of Refrigeration, Cryogenic Technologies and Eco-Energetics, Odessa, Ukraine.

Buyadgie D.I., Buyadgie O.D. (2011). The Intallation to Produce Heat or Cold (Variants) (in Ukrainian), Patent for an Invention: UA 103365 F25B 1/02 F25B 1/06. № a 2011 10382, applied for 26.08.2011, published on 10.10.2013, Bul. №19.

Buyadgie, Olexiy; Nesterov, Pavlo; Drakhnia, Oleksii; and Buyadgie, Dmytro. (2018). "Experimental Study of a Low-Temperature Compressor-Ejector Refrigeration System". *International Refrigeration and Air Conditioning Conference*. Paper 2074.

Bykov, A., Bedjanishvili, E., Kalnin, I. et.al. (1992). *Refrigeration compressors. Guidebook*. Moscow: Kolos.

Elbel, Stefan and Hrnjak, Predrag (2008) Ejector Refrigeration: An Overview of Historical and Present Developments with an Emphasis on Air-Conditioning Applications. *Proceedings of the International Refrigeration and Air Conditioning Conference*. Paper 884

Garbuz, A. (1984). *Low pressure gas ejectors for electro-chemical generators*, PhD Thesis, Odessa Technological Institute of Refrigeration Industry, Odessa, Ukraine

Sokolov, E. Y., & Zinger, N. M. (1989). *Jet Devices* (3rd ed.). Moscow: Energoatom Publiching House.