A SOLAR EJECTOR AIR-CONDITIONING AND REFRIGERATION SYSTEM

Dmytro I. Buyadgie[1], Vitaliy V. Sechenyh[2], Sergii V. Nichenko[2] and Olexiy D. Buyadgie[1]

[1] Head of R&D, Wilson Ltd, 25, Mikhailovs'ka str., office 340, 65005, Odessa, Ukraine, Phone: 380-48-7342010;

E-mail: buyadgie@wilson-ukraine.com.

[2] Ph. D., Assistant lecturers, Odessa State Academy of Refrigeration, 1/3 Dvoryanskaya Str., 65026 Odessa, Ukraine, Phone: 380-67-3990684; Fax: 380-48-7343010.

1. Introduction

Electricity consumption for cold generation purposes has become unacceptable from exergic, economic and environmental points of view. Solar powered ejector refrigeration systems (SERS) are therefore now suggested as one of the most promising and reliable cooling systems nowadays.

A solar ejector system for air-conditioning and refrigeration should be one of the main types of air conditioners. For this purpose, comprehensive investigations are required that will allow creation of a range of technical solutions, aimed primarily at increasing energy and, consequently, economic efficiency. In addition, these systems should have the simplest and most reliable design and, in fact, should become one of the elements of communication facilities, along with water and heat supplies.

In the residential buildings sector, household refrigerators could operate along with air-conditioners as an integrated system, powered by single solar collectors.

2. Calculation of ejector characteristics

2.1. Theoretical analysis of an ejector

This problem is solved in several steps. The first is the creation of functional schemes and a theoretical investigation of the basic element of the solar ejector system – the ejector.

In the first step of the calculation of the ejector parameters, properties of the working fluid must first be calculated. For this purpose a cubic equation of state of Peng-Robinson was used [1] in the form:

$$p = \frac{RT}{v-b} - \frac{a(T)}{v^2 + 2vb - b^2} \quad (1)$$

where,

$$a(T) = a_0 \left(1 + n \cdot \left(1 - \sqrt{T / T_C}\right)\right)^2, \quad (2)$$

$$a_0 = \frac{0.45724 R^2 T_C^2}{p_C}, \quad (3)$$

$$b = \frac{0.0778 R T_C}{p_C}, (4)$$

$$n = 0.37464 + 1.5422 \cdot \omega - 0.26993 \cdot \omega^2 \quad (5)$$

where p, T and v are the pressure, kPa, temperature, K, and specific volume, m^3/kg, respectively; p_C, T_C and v_C are the critical pressure, kPa, critical temperature, K, and critical specific volume, m^3/kg, respectively; ω is an acentricity factor; R is an individual gas constant, equal to R_0/M; M is molecular weight of the substance, kg/kmol; R_0 is the universal gas constant, equal to 8.314472 kJ/(kmol·K) [2].

Critical parameters of the studied refrigerants were taken from [3]. Critical parameters for poorly studied fluids or those not previously studied (such as HFE-7500) were determined using the equation of state (1) and information available in the literature about properties of the fluids. This procedure was carried out using genetic algorithms with fitness function of the form:

$$F_{Ad} = \sum \left| Y_{calc} - Y_{Lit} \right| \qquad (6)$$

This function is the sum of absolute deviations of calculated values of the fluid properties, Y_{calc}, from the values presented in the literature, Y_{Lit}. Thus, by minimizing the fitness function, we were able to identify the critical parameters of fluids, as well as the coefficients of equation (7), which describes the ideal gas specific heat:

$$Cp_0 = A + B \cdot T + C \cdot T^2 + D \cdot T^3 \qquad (7)$$

Thus, the obtained information allows calculation of both the thermal and thermodynamic properties of working fluids. With information about the thermodynamic properties of a fluid, the entrainment ratio of the ejector system can be calculated. Calculation of the entrainment ratio was carried out by the method described in [4], [5]. Based on this calculation, we obtained a preliminary assessment of the entrainment ratio and its main geometric characteristics, or "area ratio". "Area ratio" is a ratio of the nozzle exit cross-section to nozzle throat cross-section, as well as the ratio of the cross-section of the cylindrical part of the mixing chamber to the nozzle throat cross-section.

2.2. CFD modelling of the ejector flow part

As used in the first step, the procedure for calculation of the entrainment ratio is imperfect, and the results may significantly differ from the practical values of the entrainment ratio [4]. Therefore, the next step in determination of the entrainment ratio is CFD modeling of the ejector for each of the selected working fluids. Preliminary results from the first step are used as a first approximation for CFD modeling. This type of approach significantly reduces the time required to conduct CFD simulation and significantly improve the accuracy of the obtained data.

A CFD model of the ejector flow part for an air-conditioner, working on isobutane (R600a) is presented in Figure 1 and the ejector of binary ejector refrigeration system (ERS) is presented in Figure 2.

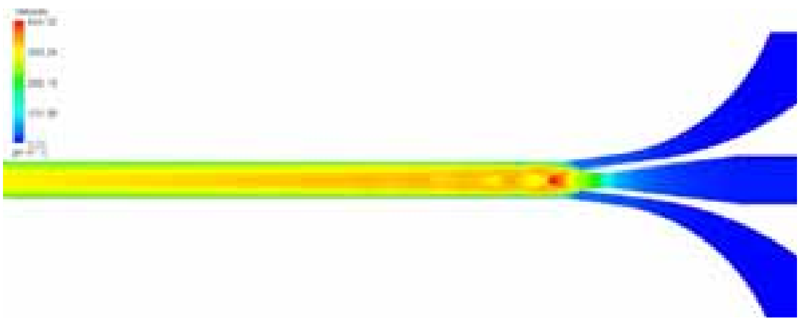

Fig. 1. Velocity profile in the flow part of an ejector (T_{ev} =12°C, T_{con} =35°C, T_{gen} =85°C) working with R600a

The results of this calculation, obtained by modeling of the ejector's flow part for all selected fluids and several mixtures, are presented in Table 1.

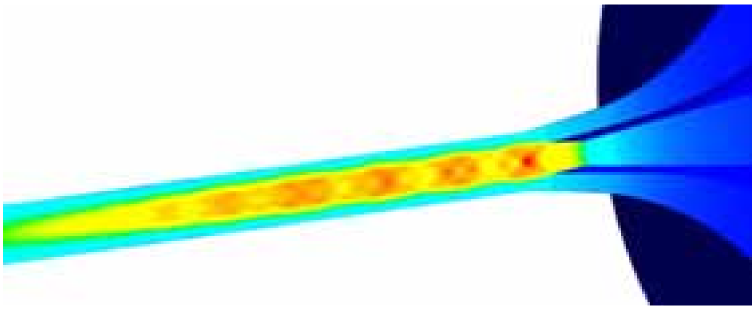

Fig. 2. Velocity profile in the flow part of an ejector (T_{ev} =12°C, T_{con} =35°C, T_{gen} =85°C) working with DME/Butane mixture.

Parameters and characteristics of ejector cycle	Working fluid						
	R600a	R600	RE170	R13T1	R365mfc	R236ea	R290/ RE170
Generation pressure, kPa	1487	1126	2467	1986	402.2	1127	3048
Condensation pressure, kPa	464.8	328.4	779.1	645.1	83.76	288.1	1075
Evaporation pressure, kPa	218.9	148	370	302.2	30.31	118.7	600
Entrainment ratio	0.55	0.51	0.54	0.44	0.35	0.37	0.60
q_{ev} / q_{gen} *	0.659	0.670	0.767	0.765	0.632	0.628	0.937
COP	0.36	0.34	0.41	0.34	0.22	0.23	0.56

** Ratio of evaporation heat to heat of steam generation*

Table 1. Results of ejector cycle calculation [2]

Since the coefficient of performance (COP) is the final feature, that allows judgment of the energy efficiency of the ERS, the values of COP are also presented in Table 1, along with the calculated entrainment ratios. The COP of a single-stage ERS is determined from the equation:

$$COP_{ej} = \omega \cdot q_{ev} / q_{gen} \qquad (8)$$

where ω is entrainment ratio, kg/kg; q_{ev} is specific cooling capacity, kJ/kg; q_{gen} specific heat load on generator, kJ/kg.

For the system, that produces cooling at different temperature levels, the value of COP is determined from expressions:

$$COP_{ej} = \frac{\sum_{i}^{N} Q_{0i} \frac{\tau_{0i}}{\tau_{01}}}{\sum_{i}^{N} Q_{Gi} \frac{\tau_{Gi}}{\tau_{G1}}} = \sum_{i}^{N} U_i \frac{q_{0i}}{q_{G1}} \frac{\frac{\tau_{0i}}{\tau_{0B}}}{\frac{\tau_{Gi}}{\tau_{GB}}} \qquad (9)$$

$$\tau_{0i} = \frac{T_C - T_{0i}}{T_{0i}} \qquad \tau_{0B} = \frac{T_C - T_{0B}}{T_{0B}}$$
$$\tau_{Gi} = \frac{T_{Gi} - T_C}{T_{Gi}} \qquad \tau_{GB} = \frac{T_{GB} - T_C}{T_{GB}} \qquad (10)$$

This research has led to creation of an ejector flow part that works on single refrigerants and their mixtures. Selection of fluids and mixtures made it possible to determine the best substance for air-conditioning and simultaneous production of cold at temperatures inherent in household refrigerators. The results of the COP calculations are presented in Table 1.

3. Application of the ejector in solar air-conditioning systems

Cold supply in the domestic sector is not limited to air-conditioning, as cooling is also used for storing and freezing food, so it would be convenient to combine this refrigeration system into one machine. In a single-stage ejector refrigeration machine, the temperature difference between condensation and evaporation usually does not exceed 30°C, which does not allow solution of the problem. Therefore, a combined system must be either a two or three-stage system, or must work on a zeotrope mixture of fluids (Fig. 3-6). A multi-stage ERS can work in either of two ways: the ejector at each step can consume working vapor from one steam generator (Fig. 3), or in subsequent stages, vapor formed after intermediate throttling in a liquid separator serves as working vapor (Fig.4, 5). In the second case, although the total consumption of heat in the steam generator produces work only in the upper stage ejector, further increases in the COP may be obtained by a cycle parameters optimization. To do this, two-stage cycles for these schemes were calculated and comparison of results is presented in Table 2 and Figure 7.

	Isobutan		**Butane**		**DME**	
$T_{Int.St.}$, °C	$COP_{Int.St.}$	$COP_{Sim.}$	$COP_{Int.St.}$	$COP_{Sim.}$	$COP_{Int.St.}$	$COP_{Sim.}$
20	0.198	0.182	0.225	0.210	0.204	0.192
25	0.192	0.179	0.219	0.206	0.199	0.189
30	0.186	0.176	0.213	0.204	0.194	0.186
35	0.179	0.173	0.206	0.201	0.189	0.184
40	0.173	0.171	0.199	0.199	0.183	0.182
	CF3I		**R365mfc**		**R236ea**	
$T_{Int.St.}$, °C	$COP_{Int.St.}$	$COP_{Sim.}$	$COP_{Int.St.}$	$COP_{Sim.}$	$COP_{Int.St.}$	$COP_{Sim.}$
20	0.212	0.200	0.267	0.245	0.228	0.206
25	0.207	0.197	0.260	0.242	0.220	0.202
30	0.202	0.194	0.251	0.238	0.212	0.198
35	0.197	0.191	0.243	0.236	0.204	0.195
40	0.191	0.189	0.234	0.233	0.195	0.192

Table 2. Results of two-stage ejector cycle calculation [2]

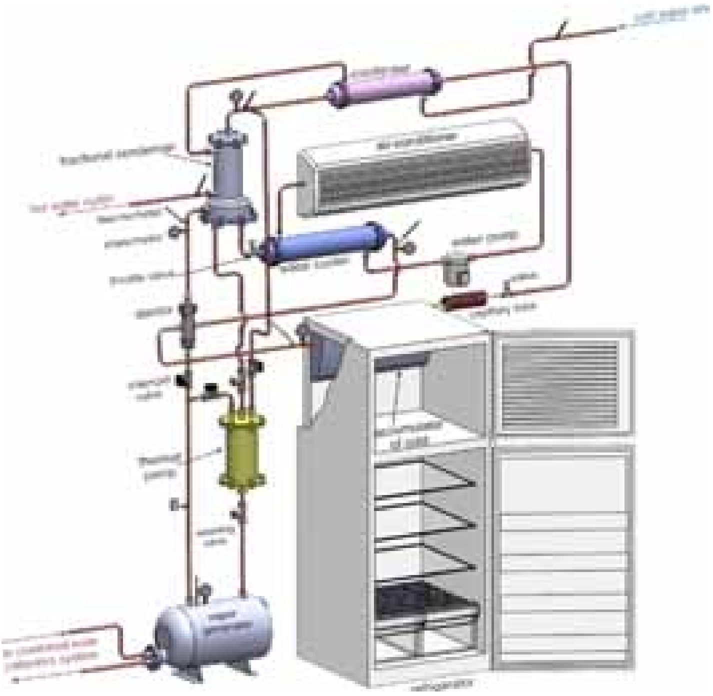

Fig. 3. Schematic of a binary ERS with one ejector

The gain in COP for the two-stage system was as high as 10%, which is significant because the contribution of bottom stage to the cooling capacity of the system is only 1-3%. The increase in the COP in the modes with high temperatures of condensation was especially notable. In this calculation, the condensation temperature was assumed to be 45 °C.

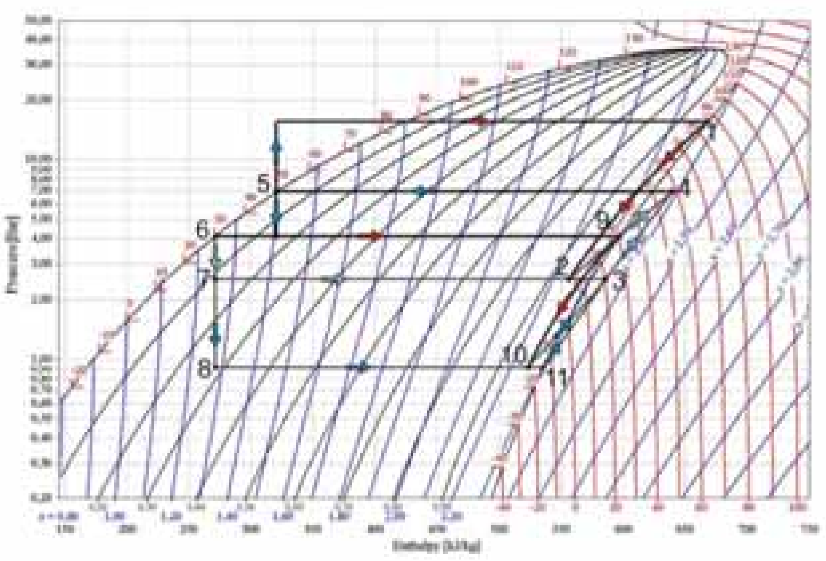

Fig. 4. Cycle of a two-stage ejector refrigerating system

An alternative to the use of graded compression in the ERS was the use of a binary and a multicomponent ERS, which can produce multi-level cold, using both a mixture with a small drop in normal boiling point and with high drop for the cascade scheme (Fig. 6). In the first case, when the vapor condensation of the low-boiling component is performed by the environment, but high-level cold is not yet achieved, the ERS can be roughly assumed to be working almost as an expander-compressor unit, i.e. the high-boiling component conducts a direct Rankine cycle and the low-boiling component conducts a refrigeration cycle. Their common path covers the ejector-condenser path, and then each flow is independent. In the second case, a part of the high-boiling component is throttled and removes condensation heat of the low-boiling component in the evaporator-condenser. In a cascade scheme, either much lower evaporation temperatures or work with a very high condensation temperature may be obtained, i.e., use of an air-cooled condenser, even at peak temperatures.

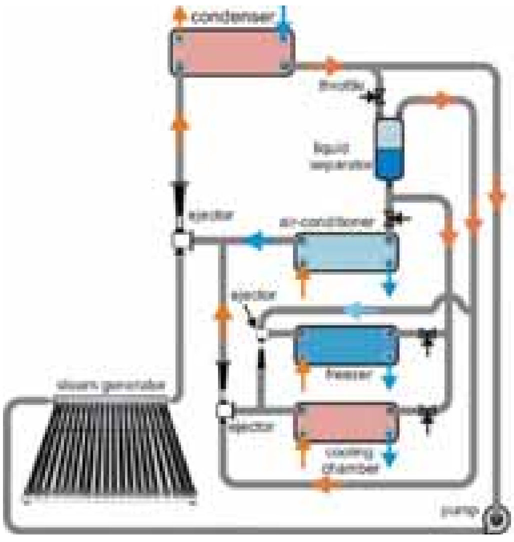

Fig. 5. Schematic of a solar three-stage ERS

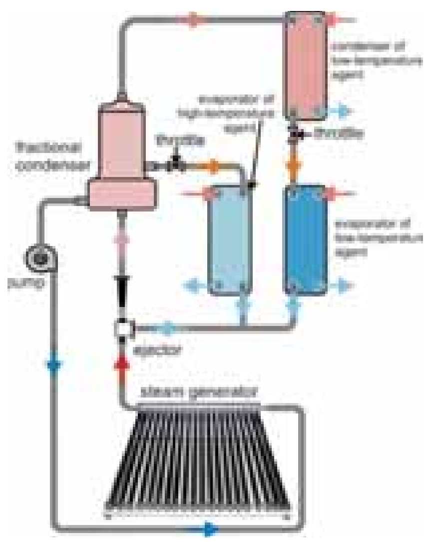

Fig. 6. Schematic of a binary ERS

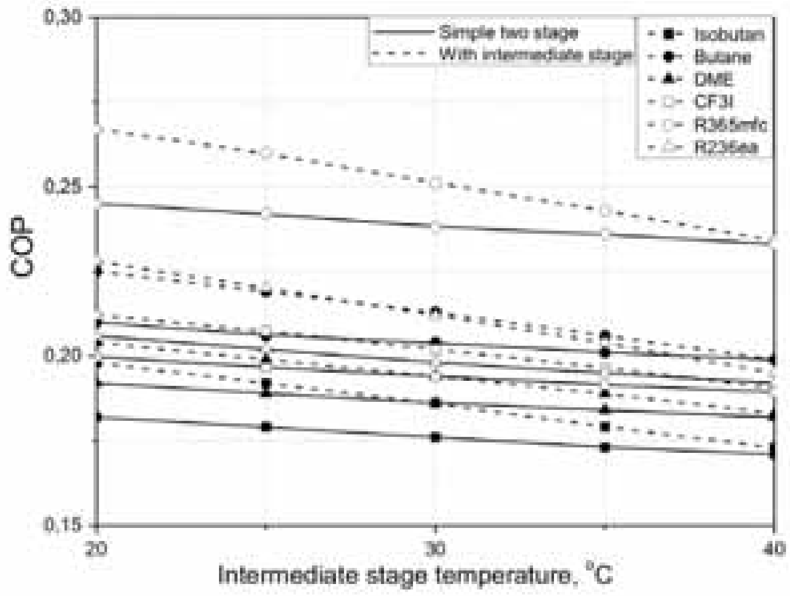

Fig. 7. Results of ejector two-stage ejector cycle calculations

3.1. Ways of air-conditioning in different climatic zones

In hot climate countries consumption of artificial cold in the housing and communal sphere is rapidly growing and does not depend on the season. In torrid zones, where there are two seasons, dry and moist, the necessity for air-conditioning is virtually the same over the year and throughout the day. In a continental climate condition, where relative humidity is between 12-20%, nighttime temperatures can drop to 3-7 °C, while the daytime temperatures can reach 35-40 °C. Therefore, the concepts of air-conditioning in these regions are fundamentally different.

In desert areas (Sahara Desert, The Karakum Desert, Kalahari Desert, etc.) it is reasonable in the winter period to accumulate the natural cold of the environment during the night and consume it during the day. This leads to significant energy savings in comparison with all means of artificial cold generation. During the summer, application of evaporative cooling of air is sufficient at a given relative humidity and to a certain ambient temperature. In the rest of the period, artificial mechanical cooling must be applied in combination with the evaporative cooling and the use of solar energy, i.e., solar thermotransformers, such as ejector refrigerating systems (ERS).

In humid tropical climates (Amazonia, Southeast Asia, equatorial Africa), air-conditioning that consists of preliminary air drying by mean of solar energy, and further air cooling with the solar ejector refrigeration system (SERS) is the most acceptable in comparison with existing systems of air-conditioning, where the moisture is removed and carried from the cooling surfaces.

In temperate climates, when parameters of air outside insignificantly differ from the parameters of air inside air-cooling is performed without moisture fall. When the temperature of the cooling surface is lower than the dew point, moisture fall is observed. This, on the one hand, decreases the heat-exchange surface of the air-cooler, but, on the other hand, leads to electric energy overconsumption for production of lower temperature cold and overconsumption of cold for moisture condensation. Large amounts of moisture occur particularly in tropical climate conditions, when outdoor air parameters significantly differ from the indoor air parameters. In this case, infiltration of hot and moist air from the outside is increased and this demands intensive moist removal. Application of two-stage air-conditioning can be one solution to this situation, which relies on preliminary dehumidification of moist air before cooling at the expense of solar energy. Preliminary solar drying in the tropics makes it possible to reduce the necessity of produced cold nearly two fold, given otherwise equal conditions. This, along with the appropriate building insulation, minimizes operational costs for air-conditioning (Fig. 8)

Moistening of the air in a dry hot climate leads to similar results, and for certain combinations of temperature and relative humidity of outside air, artificial refrigeration may not be necessary (Fig. 9)

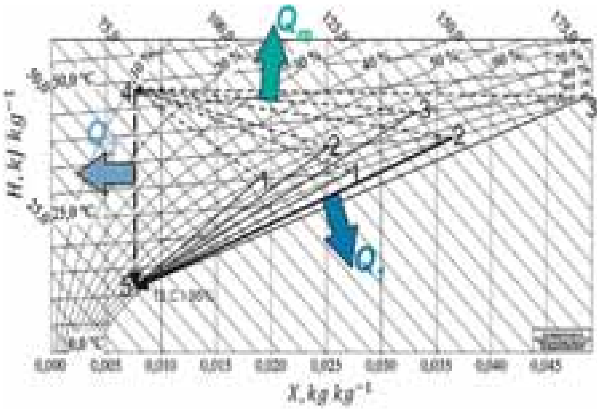

Fig. 8 Processes of two-stage cooling of moist air
Q1 – heat removed in a cooler without preliminary cooling;
Q2 – heat removed in a cooler with preliminary cooling;
Qm – heat removed along with the moisture removal.

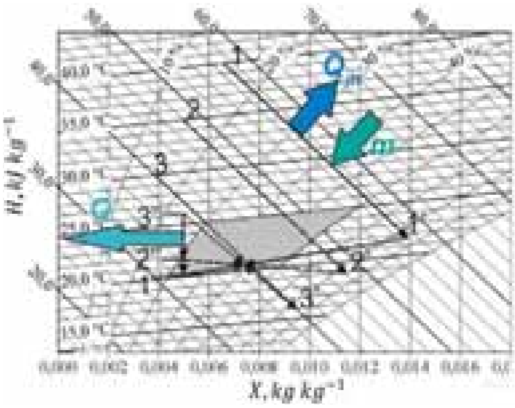

Fig. 9 Processes of two-stage cooling of dry air; Q_m – heat removed in the process of evaporative cooling; Q - heat, removed from dry air in the cooling process; m - the amount of water supplied.

4. Conclusions

In this study, an integrated approach to solving the problem of the solar cooling systems using different types of ERS is highlighted. CFD models of single-fluid and binary ejectors are shown. Solar air-conditioning is considered in different climatic conditions, which gives an opportunity to minimize energy consumption in each case. The results can be the input data for further design of ERS modules for simultaneous air-conditioning of individual houses, condominiums and public buildings and the working of household refrigerators and other consumers of low-temperature cold.

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