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Thermodynamic analysis of cascade refrigeration system using refrigerants pairs R134a-R23 and R290-R23

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Abstract

A thermodynamic energy and exergy analysis cascade refrigeration system using refrigerants pairs R134a-R23and R290-R23 is presented in this paper to optimize the operating parameters of the system. The design and operating parameters considered in this study include (1) evaporating, condensing, cascade condensing temperature and temperature difference in cascade condenser, (2) subcooling and superheating temperatures of HT and LT system, and (3) isentropic efficiency of HT and LT compressors and cascade condenser efficiency. R134a and R290 refrigerants are used in high temperature application and R23 is used in low temperature application.

Keywords: Cascade refrigeration system, Energy, Exergy, R134a, R290, R23..

Introduction

Vapour compression system can be used in temperature range from -10 ^{0}C to - 30 ^{0}C easily and low-temperature refrigeration systems are typically required in the temperature range from – 30 ^{0}C to – 100 ^{0}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, low COP values. Screw and scroll compressors have relatively flat volumetric efficiency curves and have been reported to achieve temperatures as low as – 40 ^{0}C to – 50 ^{0}C in single-stage systems. Further, the use of a single refrigerant over such a wide range of temperature results in either extremely low pressures in the evaporator and large suction volumes or extremely high pressures in the condenser. To increase volumetric efficiency and refrigerating effect and to reduce power consumption, multistage with intercooling is often employed.

Therefore, cascade systems are employed to obtain high-temperature differentials between the heat source and heat sink and are applied for temperatures ranging from −70 to 100 ^{0}C. In a cascade system a series of refrigerants with progressively lower boiling points are used in a series of single stage units. The condenser of lower stage system is coupled to the evaporator of the next higher stage system and so on. The component where heat of condensation of lower stage refrigerant is supplied for vaporization of next level refrigerant is called as cascade condenser. This system employs two different refrigerants operating in two individual cycles. They are thermally coupled in the cascade condenser. The refrigerants selected should have suitable pressure-temperature characteristics.

H.M. Getu and P.K. Bansal [1] presented a thermodynamic analysis of carbon-dioxide ammonia (R744–R717) cascade refrigeration system, to optimize the design and operating parameters of the system. M. Idrus Alhamid et al.[2] presented exergy and energy analysis of a cascade refrigeration system using R744 + R170 for low temperature applications. In this study an azeotrope mixture carbon dioxide and ethane–propane (R744+R170–R290) cascade system has been promoted as a prospective alternative solution to the use of HFC refrigerants. S. M. Zubair et al. [3] presented second-law analysis which is carried out for both two-stage and mechanical-subcooling refrigeration systems. J. Alberto Dopazo et al. [4] presented theoretical analysis of a CO_2–NH_3 cascade refrigeration system for cooling applications at low temperatures. A. D. Parekh and P. R. Tailor [5] presented a thermodynamic analysis of cascade refrigeration system using ozone friendly refrigerants pair R507A and R23. CE Vincent and MK Heun [6] has done thermoeconomic analysis and design of domestic refrigeration systems. They find that the Energy Efficiency Rating (EER) of the compressor has the most effect on system performance and economics and find that the cost of refrigerating is driven by compressor costs. A. Kilicarslan [7] presented the experimental investigation and theoretical study of a different type of two-stage vapour compression cascade refrigeration system using R134a as the refrigerant is presented.

Nomenclature

HT	High Temperature
LT	Low Temperature
COP	Coefficient of performance
COP_{MAX}	Maximum Coefficient of performance
HC	Hydrocarbons
T_0	Environment Temperature
T_C	Condensing Temperature
T_E	Evaporating Temperature
T_{CS}	Cascade Condensing Temperature
T_{ES}	Cascade Evaporating Temperature
ΔT_{CC}	The Temperature Difference in The Cascade Condenser
ΔT_{sub}	Sub cooling
ΔT_{sup}	Superheating
ΔT_a	Low side degree of superheating
ΔT_C	Low side degree of sub cooling
ΔT_b	High side degree of superheating
ΔT_d	High side degree of sub cooling
m_H	Mass Flow rate Of Refrigerant in High Temperature Cycle
m_L	Mass Flow rate Of Refrigerant in Low Temperature Cycle
X	Exergy
$\dot{X}des$	Exergy Destruction
Q_E	Heat Absorb in Evaporator
Q_C	Heat Reject In Condenser
Q_{CC}	Rate Of Heat Transfer in the cascade condenser
η_{isen}	Isentropic Efficiency
η_{II}	Exergetic Efficiency
η_{HE}	Cascade condenser efficiency
W_L	Work require for compressor in LT
W_H	Work require for compressor in HT
W_{act}	Actual Work
Cp	Specific heat at Constant Pressure

Sub-script

CC	Cascade condenser
isen	Isentropic
sub	Sub cooling
sup	Superheating
MAX	Maximum
OPT	Optimum
0	Ambient

Cascade System Description

A two-stage cascade system employs two vapour-compression units working separately with different refrigerants and interconnected in such a way that the evaporator of one system is used to serve as condenser to a lower temperature system (i.e., the evaporator from the first unit cools the condenser of the second unit).

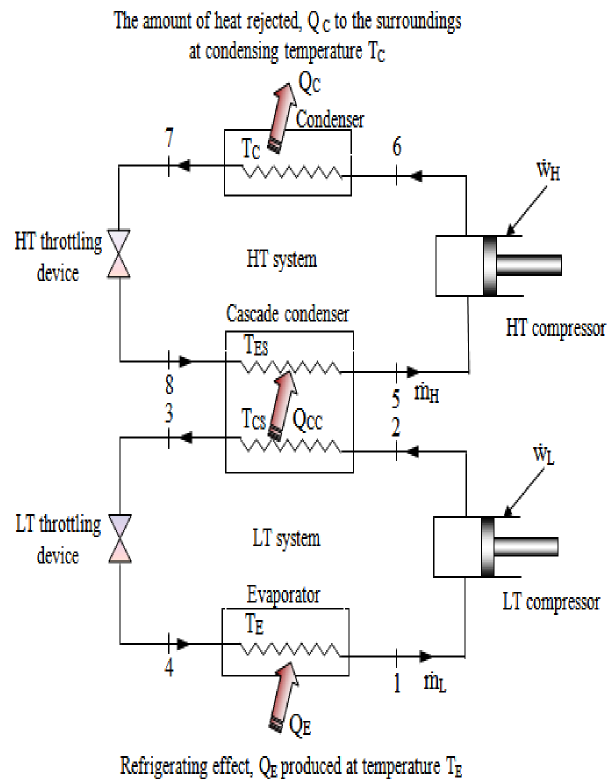

Figure 1 Schematic Diagram of Cascade System

A schematic diagram of cascade refrigeration system shown in Figure 1, the condenser of HT system, called the first or high pressure stage, is usually fan cooled by the ambient air. In some cases a water supply may be used, but air cooling is much more common. The evaporator of HT system is used to cool the condenser of LT system called the second or low-pressure stage. The unit that makes up the evaporator of HT system and the condenser of LT system is often referred to as the inter-stage or cascade condenser.

Figure 2 and 3 shows the T–s and P–h diagram of cascade refrigeration system, respectively. The condenser in this cascade refrigeration system rejects a heat of Q_C from the condenser at condensing temperature of T_C, to its warm coolant or environment at temperature of T_0. The evaporator of this cascade system absorbs a refrigerated load Q_E from the cold refrigerated space at evaporating temperature T_E. The heat rejected by condenser of LT system equals the heat absorbed by the evaporator of the HT system. T_{CS} and T_{ES} represent the condensing and evaporating temperatures of the cascade condenser, respectively.

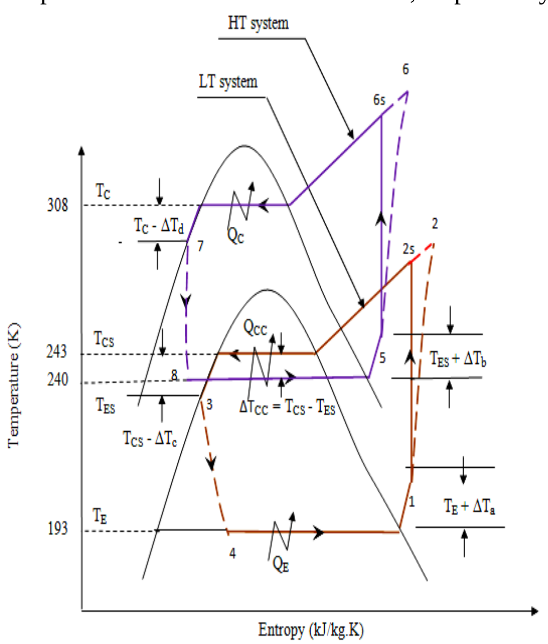

Figure 2 T – s diagram of cascade refrigeration system

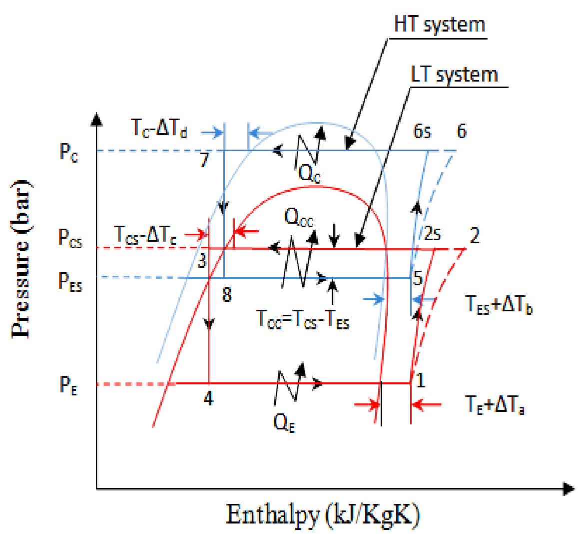

Figure 3 P – h diagram of cascade refrigeration system

The evaporating temperature (T_E), the condensing temperature (T_C), cascade condensing temperature (T_{CS}) and the temperature difference in the cascade condenser (ΔT_{CC}) are four important parameters of a cascade refrigeration system.

As stated earlier, cascade systems generally use two different refrigerants (i.e., one in each stage). One type is used for the low stage and a different one for the high stage. The reason why two refrigeration systems are used is that a single system cannot economically achieve the high compression ratios necessary to obtain the proper evaporating and condensing temperatures.

Mathematical Modeling of Cascade System

Thermodynamic analysis of cascade refrigeration system has been done for two refrigerants pairs R134a-R23 and R290-R23. The low temperature (LT) system with refrigerant R23 is used for cooling for both the systems. The high temperature (HT) system with refrigerant R134a and R290 is used to condensate the R23 of the low temperature system.

Assumptions

The thermodynamic analysis of a cascade refrigeration system was performed based on the following general assumptions:

- Cascade condenser effectiveness with isentropic efficiency for both high and low-temperature compressors is assumed to be 80%.
- Negligible pressure and heat losses/gains in the pipe networks or system components.
- Isenthalpic expansion across expansion valves.
- The dead state (ambient) conditions are 25 ^{0}C and 1 atm.
- The mass flow rate for the lower system region is 0.2 kg/min.

Governing Equations

<u>Energy Analysis:</u>

To calculate the heat transfer rates, compressors powers, and energetic and exergetic efficiencies, each cascade system component is considered as a control volume at stationary flow. Taking into account the assumptions previously made, the mass, energy and exergy balances are given by Eq. 3.1, 3.2 and 3.3, respectively.

Mass balance:

$$\sum_{in} \dot{m} = \sum_{out} \dot{m} \qquad (1)$$

Energy balance:

$$\dot{Q} - \dot{w} = \sum_{out} \dot{m}.h - \sum_{in} \dot{m}.h \qquad (2)$$

Exergy balance:

$$\dot{X}_{lost} = \sum_{out} \left(1 - \frac{T_0}{T_j}\right).\dot{Q}_j - \dot{W} + \sum_{in} \dot{m}.\psi - \sum_{out} \dot{m}.\psi \qquad (3)$$

The capacity of the evaporator is defined as:

$$\dot{Q}_E = \dot{m}_L(h_1 - h_4) \qquad (4)$$

Compressor isentropic efficiency for low-temperature circuit is given as:

$$\eta_{isen,L} = \frac{h_{2s} - h_1}{h_2 - h_1} \quad (5)$$

Whereas, for high-temperature circuit is given as:

$$\eta_{isen,H} = \frac{h_{6s} - h_5}{h_6 - h_5} \quad (6)$$

Compressor power consumption for low-temperature circuit is given as:

$$\dot{w}_L = \dot{m}_L(h_2 - h_1) \quad (7)$$

Whereas, for high-temperature circuit, it is given as:

$$\dot{w}_H = \dot{m}_H(h_6 - h_5) \quad (8)$$

Total work done or Actual work done:

$$w_{act} = \dot{w}_L + \dot{w}_H \quad (9)$$

The rate of heat transfer in the cascade condenser is determined from:

$$\dot{Q}_{CC} = \dot{m}_L(h_2 - h_3) = \dot{m}_H(h_5 - h_8) \quad (10)$$

From the above Eq. (3.10) the mass flow ratio is derived as:

$$\frac{\dot{M}_H}{\dot{m}_L} = \frac{h_2 - h_3}{h_5 - h_8} \quad (11)$$

The rate of heat rejection by the air-cooled condenser is given as:

$$\dot{Q}_H = \dot{m}_H(h_6 - h_7) \quad (12)$$

The overall COP of the system is determined as:

$$COP = \frac{\dot{Q}_E}{w_{act}} \quad (13)$$

Exergy Analysis:

Exergy analysis is usually aimed to determine the maximum performance of the system and identify the locations of exergy destruction and to show the direction for potential improvements. To evaluate the exergy losses of each systems components and the exergy loss rate of the whole system a parametric study was applied on cascade refrigeration system for both refrigerant pairs R134a-R23 and R290-R23.

Exergetic efficiency or Second law efficiency is given by:

$$\eta_{II} = \frac{w_{rev}}{w_{act}} \quad (14)$$

Exergy destruction in the system components:

Consider a cascade refrigeration system as shown in Figure 1.

$$\dot{X}_{LT\ compressor} = \dot{m}_L T_0(s_2 - s_1) \quad (15)$$

$$\dot{X}_{HT\ compressor} = \dot{m}_H T_0(s_6 - s_5) \quad (16)$$

$$\dot{X}_{condenser} = \dot{m}_H[-h_7 + h_6 + T_0(s_7 - s_6)] + \dot{Q}_H\left(1 - \frac{T_0}{T_C}\right) \quad (17)$$

$$\dot{X}_{condenser} = \dot{m}_H T_0(s_7 - s_6) + \dot{m}_H(h_6 - h_7)\left(\frac{T_0}{T_C}\right) \quad (18)$$

$$\dot{X}_{LT\ throttling\ device} = \dot{m}_L T_0(s_4 - s_3) \quad (19)$$

$$\dot{X}_{HT\ throttling\ device} = \dot{m}_H T_0(s_8 - s_7) \quad (20)$$

$$\dot{X}_{cascade\ condenser} = T_0[\dot{m}_L(s_3 - s_2) + \dot{m}_H(s_5 - s_8) \quad (21)$$

$$\dot{X}_{evaporator} = \dot{m}_L[-h_4 + h_1 + T_0(s_1 - s_4)] + \dot{Q}_L\left(1 - \frac{T_0}{T_E}\right) \quad (22)$$

$$\dot{X}_{evaporator} = \dot{m}_L T_0(s_1 - s_4) - \dot{m}_L(h_1 - h_4)\left(\frac{T_0}{T_E}\right) \quad (23)$$

Total exergy destruction:

The total exergy destruction in the system is the sum of exergy destruction in different components of the system and is given by:

$$\dot{X}_{total} = X_{LT\ compressor} + X_{HT\ compressor} + X_{condenser} + X_{LT\ throttling\ device} + X_{LT\ throttling\ device} + X_{cascade\ condenser} + X_{evaporator} \quad (24)$$

Results and Discussion

Effect of evaporating temperature on COP:

Figure 4 shows effect of evaporating temperature on COP in LT system. The evaporating temperature was varied from - 55 ^{0}C to 0 ^{0}C by keeping condensing temperature (T_C = 35 ^{0}C), temperature difference in cascade condenser (ΔT_{CC} = 3 ^{0}C), cascade condenser temperature (T_{CS} = - 30 ^{0}C) subcooling (ΔT_{sub} = 2 ^{0}C) and superheat (ΔT_{sup} = 4 ^{0}C) for both the systems constant. As shown in Figure 4 a rise in the evaporating temperature resulted in an increase in COP for both refrigerant pairs but COP of R134a is slightly higher than R290. With increase in evaporating temperature, the pressure ratio across the compressor reduces causing compressor work to reduce and refrigerating increases. The combined effect of these two factors results better performance of the system.

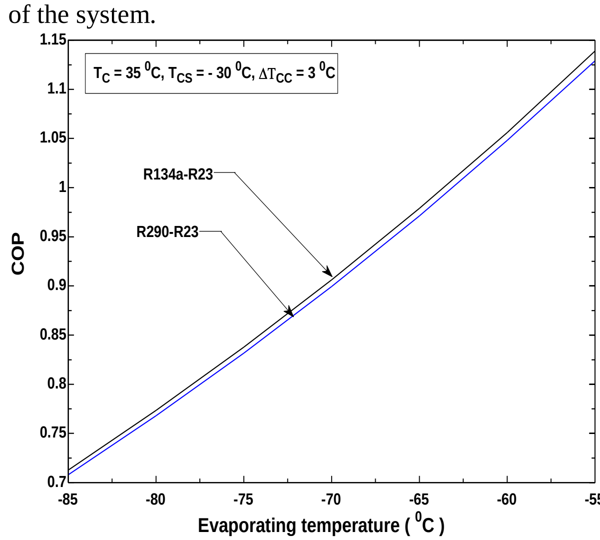

Figure 4 Effect of evaporating temperature on COP

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Effect of cascade condensing temperature on COP:

Figure 5 shows the effect of cascade condensing temperature on COP for refrigerant pairs R134a-R23 and R290-R23 cascade systems. The cascade condenser temperature was varied from - 55 ^{0}C to 0 ^{0}C while other parameters are set at standard values as explained previously. With the increase in the cascade condensing temperature (T_{CS}) Figure 5 shows that there was an optimum temperature at which the COP of the system was maximum.

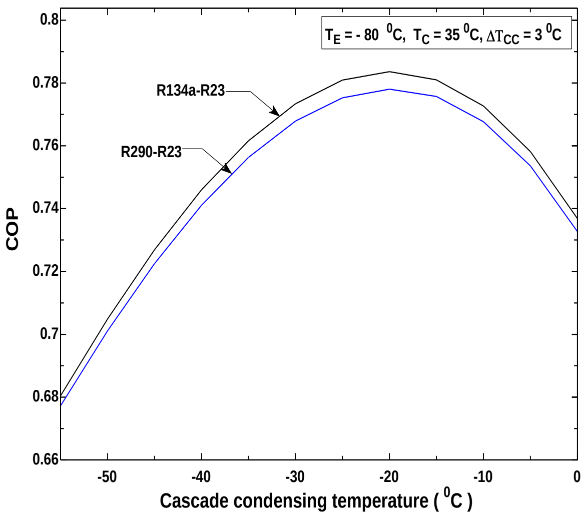

Figure 5 Effect of cascade condensing temperature on COP

This happens because for a given condensing and evaporating temperature, as the cascade condensing temperature increases, the refrigerating effect and the work done by the compressor both decreases, and combined effect of these is to increase the COP up to optimum temperature but after optimum temperature COP decreases. COP of R134a is 0.715% higher than R290 at – 20 ^{0}C cascade condensing temperature.

Effect of cascade condensing temperature on exergetic efficiency and exergy destruction:

Figure 6 and 7 shows the effect of cascade condensing temperature on exergetic efficiency and exergy destruction for refrigerant pairs R134a-R23 and R290-R23 cascade systems, respectively. The cascade condensing temperature was varied from - 55 ^{0}C to 0 ^{0}C while other parameters are set at standard values as explained previously. Similar curves are obtained for η_{II} as in case of COP, the η_{II} first increase than decreases with increase in cascade condenser temperature. This is because for a given T_C and T_E, as the T_{CS} increases, the refrigerating effect and the compressor workdone both decreases, and the combined effect of these is to increase the η_{II} up to optimum temperature but after optimum temperature η_{II} decreases. It is observed that the trends of curves of exergy destruction and exergetic efficiency are almost reverse. The X_{des} initially decreases and then increases with the increase in cascade condenser temperature from - 55 ^{0}C to 0 ^{0}C. The optimum cascade condenser temperature also corresponds to minimum value of X_{des}.

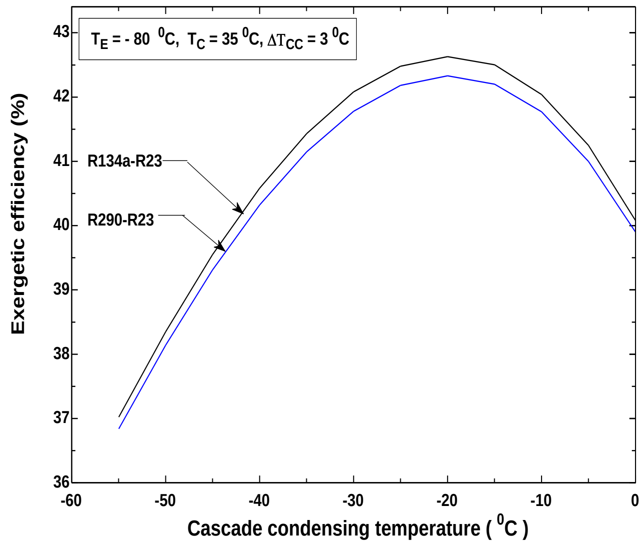

Figure 6 Effect of cascade condensing temperature on exergetic efficiency

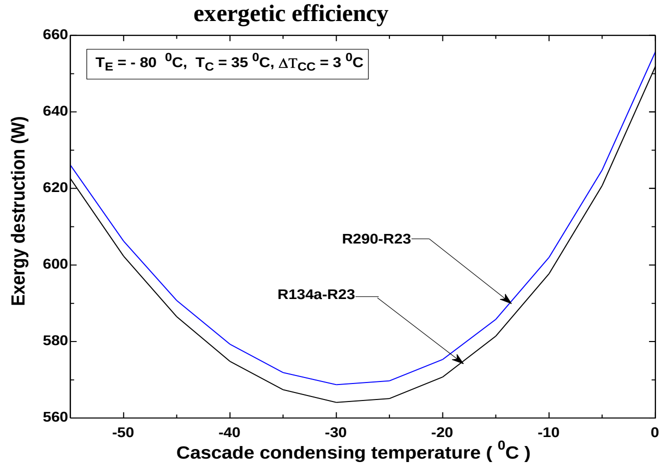

Figure 7 Effect of cascade condensing temperature on exergy destruction

Effect of degree of subcooling and superheating temperatures on COP

The effect of having different and the same degree of subcooling and superheat in systems for the refrigerant pair of R134a-R23 and R290-R23 cascade system was separately and jointly analyzed keeping the other operating parameters at standard values as explained above.

Effect of degree of subcooling (Refrigerant pair R134A-R23)

(i)Subcooling in R134a system: Degree of subcooling in this system was varied from 0 ^{0}C to 14 ^{0}C (Figure 3.9) by keeping the degree of superheat in R134A system at 0 ^{0}C and by holding the degree of subcooling and superheat in R23 system at 0 ^{0}C. It was observed that the COP of the system increased by higher degree than in the case of R23 system.

(ii)Degree of subcooling in R23: Degree of subcooling in R23 was varied from 0 ^{0}C to 14 ^{0}C (Figure 5) by keeping the degree of superheat in R23 system at 0 ^{0}C and by holding the degree of subcooling and superheat in R134A at 0 ^{0}C. The COP of the system increased but at much smaller amount than recorded for subcooling in both systems and in R134a system.

(iii)Effect of the same degree of subcooling in R134a and R23 systems: Degree of subcooling in both systems was varied simultaneously from 0 ^{0}C to 14 ^{0}C (see Figure 5) by holding the superheat at 0 ^{0}C. This resulted in an increase in the performance of the system.

This is because subcooling increases the refrigeration effect by reducing the throttling loss at no additional specific work input. Also subcooling ensures that only liquid enters into the throttling device leading to its efficient operation.

Effect of degree of superheat (Refrigerant pair R134a-R23)

(i)Degree of superheat in R134a system: Degree of superheat in R134a system was varied from 0 ^{0}C to 14 ^{0}C (Figure 5) by keeping the degree of subcooling in R134a system at 0 ^{0}C and holding the degree of subcooling and superheat in R23 system at 0 ^{0}C. This resulted in an increase in COP of the system at some higher amount than in the case of superheating in both systems and superheat in R23.

(ii)Degree of superheat in R23 system: Degree of superheat in R23 system was varied from 0 ^{0}C to 14 ^{0}C (Figure 5) by keeping the degree of subcooling in R23 system at 0 ^{0}C and holding the degree of subcooling and superheat in R134a system at 0 ^{0}C. It decreased COP of the system.

(iii)Effect of the same degree of superheat in R134a and R23 systems: Degree of superheating in both systems was varied simultaneously from 0 ^{0}C to 14 ^{0}C (Figure 5) by holding the subcooling at 0 ^{0}C. This reduces the COP of the system to some extent as compared to superheating in R134a system.

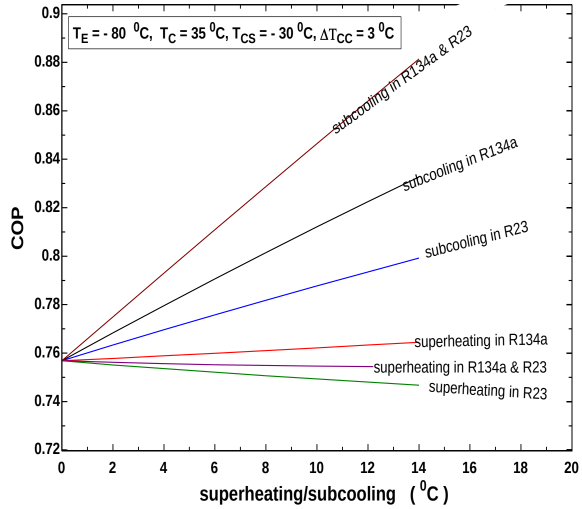

Figure 8 Effect of degree of subcooling and superheating temperatures on COP for refrigerant pair R134a-R23

The superheating increases both the refrigeration effect as well as the work of compression. Hence the COP (ratio of refrigeration effect and work of compression) may or may not increase with superheat, depending mainly upon the nature of the working fluid. Even though useful superheating may or may not increase the COP of the system, a minimum amount of superheat is desirable as it prevents the entry of liquid droplets into the compressor.

Figure 9 shows the effect of subcooling and superheating temperatures on COP for refrigerant pair R290-R23. Results show that the variation of COP for change in subcooling and superheating temperatures for refrigerant pair R290-R23 is similar as in case of R134a-R23.

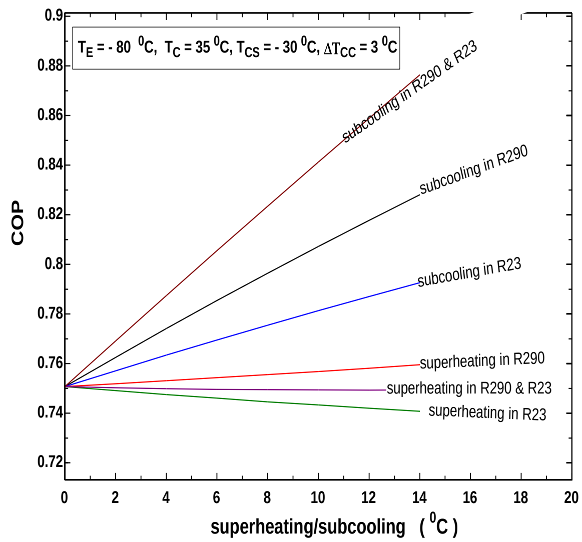

Figure 9 Effect of degree of subcooling and superheating temperatures on COP for refrigerant pair R290-R23

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Conclusion

In the present work, thermodynamic analysis and optimization of cascade refrigeration system has been carried out using R134a-R23 and R290-R23. There is effect of various operating parameters on the performance of cascade refrigeration system. So their influence over the system's COP, exergetic efficiency and entropy generation rate is reported in this analysis. The following conclusions are drawn from the present analysis:

1. The results show that the COP increased by approx 60% when the T_E increased from - 85 ^{0}C to - 55 ^{0}C. The exergetic efficiency increases with increase in T_E upto -70 ^{0}C, after -70 ^{0}C it starts decreasing.
2. The COP and exergetic efficiency increased by approx 8% when the T_{CS} increased from - 55 ^{0}C to 0 ^{0}C.
3. The results show that COP and exergetic efficiency decreases when degree of superheating increases in LT system and increases when degree of superheating increases in HT system and remain constant when degree of superheating increases in HT and LT system
4. The results show that COP and exergetic efficiency increases when degree of subcooling increases in all three cases as discussed above.
5. Also cost, availability and other properties of refrigerant R-134a are more preferable than R-290 for high temperature side so we can suggest pair of R134a-R23 for law temperature application.

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