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Energy and exergy analysis of possible alternatives to R134a in a vapour compression refrigeration cycle of a water cooler unit

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Abstract—In this paper, the energy and exergy performance of a vapour-compression refrigeration cycle of a water dispenser unit has been analyzed theoretically using different refrigerants as possible alternative substitutes to R134a. The selected low ″Global Warming Potential (GWP) ″ refrigerants are: HydroFluoroOlefins (HFO) R1234yf and R1234ze(E); HydroCarbons (HC) R290 and R600a; and HydroFluoroCarbon (HFC) R152a. The process was evaluated based on the evaporator and condenser temperatures, which range between -10 and 5ºC and between 30 and 45ºC, respectively. The theoretical model based on the first and second laws of thermodynamics has been developed using the Matlab environment. The performances of the vapor compression cycle are discussed in terms of coefficient of performance, exergy destruction and exergy efficiency. The results show that the maximum COP achieved is 5.12 and 5.10 for R152a and R600a respectively, the highest total exergy destruction is about 82.82 W for R290, the highest exergy efficiency is about 55.12% for R600a.

Keywords: water dispenser, low GWP refrigerants, vapour compression, exergy analysis.

I. INTRODUCTION

For several decades, vapor compression refrigeration systems have been widely used for different applications across many fields. These systems operate most commonly with R134a, which is a HydroFluoroCarbon (HFC) offering high performance, but also having high environmental impact (high ozone depletion potential (ODP) and global warming potential (GWP)) [1]. Environmental concerns forced the international community to limit or phase out many hydrofluorocarbons (HFCs), such as R134a, R410A, and R404A [2,3]. According to this purpose, many investigations have been conducted to find alternative refrigerants with 0 ODP and low GWP $\leq$150 for various applications based on their environmental impact, thermophysical properties, and safety. De Paula et al. [4] proposed a steady-state model of a small capacity vapour compression refrigeration system working with R290, R600a, and R1234yf. They found that R290 has the highest energy, exergy, environmental and economic performance. Heredia-Aricapa et al. [5] assessed different HFC/HFO/HC mixtures to replace refrigerants with a GWP exceeding 1300. Among these mixtures, R430A exhibited the closest performance to R134a exceeding the COP values (3.5) for all working conditions. Longo et al. [6] designed an experimental setup to test the replacement of R134a by two environmentally benign fluids: R1234yf and R1234ze(E). R1234ze(E) outperformed with a high vapor quality, whereas R1234yf is less affected by the forced convective boiling, due to its lowest pressure drop. Deymi-Dashtebayaz et al. [7] studied six pairs of refrigerants with low GWP, and concluded that R41-R61 and R41-R1234ze are the optimal refrigerants pairs with the highest COP and exergy efficiency and the lowest total cost rate. Sulaiman et al. [8] analyzed theoretically three low GWP refrigerants R1233zd(E), R1336mzz(Z), and R601 to replace R245fa in a heat pump. Their thermodynamic analysis showed that the R1233zd(E) provided a preferable trade-off between high coefficient of performance (COP) and high volumetric heating capacity (VHC) values. Noted that there are few investigations concerning water fountains in the literature, However, conventional dispensers are based principally on a vapour-compression refrigeration system using mainly R134a as working fluid.

This study models the vapour compression refrigeration system of a water cooler unit. Based on energy and exergy metrics from the second law of thermodynamics, five refrigerants with low GWP, namely R1234yf, R1234ze(E), R600a, R290 and R152a are compared to find the best overall substitute to R134a. The thermodynamic cycle is described in detail in Section II. The energy and exergy metrics are outlined in Section III. Then, Section IV presents and discusses the results, followed by the most relevant conclusions and future works in Section V.

II. SYSTEM DESCRIPTION

A schematic representation of a water cooler unit working with vapour compression refrigeration cycle along with the corresponding Pressure-Enthalpy diagram is shown in Figure 1. The system consists of compressor, condenser, evaporator, and expansion valve. It is assumed that both exchangers (evaporator and condenser) are tubular heat exchangers exchanging heat with ambient air. A piston compressor is used, the rotation speed is fixed to about 2900 rpm with a displacement volume of 3.5 cm^3. The expansion device is represented by a capillary tube.

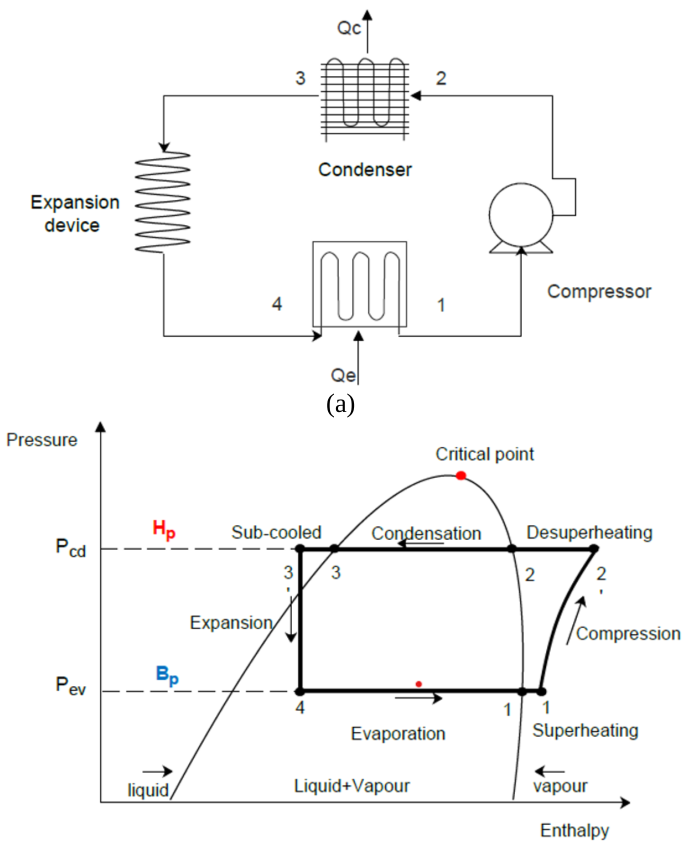

Figure 1: Schematic representation of a vapour compression refrigeration cycle and (b) corresponding pressure-enthalpy diagram.

The operating conditions considered for the thermodynamic model of the vapour compression refrigeration system are summarized in Table 1.

TABLE 1: OPERATING CONDITIONS.

Parameter	Value
Evaporator temperature (T_e)	-10 °C to 5 °C
Superheating degree	5 °C
Condenser temperature (T_c)	30 °C to 45 °C
Subcooling degree	3 °C
Reference temperature (T_0) and pressure (P_0)	24 °C, 101.3 kPa
Rotation speed of the compressor	2900 rpm
Mechanical and engine efficiencies	0.85

For the exergy and exergy analysis of the conventional compression refrigeration cycle, the refrigerants are selected as alternative solutions to R134a, with zero ODP and low value of GWP≤150. Table 2 presents the thermophysical properties of the selected refrigerants based on molecular weight (M), critical temperature and pressure (T_{crit} and P_{crit}), saturated pressure (P_{sat}), liquid and vapour density (ρ_l, ρ_v), latent heat of vaporization (L_{vap}), thermal conductivity and viscosity (λ, μ).

TABLE 2: DIFFERENT PROPERTIES OF THE STUDIED REFRIGERANTS [9].

Refrigerants	R134a	R1234yf	R1234zeE	R290	R600a	R152a
M [$g.mol^{-1}$]	102.03	114	114	44.10	58.12	66.05
T_{crit} [°C]	102	95	109.4	96.74	134.6	113.3
P_{crit} [bar]	41	34	36.3	42.5	36.3	45.1
P_{sat} [bar]	3.49	3.73	2.59	5.51	1.87	3.14
ρ_l [$kg.m^{-3}$]	1278.1	1160.4	1111.5	521.75	574.8	947.7
ρ_v [$kg.m^{-3}$]	17.1	20.7	40.6	11.9	5.01	9.89
L_{vap} [$kJ.kg^{-1}$]	194.7	160.02	154.8	367.73	349.56	301.9
λ [$W.m^{-1}.K^{-1}$]	0.089	0.074	0.078	0.103	0.097	0.106
μ [µPa.s]	250	196	269	119	187	206
GWP [-]	1430	4	6	3	4	120
ODP [-]	0	0	0	0	0	0
Flammability	A1	A2L	A2L	A3	A3	A2

III. ENERGY AND EXERGY ANALYSIS

The performances of the vapour compression refrigeration are evaluated based on the first and second laws of thermodynamics. To perform the energy and exergy analysis, the following assumptions were made:

- All processes are steady-state.
- Changes in potential and kinetic energy are negligible.
- An isenthalpic process is assumed in the expansion valve.
- Along the pipelines, pressure losses, heat losses and exergy destruction are neglected.

The generalized exergy balance for a control volume writes [10]:

$$Exd=\sum(Ex)_{in}-\sum(Ex)_{out}+[\sum(Q(1-T_0/T)_{in}-\sum(Q(1-T_0/T)_{out}] + \sum W_{in}-\sum W_{out} \qquad (1)$$

where Exd (W) is the exergy flow destruction, the first two terms are stream exergy flows, the two following terms are heat transfer exergy flows and the latest two terms presents work exergy flows. T_0 (°C) is the reference state temperature (Table 1). The exergy in any state is given by:

$$Ex=\dot{m}\,[(h-h_o)-T_o(S-S_o)] \qquad (2)$$

where $\dot{m}$ (kg/s) is the mass flow rate of the refrigerant, h_0 and S_0 are the enthalpy (kJ/kg) and entropy (kJ/kg.K) of the dead state of the refrigerant at temperature T_0 and pressure P_0. Exergetic efficiency for the whole system can be calculated as:

$$\eta_{ex} = 1-(Exd_{total}/W_{comp}) \quad (3)$$

where Exd_{total} (W) is the total exergy destruction of the system, and W_{comp} (W) is the power consumption of the compressor. The energy performance of the cycle is evaluated by the coefficient of performance COP, which is the ratio between the useful cooling generated by the evaporator over the power consumption of the compressor:

$$COP = Q_e / W_{comp} \quad (4)$$

The balance equations of mass, energy and exergy are applied for a control volume of each component to analyze the system (Table 3).

TABLE 3: ENERGY, AND EXERGY BALANCE EQUATIONS FOR EACH COMPONENT OF THE VAPOUR COMPRESSION REFRIGERATION SYSTEM.

Component	Energy balance	Exergy destruction balance
Compressor	$W_{comp} = \dot{m}\,(h_2-h_1)$	$Exd_{comp}=Ex_1+W_{comp}-Ex_2$
Condenser	$Q_c = \dot{m}\,(h_2-h_3)$	$Exd_c=Ex_2-Ex_3-Q_c\,(1-T_0/T_c)$
Evaporator	$Q_e = \dot{m}\,(h_4-h_1)$	$Exd_e=Ex_4-Ex_1+Q_e\,(1-T_0/T_e)$
Expansion valve	$h_3=h_4$	$Exd_{exp}=Ex_3-Ex_4$

IV. RESULTS AND DISCUSSION

In this investigation, a combined energy and exergy analysis has been carried out for different condensation and evaporation temperatures using 6 different refrigerants. The purpose is to find the best alternative refrigerant to R134a for vapour compression refrigeration systems.

Figure 2 shows the effect of the condenser temperature [30; 45°C] on the exergy destruction of each component of the vapour compression refrigeration cycle while keeping the evaporator temperature at 0°C. It is observed that the rise of the condenser temperature increases the exergy destruction for all refrigerants in each component due to the increase in compressor ratio, and the isentropic losses associated to the compression of a hotter gas. In fact, the highest destruction rate along the process occurs in the compressor. The refrigerant R290 provides the maximum value 60 W, whereas the lowest value is for R600a, 24 W. After the compressor, most of the exergy is destroyed in the expansion valve, followed by the condenser and the evaporator.

Figure 3 illustrates the exergy destruction at different evaporator temperatures [-10; 5°C] for a fixed condenser temperature of 30°C. The results show that the exergy destruction decreases with an increase of the evaporator temperature, since this leads to a lower compression ratio. The destruction is again more important in the compressor followed by the expansion valve, condenser and evaporator. The highest destruction rates are observed for R290: 42.8 W in the compressor, 1.9 W in the condenser, and 1.02 W and 11.27 W in the evaporator and expansion valve, respectively. The lowest values are observed once again for R600a, 16. 55 W in the compressor, 0.14 W and 0.36 W for condenser and evaporator, and 3.45 W in the expansion valve.

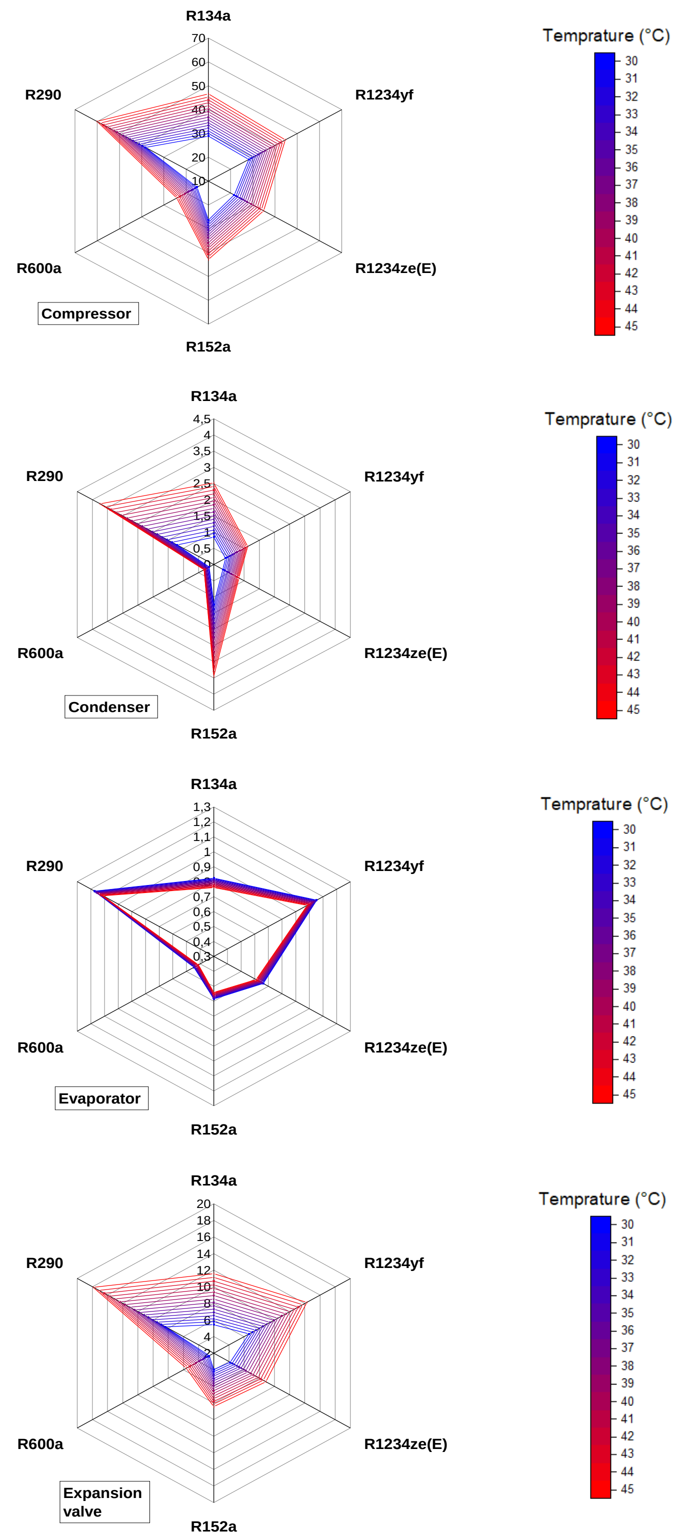

Figure 2: Exergy destruction in [W] of each component at different condenser temperatures (T_c between 30 °C and 45 °C) and fixed evaporation temperature (T_e=0°C).

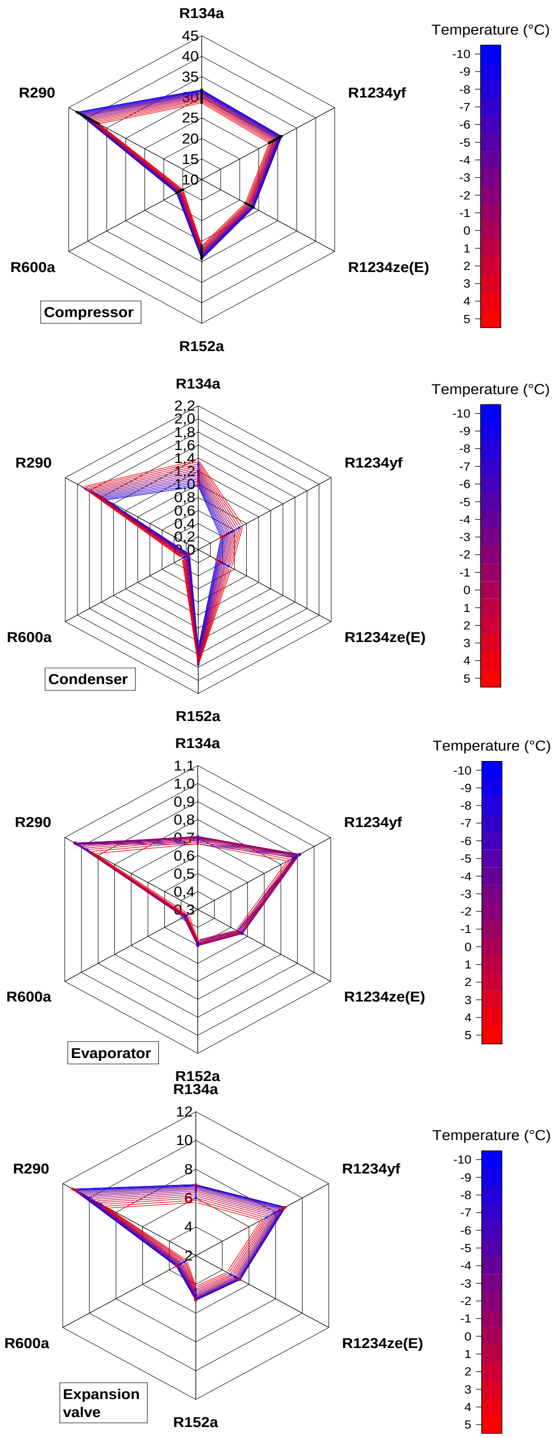

Figure 3: Exergy destruction in [W] of each component at different evaporator temperatures (T_e between -10°C and 5°C) and fixed temperature of the condenser (T_c=30°C).

The highest exergy destruction observed in the compression refrigeration system for all refrigerants are displayed in Figure 4. The results of the total exergy destruction are obtained at a condensation temperature of 45°C and an evaporator temperature of 0 °C (Fig. 4a), and a condensation temperature of 30°C and an evaporation temperature of -10 °C (Fig. 4b). It is observed that the exergy destruction for R290 is about 34.73% (Fig. 4a) and 39.76% (Fig. 4b) larger than the relative values for R134a, whereas with R600a, the system destroys 50.26% (Fig. 4a) and 49% (Fig. 4b) less energy than with R134a for both cases.

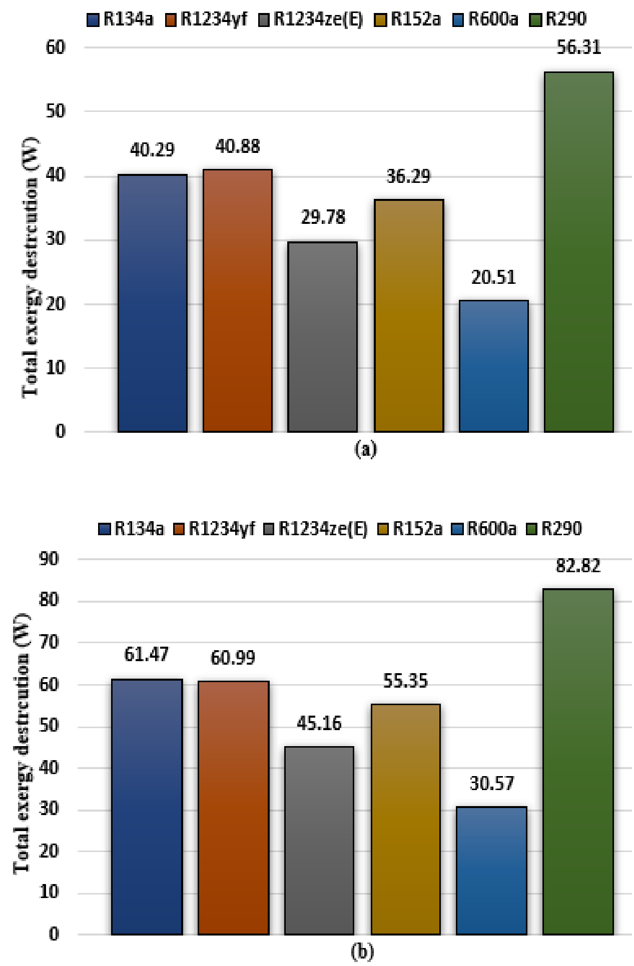

Figure 4: Maximum exergy destruction of the entire system for fixed operating conditions: (a) T_c=45°C and T_e=0°C; (b) T_c=30°C and T_e=-10°C.

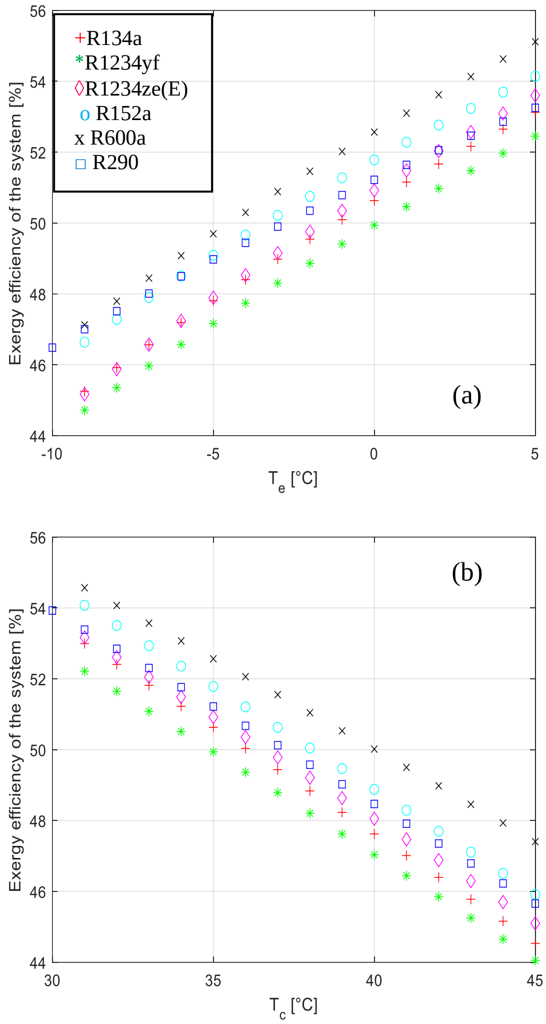

Figure 5: Exergy efficiency of the entire system against: (a) the evaporator temperature at T_c=30°C and (b) the condenser temperature at T_e=0°C.

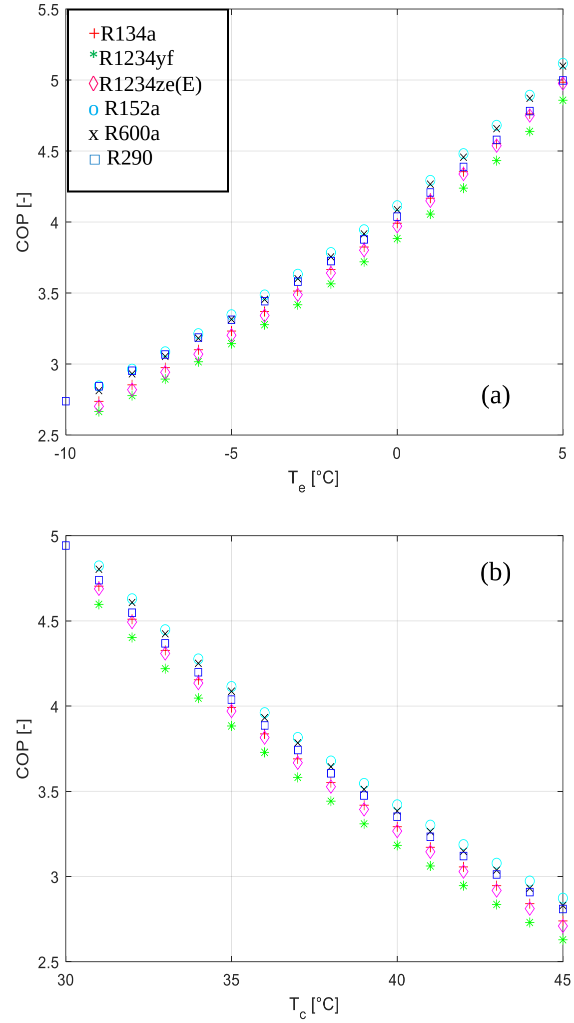

Figure 6: COP of the vapour compression refrigeration cycle against: (a) the evaporator temperature at T_c=30°C and (b) the condenser temperature at T_e=0°C.

The corresponding exergy efficiency and coefficient of performance are presented in Figures 5 and 6, respectively, for the 6 refrigerants, at different evaporator temperatures [-10; 5°C] for a fixed condenser temperature of 30°C, and condenser temperatures between [30; 45°C] keeping the evaporator temperature at 0°C. It is observed that R600a has the higher efficiency for both cases (55.11%), which is about 2% better than for R134a. The lowest value is observed for R1234yf, 52.45%. With regards to the coefficient of performance, the best value is obtained for R152a and R600a, 5.12 and 5.10 respectively. The COP of the R134a cycle is 4.9. The compressor power is sensitive to the pressure ratio due to the increase/decrease of the temperatures in the heat exchangers, which affects the COP and exergy efficiency. The R290 has the lowest compression ratio compared to the other refrigerants.

V. CONCLUSION

A theoretical analysis of the exergy and exergy performance of a water dispenser unit driven by a vapour compression refrigeration system has been conducted using 6 different refrigerants with low GWP and 0 ODP for possible drop-in replacement of R134a. It can be concluded that the COP and the exergy efficiency of the cycle decreases with the condensation temperature and increases with the evaporator temperature. A noticeable portion of exergy destruction occurs in the compressor followed by the expansion valve, the condenser, and the evaporator. The results showed that R290 exhibited the highest values of the total exergy destruction and R600a the lowest values. Moreover, R600a also showed the highest values of exergy efficiency and a COP of 5.10, which is about 4% higher than for R134a. The comparison between low GWP refrigerants makes R600a the best overall alternative solution for R134a in vapour compression refrigeration systems. Other options such as R1234ze(E) are also an attractive solution since their performances are very close to those of R134a. Further experiments are now deemed necessary to validate the theoretical results using different working fluids with low GWP.

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