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

Ebru MANCUHAN

Marmara University, Goztepe Kampusu 34722, Kadikoy, İstanbul/Turkey
E-mail: emancuhan@marmara.edu.tr

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 (LTAs) through a theoretical analysis according to the different refrigerants (R290, R404A, R407C, R50A and R22). The theoretical modeling 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 LTAs. 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 ODP and GWP considerations. Therefore, it is important to evaluate the R22 replacement options. R290 was discovered to have better performance than the R404A, R407C and R507A systems in terms of COP_{max} (1.81), GWP (11) and ODP (0) when T_E and T_C are -35^{o}C and 40^{o}C.

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

1. 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 (LTAs). 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].

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 LTAs. Lee et al. [9] evaluated theoretically the optimal condensing temperature of the cascade heat exchanger and the COP for evaporating levels between -45^oC and -55^oC. 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^oC to -30^oC. 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}. Kılıçaraslan et al. [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 and Motta [16] studied three alternatives such as HFCs (R404A, R410A) and HC (R290) to replace R22 for LTAs. As a result, R410A was discovered to be an efficient and environmentally acceptable option. Llopis 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. Modeling 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 $P_{OPT,int}$ were important to improve the performance of two-stage flash intercooler system. However, the impact of some refrigerants, which have low GWP, on the $P_{OPT,int}$ and COP_{max} values weren't sufficiently analyzed. This work will investigate the effect of low GWP refrigerants on the $P_{OPT,int}$ and COP_{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 LTAs. In LTAs (-20^{o}C to -40^{o}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 LTAs, the theoretical models suggest three correlations that are function of T_E and T_C for all refrigerants. The first one determines the $P_{OPT,int}$ and the second one determines the COP_{max} corresponding to the $P_{OPT,int}$. The third one determines the $\eta_{II,max}$ corresponding to the $P_{OPT,int}$. Therefore, the efect of T_E and T_C on the system $P_{OPT,int}$, COP_{max}, $\eta_{II,max}$ is 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.

2. Background

2.1. Properties of selected refrigerants

Natural refrigerants are increasingly used in LTAs. 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. ODP values, GWP values and the physical properties of refrigerants that are subject to this study are given in Table 1.

Table 1. The physical and environmental properties of refrigerants used in this study [22]

Refrigerants		Refrigerant's classification	Molecular weight (kg/kmol)	T_{CR} (°C)	P_{CR} (MPa)	NBP (°C)	ODP	GWP	Safety class
LTAs	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

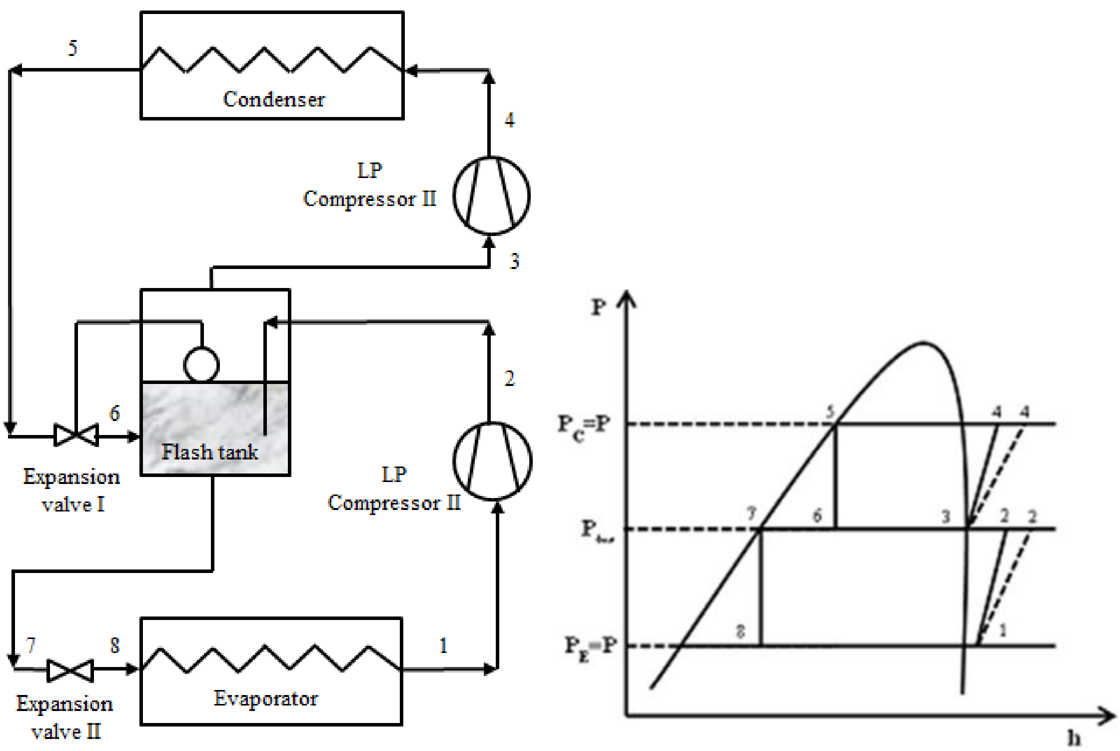
ODP and GWP values show that the correct selection of environment-friendly refrigerant is essential to reduce their harmful effects on the environment. R22 contains chlorine and it is considered one of the worst refrigerants which deplete the ozone layer. Major advantage of R507A, R404A, R407C and R22 is that they fall in A1 safety class. They are nonflammable refrigerants although they have high GWP. R290, on the other hand, has negligible GWP and high flammability which puts this natural refrigerant into the safety rating A3 [21]. 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 LTAs.

2.2. 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 Figure 1(a). At the exit of the condenser (state 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 Figure 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_{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.



(a) **(b)**

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

2.3. 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 [19]. In this section, the equations are developed for the analysis considering the state points of Figures 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,
- the evaporator cooling capacity of the system ($\dot{Q}_{Evap}$) is 6 kW.

Thermodynamic analysis is carried out using EES software [20]. 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.

2.3.1. Mass and energy analysis

The cooling capacity of the LP evaporator is defined as:

$$\dot{Q}_{Evap} = \dot{m}_{Evap}(h_1 - h_8) \tag{1}$$

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

$$\dot{m}_{Evap} = \frac{\dot{Q}_{Evap}}{(h_1 - h_8)} \tag{2}$$

Compressor power consumption for LP and HP compressors are:

$$\dot{W}_{LPC} = \dot{m}_{Evap}(h_2 - h_1) \tag{3}$$

$$\dot{W}_{HPC} = \dot{m}_{Cond}(h_4 - h_3) \tag{4}$$

Energy balance for the HP expansion valve I is:

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

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

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

Energy balance for the LP expansion valve II is:

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

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

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:

$$\dot{m}_{Cond}\ h_6 + \dot{m}_{Evap}\ h_2 = \dot{m}_{Evap} h_7 + \dot{m}_{Cond}\ h_3 \tag{9}$$

The mass flow rate ratio is determined by:

$$r = \frac{\dot{m}_{Cond}}{\dot{m}_{Evap}} = \frac{(h_2 - h_7)}{(h_3 - h_6)} \tag{10}$$

The heat transfer rate into the flash intercooler is determined as:

$$\dot{Q}_{FC} = \dot{m}_{Cond}\ (h_6 - h_3) = \dot{m}_{Evap}(h_2 - h_7) \tag{11}$$

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

$$\dot{m}_{Cond} = \dot{m}_{Evap} \frac{(h_2 - h_7)}{(h_3 - h_6)} \tag{12}$$

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

$$\dot{Q}_{Cond} = \dot{m}_{Cond}(h_4 - h_5) \tag{13}$$

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

$$COP = \frac{\dot{Q}_{Evap}}{\dot{W}_{LPC} + \dot{W}_{HPC}} = \frac{\dot{m}_{Evap}(h_1 - h_8)}{\dot{m}_{Evap}(h_2 - h_1) + \dot{m}_{Cond}(h_4 - h_3)} \tag{14}$$

2.3.2. Exergy analysis

The exergy loss rate for the LP cycle evaporator is:

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

The exergy loss rate of the LP and HP compressors are:

$$\dot{X}_{LPC} = \dot{m}_{Evap} T_o(s_2 - s_1) + \dot{W}_{LPC} \tag{16}$$

$$\dot{X}_{HPC} = \dot{m}_{Cond} T_o(s_4 - h_3) + \dot{W}_{HPC} \tag{17}$$

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

The exergy loss rate for the condenser of HP cycle is:

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

The exergy loss rates for expansion valves I and II are:

$$\dot{X}_{\mathrm{EV}_1} = \dot{m}_{Cond}[T_o(s_5 - s_6)] \tag{20}$$

$$\dot{\mathrm{X}}_{\mathrm{EV}_2} = \dot{m}_{Evap}[T_o(s_7 - s_8)] \tag{21}$$

$$\dot{X}_{Total,EV} = \dot{X}_{EV_1} + \dot{X}_{EV_2} \tag{22}$$

The exergy loss rate in the flash intercooler is:

$$\dot{X}_{FC} = \dot{m}_{Evap}[(h_2 - h_7) - T_o(s_2 - s_7)] + \dot{m}_{Cond}[(h_6 - h_3) - T_o(s_6 - s_3)] \tag{23}$$

Total exergy loss rate of the system is:

$$\dot{X}_{Lost,Total} = \dot{X}_{Evap} + \dot{X}_{Cond} + \dot{X}_{Total,EV} + \dot{X}_{Total,C} + \dot{X}_{FC} \tag{24}$$

The second law efficiency is used to measure the system performance in Eqs. (25) and (26).

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

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

In Eq. (26) $\dot{W}_{Rev}$ is the reversible power. T_0 and T_E are the dead and evaporation temperatures respectively.

3. 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_C/P_E and T_C/T_E. The COP and η_{II} values are maximized by various $P_{OPT,int}$ 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 as shown below.

Maximize COP (T_C, T_E) and η_{II} ((T_C, T_E)

Subject to:

For all refrigerants:

$$25℃ \leq T_C \leq 45℃ \quad \text{and} \quad P_{sat@T_{C=25℃}} \leq P_C \leq P_{sat@T_{C=45℃}} \tag{27}$$

For R290 and R407C:

$$-35℃ \leq T_E \leq -20℃ \quad \text{and} \quad P_{sat@T_{E=-35℃}} \leq P_E \leq P_{sat@T_{E=-20℃}} \tag{28}$$

For R404A, R507A and R22:

$$-40℃ \leq T_E \leq -20℃ \quad \text{and} \quad P_{sat@T_{E=-40℃}} \leq P_E \leq P_{sat@T_{E=-20℃}} \tag{29}$$

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.

4. 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 $P_{OPT'int}$ 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 ^oC to 45 ^oC for all refrigerants. Concerning the conditioned space, the T_E values are considered in general to be between -20 ^oC and -40 ^oC. In this temperature range, the lowest temperature that R404A R507 and R22 can reach is -40 ^oC whereas the lowest temperature that R290 and R407C can reach is -35 ^oC. As seen in the Figures 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 LTAs' 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.

4.1 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 Table 2. The unit used in the calculations is Kelvin (K).

Table 2. The correlations obtained for different refrigerants

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

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 LTAs' refrigerants. The reference point of T_E to compare different refrigerants is set to be -35 °C in LTAs.

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 Table 3.

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

T_C (K)	T_E (K)	R290			R290 [23]	R404A			R507A		
		$P_{OPT,int}$ (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

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 ^oC to -7 ^oC increases the COP values. In addition, increasing the T_C from 42 ^oC to 45 ^oC reduces the COP values of R22. In Table 4, when T_E and T_C are -20 ^oC and 45 ^oC, the R22's predicted COP and the R22's experimental COP are 2.38 and 2.1 respectively.

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

T_C (K)	T_E (K)	R407C			R22			$R22_{EXP}$ [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	-

4.2. Effects of operation parameters

4.2.1. 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 LTAs, 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 ^oC to -20 ^oC. 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 LTAs, 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 Figure 3, decreasing the T_E in LTAs from -20 ^oC to -40 ^oC decreases the COP_{max} values of refrigerants (R290, R404A, R407C, R507A and R22). R290 and R22 have the highest COP_{max} values among all five refrigerants at -35 ^oC. In addition, R290 and R22 have approximately same COP_{max} values at -35 ^oC. The COP values with refrigerant of R290 are higher than with the other HFC's refrigerants between -35 ^oC and -20 ^oC. R404A, R507A and R22 are the only refrigerant options that could be used below -35 ^oC in LTAs.

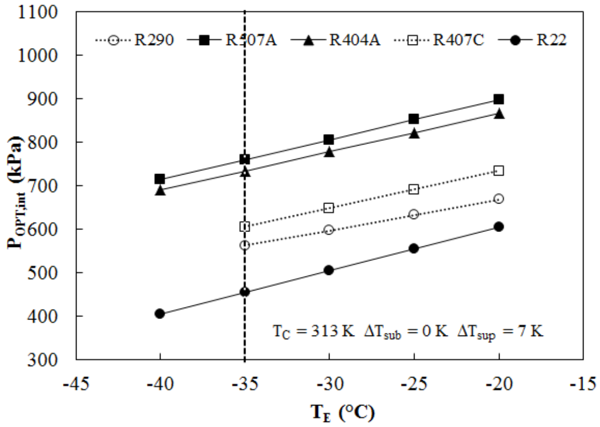


Figure 2. $P_{OPT,int}$ values versus T_E

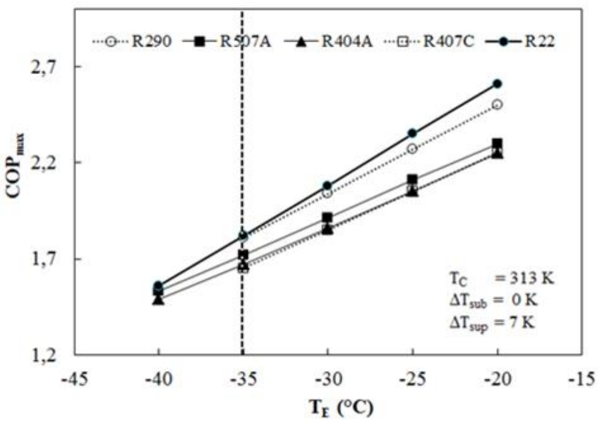


Figure 3. Effect of T_E on COP_{max}

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

In Figures 4 and 5, the variation of $P_{OPT,int}$ and COP_{max} in function of the T_C is observed from 25 ^oC to 45 ^oC. In the previous section, we determined the T_E to be -35 ^oC 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 LTAs. It is seen that increasing the T_C raises the $P_{OPT,int}$ for all the refrigerants.

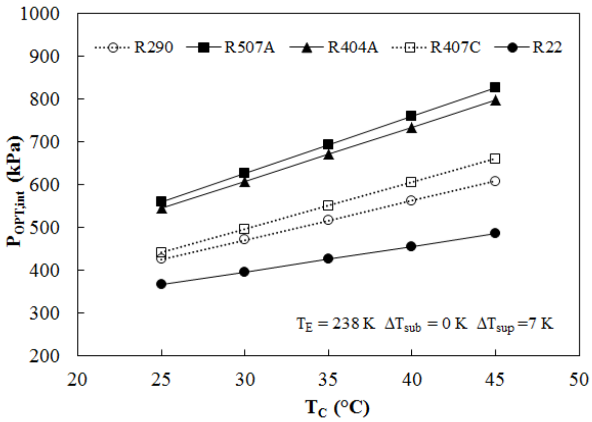


Figure 4. $P_{OPT,int}$ values versus T_C

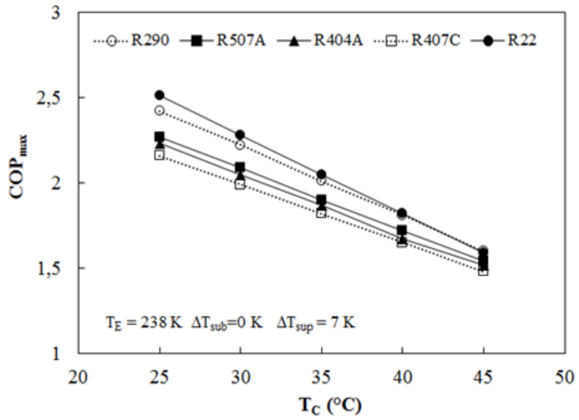


Figure 5. Effect of T_C on COP_{max}

In LTAs, 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. $P_{OPT,int}$ values of R290 are located within the values of R407C and R22. R507A and R404A have approximately same $P_{OPT,int}$ 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 LTAs, the COP_{max} values of R290 are higher than the COP_{max} values of R404A, R407C and R507A. For HFC's refrigerants, the predicted COP_{max} values are approximately same from 25 °C to 45 °C. 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.

4.2.3. Effect of T_E and T_C on the total exergy loss

In Figure 6, the system exergy loss is computed from -40°C to -20°C for LTAs. 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 LTAs, 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 Figure 7, the effect of T_C on the system's total exergy loss is calculated from 25 °C to 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 LTAs, R22's total exergy loss is predicted to be lowest while R404A's total exergy loss is predicted to be highest.

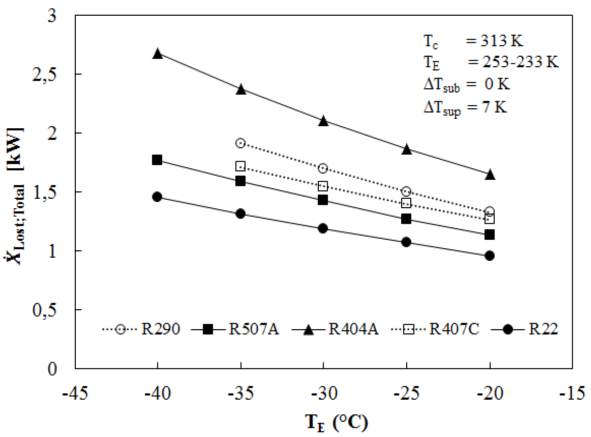


Figure 6. Effect of T_E on total exergy loss

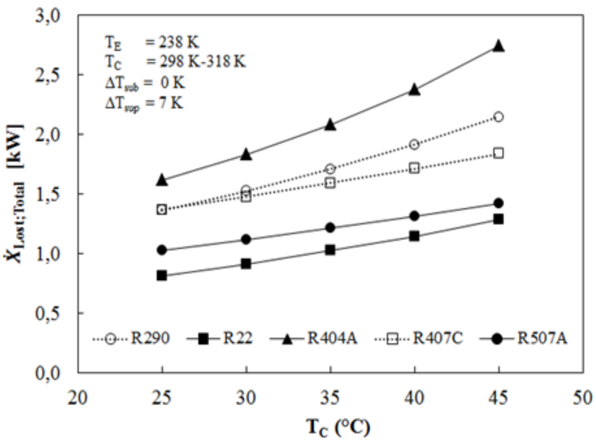


Figure 7. Effect of T_C on total exergy loss

4.2.4. 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 Figure 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 Figure 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 LTAs, the increase in $P_{OPT,int}$ value changes the system COP_{max} for all systems. When this increase is approximately same (around 175kPa) for both R404A and R507A, the system COP_{max} increases from 1.49 to 2.25 and from 1.53 to 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 to 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 to 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.

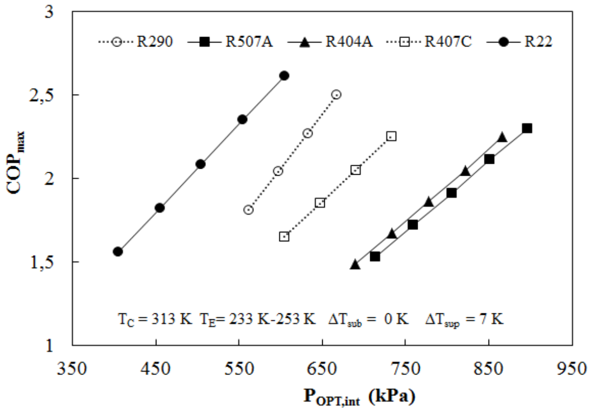


Figure 8. Effect of $P_{OPT,int}$ on COP_{max}

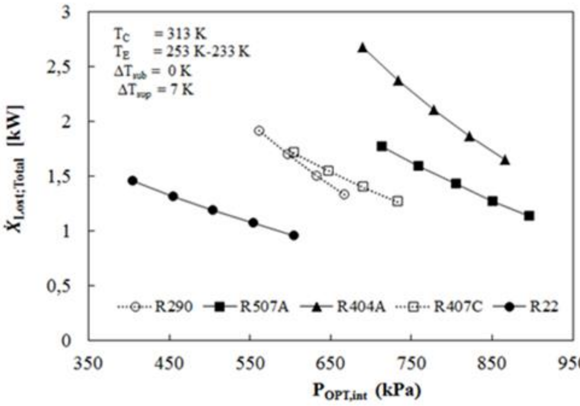


Figure 9. Effect of $P_{OPT,int}$ on total exergy loss

4.3 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 ^oC for LTAs and T_C was set to 40 ^oC. Given T_E and T_C are -35^oC and 40^oC 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 LTAs, 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 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 (^oC)	T_C (^oC)	$P_{OPT,int}$ (kPa)					
LTAs	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

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's. We tested using R290 instead of R404A, R507A, R407C and R22 in LTAs. The differences between the various refrigerant alternatives are calculated for all the former parameters in both applications. If the difference is positive for the tested refrigerants, it suggests an improvement. The negative difference on the other hand means the tested refrigerants worsen the performance.

In LTAs, it is revealed that R290 system has better performance than the R404A, R507A and R407C systems in terms of COP_{max} and GWP. 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 ^{o}C, unlike R290 and R407C that could be utilized down to -35 ^{o}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.

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
$\eta_{II_{max}}$(%)	**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

5. 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 modeling 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 that:

- 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 LTAs, 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 LTAs, 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 "LTAs, R22 and R407C have the highest and lowest COP_{max} values from 25 ^{o}C to 45 ^{o}C. On the other hand, R290 and R22 have nearly same COP_{max} values between 35 ^{o}C and 45 ^{o}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 LTAs, 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). 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 LTAs.

Nomenclature

Abbreviations

COP	coefficient of performance [-]
GWP	global warming potential
HP	high pressure
HFC	hydrofluorocarbon
HCFC	hydrochlorofluorocarbons
LP	low pressure

Latin Letters

h	specific enthalpy [kJ/kg]
$\dot{m}$	mass flow rate [kg/s]
P	pressure [kPa or bar]
$\dot{Q}$	heat transfer rate [kW]
r	mass flow rate ratio [-]
s	specific entropy [kJ/kgK]
T	temperature [°C or K]
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
Cond	condenser
Comp	compressor
E	evaporation
EV	expansion valve
Evap	evaporator
FC	flash intercooler
HPC	high pressure compressor
int	intermediate

LPC low pressure compressor
max maximum
min minimum
0 environment
OPT optimum
Rev reversible
Sub subcooling
Sup superheating

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