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NUMERICAL DESIGN AND OPERATION OF THE EJECTION REFRIGERATION SYSTEM DRIVEN BY LOW-GRADE HEAT

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ABSTRACT

Heat recovery is a common approach for effective energy management. Recently, the customers pay more attention to reduction of consumption of electric energy. On the other hand, development of technology leading to reduction of demand for electricity in industry is more and more popular. The paper summarizes the first phase of the application project dealing with developing of the ejection air-conditioning system driven by waste heat. Preliminary calculation of the proposed system operating with low-GWP working fluid will be presented and discussed. The geometry of the ejector designed for the specific case and performance operation line will also be shown.

Keywords: ejector, air-conditioning, waste heat recovery

1. INTRODUCTION

Increasing the energy demands, development of energy conversion systems as well as development the industry and technology resulting an increase in rejection of waste heat. This creates a perfect condition for development of the systems suitable for utilization fo waste heat. Air-compression systems generates a lot of megajouls of heat. The utilisation of significant amounts of heat generated by these systems is a real technical challenge due to the fact that it is in major a low-temperature heat at the level of 85 degrees C, or less even at a level lower than 70 degrees C. For this reason, as well as limitation of the conversion of this heat into electricity, a rational course of action is to use this heat for other technically and economically useful purposes than the production of electricity. One of the directions is the conversion of this heat into cooling capacity, which can be used for technological cooling or air-conditioning of industrial facilities, both in the standard conditions, where the chilled water temperature of 6 degrees C for air-conditioning units is required and also new air conditioning solutions for rooms (including industrial rooms) with air conditioning high-temperature (requiring a temperature of 16 degrees C). Paper presents development of gas ejector proposed for air-conditioning technology that utilize the industrial waste heat. Ejection refrigeration system is a modification of a well-known vapor compression cycle. Instead of pressurizing the refrigerant by a mechanical compressor, an ejector compresses refrigerant vapor flowing from evaporator and discharges it to the condenser. The motive vapor is generated in the vapor generator which is heated by low-temperature heat source. The main difference between the ejection cycle and the conventional refrigeration cycle, besides elimination of a compressor, is that the ejection cycle requires three heat sources at different temperatures, namely the vapor generator level, which is the temperature of the solar or waste heat source, a condensation level, which is the ambient temperature (actually this is a heat sink) and the evaporation temperature required for desirable cooling effect.

The basic parameters describing the ejection cycle performance in this paper are mass entrainment ratio U, Eq. (1), and condensation temperature t_c:

$$U = \frac{\dot{m}_e}{\dot{m}_g} . \tag{1}$$

The performance of the ejector depends on several quantities such as: operating pressures and temperatures at the ejector inlets and outlet, working fluid properties and ejector geometry.

The performance of the ejector depends on several quantities such as: operating pressures and temperatures at the ejector inlets and outlet, working fluid properties and ejector geometry. Many research teams have numerically and experimentally investigated the performance of the ejector cycle for various working fluids and operating parameters, e.g. Pridasawas and Lundqvist (2003), Butrymowicz et al. (2008), Gagan et al. (2018), Śmierciew et al. (2014). The efficiency of the ejector as a function of the ejector geometry were numerically as well as experimentally tested by several research teams, e.g. Zhu et al. (2009), and Varga et al. (2009).

This paper deals with own numerical studies resulting the final geometry of supersonic gas ejector operating in an ejection air-conditioning system driven by low-temperature waste heat from an air compression system. Selection of the working fluid for proposed system was based on the analysis of the energy efficiency of the system for the following fluids: isobutane, propane, R1234zeE, R1234yf, R1233zdE and R1336mzzz. The main criteria for the working fluid was GWP < 150 accoring to Regulation (EU) No 517/2014. However, taking into account the toxic and flammability classification, fluids belonging to A1 group are recommended. For this reason refrigerants R1233zdE and R1336mzzZ have been taken into consideration only.

The exemplary results of numerical studies were presented in the paper. The investigation covers the on-design and off-design operating regime of the ejector. The ejector operates at on-design condition in the case of choked both streams: primary (motive) stream and secondary stream. This operating regime covers the backpressure which is corresponding to the condensation pressure lower than some specific value of the pressure called the critical pressure p_c. At this regime, the ejector entrains the maximum amount of vapour from the evaporator, and the mass entrainment ratio U is constant. The critical pressure depends on the ejector geometry and the inlet parameters at both inlets of the ejector. The back-pressure higher than the critical pressure p_c makes the ejector starts to operate at the off-design condition. At this operating regime, the secondary flow is not choked already and the mass entrainment ratio U decreasing with increasing of the backpressure.

The assessment of the performance of the ejector as relationships between mass entrainment ratio and condensation temperature is presented in the paper.

2. ANALYSIS AND MODELLING

Development of ejector geometry was carried out in 3 stages. In the first step, for given operating conditions the main dimensions of the ejector were estimated using 1D model. In the second step these geometic parameters were used for development of numerical model for CFD simulation. Last step of ejector designing procedure was used for improvement of ejector geometry. Two sets of operating conditions have been taken into consideration (Table 1). In general, the designed systems will be driven by 200- and 600-kW heat source and should provide 35 kW and 125 kW of cold, respectively. Two levels of heat source temperature are takien into consideration: 80°C and 150 °C. The later is a supercritical value of temperature for both analyzed refrigerants. Evaporation temperature was 3°C. Taken into consideration the possible combination of two sets of motive power, two levels of motive temperature and two refrigerants a set of $2^3 = 8$ ejectors have been calculated.

Table 1. Operating conditions for ejection system

Q_{g1}	200 kW		Q_{g1}	200 kW
Q_{e1}	35 kW		Q_{e1}	35kW
Q_{g2}	600 kW		Q_{g2}	600 kW
Q_{e2}	125 kW		Q_{e2}	125 kW
t_g	80°C		t_g	150 °C
p_g	0.581 MPa		p_g	3 MPa
$t_{e,sat}$	3°C		$t_{e,sat}$	3°C
t_e	8°C		t_e	8°C
p_e	0.055 MPa		p_e	0.055 MPa

The main diameters of ejectors were estimated using 1D model based on Huang et al. (1999). Results are shown in Table 2

Table 2. Estimated dimensions of gas ejectors

no.	Q_e [kW]	L_n/D_{cr}	L_m/ D_m	L_{diff} /D_{diff}	U	refrigerant
t_g = 80°C, t_{esat} = 3°C						
1	35	3.578	9.986	4.750	0.233	R1233zdE
2	125	3.577	9.989	4.724	0.277	R1233zdE
3	35	4.539	9.997	4.744	0.249	R1336mzzz
4	125	4.540	9.992	4.741	0.297	R1336mzzz
t_g = 150°C, t_{esat} = 3°C						
5	35	9.887	9.991	4.752	0.270	R1233zdE
6	125	9.897	10.00	4.745	0.320	R1233zdE
7	35	12.97	9.986	4.742	0.307	R1336mzzz
8	125	13.05	10.00	4.747	0.365	R1336mzzz

The ejector dedicated for operation with R1233zdE is slightly smaller than for R1336mzzz due to more favbourable thermodynamic and thermokinetic properties.

2.1 Numerical model of ejector: Proper mesh with good quality is one of steps for the successful CFD numerical simulation. During meshing the geometrical model several important factors must be taken into consideration, e.g., computational power, time of calculation, required accuracy. For ejector technology numerical simulations can be divided into two groups. The first one is simulation of the operation of the ejector in general point of view. The second one is detailed simulation. Simulations from the first group allows for prediction of the performance of the ejector. Ejectors are designed for specific operating inlets and outlets parameters. In most of the cases the operating conditions are changing, and the ejector performance is also change. Therefore, the performance line of the ejection system is required. This is especially important in the refrigeration area, where experimental investigations of the ejection systems operating with new, perspective, and low-GWP refrigerants are limited. In this type of simulation in most of the cases the entrainment ratio and the compression ratio are investigated. More detailed simulations are required for the shape optimization or investigation of the flow-field inside the ejector. This can lead to improvement of the efficiency of the refrigeration ejection system. Since available analytical models based on balances equations are very simple, they are not suitable for detailed analysis of the flow-field. However, they can be effectively used for prediction of the geometry of the ejector.

Geometry of the investigated ejector was obtained by means of lumped parameter model based on gas dynamic relations and improved by CFD simulation. The inlets and outlets parameters were set as proper for heat driven refrigeration system used for air-conditioning purposes. The geometrical model was built using SpaceClaim Design Modeler belonging to ANSYS package.

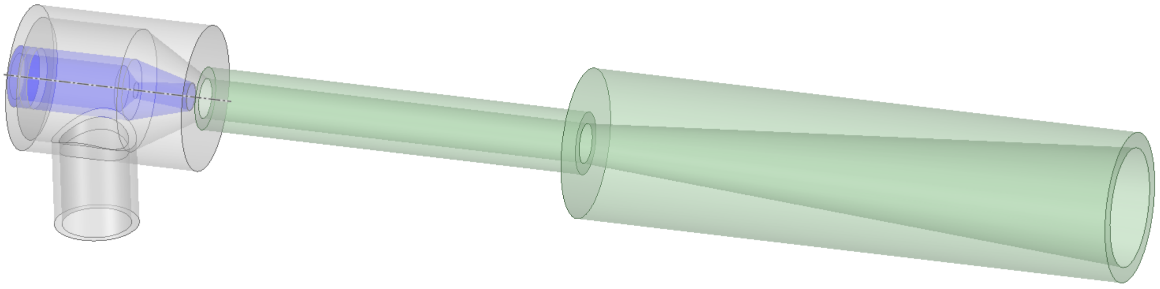

Figure 1: CAD model of the designed ejector

The discretization of the computational model was performed using a non-structural hybrid mesh. The Fluent Meshing program included in the ANSYS 2019 R2 package was used to generate the computational mesh. Local sizing of cells was used at the exit edge of the motive nozzle and at the inlet surfaces. The core of computational cells is made up of polyhedral cells which, by means of tetrahedral cells, connect to rectangular cells at the walls. Five rows of hexahedral computing cells were used against the outer walls constituting the boundary layer.

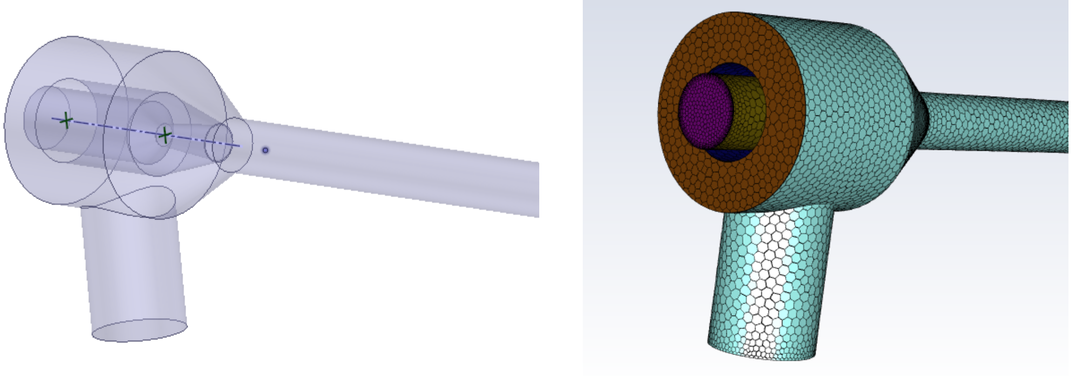

Figure 2: Control volume and discretization of the designed ejector

Table 3. Massflow rates for ejector No.1

<table>
<tr><td>$t_{g,sat}$ [°C]</td><td>t_g [°C]</td><td>p_g [MPa]</td><td>$t_{e,sat}$ [°C]</td><td>t_e [°C]</td><td>p_e [MPa]</td></tr>
<tr><td>75</td><td>80</td><td>0.581</td><td>3</td><td>8</td><td>0.055</td></tr>
<tr><td colspan="6">CFD 3D real gas model</td></tr>
<tr><td colspan="2">m_g [kg/s]</td><td colspan="2">m_e [kg/s]</td><td colspan="2">U</td></tr>
<tr><td colspan="2">0.865</td><td colspan="2">0.236</td><td colspan="2">0.272</td></tr>
<tr><td colspan="6">CFD 3D perfect gas model</td></tr>
<tr><td colspan="2">m_g [kg/s]</td><td colspan="2">m_e [kg/s]</td><td colspan="2">U</td></tr>
<tr><td colspan="2">0.806</td><td colspan="2">0.215</td><td colspan="2">0.267</td></tr>
<tr><td colspan="6">0D real gas model</td></tr>
<tr><td colspan="2">m_g [kg/s]</td><td colspan="2">m_e [kg/s]</td><td colspan="2">U</td></tr>
<tr><td colspan="2">0.882</td><td colspan="2">0.213</td><td colspan="2">0.241</td></tr>
</table>

2.2 Mesh independence test: Mesh quality was evaluated using evaluation tools available in ANSYS FLUENT Meshing, i.e. *inverse orthogonal quality* and *skewness*. Both parameters are in the range of 0-1. Values close to 0 indicate finner mesh with high quality, while values close to 1 indicate coarser mesh with unacceptable quality. Average values of quantitative parameters for this model were as follows: *inverse orthogonal quality* = 0.01 for the surface mesh and *inverse orthogonal quality* = 0.072 for the volumetric mesh. Average value of *skewness* for the volumetric mesh was $s = 0.064$. These numbers indicate that the mesh generated is of very good quality.

Mesh independent test were performed for the operating conditions given in Table 3. As a boundary condition for both streams (motive stream and entrained stream) the pressure inlets were used. Pressure outlet was used at the ejector outlet. In order to save time, the mesh sensitivity analysis was performed for an ideal gas. Mass flow rates of both streams only were were selected to control the dependence of the results on the mesh density. Meshes with a density of 261,000 control volumes and 497,000 volumes were compared. The following results were obtained:

	m_g	m_e
Mesh 1- 261k	0.805887	0.2150033
Mesh 2- 497k	0.798932	0.215928
difference %	0.863062	-0.430086422

Difference between meshes was calculated as:

$$\Delta R_{\%} = \frac{mesh1 - mesh2}{mesh1} \cdot 100\% \qquad \text{Eq. (2)}$$

As the results indicated that the difference between the mesh with a density of 261,000 volumes and the almost twice the density (497,000 volumes) was less than 1%, it was concluded that mesh with 297,000 cells is dense enough to give accurate results.

Modelling of performance line of tested ejector

Ejectors are usually designed to operate under certain conditions, commonly referred as on-design conditions. Typical operating conditions are constant motive parameters and constant suction parameters of the ejector, related to temperatrure inside evaporator. The discharge pressure of the ejector depends on the temperature of condensation inside condenser. Discharge prarameters are therefore independent of the user. As the condenser cooling conditions may vary within a fairly wide range, the saturation temperature in the condenser may also vary within a few/several Celsius degrees. The change of condensation temperature causes a change of condensation pressure and pressure ratio of the ejector. As the pressure in the condenser increases, the pressure ratio of the ejector increases. Under certain conditions, when discharge pressure is higher than so-called critical pressure the entrainment ratio decrease. These changes illustrate the operating characteristics of the ejector. The most popular are the characteristics in the form of the dependence of the suction coefficient on the saturation temperature in the condenser, u = f (tc) and the compression ratio on the suction coefficient, Π = f (u).

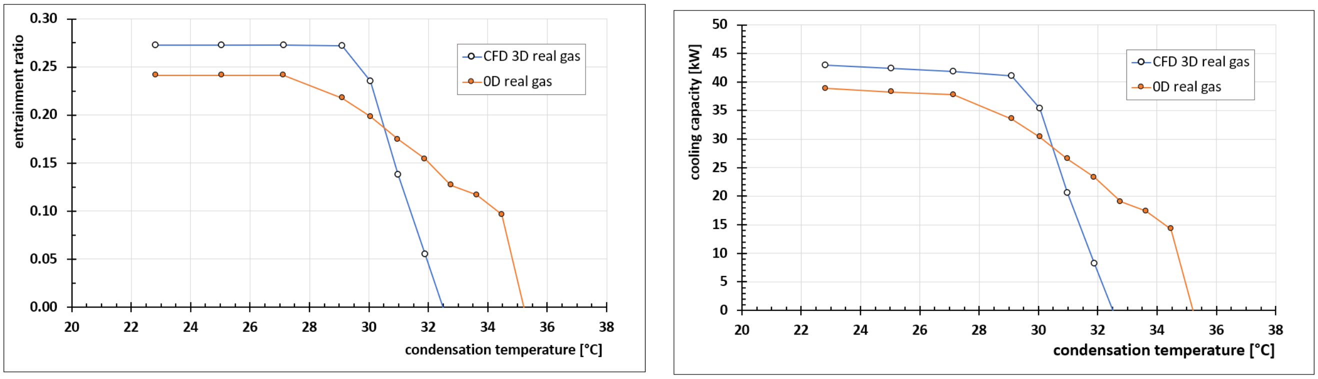

Figure 3: Performance lines of the designed ejector

The performance lines were obtained by means of lumped parameter model based on Chen *et al.* (2013) and Kumar and Ooi (2014). For comparison purposes and validation of 0D model the performance lines were also obtained by means of 3D CFD simulations. Results are shown in fig 3.

The results show the operation of the ejector in the on-design regime, where the entrainemt ratio is constant and off-design operating regime, in which the entrainemt ratio and the amount of produced cold decrease with an increase in the condensation temperature. The maximum entrainemt ratio for the 3D model is $U_{CFD} = 0.272$, while for the 0D model it is $U_{0D} = 0.241$. The differences are mainly due to the constant values of coefficients, used in the 0D model. These are primarily the isentropic efficiencies of the components of the ejector and the Fanning loss coefficient. Additionally, in 0D model the value of the isentropic exponent was also assumed as constant. Also, the 0D model uses some gas-dynamic relationships that are valid for perfect gases.

The difference between the results can also be observed during off-design operating regime. The 0D model predicts a smoother and wider range of off-design operation, while the 3D model predicts a steeper deterioration of ejector operation. The difference is mainly due to the method of modeling the vapor compression process. In the 0D model, it is assumed that the compression is obtained on a normal shock wave, regardless of the nature of the ejector operation. In off-design operating regime, the normal shock wave turns into oblique wave, which is not included in the 0D model. This is quite a significant simplification. Both models predict a roughly similar temperature (27 and 29°C) of transition from the on-design regime to the off-design regime.

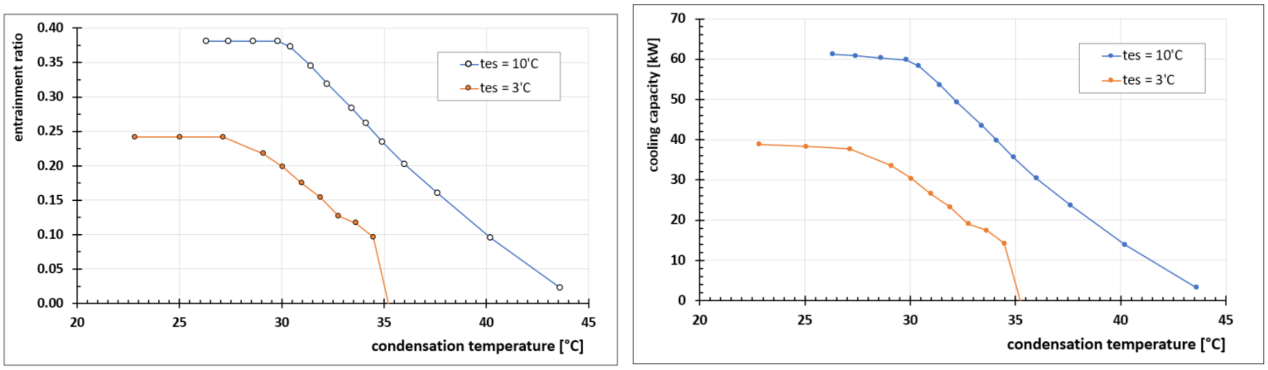

Figure 4: Performance lines of the designed ejector for two evaporating temperatures

The designed ejector was also investigated for operation with higher evaporation temperature, corresponding to high-temperature air-conditioning, in which the temperature of chilled water is 16/19°C. Results are presented in fig 4. The operating parameters were: $t_{g,sat}$ = 75°C, $t_{e,sat}$ = 10°C. The results show that the ejector is able to achieve entrainment ratio higher than 0.35 which gives a cooling capacity up to 60 kW. The critical temperature that changes the on-design to off-design operating regime is about 30°C. For the proposed ejector the cooling capacity is approximately 40 kW for standard chilled water temperature 6/12°C and up to 60 kW for chilled water temperature of 16/19°C.

Modification of the ejector geometry

Based on 3D modeling the pressure and velocity distributions along the ejector axis were opbtained and presented in fig 5. As can be seen from the figure, a shock wave appears inside the mixing chamber at the coordinate x = 0.60 m. Usually, it is assumed that during the on-design operation of the ejector, the shock wave occurs in the final part of the mixing chamber. This is also one of the major assumptions for 0D modelling. Since after the shock wave the flow is fully mixed and subsonic the flow through cylindrical mixing chamber will deteriorate the ejector operation due to additional friction. It can be seen on the figure, however, that the mixing chamber can be shortened, which reduces the dimensions of the ejector.

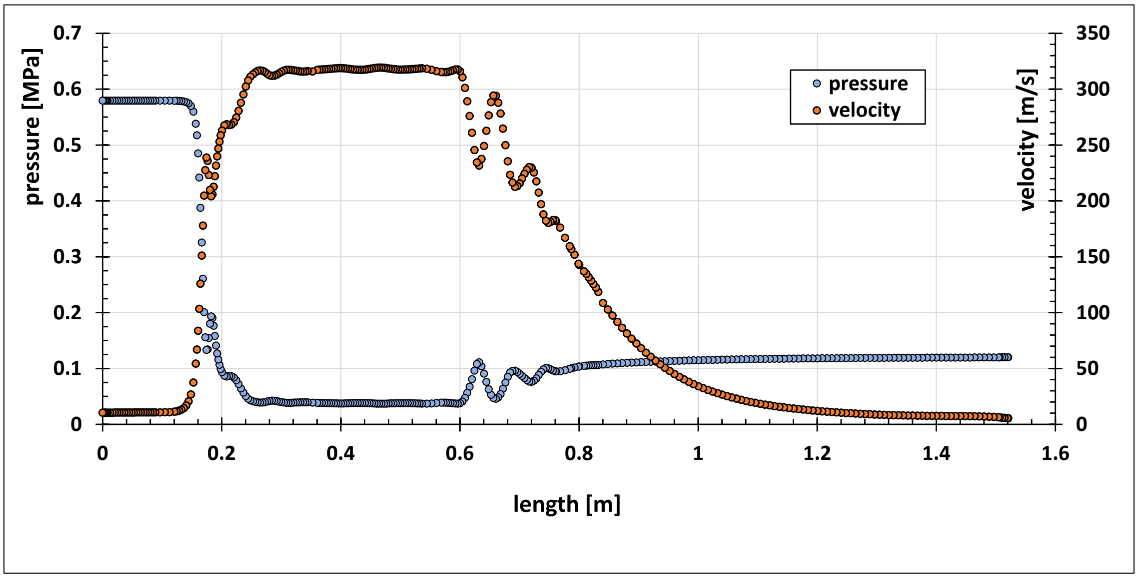

Figure 5: Pressure distribution along axis

In addition, it may be suggested that the diameter of the mixing chamber be slightly reduced. The performance characteristics show that the ejector excees the desired cooling capacity. By reducing the mixing chamber, the entrainment ratio and the cooling capacity are lowered, but it contributes to an increase in the compression ratio. As a result, it can be expected that the temperature at which the ejector changes from on-design to off-design operation regime will slightly shift towards higher temperatures. Lists of the basics and modified geometric parameters are given in table 4. Performance lines for both geometries obtained by 0D model are shown in fig. 6

Table 4. Dimensions of the designed ejectors

basic	**modified**
$t_{gsat} = 75°C$, $t_{esat} = 3°C$	**$t_{gsat} = 75°C$, $t_{esat} = 3°C$**
$L_n / D_{cr} = 3.579$	$L_n / D_{cr} = 3.579$
$L_m / D_m = 10.00$	$L_m / D_m = 7.40$
$L_{diff} / D_{diff} = 4.778$	$L_{diff} / D_{diff} = 4.829$

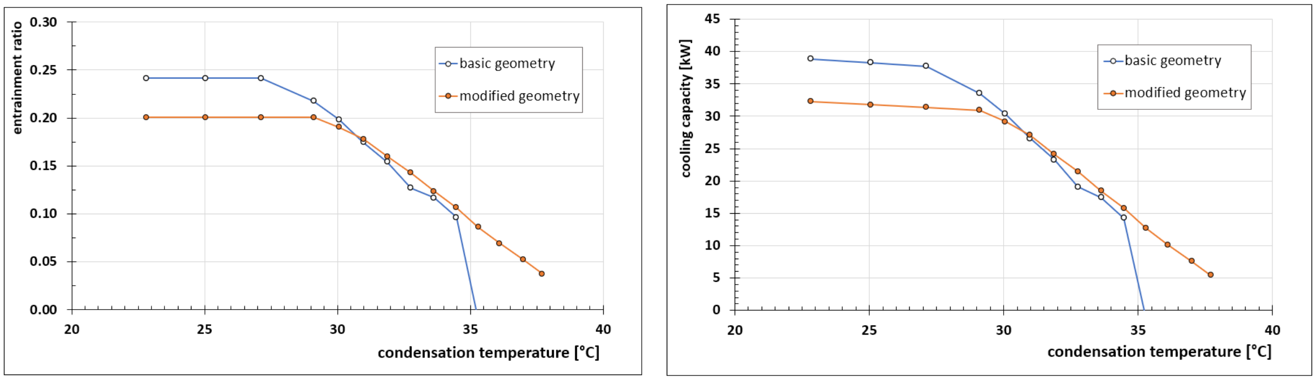

Figure 6: Performance lines of the designed ejector for two evaporating temperatures, EJ1 – basic, EJ2 - modified

Both geometries are able to operate in both on-design and off-design conditions. Changing the diameter of the mixing chamber slightly lowered the entrainment ratio and the cooling capacity. Model 0D indicates that for the EJ2 geometry, it achieves approximately 32 kW of cooling capacity with standard chilled water temperature 6/12°C. The model, as expected, indicates that reducing the diameter of the mixing chamber extended the on-design operating regime. The transition temperature of the operating characteristics $U = f(tc)$ and $\dot{Q}e = f(tc)$ corresponds to the temperature of 29°C, for the EJ1 geometry it was 27°C.

3. CONCLUSIONS

The following conclusions can be stated:

- Lumped parameter model can be effectively use for designing the gas ejectors and analysis of operation at wide range of operating parameters.
- Performance line of the gas ejector can be predicted by means of CFD simulation.
- Differences between results obtained from simulation based on 0D model, CFD 3D perfect gas and CFD 3D real gas model less than 10%.
- Modification of the ejector geometry allows for operation of the ejector as double choked at higher condensation temperatures.
- Next steps are related with manufacturing the ejector and experimental investigations. Results of this investigations will be used for validation and evaluation of the proposed designing approach.

ACKNOWLEDGEMENTS

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NOMENCLATURE

D	diameter (m)	L	length (m)
m	mass flow rate (kg/s)	T, t	temperature (K), (°C)
p	pressure (MPa)	U	mass entrainment ratio
	subscrits		
c	condensation	p	primary, motive fluid
cr	critical	s	secondary, entrained fluid
diff	diffuser	m	mixing chamber
n	motive nozzle		

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