Exergy based refrigerant selection and simulation of auto refrigeration cascade (ARC) system

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Abstract: This paper deals with the selection of refrigerants and simulation of Auto Refrigeration Cascade (ARC) system, which is generally used for low temperature applications in the range of −40 to −150°C that operates with zeotropic mixture, containing two or more refrigerants with different boiling points. The selection of refrigerants depends on the working pressure and temperature range of operation. In order to have effective phase separation, the minimum difference in the boiling points should be in the order of 40 to 80°C. Moreover, the ODP and GWP values are considered in the process of selection. Initially 52 combinations are selected and then reduced to 20, based on environmental friendliness. A further elimination process is undertaken based on suitable condensing and evaporating temperatures for effective heat transfer in cascade condenser and three refrigerant mixtures, R23/R290, R23/R600 and R125/R600, are selected. Due to some mismatch of physical properties like triple point of low boiling component in CO_2 and HFC 134a mixture at low temperature applications, it is not considered in the refrigerant selection. The mass fractions of above refrigerant pairs are calculated by applying energy balance in the cascade condenser. The above mass fraction and unit refrigeration capacity is used for the simulation of the system. Exergy analysis is used to select the efficient refrigerant pair, which gives overall minimum exergy loss by simulating the system. The exergy loss in different components in the ARC system with R23/R290 mixture (Compressor: 20 and 14.28%, Condenser: 35.01 and 30%, Separator: 16.12 and 10.65%, Cascade condenser: 8.7 and 16%, Expansion valve I: 9.04 and 16%, Expansion valve II: 7 and 15%, Evaporator: 7.3 and 7.3%) is less in comparison with R125/R600 and R23/R600 mixtures and thus an R23/R290 mixture is recommended for this specific application.

Keywords: auto refrigeration cascade system; cascade condenser; exergy; GWP; ODP; zeotropic.

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1 Introduction

A conventional cascade refrigeration system has two or more vapour compression systems; namely, high side and low side, each thermally coupled through a cascade condenser. The evaporator of the high temperature cycle is thermally coupled with the condenser of the low temperature cycle. Each stage has its own performance range; hence, the overall performance decreases with an increase in the number of circuits. To avoid performance degradation in the cascade system, ARC systems can be used for the same application with slightly higher performance. The ARC systems are used to produce low temperatures over the span of -40 to $-150°C$. The components of the refrigerant mixture used in ARC system must form a zeotropic mixture without chemical or physical interaction. In this mixture, the vapour and liquid phases remain different from each other at the equilibrium state. This characteristic of the mixture allows effective fractionation during the circulation of the mixture. The number of refrigerant components in the mixture depends on the final evaporator temperature.

Hence, ARC system can be a better alternative for cascade refrigeration systems used for low temperature applications. This paper deals with the exergy-based selection of refrigerant pairs and simulation of ARC system with selected refrigerant pairs. The performance of different refrigerant pairs is compared with different working environments and exergy losses of individual pairs are simulated and compared.

2 Auto refrigeration cascade (ARC) system

Figure 1 shows the schematic diagram of the auto refrigeration cascade system using a zeotropic mixture containing two refrigerants. The components of an ARC system operating with a two or more component zeotropic mixture basically include a compressor (A), water or air-cooled condenser (B), two or more cascade heat exchangers (D) as per the required evaporator temperature and an evaporator (E). In this case, the higher boiling component is liquefied in the condenser while the lower boiling component remains as vapour at the exit of the condenser. A phase separator (C) is used to separate the liquid and vapour phases. The low boiling vapour is condensed from the refrigeration effect of high boiling liquid, which is throttled (F) before entering the cascade condenser. The condensed low boiling liquid proceeds through another expansion device (G) and into the low temperature evaporator (E) where it cools the air or secondary fluid blown across the coil to obtain the desired low temperature. The refrigerant vapour from the evaporator and cascade condenser returns to the compressor through the suction line after the mixing process. The pressure enthalpy diagram of the ARC system is shown in Figure 2. The high boiling liquid refrigerant is throttled from state 8 to state 9 while condensation of low boiling liquid from state 4 to state 5 occurs in the cascade condenser. The condensed low boiling liquid refrigerant is throttled in the process 5–6 to produce the required low temperature. Thus, the ARC cycle is completed.

Figure 1 Auto Refrigerating Cascade (ARC) system

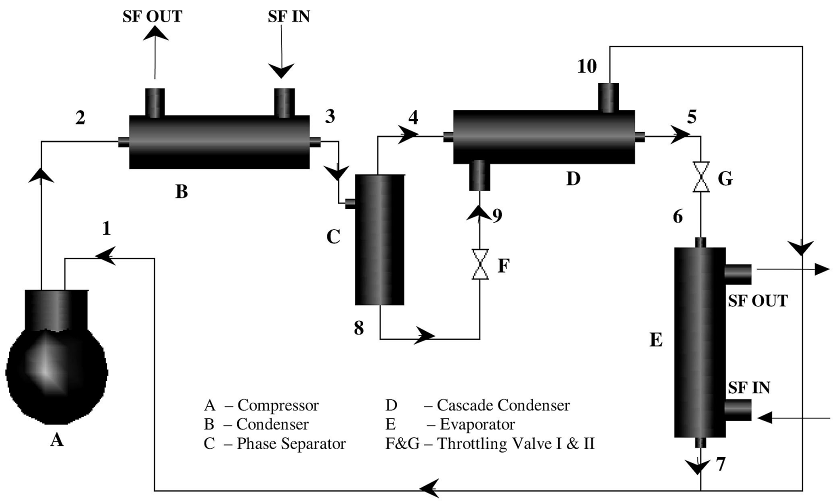
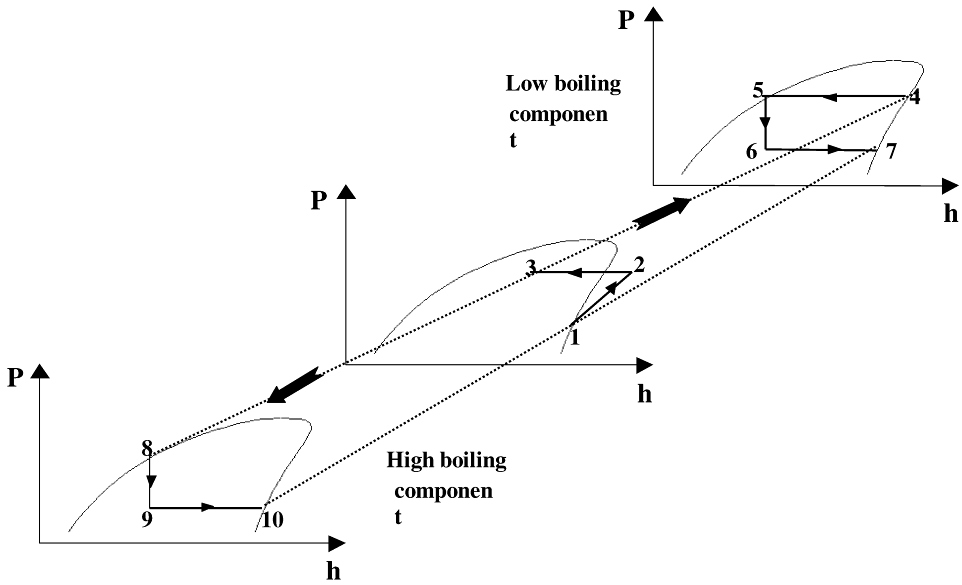


Figure 2 Pressure enthalpy diagram of ARC system



2.1 Refrigerant selection

The refrigerant pair selection for the ARC system is done with a regular approach in as well as a few factors which are specific to the ARC system as follows. The mixture should not form an azeotrope; the difference in boiling point should be in the order of 40 to 80°C and temperature overlapping in the cascade condenser for the same operating pressure must exist to ensure the condensation of a low boiling point component of the mixture (Kim and Kim, 2002; Missimer, 1996). Exergy losses at different state points are calculated and compared. The overall exergetic efficiency is considered for the selection of an efficient pair. Though CO_2 and HFC 134a mixture is used in ARC systems, it is not considered in this study, as the triple point of CO_2 is −56.57°C. The saturation pressure at the triple point is 5.182 bar. Hence CO_2 cannot be used for the applications below –56.57°C. At this temperature, possible minimum suction pressure is 5.182 bar. After the system comes to thermal equilibrium with the surrounding, the equilibrium pressure will be higher than the suction pressure, which demands a higher starting torque of the compressor. Hence, for −70°C applications, a CO_2 and HFC 134a mixture cannot be used.

3 Exergy analysis

Exergy loss is expressed as a measure of irreversibility of a thermodynamic process. The purpose of the exergy analysis is to find irreversibility losses between different state points of the system. The exergy loss at the various components can be calculated as follows, Exergy at a given point,

$$e = (h - h_0) - T_0 \times (s - s_0). \tag{1}$$

Exergy losses are calculated by making exergy balance for each component of the system. Unlike energy balance where the inflow is equal to the outflow (when there is no energy loss or gain), in exergy balance, exergy inflow is always greater than the exergy outflow due to the process irreversibility and the difference gives the exergy loss. This exergy loss has to be reduced to improve the thermodynamic efficiency of the system. Exergy balance is stated as below,

$$Exergy\ in = Exergy\ out + Irreversibility. \tag{2}$$

For a steady state process having a mechanical input 'W', in which a fluid enters and leaves the system at states points 1 and 2 respectively, the exergy loss is written as,

$$e_l = e_1 + W - e_2, \tag{3}$$

where, the changes in potential and kinetic energies are neglected. Exergy losses occurring during a cycle can be calculated by the summation of all the exergy losses in the individual components. The useful heat energy at the evaporator is given by,

$$Q_e = \dot{m}_{Low\ Boiling\ Component} \times (h_7 - h_8). \tag{4}$$

The compressor work or the input to the process is given by

$$W = \dot{m}_{mix} \times (h_2 - h_1). \tag{5}$$

The COP of the system is then calculated as

$$COP = \left(\frac{Q_e}{W}\right). \tag{6}$$

The overall exergetic efficiency of the system is defined as the ratio of exergy absorbed to the compressor work. It is given by

$$\eta_{ex} = \frac{Exergy\ of\ cold\ produced}{Exergy\ of\ work\ input}. \tag{7}$$

The selection of an appropriate dead state is important as it is explicitly used in calculating exergy. Hence changing the dead state will lead to different exergy values. In the present analysis, the reference conditions used are $T_0 = 273.15$ K, h = 200 kJ/kg, s = 1 kJ/kg K as prescribed by IIR. Flow charts for exergy loss calculation and COP with various condensing temperatures are shown in Figures 3(a)–(c) and (f) respectively.

Figure 3a Flow chart to calculate the COP and exergy loss

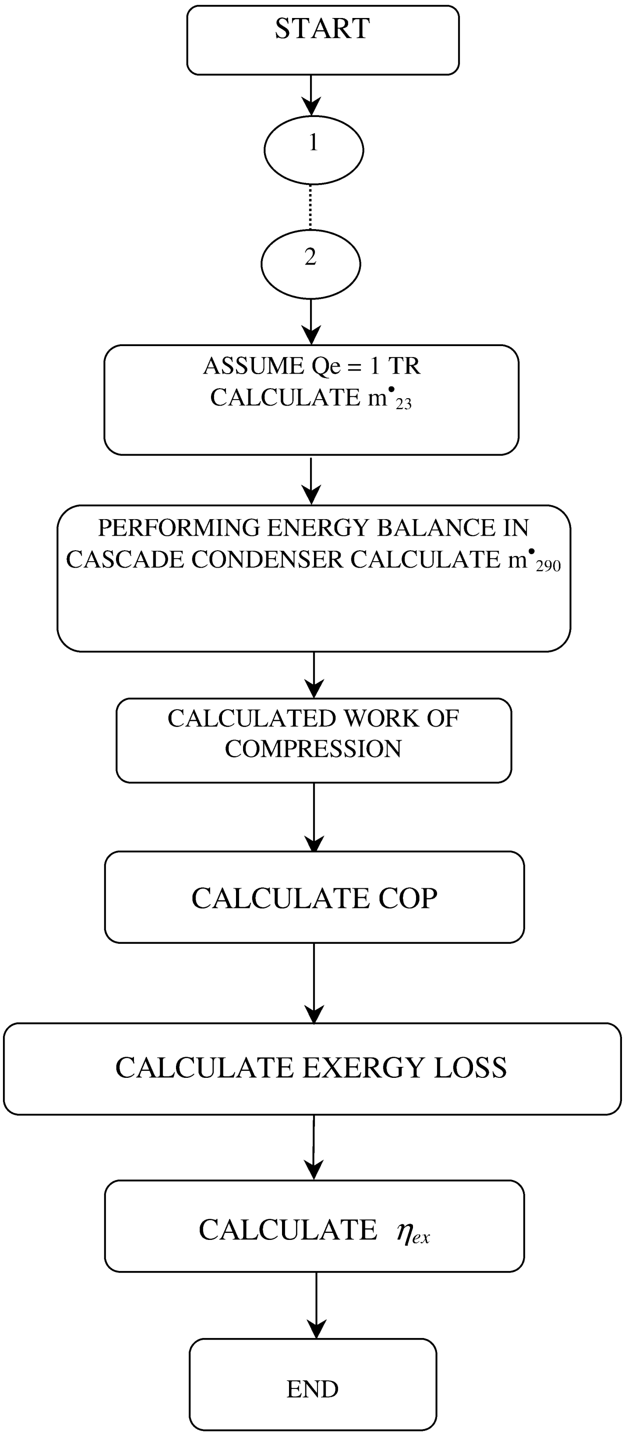


Figure 3b Flow chart to calculate the COP and exergy loss

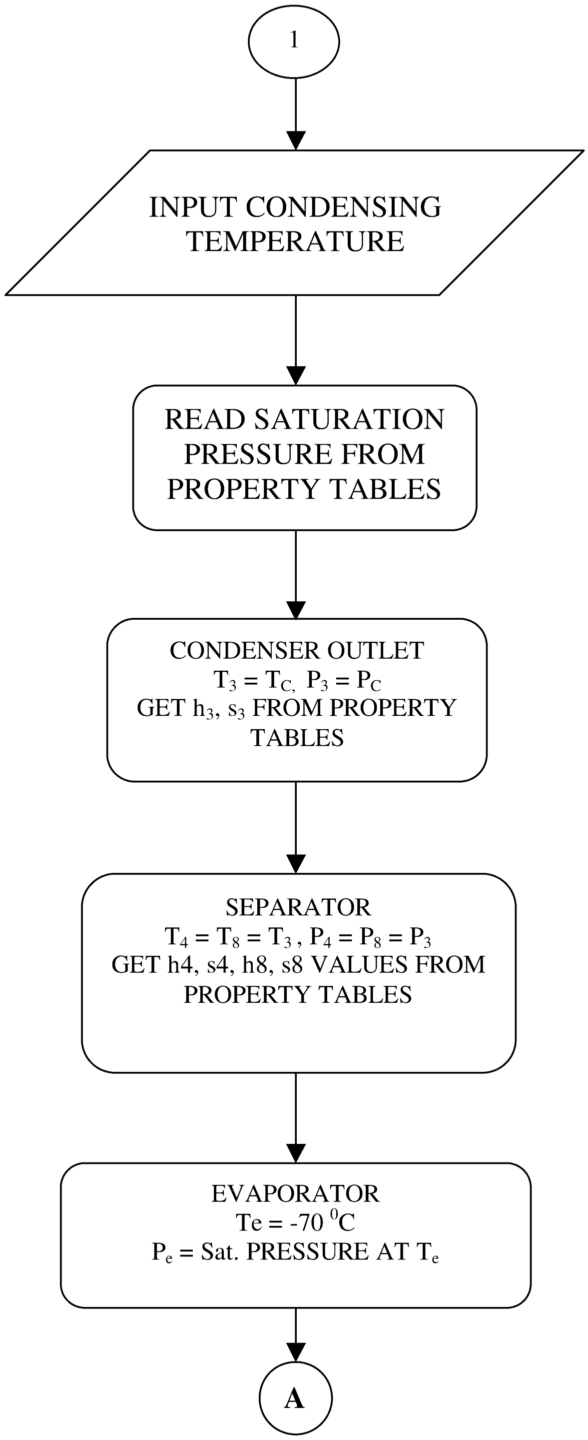


Figure 3c Flow chart to calculate the COP and exergy loss

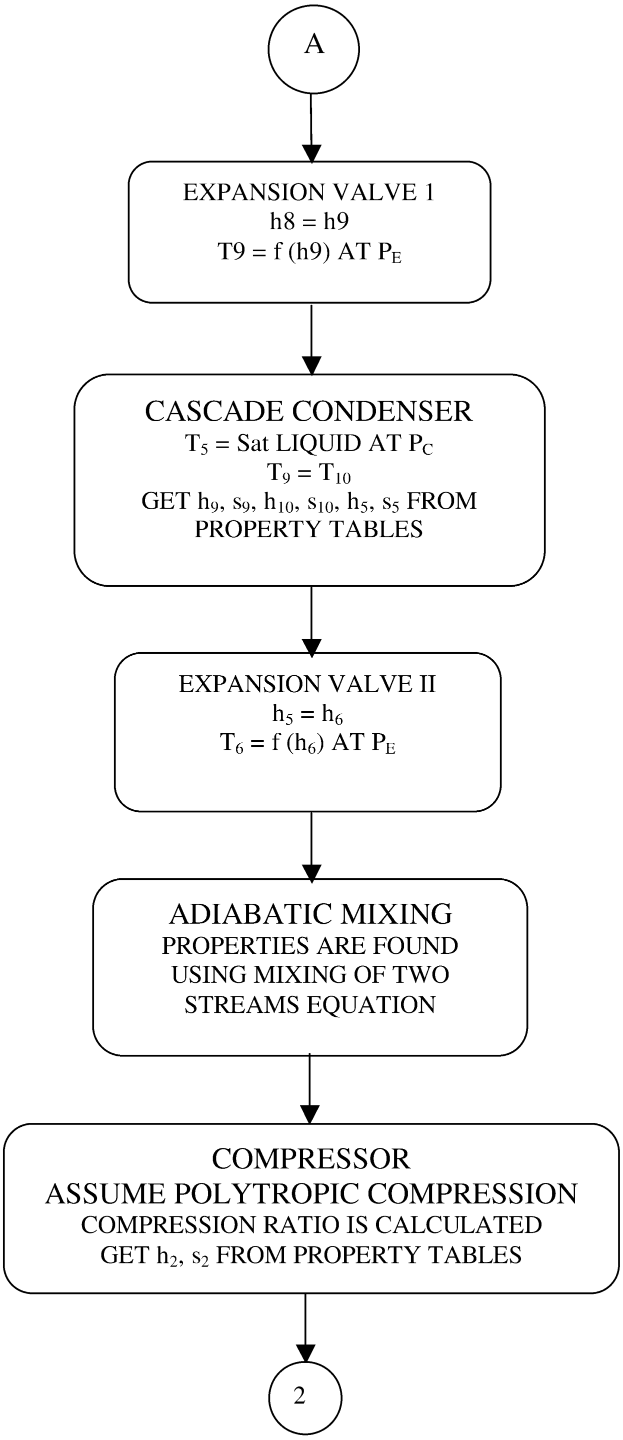


Figure 3d Flow chart to find the cascade condenser length

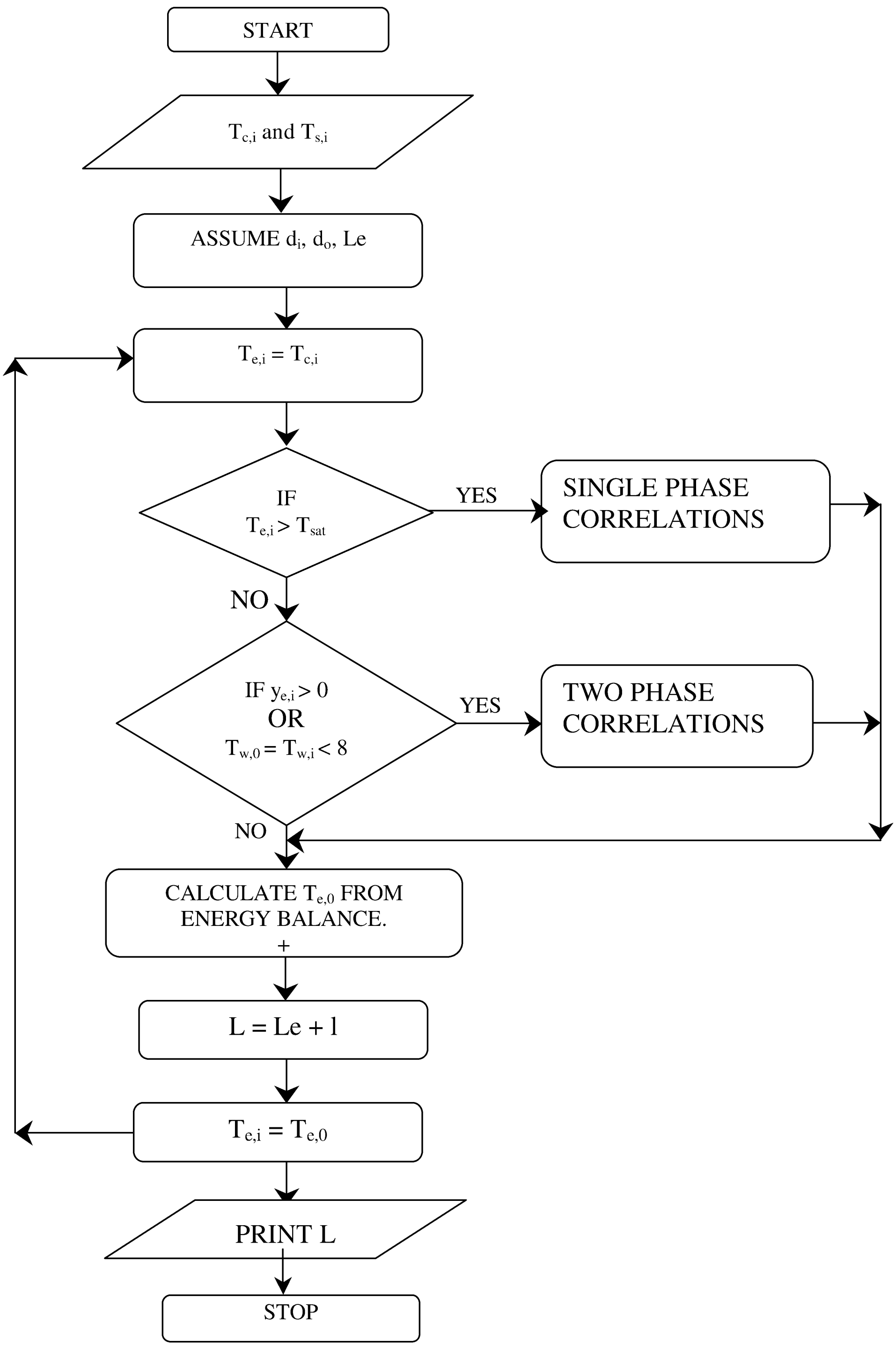


Figure 3e Flow chart to find the length of condenser and evaporator

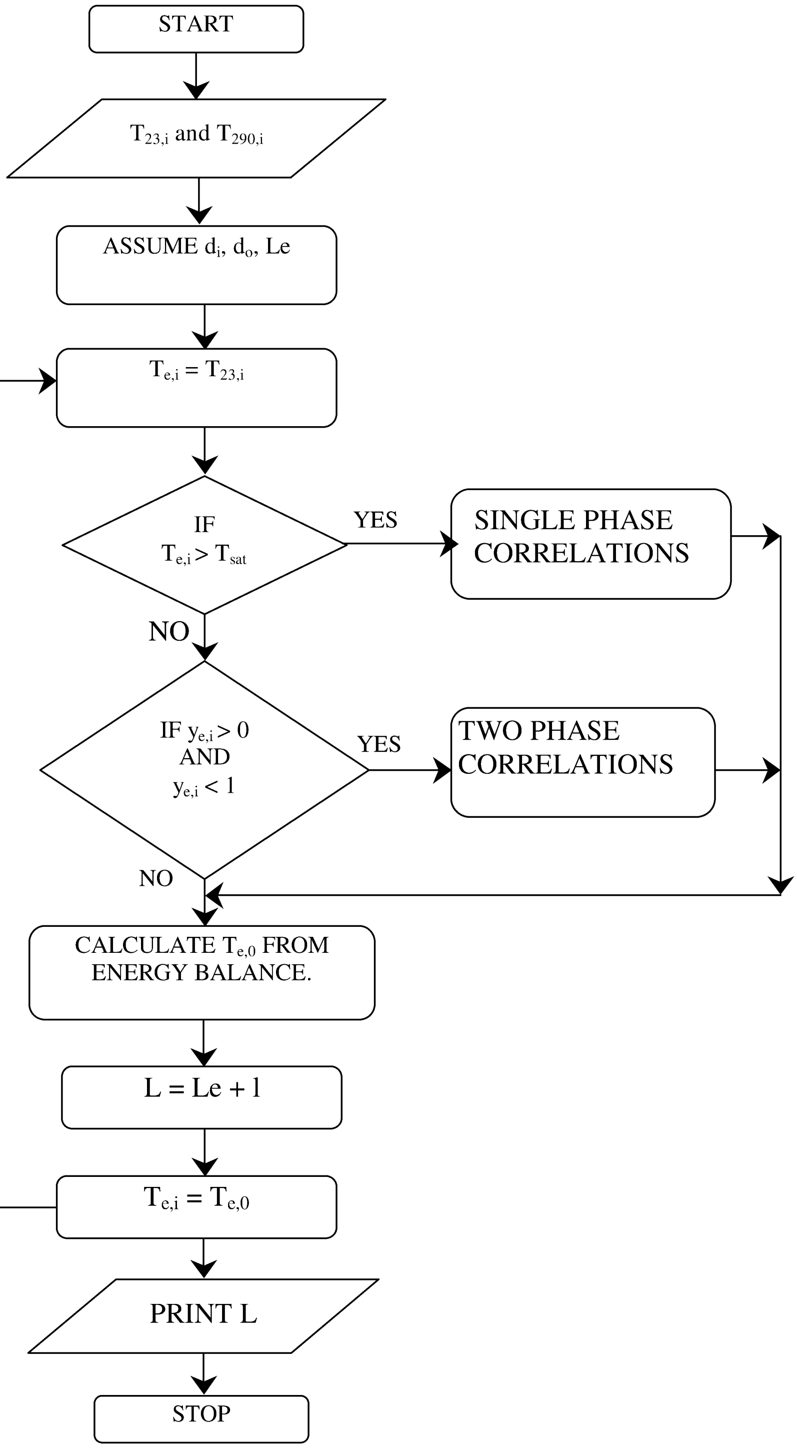
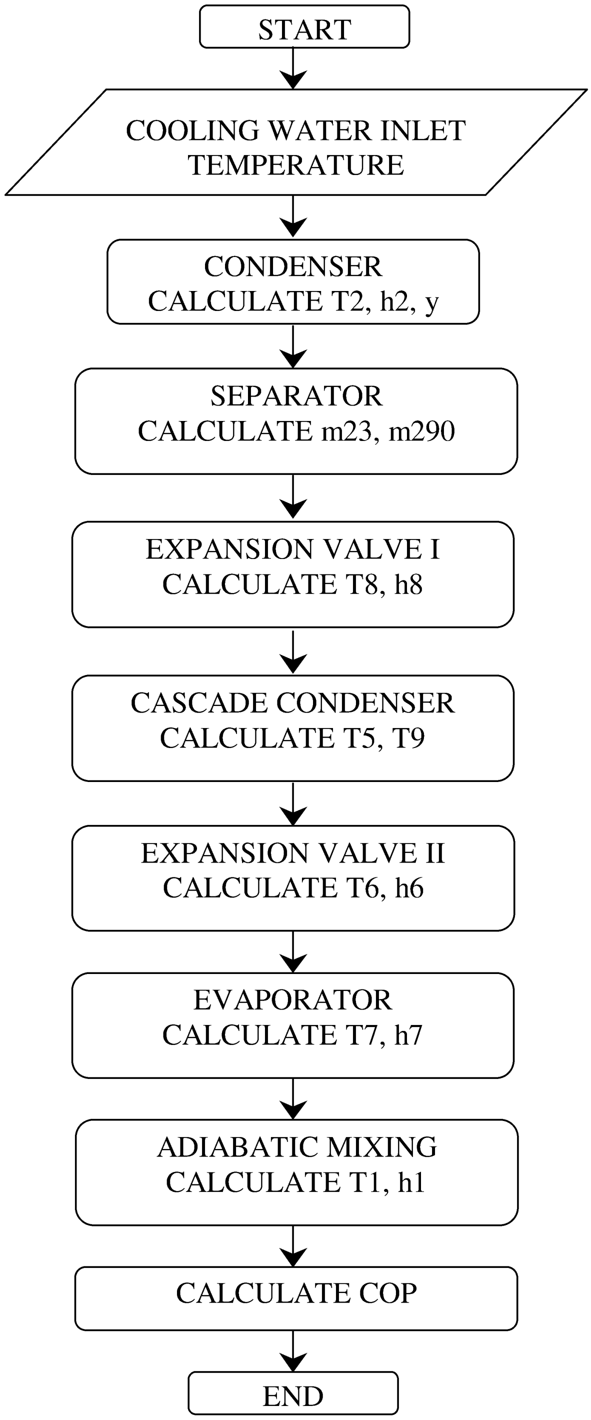


Figure 3f Flow chart to find the COP of the system for various condenser temperatures



3.1 Procedure

To carry out the exergy analysis, a programme is developed in MATHCAD 7 Professional.[1] The property values are taken from REFPROP 6.01.[2] These property values are saved as .prn extension files and used in the programme for calculating the properties at the different state points of the system. Three refrigerant pairs are chosen. They are R23/290, R125/600 and R23/600. The evaporator temperature is fixed as 203 K and the condenser temperature is varied from 303 o 343 K in steps of 10 K.

3.2 Assumptions

The compression process is polytropic having a polytropic index of 1.12 (Domanski, 1992). In the exit of the condenser the entire fraction of high boiling component is condensed. The vapour and liquid phases exist in equilibrium in the separator. Saturated liquid is obtained at the cascade condenser exit. Only the latent heat of evaporation of the higher boiling point component is predominant. Heat gained by a high boiling point component is equal to the heat lost by a low boiling point component; the mass fraction is calculated based on this assumption. Evaporator capacity is 1 TR.

3.3 Best combination of refrigerant pair

The results obtained using the MATHCAD 7[1] programme for exergy analysis of the auto refrigerating cascade system are used for the selection of a refrigerant pair. Based on the results obtained and some properties of the refrigerants, the following conclusions are evolved. The theoretical COP for the R23/290 pair is higher than the R23/600 and R125/600 pairs. The exergetic efficiency of the system is higher for the pair R23/290. The overall exergy loss of the system is lower for the R23/290 pair. The R23/290 pair requires a lesser pressure ratio when compared to the R23/600 pair and the R125/600 pair for the same operating conditions. Compressor efficiency is higher for lower molecular weight mixture i.e. for the R23/290 pair. The butane pair runs in sub atmospheric pressure, which has the problem of moisture entry into the system, leading to choking of the system. Butane, when used as a refrigerant, always needs some superheat before compression in order to avoid wet compression.

3.4 Compressor model

The compression process is defined as polytropic with an index of 1.12 (Domanski, 1992). The mass flow rate of the refrigerant changes is calculated considering the compressor displacement rate, volumetric efficiency, pressure ratio and the specific volume of the gas entering the compressor as given in the equations below

$$\eta_v = 1 - \dot{m} \times \left[\left(\frac{P_2}{P_1} \right)^{\frac{1}{n}} - 1 \right] \tag{8}$$

$$\dot{m} = \frac{V_{comp} \times \eta_v}{v_1} \tag{9}$$

$$Wc = P_1 \times 10^2 \times \nu_1 \times \frac{n-1}{n} \times \left[\left(\frac{P_2}{P_1} \right)^{\frac{n-1}{n}} - 1 \right]. \tag{10}$$

Here '$\dot{m}$' represents the clearance volume ratio and its value is taken as 0.04. The compressor model is shown in the Table 1.

Table 1 ARC system component models

Model	*Input*	*Output*
Compressor model	m_1, T_1, h_1, P_1, P_2	$m_2 = m_1$, $T_2 = f(P_1, P_2, n)$, $h_2 = f(T_2, P_2)$, $W_c = m \times (h_2 - h_1)$
Condenser model	m_1, T_1, h_1, P_1	$m_2 = m_1$, T_2, h_2, P_1, y
Cascade condenser model	R23 R290 m_2 m_1 T_2 T_1 h_2 h_1 P_2 P_1 T_1 h_1 P_1	R23 R290 m_2 m_1 T_3 T_4 h_3 h_4 P_2 P_1 $m_2 = m_1$ T_2 h_2 P_2 y
Evaporator model	T_1, h_1, P_1, Q_{in}, y_1	T_2, $P_2 = P_1$, $h_2 = f(P_2, T_2)$, $m_2 = m_1 = Q_{in}/(h_2 - h_1)$
Separator model	m_i, T_i, P_i, y_i	$P_i = P_1 = P_2$, $T_i = T_1 = T_2$, $m_1 = m_i \times y_i$, $m_2 = m_i - m_1$, $x_1 = f(P_1, T_1)$, $x_2 = f(P_2, T_2)$, $h_1 = f(P_1, T_1)$, $h_2 = f(P_2, T_2)$
Expansion device	m_1, T_1, h_1, P_1, P_2	$m_2 = m_1$, $h_2 = h_1$, $T_2 = f(P_2, h_2)$, $y = f(P_2, h_2)$
Mixing chamber model	m_1, m_2, T_1, T_2, h_1, h_2, $P_1 = P_2$	$m_0 = m_2 + m_1$, $T_0 = f(P, h_0)$, $h_0 = [(m_1 \times h_1) + (m_2 \times h_2)]/m_0$

3.5 Condenser model

The condenser model requires refrigerant inlet conditions (P_2, T_2) for the zeotropic mixture and inlet temperature (T_1) of the cooling medium. The use of zeotropic refrigerant mixture as a working fluid introduces effects like mass transport resistance in both the gas and liquid phases, as well as a change in composition during the condensation process, which, in turn, influences the available driving force. Tord and Lennart (2002) pointed out that the standard calculation methods could overestimate the heat transfer coefficient for condensation of zeotropic refrigerant mixtures by 40%. In this model the heat transfer calculations are carried out as follows,

$$\frac{1}{U} = \frac{A_0}{A_i \times h_w} + \frac{\delta \times A_0}{\lambda_w \times A_i} + \frac{1}{h} + \frac{z}{h_g}. \tag{11}$$

If the above equation is compared with the pure fluid equation usually on the overall heat transfer coefficient, it is obvious that one extra term, z/h_g is added for the zeotropic nature of the working fluid. z is defined as,

$$z = x \times c_{p,g} \times \frac{dT}{dh} \tag{12}$$

where x is the quality and dT/dh is the slope of the condensation curve. For mixtures the heat transfer coefficient h for condensation on a plain tube can be calculated from the correlation given below.

$$h = 1.51 \times \left(\frac{4}{\mu_l}\right)^{\frac{-1}{3}} \times \left(\frac{\rho_l \times (\rho_l - \rho_g) \times g}{(\mu_l)^2}\right)^{\frac{1}{3}}. \tag{13}$$

The heat transfer in a gaseous region is calculated with correlations for pure fluids, but the thermodynamic and transport properties are always estimated at actual composition and temperature of the liquid and vapour phases separately. The condenser model is shown in Table 1. The inlet conditions of the condenser are defined by m_1, P_1, T_1 and h_1 from previous steps of the simulation model. The condenser is divided into smaller elements of 0.01 m length and an initial diameter is assumed. Then considering energy balance across each element the outlet enthalpy is found. The temperature at the outlet of the element is a function of enthalpy and pressure at the corresponding composition. The output of one element acts as the input to the other, whereas the elemental length is added every time. Here the heat rejected from the refrigerant fluid at each element is known. The outlet temperature of the secondary heat transfer fluid can be found, assuming that the refrigerant gains the heat lost by the secondary fluid. Thus, the final length of the condenser is evolved. The flow chart for the calculation of condenser length is shown in Figure 3(d).

3.6 Cascade condenser model

The cascade condenser modelling is similar to the condenser model; wherein all the relations used for condenser can also be used here. The refrigerant mixture, rich in high boiling point refrigerant at low pressure is used to cool the refrigerant mixture, rich in low boiling point refrigerant at high pressure. In this model, the heat transfer coefficients are calculated using correlations applicable to pure fluids but the thermodynamic and transport properties are taken as for mixture properties at their respective composition. The cascade condenser model is shown in Table 1 and flow chart is shown in Figure 3(e).

3.7 Evaporator model

The evaporator is modelled using the same correlations as the condenser model. The refrigerant is assumed to leave the evaporator as saturated vapour without super heating. So the model incorporates only for the two-phase flow regime since no super heat is taken into account. The evaporator model is shown in Table 1. Neglecting the pressure drop in the evaporator, the exit temperature and enthalpy are obtained from the fluid property tables. The inlet condition of the evaporator i.e. temperature, quality is taken from the cascade condenser simulation programme. The evaporator capacity is taken as 1 TR.

3.8 Separator model

The separator is used to separate the two-phase zeotropic mixture into saturated vapour and saturated liquid. Because of the density difference, the vapour and the liquid phases are separated. A regular receiver can be used as the phase separator. The zeotropic vapour and zeotropic liquid are separated using their density difference. The liquid is taken from the bottom of the receiver and vapour is from the top. From the continuity equation and first law of thermodynamics: (A steady-state steady flow process),

$$\dot{m}_i = \dot{m}_1 + \dot{m}_2 \tag{14}$$

$$(\dot{m}\,h)_i = (\dot{m}\,h)_1 + (\dot{m}\,h)_2. \tag{15}$$

In this study, the pressure drop in the separator is assumed to be zero and the separator efficiency to unity. The refrigerant is in equilibrium and therefore,

$$P_i = P_1 = P_2 \tag{16}$$

$$T_i = T_1 = T_2. \tag{17}$$

From the inlet quality y_i, the outlet saturated vapour flow rate m_1 is given by

$$\dot{m}_1 = \dot{m}_i \times y_i. \tag{18}$$

The outlet saturated liquid flow rate is given by

$$\dot{m}_1 = \dot{m}_i \times (1 - y_i). \tag{19}$$

The separator model is shown in Table 1. From the composition x_i, temperature T_i and the pressure P_i at the inlet of the separator, the composition of the saturated vapour and saturated liquid are obtained from mixture properties. The enthalpy h_1, h_2 are obtained from the pressure and temperature of the respective refrigerants at the corresponding composition.

3.9 Expansion device model

The expansion device is used to control the flow of refrigerant liquid from the high-pressure side to the low-pressure side of the refrigeration system. In this study only the thermodynamic properties of the expansion valve are emphasised. From the continuity equation and the first law of thermodynamics, $\dot{m} = \dot{m}_1 = \dot{m}_2$ and $(\dot{m}\,h)_1 = (\dot{m}\,h)_2$. The process in the expansion valve is considered isenthalpic. The expansion device model is shown in Table 1. The quality of the refrigerant at the outlet of the expansion device is calculated as

$$y = \frac{h - h_f}{h_g - h_f}. \tag{20}$$

The h_f, h_g are taken at the evaporator pressure for the corresponding composition of refrigerant mixture.

3.10 Adiabatic mixing

The mixing of two different refrigerant streams is considered. The continuity equation and the first law of thermodynamics gives,

$$\dot{m} = \dot{m}_1 + \dot{m}_2 \tag{21}$$

$$(\dot{m}\,h)_1 + (\dot{m}\,h)_2 = (\dot{m}\,h). \tag{22}$$

The mixing chamber model is shown in Table 1. Assuming the inlet pressure as $P_1 = P_2$, the mixing pressure is $P_0 = P_1 = P_2$. The mixing temperature is obtained from the working fluid properties at the pressure P_0 and the enthalpy h_0.

3.11 Heat exchanger design methodology

Based on the background of the technical papers of Tord and Lennart (2002), Young Shin and Kim (1997), Belghazi et al. (2001) and Stephan (1999) a programme is written on Visual ++[3] employing a finite difference method. The output of the programme gives the length of the heat exchanger coil required for the ARC system working with the zeotropic mixture R 23/290. The overall dimensions of the heat exchanger are obtained from the programme. The transport and thermodynamic properties are taken from REFPROP 6.01[2] software for the particular composition. The heat exchanger details obtained from simulation programme are listed below:

- condenser (tube in tube): Coil length – 10.45 m, Inner tube diameter (I_0) – 6 mm and Outer tube diameter (D_0)– 15 mm
- cascade condenser (tube in tube): Coil length – 12.23 m, Inner tube diameter (I_0) – 6 mm and Outer tube diameter (D_0) – 15 mm
- evaporator (tube in tube): Coil length – 10 m, Inner tube diameter (I_0) – 6 mm and Outer tube diameter (D_0) – 15 mm.

The programme employs a finite difference method to calculate the length of the condenser coil by considering the condenser as a single coil neglecting the effects of bends.

4 Results and discussion

The results obtained in this analysis are broadly classified into two categories vis. Selection refrigerant pair based on exergy analysis and performance plots from the simulation programme. The evaporator temperature is taken as 203 K for all condensing temperatures as the application is focused on 203 K. Figure 4 shows the variation of exergy loss of different components with respect to various condenser temperatures at an evaporator temperature of 203 K for the R23/290 refrigerant pair. The minimum possible exergy loss for R23/290 is 389 kJ/kg at the condensing temperature of 325 K. Figure 5 shows the variation of exergy loss of different components with respect to various condenser temperatures at an evaporator temperature of 203 K for R23/600 refrigerant pair. The minimum possible exergy loss for R23/600 is 419 kJ/kg at the condensing temperature of 313 K. Figure 6 shows the variation of exergy loss of different components with respect to various condenser temperatures at an evaporator temperature of 203 K for the R125/600 refrigerant pair. The minimum possible exergy loss for R125/600 is 324 kJ/kg at the condensing temperature of 324 K. By comparing the above three exergy loses, it is clear that the refrigerant pair R23/290 has less value of 325 kJ/kg. The minimum exergy loss has been recorded at between the condensing temperatures of 313 and 325 K due to the lesser process irreversibility and optimum system performance.

Figure 4 Exergy loss of various components for different condensing temperatures for R23/R290 pair at t_e = 203 K

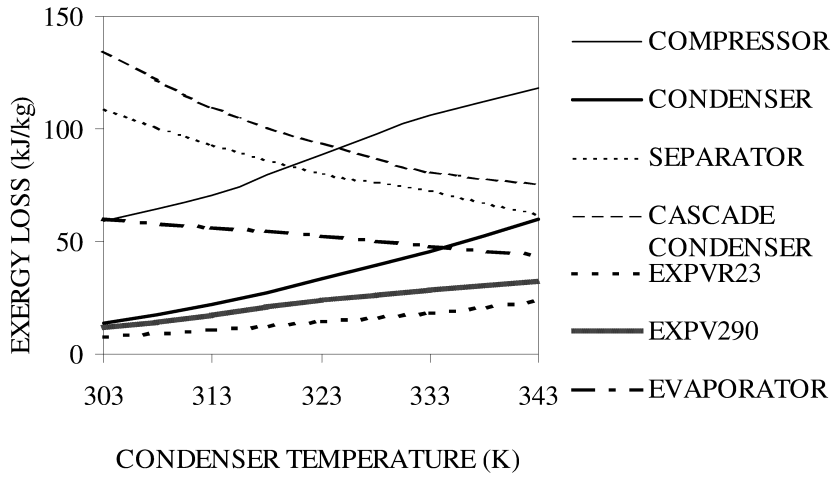


Figure 5 Exergy loss of various components for different condenser temperatures for R23/R600 pair at t_e = 203 K

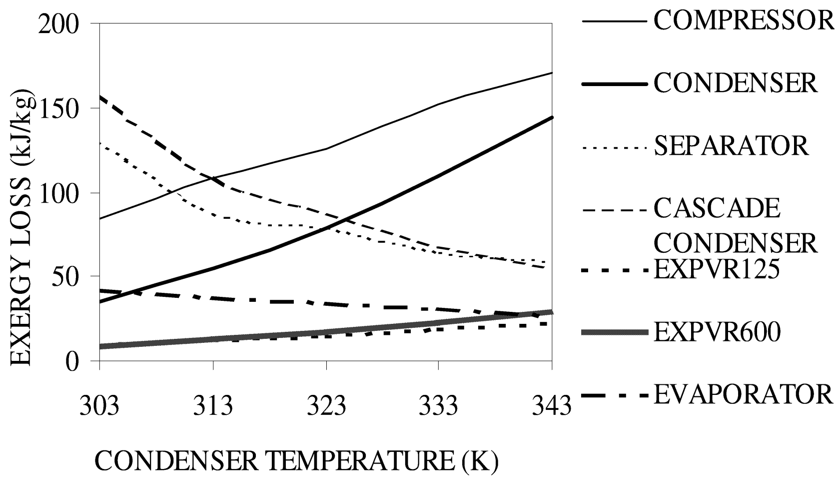


Figure 6 Exergy loss of various components for different condenser temperatures for R125/R600 pair at $t_e = 203\,K$

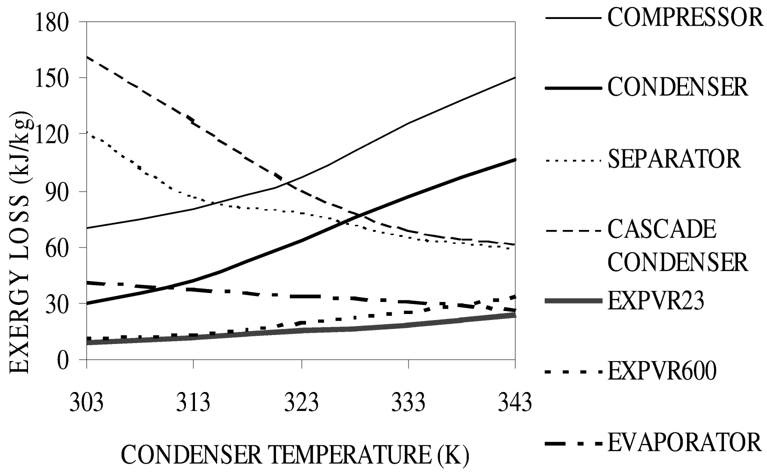


Figure 7 shows the variation of COP of ARC system for different condenser temperatures. It is evident that the COP decreases approximately 50% when the condenser temperature increases. The COP of the system is recorded at 303 K maximum but it is not recommended due to its higher exergy losses. Figure 8 shows the variation of exergetic efficiency for different condenser temperatures. The graph concludes that the exergetic efficiency of the R23/290 pair is better than the R23/600 and R125/600 pair. It is seen from the graphs that the compressor exergy losses are greater when the condensing temperature increases for a fixed evaporator temperature due to the system irreversibility increasing with the work of compression. The condenser exergy losses are generated mainly due to the larger temperature difference between the condenser temperature and the ambient. So as the condenser temperature increases, the overall temperature difference increases and hence the exergy losses also increase. The losses can be minimised by increasing the heat transfer area, which in turn increases the cost. Optimum values are to be found between minimum exergy loss and cost. By decreasing the temperature of the refrigerant coming out of the heat exchanger, the losses at the expansion valves can be minimised. Figure 9 shows the variation of refrigerant temperature along the condenser length. It is seen from the graph that the temperature of the refrigerant decreases along the length of the condenser. It is noted that 20% of the condenser length is used for de-superheating and the remaining 80% for the two-phase region. Figure 10 shows the variation of vapour quality of the refrigerant along the length of the condenser. It is seen that the vapour quality decreases gradually in the two-phase region and the refrigerant is not condensed fully i.e. the dew point of the refrigerant mixture is not reached. This is mainly because the dew point of the zeotropic mixture is well below the secondary fluid inlet temperature. Figure 9 shows the change in the temperature of the low boiling point refrigerant along the length of the cascade condenser. It is observed from the graph that 17% of the total heat exchanger length is used for de-superheating and the remainder is used for condensation. The change in temperature in the two-phase region signifies the glide phenomenon that occurs during the constant heat rejection process in a zeotropic mixture. Figure 10 shows the variation of vapour quality of low boiling point refrigerant along the length of the heat exchanger. The refrigerant leaves the heat exchanger as saturated liquid since no sub-cooling is assumed in the calculation. Figure 9 shows the variation of temperature along the evaporator coil length. It is seen from the graph that the temperature increases along the length of the evaporator. Figure 10 shows the variation of quality along the evaporator coil length. The vapour quality increases as it passes through the coil by picking up the product load. Figure 11 shows the variation of COP for various cooling water inlet temperatures of the condenser. The COP of the system is found to be 1.08 for the cooling water inlet temperature of 303 K. It is observed from the graph that the COP decreases as the cooling water inlet temperature increases. The vapour quality exit at the condenser plays a major contribution to the COP as it determines the mass flow rate of the liquid and vapour phases after the separator.

Figure 7 Variation of COP for various condenser temperature

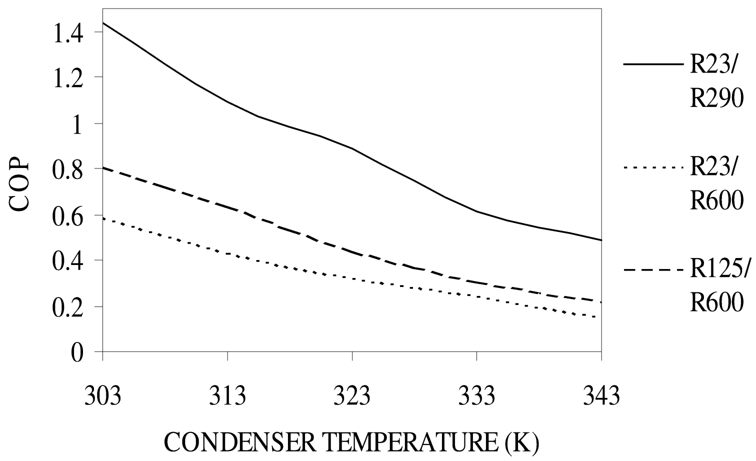


Figure 8 Variation of exergetic efficiency for various condenser temperature

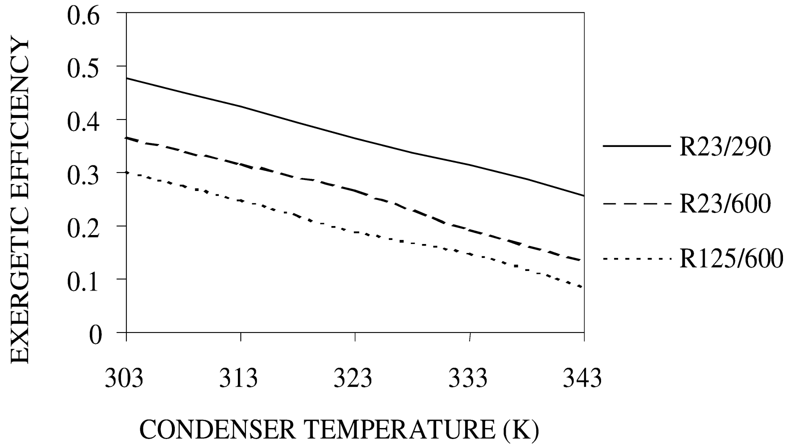


Figure 9 Variation of temperature along the length of heat exchangers

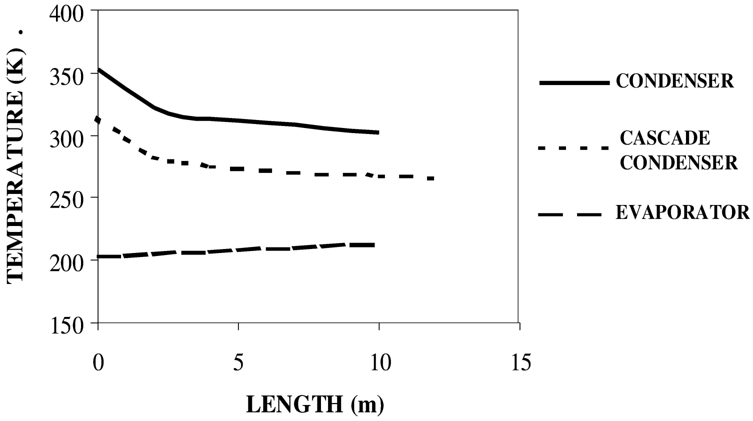


Figure 10 Variation of refrigerant vapour quality along the length of heat exchangers

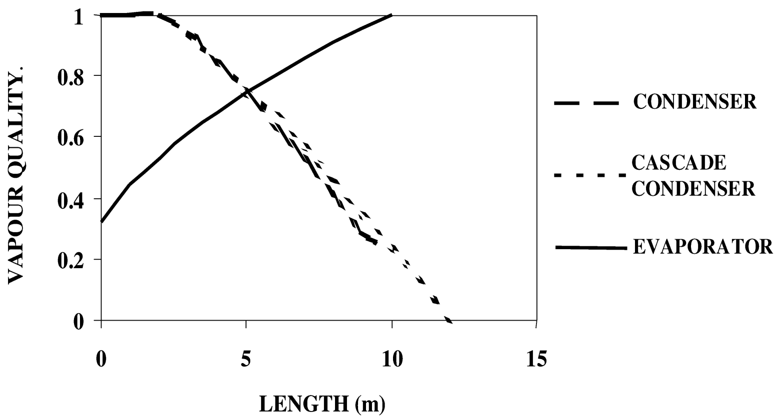
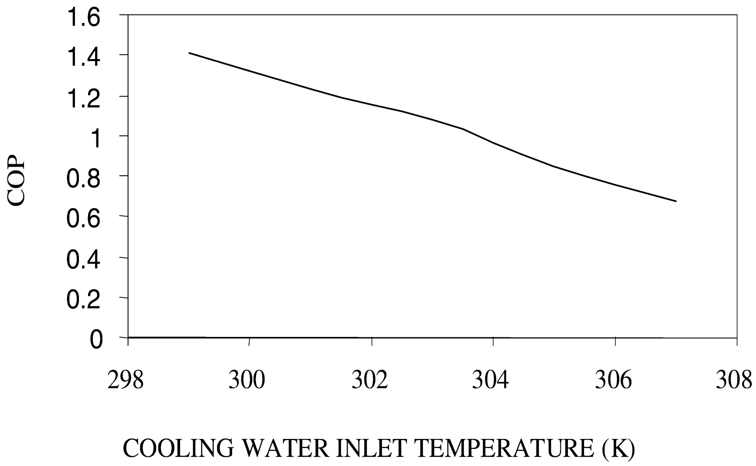


Figure 11 Variation of COP for various cooling water inlet temperature (R23/290)



5 Conclusions

The Auto Refrigerating Cascade (ARC) system can be a promising method for producing low temperature applications. Even cryogenic temperatures can be obtained by having a proper refrigerant mixture for the ARC system. The selection of refrigerant for ARC system is rather difficult, since a mixture of refrigerants is used. Some important aspects like the zeotropic nature of the refrigerant, boiling point difference for proper fractionation, molecular weight and second law analysis have to be considered. The refrigerant pair R23/R290 has been selected based on the features mentioned in the results and discussion. The second law analysis is performed on the three refrigerant pairs i.e. R23/600, R23/290 and R125/600; for the condenser temperature of 303 K the following are concluded (Table 2).

Table 2 Summary of exergy losses in individual components of ARC system

Component name	*% Less exergy loss of R23/R290 mixture over R125/R600*	*% Less exergy loss of R23/R290 mixture over R23/R600*
Compressor	20.00	14.28
Separator	35.01	30.00
Condenser	16.12	10.65
Cascade condenser	8.70	16.00
Expansion valve I	9.04	16.00
Expansion valve II	7.00	15.00
Evaporator	7.30	7.30

The exergetic efficiency of the system with R23/290 pair has been found to be higher than with that of R23/600 pair by 23% and R125/600 pair by 25% for the condenser temperature of 303 K. The overall system performance of R23/R290 mixture is high in comparison with R23/600 and R125/600 for various cooling water inlet temperatures. For this specific low temperature application, CO_2 cannot be used because of its triple point and higher suction pressure.

Nomenclature

e	Specific exergy (kJ/kg)
h	Enthalpy (kJ/kg)
T	Temperature (K)
s	Entropy (kJ/kg.K)
$\dot{m}$	Mass flow rate (kg/s)
W	Compressor work (kJ/kg)
η_{ex}	Exergetic efficiency (%)
η_v	Volumetric efficiency (%)

n	Index of compression
V_{Comp}	Volume flow rate (m^3/s)
V	Specific volume (kg/m^3)
P	Pressure (kPa)
U	Overall heat transfer coefficient (W/m^2K)
A	Area (m^2)
ρ	Density (kg/m^3)
g	Acceleration due to gravity (m/s^2)
μ	Dynamic viscosity (m^2/s)
λ	Thermal conductivity (W/mK)
SF	Secondary heat transfer fluid
mix	Mixture

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Notes

[1] MATHCAD 7 – Professional, Mathsoft Engineering and Education.

[2] REFPROP, Thermodynamic and Transport Properties of Refrigerants and Refrigerant Mixtures, NIST Standard Reference Database 23, Version 6.01, 1998.

[3] Visual C++, Microsoft Corporation, 1997.