

THE EFFECT OF EJECTOR GEOMETRY ON PERFORMANCE OF EJECTOR-ABSORPTION REFRIGERATION SYSTEM

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ABSTRACT

In this study, the effect of ejector geometry on performance of an ejector-absorption refrigeration system (EARS) operating with the aqua/ammonia was investigated. By using of the ejector, the obtained absorber pressure becomes higher than the evaporator pressure and thus the system works with triple-pressure-level. The ejector has four functions: it (i) aided pressure recovery from the evaporator, (ii) upgraded the mixing process and the pre-absorption by the weak solution of the ammonia coming from the evaporator, (iii) increased absorber temperature and (iv) pre-absorption in the ejector improves the efficiency of the EARS. Performance improvement by ejector geometry under maximum COP and ECOP conditions is around 2.3% for COP and 4.7% for ECOP. CFD analysis of the geometry of the ejector also functions defined in theory has been found to fulfill visually.

Keywords: Absorption refrigeration system, COP, ejector

EJEKTÖR GEOMETRİSİNİN EJEKTÖRLÜ ABSORPSİYONLU SOĞUTMA SİSTEMİ PERFORMANSINA ETKİSİ

ÖZET

Bu alıřmada, amonyak/sui le alıřan ejektörlü absorpsiyonlu soğutma sisteminde ejector geometrisinin system performansı üzerine etkileri araştırılmıştır. Ejektör kullanılmasıyla, elde edilen absorber basıncı evedaporator basıncından daha yüksek olur ve böylece system üçlü basınçla alışır. Ejektörün dört fonksiyonu vardır: (I) evedaporatördeki basıncın iyileştirilmesine yardım eder, (ii) evedaporatörden gelen amonyakça zayıf eriyiğın ön absorplanmasını sağlar ve karışım işlemini iyileştirir, (iii) absorber sıcaklığını artırır, (iv) ejektördeki ön absorplama sistemin verimini iyileştirir. Maksimum COP ve ECOP değerlerinde ejector COP değerini %2.3, ECOP değerinde de %4.7 iyileşme sağlamıştır. Ayrıca CFD analizi ile ejector geometrisinin teorik olarak tanımlanan fonksiyonlarını görsel olarak yerine getirdiği tespit edilmiştir.

Anahtar Kelimeler: Absorpsiyonlu Soğutma Sistemi, COP, ejektör

NOMENCLATURE	
c_p	Specific heat at constant pressure (J/kg K)
E	Exergy rate (W)
f	Circulation ratio
h	Specific enthalpy (J/kg)
\dot{m}	Mass flow rate (kg/s)
p	Pressure (bar)
\dot{q}	Heat transfer rate (W)
s	Specific heat at constant pressure (J/kg K)
T	Exergy rate (W)
T_o	Circulation ratio
x	Specific enthalpy (J/kg)
\dot{W}	Mass flow rate (kg/s)
v	Pressure (bar)

SUBSCRIPTS	
a	Condenser
c	Evaporator
e	Generator
g	Pump
p	Total
t	Component
i	Condenser

1. INTRODUCTION

Ejector-absorption refrigeration systems (EARSs) are a promising system for recovering low-level waste heat and are attractive for renewable energy such as geothermal energy, solar energy, etc. In recent years, large quantities of heat from industries have been rejected to the atmosphere as wastewater or waste steam, which not only wastes energy but also pollutes the atmosphere [1]. The EARS can effectively recover about 50% of the waste heat and reuse it in industrial processes, especially in petrochemical and chemical plants. The objective of this is to analyse EARS thermodynamically and to determine the performance as a function of the operating levels of temperatures.

The choice of the most appropriate refrigerant/absorbent pair is as important as system design and optimisation of parameters. Though the water/lithium bromide mixture has been the only mixture used commercially in the EARSs, it has some major disadvantages such as the low working pressure, its limited gross temperature lift. For these reasons, new working pairs have been proposed and used in EARSs [1-2-3-4-5]. As shown in these studies, the aqua-ammonia combination has acceptable thermo physical properties for the EARSs. In order to improve the EARS performance, researches on the design of system components and applications of various configurations on the system have been still constantly going on.

In the present study, in the EARS with working aqua-ammonia was used a specially ejector was located at the absorber inlet. As is well known, ejectors can increase the pressure and do not consume mechanical energy directly, which are the main characteristics of ejectors [6-7-8-9]. Due to these characteristics, applying an ejector may be simpler and safer technologically than applying mechanical devices, which can increase pressure, such as a compressor, pump, etc. Besides the ejector's very simple configuration, the systems combining ejectors and other devices are also very simple [10].

In this work, a combination of computerized simulation program was developed to examine the influence of the different ejector geometry on

performance of the EARS. In addition, exergy losses for each component in the system were calculated at selected working temperature. The results of the exergy analyses can be used to identify the effect ejector geometry on performance of EARS.

2. SYSTEM DESCRIPTION

EARS is a device, which can deliver heat at a higher temperature than the temperature of the fluid by which it is fed. The characteristics of the EARS are [11].

- It can transfer low-grade heat to high-grade heat, while it consumes only little work done by the pumps
- It has no rotary device except pumps, so it is very simple in configuration, easy to operate and maintain and has a higher life expectancy
- It can reduce energy loss and discharge of CO₂, thus reducing the greenhouse effect.

The EARS basically consists of: an evaporator, a generator, a condenser, an absorber, an economizer, an ejector, pumps, valve (Figure 1). A quantity of waste heat q_g is added at a relatively low temperature T_g to the generator to vaporise the working fluid. The vaporised working fluid flows to the condenser delivering an amount of heat q_c at a reduced temperature T_c . The liquid leaving the condenser is pumped to the evaporator where it is evaporated by using a quantity of waste heat q_e at an intermediate temperature T_e . After this, the vaporised working fluid flows to the absorber where it is absorbed for the strong solution coming from the generator and delivers heat q_a at a high temperature T_a . Finally the weak salt solution is returned to the generator preheating the strong solution in the economiser before repeating the cycle again.

It can be seen in Figure 1 that the EARS operates with three pressure levels and three temperature levels when the same waste heat is supplied to the generator and the evaporator. The highest temperature is obtained in the

absorber when the heat is supplied to the generator and the evaporator at an intermediate temperature. Therefore, upgraded heat is delivered from the absorber.

A schematic representation of an ejector is given Figure 2. An ejector cycle is similar to a mechanical vapor-compression system except that a liquid pump, boiler and ejector replace the compressor. Operation at two different pressure levels for vapor-compression systems is changed to three different pressure levels for ejector cycles; ejector cycles may be powered by thermal energy [6].

The ejector mainly consists of a nozzle, a mixing tube and a diffuser. The ejector is characterized by the fact that there are no moving parts and no requirement for additional energy source. At the nozzle outlet, production of a full cone spray of small high-velocity drops of a weak solution serves to transform potential energy into kinetic energy. At the diffuser inlet, the stream of drops mixes with the refrigerant vapour coming from the evaporator. The diffuser pressure recovery is obtained by transforming the droplet kinetic energy into potential energy. The velocities of both the absorbent drops and the refrigerant gas are reduced. The influences of the ejector design parameters on the pressure recovery; temperature and concentration of the refrigerant in the solution and velocities of the gas and liquid drops have previously been described by Levy in [12-13-14].

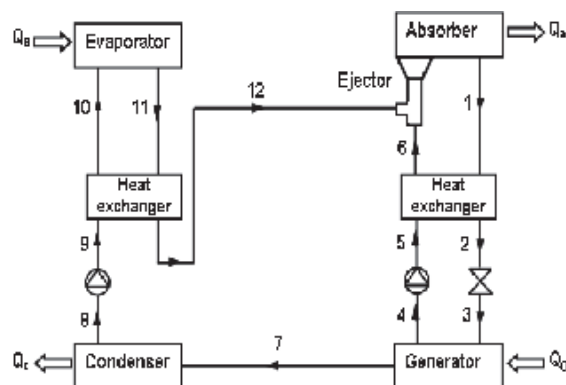


Figure 1. Schematic representation Ejector-Absorption Refrigeration System (EARS)

3. THERMODYNAMIC ANALYSIS

3.1. ANALYSIS OF EJECTOR

The heated solution leaving the heat exchanger (stage 6) as a primary fluid enters the nozzle, of the ejector in which it expands rapidly. In the exit of the nozzle the solution gets very rapid with very low pressure and draws the working fluid vapor (stage 12) from the evaporator as the secondary fluid. The two fluids mix fully in the mixing tube of ejector. The mixture leaves the diffuser at the absorber pressure, $P_a = P_{1'}$, and enters the absorber. To make easy the design of the ejector for absorption systems, a numerical model of simultaneously mass, momentum and heat transfer equations for ejector system was evaluated in Ref. [12-13-14].

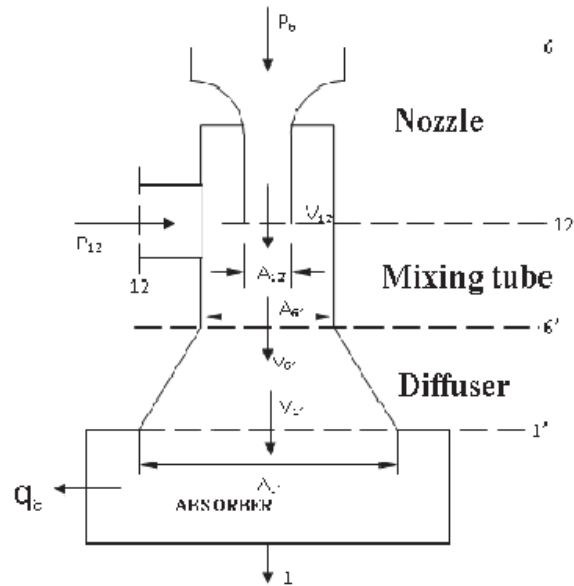


Figure 2. A schematic representation an ejector- absorber model

The exit velocity from the nozzle is calculated from

$$V_{12'}^2 = \eta_N 2(P_6 - P_{12'}) / \rho_6 \quad (1)$$

The mass flow rate is

$$\dot{m}_6 = \rho_6 V_{12'} A_{12'} \quad (2)$$

The mixture mass flow rate is

$$\dot{m}_{6'} = \dot{m}_6 + \dot{m}_{12} = A_{6'} \cdot V_{6'} \cdot \rho_{6'} = \dot{m}_1 \quad (3)$$

A momentum balance of the mixing tube yields:

$$\dot{m}_6 \cdot V_{12'} + P_{12'} \cdot A_{6'} = (\dot{m}_6 + \dot{m}_{12}) V_{6'} = P_{6'} \cdot A_{6'} \quad (4)$$

Combining the above equations, we can obtain the expression for the pressure rise in the mixing tube from

$$\frac{P_{6'} - P_{12'}}{\frac{1}{2} \rho_{12'} \cdot V_{12'}^2} = 2 \left(\frac{A_{12'}}{A_{6'}} \right) - 2 \left(\frac{\dot{m}_{12} + \dot{m}_6}{\dot{m}_6} \right)^2 \left(\frac{\rho_6}{\rho_{6'}} \right) \left(\frac{A_{12'}}{A_{6'}} \right) \quad (5)$$

The energy equation is

$$\eta_D \left(\frac{\rho_{6'} \cdot V_{6'}^2 - \rho_{1'} \cdot V_{1'}^2}{2} \right) = \frac{1}{\rho_{1'}} (P_{1'} - P_{6'}) \quad (6)$$

The diffuser exit velocity is

$$V_{1'} = \frac{1}{A_{1'} \cdot \rho_{1'}} \dot{m}_1 \quad (7)$$

The absorber pressure is $P_a = P_1 = P_{1'}$ and $\dot{m}_a = \dot{m}_{1'}$

The compression ratio, ε , is defined as the ratio of the absorber pressure and the evaporator pressure

$$\varepsilon = P_a / P_e \quad (8)$$

When we know the structure parameters of the ejector, such as $A_{12'}$, $A_{6'}$ and $A_{1'}$, and each component efficiencies of ejectors, we can make appropriate assumptions. The rates for ejector geometric areas are defined as follows:

$$\alpha = A6/A/12 \quad (9)$$

$$\beta = A12'/A/6' \quad (10)$$

$$\gamma = A6'/A/1 \quad (11)$$

3.2. ANALYSIS OF THE EARS CYCLE

3.2.1. Thermal Physical Properties of Working Fluids

As a function of saturation temperature the saturation pressure of the aqua-ammonia mixture is given by Bourseau and Bugarel [15] as;

$$\text{Log } P = A - \frac{B}{T} \quad (12)$$

Where the values of A and B are defined as follows:

$$A = 10.440 - 1.767x + 0.9823x^2 + 0.3627x^3 \quad (13)$$

$$B = 2013.8 - 2155.7x - 1540.9x^2 - 194.7x^3 \quad (14)$$

In this work, the equations derived by Ziegler and Trepp and Shultz [16-18]. For aqua-ammonia mixtures using the Gibbs free energy functions in both the liquid and gas phases in are used. The specific volume, enthalpy and entropy are expressed as a function of the Gibbs free energy, respectively, as follows:

$$v = \left[\frac{\partial g(T, p)}{\partial p} \right]_T \quad (15)$$

$$h = -T^2 \frac{\partial}{\partial T} \left[\frac{g(T, p)}{T} \right]_p \quad (16)$$

$$s = - \left[\frac{\partial g(T, p)}{\partial T} \right]_p \quad (17)$$

3.3. ENERGY METHOD (THE FIRST LAW ANALYSIS)

The mass fraction of the mixture at different points of the system (Figure 1) is calculated using the corresponding temperature data given in Table 1.

Table 1. Data used in the analysis

Condenser temperature, T_c	25 °C
Evaporator temperature, T_e	-10, 0, 10 °C
Generator temperature, T_g	40-130 °C
Absorber temperature, T_a	30 °C
Ambient temperature, T_o	20 °C
Economizer effective	0.9

The heat transfer rates in some of the components of the EARS are given as:

$$\text{Evaporator } \dot{q}_e = \dot{m}_3(h_4 - h_3) \quad (18)$$

$$\text{Condenser } \dot{q}_c = \dot{m}_1(h_1 - h_2) \quad (19)$$

$$\text{Generator } \dot{q}_g = \dot{m}_7 h_7 - \dot{m}_1 h_1 - \dot{m}_8 h_8 \quad (20)$$

$$\text{Absorber } \dot{q}_a = \dot{m}_1(h_1' - h_1) \quad (21)$$

The coefficient of performance of the EARS is equal to the heat load in the absorber per unit heat load in the generator and the evaporator plus the work done by the pumps.

$$\text{COP} = \frac{\dot{q}_e}{\dot{q}_g + \dot{W}_{p1} + \dot{W}_{p2}} \quad (22)$$

The exergetic coefficient of performance of the EARS is defined as:

$$\text{ECOP} = \frac{\dot{q}_e(1 - T_0/T_e)}{\dot{q}_g(1 - T_0/T_g) + \dot{W}_{p1e} + \dot{W}_{p2e}} \quad (23)$$

4. RESULTS AND DISCUSSION

At each reference point of the EARS the mass flow rate and enthalpy of the mixture are determined for different working conditions. Using these parameters the COP and ECOP of the EARS are calculated within the prescribed temperature limits. The working temperatures given in Table 1 were used for the different working conditions of the EARS.

Figure 3 shows the changes in COP value with different generator and evaporator temperatures based on α . System performance (COP) is highest with high evaporator temperatures and high α values. Figure indicates that high α value positively affects the system performance for all evaporator temperatures. In other words, the small cross-sectional area of liquid coming from evaporator and entering in ejector makes positive contributions to system performance. From this regard, the effects of β in ejector geometry on system performance were investigated by keeping α constant at 2 and the results are given in Figure 4.

In Figure 4, the highest performance was obtained with $\beta=0.5$ for all evaporator temperatures. In Figure 5, γ value was changed by keeping both rates constant ($\alpha=2$, $\beta=0.5$). Here, maximum performances were obtained with $\gamma=0.5$ for all evaporator temperatures. The system functions at maximum performance on condition that generator temperature is 60°C , and other parameters are as follows: $T_e=10^{\circ}\text{C}$, $\alpha=2$, $\beta=0.5$ and $\gamma=0.5$. Similarly, ECOP of the system was calculated considering the abovementioned geometric rates and given in Figure 6-8. Maximum ECOP reached 37.7%, while other parameters occurred as follows: $T_e=-10^{\circ}\text{C}$, $T_g=84^{\circ}\text{C}$, $\alpha=2$, $\beta=0.5$, and $\gamma=0.5$. It was aimed to make both COP and ECOP performances of the system maximum by changing geometric rates. Thus, analytic results are established for optimum ejector design that makes system performance maximum (Table 1 and 2). Improvements for the maximum COP values of the system at the values given in Table 1 ($T_e=0^{\circ}\text{C}$ and $T_g=86^{\circ}\text{C}$) are determined as 2.3% at $\beta=0.5$ and 4.7% at $\gamma=0.5$. Similarly, improvements for ECOP were calculated as 2.3% at $\beta=0.5$ and 4.7% at $\gamma=0.5$ on average. Under maximum performance conditions, ECOP values occurred at lower generator temperatures compared to COP values. The properties of Ejector Geometry given in the Abstract section were improved at $\alpha=2$, $\beta=0.5$ and $\gamma=0.5$.

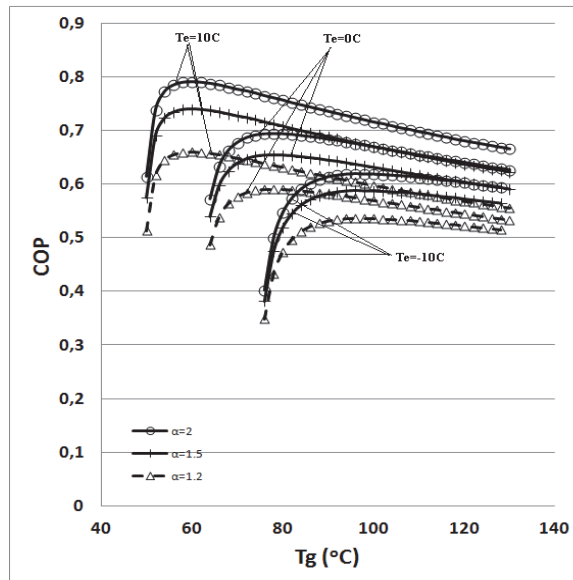


Figure 3. The variation of COP of the EARS at different evaporator and generator temperatures with different the rates of α ($T_a=30^\circ\text{C}$ $T_c=25^\circ\text{C}$)

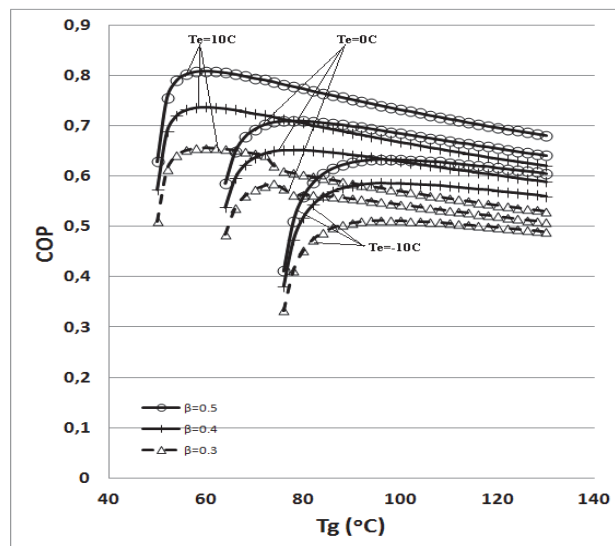


Figure 4. The variation of COP of the EARS at different evaporator and generator temperatures with different the rates of β ($\alpha=2$, $T_a=30^\circ\text{C}$, $T_c=25^\circ\text{C}$)

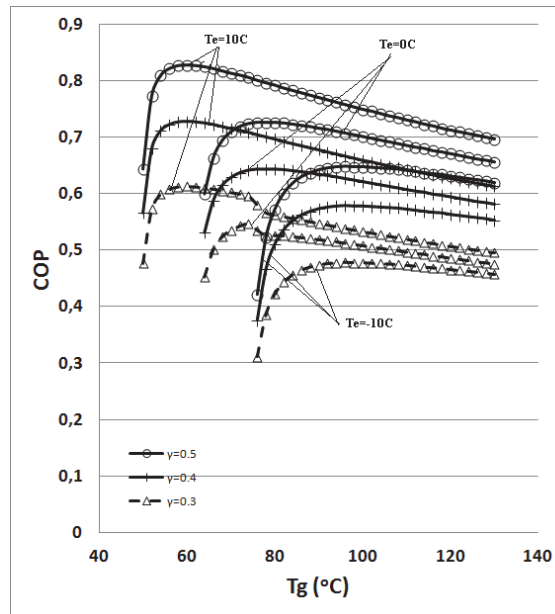


Figure 5. The variation of COP of the EARS at different evaporator and generator temperatures with different the rates of γ ($\alpha=2, \beta=0.5, T_a=30^\circ\text{C}, T_c=25^\circ\text{C}$)

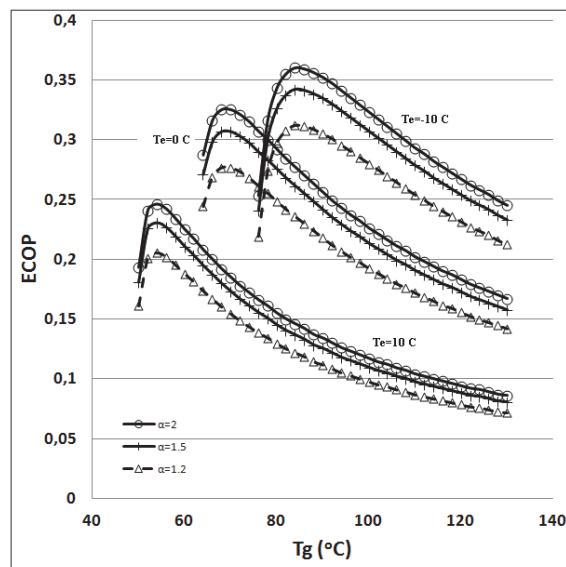


Figure 6. The variation of ECOP of the EARS at different evaporator and generator temperatures with the different rates of α ($T_a=30^\circ\text{C}, T_c=25^\circ\text{C}$)

5. CFD ANALYSES OF VARIOUS EJECTOR GEOMETRIES

In order to observe the effects of variations in ejector geometry upon the degree to which an ejector accomplishes the four expected operational requirements mentioned in the abstract, a CFD flow analysis was performed. In CFD evaluations, the ejector input conditions were considered in accordance with the results of the thermodynamic analysis made based on the operating point parameters listed in Table 1.

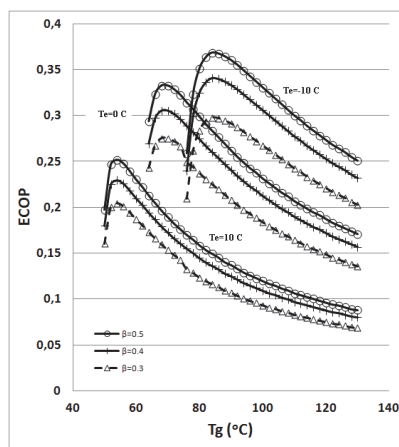


Figure 7. The variation of ECOP of the EARS at different evaporator and generator temperatures with the different rates of β ($\alpha=2$, $T_a=30^\circ\text{C}$, $T_c=25^\circ\text{C}$)

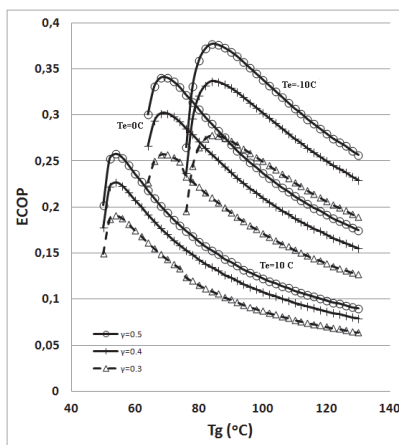


Figure 8. The variation of ECOP of the EARS at different evaporator and generator temperatures with the different rates of γ ($\alpha=2$, $\beta=0.5$, $T_a=30^\circ\text{C}$, $T_c=25^\circ\text{C}$)

Table 2. Maksimum COP values

Te (°C)	Tg (°C)	$\alpha=2$	$\alpha=1.5$	$\alpha=1.2$
-10	96	0,619	0,58805	0,536302
0	86	0,688	0,648784	0,585398
10	60	0,79	0,739361	0,65951
		$\beta=0.5$	$\beta=0.4$	$\beta=0.3$
-10	96	0,633237	0,586156	0,511793
0	86	0,703824	0,646695	0,558645
10	60	0,80817	0,736958	0,655751
		$\gamma=0.5$	$\gamma=0.4$	$\gamma=0.3$
-10	96	0,648086	0,578771	0,47827
0	86	0,720329	0,638547	0,522054
10	60	0,827162	0,727672	0,612799

Table 3. Maksimum ECOP values

Te (°C)	Tg (°C)	$\alpha=2$	$\alpha=1.5$	$\alpha=1.2$
-10	84	0,36	0,342	0,311904
0	70	0,325	0,306475	0,276532
10	54	0,246	0,230231	0,205366
		$\beta=0.5$	$\beta=0.4$	$\beta=0.3$
-10	84	0,36828	0,340899	0,29765
0	70	0,332475	0,305479	0,274956
10	54	0,251658	0,229483	0,204196
		$\gamma=0.5$	$\gamma=0.4$	$\gamma=0.3$
-10	84	0,376916	0,336603	0,278154
0	70	0,340288	0,30163	0,256947
10	54	0,257572	0,226592	0,190821

Figures 9-11 depict the pressure distributions inside the mixing region for $\alpha = 1.2, 1.5$ and 2 , respectively. According to the pressure distributions of the cooler and the absorber at the mixing point, it can be seen that a higher amount of pressure was obtained for $\alpha = 2$ (Figure 11). The new operating pressure, the pressure that yielded a triple-pressure system and which the ejector was expected to yield, was achieved for the high value of α .

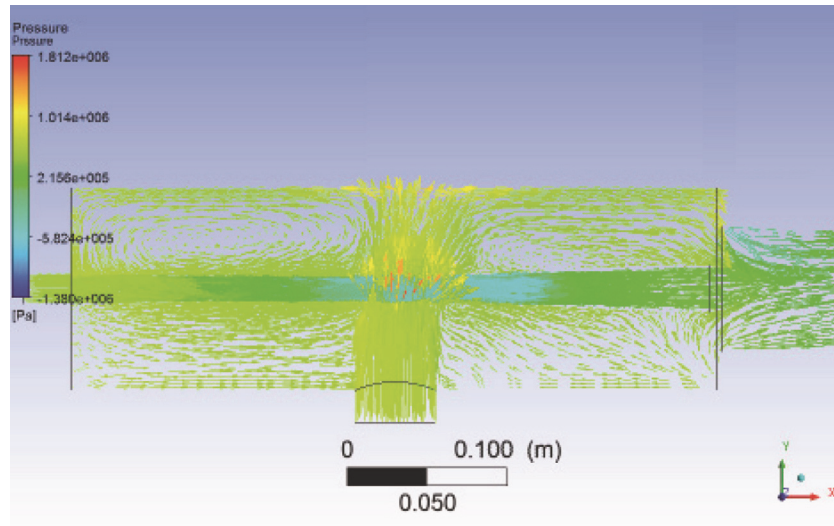


Figure 9. Mixing region CFD results for $\alpha=1.2$

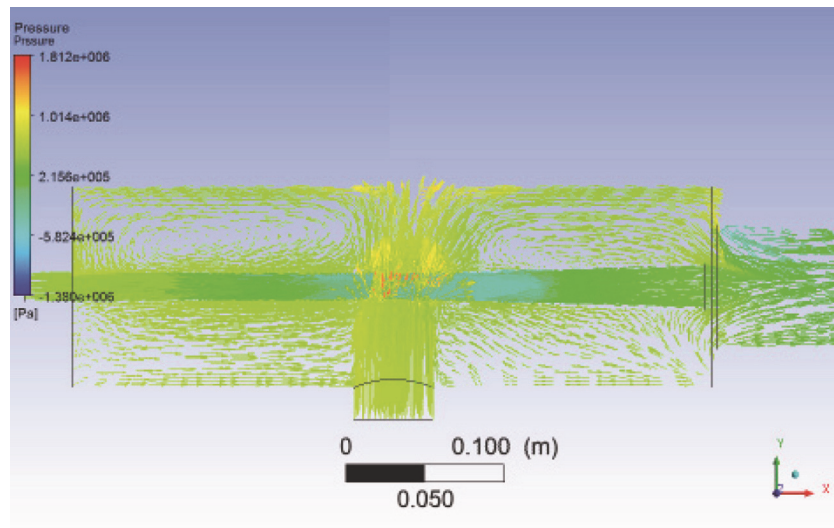


Figure 10. Mixing region CFD results for $\alpha=1.5$

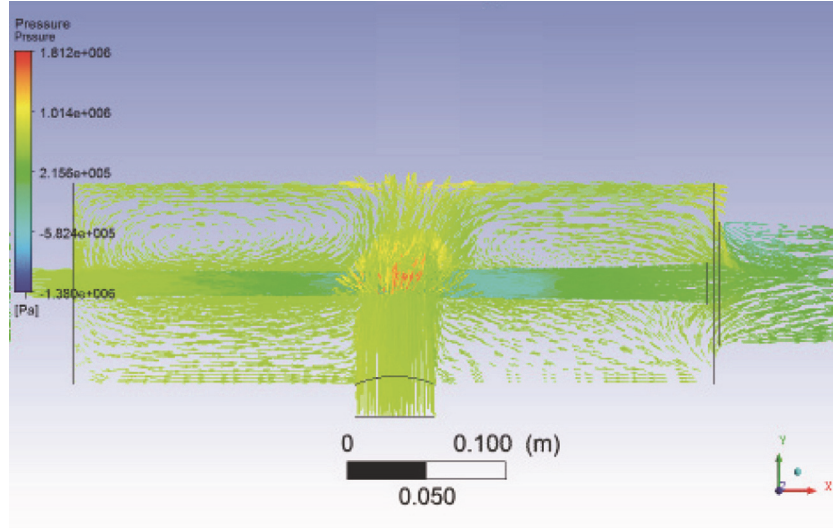


Figure 11. Mixing region CFD results for $\alpha=2$

Figures 12-14 show the CFD results regarding the change in the bulk concentrations of the cooler and the absorber fluids inside the ejector for $\beta = 0.3, 0.4$ and 0.5 in respective order. According to Figure 14, the absorber's cooler fluid preabsorption capacity, an operational figure of merit for the ejector, was significantly higher at $\beta = 0.5$. The more the absorber absorbs the cooler, the more the cooling capacity increases. The ejectors in turn preabsorb the dilute cooler solution coming from the evaporator.

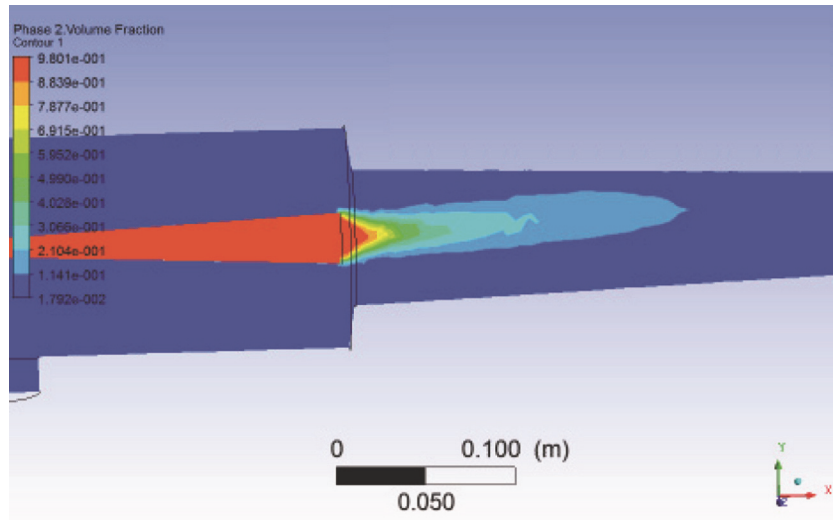


Figure 12. Mixing region CFD results for $\beta=0.3$

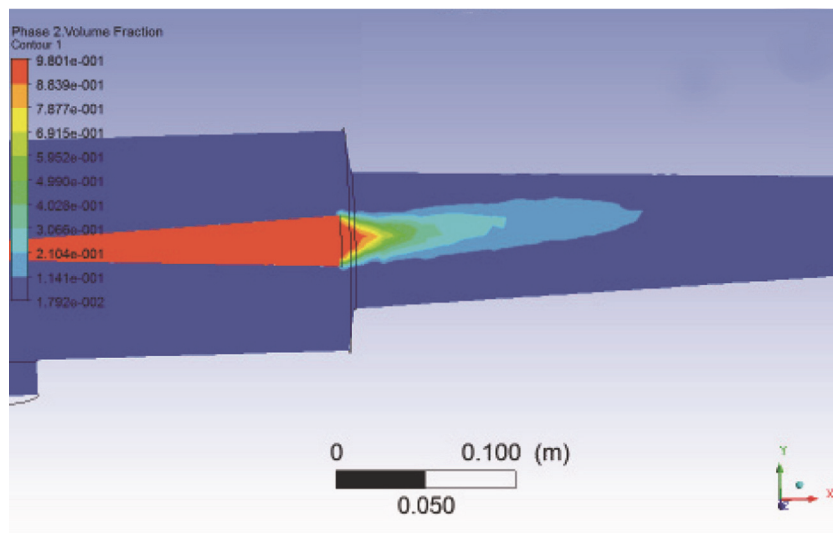


Figure 13. Mixing region CFD results for $\beta=0.4$

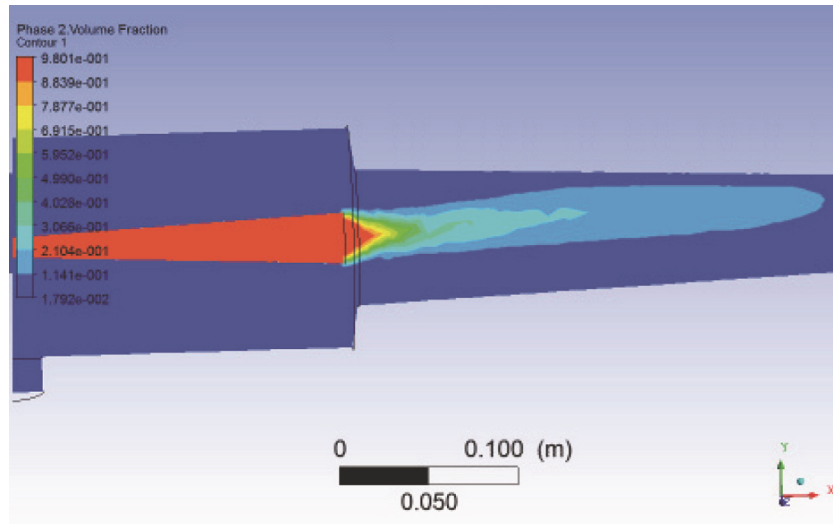


Figure 14. Mixing region CFD results for $\beta=0.5$

As Figure 15 indicates, at $\beta = 0.5$, the cooler concentration decreased whereas the absorber concentration increased. This implies that the geometry for which the absorber preabsorbs the cooler develops at $\beta = 0.5$.

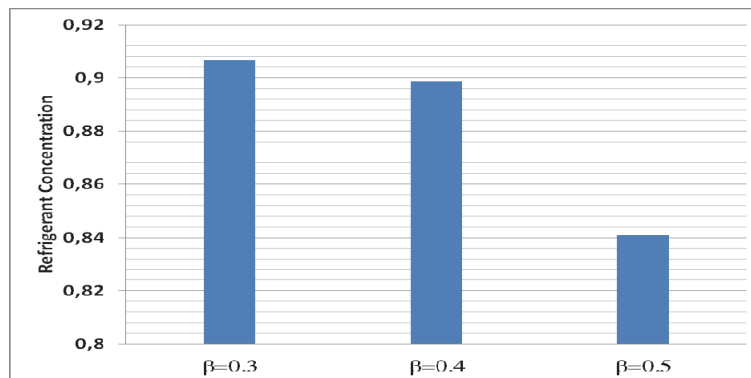


Fig.15. Variation of refrigerant concentration for different β values

Figures 16-18 plot the temperature changes for $\gamma = 0.3, 0.4$ and 0.5 , respectively. The desired absorber temperature at the exit of the ejector mixing region was attained for $\gamma = 0.5$.

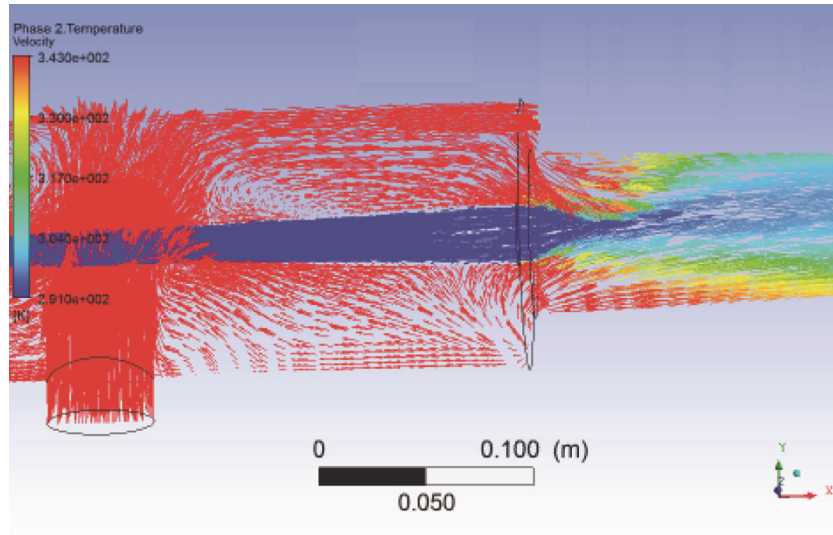


Figure 16. Mixing region CFD results for $\gamma=0.3$

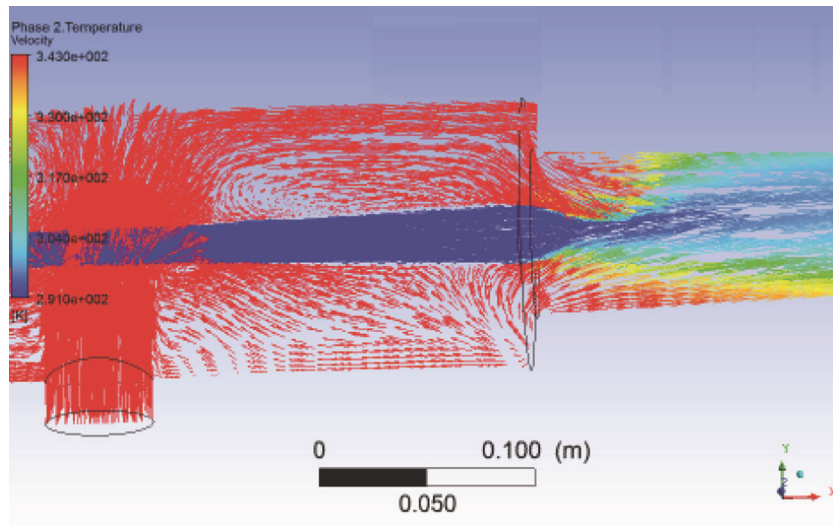


Figure 17. Mixing region CFD results for $\gamma=0.4$

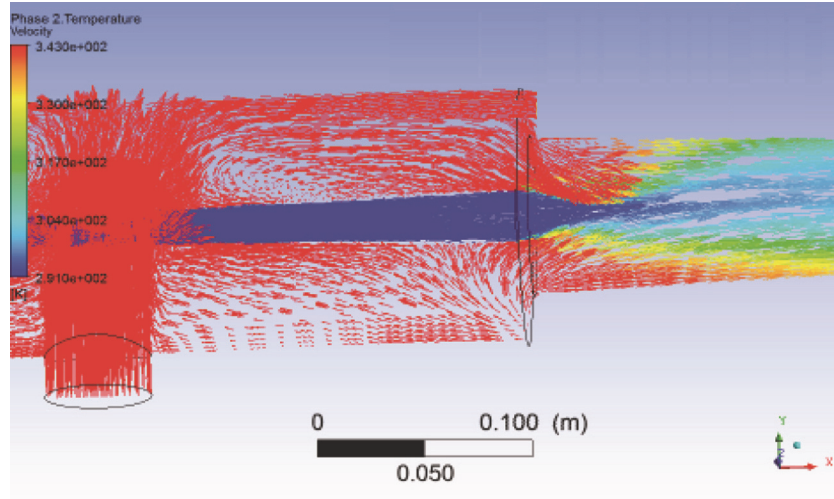


Figure 18. Mixing region CFD results for $\gamma=0.5$

6. CONCLUSION

Performance improvements obtained by using ejector in absorption cooling systems with quite low thermal efficiency was found directly related to ejector geometry with the help of thermodynamic simulation developed in the study. This study demonstrates the effects of ejector geometry rates on both performance and exergetic performance. The study will contribute to literature in that it shows even small effects of ejector geometry on performance in terms of design parameters.

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