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Effects of Variation in Evaporator and Condenser Temperature on Cascade Condenser Temperature, COP and Second Law Efficiency of a Cascade Refrigeration System

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ABSTRACT

In the present study, the effects of variation in evaporator and condenser temperatures on first and second law efficiency of the cascade system using NH_3/CO_2 (Ammonia-Carbon dioxide) and C_3H_6 /CO_2 (Propylene -Carbon dioxide) pairs have been carried out. The optimum temperature in cascade condenser temperature corresponding to maximum exergetic efficiency is also determined under for these conditions. It is observed that maximum COP and maximum exergetic efficiency occur at the same cascade condenser temperature. It is observed that optimum cascade temperature increases with increase in evaporator temperature, condenser temperature and approach in cascade condenser. The optimum temperature in cascade condenser for C_3H_6/CO_2 pair is higher than that for NH_3/CO_2 pair. NH_3/CO_2 pair offers better exergetic efficiencies at optimum cascade condenser temperature than C3H6/CO2 pair. This also means that overall exergy destruction in NH_3/CO_2 pair is less than C_3H_6/CO_2 pair.

Keywords:*Cascade Refrigeration System; Exergy; Optimum Temperature in Cascade Condenser; Natural Refrigerants; NH_3/CO_2; C3H_6 /CO_2*

1.0Introduction

Low temperature refrigeration systems are normally required in the temperature range from -30 °C to -100°C in various industries such as pharmaceutical, food , chemical, blast freezing and liquefaction of gases. The application of multi-stage vapour compression refrigerating systems is not desirable for attaining very low temperatures due to the solidification temperature of the refrigerant, low evaporator pressure, enormously large specific volume and difficulties encountered in the operation of mechanical equipment such as compressor with the use of a single refrigerant. These problems are usually overcome by adopting a cascade refrigeration system where two or more independent vapour compression systems are cascaded.

Gupta [1] has numerically optimized the cascaded refrigeration-heat pump system for maximum overall COP and minimum operating costs with refrigerants R-12 in high temperature circuit and R-13 in low temperature circuit. Kanoğlu [2] accomplished the exergy analysis of the multistage cascade refrigeration cycle used for natural gas liquefaction. The relations for the total exergy destruction, exergetic efficiency and minimum work requirement for the liquefaction of natural gas in the cycle are developed. It was shown that the minimum work depends only on the properties of the incoming and outgoing streams of natural gas, and it increases with decreasing liquefaction temperature. Rattsand Brown [3] performed the cascading of an ideal vapour compression cycle for determining the optimal intermediate temperatuhres based on the entropy generation minimization method. Agnew and Ameli [4] optimized two stage cascade refrigeration system for minimum power consumption and a given refrigeration rate using finite time thermodynamics approach for refrigerants R717 and R508b in high temperature circuit and low temperature circuit respectively. This pair was found to exhibit better performance in comparison to R12 and R13 pair. Nicola et al. [5] carried out the first law performance of a cascade refrigeration cycle, operating with ammonia in high temperature circuit and blends of CO2 and HFCs in low temperature circuit, for those applications where temperatures below triple point of CO2 (216.58 K) are needed. Their results show that the R744 blends are an attractive option for the low-temperature circuit of cascade systems operating at temperatures approaching 200 K. Bhattacharyya et al. [6] carried out the analysis of a cascade refrigeration

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system for simultaneous heating and cooling with a CO2 based high temperature cycle and C3H8 (Propane) based low temperature cycle. They predicted the optimum performance of the system with variation in the design parameters and operating variables. This cascaded system can operate simultaneously between refrigerating space temperature of -40°C and a heating output temperature of about 120°C. Moreover, propane vindicates itself as a better refrigerant than ammonia due to its non-toxic nature. However, its flammability remains a concern. Lee et al. [7] optimised condensing temperature of a two stage cascade refrigeration system for ammonia and carbon dioxide for maximization of COP and minimization of exergy loss. It was deduced that optimal condensing temperature increased with condensation and evaporation temperatures. The effects of sub-cooling and superheating were not taken into the consideration.

The computation of exergetic efficiency was also not performed. Kruse and Rüssmann [8] investigated the COP of a cascade refrigeration system using N2O (Nitrous oxide) as refrigerant for the low temperature cascade stage and various natural refrigerants like NH3, C3H8, propene, CO2 and N2O itself for the high temperature stage. They compared its result with a conventional R23/HFC134a cascade refrigeration system for heat rejection temperatures between 25 to 55 °C. They concluded that by substituting the lower stage refrigerant R23 by N2O practically achieved the same energetic performance with high stage fluids R134a, ammonia and hydrocarbons. Niu and Zhang [9] carried out the experimental study of a cascade refrigeration system with R290 in high temperature circuit and a blend of R744/R290 in low temperature circuit. The performance of the blend was compared with R13 in low temperature circuit.

The blend showed good cycle performance compared with R13 and is considered as a promising alternative refrigerant to R13 when the evaporator temperature is higher than 201 K. Getu and Bansal [10] carried out the energy analysis of a carbon dioxide–ammonia (R744/R717) cascade refrigeration system. Their study involved the examination of the effects of evaporating, condensing and cascade condenser temperatures, sub-cooling and superheating in both high and low temperature circuits on optimum COP. They employed a multi-linear regression analysis and developed mathematical expressions for maximum COP, the optimum evaporating temperature of R717 and the optimum mass flow ratio of R717 to that of R744 in the cascade system. Their study did not include the exergy analysis approach to achieve maximum exergetic efficiency. Bhattacharyya et al. [11] carried out the analysis of an endoreversible two-stage cascade cycle and analytically obtained the optimum intermediate temperature for maximum exergy and refrigeration effect.

They also developed a comprehensive numerical model of a trans-critical CO2/C3H8 cascade system and verified the theoretical results. Bansal and Jain, [12] reviewed the literature on cascade refrigeration system.

They reported that a cascade refrigeration system is normally required for producing low temperatures ranging from (-)30°C to (-)100°C for various industries such as pharmaceutical, food, chemical, blast freezing and liquefaction of gases. The refrigerants specified for use in high temperature circuit are HCFC22, HFC134a, R507A, ammonia, propane, and propylene whereas carbon dioxide, HFC23 and R508B are suitable for use in low temperature circuit. The refrigerant pairs that have received the most attention in recent years are R717/R744 (ammonia / carbon-dioxide) and R1270/R744 (propylene / carbon-dioxide) for applications down to (-) 54°C. Dopazo et al. [13] carried out theoretical analysis of a NH3/CO2 cascade refrigeration system for cooling applications at low temperatures. The results have been presented for optimization of coefficient of performance in the evaporation temperature range (-)55°C to (-)30°C in low temperature circuit, 25 to 50°C condensation temperature in high temperature circuit and (-)25 to 5°C in cascade condenser. The approach temperature was varied between 3-6°C.

The effect of compressor isentropic efficiency on system COP is also examined. The results show that, when following both exergy analysis and energy optimization methods, an optimum value of cascade condenser temperature is achieved. However in this study, effect of sub-cooling and superheating for determining the optimum cascade condenser temperatures is not included.

Thus from literature review it is obvious that natural refrigerants are attracting the interest of scientists and a lot of work is being done in this area. The refrigerant pairs which have garnered the attention are NH3 (R717) and R508b, R717 and R744, R744 and R290 and R717 and blends of R744 and HFCs and R717 and C3H6 (R1270). Moreover, the studies cited above pertain to energy analysis and in very few studies the exergy analysis of cascade systems has been presented. In the studies pertaining to cascade system, analysis of R1270/R744 is not presented.

Hence in the present study the effect of variation in evaporator and condenser temperatures on COP and second law efficiency of a cascade refrigeration system is investigated using refrigerant pairs NH_3/CO_2 (R717/R744) and C_3H_6/CO_2 (R1270/ R744).

2.0 Description Of Cascade Refrigeration System

Fig. 1(a) schematically represents a two stage cascade system and Fig. 1(b) presents the corresponding pressure enthalpy diagram. This refrigeration system comprises two separate refrigeration circuits- the high-temperature circuit (htc) and the low-temperature circuit (ltc). Each circuit has a different refrigerant suitable for that temperature with lower temperature units progressively using lower boiling point refrigerants.

The lower boiling point refrigerant will have higher saturation pressure at low temperatures that keeps the ingress of air under control and requires a smaller compressor for the same refrigerating effect due to higher density of suction vapours.

The circuits are thermally connected to each other through a cascade-condenser, which acts as an evaporator for the 'htc' and a condenser for the 'ltc'. Fig. 1(a) indicates that the condenser in this cascade refrigeration system rejects heat 'Q_C' from the condenser at condensing temperature '*Tc*' to its warm coolant or environment at temperature '*T0*'.

The evaporator of the cascade system absorbs a refrigerated load '*Q*e' from the cold refrigerated space at '*Tr*' to the evaporating temperature 'Te'. The heat absorbed by the evaporator of the 'ltc' plus the work input to the 'ltc' compressor equals the heat absorbed by the evaporator of the 'htc'.

'Tmc' and 'Tme' represent the condensing and evaporating temperatures of the cascade condenser, respectively. Approach is designated as '*A*' and it represents the difference between the condensing temperature (*Tmc*) of 'ltc' and the evaporating temperature (*Tme*) of 'htc'.

The evaporating temperature (*Te*), the condensing temperature (*Tc*), and the temperature difference in the cascade-condenser (*A*) are three important design parameters of a cascade refrigeration system.

3.0 Thermodynamic Analysis Of Cascade Refrigeration System

The thermodynamic analysis of the two stage cascade refrigeration system involves the application of principles of mass conservation, energy conservation and exergy balance.

3.1 Mass balance

The mass flow rates are and in *'mltc'* and 'm*htc*' and *'htc*' respectively.

3.2 Energy balance

The energy balance across evaporator is given by:

$$\dot{Q}_e = \dot{m}_{ltc}(h_1 - h_4) \tag{1}$$

Energy balance across cascade condenser is given by:

$$\dot{m}_{ltc}(h_2 - h_3) = \dot{m}_{htc}(h_5 - h_8) \tag{2}$$

Power required to operate the compressors is given by:

$$\dot{W}_{comp\_total} = \dot{W}_{comp\_ltc} + \dot{W}_{comp\_htc} = \dot{m}_{ltc}(h_2 - h_1) + \dot{m}_{htc}($$

Coefficient of performance of cascade refrigeration system is given by:

$$COP = \frac{\dot{Q}_e}{\dot{W}_{comp\_ltc} + \dot{W}_{comp\_htc}} \tag{4}$$

3.3 Exergy balance

The second law of thermodynamics derives the concept of exergy, which always decreases due to thermodynamic irreversibility. Exergy [14] is defined the measure of usefulness, quality or potential of a stream to cause change and an effective measure of the potential of a substance to impact the environment. When the kinetic and potential energies are neglected, specific exergy of a fluid stream [15] can be defined as:

$$e = (h - h_0) - T_0(s - s_0) \tag{5}$$

wheree is the specific exergy of the fluid at temperature T. The terms h and sare the enthalpy and entropy of the fluid, whereas, hoand so are the enthalpy and entropy of the fluid at environmental temperature (or dead state temperature) To(is in all cases absolute temperature is used in K). According to Bejan et al. [16], the exergy balance applied to a fixed control volume is given by the equation (6).

$$\sum \dot{m}_i e_i - \sum \dot{m}_e e_e + \dot{Q}\left(1 - \frac{T_0}{T}\right) - \dot{W} - \dot{ED} = 0 \tag{6}$$

or $$\sum \dot{E}_i - \sum \dot{E}_e + \dot{Q}\left(1 - \frac{T_0}{T}\right) - \dot{W} - \dot{ED} = 0 \tag{7}$$

The first two terms are exergy input and output rates of the flow, respectively. The third term is the exergy associated with heat transfer.Q, which is positive if it is entering into the system.

It is can also be regarded as work obtained by Carnot engine operating between T and T0, and is therefore equal to maximum reversible work that can be obtained from heat energy .

Q .Wis the mechanical work transfer to or from the system, and the last term (ED) is exergy destroyed due to the internal irreversibilities. The principle exergy destruction factors in a process are friction, heat transfer under temperature difference and unrestricted expansion. Using equation (6) and (7), total exergy destruction in system components have been calculated.

3.3.1 Exergeticefficiency

The exergetic efficiency is the ratio between exergy in product to the exergy in fuel and is given by equation (16):

$$\eta_{e_x} = \frac{Exergy\ inproduct}{Exergy\ offuel} = \frac{\dot{EP}}{\dot{EF}} = \frac{\dot{Q}_e\left|(1-T_0/T_r)\right|}{\dot{W}_{comp\_ltc}+\dot{W}_{comp\_htc}} \quad (8)$$

or $$\eta_{e_x} = \frac{\dot{Q}_e\left|(1-T_0/T_r)\right|}{\dot{W}_{comp\_total}} = \frac{COP_{crs}}{COP_{rr}} \quad (9)$$

whereCOPcrsis coefficient of performance of cascade refrigeration system and COPrris the coefficient of performance of reversible refrigerator operating between dead state temperature T0 and Tr .

4.0 Results and Discussion

A computational model is developed for carrying out the energetic and exergetic analysis of the cascade system using Engineering Equation Solver software (Klein and Alvarado[19]). The input data specified below, for the computation of results shown in figures (2) through (9) is referred from Bansal and Jain [12]:

1. Refrigeration capacity ($\dot{Q}_e$) : 1 TR
2. Sub cooling of refrigerant leaving *'htc'* condenser ($\Delta T_{sb,c}$) : 5 °C
3. Superheating of suction vapour in *'ltc'* evaporator ($\Delta T_{sh,e}$) : 10 °C
4. **Isentropic efficiency of compressors (η_{comp}) : 70 %**
5. **Difference between evaporator and space temperature : (Tr-Te) = 10 °C**
6. **Evaporator temperature (T_e) (in steps of 5°C) : -55°C to -15 °C**
7. **Condenser temperature (T_c) (in steps of 10 °C) : 30 °C to 60 °C**
8. **Dead state temperature (T_0) and pressure (P_0) are 25 °C and 1.01325 bar respectively.**
9. Reference enthalpy (h_o) and entropy (s_o) of the working fluids have been calculated corresponding to the dead-state temperature (T_0) of 25°C.
10. **Heat losses and pressure drops in connecting lines and various components are neglected.**

Figs. 2(a) and (b) represent the comparison of present results obtained using the computer code developed for performance analysis of two stage cascade system with research of Bansal and Jain [12]. In Fig. 2(a) the variation of COP versus approach in cascade condenser is shown and in Fig. 2(b) variation of COP versus temperature in cascade condenser (Tcc) is presented. It is observed that results calculated using the present model are in agreement with the theoretical results reported by Bansal and Jain [12] for ammonia / carbon dioxide pair and the difference in results is less than 0.5%. The increase in approach causes a drop in COP because of increase in cascade condenser temperature and hence pressure ratio across compressor in 'ltc' increases thereby increasing the compression work in 'ltc'. This enhancement of compression work causes a reduction in COP in 'ltc' and COP of the cascade system also.

The variation of COP with temperature in cascade condenser shows that there exists a maximum value of COP corresponding to which cascade condenser temperature is optimum. This happens because the pressure ratio of 'ltc' compressor increases with increase in cascade condenser temperature causing a reduction in COP in 'ltc' because of increase in compressor work in 'ltc' whereas the reverse happens in 'htc' and hence there exists an optimum cascade condenser temperature 'Tcc_opt' corresponding to which total compression work is minimum and hence COP is maximum. The results of propylene/carbon-dioxide are also shown in this figure and it can be observed that the COP curve is identical for this pair of refrigerants however the COP offered is lower in comparison to ammonia/ carbon-dioxide pair.

4.1 Effect of cascade condenser temperature

Fig. 3 presents the variation of exergetic efficiency and total exergy destruction versus cascade condenser temperature. It is observed that total exergy destruction decreases up to certain cascade condenser temperature and further increases with increase in cascade condenser temperature. The exergetic efficiency shows a reverse trend in comparison to total exergy destruction. The reason for such a behaviour of exergetic efficiency can be explained on the basis of total compressor power required in 'ltc' and 'htc'. The total compressor power required is lowest at a particular cascade

capacity ($\dot{Q}_e$) is constant and the term $\left(1-\frac{T_0}{T_r}\right)$ is also constant since both dead state temperature (T_0) and cold room temperature (T_r) are constants. Hence the input exergy given by $\dot{Q}_e\left(1-\frac{T_0}{T_r}\right)$ is a constant value. Thus exergetic efficiency given by $\eta_{e_x} = \frac{\left|\dot{Q}_e\left(1-\frac{T_0}{T_r}\right)\right|}{\dot{W}_{comp\_total}}$ is highest at a specific cascade condenser temperature corresponding to which total compressor power required is lowest. This specific temperature, corresponding to which the exergetic efficiency is highest, is optimum cascade condenser temperature. It is observed from Figs. (2) and (3) that the optimum cascade condenser temperature corresponding to maximum COP and maximum exergetic efficiency is identical. Fig (3) also depicts that R717/R744 shows better exergetic efficiency as compared to R1270/R744.

The Influence of Various Design Parameters which affect the optimum Cascade condenser temperature, maximum COP and maximum exergetic efficiency are (i) Evaporator temperature (ii) Condenser temperature (iii) Approach in cascade condenser (iv) Isentropic efficiencies of compressors in ‘ltc’ and ‘htc’ (v) Sub-cooling of refrigerant exiting condenser in ‘htc’ and (vi) Superheating in evaporator in ‘ltc’.

In the present study the effect of evaporator and condenser temperature is considered.

Table 1: Comparison of Optimum Cascade Condenser Temperature, COP Max And Maximum Exergetic Efficiency of R717/744 And R1270/R744 Pairs

A=0 °C, Tc = 50 °C, Te= -45°C							
η_comp		Tcc (°C)		COP_{max}		η_{ex_max}	
Ltc	*htc*	R717/ R744	R1270/ R744	R717/ R744	R1270/ R744	R717/ R744	R1270/ R744
1	1	-13.02	-7.43	1.83	1.69 (-7.6%)	0.461	0.425 (-7.8%)
0.8	1	-22.39	-16.18	1.68	1.53 (-8.9%)	0.423	0.385 (-8.98%)
1	0.8	-4.95	-0.32	1.58	1.47 (-6.9%)	0.398	0.371 (-6.8%)
0.8	0.8	-14.16	-8.11	1.43	1.31 (-8.4%)	0.359	0.331 (-7.8%)
0.7	0.7	-15.10	-8.66	1.23	1.13 (-8.13%)	0.31	0.284 (-8.39%)

Fig 1: Cascade Cycle

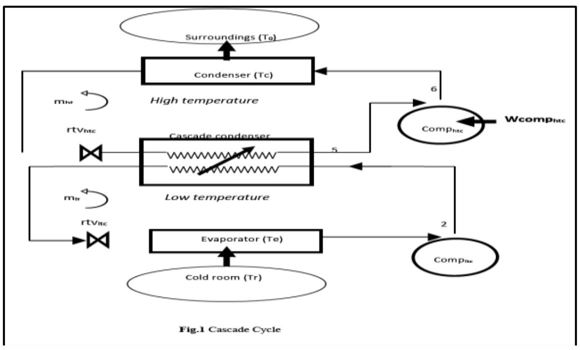
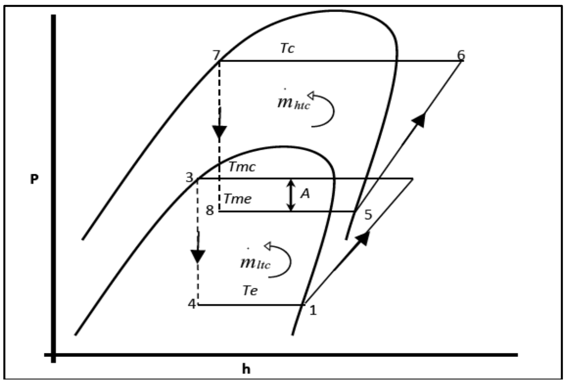


Fig.1 Cascade Cycle

Fig 1: (b) P-h Diagram of Cascade Refrigeration System



4.2 Effect of evaporator temperature

Fig. (4) shows the variation of optimum cascade condenser temperature and maximum COP with evaporator temperature and the influence of isentropic efficiency of ‘ltc’ and ‘htc’ compressors on optimum temperature in cascade condenser and maximum COP. It is evident from this Fig. that the increase in evaporator temperature increases the optimum temperature in cascade condenser and maximum COP. The increase in optimum cascade condenser temperature is attributed to decrease in overall working temperature range and it also reduces the pressure ratio in ‘ltc’ and ‘htc’. Hence compressor works decreases and COP of the cascade system increases. It is observed that decrease in isentropic efficiency of the ‘ltc’ compressor from 1 to 0.8 (keeping the isentropic efficiency of compressor in ‘htc’ = 1) causes the optimum cascade condenser temperature to decrease whereas the reverse happens in case when isentropic efficiency of the ‘htc’ compressor decreases to 0.8 from 1 (keeping the isentropic efficiency of the ‘ltc’ compressor =1). This effect is nearly compensated when the efficiencies of both ‘ltc’ and ‘htc’ compressors reduce from 1 to 0.8 and the optimum cascade condenser temperature obtained in this particular case is very near (but lower) to the optimum cascade condenser when assuming isentropic efficiencies of both the compressors are taken as 1. One more observation that is important to highlight here is that 20% reduction in the efficiency of ‘ltc’ compressor causes about 10K drop in optimum cascade condenser temperature as compared to about 8K rise in cascade condenser temperature for the same decrease in efficiency of ‘htc’ compressor.

On the other hand 20% drop in isentropic efficiency of ‘ltc’ compressor causes the system COP to drop by 7.3% to 8.9% as compared to a drop of 13-14% in COP for 20% decrease in isentropic efficiency of ‘htc’ compressor.

The variation in EDRmin and maximum exergetic efficiency is represented in Fig. (5). Two main characteristics of this Fig. are decreasing trend of maximum exergetic efficiency with increase in evaporator temperature and decrease in maximum value of exergetic efficiency with decrease in isentropic efficiency of either of the compressors. In this case also, it is crucial to emphasize that the reduction in isentropic efficiency of 'htc' compressor by 20% causes about 13-14% reduction in maximum value of exergetic efficiency as compared to 7.3-9% reduction when the isentropic efficiency of 'ltc' compressor reduces by same amount.

Thus once again it is confirmed that lowering of isentropic efficiency of compressor in 'htc' (i.e compressor for ammonia) has more damaging effect on system performance as compared to carbon dioxide compressor. The trends of curves of minimum EDR are just opposite to maximum exergetic efficiency curves. This fact is also highlighted in equation (15) given above.

Fig. (6) illustrates the effect of variation in evaporator temperature on optimum temperature in cascade condenser, maximum COP, maximum exergetic efficiency and minimum exergy destruction ratio for R1270/R744 pair.

Fig.7 shows the effect of variation in evaporator temperature on minimum EDR and maximum exergetic efficiency. The trends for optimum cascade condenser temperature, maximum COP, maximum exergetic efficiency and minimum EDR are similar to the trends of these parameters presented in Figs. (4) and (5) for R717/R744. The 20% reduction in isentropic efficiency of 'ltc' compressor (i.e. carbon dioxide compressor) is accountable for lowering both maximum COP and maximum exergetic efficiency by 8.7% and 10.2% corresponding to evaporator temperatures of -35°C and -55°C respectively.

Similar to above, the reduction in maximum values of COP and exergetic efficiency is 13.1% and 12.6% for identical temperature conditions when isentropic efficiency of 'htc' compressor reduces by 20%.

Table 1 presents the comparison of the two pairs of refrigerants considered for various conditions of isentropic efficiencies of 'ltc' and 'htc' compressors.

It is observed that R717/R744 refrigerant pair offers better performance in terms of maximum COP and maximum exergetic efficiency as specified in the table.

The values given in braces show the percentage difference by which the values of maximum COP and maximum exergetic efficiency for refrigerant pair R1270/R744 are lower than the corresponding values for R717/R744.

4.3 Effect of condenser temperature

Figs. (8) and (9) depict the effect of condenser temperature on optimum cascade condenser temperature, maximum COP, minimum EDR and maximum exergetic efficiency respectively for R717/R744.

Simultaneously these Figs. also present the effect of isentropic efficiencies 'ltc' and 'htc' compressors on above mentioned parameters. The optimum temperature increases with increase in condenser temperature. This happens because of increase in overall working temperature range.

The maximum COP of the system reduces since the increase in condenser temperature causes the pressure ratios of the 'ltc' and 'htc' compressors to increase and hence the power input increases thereby reducing the maximum COP.

The effect of isentropic efficiencies of compressors on optimum temperature in cascade condenser and maximum COP are similar to the trends observed in the case of variation of evaporator temperature depicted in Fig. (4) and explained in corresponding para.

Fig. (9) shows that maximum exergetic efficiency reduces with increase in condenser temperature.

This decreasing trend of exergetic efficiency is achieved because of increase in input exergy i.e. total compressor power required for the same output exergy of the corresponding to a constant cooling capacity.

The effect of isentropic efficiencies of the compressors is similar to the trends that were achieved in case when evaporator temperature was varied (Refer Fig. (5)).

The variation of optimum temperature in cascade condenser and maximum COP for R1270/R744 is illustrated in Fig.(10). The variation in maximum exergetic efficiency and minimum EDR are presented in Fig.(11).

The explanation of trends of these curves is similar as has been discussed for R717/R744. The comparison of the optimum temperatures, maximum COP and maximum exergetic efficiencies for a particular set of data is already presented for these two pairs of refrigerants in Table (1).

5.0 Conclusions

In this study, a detailed energy and exergy analysis of a two stage cascade refrigeration system has been carried out for R717/R744 and R1270/R744 pairs of the refrigerants for the computation of optimum cascade condensertemperature.

The effects of various parameters are also computed. The following conclusions can be drawn from the above analysis:-

1 The COP and exergetic efficiency increase with increase in cascade condenser temperature, achieve maximum values at a particular cascade condenser temperature and decrease with further increase in cascade condenser temperature. This specific cascade condenser temperature, corresponding to which both COP and exergetic efficiency are maximum, is designated as optimum cascade condenser temperature. The optimum cascade condenser temperature also corresponds to the condition of minimum total exergy destruction in the system.

2 The optimum value of cascade condenser temperature increases with increase in evaporator and condenser temperatures. The decrease in isentropic efficiency of the 'ltc' compressor causes the optimum cascade condenser temperature to reduce whereas optimum cascade condenser temperature increases with decrease in isentropic efficiency of the 'htc' compressor. The maximum COP increases with increase in evaporator temperature whereas reverse happens when condenser temperature increases. The maximum exergetic efficiency reduces with increase in evaporator and condenser temperatures. The decrease in values of maximum COP and maximum exergetic efficiency is higher when the isentropic efficiency of 'htc' compressor is reduced as compared to the identical decrease in the value of isentropic efficiency of 'ltc' compressor.

3 The values of maximum COP and maximum exergetic efficiency of R717/R744 are 7-9% higher than R1270/R744 pair.

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Nomenclature

A	Approach
COP	
$\dot{E}$	Exergy rate of fluid (kW)
$\dot{ED}$	**Exergy destruction rate (kW)**
EDR	**Exergy destruction ratio**
$\dot{EF}$	Exergy rate of fuel (kW)
$\dot{EP}$	Exergy rate of product (kW)
h	Enthalpy (kJ/kg)
$\dot{m}$	Mass flow rate (kg/s)
P	Pressure (kPa)
$\dot{Q}$	Rate of heat transfer (kW)
s	Entropy (kJ/kg K)
T	Temperature (°C)
Tmc	Condensation temperature in cascade condenser
Tme	Evaporation temperature in cascade condenser
W	Power (W)

Subscript

c	Condenser
cc	Cascade condenser
comp	Compressor
crs	Cascade refrigeration system
e	Evaporator
ex	Exergetic
htc	High temperature circuit
ltc	Low temperature circuit
opt	Optimum
r	Refrigerant, room
rr	Reversible refrigerator
s	Isentropic
rtv, t	Refrigerant throttle valve
total	Total
0	Dead state

Greek letters

$\Delta T_{sb,c}$	Sub-cooling in condenser
$\Delta T_{sh,e}$	Superheating in evaporator
η	Efficiency
δ	Efficiency defect