PARAMETRIC ANALYSIS OF AN EJECTOR REFRIGERATION CYCLE ACTIVATED BY RENEWABLE ENERGY

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ABSTRACT

The refrigeration and cooling systems activated by waste heat and renewable energy sources can achieve high energy efficiency. In this paper, a performance investigation and working fluid choice for an ejector refrigeration cycle (ERC) activated by renewable energy source are conducted. In the ejector mathematical model, the mixing process through the ejector constant-area section is supposed to be constant-pressure process. The performance of some selected hydrocarbons, hydrofluoroolefins, hydrofluoroethers, and hydrofluorocarbons refrigerants in the ERC is examined and compared. The behaviour of the ejector is represented by the ejector area ratio (EAR) and entrainment ratio (ω), whereas the cycle performance is represented by the coefficient of performance (COP). The impacts of some selected operating parameters as the condenser, evaporator, and generator temperatures on the ejector behaviour and cycle performances are also examined. The results show that the best ejector behaviour and cycle performance are achieved with the highest critical temperature dry refrigerant, i.e. cyclopentane under all examined operating conditions. From energy efficiency and environmental issues viewpoints, cyclopentane can be considered the most appropriate refrigerant amongst all examined working fluids. However, additional precautions should be taken against its flammability. The highest COP value, the corresponding ω , and the needed EAR using cyclopentane are 0.81, 1.03, and 17.3, respectively with a condenser temperature of 25 °C and the typical values for the remaining operating parameters.

KEYWORDS: *Ejector Refrigeration Cycle & Working Fluids & Entrainment Ratio & Renewable Energy*

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1. INTRODUCTION

Recently, world energy consumption is continuously increasing. Due to environmental problems and fossil fuel shortages, it is important to use waste heat and renewable energy. Power and refrigeration systems, which can be activated by waste heat or renewable energy sources, like solar or geothermal energy and waste heat, have been examined and developed. Waste heat is available in a broad range of temperatures based on the industrial processes. The thermally-activated refrigeration systems driven by renewable heat sources have gained extensive interest. The required power to drive these systems is generally heat and a small amount of mechanical energy, required to flow the refrigerant. These systems can be considered good alternatives for traditional vapor compression systems due to their slight environmental impacts and energy consumption [1, 2]. The ejector refrigeration cycle (ERC) is a promising cycle for converting thermal energy to refrigeration. The ERC has numerous pros, for instance, simple structure, low capital cost, low running and maintenance cost, and long lifetime [3].

The ejector makes entrainment, mixing, and compression without restrictions with respect to refrigerant. Unfortunately, because of less coefficient of performance (COP) of the ERC, these cycles aren't spread in the refrigeration markets. To widen the ERC utilization, further development is needed to boost their performance [4]. Numerous experimental and theoretical studies were conducted to increase the comprehension of the ejector design and processes to increase the cycle performance. The efficiencies of the ejector components and operating parameters have a large influence on the ejector behaviour, and different refrigerants perform differently in the ERC [5]. The dependence of on the ejector behaviour and the cycle performance on the operating parameters and ejector geometry was analytically tested by computational fluid dynamics models, thermodynamic ejector models, besides the experimental studies [6- 8]. Many studies were performed for comparing the performance of working fluids in ERCs [9, 10]. Also, refrigerant mixtures were used for ERC by Yan et al. [11].

Based on the literature survey, there's no settlement which refrigerant has the best performance in the ERC. Therefore, it's clear that further studies are still necessary regarding choice of the most proper refrigerant for ERC. In the present study, the performance of 10 refrigerants in the ERC is evaluated and compared theoretically. The study highlights on the choice of the most suitable refrigerant for ERC driven by renewable energy source with a temperature of nearly 115°C and down to 65° C. Ten common and new pure hydrofluorocarbons (HFCs), hydrocarbons (HCs), hydrofluoroolefins (HFOs), and hydrofluoroethers (HFEs) are used as refrigerants for ERC. The inspected substances are RE170, R245ca, R1234ze (E), R1234yf, RE245cb2, isohexane (R602a), cyclopentane, isopentane (R601a), neopentane (R601b), and diethyl ether (R610). The refrigerant accomplishes the top most entrainment ratio (ω) and COP will be recommended. The effects of some selected operating parameters on the ejector behaviour and ERC performance are studied and discussed as well. The one-dimensional mathematical model for ejector developed by Saleh [5] is used to specify the ejector behaviour and the cycle performance.

2. FEATURES OF THE EJECTOR REFRIGERATION CYCLE

The ERC diagram and the cycle processes in the pressure-enthalpy diagram are demonstrated in Figure 1. The points from 0 to 5 are found within the ejector and will be appearing later in Figure 3. The ERC mainly contains generator, ejector, expansion valve, pump, condenser, and evaporator. During the process 8-9, heat is added ($\dot{Q}_g$) to the primary flow from renewable energy source. The primary flow leaves the generator as saturated vapor and enters the nozzle of the ejector. Then it entrains the low-pressure secondary flow from the evaporator. The two flows subjected to a mixing process inside the ejector and then pressure recapture. Then the mixed two streams pass through the condenser and reject heat to the surroundings ($\dot{Q}_c$) during process 6-7. The condensate distributed in two streams. The secondary flow is the first stream with a mass flow rate ($\dot{m}_s$). This stream flows to the evaporator through the expansion valve during the process 7-10. During the process 10-11 the secondary flow evaporates and therefore gives the refrigeration effect ($\dot{Q}_e$). The primary flow is the second stream with a mass flow rate ($\dot{m}_p$). For completing the cycle, the circulating pump pumped back the primary stream to the generator during the process 7-8.

Refrigerants are categorized as dry, isentropic, and wet based on the shape of their temperature against entropy diagram. In the case of wet refrigerant, slight droplets may be formed at the outlet of the ejector nozzle. This may extremely affect the ejector gas dynamic process and accordingly the ejector behaviour. This may be avoided by superheating the primary stream before arriving the nozzle to be sure that the nozzle expansion process happens in the superheated region. On the other side, in the case of isentropic and dry refrigerants there is no phase change through the

expansion process and the superheat is not necessary [9]. The shapes of T-s diagrams of the examined refrigerants are shown in Figure 2.

The refrigerant has a significant effect on the ejector behaviour and cycle performance. A proper refrigerant accomplishes best ejector behaviour and cycle performance and a minimum environmental impact. During refrigerant choice for the ERC, the subsequent matters should be considered: (i) thermodynamic concern: energy efficiency, (ii) environmental concern: ozone depletion potential (ODP), global warming potential (GWP), and atmospheric lifetime (ALT), (iii) safety aspects: auto ignition, toxicity, and flammability and (iv) availability and cost.

Figure 1: ERC and its p-h Diagram

Figure 2: T-s Diagrams of the Examined Refrigerants

Many studies used chlorofluorocarbons (CFCs) and hydrochlorofluorocarbons (HCFCs) as refrigerants for ERC [2, 7]. Such refrigerants were forbidden due to their environmental effects. Thus, HFCs were suggested for ERC because of their zero ODP [6, 13, 14]. HFCs are now being regulated because they have large GWP values. Therefore, still there is screening for alternative refrigerants, which may have a higher ERC performance and lower environmental impacts. Hydrocarbons (HCs) may be a good alternative, which have too low GWP values and excellent thermophysical properties. The HCs have many advantages, for instance, environmentally friendly, non-toxic, and chemically stable. Conversely, HCs are highly flammable, which is the only challenge against HCs. But, with proper safety defenses, the flammability will not constitute a challenge in utilizing HCs. Consequently, several HCs as R600, R600a were used as refrigerants in ERC [6, 15].

In the present study, the performance of a thermally-driven ERC utilizing ten HFCs, HCs, hydrofluoroethers (HFEs), and hydrofluoroolefins (HFOs) refrigerants is examined. The examined refrigerants are RE170, R245ca, R1234ze(E), R1234yf, RE245cb2, R602a, cyclopentane, R601a, R601b, and R610. The main properties of these refrigerants are listed in Table 1 [16].

Table 1: Properties of the Inspected Refrigerants for ERC

Refrigerant	Chemical Formula	Physical Properties					Environmental Data			Safety Data	
		M	NBP	T _{crit}	P _{crit}	$v_{crit} \times 10^3$	ALT	ODP	GWP	LFL	Safety
		g/mol	°C	°C	MPa	m ³ /kg	Year		100 yr	%	Group
R1234yf	CF ₃ CF=CH ₂	114.0	-29.5	94.7	3.38	0.002	0.029	0	<1	6.2	A2L
R1234ze(E)	CHF=CHCF ₃	114.0	-18.9	109.4	3.635	2.04	0.045	0	<1	7.6	A2L
RE170	(CH ₃) ₂ O	46.1	-24.8	127.2	5.337	3.65	0.015	0.0	1	3.4	A3
RE245cb2	CF ₃ CF ₂ -O-CH ₃	150.1	5.6	133.7	2.886	2.0	4.90	0	680	—	—
R601b	C(CH ₃) ₄	72.15	9.5	160.59	3.196	4.24		0.0	~20	1.4	—
R245ca	CHF ₂ -CF ₂ -CH ₂ F	134.1	25.3	174.4	3.94	1.90	6.5	0	726	7.1	—
R601a	CH ₃ CH ₂ CH(CH ₃) ₂	72.15	27.8	187.2	3.378	4.24	12	0	4	1.32	A3
R610	C ₄ H ₁₀ O	74.12	34.454	193.55	3.644	3.79	12	0.0	4	1.9	—
R602a	(CH ₃) ₂ CH(CH ₂) ₂ CH ₃	86.2	60.2	224.6	3.040	4.27	---	0.0	~20	1.2	A3
cyclopentane	C ₅ H ₁₀	70.1	49.3	238.6	4.571	3.73	0.007	0.0	<0.1	1.1	A3

3. PROCESSES AND MATHEMATICAL MODEL OF THE EJECTOR

The ejector sections and the change of velocity and pressure with the position in the ejector are shown in Figure 3. The description of the ejector operating principles as follows: (1) the primary flow increases its velocity and expands in the nozzle of the ejector, (2) the nozzle exhausts the primary flow with supersonic speed and low mixing pressure (P_m), (3) due to the difference between primary and secondary flows pressures, the secondary stream enters the suction chamber and its velocity grows, (4) because of the velocity difference between the two flows, they exit from suction chamber without mixing. The two streams begin to mix in the mixing chamber where the secondary stream velocity continues to increase while the primary stream velocity continues to decrease due to the mixing process till they are completely mixed in section 3, (5) before delivery from the mixing chamber, a compression shock is encouraged which changes the velocity from supersonic to subsonic and rises the pressure, (6) in the diffuser section the pressure continues to increase while the velocity reduces and finally the refrigerant delivers to the condenser.

Figure 3: Ejector Diagram and Velocity and Pressure Change as a Function of Position in the Ejector

The ejector design classified into two types based on the location of the nozzle outlet position. If the nozzle outlet locates in the suction chamber, the mixing process of the secondary and primary streams happens with a fixed pressure in the suction chamber. In this case, the ejector is a constant-pressure mixing ejector type. On the other hand, if the nozzle outlet locates in the constant-area part, the mixing of the two flows happens in the constant-area section. This type is called a constant-area mixing ejector. Both ejector types have been widely studied. The constant-pressure ejector type is widely used because it has a better behaviour than the constant-area type [17, 18].

In the present study, the mathematical model established by Saleh [5] is utilized to estimate the ejector behaviour and the performance of the ERC. The mathematical model evaluates both the ejector behaviour and the cycle performance and calculates the required ejector area ratio (EAR) as well. The ejector behaviour is defined by EAR and ω and the performance of the ERC is defined by the cycle COP. The mixing process through the ejector constant-area section is supposed to be constant-pressure process in the ejector mathematical model. The evaporator pressure (P_e) is greater than P_m in the real ejector process to permit the secondary stream to entrain. To make the ejector analysis simple, the P_e is assumed equal to P_m [12]. Nevertheless, in the mathematical model in this study P_m is assumed less than P_e and takes into account the compression shock in the mixing chamber. Additional details about the mathematical model principles can be found in Saleh [5]. The primary equations of the model are presented in the next subsections.

3.1 Primary Stream

The primary stream velocity at the exit of the nozzle (V_2) is given by:

$$V_2 = \{2\eta_n(h_9 - h_{2i})\}^{(1/2)} \quad (1)$$

where η_n is the nozzle efficiency, h_9 is the enthalpy at the nozzle inlet, and h_{2i} is the ideal enthalpy at the nozzle outlet.

$$\eta_n = \frac{h_9 - h_2}{h_9 - h_{2i}} \quad (2)$$

where h_2 is the actual enthalpy at the nozzle outlet.

The expansion of primary stream in the nozzle is expressed using Mach number at point 2 (M_{p2}) as El-Dessouky et al. [19]

$$M_{p2} = \left\{ \frac{2\eta_n}{\gamma-1} \left[\left(\frac{P_g}{P_m} \right)^{\left(\frac{\gamma-1}{\gamma} \right)} - 1 \right] \right\}^{0.5} \quad (3)$$

where P_g is the generator pressure, and γ is the specific heat ratio.

3.2 Secondary Stream

In many previous studies, the secondary stream velocity at the outlet of the nozzle (V_1) was ignored due to the supposition of P_m equals P_e [20]. But, V_1 cannot be ignored due to the difference between P_m and P_e values. So, V_1 is given by: can be expressed as follows.

$$V_1 = \sqrt{2(h_{11} - h_1)} \quad (4)$$

where h_1 and h_{11} are the secondary stream enthalpies at the nozzle and the evaporator exits.

The secondary stream Mach number (M_{s2}) at the nozzle outlet can be written as:

$$M_{s2} = \left\{ \frac{2}{\gamma-1} \left[\left(\frac{P_e}{P_m} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right] \right\}^{0.5} \quad (5)$$

3.3 Mixing Process

Both secondary and primary flows are completely mixed in the mixing chamber before formation the shock at section 4. The momentum equation governed the ideal mixing process is given by:

$$\dot{m}V_4 = \dot{m}_sV_1 + \dot{m}_pV_2 \quad (6)$$

where $\dot{m}$, $\dot{m}_s$, and $\dot{m}_p$ are the total, secondary, and primary streams mass flow rates, and V_4 , V_1 , and V_2 are the velocities in the particular positions in the ejector as exhibited in Figure 3.

The entrainment ratio, ω is given by:

$$\omega = \frac{\dot{m}_s}{\dot{m}_p} \quad (7)$$

The ideal velocity (V_{4i}) is given by:

$$V_{4i} = \frac{\omega V_1 + V_2}{\omega + 1} \quad (8)$$

The mixing efficiency (η_m) is given by:

$$\eta_m = \frac{V_4^2}{V_{4i}^2} \quad (9)$$

The actual velocity V_4 is given by

$$V_4 = (\eta_m)^{0.5} * \frac{\omega V_1 + V_2}{\omega + 1} \quad (10)$$

The critical Mach number at position 4 (M_4^*) can be found using the critical Mach number of the secondary stream (M_{s2}^*) and the primary stream (M_{p2}^*) at point 2 and is given by:

$$M_4^* = (\eta_m)^{0.5} * \frac{\{M_{p2}^* + \omega M_{s2}^* (T_e/T_g)^{1/2}\}}{\sqrt{(\omega+1)(1+\omega T_e/T_g)}} \quad (11)$$

where T_g and T_e are the generator and the evaporator temperatures.

The relationship between the Mach number at any position in the ejector and the critical Mach number is given by:

$$M = \left\{ \frac{2M^{*2}}{(1+\gamma) - M^{*2}(\gamma-1)} \right\}^{0.5} \quad (12)$$

The Mach number (M_4) of the mixed flow at section 4 is given by:

$$M_4 = \sqrt{\frac{2M_4^{*2}}{(1+\gamma) - M_4^{*2}(\gamma-1)}} \quad (13)$$

3.4 Mixed Flow Across the Diffuser

The diffuser efficiency, η_d , is given by

$$\eta_d = \frac{h_{6i} - h_4}{h_6 - h_4} \quad (14)$$

where h_6 and h_{6i} are the ejector exit actual and isentropic enthalpies.

The ω can be re-expressed as

$$\omega = \frac{\sqrt{2\eta_n(h_9 - h_{2i})} - \sqrt{\{2(h_{6i} - h_4)/(\eta_m\eta_d)\}}}{\sqrt{\{2(h_{6i} - h_4)/(\eta_m\eta_d)\}} - \sqrt{2(h_{11} - h_1)}} \quad (15)$$

The Mach number downstream and upstream of the shock is given by

$$M_5 = \sqrt{\frac{\{M_4^2 + 2/(\gamma-1)\}}{\{2\gamma M_4^2/(\gamma-1)\} - 1}} \quad (16)$$

The pressure ratio (P_5/P_4) through the shock is given by

$$\frac{P_5}{P_4} = \frac{\gamma M_4^2 + 1}{\gamma M_5^2 + 1} \quad (17)$$

The pressure recovery process of the mixed flow is given by

$$\frac{P'_6}{P_5} = \left[\frac{(\gamma-1)}{2} M_5^2 + 1 \right]^{\gamma/(\gamma-1)} \quad (18)$$

The calculated pressure P'_6 at the ejector outlet has to compare with the condenser pressure (P_6). Thus, to get ω and P_6 , two iterative processes have to do in the calculation program.

3.5 Ejector Geometry and ERC Performance

The EAR is the ratio between the mixing chamber area (A_3) and the area of the nozzle throat (A_1). The EAR is given by the following equation [12]

$$EAR = \frac{A_3}{A_1} = \frac{\left\{ P_g \{2/(\gamma+1)\}^{1/(\gamma-1)} \sqrt{\{(\omega+1)(1+\omega T_e/T_g)(1-2/(\gamma+1))\}} \right\}}{\left\{ P_6 (P_5/P_6)^{1/\gamma} \sqrt{1-(P_5/P_6)^{(\gamma-1)/\gamma}} \right\}} \quad (19)$$

The added heat rate to the refrigerant in the generator, $\dot{Q}_g$, and the refrigeration capacity, $\dot{Q}_e$, can be calculated as by the following equations:

$$\dot{Q}_g = \dot{m}_p(h_9 - h_8) \quad (20)$$

$$\dot{Q}_e = \dot{m}_s(h_{11} - h_{10}) = \omega \dot{m}_p(h_{11} - h_{10}) \quad (21)$$

The COP of the ERC is the ratio between $\dot{Q}_e$ and both $\dot{Q}_g$ and the required pump power ($\dot{W}_{\text{pump}}$). The $\dot{W}_{\text{pump}}$ is less than 1% of $\dot{Q}_g$, so it can be neglected [21]. The COP of the ERC is given by

$$COP_{\text{ERC}} = \frac{\dot{Q}_e}{\dot{W}_{\text{pump}} + \dot{Q}_g} = \omega \frac{h_{11} - h_{10}}{h_9 - h_8} \quad (22)$$

A computer program is written to evaluate the ejector behaviour and the ERC performance utilizing different

refrigerants under various operating conditions. The computer program calculates the COP and the corresponding ω in addition to the necessary EAR using operating parameters, and refrigerant properties. The thermodynamic properties of the examined refrigerants are found using REFPROP 9.1 [22].

The typical ERC operating conditions and their ranges are as follows: the typical, condenser temperature (T_c), T_e , and T_g are 35 °C, 10 °C, and 90 °C, respectively. The inspected ranges of T_g , T_c , and T_e are (60 °C to 110 °C), (25 °C to 45 °C), and (-5 °C to 15 °C), respectively. The η_n , η_m , and η_d , are fixed at 0.9.

4. RESULTS AND DISCUSSIONS

The ejector behaviour and the ERC performance utilizing different refrigerants are investigated. The inspected refrigerants are RE170, R245ca, R1234ze(E), R1234yf, RE245cb2, R602a, cyclopentane, R601a, R601b, and R610. A one-dimensional model for the ERC is used to assess the ejector behaviour and ERC performance. Among all the proposed refrigerants, R245ca and RE245cb2 have very large GWP as shown in Table 1, however they have zero ODP. The remaining eight refrigerants have zero ODP and very low GWP, however they are flammable or mild-flammable. Consequently, additional precautions should be taken against the safety concerns. Hence, from the safety and environmental concerns viewpoints, there is no refrigerant meets all requirements. Every refrigerant has its own advantages and disadvantages. Hence, the energy efficiency criterion will be considered to select the most suitable refrigerant for the ERC.

Table 2 shows the results of the ejector behaviour and the ERC performance with the typical working conditions using all examined refrigerants. The types of T-s diagram of the inspected refrigerants are presented in Table 2 as well. The results demonstrate cyclopentane with the highest critical temperature (T_{crit}) attains the uppermost values of ω and COP. These values are 0.380 and 0.511, respectively and the corresponding EAR is 8.96. Reversely, R1234yf with the bottommost T_{crit} attains the bottommost COP and ω values. These values are 0.122 and 0.145, respectively and the corresponding EAR is 4.87. Hence, from the energy efficiency criterion standpoint, cyclopentane is the most appropriate refrigerant for ERC to recover thermal energy. The use cyclopentane as a refrigerant for ERC requires supplementary considerations regarding concerning its flammability.

Table 2: The Performance of Typical ERC

Refrigerant	Refrigerant Type	COP	ω	EAR
R1234yf	dry	0.122	0.145	4.87
R1234ze(E)	dry	0.239	0.302	5.77
RE170	wet	0.327	0.281	6.10
RE245cb2	dry	0.295	0.430	7.06
R601b	dry	0.447	0.334	7.40
R245ca	dry	0.347	0.470	8.53
R601a	dry	0.342	0.482	7.65
R610	dry	0.493	0.371	8.70
R602a	dry	0.485	0.360	8.73
Cyclopentane	dry	0.511	0.380	8.96

The ejector behaviour and the ERC performance mainly depend on ejector geometry, working conditions, and working fluid. The impacts of some selected working conditions as T_c , T_g , and T_e on the ejector behaviour and cycle performance are discussed in the succeeding sections

4.1 Impact of Generator Temperature

The variations of the ejector behaviour (ω and EAR) and the ERC performance (COP) with T_g are displayed in Figure 4. The ω improves with the enhance of T_g for all investigated refrigerants as displayed in Figure 4a. This is due to the fact that at constant T_c and T_e , the increase of T_g leads to growth the generator pressure. This results in enhance the entrainment effect of the ejector. Hence, further secondary flow could be entrained, and therefore increases ω . Due to the increase of ω , the EAR should be increased as shown in Figure 4b in order to provide a suitable flow area. Among all the examined refrigerants, cyclopentane achieves the maximum ω , whereas R1234yf achieves the minimum values for all inspected T_g . It should be noticed that R1234ze(E) and R1234yf have been inspected in a small domain of T_g because they have low critical temperatures as listed in Table 1. In the case of cyclopentane, the increase of T_g from 60°C to 110°C leads to increase ω nearly four times and accordingly the needed EAR increases almost 4.78 times.

The alterations of COP as a function of T_g for all examined refrigerants are shown in Figure 4c. Generally for all refrigerants as T_g increases, the entrained secondary flow increases. This is due to the better suction effect and the higher entry energy which leads to improve COP. Regardless of the refrigerant type, the tendency of COP variations are similar to that of ω . The equation 22 interprets this where the COP value is directly proportional to ω . Among all examined refrigerants, cyclopentane achieves the highest COP values in the inspected T_g domain, whereas R1234yf attains the lowest COP values. From energy efficiency standpoint, cyclopentane is the most suitable refrigerant among all examined refrigerants for ERC for recovering low-temperature renewable energy. Irrespective of refrigerant type, the maximum COP is less than 0.5 in the inspected T_g domain from 60 °C to 110 °C, as shown in Figure 4c. To some extent, the small COP values limit the spread of ERC in the refrigeration and air conditioning market.

Figure 4: The Impact of T_g on ω (a), EAR (b), and COP (c)

4.2. Impact of Condenser Temperature

The variations of ejector behaviour and ERC performance for all examined refrigerants as a function of T_c are shown in Figure 5. The fixed-geometry ejector has a critical back or discharge pressure and the entrainment ratio is controlled by this pressure [7]. The performance of the ejector can be classified into three operating modes based on the discharge pressure: (a) subcritical mode in which the discharge pressure is larger than critical discharge pressure; (b) critical mode in which the discharge pressure is lower than the critical pressure and it is described by a constant ω which keep the COP and refrigeration effect constant; and (c) malfunction mode in which the discharge pressure is larger than the ejector operational mode limiting condition. In this mode, the entrained flow reverses back to the evaporator and the ejector wholly misses its purpose. Such mode doesn't happen with the variable-geometric ejector (VGE). It can be detected from Figure 5a that there are no turning points in ω which always decreases as T_c grows. Such working mode is defensible by varying EAR. The growth of T_c results to a continuous steady decline in ω value instead of a sharp drop. At high T_c the ω value is very small and the ejector will not stop working as shown in Figure 5a. Hence, the VGE is a good solution for widening the domain of T_c , in this case the EAR should be reduced with the decrease of ω as displayed in Figure 5b. The mixing pressure grows with a large degree with the growth of T_c . This leads to reduce the difference between the mixing and secondary stream pressures at a constant evaporator pressure. Therefore, the entrained secondary flow decreases. This interprets the decline of ω with the growth of T_c . Also the increase of the discharge pressure forces the shock to move in the direction of the nozzle exit, which leads to decrease the entrainment effect [12].

The variations of COP with T_c are shown in Figure 5c. The COP and ω have a similar tendencies with the T_c alteration as can be detected from Figs. 5a and 5c. The COP grows with the decrease of T_c irrespective of the refrigerant. In conclusion, for better ejector behaviour and cycle performance it is preferable to operate the ERC with low T_c .

Among all the inspected refrigerants, cyclopentane has the largest ω and COP values and is the most sensitive refrigerant to the T_c variations as can be detected from Figure 5. In the case of cyclopentane, the increase of T_c from 25 °C to 45 °C leads to decrease COP, ω and EAR by nearly 4.5, 4, and 3.2 times, respectively. Conversely, R1234yf has the lowest COP and ω values and therefore the minimum EAR in the inspected domain of T_c . R1234yf is the lowest sensitive refrigerant to the T_c variations. Using R1234yf, an increase of one degree Celsius in T_c results in nearly 1.7 %, 1.9 %, and 24.5 % reduction in COP, ω and EAR, respectively in the inspected domain of T_c .

Figure 5: Impact of T_c on ω (a), EAR (b), and COP (c)

4.2 Impact of Evaporator Temperature

The variations of ejector behaviour and ERC performance with T_e for all examined refrigerants are shown in Figure 6. The ejector behaviour and the ERC performance against T_e variation tendencies are a large extent similar to those against T_g . As can be detected from Figs. 6a, 6b and 6c, the COP, ω and EAR increase with the growth of T_e for all examined refrigerants in the inspected T_e domain. At constant T_c and T_g , the secondary flow pressure grows as T_e increases, which contributes to boost entrainment effect. This leads to boost both ω and COP as can be detected from Figs. 6a and 6b. So, the EAR should be increased as shown in Figure 6b. The EAR increases nearly linear with the increase of T_e for all examined refrigerants. The EAR is not sensitive to the alteration of T_e compared to the alteration of both T_c and T_g . This is because the primary stream flow, which is unchanged in this case due to fixation of T_g , mainly contributes in the production of an aerodynamically convergent nozzle where the secondary stream achieves a peak value and the stream chokes whereas the impact of the T_e on the aerodynamically choked area is very small. Nevertheless, the growth of secondary stream pressure forces the active location of the convergent nozzle to transfer downstream, which permit for additional secondary stream to flow [12].

As can be detected from Figure 6, among all inspected refrigerants, cyclopentane has the highest ω and COP values and has the highest sensitivity to deviations of T_e compared with other examined refrigerants. In the case of cyclopentane, a one degree Celsius rise in T_e leads to nearly 1.8 %, 2.2 %, and 17.8 % growth in COP, ω and EAR, respectively. On the other side, R1234yf has lowest COP and ω values and hence the lowest EAR in the studied T_e domain. R1234yf has the lowermost sensitivity to deviations of T_e in comparison with other examined refrigerants. In the case of R1234yf, the variation of T_e from -5 °C to 15 °C leads to increase COP, ω and EAR by nearly 2.5, 1.8, and 1.1 times, respectively.

The parametric investigation of the ERC which displayed in Figs. 4-6 show that the ejector behaviour and the ERC performance are more sensitive to alteration of T_c than either T_e or T_g regardless the refrigerant type. To maintain the ERC works at the best conditions within a given range of working parameters, the EAR must be changed to compensate the variation of ω due to variation of operating parameters, as shown in Figs. 4-6. Overall, the performance of fixed-geometry ejector is less than the VGE [12]. The VGE can be utilized to guarantee that the ERC operates at its best conditions.

Figure 6: Impact of T_e on ω (a), EAR (b), and COP (c)

5. CONCLUSIONS

The performance of an ejector refrigeration cycle activated by renewable energy was investigated. A one-dimensional model was used to calculate the ejector behaviour and the ejector refrigeration cycle performance. The performance of many different refrigerants, i.e. RE170, R245ca, R1234ze (E), R1234yf, RE245cb2, R602a, cyclopentane, R601a, R601b, and R610 in the cycle was inspected. The impacts of some selected operating parameters on the ejector behaviour and cycle performance were investigated as well.

The parametric investigation of the cycle shows that each parameter has similar impacts on the ejector behaviour and the cycle performance regardless the refrigerant. For more details, the growing of generator and evaporator temperatures in addition to the decrease of condenser temperature enhance the ejector behaviour and cycle performance. The impact of condenser temperature on the ejector behaviour and the cycle performance is more pronounced in comparison with the impacts of evaporator or generator temperatures regardless the refrigerant. Furthermore, the variations trends of entrainment ratios are similar to those of COP trends. To maintain optimum ejector behaviour and cycle performance under various operating conditions, the ejector area ratio has to increase with the entrainment ratio enhancement.

The highest ejector behaviour and cycle performance are achieved using cyclopentane under all inspected operating conditions. Accordingly, cyclopentane is the most suitable refrigerant for ejector refrigeration cycle from the energy efficiency standpoint. However, additional precautions should be taken against its safety concerns. The highest COP value, the corresponding ω , and the needed EAR using cyclopentane are 0.81, 1.03, and 17.3, respectively with condenser temperature of 25 °C and the typical values for the remaining operating parameters.

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