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Ejector-Based Air Conditioner Utilizing Natural Refrigerants

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ABSTRACT

The paper deals with modeling of the ejection cycle, especially for solar air-conditioning. The presented approach for ejector analysis has been proposed, based on formulation of performance curve of the installation and performance curve of an ejector. The operation point of the whole system is found at the interception of these two curves. The exemplary performance curve of the ejector has been showed experimentally and numerically.

1. INTRODUCTION

Residential and commercial air-conditioning consumes over 15% of all electric energy generated in the USA and creates two sources of environmental pollution: 1) the ozone-depletion effect of traditional refrigerants belonging to CFC and HCFC groups, and 2) the emission of greenhouse gases connected with the generation of electricity. Both sources are contributing significantly to the global warming effect. Additionally, with energy cost rising constantly, industry is looking to reduce electricity expenses as a means of lowering their fixed costs in order to stay competitive.

This paper presents the development of air-conditioning technology that completely eliminates the ozone depletion effect by using natural refrigerants and also dramatically reduces the need for electric power. This is accomplished by using free or inexpensive heat – either solar or waste heat, as the main source of energy instead of electricity. The described system is a modification of a well-known vapor compression cycle (VCC) and it uses our previously developed ejector device for non-mechanical compression. Instead of pressurizing the refrigerant by a mechanical compressor, a pump compresses the liquefied refrigerant, then heat is added to evaporate it and finally the refrigerant is re-compressed in an ejector without any mechanical energy spent. The main difference between this cycle and the conventional refrigeration cycle (reverse Rankine cycle), besides elimination of a compressor, is that it requires three heat sources at different temperatures rather than two, namely at the generator level, which is the temperature of the solar or waste heat source, at a condensing level, which is the ambient temperature (actually this is a heat sink) and the evaporator temperature required for cooling effect.

The theoretical analysis of the cycle combined with laboratory experiments conducted on a specially constructed test-stand is described here. A new method of analysis was proposed based on the set of two performance characteristics: the first for an ejector and the second one for the rest of the refrigeration system. The operating parameters of the whole system are determined as the intersection of these two characteristic lines. This is a very similar method as in the case of the well-known liquid mechanics procedure for pump selection for a given pipeline system.

The main objective of numerical calculations and experimental evaluations was to determine the ejector performance, which is necessary to find the operating point of the system. The analysis presented in this paper demonstrates that the proposed system is feasible from a thermodynamic standpoint and, furthermore, that the amount of heat required to produce a given cooling load is reasonable and achievable from the solar or waste heat sources.

2. AIR-CONDITIONING CYCLE

The schematics of the simplest ejection refrigeration system together with its thermodynamics cycle in log*(p)-h* coordinates are presented in Fig. 1. Liquid refrigerant is passed through the pump to the generator (point 8). The liquid is then heated in the generator by either energy from solar panels or waste heat. First stage of heating produces saturated vapor, which is then heated further and leaves the generator in a superheated condition (point 1). The degree of vapor superheat is the function of the generator capacity as well as mass flow rate. Such superheated vapor enters the motive nozzle of the ejector and undergoes an expansion from the generator pressure p_g to the lower pressure, which is an evaporation pressure p_e (point 2). The ejector sucks in vapor from the evaporator (point 7), and mixes it with expanded vapor (point 2) and in consequence, the mixed vapor in state 3 is obtained. The pressure of the working fluid initially rises slightly as a result of the momentum exchange, and then it rises more in the diffuser up to the point 4, achieving the level of the condensation pressure p_c. Compressed vapor enters the condenser, where it condenses and may also subcool depending on the conditions of the cooling in the condenser. The working fluid leaves the condenser in the liquid state (point 5). It is then divided to two parts: one part flows to the generator through the circulating pump, meanwhile the remaining part flows to the evaporator through the expansion (throttling) valve, in which it is throttled to the evaporation pressure of p_e, achieving the condition of wet vapor (point 6). Through the boiling in the evaporator, the working fluid absorbs cooling capacity Q_o from the refrigerated medium, which may be the air circulating in the air-conditioned room, or the ice water, which in the turn will cool the air-conditioned room.

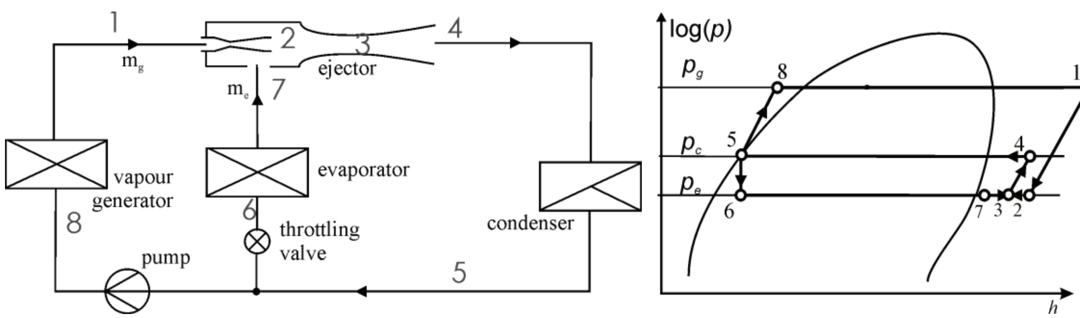


Figure 1: Schematic diagram of ejector device and ejector cycle in log*(p)-h* co-ordinates

The main difference between this cycle and the conventional refrigeration cycle (reverse Rankine cycle) besides elimination of a compressor, is that it requires three heat sources at different temperatures rather than two, namely at the generator level (8-1) which is the temperature of the solar source, 70-100°C, a condensing level (4-5), which is the ambient temperature of 25-35°C and the evaporator temperature (6-7) at approx. 5-8°C.

3. APPROACH TO CYCLE ANALYSIS

In order to analyze the above system, we have proposed our own approach for modeling of ejection cycles. The main concept of this method is to split the cycle and separate clean thermodynamic relationship from the parameters characterizing the operation of ejector (Fig.1). This will allow to separate the investigation of the flow in an ejector, which is rather complicated from the operation of the entire system, which on the other hand can be described by relatively simple equations.

The applications of existing models based on the conservations equations only, requires the assumption of a driving pressure. However, both, the condensation pressure and the evaporation pressure are fixed parameters in cooling devices. They depend upon the thermal capacity of condenser, the temperature and the flow rate of cooling water, flow conditions through the condenser, etc. In addition, the performance of the condenser is effected by external conditions, such as ambient temperature. Similarly, the evaporation pressure is a function of the cooling capacity of the evaporator, as well as the required cooling temperature. As far as prediction of the driving (motive) pressure in an ejector, it was shown (Smierciew K. et. al, 2008) that it was necessary to model the relationship between the compression ratio and mass entrainment ratio $\Pi = f(U)$ for the entire ejection process. The operating point of the

system was then determined as an intersection of the performance curve of an ejector and the system capacity curve. Based on the real performance curve of the ejector, the required motive pressure could be predicted for a given cooling capacity, geometry of the ejector, and assumed operating parameters of the system.
Therefore, the main task in the proposed method for the compression cycle analysis will be to prepare two characteristics (one for the ejector and another for the entire system), describing the dependence of dimensionless compression ratio Π on the entrainment ratio U. The knowledge of such performance curves for the system and for the ejector will allow to find the operating point of the system. This is a very similar method as in the case of the well-known procedure from liquid mechanics for pump selection for a given pipeline system.
The compression ratio is given by the following equation:

$$\Pi = \left(p_c - p_e\right) / \left(p_g - p_e\right) \tag{1}$$

The operation of the system is described by the energy conservation equation only. Because p_g and U are unknown parameters in ejection systems, and there is no available information about the p_g value in the energy conservation equation, there is a need to seek other relations in order to determine the value of the driving vapor pressure.
For every ejector the individual operation curve can be prepared. Preparing the performance curve is rather a difficult task in an absence of experimental investigations, and the only way is the numerical modeling.
The study of an exact model is complicated due to various processes occurring in an ejector such as: mixing of the streams, mass, momentum and energy transfer, eddies, separation of the stream from boundaries, shock waves, etc. Because of all these processes the model of ejector operation becomes complex. However, results received from experiments carried over on similar ejectors under similar operating conditions, can be applied to simplify the entire modeling task.

3.1 Performance of the system

The performance of the system results from the energy balance and does not contain any information about the ejector, therefore it treats the ejector as a “black box”. The energy conservation law for the entire cooling system can be written as

$$\dot{Q}_c = \dot{Q}_e + \dot{Q}_g + \dot{P}_p \tag{2}$$

where $\dot{Q}_c$ is capacity of the condenser, $\dot{Q}_e$ is the cooling capacity of the evaporator, $\dot{Q}_g$ is energy input to the generator, and $\dot{P}_p$ is power of the pump. Because of the very small value of $\dot{P}_p$ in comparison with remaining components in formula (2), it can be neglected. Now, the individual thermal capacity of each device can be expressed as follows:

$$\dot{Q}_e = \dot{m}_e \cdot (h_7 - h_6) \tag{3}$$

$$\dot{Q}_g = \dot{m}_g \cdot (h_1 - h_8) \tag{4}$$

$$\dot{Q}_c = \left(\dot{m}_g + \dot{m}_e\right) \cdot (h_4 - h_5) \tag{5}$$

after suitable transformations we can obtain:

$$\left(\dot{m}_g \cdot h_1 + \dot{m}_e \cdot h_7\right) = (\dot{m}_g + \dot{m}_e) \cdot h_4 \tag{6}$$

For a refrigerating plant, the quantity of transferred heat has to be known, therefore the cooling capacity of the evaporator $\dot{Q}_e$ is also known. From Eq.(3) the mass flow of refrigerant flowing through the evaporator can be determined. The enthalpy h_7 is a function of pressure and temperature of vaporization: $h_7 = h(p_e, t_e)$ and $t_e = t_s(p_e) + \Delta T_e$, where $t_s(p_e)$ is saturation temperature and ΔT_e is the vapor superheat. Then, the enthalpy h_4 can be calculated from the following equation:

$$h_4(p_g,U)=\frac{h_1(p_g)+U\cdot h_7}{1+U} \tag{7}$$

In order to describe the operation of the cooling system, the processes occurring inside of the ejector are not considered. However, the condition of refrigerant at the ejector outlet has to be known. This leads to the assumptions that mixing takes place at constant pressure and processes of expansion and compression are isentropic. If the pressure and the amount of superheat of driving fluid are known, then temperature of fluid is: $t_g = t_s(p_g)+\Delta T_g$, and the enthalpy h_1 can be obtained as a function $h_1 = h(p_g, t_g)$. Assuming an initial entrainment ratio U, the enthalpy at the ejector outlet can be found from Eq. (7). Energy conservations equation between inlet and outlet of the driving nozzle can be written as:

$$h_1(p_g)=h_2+\frac{1}{2}w_2^2 \tag{8}$$

And energy equation for a mixing chamber:

$$h_2+\frac{1}{2}w_2^{\,2}+U\cdot h_7=(1+U)\left(h_3+\frac{1}{2}w_3^{\,2}\right) \tag{9}$$

Further assuming zero velocity of vapor after mixing process (state 3), expressing the outlet velocity from the motive nozzle using the difference of enthalpies $w_2^2=2(h_1-h_2)$, and making a few other transformation, gives:

$$h_3+\frac{h_1(p_g)-h_2}{(1+U)^2}=\frac{h_1(p_g)+U\cdot h_7}{(1+U)} \tag{10}$$

Thus, the enthalpy h_3 is a function of p_g and entrainment ratio U can be expressed as follows:

$$h_3=\frac{h_1(p_g)+U\cdot h_7}{(1+U)}-\frac{h_1(p_g)-h_2}{(1+U)^2}=h_4(p_g,U)-\Delta h_2(p_g,U) \tag{11}$$

where

$$\Delta h_2(p_g,U)=\frac{h_1(p_g)-h_2}{(1+U)^2} \tag{12}$$

is the kinetic energy changed at the diffuser resulting in static pressure rise. It needs to note that the above result of h_3 is not a total enthalpy but a static enthalpy. The total enthalpy is:

$$h_{3'}(p_g,U)=h_3(p_g,U)+h_2=h_4 \tag{13}$$

Calculating the entropy at this state, pressure at the outlet from the ejector can be defined as $p_4 = p(s_{3=4}, h_4)$. This pressure has to be compared with assumed p_c, and in case of a significant difference, the entrainment ratio U should be recalculated. These calculations should be repeated with different U values, until the difference between p_4 and p_c is not significant. As a result, one point of the characteristic $\Pi = f(U_i)$ is determined. In order to obtain the entire characteristics of the system, the above calculations should be repeated for different values of the driving vapor pressure and data points plotted.

3.2 Performance of the ejector

The model describing the work of isentropic ejector was developed for the purpose of this project. For a given ejector geometry, specific parameters describing the processes inside the ejector are: p_e, ΔT_e, p_c, ΔT_g . Additionally, the driving vapor pressure p_g, was initially assumed. The enthalpy and density of driving vapor was calculated as functions: $h_1 = h(p_g, t_g)$ and $\rho_1 = \rho(p_g, t_g)$. As the consequence of expanding in the nozzle, vapor achieves pressure $p_2 = p_7 = p_e$. The remaining parameter of expanded vapor can be found as: $h_2 = h(p_g, s_{1=2})$, $t_2 = t(p_g, s_2)$, $\rho_2 = \rho\,(p_2, t_2)$. Constant value of specific heat was assumed. The critical velocity in the nozzle throat is given by:

$$w_{cr} = \sqrt{2 \cdot \frac{\kappa}{\kappa - 1} \cdot p_g \cdot \rho_1^{-1} \cdot \left(1 - \beta^{(\kappa-1)/\kappa}\right)} \tag{14}$$

The mass flow rate of motive vapor can be calculated using the following equation:

$$\dot{m}_g = p_g \cdot A_t \cdot \sqrt{\frac{\kappa}{T_g \cdot R} \cdot \left(\frac{2}{\kappa+1} 1 - \beta\right)^{(\kappa+1)/(\kappa-1)}} \tag{15}$$

where A_t is the cross-section area of the throat of the nozzle, κ is heat capacity ratio $\kappa = c_p/c_v$, β is the critical pressure ratio. Because two streams with different temperatures enter the mixing chamber separately, their average temperature was determined. The temperature of a mixture at the end of the cylindrical part of ejector is given by an isentropic relationship:

$$t_m(p_m, U) = \frac{t_2 + U \cdot t_7}{1 + U} \cdot \left(\frac{p_m(U)}{p_2}\right)^{(\kappa-1)/\kappa} \tag{16}$$

In the above equation $p_m(U)$, pressure of the mixture is an unknown parameter that depends on the entrainment ratio U. Using a similar isentropic relationship as in eqn. (16), the density of the mixture $\rho_m(p_m)$ and the outlet temperature $t_4 = t_c$ can be calculated. Velocity of the mixture can be calculated from mass balance equation and velocity at the ejector outlet can be obtained from the continuity equation:

$$w_m(p_m, U) = \frac{\dot{m}_g \cdot (1 + U)}{A_m \cdot \rho_m(p_m)} \tag{17}$$

$$w_4(p_m, U) = \frac{\dot{m}_g \cdot (1 + U)}{A_d \cdot \rho_4} \tag{18}$$

where A_d is the cross-section area of the of diffuser, and $\rho_4 = \rho(p_4, t_4)$ is the density of the mixture at the outlet of the diffuser. Velocity achieved by the secondary vapor at the inlet to the mixing chamber is given by the following equation:

$$w_7(U) = \frac{\dot{m}_g \cdot U}{A_7 \cdot \rho_7} \tag{19}$$

where, area for secondary fluid at the mixing chamber inlet $A_7 = A_m - A_2$, A_m is the mixing chamber area, A_2 driving nozzle outlet area and $\rho_7 = \rho(p_e, t_e) \approx \rho_2$.

Values of p_m and U are unknown, it is therefore necessary to come up with an additional relationship in order to find the solution. This can be a momentum conservation equation, written separately for a mixing chamber Eq. (20) and a diffuser Eq. (21):

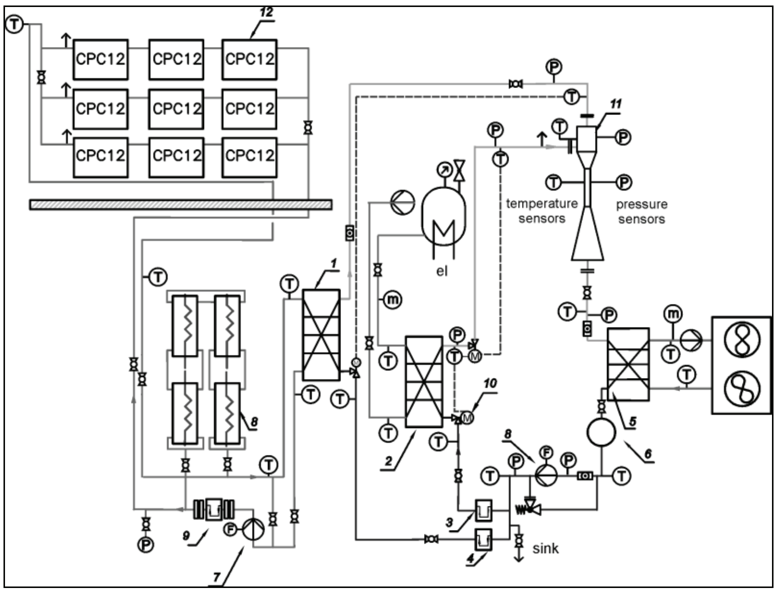
$$\frac{A_m}{\dot{m}_g}(p_2 - p_m) - (1 + U) \cdot w_m + w_2 + U \cdot w_7 = 0 \tag{20}$$

$$A_m p_m + \frac{1}{2}\left(A_d - A_m\right)\left(p_m + p_4\right) - A_d p_4 = \left(\dot{m}_g + \dot{m}_e\right)\left(w_4 - w_m\right) \tag{21}$$

Equations (20) and (21) are simultaneous equations with two unknowns: mixture pressure p_m and entrainment ratio U., which now can be determined. The assumptions, taken for writing the above momentum conservation equations should be noted. Specifically, friction forces were neglected and shock wave were not considered. While it is generally recognized that the pressure rise in the ejector is caused by the shock wave, our approach assumes that the compression is the result of momentum exchange only.

4. EXPERIMENTAL RESULTS

The operating point of the ejector should be known in order to develop a methodology for its design. It can be found either theoretically by modeling or experimentally. Within this project, we considered both approaches for isobutene used as a refrigerant. For the experiments, the testing rig was erected according to the schematics shown in Fig. 2. The ejector has been initially designed, based on the above presented theory and its approximate geometry is shown in Fig. 3. The system was instrumented with temperature and pressure sensors, mass flowmeters for refrigerant and coolant. Separate heat load system was built for an evaporator and a heat rejection system – for a condenser. Both were filled with a glycol solution.



1	vapor generator
2	evaporator
3	secondary fluid mass flowmeter
4	primary fluid mass flowmeter
5	condenser
6	liquid refrigerant storage
7	glycol pump
8	refrigerant pump
9	glycol mass flowmeter
10	throttling valve
11	ejector
12	solar collector

Figure 2: Schematic of the testing rig.

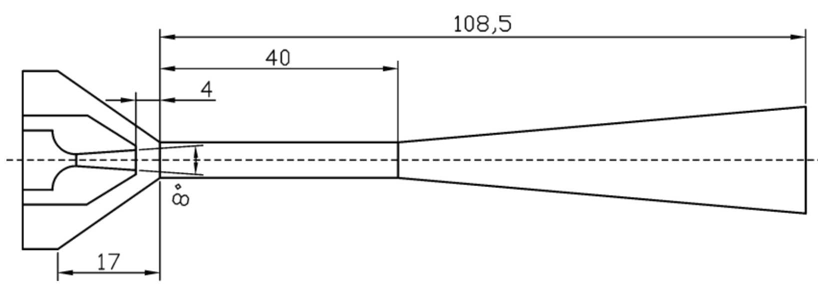
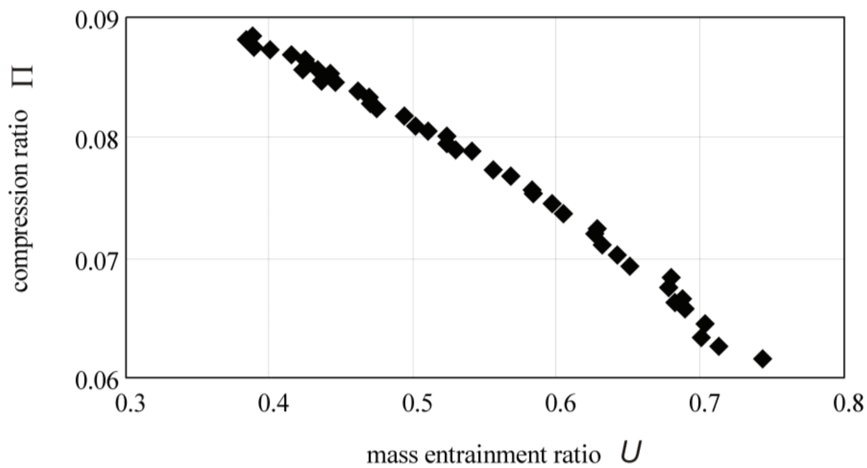


Figure 3: Axi-symmetric cross-section of tested ejector

The main source of heat was a solar system consisting of 12 vacuum solar collectors installed on the roof of the laboratory. The maximum amount of solar heat was approximately 20 kW. Additionally, electric heaters were installed to provide an auxiliary heat source.

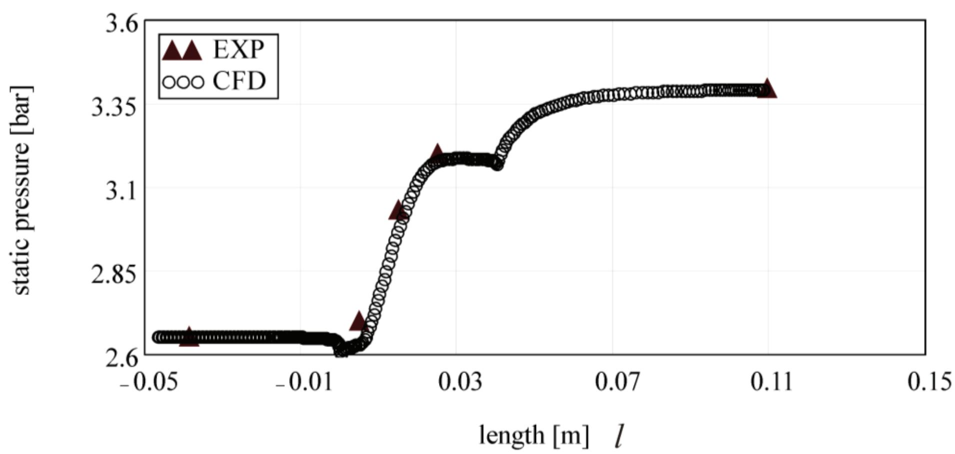
Preliminary tests have confirmed the stable operation of the system under air-conditioning conditions suitable for ceiling-cooling. Systematic experimental investigations were carried out for a wide range of operation parameters of the ejector including entrainment ratio, pressure rise, as well as pressure and temperature profiles in the mixing chamber and the diffuser. An exemplary ejector performance is presented in Fig. 4.



generator pressure p_g = 9.1 bar	condensing pressure p_c = 2.20 bar
evaporation pressure p_e = var	mass flow rate m_g = 30.07 kg/h

Figure 4: Experimental compression ratio *Π* versus mass entrainment ratio *U.*

Commercial CFD code was used for modeling of the isobutene ejector operating in the superheated vapor region. A two-dimensional axisymmetrical model was applied (Butrymowicz et. al, 2008) for calculations presented in this paper. The thermodynamic properties of the working fluid are implemented from the NIST database. A six-equation turbulence model was applied, in which Reynolds stresses are calculated directly. The results of the simulation in comparison with experimental results are presented in Fig. 5. Boundary condition used in CFD simulation are shown in Table 1.



l = 0.0 - inlet to the mixing chamber

Figure 5: Comparison of calculated static pressure profile with experimental data

Table 1: Comparison of CFD results with experimental results

p_g [bar]	t_g [°C]	p_e [bar]	t_e [°C]	p_c [bar]	t_c [°C]	primary flow m_g [kg/h]		secondary flow m_e [kg/h]	
						EXP	**CFD**	**EXP**	**CFD**
10.68	78.9	2.65	20	3.4	54.3	35.2	35.5	15.3	18.7

The numerical results show reasonably good agreement with experimental data, especially in terms of mass flow rate of the motive vapor $\dot{m}_g$. However, some discrepancy for mass flow rate of the secondary fluid $\dot{m}_e$ was observed. Entrainment ratio U in CFD simulation is higher (U_{CFD} = 0.526) than measured values (U_{EXP} = 0.435) for tested operation parameters. It is possible that the above phenomenon is caused by presence of thermocouples with the diameter of 0.5 mm inside the mixing chamber of the ejector. The measuring caps of these thermocouples were located at the center line of the mixing chamber. The presence of the thermocouples could be included in numerical calculations. However, this requires application of the three-dimensional model, which is planned for the future.

5. CONCLUSIONS

On the basis of the presented results, the following was concluded:

- The model was formulated for an ejection air-conditioning cycle. It consists of the thermodynamic performance of the system and the characteristics of the ejector.
- The performance of the ejector may be found by numerical modeling or by experimental investigations. Both approaches have been presented in this paper.
- The reasonable agreement between theoretical CFD results and experimental test has been found for isobutene used as a refrigerant.
- Further investigations with natural refrigerants are needed to prepare more general information and more accurate models on relevant ejector parameters.
- Based on modeling as well as experimentation, the performance of the ejector can be predicted.

The described air-conditioner is intended for residential and commercial buildings, wherever either low-grade waste heat or solar heat is available. However many other applications are possible, for example, there is a great potential of using the developed technology for refrigeration and air-conditioning purposes in Third World countries and in remote areas where electric energy is unavailable. Small solar-based units can be developed for storage of medicines, perishable food, and to cool field clinics to name just few possibilities.

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