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

by

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Original scientific paper
https://doi.org/10.2298/TSCI180921011M

The aim of this study is to investigate the effect of low global warming potential refrigerants on the optimum intermediate pressure ($P_{\text{OPT,int}}$) 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 $P_{\text{OPT,int}}$. The linear regression method is then used to derive three correlations of $P_{\text{OPT,int}}$ with the maximum values of COP and Second law efficiency according to $T_E$ and $T_C$. Hence, the $P_{\text{OPT,int}}$ 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 $P_{\text{OPT,int}}$ in low temperature applications. The R507A system has the highest $P_{\text{OPT,int}}$ values and R22 system has the lowest $P_{\text{OPT,int}}$ 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 (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, $P_{\text{int}}$, corresponding to the minimum compressor work. Therefore, it is important to determine the optimum intermediate pressure, $P_{\text{OPT,int}}$, 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 $\eta_{II}$. Kilicaslan 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 $P_{\text{OPT,int}}$ were important to improve the performance of two-stage flash intercooler system. However, the impact of some refrigerants, which have low GWP, on the $P_{\text{OPT,int}}$ and COP$_{\text{max}}$ values were not sufficiently analyzed. This work will investigate the effect of low GWP refrigerants on the $P_{\text{OPT,int}}$ and COP$_{\text{max}}$ 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 $T_{C}$ for all refrigerants. The first one determines the $P_{\text{OPT,int}}$ and the second one determines the COP$_{\text{max}}$ corresponding to the $P_{\text{OPT,int}}$. The third one determines the $\eta_{\text{II,max}}$ corresponding to the $P_{\text{OPT,int}}$. Therefore, the effects of $T_{E}$ and $T_{C}$ on the system $P_{\text{OPT,int}}$, COP$_{\text{max}}$, $\eta_{\text{II,max}}$ 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>
<thead>
<tr>
<th>Refrigerants</th>
<th>Refrigerant’s classification</th>
<th>Molecular weight [kg·mol⁻¹]</th>
<th>$T_{CR}$ [°C]</th>
<th>$P_{CR}$ [MPa]</th>
<th>NBP [°C]</th>
<th>ODP</th>
<th>GWP [g·mol⁻¹·yr⁻¹]</th>
<th>Safety class</th>
</tr>
</thead>
<tbody>
<tr>
<td>LTA</td>
<td></td>
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<td></td>
<td></td>
<td></td>
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<td></td>
</tr>
<tr>
<td>R404A</td>
<td>HFC</td>
<td>97.60</td>
<td>72.12</td>
<td>3.765</td>
<td>–46.5</td>
<td>0</td>
<td>3921</td>
<td>A1</td>
</tr>
<tr>
<td>R507A</td>
<td>HFC</td>
<td>98.60</td>
<td>70.5</td>
<td>3.70</td>
<td>–46.7</td>
<td>0</td>
<td>3985</td>
<td>A1</td>
</tr>
<tr>
<td>R407C</td>
<td>HFC</td>
<td>86.20</td>
<td>86.11</td>
<td>4.63</td>
<td>–42.0</td>
<td>0</td>
<td>1600</td>
<td>A1</td>
</tr>
<tr>
<td>R22</td>
<td>HCFC</td>
<td>86.47</td>
<td>96.14</td>
<td>4.99</td>
<td>–40.81</td>
<td>0.05</td>
<td>1810</td>
<td>A1</td>
</tr>
<tr>
<td>R290</td>
<td>HC</td>
<td>44.09</td>
<td>134.6</td>
<td>4.23</td>
<td>–42.09</td>
<td>0</td>
<td>11</td>
<td>A3</td>
</tr>
</tbody>
</table>

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 A1 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 A3 [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 \( P_{\text{int}} \) 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 \( (P_{E} = P_{1} = P_{5}) \), 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)](image)

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 \( \text{COP}_{\text{max}} \) and the \( \eta_{\text{Imax}} \) 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, $\Delta T_{sup}$, is 7 °C,

– saturated liquid state at the exit of condenser, $\Delta T_{sub}$, is 0 °C, and

– the evaporator cooling capacity of the system, $Q_{evap}$, 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:

$$\dot{Q}_{evap} = \dot{m}_{evap} (h_1 - h_8)$$

(1)

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

$$\dot{m}_{evap} = \frac{\dot{Q}_{evap}}{h_1 - h_8}$$

(2)

Compressor power consumption for LP and HP compressors:

$$W_{LPC} = \dot{m}_{evap} (h_2 - h_1)$$

(3)

$$W_{HPC} = \dot{m}_{cond} (h_4 - h_3)$$

(4)

Energy balance for the HP expansion valve I:

$$h_5 = h (P = P_{crit}, x = 0)$$

(5)

$$h_3 = h_6$$

(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:

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

(7)

$$h_7 = h_8$$

(8)

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

$$\dot{m}_{cond} h_6 + \dot{m}_{evap} h_2 = \dot{m}_{evap} h_7 + \dot{m}_{cond} h_3$$

(9)

The mass-flow rate ratio is determined:

$$r = \frac{\dot{m}_{cond}}{\dot{m}_{evap}} = \frac{(h_2 - h_7)}{(h_3 - h_6)}$$

(10)
The heat transfer rate into the flash intercooler is determined:
\[
\dot{Q}_{FC} = \dot{m}_{\text{cond}}(h_0 - h_5) = \dot{m}_{\text{evap}}(h_2 - h_7)
\] (11)

The refrigerant mass-flow rate of HP cycle is defined:
\[
\dot{m}_{\text{cond}} = \dot{m}_{\text{evap}} \frac{(h_2 - h_7)}{(h_5 - h_6)}
\] (12)

The heat transfer rate of the HP cycle condenser is defined:
\[
\dot{Q}_{\text{cond}} = \dot{m}_{\text{cond}}(h_4 - h_5)
\] (13)

And finally, the overall COP of the system with flash intercooling is determined:
\[
\text{COP} = \frac{\dot{Q}_{\text{evap}}}{W_{\text{LPC}} + W_{\text{HPC}}} = \frac{\dot{m}_{\text{evap}}(h_1 - h_8)}{\dot{m}_{\text{evap}}(h_2 - h_7) + \dot{m}_{\text{cond}}(h_4 - h_5)}
\] (14)

**Exergy analysis**

The exergy loss rate for the LP cycle evaporator:
\[
\dot{X}_{\text{evap}} = \dot{m}_{\text{evap}} \left[ (h_1 - h_8) + T_o (s_1 - s_8) \right] + \dot{Q}_{\text{evap}} \left( \frac{T_o}{T_E} \right)
\] (15)

The exergy loss rate of the LP and HP compressors:
\[
\dot{X}_{\text{LPC}} = \dot{m}_{\text{evap}} T_0 (s_2 - s_1) + W_{\text{LPC}}
\] (16)

\[
\dot{X}_{\text{HPC}} = \dot{m}_{\text{cond}} T_0 (s_4 - h_3) + W_{\text{HPC}}
\] (17)

\[
\dot{X}_{\text{total,C}} = \dot{X}_{\text{LPC}} + \dot{X}_{\text{HPC}}
\] (18)

The exergy loss rate for the condenser of HP cycle:
\[
\dot{X}_{\text{cond}} = \dot{m}_{\text{cond}} \left[ (h_4 - h_5) + T_0 (s_4 - s_5) \right] - \dot{Q}_{\text{cond}} \left( \frac{T_o}{T_C} \right)
\] (19)

The exergy loss rates for expansion valves I and II:
\[
\dot{X}_{\text{EV}_1} = \dot{m}_{\text{cond}} \left[ T_0 (s_3 - s_6) \right]
\] (20)

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

\[
\dot{X}_{\text{total,Ev}} = \dot{X}_{\text{EV}_1} + \dot{X}_{\text{EV}_2}
\] (22)

The exergy loss rate in the flash intercooler:
\[
\dot{X}_{\text{FC}} = \dot{m}_{\text{evap}} \left[ (h_2 - h_7) - T_0 (s_2 - s_7) \right] + \dot{m}_{\text{cond}} \left[ (h_6 - h_3) - T_0 (s_6 - s_3) \right]
\] (23)

Total exergy loss rate of the system:
\[
\dot{X}_{\text{lost, total}} = \dot{X}_{\text{evap}} + \dot{X}_{\text{cond}} + \dot{X}_{\text{total,Ev}} + \dot{X}_{\text{total,C}} + \dot{X}_{\text{FC}}
\] (24)
The Second law efficiency is used to measure the system performance in eqs. (25) and (26):
\[ \eta_{II} = \frac{W_{REV}}{W_{Act}} \]
(25)
\[ W_{REV} = Q_{evap} \left( \frac{T_{th}}{T_{evap}} - 1 \right) \]
(26)

In eq. (26) is the reversible power. The \( T_{th} \) and \( T_{evap} \) 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 \( P_{C}/P_{E} \) and \( T_{C}/T_{E} \). The COP and \( \eta_{II} \) 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.

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

- for all refrigerants
  \[ 25^\circ C \leq T_{C} \leq 45^\circ C \] and \( P_{sat at T_{C}=25^\circ C} \leq P_{C} \leq P_{sat at T_{C}=45^\circ C} \)
  (27)

- for R290 and R407C
  \[ -35^\circ C \leq T_{E} \leq -20^\circ C \] and \( P_{sat at T_{E}=-35^\circ C} \leq P_{E} \leq P_{sat at T_{E}=-20^\circ C} \)
  (28)

- for R404A, R507A, and R22
  \[ -40^\circ C \leq T_{E} \leq -20^\circ C \] and \( P_{sat at T_{E}=-40^\circ C} \leq P_{E} \leq P_{sat at T_{E}=-20^\circ C} \)
  (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 \( \eta_{II} \) 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 25-45 °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 cho-
Mancuhan, E.: Comparative Evaluation of a Two-Stage Refrigeration System ...  
THERMAL SCIENCE: Year 2020, Vol. 24, No. 2A, pp. 815-830

sen 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 \( P_{\text{opt, int}} \) is calculated from the given \( T_E \) and \( T_C \). Then, the \( \text{COP}_{\text{max}} \) and \( \eta_{\text{II, max}} \) are calculated from the former \( P_{\text{opt, int}} \) 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**

<table>
<thead>
<tr>
<th>Refrigerants of LTA</th>
<th>( P_{\text{opt, int}} )</th>
<th>( \text{COP}_{\text{max}} )</th>
<th>( \eta_{\text{II, max}} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>R290</td>
<td>( P_{\text{opt, int, R290}} = -3970 + 7.0918T_E + 9.0857T_C )</td>
<td>( \text{COP}_{\text{max, R290}} = 3.5739 + 0.0466T_E - 0.0410T_C )</td>
<td>( \eta_{\text{II, max, R290}} = 34.6534 + 0.01472T_E + 0.0606T_C )</td>
</tr>
<tr>
<td>R507A</td>
<td>( P_{\text{opt, int, R507A}} = -5575.2 + 9.102T_E + 13.319T_C )</td>
<td>( \text{COP}_{\text{max, R507A}} = 3.954 + 0.03868T_E - 0.03655T_C )</td>
<td>( \eta_{\text{II, max, R507A}} = 44.143 + 0.038T_E + 0.0056T_C )</td>
</tr>
<tr>
<td>R404A</td>
<td>( P_{\text{opt, int, R404A}} = -5297.7 + 8.7774T_E + 12.596T_C )</td>
<td>( \text{COP}_{\text{max, R404A}} = 3.8127 + 0.0381T_E - 0.0357T_C )</td>
<td>( \eta_{\text{II, max, R404A}} = 35.512 + 0.02216T_E + 0.0441T_C )</td>
</tr>
<tr>
<td>R407C</td>
<td>( P_{\text{opt, int, R407C}} = -4870.4 + 8.5734T_E + 10.9743T_C )</td>
<td>( \text{COP}_{\text{max, R407C}} = 2.8501 + 0.03987T_E - 0.0357T_C )</td>
<td>( \eta_{\text{II, max, R407C}} = 34.510 + 0.0105T_E + 0.054T_C )</td>
</tr>
<tr>
<td>R22</td>
<td>( P_{\text{opt, int, R22}} = -3787.72 + 9.9696T_E + 5.9748T_C )</td>
<td>( \text{COP}_{\text{max, R22}} = 3.7735 + 0.0523T_E - 0.046T_C )</td>
<td>( \eta_{\text{II, max, R22}} = 39.996 + 0.01694T_E + 0.04234T_C )</td>
</tr>
</tbody>
</table>

The proposed correlations are used to predict the \( P_{\text{opt, int}} \), \( \text{COP}_{\text{max}} \), and \( \eta_{\text{II, max}} \) values from different operating parameters \( T_E \) and \( T_C \). 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 $P_\text{OPT,int}$, COP$_\text{max}$, and $\eta_{II,\text{max}}$ for LTA with different refrigerants

<table>
<thead>
<tr>
<th>$T_c$ [K]</th>
<th>$T_r$ [K]</th>
<th>$P_\text{OPT,int}$ [kPa]</th>
<th>COP$_\text{max}$</th>
<th>$\eta_{II,\text{max}}$ [%]</th>
<th>$R290$ [23]</th>
<th>$R404A$</th>
<th>$R507A$</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>$R290$</td>
<td>$R290$ [23]</td>
<td>$R404A$</td>
<td>$R507A$</td>
<td></td>
<td></td>
</tr>
<tr>
<td>318</td>
<td>263</td>
<td>NA</td>
<td>NA</td>
<td>NA</td>
<td>NA</td>
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<td>NA</td>
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<td>NA</td>
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<tr>
<td>253</td>
<td>713.5</td>
<td>2.30</td>
<td>57.65</td>
<td>928.5</td>
<td>2.09</td>
<td>55.14</td>
<td>963.1</td>
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<td>248</td>
<td>678.0</td>
<td>2.07</td>
<td>57.57</td>
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Mancuhan, E.: Comparative Evaluation of a Two-Stage Refrigeration System...

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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 $P_{\text{OPT, int}}$ 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$_{\text{max}}$ and $P_{\text{OPT, int}}$ is examined for different refrigerants of system. Figure 2 depicts the effect of $T_E$ on the calculated $P_{\text{OPT, int}}$ values which maximize the COP. It is seen that decreasing the $T_E$ decreases the $P_{\text{OPT, int}}$ for all the refrigerants. For LTA, the largest $P_{\text{OPT, int}}$ values are observed when R507A is used as refrigerant whereas the lowest $P_{\text{OPT, int}}$ values are found when R22 is used. The $P_{\text{OPT, int}}$ 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$_{\text{max}}$ values of refrigerants (R290, R404A, R407C, R507A, and R22). The R290 and R22 have the highest COP$_{\text{max}}$ values among all five refrigerants at –35 °C. In addition, R290 and R22 have approximately same COP$_{\text{max}}$ 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.

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

In figs. 4 and 5, the variation of $P_{\text{OPT, int}}$ and COP$_{\text{max}}$ 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 $P_{\text{OPT, int}}$ values which maximize the COP for LTA. It is seen that increasing the $T_C$ raises the $P_{\text{OPT, int}}$ for all the refrigerants.

In LTA, the largest values of $P_{\text{OPT, int}}$ are observed when R507A is used whereas the lowest values of $P_{\text{OPT, int}}$ are calculated when R22 is used. The $P_{\text{OPT, int}}$ values of R290 are located within the values of R407C and R22. The R507A and R404A have approximately same $P_{\text{OPT, int}}$...
values between 25 °C and 45 °C. Figure 5 reveals that increasing the $T_c$ reduces the COP$_{\text{max}}$ values for all studied refrigerants. In LTA, the COP$_{\text{max}}$ values of R290 are higher than the COP$_{\text{max}}$ values of R404A, R407C, and R507A. For HFC refrigerants, the predicted COP$_{\text{max}}$ values are approximately same from 25 °C to 45 °C. The R290 and R22 have nearly same COP$_{\text{max}}$ values between 35 °C and 45 °C. On the other hand, R290 has the predicted COP$_{\text{max}}$ 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 the system. For LTA, R22’s total exergy loss is predicted to be lowest while R404A’s total exergy loss is predicted to be highest.

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

The influence of the $P_{\text{OPT, int}}$ on the COP$_{\text{max}}$ and the system total exergy loss is investigated and compared for different refrigerants. In fig. 8, the COP$_{\text{max}}$ 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.

**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

<table>
<thead>
<tr>
<th>Refrigerants</th>
<th>Operating Parameters</th>
<th>Mass ratio ($\dot{m}_H/\dot{m}_L$)</th>
<th>$W_{total}$ [kW]</th>
<th>$\dot{X}_{lost,min}$ [kW]</th>
<th>$\eta_{II,max}$ [%]</th>
<th>COP$_{max}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>LTA</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>R22</td>
<td>–35</td>
<td>40</td>
<td>455</td>
<td>1.48</td>
<td>3.30</td>
<td>1.32</td>
</tr>
<tr>
<td>R290</td>
<td>–35</td>
<td>40</td>
<td>562</td>
<td>1.50</td>
<td>3.31</td>
<td>1.42</td>
</tr>
<tr>
<td>R407C</td>
<td>–35</td>
<td>40</td>
<td>605</td>
<td>1.59</td>
<td>3.64</td>
<td>1.71</td>
</tr>
<tr>
<td>R404A</td>
<td>–35</td>
<td>40</td>
<td>734</td>
<td>1.65</td>
<td>3.58</td>
<td>1.63</td>
</tr>
<tr>
<td>R507A</td>
<td>–35</td>
<td>40</td>
<td>760</td>
<td>1.66</td>
<td>3.53</td>
<td>1.59</td>
</tr>
</tbody>
</table>

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\textsubscript{max}, η\textsubscript{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 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 worsen the performance.

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

<table>
<thead>
<tr>
<th>Parameter</th>
<th>R290</th>
<th>R404A</th>
<th>R507A</th>
<th>R407C</th>
<th>R22</th>
<th>Difference [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>COP\textsubscript{max}</td>
<td>1.81</td>
<td>1.67</td>
<td>1.72</td>
<td>1.65</td>
<td>1.82</td>
<td>-7.73</td>
</tr>
<tr>
<td>η\textsubscript{II, max} [%]</td>
<td>57.12</td>
<td>54.58</td>
<td>54.94</td>
<td>53.91</td>
<td>57.28</td>
<td>-4.45</td>
</tr>
<tr>
<td>$X_{\text{tot, min}}$ [kW]</td>
<td>1.42</td>
<td>1.63</td>
<td>1.59</td>
<td>1.71</td>
<td>1.32</td>
<td>14.79</td>
</tr>
<tr>
<td>$W_{\text{tot}}$ [kW]</td>
<td>3.31</td>
<td>3.58</td>
<td>3.53</td>
<td>3.64</td>
<td>3.30</td>
<td>8.48</td>
</tr>
</tbody>
</table>

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

The R404A and R407C systems have the approximately same COP\textsubscript{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\textsubscript{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\textsubscript{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\textsubscript{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 $P_{\text{int}}$ 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\textsubscript{II} are computed from the predicted values of $P_{\text{OPT, int}}$. The influence of $T_e$ and $T_c$ on the system $P_{\text{OPT, int}}$, COP\textsubscript{max}, η\textsubscript{II, max} is evaluated for LTA’s refrigerants. From the results, it can be concluded as follow.

- For all systems, the increase in $P_{\text{OPT, int}}$ value causes the increase of system COP\textsubscript{max}. It is also realized that the lowest increase of $P_{\text{OPT, int}}$ value provides the highest increase of COP\textsubscript{max} value for R290. In LTA, R507A system has the highest $P_{\text{OPT, int}}$ values and R22 system has the lowest $P_{\text{OPT, int}}$ values at their respective $T_c$ and $T_e$ values.
- Decreasing $T_e$ reduces the COP\textsubscript{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\textsubscript{max} within their respective $T_e$ interval.
The COP$_{\text{max}}$ 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$_{\text{max}}$ values from 25-45 °C. On the other hand, R290, and R22 have nearly same COP$_{\text{max}}$ 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$_{\text{max}}$, ηII$_{\text{max}}$, 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$_{\text{max}}$ (1.81), GWP (11), and ODP (0). The R404A, R407C, and R507A systems COP$_{\text{max}}$ are lower than the R290 system’s COP$_{\text{max}}$ 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$^{-1}$]
- $m$ – mass-flow rate, [kg/s]
- $P$ – pressure, [kPa or bar]
- $P_{\text{CR}}$ – critical pressure, [MPa]
- $Q$ – heat transfer rate, [kW]
- $r$ – mass-flow rate ratio, [–]
- $s$ – specific entropy, [kJkg$^{-1}$K$^{-1}$]
- $T$ – temperature, [°C or K]
- $T_{\text{CR}}$ – critical temperature, [°C]
- $\Delta T$ – temperature difference, [–]
- $W$ – power, [kW]
- $\dot{X}$ – rate of exergy loss, [kW]
- $\eta$ – efficiency, [%]
- $\eta_{\text{II}}$ – second law efficiency, [%]
- EXP – experimental
- EV – expansion valve
- evap – evaporator
- FC – flash intercooler
- HPC – high pressure compressor
- int – intermediate
- LPC – low pressure compressor
- max – maximum
- OPT – optimum
- Rev – reversible
- sub – subcooling
- sup – superheating

**Greek Letters**

- η – efficiency, [%]
- ηII – second law efficiency, [%]
- 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

**Subscripts**

- Act – actual
- C – condensation
- CR – critical
- cond – condenser
- E – evaporation
- Exp – experimental
- EV – expansion valve
- EV – expansion valve
- int – intermediate
- LPC – low pressure compressor
- max – maximum
- OPT – optimum
- Rev – reversible
- sub – subcooling
- sup – superheating

**References**


***, ANSI/ASHRAE Standard 34, Designation and Safety Classification of Refrigerants, 2016


***, EES., Engineering Equation Solver, Academic Commercial, V10.326, f Chart Software Inc., 2017