Energy analysis and performance evaluation of a novel multi-evaporator ejector refrigeration cycle (ERC)

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This paper presents a theoretical analysis of triple-evaporator ejector refrigeration cycle (TEERC) for triple applications of cooling, freezing, and ventilation, based upon the first and second laws of thermodynamics. Nine appropriate working fluids (i.e., R717, R152a, R134a, R290, cis-2-butene, butane, isobutene, isobutane, R236fa) are presented for the proposed cycle based on the working fluid characteristics, cycle efficiency, and environmental consideration. Energetic and exergetic analyses of the proposed cycle have been performed leading to the determination of the main source of the irreversibility of the whole cycle. It was found that the generator has the main source of irreversibility which is followed by the ejector and condenser, respectively. The maximum and minimum coefficients of performance (COP) are obtained for R717 and R236fa by the values of 0.333 and 0.268, respectively. On the other hand, the maximum and minimum exergy efficiencies are calculated for R717 and isobutene by the values of 21.43% and 12/51 %, respectively. Also, using R717 as the best working fluid in this investigation, the ventilation, cooling and freezing capacities are obtained 11.68 kW, 3.86 kW, and 1.904 kW, respectively. At last, sensitivity analysis of some key parameters has been conducted in order to understand the characteristics of the proposed cycle, comprehensively. It has been shown that increasing of the evaporators and generator temperatures and decreasing of the condenser temperature increase both COP and exergy efficiency. Moreover, among all influential parameters, the ejector mass entrainment ratio has a stronger effect on the freezing, ventilation, and cooling capacities.

keywords: *Triple-evaporator ejector refrigeration cycle (TEERC); Cooling; Freezing; Ventilation; Energetic analysis; Exergetic analysis; Working fluid selection; Low-temperature heat source.*

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NOMENCLATURE

COP coefficient of performance

e specific exergy $(kJ.kg^{-1})$

$\dot{E}$ exergy rate (kW)

ERC ejector refrigeration cycle

h specific enthalpy $(kJ.kg^{-1})$

$\dot{m}$ mass flow rate $(kg.s^{-1})$

P pressure (MPa)

pf primary flow

$\dot{Q}$ heat transfer rate (KW)

s specific entropy $(kJ.kg^{-1}K^{-1})$

sf secondary flow

T temperature (K)

$TEERC$ Triple evaporator ERC

$Th.V$ throttling valve

in inlet

e evaporator

ej ejector

Ex exergy

F fuel

g generator

is isentropic

KN kinetical

L loss

m mixer

n nozzle

out outlet

P product

pf primary flow

PH physical

1. INTRODUCTION

Many researches have been done to explore the ejector refrigeration cycle (ERC) to utilize low-temperature heat sources (such as geothermal resources, solar energy, industrial water heats, etc.) to produce the required refrigeration. The scale of this refrigeration production can be either micro-scale or macro-scale which must be taken into account, appropriately. The technology of ejector refrigeration cycle can be much more suitable for micro-scales cooling production purposes, since the ejector produces some side effects (such as noise, vibration, etc.) in macro-scales.

In the vicinity of this topic, many researches have been done to introduce this awesome technology more comprehensively. Ghaebi et al. [1] presented energetic analysis of the standard ejector refrigeration cycle using six different working fluids, namely: isobutane, R152a, R134a, R600a, R290, and R245fa. They demonstrated that among all proposed working fluids, isobutane can be the best choice for the ERC on account for possessing of high coefficient of performance (COP) and low environmental impact. They had also conducted a parametric study for the ERC based on the first law of thermodynamics. They had shown that one can have a high COP for the ERC by increasing the evaporator temperature or decreasing the condenser temperature. Chen et al. [2] presented a thorough investigation on working fluid selection for ERC using three different types of working fluids (wet, dry, and isentropic fluids). They had concluded that R600 is the best working fluid due to its relatively high COP as well as low environmental impact. Smierciew et al. [3] presented an experimental investigation on the ejector air-conditioning using isobutane as working fluid. They had calculated the COP and loss coefficient of the ejector. Tashtoush et al. [4] investigated performance of the ejector refrigeration cycle at the critical mode, using R134a as an appropriate working fluid with consideration of the performance characteristics as well as the environmental impact. They had also specified some appropriate ranges for the condenser temperature, evaporator pressure and generator temperature. They had found that COP falls into the range of the 0.59-0.67 at the condenser back pressure of 24 bar.

In recent decades, the application of the ERC in the combined cooling, heating, and power (CCHP) systems has been highlighted due to its applicable mechanism [5,6]. For example, Ghaebi et al. [7] presented a novel combined cooling and power cycle by an appropriate integration of an organic Rankine cycle (ORC) and an ejector refrigeration cycle (ERC) to produce cooling output and power output, simultaneously. They had presented R113 and isobutane as the working fluid of the ORC and ERC, respectively, showing that this arrangement will result in maximum thermal efficiency of 34.69 %. They had also demonstrated that the proposed cycle with this pair of working fluids has the maximum exergy efficiency of 52.53 %. Wang et al. [8] presented a novel combined power and ejector refrigeration cycle for the cogeneration purposes, using Rankine cycle and the ejector refrigeration cycle. They have analyzed the proposed cycle using the concept of the first and second laws of thermodynamics. The analysis of this group had demonstrated that the generator has the main source of the irreversibility which is followed by the ejector and turbine, respectively. Habibzadeh et al. [9] combined ERC and ORC in a novel way using different working fluids of R123, R141b, R245fa, R600a, and R601a based on the classical laws (first- and second-laws). They had concluded that R601a has the highest thermal efficiency and the lowest overall exergy destruction rate. The application of ejector in the CCHP systems is much more interesting for solar energy heat sources, too [10,11]. In these researches, the concept of the ERC has been developed in a more general way. Xu et al. [12] combined a Brayton cycle and a transcritical ejector refrigeration cycle (ERC) to present a modified CCHP system, using supercritical CO2. Energy and exergy analysis of the proposed system were conducted showing that the exergy efficiency of the modified system is (10.4-22.5) % higher than that of the basic one. Furthermore, the parametric study of the system demonstrated that an increase in the turbine inlet temperature increases the power output and exergy efficiency, considerably. More recently, Megdouli et al. [13] combined vapor compression refrigeration cycle (VCRC) and ERC to introduce a new hybrid VCRC (HVCRC). Energy, exergy, and parametric studies were carried out using CO2 as a refrigerant. The results of this group demonstrated that the COP of the HVCRC is increased 25 %, while input power reduced by 20 % compared to that of the VCRC for the same cooling capacity. Li et al. [14] compared the performance of R1234yf and R134a in the ejector expander refrigeration cycle (ERC), reporting that R1234yf has a lower COP than R134a,. Liu et al. [15] presents a modified vapor compression refrigeration cycle (MVRC) for domestic applications in refrigerator/freezers, using zeotropic mixture R290/R600a. In their work, exergy efficiency raised up to 6.71 %, while the total exergy loss reduced up to 24.47 %. Investigation of the ejector as an expander is also boosted in recent years for energy's waste reduction purposes, based on the classical laws. Through these studies, the ejector has been used as an expander instead of the throttling valve in order to reduce this component exergy destruction rate [16,17]. This application of the ejector has been also investigated, experimentally [18]. For example, Ghaebi et al. [19] enhanced the performance of the traditional combined power and ejector refrigeration (TCPER) cycle by employing of an ejector expander in place of expansion valve based on the first- and second-law-efficiencies. They demonstrated that the use of an ejector in place of the expansion valve in the TCPER cycle can increase the cooling capacity up to 9.5 %, while the net produced work remains nearly constant.

This work aims at introducing of triple-evaporator ejector refrigeration cycle (TEERC) for three different production purposes. According to the above mentioned literatures, no investigation on use of ERC for various applications are carried out. To bridge this gap, three evaporators are used in parallel working at different temperatures for various applications for cooling, freezing, and ventilation. Application of this arrangement will be more highlighted in districted areas for multi-objectives cooling productions. The main goal of this paper is versatile and multi-objective which can be summarized as below:

- To produce simultaneous freezing output, cooling output, and the ventilation output from the ERC, using the triple-evaporator concept.
- To analyze the proposed cycle based on the first and second laws of thermodynamics.
- To suggest a couple of more appropriate working fluids for the proposed cycle based on the performance consideration as well as the environmental impact of the working fluids.
- To study the effect of some key parameters on the first- and second-law efficiencies.

2. CYCLE DESCRIPTION

Fig. 1 shows the schematic of the proposed TEERC for cooling production purposes. The proposed cycle consists from a vapor generator, an ejector, a condenser, a pump, three evaporators, and three throttling valves (TVs). The operation of the proposed cycle is simple and is as follows:

The heated working fluid in the generator enters the ejector as a primary flow (point 1) and draws the low pressure secondary flow of the superheated vapor of the evaporators (point 20) into the ejector. These two working fluids are then mixed at mixing chamber (point 3), and then enters into the condenser, where the condensation process happens by rejecting heat into the surroundings. The saturated liquid of the condenser is then divided into two streams. One stream goes through the pump (point 6) and then enters the generator (point 7), while completing the ERC operation process. On the other hand, the rest of the stream (point 8) is again divided into two parts. One goes into the throttling valve 1 (point 9) and then enters into the evaporator 1 (point 10), while ventilating the environment. The second part itself (point 11) divided into two parts, again. One part goes through the throttling valve 2 (point 12) and then enters into the evaporator 2 (point 13), while producing cooling outlet for refrigeration purposes. The second part (point 14) also enters the throttling valve 3, while expanding into the state 15 in an isobar process. This two-phase flow, then enters the evaporator 3 and produces the freezing output. In order to prevent bubble formation in the ejector, the outlets of all evaporators are superheated. The outlet flows of the evaporators are then mixed into a single flow (point 20), and then enters the ejector, as pointed out earlier.

3. MATHEMATICAL MODELING

A. THERMODYNAMICS ASSUMPTIONS

For the thermodynamic modeling of the developed cycle, a thermodynamic code is arranged on Engineering Equation Solver (EES) software which is written based on specific thermodynamic assumptions. These thermodynamic assumptions are as follows: [3,19,20]:

- All processes included in the cycle are at steady state.
- The flow inside the ejector system is one-dimensional flow.

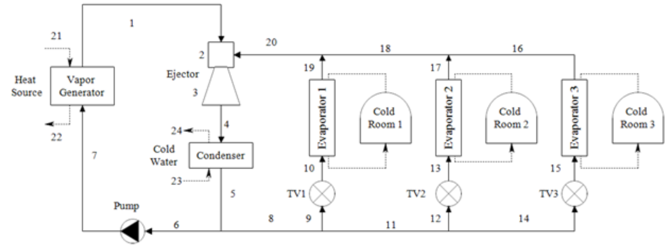

Fig. 1. The schematic of the proposed triple-evaporator ejector refrigeration cycle (TEERC).

- Temperature of the evaporator, generator, condenser, and pump in all process are assumed to be constant and are in the thermodynamic equilibrium.
- We treat ejector as a black-box model. In other words, we consider just inlet and outlet parameters of the ejector.
- Kinetic energy at the inlet and outlet of all components is negligible.
- Pressure and heat losses inside of the evaporator, generator, condenser, pump, and ducts are negligible.
- The reference state pressure and temperature are 0.101 MPa and 298 K.
- The isentropic efficiency of the pump is assumed 90 %.
- In mixing chamber of the ejector, constant pressure assumption is taken into the consideration.
- There is no heat transfer between the ejector and surroundings.
- The flow through the expansion valve is assumed to be an isenthalpic flow.
- The working fluid at the outlet of the generator is assumed at saturated vapor state. In contrary, the working fluid at the outlet of the condenser is assumed to be in saturated liquid state.
- Ejector nozzle, mixing, and diffuser efficiencies are 80, 95 and 80 %, respectively.

B. ENERGY ANALYSIS

From the first law of thermodynamics, once conservation of mass and energy are applied to each component, the properties of all states can be determined. These governing equations for energetic analysis of a cycle can be written as follows:

$$\sum \dot{m}_{in} - \sum \dot{m}_{out} = 0 \quad \textbf{(1)}$$

$$\sum (\dot{m}h)_{in} - \sum (\dot{m}h)_{out} + \sum \dot{Q}_{in} - \sum \dot{Q}_{out} + \dot{W} = 0 \quad \textbf{(2)}$$

The mass entrainment ratio of the ejector is another essential parameter in ejector refrigeration cycles. This parameter is defined as the mass flow rate of the secondary flow ($\dot{m}_{sf}$) to the primary flow ($\dot{m}_{pf}$) :

$$U = \dot{m}_{sf} / \dot{m}_{pf} \quad \textbf{(3)}$$

in which both $\dot{m_{sf}}$ and $\dot{m_{pf}}$ are in (kg/s).

The ejector entrainment ratio (U) is defined as the ejector suction mass flow rate at state (sf) divided by the motive mass flow rate at state (pf). Therefore, for 1 kg of the mixture in the ejector, the suction mass flow rate is $U/(1+U)kg$ and the motive mass flow rate is $1/(1+U)kg$.

The drive steam enters the ejector and expands to suction pressure (P_{suc}) with a nozzle efficiency defined as [20]:

$$\eta_n = \frac{h_{pf} - h_n}{h_{pf} - h_{n,s}} \quad \textbf{(4)}$$

in which $h_{n,s}$ is the corresponding isentropic state (n).

The energy balance between states (pf) and (n) is:

$$h_{pf} - h_n = \frac{1}{2}u_n^2 \quad \textbf{(5)}$$

By applying momentum conservation in the mixing section (n-m):

$$u_m = u_n/(1+U) \quad \textbf{(6)}$$

The energy balance for the ejector as a control volume can be written as follow:

$$h_{out} = h_{pf}/(1+U) + h_{sf}U/(1+U) \quad \textbf{(7)}$$

The mixing efficiency is given as:

$$\eta_m = \frac{u^2_{m'}}{u_m^2} \quad \textbf{(8)}$$

where,$u_{m'}$ is the corrected form of u_m, in order to account for mixing section losses. The energy balance equation between states (m) and (out) is:

$$h_{out} - h_m = \frac{1}{2}u^2_{m'} \quad \textbf{(9)}$$

The mixture recovers pressure in the ejector diffuser with a given efficiency of:

$$\eta_d = \frac{h_{out,s} - h_m}{h_{out} - h_m} \quad \textbf{(10)}$$

where $h_{out,s}$ is the corresponding isentropic enthalpy at the outlet of the ejector.

C. EXERGY ANALYSIS

Exergy of a system is defined as the maximum theoretical useful work which can be obtained as the system interacts to the equilibrium. If a system comes into an equilibrium state with its environment, then in this case its exergy will be completely destroyed. So, we can conclude that the value of exergy cannot be negative. Exergy destruction is also a vital parameter in exergy analysis, which indicates the main source of system loss by the means of each component. In the absence of magnetic, electrical, nuclear, and surface tension effects, the rate of total exergy of the system $(\dot{E}_{total})$can be divided into four components: physical exergy rate $(\dot{E}_{PH})$, kinetic exergy rate $(\dot{E}_{KN})$, potential exergy rate $(\dot{E}_{PT})$, and chemical exergy rate $(\dot{E}_{CH})$, [21]:

$$\dot{E}_{total} = \dot{E}_{PH} + \dot{E}_{KN} + \dot{E}_{PT} + \dot{E}_{CH} \quad \textbf{(11)}$$

and the specific exergy can be expressed as:

$$e_{total} = e_{PH} + e_{KN} + e_{PT} + e_{CH} \quad \textbf{(12)}$$

in which

$$e = \dot{E}/\dot{m} \quad \textbf{(13)}$$

which is more convenient to work with it. The sum of the kinetic, potential, and physical exergies of a system is called thermomechanical exergy.

The specific physical exergy of a closed system for different working fluids can be calculated from the following equation:

$$e_{PH} = h - h_0 - T_0(s - s_0) \quad \textbf{(14)}$$

in which are specific enthalpy and entropy of the substance, respectively, and h_0, s_0 are those parameters at reference state (dead state) of known pressure and temperature of (P_0, T_0).

The total and each component exergy destruction rate can be written as:

$$\dot{E}^i_F = \dot{E}^i_P + \dot{E}^i_D + \dot{E}^i_L \quad \textbf{(15)}$$

in which $\dot{E}_P$ an d $\dot{E}_F$ are the rates of generated product and supplied fuel of element i, respectively. On the other hand, $\dot{E}_L$ and $\dot{E}_D$ are the rates of exergy loss and exergy destruction of the element i, respectively.

The exergetic efficiency of element i (η^i_{Ex}) and total exergetic efficiency of the system ((η^{total}_{Ex})) can be expressed as:

$$\eta^i_{Ex} = \dot{E}^i_P/\dot{E}^i_F \quad \textbf{(16)}$$

$$\eta^{total}_{Ex} = \dot{E}^{total}_P/\dot{E}^{total}_F \quad \textbf{(17)}$$

4. WORKING FLUID SELECTION

Selection of an appropriate working fluid can be the number one issue in improving of the COP and exergetic efficiencies. The appropriate working fluid can be chosen based upon two important factors: having the highest efficiency and being eco-friendly working fluid. So, we must reach to a trade-off between these two factors. One of the most important alternative working fluids is HCs and HFCs due to their zero ODP^+ and low GWP^+_+ characteristics. But, on the other hand, HC refrigerants have flammability issues, which restrict their usages in the ORC. However, the reduction in flammability can be yielded by choosing an appropriate mechanism.

In this paper we have suggested 9 different appropriate working fluids (i.e., R717, isobutane, isobutene, butene, cis-2-butene, R134a, R152a, R236fa, R290) for the proposed cycle, theoretically. Among all presented working fluids, isobutane and R717 are the best choice from the environmental impact's and thermal efficiency's point of views, respectively [3,22,23].

5. RESULTS AND DESCUSSION

A. Model Validation

To show the accuracy of the mathematical manipulation of the equations, an appropriate code has been written in Engineering Equation Solver (EES) to compare the results with reference [3]. The results of all comparison have been listed in Table 1.

This section also presents the obtained results from energetic and exergetic analyses of TEERC. But presenting all obtained results for different working fluids seems to be useless and cumbersome. So, some of thermodynamic results for the best case of study (i.e., R717) have been presented. But before we proceed further, we have specified some required input flow parameters which will come in handy for both energetic and exergetic analyses (Table 2).

Table 1. Model validation between present work and experiment [3].

Parameter	Present work	Experiment	Relative error (%)
Generator heat $Q_g(KW)$	8.713	9	3.18
Cooling capacity $Q_e(KW)$	1.706	1.75	2.51
Condenser load $Q_c(KW)$	9.826	11.28	12.89
Pump power $W_{pu}(KW)$	0.01701	0.02	17.64
Mass entrainment ratio U	0.2422	0.24	0.91
Coefficient of performance (COP)	0.1954	0.19	2.84

Table 2. Some required input parameters for thermodynamic modeling.

Parameter	Value
Generator temperature $T_g(K)$	330
Evaporator 1 superheated pressure $P_{es1}(MPa)$	0.08558
Evaporator 2 superheated pressure $P_{es2}(MPa)$	0.1458
Evaporator 3 superheated pressure $P_{es3}(MPa)$	0.2093
Evaporator 1 temperature $T_{e1}(K)$	283
Evaporator 2 temperature $T_{e2}(K)$	273
Evaporator 3 temperature $T_{e3}(K)$	260
Condenser temperature $T_c(K)$	302
Mass entrainment ratio U	0.34
Mass flow rate of steam $\dot{m}_{st}(kg.s^{-1})$	0.5
Cold room 1 temperature $T_{room1}(K)$	286
Cold room 2 temperature $T_{room2}(K)$	276
Cold room 3 temperature$T_{room3}(K)$	264

Table 3. Output data obtained from energetic analysis of TEERC.

Working Fluid	$Q_{e1}(KW)$	$Q_{e2}(KW)$	$Q_{e3}(KW)$	$Q_g(KW)$	$Q_c(KW)$	COP
R717	11.7	3.865	1.906	52.28	67.5	0.333
R152a	10.86	3.528	1.702	52.28	66.01	0.306
R134a	10.54	3.391	1.613	52.28	65.45	0.296
R290	10.55	3.393	1.613	52.28	65.5	0.295
Cis-2-butene	10.38	3.35	1.603	52.28	65.17	0.293
Butene	10.17	3.268	1.554	52.28	64.71	0.286
Isobutene	10.19	3.267	1.549	52.28	64.95	0.285
Isobutane	9.857	3.139	1.475	52.28	64.09	0.276
R236fa	9.627	3.04	1.409	52.28	63.65	0.268

Table 4. Calculated exergy properties for different components of the proposed TEERC using R717.

Working Fluid	$\dot{E}_F^{total}$(kW)	$\dot{E}_F^{total}$(kW)	$\dot{E}_D^{total}$(kW)	η_{ex}^{total}(%) R717
4.686	0.986	2.872	21.04	
R152a	4.686	0.8071	3.097	17.22
R134a	4.686	0.7269	3.191	15.51
R290	4.686	0.632	3.292	13.49
Cis-2-butene	4.686	0.8484	3.058	18.1
Butene	4.686	0.8017	3.131	17.11
Isobutene	4.686	0.5591	3.405	11.93
Isobutane	4.686	0.743	3.207	15.85
R236fa	4.686	0.7325	3.223	15.63

B. ENERGY RESULTS

Table 3 gives the obtained results of energy analysis for the proposed TEERC. The coefficient of performance of the system falls into the range of 0.268-0.333. The maximum COP is obtained for R717, while the minimum value of COP is obtained for R236fa. The ventilation capacity (Q_{e1}), cooling capacity (Q_{e2}), and freezing capacity (Q_{e3}), are also maximum for the R717. Using R717, ventilation capacity, cooling capacity, and freezing capacity are obtained 11.7 kW, 3.86 kW, and 1.90 kW, respectively.

C. EXERGY RESULTS

Table 4 gives some of the calculated key parameters of the exergy analysis, i.e., the total exergy rate of fuel, the total exergy rate of product, the total exergy rate of destruction, and exergy efficiency for the TEERC, using different working fluids. The overall exergy efficiency and exergy destruction for the proposed cycle fall into the range of (11.93-20.19) % and (2.925-3.405) kW, respectively. The maximum exergy efficiency and the minimum exergy efficiency of the cycle obtained when R717 and isobutene are used in the cycle, respectively. Therefore, selection of R717 is also appropriate for second law of thermodynamic.

D. PARAMETRIC STUDY

Fig. 2 has been plotted to show the effect of the generator temperature on some of the critical design parameters. Theses parameters are: COP, exergy efficiency, ventilation/input heat ratio, cooling/input heat ratio, and freezing/input heat ratio. As shown in this figure, the Cop and exergy efficiency are improved by increasing of the generator temperature. This is mainly due to the fact an increase in the generator temperature increases

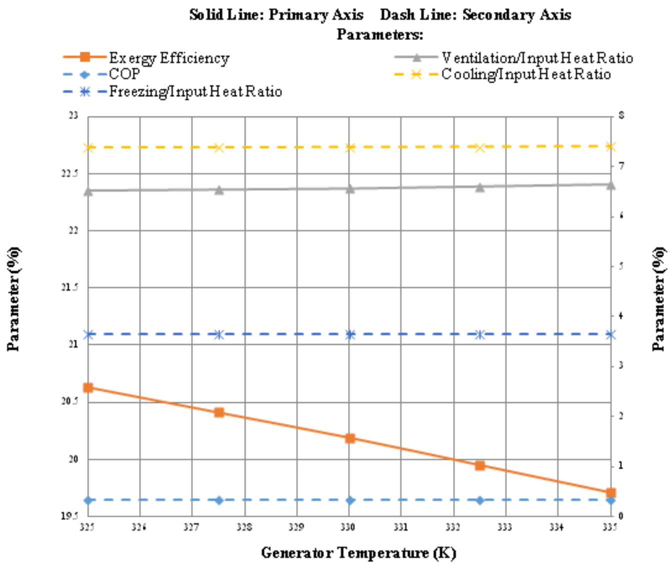

Fig. 2. Effect of generator temperature on the: COP, exergy efficiency, ventilation/input heat ratio, cooling/input heat ratio, and freezing/input heat ratio, using R717.

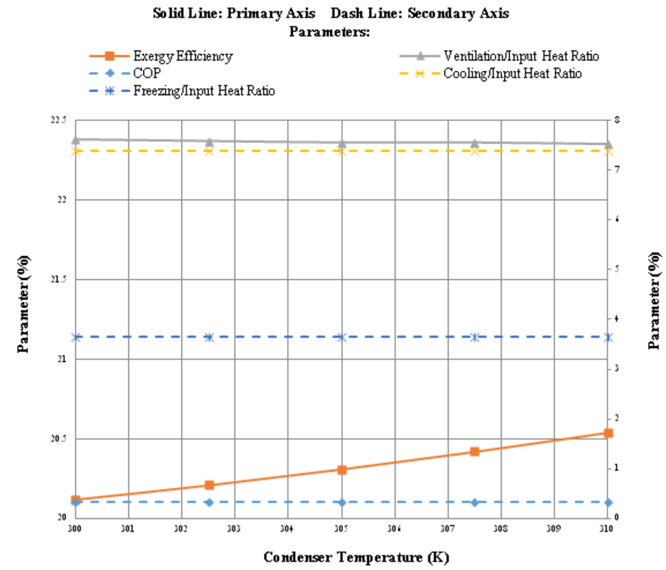

Fig. 3. Effect of condenser temperature on the: COP, exergy efficiency, ventilation/input heat ratio, cooling/input heat ratio, and freezing/input heat ratio, using R717.

the mass flow rate of the ejector, which gives a rise in the evaporators' mass flow rate. Hence, the ventilation, cooling, and freezing capacities are augmented with respect to increase of the generator temperature.

Fig.3 has been plotted to show the effect of the condenser temperature on the different important prescribed parameters. As shown, the cooling/input heat, ventilation/input heat, and freezing/input heat ratios are decreased by increasing of the condenser temperature, slightly. This is mainly due to the fact that increasing of the condenser temperature increases the rejecting heat of condensation through the cycle operation. This will result in a reduction in the COP of the cycle. But, on the other hand, the maximum theoretical work of the generator is increased as the condenser temperature increases, resulting in a rise in the overall exergy efficiency.

Fig.4 shows the effect of the evaporator 1 temperature on the different proposed important parameters. As illustrated, the cooling/input heat and freezing/input ratios are constant with respect to the evaporator 1 temperature variation, since the temperature of evaporator 1 has no effect on that of evaporators 2 and 3. On the other hand, increasing of the evaporator 1 temperature increases the ventilation capacity, since the enthalpy difference throughout the evaporator 1. As a result of this variation, the COP is increased. Meanwhile, the exergy of ventilation is increased as evaporator 1 temperature increases, and thus the exergy efficiency of the cycle is increased.

Fig.5 shows the effect of the evaporator 2 temperature on the different proposed important parameters. As illustrated, the freezing/input heat and ventilation/input heat ratios are constant with respect to the evaporator 2 temperature variation, since the evaporator 2 temperature variations have no effect on the evaporators 1 and 3. Increasing of the evaporator 2 temperature increases the cooling capacity, and hence the COP of the system is increased since the input heat capacity is constant under this variation. Moreover, the exergy efficiency of the system increases with a rise in the evaporator 2 temperature. This is because, exergy produced for cooling through the evaporator

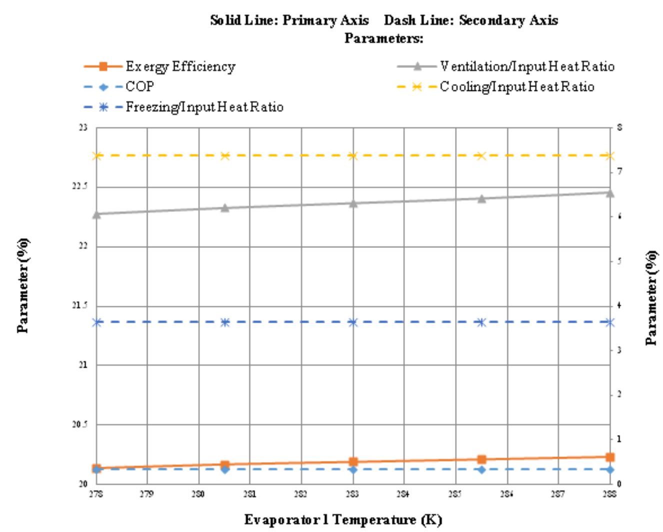

Fig. 4. Effect of evaporator 1 temperature on the: COP, exergy efficiency, ventilation/input heat ratio, cooling/input heat ratio, and freezing/input heat ratio, using R717.

2 is increased, resulting in increasing of the overall exergy of products.

Finally, to investigate the effect of the evaporator 3 temperature on the aforementioned performance parameters, Fig. 6 is depicted. As the figure indicates, it can be said that the cooling/input heat and ventilation/input heat ratios are constant with respect to the evaporator 3 temperature variation, since no variation in evaporators 1 and 2 temperatures are observed. Therefore, among all different produced refrigeration capacities, only freezing capacity is increased as the evaporator 3 temperature is increased, as expected earlier. This will result in an increase in the COP of the system. In addition, increasing of the evaporator 3 temperature increases the exergy of freezing, so slightly. As a result, the exergy efficiency of the cycle is in-

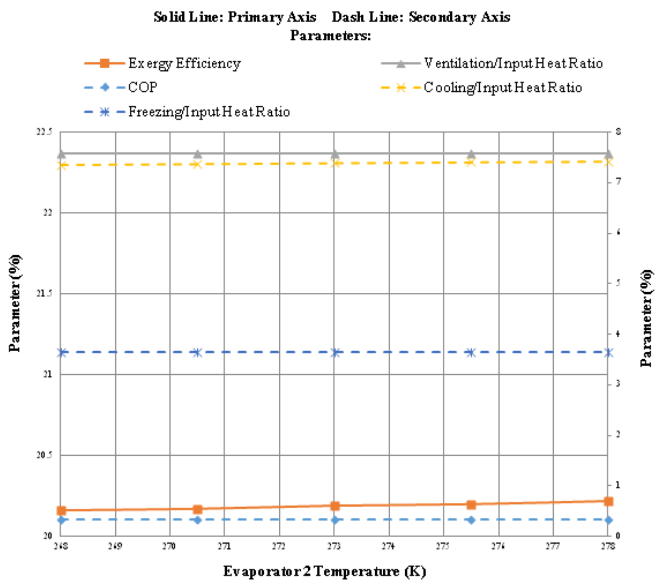

Fig. 5. Effect of evaporator 2 temperature on the: COP, exergy efficiency, ventilation/input heat ratio, cooling/input heat ratio, and freezing/input heat ratio, using R717.

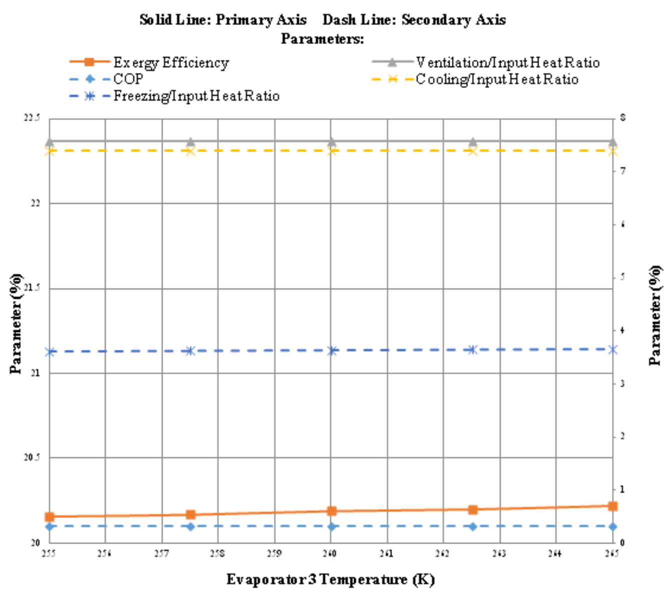

Fig. 6. Effect of evaporator 3 temperature on the: COP, exergy efficiency, ventilation/input heat ratio, cooling/input heat ratio, and freezing/input heat ratio, using R717.

creased.

Fig. 7 has been plotted to show the effect of the ejector mass entrainment ratio on the COP, exergy efficiency, ventilation/input heat ratio, cooling/input heat ratio, and freezing/input heat ratio. As shown in this figure, the Cop is increased by increasing of the ejector mass entrainment ratio, since three different refrigeration capacities (i.e., ventilation, cooling, and freezing capacities) are augmented with an increase of the mass entrainment ratio. The reason behalf of this phenomenon is that an increase in the ejector mass entrainment ratio increases the mass flow rate of the secondary flow which is proportional with

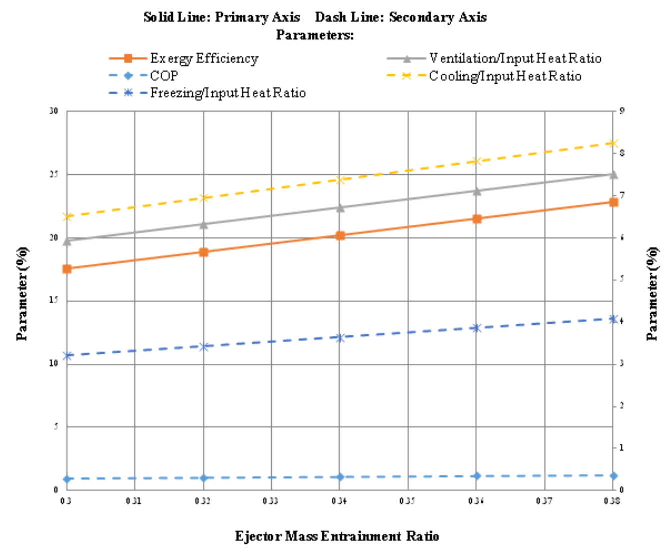

Fig. 7. Effect of ejector mass entrainment ratio on the: COP, exergy efficiency, ventilation/input heat ratio, cooling/input heat ratio, and freezing/input heat ratio, using R717.

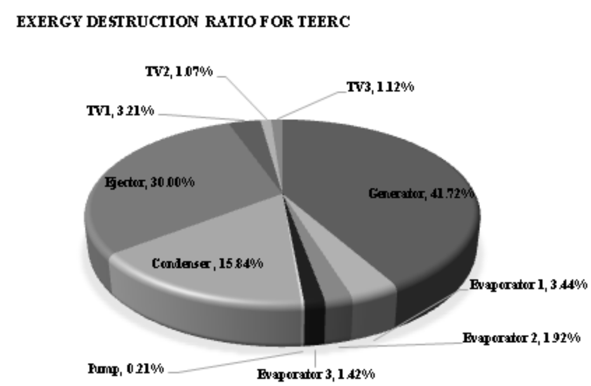

Fig. 8. Contribution of each component on the irreversibility of the proposed TEERC, using R717.

each aforementioned capacity. In addition, the exergy efficiency is increased by increasing of the ejector mass entrainment ratio, since ventilation, cooling, and freezing exergies are increases.

Fig. 8 shows the accountability of each component for the exergy loss of the whole cycle. This figure, like the previous figures are plotted for the best case of the study, R717. As illustrated, the generator accounts for the main source of the irreversibility which is followed by the ejector and condenser, respectively. So, implementing appropriate methods to reduce of the exergy destruction of these three components is the number one issue for cycle design purposes. One can design a suitable heat exchanger for the generator or condenser using some appropriate techniques.

6. CONCLUSION

A theoretical analysis of triple-evaporator ejector refrigeration cycle (TEERC) for triple-production of cooling output, freezing output, and ventilation output was proposed. Nine appropriate working fluids (i.e., R717, R152a, R134a, R290, cis-2-butene, butane, isobutene, isobutane, R236fa) were presented for the proposed cycle based on the working fluid characteristics, cycle efficiency, and environmental consideration. Energetic and exergetic analyses of the proposed cycle were performed in order to determine the main source of the irreversibility of the whole cycle. It was found that the generator has the main source of irreversibility which is followed by the ejector and condenser, respectively. This thorough investigation has been resulted in the following multiple conclusions:

• The maximum and minimum coefficient of performance (COP) are obtained for R717 and R236fa by the values of 0.333 and 0.268.

• The maximum and minimum exergy efficiencies are calculated for R717 and isobutene by the values of 21.43% and 12/51 %, respectively.

• The Cop and exergy efficiency are increased by increasing of the ejector mass entrainment ratio.

• The Cop and exergy efficiency are improved by increasing of the generator temperature.

• Increasing of the condenser temperature results in a reduction in the COP of the cycle and an increase in the overall exergy efficiency.

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