SIMULATION OF THERMODYNAMIC ANALYSIS OF CASCADE REFRIGERATION SYSTEM WITH ALTERNATIVE REFRIGERANTS

Mr. PARTHIBAN KASI
Assistant Professor, Department of Mechanical Engineering, Velammal Institute of Technology, Chennai, India

ABSTRACT

The main aim of this project is to analyses the cascade refrigeration system by employing various alternative refrigerant pairs and choosing the best pair for higher temperature circuit (HTC) and lower temperature circuit (LTC). The analysis was done in various refrigerants pairs which are R134a/R23, R290/R23, R404A/R23, R407C/R23, R410A/R23, R134a/R508B, R290/R508B, R404A/R508B, R407C/R508B, R410A/R508B, R134a/R170, R290/R170, R404A/R170, R407C/R170 and R410A/R170. Assuming the degrees of sub cooling and superheating as 5°C and 10°C, respectively. The condenser temperature in higher temperature circuit (HTC) was varied from 30°C to 50°C and evaporator temperature in lower temperature circuit (LTC) was varied from -70 to -50°C. The intermediate heat exchanger temperature is about -20°C. Furthermore, the efficiencies of the compressors are assumed to be equal as 0.7. It has been found that the coefficient of performance (COP) of the cascade refrigeration system increases and the mass flow rate of higher temperature circuit increases, along with the work of compressor with rise in evaporator temperature for all refrigerant pairs. On the other hand, the COP of the cascade refrigeration system decreases and the mass flow rate of higher temperature circuit increases, also there is increase in work of compressor with increase in condenser temperature. Finally, the refrigerant pair R134a–R170 has the highest COP and lowest mass flow rate, while R404A–R508B has the lowest COP and highest mass flow rate.

Keywords: Cascade Refrigeration System, Heat Exchanger, Condenser, Evaporator, COP, etc.

1. INTRODUCTION

Vapor compression cycle can be used in the temperature range of -10 to -30°C easily. And low-temperature refrigeration systems are typically required in the temperature range from -30°C to –100°C for applications in food, pharmaceutical, chemical, and other industries, e.g., blast freezing,
cold storages, liquefaction of gases such as natural gas, etc. At such low temperatures, single-stage compression systems with reciprocating compressors are generally not feasible due to high pressure ratios. A high pressure ratio implies high discharge and oil temperatures and low volumetric efficiencies and, hence it has low COP values. Screw and scroll compressors have relatively flat volumetric efficiency curves and have been reported to achieve temperatures as low as –40°C to –50°C in single-stage systems.

Cascade refrigeration cycle can be used to achieve low temperatures, where series of single-stage units are used that are thermally coupled through evaporator/condenser cascades, as shown in Fig. 1 for a two-circuit cascade unit. Each circuit has a different refrigerant suitable for that temperature, the lower temperature units are progressively using lower boiling point refrigerants. Generally, two-circuit and rarely three-circuit cascade systems are used. In general, if the desired temperature it can be easily achieved in a single-stage machine, it will be more efficient than a cascade system due to irreversibility and losses associated with a large number of components.

2. THERMODYNAMIC ANALYSIS OF CASCADE REFRIGERATION SYSTEM

Fig. 2 shows vapor compression cascade refrigeration system under consideration, which consists of low and high side refrigeration systems indicated as A and B. Refrigeration systems A and B are coupled to each other by means of a heat exchanger in which the total heat from refrigeration system A is rejected to refrigeration system B. The refrigerants flowing in both systems are usually different from each other although there are some cases where the same refrigerant can be used in both systems.

Fig. 1: Cascade refrigeration system

Fig. 2: P-H diagram of cascade refrigeration system
Thermodynamic analysis is based on the energy and irreversibility analyses of the elements of the two stage vapor compression cascade refrigeration system. Each element in the system is treated as a control volume. The equations energy and continuity for a control volume can be written as

\[
\frac{d}{dt}
\left[
Q_{\text{c,v}} + \sum m_i \left( h_i + \frac{V_i^2}{2} + gZ_i \right)
\right] = \sum \left[ \frac{\partial}{\partial t} \left( m_i \left( h_i + \frac{V_i^2}{2} + gZ_i \right) \right) \right] + W_{\text{c,v}}
\]

where \( Q_{\text{c,v}} \), \( W_{\text{c,v}} \) and \( E_{\text{c,v}} \) are the heat transfer rate to the control volume, the actual power done by the control volume and the energy within the boundary of the control volume respectively, while \( m, h, \frac{V^2}{2}, gZ \) are the mass flow rate specific enthalpy, specific kinetic energy and potential energy of the fluid at the inlet or outlet conditions.

It is assumed that subcooling occurs in the liquid line while superheating occurs in the suction line inside the refrigerated space, changes in kinetic and potential energies and pressure drop through the cycle are negligible. It is assumed that steady-state and uniform flow conditions exist through the components of the cascade refrigeration system, in other words, the condition of the mass at each point of the components doesn’t change with time. It is also assumed that isentropic efficiencies of the compressors which are equal, and the heat loss from the compressors, heat exchanger and expansion valves are negligible.

\[
\frac{d}{dt} E_{\text{c,v}} = 0
\]

\[
\sum m_o = \sum m_i = m_A
\]

The heat transfer rate of the evaporator (refrigeration capacity) \( Q_e \) is

\[
Q_e = m_A q_e
\]

where \( q_e \) and \( m_A \) are the specific refrigeration capacity of system A and the refrigerant mass flow rate flowing through system A, respectively.

The total work input to the compressors in both systems can be calculated from

\[
W_{\text{tot}} = m_A W_{\text{compA}} + m_B W_{\text{compB}}
\]

where \( m_B \) is the refrigerant mass flow rate flowing through system B, while \( W_{\text{compA}} \) and \( W_{\text{compB}} \) are the specific work of compressions in system A and system B, respectively.

The coefficient performance of the cascade system \( \text{COP}_{\text{cas}} \) is

\[
\text{COP}_{\text{cas}} = \frac{Q_e}{W_{\text{tot}}}
\]
2.1 Low Temperature Side of the Cascade Refrigeration System

As mentioned earlier, the low temperature side of the cascade refrigeration system consists of compressor A, one side of the heat exchanger, thermostatic expansion valve A, and the evaporator.

In the refrigeration cycle A, the refrigerant enters compressor A at state 1 as superheated vapour at low pressure and low temperature. From state 1 to 2, the refrigerant is compressed and discharged as superheated vapour at high pressure and high temperature. Then, it enters the heat exchanger where it condenses and rejects heat to the evaporator of system B. It leaves the heat exchanger at state 3 as high pressure saturated liquid. From state 3 to 4, the refrigerant is sub cooled as a result of heat transfer to the condensing medium. At state 4, the refrigerant enters expansion valve A, where its pressure is reduced in a throttling process to the evaporator pressure. After leaving the valve at state 5, it enters the evaporator, where it absorbs heat from the refrigerated space. At state 6, it leaves the evaporator as saturated vapour at low pressure and low temperature.

When the assumptions mentioned previously are taken into consideration, specific work of compression for the compressor can be written as

\[ w_{comp A} = h_2 - h_1 \]

where \( h_1 \) and \( h_2 \) are the specific enthalpies across compressor A.

The refrigerants are simulated as ideal gases during compression process. Hence the specific work of compression can also be expressed by

\[ w_{comp} = \frac{m}{\eta_{comp}} \left( h_2 - h_1 \right) \]

where \( P_c, P_e, \) and \( T_{comp,i} \) are the condenser pressure, evaporator pressure, and the temperature at compressor inlet, respectively, while \( \eta_{isent} \) is the isentropic efficiency of the compressor; \( c_p \) and \( K \) are constant pressure specific heat and specific heat ratio of the refrigerant, respectively. Isentropic efficiency of the compressor can be expressed in terms of polytrophic efficiency \( \eta_{pol} \); pressure ratio and specific heat ratio.

The specific heat rejected in the heat exchanger can be written as

\[ q_{HE - A} = (h_2 - h_4) \]

where \( h_4 \) is the enthalpy of the refrigerant at the expansion valve inlet. The enthalpy of the sub cooled liquid at state 4 can be calculated by assuming that saturated liquid at point 3 is cooled to state 4 at a constant pressure. Therefore, specific enthalpy at state 4 can be expressed as

\[ h_4 = h_3 - c_{p,l} (T_3 - T_4) \]

where \( c_{p,l} \) is the specific heat at constant pressure of refrigerant in liquid state between states 3 and 4, \( h_3 \) the enthalpy of saturated liquid at the heat exchanger exit, \( T_3 \) the saturation temperature in the heat exchanger and \( T_4 \) is the temperature at the expansion valve inlet.

During the throttling process in the expansion valve, it is assumed that there is no heat transfer to the environment, which results in

\[ h_4 = h_5 \]
The specific refrigeration capacity of refrigeration cycle A can be written as

\[ q_e = m_r (h_1 - h_5) \]

where \( h_5 \) is the enthalpy of refrigerant at evaporator inlet. The enthalpy of superheated vapour at state 1 can be calculated by assuming that saturated vapour at state 6 which is heated to state 1 at a constant pressure. Therefore, specific enthalpy at state 1 can be expressed as

\[ h_1 = h_6 + c_{p,v} (T_1 - T_6) \]

Where \( c_{p,v} \) is the constant pressure specific heat of refrigerant in vapor state between states 6 and 1, \( T_6 \) is the evaporator temperature, and \( T_1 \) is the temperature at the compressor inlet.

### 2.2 High Temperature Side of the Cascade Refrigeration System

In refrigeration system B, the same processes exist and same assumptions are valid as in system A. The specific work of compression for the adiabatic compression process in system B can be written as

\[ w_{compB} = h_8 - h_7 \]

where \( h_7 \) and \( h_8 \) are the specific enthalpies across compressor B.

\[ w_{comp} = \frac{m_r}{\eta_{comp}} (h_8 - h_7) \]

The rate of enthalpy change of the condenser can be determined from

\[ q_c = m_r (h_{10} - h_8) \]

where \( h_{10} \) is the enthalpy of the refrigerant at the expansion valve inlet. The enthalpy of sub cooled liquid can be calculated using the same approach. Therefore, specific enthalpy at state 10 can be expressed as

\[ h_{10} = h_9 - c_{p,l} (T_{10} - T_9) \]

where \( c_{p,l} \) is the constant pressure specific heat of the refrigerant in liquid state between states 9 and 10, \( T_9 \) is the condenser temperature and \( T_{10} \) is the temperature at the expansion valve inlet.

The assumption in refrigeration system A can also be used for the throttling process in expansion valve B,

\[ h_{10} = h_8 \]

where \( h_{10} \) and \( h_{11} \) are the enthalpies of refrigerant at the expansion valve inlet and exit, respectively.
The specific heat absorbed in the heat exchanger can be written as

\[ q_{HE - B} = (h_7 - h_{11}) \]

The specific enthalpy of superheated vapor at state 7 can be expressed by

\[ h_7 = h_{12} + c_{p,v}(T_7 - T_{12}) \]

where \( c_{p,v} \) is the constant pressure specific heat of the superheated refrigerant vapor between states 12 and 7, \( h_{12} \) is the enthalpy of saturated vapor at the heat exchanger exit, \( T_{12} \) is the saturation temperature in the heat exchanger, and \( T_7 \) is the temperature at the compressor inlet.

### 2.3 Heat Exchanger

Combining refrigeration systems A and B, the heat exchanger is the key component of the cascade system. The heat rejected from the refrigeration system A is transferred to the refrigeration system B in the heat exchanger. It is also well insulated to provide that the total heat rejected from refrigeration cycle A be equal to the total heat absorbed by the refrigeration system B. As a result, the first law of thermodynamics for the heat exchanger can be written as

\[ m_B(h_7 - h_{11}) = m_A(h_2 - h_4) \]

A computer code based on the thermodynamic models presented above has been developed to calculate the COPs and mass ratio of various refrigerant pairs.

### 3. SIMULATION METHODOLOGY

The engineering calculation and simulation of the cascade refrigeration cycle system requires the availability of simple and efficient mathematical methods for the determination of thermodynamic property values of the operating fluid. Values of the thermodynamic property are necessary both at the key points in the cycle and along the process taking in the various components.

#### 3.1 Input Data for Simulation

- Condenser Temperature: \( T_c = 30 \) to 50°C
- Evaporator Temperature: \( T_E = -70 \) to -50°C
- Intermediate temperature: \( T_{INT} = -20°C \)
- Compressor efficiency of the system: \( \eta_{comp} = 0.7 \)
- Refrigerant in high temperature circuit: R-134a, R-290, R-407C, R-404A, R-410A
- Refrigerant in low temperature circuit: R-23, R-508B, R-170
- Sub cooling (in HTC and LTC): 5°C
- Superheating (in HTC and LTC): 10°C

By carrying out the thermodynamic analysis of the system for the conditions stated above the values at various state points of the cascade refrigeration cycle have been obtained.
In the present work, following parameters have been computed.
- Coefficient of performance of a cascade refrigeration cycle.
- Mass Flow rate of HTC.
- Work of compression.

3.2 Selection of Working Fluid
Many factors need to be considered
- Ozone depletion potential
- Global warming potential
- Combustibility
- Thermal factors
- Toxicity
- Cost
- Environmental friendliness, etc.

3.2.1 Low Temperature Circuit

<table>
<thead>
<tr>
<th>REFRIGERANTS</th>
<th>PROPERTIES</th>
</tr>
</thead>
<tbody>
<tr>
<td>R-23 (Trifluormethane)</td>
<td>BP: -82.1°C</td>
</tr>
<tr>
<td></td>
<td>Tcrit:25.9°C</td>
</tr>
<tr>
<td></td>
<td>ODP:0, GWP:12000</td>
</tr>
<tr>
<td>R-170 (Ethane)</td>
<td>BP: -88.6°C</td>
</tr>
<tr>
<td></td>
<td>Tcrit:32.2°C</td>
</tr>
<tr>
<td></td>
<td>ODP:0, GWP:20</td>
</tr>
<tr>
<td>R-508B</td>
<td>BP: -87.6°C</td>
</tr>
<tr>
<td></td>
<td>Tcrit:11.2°C</td>
</tr>
<tr>
<td></td>
<td>ODP:0, GWP:1300</td>
</tr>
</tbody>
</table>

3.2.2 Higher Temperature Circuit

<table>
<thead>
<tr>
<th>REFRIGERANT</th>
<th>PROPERTIES</th>
</tr>
</thead>
<tbody>
<tr>
<td>R-134a (Tetrafluoroethane)</td>
<td>Boiling point: -26.1°C</td>
</tr>
<tr>
<td></td>
<td>Critical temperature:101.1°C</td>
</tr>
<tr>
<td></td>
<td>ODP: 0, GWP:1300</td>
</tr>
<tr>
<td>R-290 (Propane)</td>
<td>Boiling point: -42.2°C</td>
</tr>
<tr>
<td></td>
<td>Critical temperature:96.6°C</td>
</tr>
<tr>
<td></td>
<td>ODP: 0, GWP:20</td>
</tr>
<tr>
<td>R-407C</td>
<td>Boiling point: -43.6°C</td>
</tr>
<tr>
<td></td>
<td>Critical temperature:85.8°C</td>
</tr>
<tr>
<td></td>
<td>ODP: 0, GWP:1800</td>
</tr>
<tr>
<td>R-404A</td>
<td>Boiling point: -46.2°C</td>
</tr>
<tr>
<td></td>
<td>Critical temperature:72°C</td>
</tr>
<tr>
<td></td>
<td>ODP: 0, GWP:3900</td>
</tr>
<tr>
<td>R-410A</td>
<td>Boiling point: -51.3°C</td>
</tr>
<tr>
<td></td>
<td>Critical temperature:70.5°C</td>
</tr>
<tr>
<td></td>
<td>ODP: 0, GWP:2100</td>
</tr>
</tbody>
</table>
4. RESULTS & DISCUSSIONS

Using the developed computer code, the COPs and mass flow rates of the HTC refrigerant in the cascade refrigeration system and work of compression are determined as functions of the evaporator temperature, condenser temperature, the intermediate temperatures of the lower and higher temperature systems in the heat exchanger and the compressor efficiency. Then, COPs and mass flow rates and work of compression of the considered pairs are compared and the best refrigerant pair among them is selected.

4.1 Variation of COP with the Condenser Temperature

The changes in the COP of the cascade system will vary in the condenser temperature for an evaporator temperature of -70°C intermediate temperature as -20°C and compressor efficiency of 0.7 respectively. As the condenser temperature of the cascade system increases, the condenser pressure increases, while the evaporator temperature, intermediate temperature and compressor efficiency are kept in constant. On increasing the condenser pressure, more compressor power is obtained, which makes the COP of the cascade system for all considered refrigerant pairs to decrease.

![Fig.3: COP as a function of condenser temperature](image)

4.2 Variation of mass flow rate with the Condenser Temperature

![Fig.4: Mass Flow Rate as a Function of Condenser Temperature](image)
The changes in the mass flow rate of the HTC refrigerant will change in the cascade refrigeration system with respect to the condenser temperature for an evaporator temperature of -70°C, intermediate temperature as -20°C and compressor efficiency of 0.7 respectively. As the condenser temperature of the cascade system increases, the condenser pressure increases, while the evaporator temperature, intermediate temperature and compressor efficiency are kept in constant. On increasing the condenser pressure, more compressor power is obtained, which causes the mass flow rate of the HTC refrigerant in the cascade refrigeration system for all considered refrigerant pairs to increase. As the condenser temperature of the cascade system increases, the mass flow rate of the HTC refrigerant in the cascade refrigeration system increases. It is seen that R290–R23 has the lowest mass flow rate, while R134a–R23 has the highest one. The mass flow rate of R404A–R23, R407C–R23 and R410A–R23 place in the middle range and increases and they are parallel to each other.

4.3 Variation of work of compression with the Condenser Temperature

![Fig.5: Changes in Compressor Work with Condenser temperature](image)

The change in compressor work of the refrigerant pairs increase with respect to the condenser temperature. The figure shows, the evaporator temperature is -70°C, intermediate temperature is -20°C and compressor efficiency is 0.7 which are kept constant. As the condenser temperature increases, the low temperature circuit compressor’s work done \(W_{\text{COMP1}}\) decreases, but the high temperature circuit compressor’s work done \(W_{\text{COMP2}}\) increases, and the combined effect of these are to increase the work done. In the considered condenser temperature range, R404A–R23 has the highest work done values, while R134a–R23 has the lowest ones. The work done of refrigerant pairs placed in the middle range, namely R290–R23, R407C–R23 and R410A–R23, increases and they are parallel to each other.

4.4 Variation of COP with the Evaporator Temperature

The changes in the COP of the cascade system with respect to the evaporator temperature for an condenser temperature of 40°C, intermediate temperature as -20°C and compressor efficiency of 0.7, respectively. When condenser temperature, intermediate temperature and compressor efficiency which are kept constant. The refrigeration effect of the cascade system increases and the specific work of compression decreases on increasing the evaporator temperature. Consequently, the COP of the cascade refrigeration system increases. It is seen that R134a–R23 has the highest COP, while R404A–R23 has the lowest one. The COPs of R290–R23, R407C–R23, and R410A–R23 place in the middle range and increase parallel to each other.
4.5 Variation of mass flow rate with the Evaporator Temperature

The variation in mass flow rate with the evaporator temperature for a constant condenser temperature is 40°C, intermediate temperature is -20°C and compressor efficiency is 0.7. When the condenser temperature, intermediate temperature and compressor efficiency which are kept constant. The refrigeration effect of the cascade system increases and the specific work of compression decreases on increasing the evaporator temperature. Consequently, the mass flow rate of the HTC cascade refrigeration system increases. In the considered evaporator temperature range, R410A–R23 has the lowest mass flow rate values, while R404A–R23 has the highest ones. The mass flow rate of refrigerant pairs placing in the middle range, namely R134a–R23, R407C–R23 and R290–R23, increases parallel to each other.
4.6 Variation of work of compression with the Evaporator Temperature

The changes in compressor work of the refrigerant pairs with respect to the evaporator temperature. The figure shows, the condenser temperature of 40°C, intermediate temperature as -20°C and compressor efficiency of 0.7 which are kept constant. As the evaporator temperature increases, the low temperature circuit compressor’s work done ($W_{COMP1}$) decreases but the high temperature circuit compressor’s work done ($W_{COMP2}$) increases, and the combined effect of these are to increase the work done. In the considered evaporator temperature range, R404A–R23 has the highest work done values, while R134a–R23 has the lowest ones. The work done of refrigerant pairs placing in the middle range, namely R290–R23, R407C–R23 and R410A–R23, increase parallel to each other.

4.7 Variation of COP with the Condenser Temperature

The changes in COP of the cascade system will vary in the condenser temperature for an evaporator temperature of -70°C intermediate temperature as -20°C and compressor efficiency of 0.7 respectively. As the condenser temperature of the cascade system increases, the condenser pressure increases, while the evaporator temperature, intermediate temperature and compressor efficiency are kept in constant. On increasing the condenser pressure, more compressor power is obtained, which makes the COP of the cascade system for all considered refrigerant pairs to decrease. In the considered condenser temperature range, R134a–R170 has the highest COP values, while R404A–R170 has the lowest ones. The COPs of refrigerant pairs placing in the middle range, namely R410A–R170, R407C–R170 and R290–R170, decrease parallel to each other.
4.8 Variation of mass flow rate with the Condenser Temperature

The changes in the mass flow rate of the HTC refrigerant in the cascade refrigeration system with respect to the condenser temperature for an evaporator temperature of -70°C, intermediate temperature as -20°C and compressor efficiency of 0.7, respectively. As the condenser temperature of the cascade system increases, the condenser pressure also increases, while the evaporator temperature, intermediate temperature and compressor efficiency are kept constant. On increasing the condenser pressure, more compressor power is required, which causes the mass flow rate of the HTC refrigerant in the cascade refrigeration system for all considered refrigerant pairs to increases. As the condenser temperature of the cascade system increases, the mass flow rate of the HTC refrigerant in the cascade refrigeration system increases. It is seen that R404A–R170 has the lowest mass flow rate, while R407C–R170 has the highest one. The mass flow rate of R134a–R170, R290–R170, and R410A–R170 place in the middle range and increases any they are parallel to each other.
4.9 Variation of work of compression with the Condenser Temperature

The changes in compressor work of the refrigerant pairs with respect to the condenser temperature. The figure shows, the evaporator temperature of -70°C, intermediate temperature as -20°C and compressor efficiency of 0.7 are kept constant. As the condenser temperature increases, the low temperature circuit compressor's work done (W\text{COMP1}) decreases but the high temperature circuit compressor's work done (W\text{COMP2}) increases, and the combined effect of these are to increase the work done. In the considered condenser temperature range, R404A–R170 has the highest work done values, while R134a–R170 has the lowest ones. The work done of refrigerant pairs placing in the middle range, namely R290–R170, R407C–R170 and R410A–R170, increase parallel to each other.

4.10 Variation of COP with the Evaporator Temperature

The changes in the COP of the cascade system with respect to the evaporator temperature for an condenser temperature of 40°C, intermediate temperature as -20°C and compressor efficiency of 0.7 respectively. When condenser temperature, intermediate temperature and compressor efficiency
which are kept constant. The refrigeration effect of the cascade system increases and the specific work of compression decreases on increasing the evaporator temperature. Consequently, the COP of the cascade refrigeration system increases. It is seen that R134a–R170 has the highest COP, while R404A–R170 has the lowest one. The COPs of R290–R170, R407C–R170, and R410A–R170 place in the middle range and increase parallel to each other.

4.11 Variation of mass flow rate with the Evaporator Temperature

The variation in mass flow rate with the evaporator temperature for a constant condenser temperature of 40°C, intermediate temperature of -20°C and compressor efficiency is 0.7. When the condenser temperature, intermediate temperature and compressor efficiency which are kept constant. The refrigeration effect of the cascade system increases and the specific work of compression decreases on increasing the evaporator temperature. Consequently, the mass flow rate of the HTC cascade refrigeration system increases. In the considered evaporator temperature range, R404A–R170 has the lowest mass flow rate values, while R134a–R170 has the highest ones. The mass flow rate of refrigerant pairs placing in the middle range, namely R410A–R170, R407C–R170 and R290–R170, increases parallel to each other.

4.12 Variation of work of compression with the Evaporator Temperature

The changes in compressor work of the refrigerant pairs with respect to the evaporator temperature. In this figure, the condenser temperature of 40°C, intermediate temperature as -20°C and compressor efficiency of 0.7 are kept constant. As the evaporator temperature increases, the low temperature circuit compressor’s work done (W_{COMP1}) decreases but the high temperature circuit compressor’s work done (W_{COMP2}) increases, and the combined effect of these are to increase the work done. In the considered evaporator temperature range, R404A–R170 has the highest work done
values, while R134a–R170 has the lowest ones. The work done of refrigerant pairs placing in the middle range, namely R290–R170, R407C–R170 and R410A–R170, increase parallel to each other.

![Fig.14: Compressor Work as a Function of Evaporator Temperature](image)

4.13 Variation of COP with the Condenser Temperature

The changes in the COP of the cascade system will vary in the condenser temperature for an evaporator temperature of -70°C intermediate temperature as -20°C and compressor efficiency of 0.7 respectively. As the condenser temperature of the cascade system increases, the condenser pressure increases, while the evaporator temperature, intermediate temperature and compressor efficiency are kept in constant. On increasing the condenser pressure, more compressor power is obtained, which makes the COP of the cascade system for all considered refrigerant pairs to decrease. In the considered condenser temperature range, R290–R508B has the highest COP values, while R404A–R508B has the lowest ones. The COPs of refrigerant pairs placing in the middle range, namely R410A–R508B, R407C–R508B and R134a–R508B, decrease parallel to each other.

![Fig.15: Variation in COP as a function of condenser temperature](image)
4.14 Variation of mass flow rate with the Condenser Temperature

Fig. 16: Variation in Mass Flow Rate as a Function of Condenser Temperature

The changes in the mass flow rate of the HTC refrigerant in the cascade refrigeration system with respect to the condenser temperature for an evaporator temperature of -70°C, intermediate temperature as -20°C and compressor efficiency of 0.7, respectively. As the condenser temperature of the cascade system increases, the condenser pressure also increases, when evaporator temperature, intermediate temperature and compressor efficiency which are kept constant. On increasing the condenser pressure, more compressor power is required, which causes the mass flow rate of the HTC refrigerant in the cascade refrigeration system for all considered refrigerant pairs to increases. As the condenser temperature of the cascade system increases, the mass flow rate of the HTC refrigerant in the cascade refrigeration system increases. It is seen that R404A–R508B has the lowest mass flow rate, while R407C–R508B has the highest one. The mass flow rate of R134a–R508B, R290–R508B, and R410A–R508B place in the middle range and increases parallel to each other.

4.15 Variation of work of compression with the Condenser Temperature

The changes in compressor work of the refrigerant pairs with respect to the condenser temperature. In this figure, the evaporator temperature of -70°C, intermediate temperature as -20°C and compressor efficiency of 0.7 which are kept constant. As the condenser temperature increases, the low temperature circuit compressor’s work done (W_{COMP1}) decreases but the high temperature circuit compressor’s work done (W_{COMP2}) increases, and the combined effect of these are to increase the work done. In the considered condenser temperature range, R404A–R508B has the highest work done values, while R290–R508B has the lowest ones. The work done of refrigerant pairs placing in the middle range, namely R134a–R508B, R407C–R508B and R410A–R508B, increase parallel to each other.
4.16 Variation of COP with the Evaporator Temperature

The changes in the COP of the cascade system with respect to the evaporator temperature for an condenser temperature of 40°C, intermediate temperature as -20°C and compressor efficiency of 0.7, respectively. When condenser temperature, intermediate temperature and compressor efficiency which are kept constant. The refrigeration effect of the cascade system increases and the specific work of compression decreases on increasing the evaporator temperature. Consequently, the COP of the cascade refrigeration system increases. It is seen that R290–R508B has the highest COP, while R404A–R508B has the lowest one. The COPs of R134a–R508B, R407C–R508B, and R410A–R508B place in the middle range and increase parallel to each other.
4.17 Variation of mass flow rate with the Evaporator Temperature

The variation in mass flow rate with the evaporator temperature for a constant condenser temperature of 40°C, intermediate temperature of -20°C and compressor efficiency is 0.7. When the condenser temperature, intermediate temperature and compressor efficiency are kept constant. The refrigeration effect of the cascade system increases and the specific work of compression decreases on increasing the evaporator temperature. Consequently, the mass flow rate of the HTC cascade refrigeration system increases. In the considered evaporator temperature range, R404A–R508B has the lowest mass flow rate values, while R407C–R508B has the highest ones. The mass flow rate of refrigerant pairs placing in the middle range, namely R410A–R508B, R134a–R508B and R290–R508B, increases parallel to each other.

4.18 Variation of work of compression with the Evaporator Temperature

The changes in compressor work of the refrigerant pairs with respect to the evaporator temperature. In this figure, the condenser temperature of 40°C, intermediate temperature as -20°C and compressor efficiency of 0.7 are kept constant. As the evaporator temperature increases, the low temperature circuit compressor’s work done (W\text{COMP1}) decreases but the high temperature circuit compressor’s work done (W\text{COMP2}) increases, and the combined effect of these is to increases work done. In the considered evaporator temperature range, R404A–R508B has the highest work done values, while R290–R508B has the lowest ones. The work done of refrigerant pairs placing in the middle range, namely R134a–R508B, R407C–R508B and R410A–R508B, increase parallel to each other.
The changes in compressor work of the refrigerant pairs with respect to the evaporator temperature. The figure shows, the condenser temperature of 40°C, intermediate temperature as -20°C and compressor efficiency of 0.7 are kept constant. As the evaporator temperature increases, the low temperature circuit compressor’s work done ($W_{COMP1}$) decreases but the high temperature circuit compressor’s work done ($W_{COMP2}$) increases, and the combined effect of these is to increases work done. In the considered evaporator temperature range, R404A–R508B has the highest work done values, while R290–R508B has the lowest ones. The work done of refrigerant pairs placing in the middle range, namely R134a–R508B, R407C–R508B and R410A–R508B, increase parallel to each other.

CONCLUSION

Coefficient of performance, mass flow rate and compressor work of the cascade refrigeration system for the various refrigerant pairs have been analyzed and compared with the evaporator temperature, condenser temperature, the temperature difference between the saturation temperatures of the lower and higher temperature systems in the heat exchanger ($\Delta T$), intermediate temperature and the compressor efficiency of the cascade refrigeration system.

Refrigerant pair R134a–R170 is found to be best pair for the vapour compression cascade refrigeration systems among all considered pairs.

NOMENCLATURE

$C_p$ - constant pressure specific heat of (kJ kg$^{-1}$ K$^{-1}$)
$E$ - Energy (kJ)
$h$ - Specific enthalpy of refrigerant (kJ kg$^{-1}$)
$K$ - Ratio of constant specific heats
$m$ - Mass flow rate (kg s$^{-1}$)
$q$ - Specific capacity (kJ kg$^{-1}$)
$Q$ - Heat transfer rate (kW)
$P$ - Pressure (kPa)
$T$ - Temperature (K)
$\Delta T$ - temperature difference between the saturation temperatures of the lower and higher temperature systems in the heat exchanger (K)
$w$ - Specific compressor capacity (kJ kg$^{-1}$)

Greek words

$\eta_c$ - efficiency of the compressor

Subscripts

A refrigeration system A
B refrigeration system B
Abbreviations

BP  Boiling Point
HTC  Higher Temperature circuit
LTC  Lower Temperature circuit
COP  Coefficient of Performance
ODP  Ozone Depletion Potential
GWP  Global Warming Potential

REFERENCES


