COMPARATIVE EVALUATION OF A TWO-STAGE REFRIGERATION SYSTEM WITH FLASH INTERCOOLING USING DIFFERENT REFRIGERANTS

by

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The aim of this study is to investigate the effect of low global warming potential refrigerants on the optimum intermediate pressure ($P_{OPT,int}$) and performance (COP) values of a refrigeration system with flash intercooling. For realize, the optimum operating parameters of system were determined in low temperature applications through a theoretical analysis according to the different refrigerants (R290, R404A, R407C, R507A, and R22). The theoretical modelling of system is done by optimizing the intermediate pressure at given evaporation (T_E) and condensation (T_C) temperatures for selected refrigerants. After optimization, the maximized values of COP and Second law efficiency are computed from the predicted values of $P_{OPT,int}$. The linear regression method is then used to derive three correlations of $P_{OPT,int}$, maximum values of COP and Second law efficiency according to T_E and T_C. Hence, the $P_{OPT,int}$ values maximizing the system performance are found from various T_E and T_C values for each refrigerant. Due to calculations, increasing T_E and T_C cause the increase in $P_{OPT,int}$ in low temperature applications. The R507A system has the highest $P_{OPT,int}$ values and R22 system has the lowest $P_{OPT'int}$ values. Although R22 system has slightly more efficient than R290 system, it is being phased out worldwide because of the risk of ozone depletion potential and global warming potential considerations. Therefore, it is important to evaluate the R22 replacement options. The R290 was discovered to have better performance than the R404A, R407C and R507A systems in terms of COP_{max} (1.81), global warming potential (11), and ozone depletion potential (0) when T_E and T_C are -35°C and 40 °C.

Key words: *optimization, low temperature applications, COP, Second law efficiency, exergy loss*

Introduction

In many refrigeration applications, the pressure difference between evaporation and condensation is low for the simple vapor compression systems. For high temperature or pressure difference, the multi-stage systems (either staged compression or cascade system) should be suggested medium and low temperature applications (LTA). In particular, two-stage systems use one refrigerant to optimize the inter-stage conditions. The previous thermodynamic models show that two-stage cycles with flash tank have better performance than the single-stage cycles. Their performance depends significantly on the intermediate pressure, P_{int}, corresponding to the minimum compressor work. Therefore, it is important to determine the optimum intermediate pressure, $P_{OPT,int}$, for different operating conditions to improve the performance of the two-stage cycles with flash tank [1-4].

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In literature, there are many theoretical and experimental studies about two-stage cycles with different configurations to improve their performance. Mbarek *et al.* [5] analyzed theoretically three different configurations of R134a two-stage refrigeration system. All three systems have the same main components, but they differ in the arrangement of the flash tank to improve energy efficiency. It is determined that the place of the flash tank is an important parameter for computation of the system performance. Jiang *et al.* [6] analyzed six different configurations of two-stage compression systems using the refrigerants R22 and ammonia. They evaluated the role of the parameters that influence the optimum P_{int} in the design of different systems. Yuan and Xia [7] investigated experimentally the performances of a heat pump with a flash-tank and a heat pump with a sub-cooler using R22. At low ambient temperatures, the heat pump system with the flash tank is more efficient than the sub-cooler system for heating performances. Arora and Dhar [8] determined the P_{int} of the two-stage compression system. They found that the different configurations and various operating conditions of optimum P_{int} are important to improve the performance of two-stage flash intercooler system. Torrella *et al.* [1] analyzed experimentally the inter-stage working conditions of a R404A two-stage compression system which operates with two different configurations in medium and low refrigeration applications. They showed that the P_{int} pressure depends on the cycle configuration and the operating conditions.

In literature, there are many theoretical studies which analyze the effects of various operation parameters on the COP of cascade systems using the natural and synthetic refrigerant pairs for LTA. Lee *et al.* [9] evaluated theoretically the optimal condensing temperature of the cascade heat exchanger and the COP for evaporating levels between –45 °C and –55 °C. Dopaza *et al.* [10] also theoretically analyzed the influence of the cycle parameters on its efficiency and evaluated the optimal condensing temperature. Calculated results show that the COP increases 70% when the evaporation temperature varies from –55 °C to –30 °C. Getu and Bansal [11] analyzed theoretically cascades of CO_2 with ammonia, propane, propylene, ethanol, and R404A concluding that the best couple from an energy point of view was ethanol/CO_2 followed by NH_3/CO_2. Yilmaz *et al.* [12] examined the effect of operating conditions on a CO_2/R404A cascade system's performance in terms of the COP and the η_{II}. Kilicaraslan and Hosoz [13] determined and compared the COP and irreversibility of the cascade system using a large family of environment friendly refrigerant pairs. Dokandari *et al.* [14] theoretically investigated the ejector utilization's effect on the performance of the conventional CO_2/NH_3 cascade system. They indicated that the employment of the ejectors has considerable effect on the performance of the conventional cascade system. Yilmaz *et al.* [15] examined mathematically the performance of a two-stage subcritical CO_2/NH_3 cascade refrigeration system for different operating conditions. They proposed correlations to predict the maximum COP for given operation parameters.

In literature, there are few theoretical and experimental studies to evaluate the alternative refrigerants with respect to performance and environmental considerations in refrigeration applications [16-18]. Spatz *et al*., [16] studied three alternatives such as hydrofluorocarbon (HFC) (R404A, R410A) and HC (R290) to replace R22 for LTA. As a result, R410A was discovered to be an efficient and environmentally acceptable option. Lopis *et al.* [17] presented the experimental evaluation of refrigerants R404A and R507A in the double-stage refrigeration plant. They determined that the performance of R404A system is slightly higher than R507A system's when the plant operates without inter-stage system at low evaporation temperatures. Pansulla and Allgood [18] reported the results of a study comparing R449A with R22 in low and medium evaporation temperatures. Modelling and experiment results of the system suggest that R449A can replace R22.

Previous work in this area mainly focused on the theoretical and experimental analysis of two-stage cycles with different configurations. These studies found that the different configurations and various operating conditions of $P_{\mathrm{OPT,int}}$ were important to improve the performance of two-stage flash intercooler system. However, the impact of some refrigerants, which have low GWP, on the $P_{\mathrm{OPT,int}}$ and $\mathrm{COP}_{\mathrm{max}}$ values were not sufficiently analyzed. This work will investigate the effect of low GWP refrigerants on the $P_{\mathrm{OPT,int}}$ and $\mathrm{COP}_{\mathrm{max}}$ values of system through the thermodynamic analysis.

This study proposes a theoretical model of a two-stage refrigeration system with flash intercooling for different refrigerants at LTA. In LTA (–20 °C to –40 °C), the performance of the system is theoretically investigated for the replacement of synthetic refrigerants (R404A, R507A, R407C, and R22) with the natural alternative (R290). In LTA, the theoretical models suggest three correlations that are function of T_{E} and $\mathrm{T_C}$ for all refrigerants. The first one determines the $P_{\mathrm{OPT,int}}$ and the second one determines the $\mathrm{COP}_{\mathrm{max}}$ corresponding to the $P_{\mathrm{OPT,int}}$. The third one determines the $\eta_{\mathrm{II,max}}$ corresponding to the $P_{\mathrm{OPT,int}}$. Therefore, the effects of T_E and T_C on the system $P_{\mathrm{OPT,int}}$, $\mathrm{COP}_{\mathrm{max}}$, $\eta_{\mathrm{II,max}}$ are evaluated for LTA's refrigerants. Furthermore, the harmful environmental effects of applications are compared across all refrigerants. The models of all refrigerants could be used to develop data for the future experimental refrigeration applications.

Background

Properties of selected refrigerants

Natural refrigerants are increasingly used in LTA. In order to make a decision the best refrigerant in the applications, significant characteristics for instance ozone depletion potential (ODP), global warming potential (GWP), toxicity, flammability, *etc.* should be analyzed with the operating conditions. The ODP values, GWP values and the physical properties of refrigerants that are subject to this study are given in tab. 1.

Table 1. The physical and environmental properties of refrigerants used in this study [19]

Refrigerants		Refrigerant's classification	Molecular weight [kgk^{-1}mol^{-1}]	T_{CR} [°C]	P_{CR} [MPa]	NBP [°C]	ODP	GWP	Safety class
LTA	R404A	HFC	97.60	72.12	3.765	–46.5	0	3921	A1
	R507A	HFC	98.60	70.5	3.70	–46.7	0	3985	A1
	R407C	HFC	86.20	86.11	4.63	–42.0	0	1600	A1
	R22	HCFC	86.47	96.14	4.99	–40.81	0.05	1810	A1
	R290	HC	44.09	134.6	4.23	–42.09	0	11	A3

The ODP and GWP values show that the correct selection of environment-friendly refrigerant is essential to reduce their harmful effects on the environment. The R22 contains chlorine and it is considered one of the worst refrigerants which deplete the ozone layer. Major advantage of R507A, R404A, R407C, and R22 is that they fall in A_1 safety class. They are non-flammable refrigerants although they have high GWP. The R290, on the other hand, has negligible GWP and high flammability which puts this natural refrigerant into the safety rating A_3 [20]. Hence, using R290 requires additional safety measures. Therefore, this study investigates possible alternatives of R290. R404A, R507A, R407C, R22, and R290 are also investigated and compared in detail in order to decide the environment friendly and efficient refrigerant in LTA.

Two-stage refrigeration system with flash intercooling

The two-stage refrigeration system with flash intercooling is selected to investigate in detail for different refrigerants. In two-stage refrigeration cycle, the flash tank is located between the high and low pressure (HP and LP) cycles. The schematic diagram of a two-stage cycle with flash intercooling is shown in fig. 1(a). At the exit of the condenser – 5, the refrigerant is saturated liquid. Then the refrigerant is throttled to the P_{int} and enters to the flash intercooler which cools the refrigerant to state – 7. It is then throttled to the evaporator pressure at state – 8. The superheated vapor from the evaporator at state – 1, is compressed in the LP compressor to state – 2, when it enters the flash intercooler. De-superheating of the vapor takes place in the flash intercooler by evaporation of liquid refrigerant. Flash intercooler increases the mass-flow rate of refrigerant to HP compressor and reduces the mass-flow rate of refrigerant coming to the evaporator. Saturated vapor from the flash intercooler at state – 3, is compressed to the state – 4, and superheated vapor is cooled in the condenser.

As shown in the *P*-*h* diagram in fig. 1(b), there are evaporation or low pressures ($P_E = P_1 = P_8$), condensation or high pressures ($P_C = P_4 = P_5$) and intermediate pressures ($P_{\text{int}} = P_2 = P_3 = P_6 = P_7$) between the two compression stages. The HP corresponds to the T_C and the LP corresponds to the T_E.

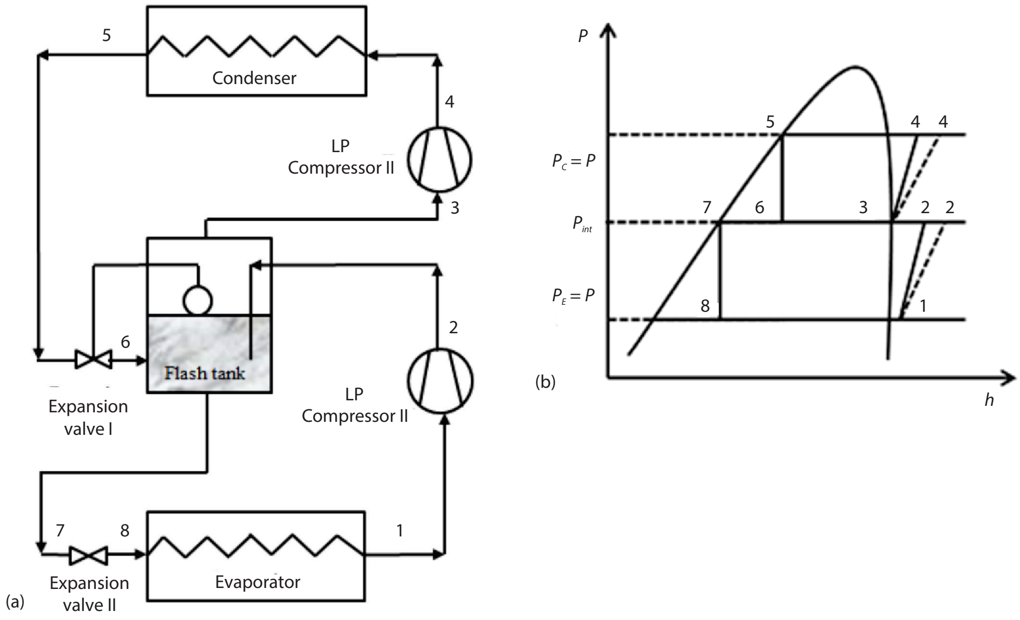

Figure 1. Schematic of the system with flash intercooling (a) and the system *P-h* diagram (b)

Thermodynamic analysis

The thermodynamic model of the two-stage refrigeration system with flash intercooling is developed based on First and Second laws of thermodynamics. Mass, energy and exergy equations are derived for both low and high pressure cycles. The COP_{max} and the η_{IImax} values are computed for various operating conditions for all investigated refrigerants [21]. In this section, the equations are developed for the analysis considering the state points of figs. 1(a) and 1(b).

In the analysis, the following assumptions are taken into account:
- heat gains and pressure and other losses are ignored in all system components,
- potential and kinetic energy changes are neglected,
- isentropic compressor efficiencies are assumed to be 0.80 in low and high pressure cycles,
- isenthalpic expansion of refrigerants is in expansion valves,
- saturated vapor state at the exit of evaporator, ΔT_{sup}, is 7 °C,
- saturated liquid state at the exit of condenser, ΔT_{sub}, is 0 °C, and
- the evaporator cooling capacity of the systemm, $\dot{Q}_{evap}$, is 6 kW.

Thermodynamic analysis is carried out using EES software [22]. The EES is a general equation solving program that can numerically solve coupled non-linear algebraic and differential equations. It is also capable to perform optimization studies.

Mass and energy analysis

The cooling capacity of the LP evaporator is defined:

$$\dot{Q}_{\text{evap}} = \dot{m}_{\text{evap}} \left(h_1 - h_8 \right) \tag{1}$$

The refrigerant mass-flow rate of LP cycle is defined:

$$\dot{m}_{\text{evap}} = \frac{\dot{Q}_{\text{evap}}}{\left(h_1 - h_8 \right)} \tag{2}$$

Compressor power consumption for LP and HP compressors:

$$\dot{W}_{\text{LPC}} = \dot{m}_{\text{evap}} \left(h_2 - h_1 \right) \tag{3}$$

$$\dot{W}_{\text{HPC}} = \dot{m}_{\text{cond}} \left(h_4 - h_3 \right) \tag{4}$$

Energy balance for the HP expansion valve I:

$$h_5 = h\left(P = P_C, x = 0 \right) \tag{5}$$

$$h_5 = h_6 \tag{6}$$

The intermediate pressure between the two compression stages is calculated from the derived correlation of $P_{\text{OPT,int}}$ for the selected refrigerant.

Energy balance for the LP expansion valve II:

$$h_7 = h\left(P = P_{\text{int}}, x = 0 \right) \tag{7}$$

$$h_7 = h_8 \tag{8}$$

The mass-flow rates of refrigerants through the condenser and evaporator are different for system with flash inter-cooling. The ratio mass-flow rates can be obtained from an energy balance on the flash intercooler. Energy balance of flash intercooler is defined as:

$$\dot{m}_{\text{cond}} h_6 + \dot{m}_{\text{evap}}\, h_2 = \dot{m}_{\text{evap}} h_7 + \dot{m}_{\text{cond}} h_3 \tag{9}$$

The mass-flow rate ratio is determined:

$$r = \frac{\dot{m}_{\text{cond}}}{\dot{m}_{\text{evap}}} = \frac{\left(h_2 - h_7 \right)}{\left(h_3 - h_6 \right)} \tag{10}$$

The heat transfer rate into the flash intercooler is determined:

$$\dot{Q}_{\mathrm{FC}} = \dot{m}_{\mathrm{cond}}\left(h_6 - h_3\right) = \dot{m}_{\mathrm{evap}}\left(h_2 - h_7\right) \tag{11}$$

The refrigerant mass-flow rate of HP cycle is defined:

$$\dot{m}_{\mathrm{cond}} = \dot{m}_{\mathrm{evap}}\frac{\left(h_2 - h_7\right)}{\left(h_3 - h_6\right)} \tag{12}$$

The heat transfer rate of the HP cycle condenser is defined:

$$\dot{Q}_{\mathrm{cond}} = \dot{m}_{\mathrm{cond}}\left(h_4 - h_5\right) \tag{13}$$

And finally, the overall COP of the system with flash intercooling is determined:

$$COP = \frac{\dot{Q}_{\mathrm{evap}}}{\dot{W}_{\mathrm{LPC}} + \dot{W}_{\mathrm{HPC}}} = \frac{\dot{m}_{\mathrm{evap}}\left(h_1 - h_8\right)}{\dot{m}_{\mathrm{evap}}\left(h_2 - h_1\right) + \dot{m}_{\mathrm{cond}}\left(h_4 - h_3\right)} \tag{14}$$

Exergy analysis

The exergy loss rate for the LP cycle evaporator:

$$\dot{X}_{\mathrm{evap}} = \dot{m}_{\mathrm{evap}}\left[\left(h_1 - h_8\right) + T_o\left(s_1 - s_8\right)\right] + \dot{Q}_{\mathrm{evap}}\left(\frac{T_0}{T_E}\right) \tag{15}$$

The exergy loss rate of the LP and HP compressors:

$$\dot{X}_{\mathrm{LPC}} = \dot{m}_{\mathrm{evap}}T_0\left(s_2 - s_1\right) + \dot{W}_{\mathrm{LPC}} \tag{16}$$

$$\dot{X}_{\mathrm{HPC}} = \dot{m}_{\mathrm{cond}}T_0\left(s_4 - h_3\right) + \dot{W}_{\mathrm{HPC}} \tag{17}$$

$$\dot{X}_{\mathrm{total,C}} = \dot{X}_{\mathrm{LPC}} + \dot{X}_{\mathrm{HPC}} \tag{18}$$

The exergy loss rate for the condenser of HP cycle:

$$\dot{X}_{\mathrm{cond}} = \dot{m}_{\mathrm{cond}}\left[\left(h_4 - h_5\right) + T_0\left(s_4 - s_5\right)\right] - \dot{Q}_{\mathrm{cond}}\left(\frac{T_0}{T_C}\right) \tag{19}$$

The exergy loss rates for expansion valves I and II:

$$\dot{X}_{\mathrm{EV}_1} = \dot{m}_{\mathrm{cond}}\left[T_0\left(s_5 - s_6\right)\right] \tag{20}$$

$$\dot{X}_{\mathrm{EV}_2} = \dot{m}_{\mathrm{evap}}\left[T_0\left(s_7 - s_8\right)\right] \tag{21}$$

$$\dot{X}_{\mathrm{total,EV}} = \dot{X}_{\mathrm{EV}_1} + \dot{X}_{\mathrm{EV}_2} \tag{22}$$

The exergy loss rate in the flash intercooler:

$$\dot{X}_{\mathrm{FC}} = \dot{m}_{\mathrm{evap}}\left[\left(h_2 - h_7\right) - T_0\left(s_2 - s_7\right)\right] + \dot{m}_{\mathrm{cond}}\left[\left(h_6 - h_3\right) - T_0\left(s_6 - s_3\right)\right] \tag{23}$$

Total exergy loss rate of the system:

$$\dot{X}_{\mathrm{lost,total}} = \dot{X}_{\mathrm{evap}} + \dot{X}_{\mathrm{cond}} + \dot{X}_{\mathrm{total,EV}} + \dot{X}_{\mathrm{total,C}} + \dot{X}_{\mathrm{FC}} \tag{24}$$

The Second law efficiency is used to measure the system performance in eqs. (25) and (26):

$$\eta_{\mathrm{II}} = \frac{\dot{W}_{\mathrm{Rev}}}{\dot{W}_{\mathrm{Act}}} \tag{25}$$

$$\dot{W}_{\mathrm{Rev}} = Q_{\mathrm{evap}} \left(\frac{T_0}{T_{\mathrm{E}}} - 1 \right) \tag{26}$$

In eq. (26) is the reversible power. The T_0 and T_{E} are the dead and evaporation temperatures, respectively.

Optimization method

The conjugate direct method (or the direct search method) in the EES software is used to find the maximum/minimum value of a decision variable within a certain range for design variables. In this study, the objective of optimization is to find the best possible system performance of each refrigerant for different operating conditions. Therefore, the P_{int} is initially optimized for different values of $P_{\mathrm{C}}/P_{\mathrm{E}}$ and $T_{\mathrm{C}}/T_{\mathrm{E}}$. The COP and η_{II} values are maximized by various $P_{\mathrm{OPT,int}}$ values that are determined from the former optimization. Model equations can be expressed as a function of two operating or design parameters for various refrigerants.

Maximize COP (T_{C}, T_{E}) and η_{II} (T_{C}, T_{E}). Subject to:

– for all refrigerants

$$25\,^{\circ}\mathrm{C} \le T_{\mathrm{C}} \le 45\,^{\circ}\mathrm{C} \;\text{ and }\; P_{\mathrm{sat\ at}\ T_{\mathrm{C}}=25^{\circ}C} \le P_{\mathrm{C}} \le P_{\mathrm{sat\ at} T_{\mathrm{C}}=45^{\circ}C} \tag{27}$$

– for R290 and R407C

$$-35\,^{\circ}\mathrm{C} \le T_{\mathrm{E}} \le -20\,^{\circ}\mathrm{C} \text{ and } P_{\mathrm{sat\,at}\ T_{\mathrm{E}}=35^{\circ}\mathrm{C}} \le P_{\mathrm{E}} \le P_{\mathrm{sat\,at}\ T_{\mathrm{E}}=-20^{\circ}\mathrm{C}} \tag{28}$$

– for R404A, R507A, and R22

$$-40\,^{\circ}\mathrm{C} \le T_{\mathrm{E}} \le -20\,^{\circ}\mathrm{C} \;\text{ and }\; P_{\mathrm{sat\ at}\ T_{\mathrm{E}}=-40^{\circ}\mathrm{C}} \le P_{\mathrm{E}} \le P_{\mathrm{sat\ at}\,T\,T_{\mathrm{E}}=-20^{\circ}\mathrm{C}} \tag{29}$$

The lower and upper bound of saturation vapor pressure are specific to the studied refrigerants in constraints (28) and (29). The constraint (27) is for T_{C} and it is general for all the refrigerants.

Results and discussion

The mathematical model of a system is implemented in EES in order to estimate the performance parameters such as the COP, the η_{II} and the $P_{\mathrm{OPT,int}}$ values for LTA's refrigerants (R290, R404A, R507A, R407C, and R22).

The system's operating conditions are chosen based on both the conditioned space and the ambient conditions. In case studies of the ambient conditions, the T_{C} values are varied from 25-45 °C for all refrigerants. Concerning the conditioned space, the T_{E} values are considered in general to be between –20 °C and –40 °C. In this temperature range, the lowest temperature that R404A, R507, and R22 can reach is –40 °C whereas the lowest temperature that R290 and R407C can reach is –35 °C. As seen in the figs. 2 and 3, all refrigerants can be used at the evaporation temperatures –35 °C, since it is the only applicable temperature for all refrigerants. The T_{E} values of –35 °C are chosen as reference point for comparison of LTA refrigerants (the vertical dotted line at –35 °C). In calculations, the degrees of subcooling and superheat are chosen to be 0 °C and 7 °C, respectively. The evaporator cooling capacity is kept constant at 6 kW as in a medium scale supermarket.

Optimization study results

The linear regression method is applied in two-variable optimization calculations using EES software. Three correlations are computed for each refrigerant from the two operating condition variables T_E and T_C. First, the $P_{OPT,int}$ is calculated from the given T_E and T_C. Then, the COP_{max} and $\eta_{II,max}$ are calculated from the former $P_{OPT,int}$ values. The computed correlations are presented in tab. 2. The unit used in the calculations is Kelvin.

Table 2. The correlations obtained for different refrigerants

Refrigerants of LTA			
R290			
$P_{OPT,int}$	$P_{OPT,int,\,R290} = -3970 + 7.0918T_E + 9.0857T_C$	$R^2 = 99.67\%$
COP_{max}	$COP_{max,\,R290} = 3.5739 + 0.0466T_E - 0.0410T_C$	$R^2 = 98.85\%$
$\eta_{II,max}$	$\eta_{IImax,\,R290} = 34.6534 + 0.01472T_E + 0.0606T_C$	$R^2 = 90.61\%$
R507A		
$P_{OPT,int}$	$P_{OPT,int,\,R507A} = -5575.2 + 9.102T_E + 13.319T_C$	$R^2 = 99.46\%$
COP_{max}	$COP_{max,\,R507A} = 3.954 + 0.03868T_E - 0.03655T_C$	$R^2 = 99.13\%$
$\eta_{II,max}$	$\eta_{IImax,\,R507A} = 44.143 + 0.038T_E + 0.0056T_C$	$R^2 = 98.16\%$
R404A		
$P_{OPT,int}$	$P_{OPT,int,\,R404A} = -5297.7 + 8.7774T_E + 12.596T_C$	$R^2 = 99.58\%$
COP_{max}	$COP_{max,\,R404A} = 3.8127 + 0.0381T_E - 0.0357T_C$	$R^2 = 99.11\%$
$\eta_{II,max}$	$\eta_{IImax,R404A} = 35.512 + 0.02216T_E + 0.0441T_C$	$R^2 = 93.66\%$
R407C		
$P_{OPT,int}$	$P_{OPT,int,\,R407C} = -4870.4 + 8.5734T_E + 10.9743T_C$	$R^2 = 99.63\%$
COP_{max}	$COP_{max,\,R407C} = 2.8501+0.0398T_E - 0.0341T_C$	$R^2 = 99.07\%$
$\eta_{II,max}$	$\eta_{IImax,\,R407C} = 34.510 + 0.0105T_E + 0.054T_C$	$R^2 = 97.85\%$
R22		
$P_{OPT,int}$	$P_{OPT,int,\,R22} = -3787.72 + 9.9696T_E + 5.9748T_C$	$R^2 = 99.33\%$
COP_{max}	$COP_{max,\,R22} = 3.7735 + 0.0523T_E - 0.046T_C$	$R^2 = 96.60\%$
$\eta_{II,max}$	$\eta_{IImax,\,R22} = 39.996 + 0.01694T_E + 0.04234T_C$	$R^2 = 99.41\%$

The proposed correlations are used to predict the $P_{OPT,int}$, COP_{max}, and $\eta_{II,max}$ values from different operating parameters T_C and T_E. Table 3 presents these predictions for LTA's refrigerants. The reference point of T_E to compare different refrigerants is set to be –35 °C in LTA.

Quadha *et al.* [23] performed a detailed energy and exergy analysis of a two-stage refrigeration cycle using R290 and ammonia. The calculated COP of both refrigerants decreases with the increases of T_C between 30 °C and 60 °C by choosing the T_E constant at –30 °C. The predicted COP values of R290 can be seen in detail in tab. 3.

Table 3. Predictions of $P_{OPT,int}$, COP_{max}, and $\eta_{II,max}$ for LTA with different refrigerants

T_C [K]	T_E [K]	*R*290			*R*290 [23]	*R*404A			*R*507A		
		$P_{OPT,int}$ [kPa]	COP_{max}	$\eta_{II,max}$ [%]	COP_{max}	$P_{OPT,int}$ [kPa]	COP_{max}	$\eta_{II,max}$ [%]	$P_{OPT,int}$ [kPa]	COP_{max}	$\eta_{II,max}$ [%]
318	263	NA	NA	NA	–	NA	NA	NA	NA	NA	NA
	258	NA	NA	NA	–	NA	NA	NA	NA	NA	NA
	253	713.5	2.30	57.65	–	928.5	2.09	55.14	963.1	2.12	55.54
	248	678.0	2.07	57.57	–	884.6	1.90	55.03	917.6	1.92	55.35
	243	642.6	1.83	57.50	2.0	840.7	1.71	54.91	872.1	1.73	55.16
	238	**607.1**	**1.60**	**57.43**	–	**796.9**	**1.52**	**54.80**	**826.6**	**1.54**	**54.97**
	233	NA	NA	NA	–	753.0	1.33	54.69	781.1	1.34	54.78
313	263	NA	NA	NA	–	NA	NA	NA	NA	NA	NA
	258	NA	NA	NA	–	NA	NA	NA	NA	NA	NA
	253	668.1	2.50	57.35	–	865.5	2.25	54.92	896.5	2.30	55.51
	248	632.6	2.27	57.27	–	821.6	2.05	54.80	851.0	2.11	55.32
	243	597.1	2.04	57.20	2.2	777.8	1.86	54.69	805.5	1.91	55.13
	238	**561.7**	**1.81**	**57.12**	–	**733.9**	**1.67**	**54.58**	**760.0**	**1.72**	**54.94**
	233	NA	NA	NA	–	690.0	1.49	54.47	714.5	1.53	54.75
308	263	NA	NA	NA	–	NA	NA	NA	NA	NA	NA
	258	NA	NA	NA	–	NA	NA	NA	NA	NA	NA
	253	622.6	2.71	57.04	–	802.6	2.45	54.70	829.9	2.48	55.48
	248	587.2	2.48	56.97	–	758.7	2.25	54.58	784.4	2.29	55.29
	243	551.7	2.24	56.90	2.4	714.8	2.06	54.47	738.9	2.10	55.10
	238	**516.3**	**2.01**	**56.82**	–	**670.9**	**1.87**	**54.36**	**693.4**	**1.90**	**54.91**
	233	NA	NA	NA	–	627.0	1.68	54.25	647.9	1.71	54.72
303	263	NA	NA	NA	–	NA	NA	NA	NA	NA	NA
	258	NA	NA	NA	–	NA	NA	NA	NA	NA	NA
	253	577.2	2.92	56.74	–	739.6	2.62	54.47	763.3	2.67	55.45
	248	541.7	2.68	56.67	–	695.7	2.43	54.36	717.8	2.47	55.26
	243	506.3	2.45	56.59	2.61	651.8	2.24	54.25	672.3	2.28	55.07
	238	**470.8**	**2.22**	**56.52**	–	**607.9**	**2.05**	**54.14**	**626.8**	**2.09**	**54.88**
	233	NA	NA	NA	–	564.0	1.86	54.03	581.3	1.89	54.69
298	263	NA	NA	NA	–	NA	NA	NA	NA	NA	NA
	258	NA	NA	NA	–	NA	NA	NA	NA	NA	NA
	253	531.8	3.12	56.44	–	676.6	2.80	54.25	696.7	2.85	55.43
	248	496.3	2.89	56.36	–	632.7	2.61	54.14	651.2	2.65	55.24
	243	460.9	2.66	56.29	–	588.8	2.42	54.03	605.7	2.46	55.05
	238	**425.4**	**2.42**	**56.22**	–	**544.9**	**2.23**	**53.92**	**560.2**	**2.27**	**54.86**
	233	NA	NA	NA	–	501.0	2.04	53.81	514.7	2.07	54.67

Table 4. Prediction values ($P_{OPT,int}$, COP$_{max}$, and, $\eta_{II,max}$) and experimental data for LTA's refrigerants

T_C [K]	T_E [K]	*R*407C			*R*22			*R*22EXP [7]
		$P_{OPT,int}$ [kPa]	COP_{max}	$\eta_{II,max}$ [%]	$P_{OPT,int}$ [kPa]	COP$_{max}$	$\eta_{II,max}$ [%]	COP$_{max}$
318	263	NA	NA	NA	NA	NA	NA	–
	258	NA	NA	NA	NA	NA	NA	–
	253	788.5	2.08	54.34	634.6	2.38	57.75	**2.1**
	248	745.6	1.88	54.29	584.7	2.12	57.66	**1.8**
	243	702.8	1.68	54.23	534.9	1.85	57.58	–
	238	**659.9**	**1.48**	**54.18**	**485.0**	**1.59**	**57.49**	–
	233	NA	NA	NA	435.2	1.33	57.41	–
313	263	NA	NA	NA	NA	NA	NA	–
	258	NA	NA	NA	NA	NA	NA	–
	253	733.6	2.25	54.07	604.7	2.61	57.53	**2.2**
	248	690.8	2.05	54.02	554.9	2.35	57.45	**1.9**
	243	647.9	1.85	53.96	505.0	2.08	57.36	–
	238	**605.0**	**1.65**	**53.91**	**455.2**	**1.82**	**57.28**	–
	233	NA	NA	NA	405.3	1.56	57.20	–
308	263	NA	NA	NA	NA	NA	NA	–
	258	NA	NA	NA	NA	NA	NA	–
	253	678.8	2.42	53.80	574.8	2.84	57.32	–
	248	635.9	2.22	53.75	525.0	2.58	57.24	–
	243	593.0	2.02	53.69	475.1	2.31	57.15	–
	238	**550.2**	**1.82**	**53.64**	**425.3**	**2.05**	**57.07**	–
	233	NA	NA	NA	375.4	1.79	56.98	–
303	263	NA	NA	NA	NA	NA	NA	–
	258	NA	NA	NA	NA	NA	NA	–
	253	623.9	2.59	53.53	545.0	3.07	57.11	–
	248	581.0	2.39	53.48	495.1	2.81	57.03	–
	243	538.1	2.19	53.42	445.3	2.54	56.94	–
	238	**495.3**	**1.99**	**53.37**	**395.4**	**2.28**	**56.86**	–
	233	NA	NA	NA	345.6	2.02	56.77	–
298	263	NA	NA	NA	NA	NA	NA	–
	258	NA	NA	NA	NA	NA	NA	–
	253	569.0	2.76	53.26	515.1	3.30	56.90	–
	248	526.1	2.56	53.21	465.2	3.04	56.81	–
	243	483.3	2.36	53.15	415.4	2.77	56.73	–
	238	**440.4**	**2.16**	**53.10**	**365.5**	**2.51**	**56.65**	–
	233	NA	NA	NA	315.7	2.25	56.56	–

Yuan and Xia [7] investigated and compared experimentally the performances of a heat pump with a flash-tank and a heat pump with a sub-cooler using R22. For the system with flash-tank, they determined that increasing the T_E from –25 °C to –7 °C increases the COP values. In addition, increasing the T_C from 42-45 °C reduces the COP values of R22. In tab. 4, when T_E and T_C are –20 °C and 45 °C, the R22's predicted COP and the R22's experimental COP are 2.38 and 2.1, respectively.

Effects of operation parameters

Effect of T_E on $P_{OPT,int}$ and maximized COP

The T_E is considered in general to be between –40 °C and –20 °C by keeping the T_C constant at 40 °°C. In LTA, T_E values range from –40 °C to –20 °C for R507A, R404A, and R22. For R290 and R407C, T_E values change from –35 °C to –20 °C. The influence of T_E on the COP_{max} and $P_{OPT,int}$ is examined for different refrigerants of system. Figure 2 depicts the effect of T_E on the calculated $P_{OPT,int}$ values which maximize the COP. It is seen that decreasing the T_E decreases the $P_{OPT,int}$ for all the refrigerants. For LTA, the largest $P_{OPT,int}$ values are observed when R507A is used as refrigerant whereas the lowest $P_{OPT,int}$ values are found when R22 is used. The $P_{OPT,int}$ values of R290 system lie within the values of R407C and R22 systems. In fig. 3, decreasing the T_E in LTA from –20 °C to –40 °C decreases the COP_{max} values of refrigerants (R290, R404A, R407C, R507A, and R22). The R290 and R22 have the highest COP_{max} values among all five refrigerants at –35 °C. In addition, R290 and R22 have approximately same COP_{max} values at –35 °C. The COP values with refrigerant of R290 are higher than with the other HFC's refrigerants between –35 °C and –20 °C. R404A, R507A, and R22 are the only refrigerant options that could be used below –35 °C in LTA.

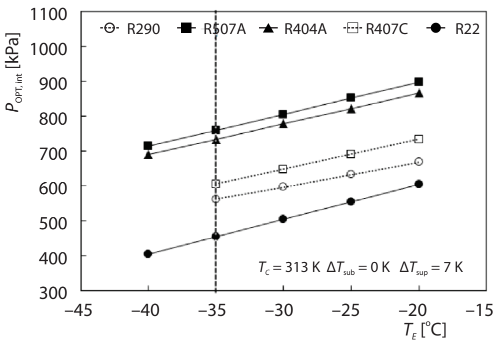

Figure 2. The $P_{OPT,int}$ values *vs.* T_E

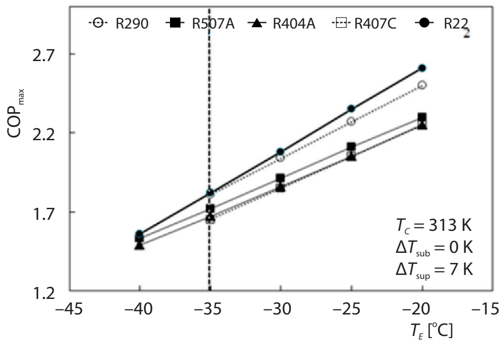

Figure 3. Effect of T_E on COP_{max}

Effect of T_C on $P_{OPT,int}$ and maximized COP

In figs. 4 and 5, the variation of $P_{OPT,int}$ and COP_{max} in function of the T_C is observed from 25 °C to 45 °C. In the previous section, we determined the T_E to be –35 °C as it was the only applicable T_E for all refrigerants. Figure 4 depicts the effect of T_C on the calculated $P_{OPT,int}$ values which maximize the COP for LTA. It is seen that increasing the T_C raises the $P_{OPT,int}$ for all the refrigerants.

In LTA, the largest values of $P_{OPT,int}$ are observed when R507A is used whereas the lowest values of $P_{OPT,int}$ are calculated when R22 is used. The $P_{OPT,int}$ values of R290 are located within the values of R407C and R22. The R507A and R404A have approximately same $P_{OPT,int}$

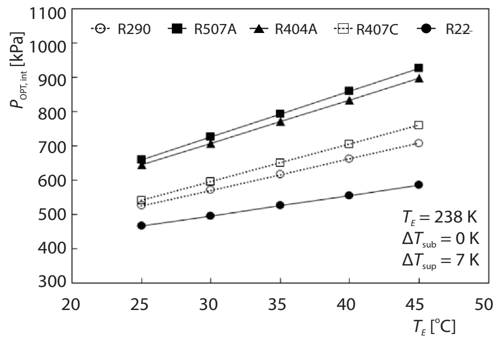

Figure 4. The $P_{OPT,int}$ values *vs.* T_C

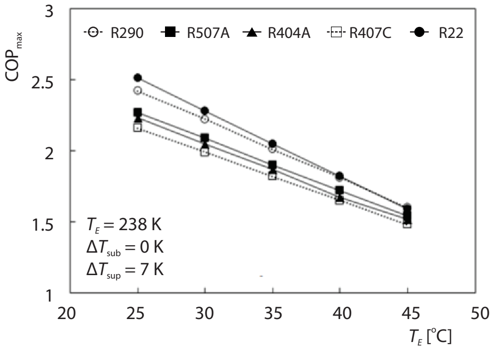

Figure 5. Effect of T_C on COP_{max}

values between 25 °C and 45 °C. Figure 5 reveals that increasing the T_C reduces the COP_{max} values for all studied refrigerants. In LTA, the COP_{max} values of R290 are higher than the COP_{max} values of R404A, R407C, and R507A. For HFC refrigerants, the predicted COP_{max} values are approximately same from 25 °C to 45 °C. The R290 and R22 have nearly same COP_{max} values between 35 °C and 45 °C. On the other hand, R290 has the predicted COP_{max} values 2.42 and 1.60 when the condensation temperatures are 25 °C and 45 °C, respectively.

Effect of T_E and T_C on the total exergy loss

In fig. 6, the system exergy loss is computed from –40 °C to –20 °C for LTA. During these computations, the T_C is kept constant at 40 °C for all refrigerants. Increasing T_E reduces the total exergy loss of the system for all investigated refrigerants. In LTA, it is found that R22 system has the lowest total exergy loss whereas R404A system has the highest total exergy loss when the T_E varies from –40 °C to –20 °C. Total exergy loss value of R290 system is located within the values of R404A and R407C systems for the T_E between –35 °C and –20 °C.

In fig. 7, the effect of T_C on the system's total exergy loss is calculated from 25-45 °C for all refrigerants. The evaporation temperatures are kept constant at –35 °C for LTA's refrigerants. Increasing T_C increases the total exergy loss of system. For LTA, R22's total exergy loss is predicted to be lowest while R404A's total exergy loss is predicted to be highest.

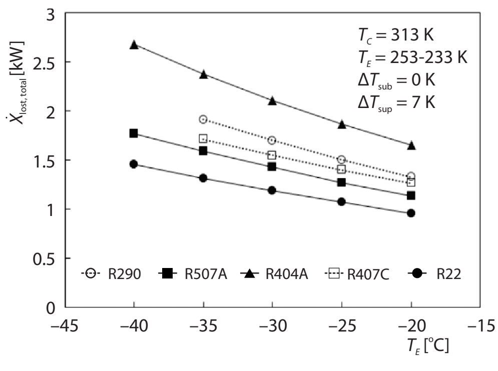

Figure 6. Effect of T_E total exergy loss

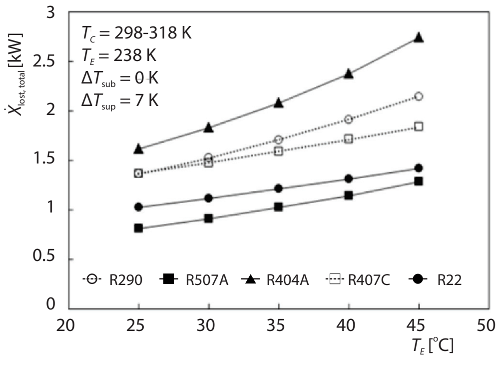

Figure 7. Effect of T_C total exergy loss

Effect of $P_{OPT,int}$ on maximized COP and total exergy loss

The influence of the $P_{OPT,int}$ on the COP_{max} and the system total exergy loss is investigated and compared for different refrigerants. In fig. 8, the COP_{max} values of the system increase for different refrigerants when the $P_{OPT,int}$ rises. In the same $P_{OPT,int}$ intervals of fig. 9, the system total exergy loss is diminished for different refrigerants. The decrease in the total exergy loss of the system implicitly yields the increase in the system COP_{max}.

In LTA, the increase in $P_{OPT,int}$ value changes the system COP_{max} for all systems. When this increase is approximately same (around 175 kPa) for both R404A and R507A, the system COP_{max} increases from 1.49-2.25 and from 1.53-2.30, respectively. It is found that the increase in $P_{OPT,int}$ value (around 100 kPa) causes the increase of system COP_{max} from 1.81-2.50 for R290. It is also realized that the increase in $P_{OPT,int}$ value (about 150 kPa) causes the increase of system COP_{max} from 1.82-2.61 for R22. As a result, it is concluded that the lowest increase of $P_{OPT,int}$ value provides the highest increase of COP_{max} value for R290.

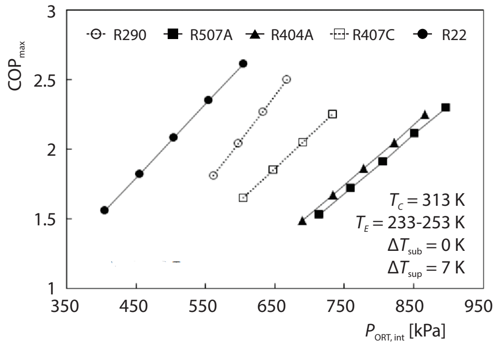

Figure 8. Effect of $P_{OPT,int}$ on COP_{max}

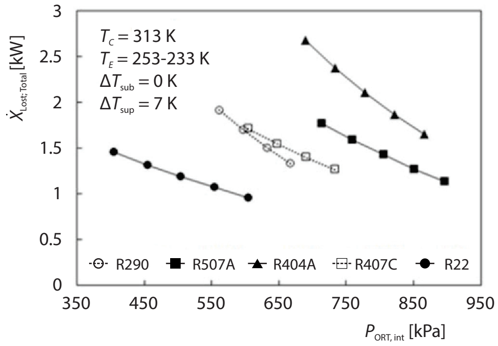

Figure 9. Effect of $P_{OPT,int}$ total exergy loss

Results of thermodynamic analysis

Table 5 summarizes the thermodynamic analysis results of the refrigeration system with flash intercooling. In the computations, the T_E was chosen to be –35 °C for LTA and T_C was set to 40 °C.

Table 5. Summary of the thermodynamic analysis for various refrigerants

Refrigerants		Operating Parameters			Mass ratio ($\dot{m}_H/\dot{m}_L$)	W_{total} [kW]	$\dot{X}_{lost,min}$ [kW]	$\eta_{II,max}$ [%]	COP_{max}
		T_E [°C]	T_C [°C]	$P_{OPT,int}$ [kPa]					
LTA	R22	–35	40	455	1.48	3.30	1.32	57.28	1.82
	R290	**–35**	**40**	**562**	**1.50**	**3.31**	**1.42**	**57.12**	**1.81**
	R407C	–35	40	605	1.59	3.64	1.71	53.91	1.65
	R404A	–35	40	734	1.65	3.58	1.63	54.58	1.67
	R507A	–35	40	760	1.66	3.53	1.59	54.94	1.72

Given T_E and T_C are –35 °C and 40 °C for R290, the computed $P_{OPT,int}$ was 562 kPa. Torrella *et al.* [1] determined experimentally that the increase of inter-stage pressure is a consequence of the increase of refrigerant mass-flow rates ratio, $\dot{m}_H/\dot{m}_L$. In this study, calculations predicted that the increase in the $\dot{m}_H/\dot{m}_L$ increases the $P_{OPT,int}$ for all studied refrigerants. In LTA, the calculated COP_{max} and $\eta_{II,max}$ values of R290 system are better than the values of R507A, R404A, and R407C systems. For R290 and R22, the predicted COP_{max}, $\eta_{II,max}$ values are nearly same. Its COP_{max} and $\eta_{II,max}$ values are 1.81 and 57.12%, respectively.

Table 6 compares some important system parameters such as COP_{max}, $\eta_{II,max}$, total exergy lost and total work input of system within each other for the LTA. We tested using R290 instead of R404A, R507A, R407C, and R22 in LTA. The diferences between the various refrigerant alternatives are calculated for all the former parameters in both applications. If the difference is positive for the tested refrigerants worsen the performance.

Table 6. Comparison of parameters between the systems using natural and synthetic refrigerants

Parameter	R290	Alternatives				Difference [%]			
		R404A	R507A	R407C	R22	R404A	R507A	R407C	R22
COP_{max}	**1.81**	1.67	1.72	1.65	1.82	–7.73	–4.97	–8.84	0.55
η_{IImax} [%]	**57.12**	54.58	54.94	53.91	57.28	–4.45	–3.82	–5.62	0.28
$\dot{X}_{lost,min}$ [kW]	**1.42**	1.63	1.59	1.71	1.32	14.79	11.97	20.42	–7.0
W_{total} [kW]	**3.31**	3.58	3.53	3.64	3.30	8.48	6.97	10.30	-0.30

In LTA, it is revealed that R290 system has better performance than the R404A, R507A, and R407C systems in terms of COP_{max} and GWP. The R404A, R507A, and R407C systems' COP_{max} are lower than the R290 system's COP_{max} by 7.73%, 4.97% and 8.84%, respectively.

The R404A and R407C systems have the approximately same COP_{max} values (1.67 and 1.65). On the other hand, R404A system has the highest GWP (3922) and R407C system has the lowest GWP (1600). Although R507A system has similar COP_{max} and GWP values, it slightly outperforms R404A and it is still worse than R290. Hence, R404A, R407C, and R507A are the bad options among the considered LTA refrigerants in terms of the COP_{max} and GWP. The R22 system is slightly more efficient (0.55%) than the R290 system. However, due to the ongoing global phase out of R22, it is needed to be replaced by alternative refrigerants. R404A, R507A, and R22 can be utilized for the evaporation temperatures down to –40 °C, unlike R290 and R407C that could be utilized down to –35 °C. R290 is also in A3 safety class while R404A, R407C, R507A, and R22 are in A1 safety class. The biggest advantage of R290 system over the other four alternatives is that it has negligible GWP (11), high COP_{max} (1.81) and 0 of ODP.

Conclusions

Different refrigerants are analyzed and compared theoretically in EES based on the First and Second laws of thermodynamics for a refrigeration system with flash intercooling. The theoretical modelling is done by optimizing the P_{int} at given T_E and T_C values for all LTA's refrigerants (R290, R404A, R507A, R407C, and R22). After optimization, the maximized values of COP and η_{II} are computed from the predicted values of $P_{OPT,int}$. The influence of T_E and T_C on the system $P_{OPT,int}$, COP_{max}, $\eta_{II,max}$ is evaluated for LTA's refrigerants. From the results, it can be concluded as follow.

- For all systems, the increase in $P_{OPT,int}$ value causes the increase of system COP_{max}. It is also realized that the lowest increase of $P_{OPT,int}$ value provides the highest increase of COP_{max} value for R290. In LTA, R507A system has the highest $P_{OPT'int}$ values and R22 system has the lowest $P_{OPT'int}$ values at their respective T_C and T_E values.
- Decreasing T_E reduces the COP_{max} of the system for all investigated refrigerants. However, decreasing T_E increases the total exergy loss of the system for all studied refrigerants. In LTA, all systems have similar change trend in COP_{max} within their respective T_E interval.

- The COP_{max} values decrease with increasing T_C for selected systems. However, increasing T_C increases the total exergy loss of system for all refrigerants. In LTA, R22, and R407C have the highest and lowest COP_{max} values from 25-45 °C. On the other hand, R290, and R22 have nearly same COP_{max} values between 35 °C and 45 °C.

In addition, the thermodynamic analysis includes the comparison of LTA's refrigerants with respect to the significant system parameters such as COP_{max}, $\eta_{II,max}$ and total exergy loss of system. In LTA, it is revealed that R290 (HC) system has better performance than the R404A, R407C and R507A (HFC) systems in terms of COP_{max} (1.81), GWP (11), and ODP (0). The R404A, R407C, and R507A systems COP_{max} are lower than the R290 system's COP_{max} by 7.73%, 8.84%, and 4.97%, respectively. The R22 (HCFC) system is slightly more efficient (0.55%) than the R290 system. However, due to the ongoing global phase out of R22, it is needed to be replaced by alternative refrigerants in LTA.

Nomenclature

h – specific enthalpy, [kJkg^{-1}]
$\dot{m}$ – mass-flow rate, [kgs^{-1}]
P – pressure, [kPa or bar]
P_{CR} – critical pressure, [MPa]
$\dot{Q}$ – heat transfer rate, [kW]
r – mass-flow rate ratio, [–]
s – specific entropy, [kJkg^{-1}K^{-1}]
T – temperature, [°C or K]
T_{CR} – critical temperature, [°C]
ΔT – temperature difference, [–]
$\dot{W}$ – power, [kW]
$\dot{X}$ – rate of exergy loss, [kW]

Greek Letters

η – efficiency, [%]
η_{II} – second law efficiency, [%]

Subscripts

Act – actual
C – condensation
CR – critical
cond – condenser
E – evaporation
EXP – experimental
EV – expansion valve
evap – evaporator
FC – flash intercooler
HPC – high pressure compressor
int – intermediate
LPC – low pressure compressor
max – maximum
0 – environment
OPT – optimum
Rev – reversible
sub – subcooling
sup – superheating

Acronyms

COP – coefficient of performance, [–]
GWP – global warming potential
HP – high pressure
HFC – hydrofluorocarbon
HCFC – hydrochlorofluorocarbons
LP – low pressure
NBR – normal boiling point, [°C]
ODP – ozone depletion potential

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