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Exergy and Performance Analysis of Three Stage Auto Refrigerating Cascade (3 Stage ARC) System Using Zeotropic Mixture of Eco-Friendly Refrigerants

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Abstract – *To Rule out the CFC's from Vapour Compression Refrigeration, the Zeotropic mixture of HC's and HFC's being the only alternates to cater to the needs of Cryo-cooling and Cryo-chamber technology, the Three Component Zeotropic Mixture of R1270/R170/R14 was studied for the existence of the 3 stage ARC system. Exergy Analysis was carried out on this system for Confirmation of the 3 stage ARC System and the results confirmed the Existence of the system. The effect of mass fraction on Coefficient of Performance (COP), Exergy lost, Exergic efficiency, Efficiency defect and the Evaporating temperature achieved were investigated for different mass fractions. In accordance with the Environmental issues and the process of sustainable development, the Three Component Zeotropic Mixture of R1270/R170/R14 with the mass fraction of 0.265:0.18:0.55 was performing best with the suggestion of an alternative refrigerant for Three stage Auto Refrigerating Cascade (3 stage ARC) System operating at Very low evaporating temperature in the range of 183K (-90^{0}C) at COP of 0.267 and comparatively increased Exergic efficiency up to 6.02% (66.9%). The better performance with COP of 0.316 and the Exergic efficiency of 63.1% at around 194K (-79^{0}C) have eliminated based on the fact that the lower most evaporating temperature was not achieved.*

***Keywords**: Exergy Analysis, Exergic Efficiency, Efficiency Defect, 3 Stage ARC, Zeotropic Mixture of R1270/R170/R14, Performance Analysis, COP*

Nomenclature

ψ	Exergy flow (J kg^{-1})
η_x	Exergy Efficiency (%)
δ	Efficiency Defect
COP	Coefficient of performance
h	Specific Enthalpy (J kg^{-1})
T_0	Ambient temperature (^{0}K)
$T_{evaporator\text{-}I}$	Evaporating temperature of cascade condenser-I [Refirigerant-1] (^{0}K)
$T_{evaporator\text{-}II}$	Evaporating temperature of cascade condenser-II [Refrigerant-II] (^{0}K)
$T_{evaporator\text{-}III}$	Evaporating temperature of Evaporator-III [Refrigerant-III] (^{0}K)
T_{ACC}	Condensing temperature of Air cooled condenser [Refrigerant-I] (^{0}K)
$T_{condenser\text{-}I}$	Condensing temperature of cascade condenser-I [Refrigerant-II] (^{0}K)
$T_{condenser\text{-}I}$	Condensing temperature of cascade condenser-II [Refrigerant-III] (^{0}K)
s	Specific Entropy (J kg^{-1} K^{-1})
w_c	Compressor work input (W)
$\dot{m}$	Mass (Total Refrigerant) flow rate ($\dot{m} = m_{RI} + m_{RII} + m_{RIII}$) (kg s^{-1})
m_{RI}	Mass Flow rate of Refrigerant I (kg s^{-1})
m_{RII}	Mass Flow rate of Refrigerant II (kg s^{-1})
m_{RIII}	Mass Flow rate of Refrigerant III (kg s^{-1})
Q_{ACC}	Heat removed at Air cooled condenser [Heat rejected by The Mixture of First, second and third refrigerant at the Air cooled condenser as Hot fluid link] (W)
$Q_{condenser\text{-}I}$	Heat removed at condenser-I [Heat rejected by The Mixture of second and third refrigerant at the inner tube of cascade condenser-I as Hot fluid link] (W)
$Q_{condenser\text{-}II}$	Heat removed at condenser-II [Heat rejected by third refrigerant at the inner tube of cascade condenser-II as Hot fluid link] (W)
$Q_{evaporator\text{-}III}$	Refrigerating effect [Refrigerant-III] (W)
$Q_{evaporator\text{-}II}$	Refrigerating effect at cascade condenser-II [Refrigerant-II] (W)
$Q_{evaporator\text{-}I}$	Refrigerating effect at cascade condenser-I [Refrigerant-I] (W)
I_{Total}	Exergy loss of the system (J s^{-1})
I_{comp}	Exergy loss at compressor (J s^{-1})
I_{ACC}	Exergy loss at Air cooled condenser (J s^{-1})
$I_{condenser\text{-}I}$	Exergy loss at cascade condenser-I in hot fluid link by mixture of second and third refrigerant (J s^{-1})
$I_{condenser\text{-}II}$	Exergy loss at cascade condenser-I in hot fluid link by third refrigerant (J s^{-1})

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$I_{TEV\text{-}I}$	Exergy loss at Thermostatic Expansion valve-I [Refrigerant-I] ($J\ s^{-1}$)
$I_{TEV\text{-}II}$	Exergy loss at Thermostatic Expansion valve-II [Refrigerant-II] ($J\ s^{-1}$)
$I_{TEV\text{-}III}$	Exergy loss at Thermostatic Expansion valve-III [Refrigerant-III] ($J\ s^{-1}$)
$I_{evaporator\text{-}I}$	Exergy loss at cascade condenser-I in clod fluid link by first refrigerant ($J\ s^{-1}$)
$I_{evaporator\text{-}II}$	Exergy loss at cascade condenser-II in clod fluid link by second refrigerant ($J\ s^{-1}$)
$I_{evaporator\text{-}III}$	Exergy loss at Evaporator as cold link by Third refrigerant ($J\ s^{-1}$)

I. Introduction

Ozone level in the Arctic region as was measured as 320 DU (Dobson units) or about 150 DU below spring time as normal as of 450DU in 1956 at first as an unknown phenomenon of Ozone Hole by G.M.B Dobson.

But the CFC's lead the world of Refrigeration industries for over 6 decades before the harmful effect on the Ozone layer was identified in 1974 by Frank Sherwood Rowland and his postdoctoral associate Mario J. Molina.

A Steady decline of 4% per decade in Total Volume of Stratospheric Ozone was found and much greater depletion level during springtime was identified in early 1980's. The results of global warming effect of these refrigerants 8500 times greater than CO_2 over hundred years have directed the HVAC Engineers towards Production of HFC's and HC's as well as the ban on usage of CFC's and HCFC's. The Montreal Protocol which was an agreement signed on 16 September 1987 and came into force on 1 January 1989 has also enriched the recent researches.

The London, Copenhagen, Montreal and Beijing Amendments came into force on 10th August 1992, 14th June 1994, 10th November 1999 and 25th February 2002 established the final expiration of HCFC's as 2030. So the search continues on the possible alternatives for the refrigerants, which have thermophysical acceptance, thermodynamically acceptable and harmless for the environment. Eric W. Lemmon and Richard T Jacobsen [1] have reported that No additional parameters were required to model the ternary mixture and also slight systematic offsets are seen in several locations for example, The R-32/125/134a system is unique from a modeling standpoint since it combines the three mixture equations the individual equations for R-32/125 and R-32/134a, and the generalized equation for R-125/134a.

Ciro Aprea and Angelo Maiorino [2] have studied COP improvement by Employing pressure control at gas-cooler outlet and quantified as 6.6–8.5% Under minimum pressure working condition at different T ambient.

Michael Uhlmann and Stefan S. Bertsch [3] have derived the improved control strategies for heat pumps using on-off cycling as capacity control through the evidence of performance losses of 1 - 2% for short cycling times of air source heat pumps and 5% efficiency gain in the case of geothermal heat pumps by the recovery of the ground probe during the off time.

J.H. Lee et al [4] have suggested the numerical code to predict the performance of Condenser and found the acceptable deviation of calculated and experimental values with the use of R-22 were 10.1% greater than experiment data and with the use of R-407C the results were 10.7% less than experiment data and thus they suggested the numerical code to be used as a design tool to develop better condenser paths.

Zanjun Gao et al [5] have studied the absorption refrigeration system with the binary mixture of Trifluoromethane (R23) and N,N-dimethyl formamide as a promising new working fluid and found the average relative deviation of 1.8% between experimental and calculated values of system parameters.

Pradeep Bansal, Edward Vineyard and Omar Abdelaziz [6] have studied the alternative technologies for refrigeration such as thermo-acoustic refrigeration, thermoelectric refrigeration, thermo-tunneling, magnetic refrigeration, Sterling cycle refrigeration, pulse tube refrigeration, Malone cycle refrigeration, absorption refrigeration, adsorption refrigeration, and compressor driven metal hydride heat pumps. They suggested going for integrated heat pump system serving both heating and air conditioning applications in domestic applications.

X. Boissieux, M.R. Heikal and R.A. Johns [7] studied with the use of Mixtures R407C (R32/R125/R134a of quality 0.23:0.25:0.52), R404a (R125/R143a/R134a of quality 0.44:0.52:0.04) and Isceon 59 (R125/R134a/R600 of quality (0.47:0.50:0.03) and stated that the Dobson and Chato correlation provided the best prediction for these refrigerant mixtures and the Shah correlation fitted the measurements of the local heat transfer coefficients well and seems to cope well with refrigerant mixtures.

Ruixiang Wang, Qingping Wu and Yezheng Wu [8] have confirmed the normal working of Residential Air Conditioners with the use of the mixture of R410a/MNRO as working fluid. The cooling/heating Energy Efficiency Ratio of the Residential Air Conditioners increased about 6% by replacing the Polyol-Easter oil VG 32 lubricant with MNRO (mineral-based nano-refrigeration oil).

Jianyong Chen and Jianlin Yu [9] have Confirmed the new refrigeration cycle having the Evaporator circuit of two branches to realize Lorentz cycle with the advantage of Temperature glide (NRC) using the binary non-azeotropic refrigerant mixture (R32/R134a) results in 8 to 9% COP raise and 9.5% increase in Volumetric Refrigerating Capacity.

M.H. Kim et al [10] have predicted the mean deviation of the calculated condensation heat transfer coefficients for the binary Zeotropic mixture of R134a/R123 in the horizontal smooth tube was about 10.3% and he also suggest that the high mass flux transfer have slight effect on condensation heat transfer.

Mark O McLinden et al [11] suggested that halocarbon refrigerants, propane, ammonia, and carbon dioxide, R11, R12, R13, R22, R23, R114, R115, R125, R142b, R143a as well as mixtures of these fluids the average absolute deviations between the calculated and experimental values of thermal conductivity arranged from 1.08 to 5.57% for the pure Fluids, and 2.98 to 9.40% deviations for the mixtures given below R125/134a, R32/propane, R32/134a, Propane/R134a, R32/125, R32/125/134a.

Jianlin Yu, Hua Zhao and Yanzhong Li [12] have proved that an outstanding merit in decreasing the pressure ratio of compressor as well as increasing the COP. For Novel Autocascade Refrigeration system with Ejector using the mixture of R134a/R23 of Quality 0.85:0.15 operated at the condenser outlet temperature of 400C, the evaporator inlet temperature of -40.3^{0}C, the pressure ratio of the ejector reaches to 1.35, the pressure ratio of compressor is reduced by 25.8% and the COP is improved by 19.1% over the conventional Autocascade refrigeration cycle.

K. Comakli et al [13] was observed that the most effective parameters are found to be the condenser air inlet temperature for COP and Exergetic efficiency for the heat pump with the use of mixture of R404a and R22 and has no Influence of R404a mixture and suggests that the Pure R22 is the only solution for Heat pump application.

Primal Fernando et al [14] have predicted that the heat transfer coefficients of Evaporator using Propane as Refrigerant was found to increase with increasing heat flux, which was accompanied by an increase in the Propane mass flux and increasing evaporation temperatures as follows evaporation temperatures ranged from -15 to +10^{0}C and the heat and mass fluxes ranged from 200^0 to 9000 W/m^{-2} and 13 to 66 kg m^{-2}s^{-1}.

The refrigerant flow was laminar with the liquid Reynolds number ranging from 171 to 877.

Ki-Jung Park and Dongsoo Jung [15] have reduced the charge of the system up to 58% due to its low liquid density and found that the COP of the system using Mixture of R170/R290 of quality 0.06:0.94 was higher than that of R22 and have 16.6–28.2^{0}C lower compressor discharge temperatures, this in turn increases the life of the system. Primal Fernando et al [16] have performed a study on traditional refrigeration system under typical Swedish condition and found that the charge of about 200 grams is the best choice for the heat pump providing a COP between 3.5 and 4 using R290 as a Refrigerant.

Dongoo Jung et al [17] have Examined the Performance of R290 / R600a Mixture of quality or mass fraction 0.6 : 0.4 in Domestic Refrigerators and Suggested the COP increase of 2.3%. Three to four Percentage (3-4%) of Higher Energy efficiency at faster Cooling rate as well as shorter compressor on-time and Lower compressor Dome temperatures were confirmed with this mixture compared to CFC12 – R12.

D.J. Cleland, R.W. Keedwell and S. R. Adams [18] have quantified the system performances and stated that the system with mixture of propane and ethane (Care-50) reduced energy use by 6–8% under similar system cooling capacity relative to HCFC-22. With propane (Care-40), energy use decreased by 5% but cooling capacity was 9% lower.

S.Anand, S.K. Tyagi [19] have revealed the fact that the exergy destruction is least when the system is 25% charged and also COP of the system is high when the system is 50% charged due to higher refrigerating effect and reduced compressor work and Exergy efficiency of the system is highest when the system is 100% charged. So he suggested that the system must run with the optimum balance between the exergy efficiency and energy savings.

Jianfeng Wu et al [20] achieved a minimum no-load temperature of -197.7^{0}C (about 75.7 K), -174^{0}C (about 99 K) was obtained at 110 W cooling capacities with the mixed-gases refrigeration using dual mixed-gases Joule Thomson refrigeration system. They achieved the lowest temperature of -192^{0}C (about 81 K) with an effective preservation volume of 80 L at a relatively faster cooling-down rate in cryogenic chamber and found 2.5 hours to reach -180^{0}C, and 5 hours to reach -190^{0}C.

Liu Jie et al [21] suggested the start up processes in different situations may cause some liquid superheats and evaporator temperature overshoots, but they will not affect much on the steady state operation of the MPCL (Mechanically Pumped Two-phase Cooling Loop).

Andrey Rozhentsev and Vjacheslav Naer [22] have studied the stationary modes the operating parameters of the system Using 3-component non-azeotropic mixtures of hydrocarbons – isobutane ($CH(CH_3)_2CH_3$) / ethane (CH_3CH_3) / methane CH_4) as a refrigerant which corresponded to their design / calculated values varying with the refrigerant working mixture compositions and ambient temperatures (that is the heat load) within the following ranges: discharge pressure – (12.3, ..., 13.4) bar; suction pressure – (0.8, ..., 1.3) bar; compressor input power – (385, ..., 435) W. The air temperature in the low-temperature chambers of the 1st and 2nd type for the considered modes was as low as (-80, ..., -95)^{0}C while the average temperature along the evaporator was of (-85, ...,-105)^{0}C, correspondingly.

D.Y. Lee et al [23] have predicted the effect of the refrigerant change from R22 to R407C on the Chiller performance and measured that the cooling capacity decreases by 10–20% and the COP by 20–30% depending on the temperature condition. It is found that the main reason for the decrease in the Chiller performance is the decrease in the heat transfer coefficient of R407C compared with that of R22 based on the three factors which are thermodynamic properties, compressor efficiency, and heat transfer.

Dave Sajjan, Tord Karlsson and Lennart Vamling [24] have investigated the performance of the system of R22 with the retrofit of R407C. The experimental and theoretical investigation made confirmed the drop in shell-and-tube condenser performance and quantified that the reduction in performance can be as large as 70% compared to the full condenser load at lower condenser loads.

Andrey Rozhentsev [25] Investigated the Auto Refrigerating Cascade System (Linde Cycle) and achieved -75^{0}C and the following operating conditions working with the Zeotropic mixture of n-butane / ethylene with the quality of 0.6:0.4 under the total mass charge of 110grams between the ambient temperature of (-5, ..., +43)^{0}C and working conditions he achieved: The air temperature in the chamber (-86, ..., -63)^{0}C; discharge pressure (4.5, ..., 13.5) bar; suction pressure (1.3, ..., 3.5) bar; Input power (300, ..., 560) W; discharge temperature (45, ..., 110)^{0}C; and the compressor dome temperature (25, ..., 70)^{0}C. A. Johansson and P. Lundqvist [26] have found the charged composition and the circulated composition as well as the leak composition differ as well as the differed compositions will not affect the cycle performance using the Zeotropic mixtures like 407C. He has also suggested the predictive model to determine the circulating composition and suggested that this will be same for Zeotropic mixtures consisting 3 and more components. Kai Du et al [27] have suggested the use of Zeotropic mixture of R134a & R23 of quality 0.7:0.3 as an alternate working fluid for Auto Cascade Refrigerating system to obtaining High COP. He also suggested raising the mass flex of high boiling Liquid Refrigerant and reducing the Low boiling Liquid Refrigerant without any alteration in its Discharge condition. Yijian He and Guangming Chen [28] have studied the properties of the system and stated that the mixture of R23+R32+R134a/DMF as working pair of a novel auto-cascade absorption refrigeration system has gained better performances with a refrigerating temperature as low as -50^{0}C. They made it clear that the mixture of R23+R32+R134a/DMF as working pair shows a rapider lowering rate of refrigerating temperature than that of an auto-cascade absorption refrigeration system using R23 + R134a/DMF as its working pair. Giovanni Di Nicola et al [29] have investigated the feasibility of R744 blends as an attractive option for the low-temperature-circuit in cascade systems operating at temperatures approaching 200 K.

P. Thangavel et al [30] have investigated the feasibility of Hydrocarbon Mixtures as alternative refrigerants and proved they are the viable solution for VCR system without any modification in the system. S. Frikhal and M.S. Abid [31] have investigated performance of combined refrigeration cycles conventional cascade (CC) and the integrated cascade (IC)] using finite time thermodynamic (FTT) analysis and was found that at fixed condensing and evaporating temperatures and for same intermediate heat-exchanger temperature ratio, the IC is more efficient than the CC system. The coefficient of performance of the IC system can be more enhanced under the condition of minimum intermediate heat exchanger irreversibilities (less glide between the two counter-flow intermediate heat exchanger).

Even though the studies revealed many truth these studies does not deal with ARC and its existence with respect to 3 component Zeotropic mixture.

So this study is the stepping stone for the future in Cryo-technologies.

II. Experimental Setup and Procedure

Three stage Auto Refrigerating cascade system was fabricated as per the design parameters. The detailed photographic view of the system is shown in Fig. 1 and the line diagram of the setup is shown in Fig. 2.

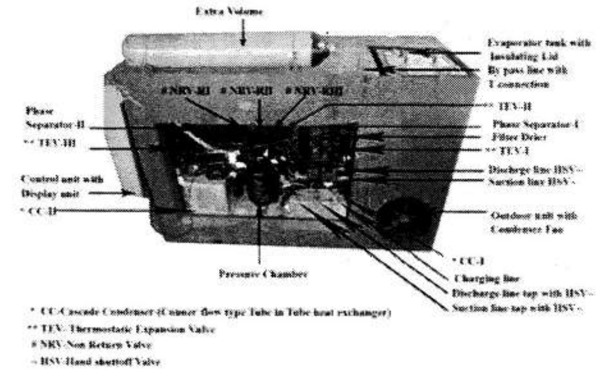
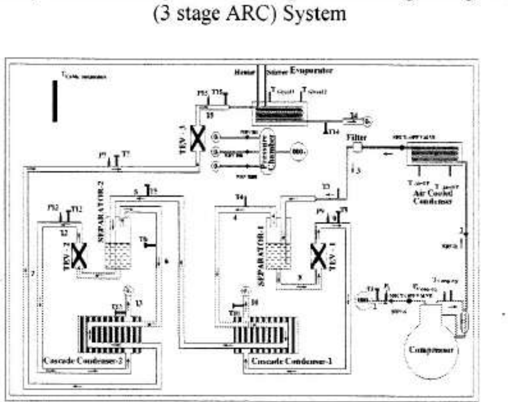


Fig. 1. Experimental setup of Three Stage Auto Refrigerating Cascade (3 stage ARC) System



State Points

1 - Low pressure Vapour of RI,II&III
2 - High Pressure - Super Heated Vapour of RI,II & III
3 - Sub Cooled Liquid of RI + super heated Vapour of RII&III
4 - Super heated Vapour of RII,III
5 - Sub cooled Liquid of RII + Super heated Vapour of RIII
6 - Super Heated Vapour of RIII
7 - Subcooled Liquid of RIII
8 - Sub Cooled Liquid of RI
9 - Wet Mixture of RI
10 - Super Heated Vapour of RI
11 - Sub Cooled Liquid of RII
12 - Wet Mixture of RII
13 - Super Heated Vapour of RII
14- Super Heated Vapour of RIII
15- Wet Mixture of RIII

Fig. 2. Line Diagram of Three Stage Auto Refrigerating Cascade (3 stage ARC) System

The compressor used in this system is Kirloskar of 4500 BTU capacity. A four row air cooled condenser of 2TR capacity is used to facilitate the proper condensing area. An oil separator was connected to separate the oil from the compressor discharge and the separated oil was directed to the compressor doom through the suction line of the compressor. The compressors oil return has been ensured and the Filter drier is also used in the system after Air cooled condenser for proper filtration and block free working of the system.

Since the pressure may go up to 700psig during operation and 450psig while system gets thermal equilibrium with atmosphere, a thick cylinder has been considered for phase separation. To ensure the proper working of the three stage auto refrigerating cascade system an Extruded copper tube welded at top and bottom by copper plate were considered for phase separator I & phase separator-II. Each separator is provided with one input line, one copper line at the top of the separator which provides the high pressure line for the medium and low boiling refrigerants and the separator ensures perfect separation of liquid refrigerant through gravity separation.

The air cooled condenser must remove all the heat of refrigerants combined together and must condense the high boiling refrigerant which is the largest mass fraction of all three circulating refrigerants must cools the Medium boiling refrigerant and then the medium boiling refrigerant must cool the low boiling refrigerant to ensure that the circuit works continuously. For the realization of these different requirements of each stage this three stage ARC system considered the Thermostatic Expansions valves with orifice sizes decreasing continuously in size to accommodate the variation in cooling load.

The system is provided with the Extra volume to accommodate the excess density gas during the thermal equilibrium of the ARC. Cascade condenser-I and cascade condenser-II are used as counter flow heat exchangers for the better and efficient heat transfer performance between the refrigerants flowing in the circuit. Since the heat rejected by the low boiling refrigerant must be equal to the heat gained by the medium boiling refrigerant at Cascade condenser-II and heat rejected by the mixture of medium and low boiling refrigerants should be equal to heat gained by the high boiling refrigerant in cascade condenser-I, these two cascade condensers were insulated with PUF insulation around them by a box. Flow of hot fluid inside the inner tube ensures the heat transfer to the cold fluid flowing in outer tube of cascade condenser in both the cases.

Pressure chamber is used to provide extra volume as well as mixing chamber at suction side of the system through three different Non-return valves. The system was equipped with two number of wattmeter to indicate the power consumption of the compressor and the heat load given to the evaporator. The heat was connected through On-off control.

Around twelve numbers of P type RTD sensors with an accuracy of ±1°C & around 13 numbers of 3 wire RTD sensors with ±1°C were connected at different positions as stated state points of the system. Totally six number of WIKA [0-100Kg/cm2 (0-1400psi)] pressure gauges were used in this system for continuous monitoring of system pressure at suction and discharge side. The readings confirm that no considerable variation along the length of the piping. So decided to used individual suction pressure of each refrigerant after each expansion, discharge pressure and Discharge side pressure just at the end of the high pressure line of low boiling refrigerant which travels more distance from the compressor [just before TEV-III]. Thermo wells were set for mounting the temperature sensors.

A heater was used to provide load to the evaporator. A stirrer was also used to enhance the heat transfer in the secondary fluid in the evaporator tank. Evaporator tank was insulated with PUF insulation around it. The Perfect insulation is being ensured inside the evaporator tank.

The total leak in the setup was less than 5% of the evaporator load.

All the parameters of the system and mainly the power consumption were observed after the system has reached the study state condition. Values obtained were used for this study and the characteristic curves were plotted and the performance of ARC system was studied.

III. Exergy Analysis

Generally first and second law of thermodynamics are used to analyze the system performance. In an energy analysis, based on the first law of thermodynamics, all forms of energy are considered to be equivalent. The loss of quality of energy is not taken into account. For example, the change of the quality of thermal energy as it is transferred from a higher temperature source to a lower temperature sink, cannot be demonstrated in an energy analysis and also shows the energy flow to be continuous.

Exergy analysis and the concept of second law of efficiency has invoked considerable interest in recent years due to the fact that its application leads to a better understanding of the process of energy transfer, and helps to identify the thermodynamic losses clearly. However, this analysis can be a complementary to the energy analysis but cannot replace it. Studies applying second law analysis for refrigeration process though available are limited to certain refrigerants and particular operating conditions. The importance of second law analysis as compared to the conventional energy conservation analysis is mainly due to the effect that the later does not take in to account the quality of energy, and assigns high quality forms of energy to low quality process. Moreover, it results in assigning efficiency values greater than unity for certain refrigeration process and heat pump applications, thus departing from the efficiency concepts. Second law of thermodynamic analysis takes in to account the quantity of energy consumed and the quality of the energy conservation.

Besides it permits to identify the losses occurring in different components of the system and thus improve thermodynamic efficiency.

Exergy of a system denotes the maximum amount of work that can be obtained when the system is allowed to come back to equilibrium with the surroundings. The state of the surrounding is called 'dead state'. Two types of dead states come in to play.

The system in thermal and mechanical equilibrium with the surroundings is said to be in "restricted dead state". In this state, the system is not permitted to mix or enter in to chemical reaction with the surroundings and the maximum amount of work obtained is called "thermo-mechanical availability". The second, called as "absolute dead state" is achieved when the system is in thermal, mechanical and chemical equilibrium with the surroundings.

Thus the concentration difference between the system at restricted dead state and the surroundings at absolute dead state could be used to produce a certain quantity of work called "chemical availability".

It is important to note that chemical availability plays a dominant role in exergy analysis in process involving mixtures of chemical substances or in chemical reactions (e.g. combustion) and should not be ignored. Hence, for vapour compression refrigeration process it is sufficient to consider only the "thermo-mechanical equilibrium".

Under the assumption that the change of kinetic and potential energy is negligible and the ambient temperature is T_0, the exergy is given by the equations:

$$\psi = h - T_0 s \tag{1}$$

$$\psi = (h - h_0) - T_0(s - s_0) \tag{2}$$

For the Three stage Auto Refrigerating Cascade system the component wise the exergy balance equation can be written as follows

a) For Compressor:

Compressor work:

$$w_c = \dot{m}(h_2 - h_1) \tag{3}$$

where:

$$\dot{m} = m_{RI} + m_{RII} + m_{RIII}$$

The exergy loss (due to irreversibility) in the compressor:

$$I_{comp} = \dot{m}(h_1 - T_0 s_1) + w_c - \dot{m}(h_2 - T_0 s_2) \tag{4}$$

b) For Air cooled Condenser:

Heat removed at Air Cooled condenser:

$$Q_{ACC} = \dot{m}(h_2 - h_3) \tag{5}$$

where:

$$\dot{m} = m_{RI} + m_{RII} + m_{RIII}$$

The exergy loss (due to irreversibility) in the Air cooled condenser:

$$I_{ACC} = \dot{m}(h_2 - T_0 s_2) - \dot{m}(h_3 - T_0 s_3) + -Q_{ACC}\left(1 - \frac{T_0}{T_{ACC}}\right) \tag{6}$$

c) For Condenser-I (Cascade Condenser-I) (Hot fluid flow): Heat removed at condenser-1:

$$Q_{Condenser-I} = \dot{m}_{RII+RIII}(h_4 - h_5) \tag{7}$$

The exergy loss (due to irreversibility) in the condenser-1:

$$I_{Condenser-I} = \dot{m}_{RII+RIII}(h_4 - T_0 s_4) + -\dot{m}_{RII+RIII}(h_5 - T_0 s_5) - Q_{condenser-I}\left(1 - \frac{T_0}{T_{condenser-I}}\right) \tag{8}$$

d) For Condenser-II:

(Cascade Condenser-II: Hot fluid flow):

Heat removed at condenser-II:

$$Q_{Condenser-II} = \dot{m}_{RII+RIII}(h_6 - h_7) \tag{9}$$

The exergy loss (due to irreversibility) in the condenser-II:

$$I_{Condenser-II} = \dot{m}_{RIII}(h_6 - T_0 s_6) - \dot{m}_{RIII}(h_7 - T_0 s_7) + -Q_{condenser-II}\left(1 - \frac{T_0}{T_{condenser-II}}\right) \tag{10}$$

e) For Thermostatic Expansion Valve-I:

The exergy loss (due to irreversibility) in the Thermostatic Expansion valve-I:

$$I_{TEV-I} = \dot{m}_{RI}(h_8 - T_0 s_8) - \dot{m}_{RI}(h_9 - T_0 s_9) \tag{11}$$

Since the enthalpy is constant during the expansion process, we know that $h_8 = h_9$, the above equation can be written as:

$$I_{TEV-I} = \dot{m}_{RI} T_0 (s_9 - s_8) \tag{12}$$

f) For Thermostatic Expansion Valve-II:

The exergy loss (due to irreversibility) in the Thermostatic Expansion valve-I:

$$I_{TEV-II} = \dot{m}_{RII}(h_{11} - T_0 s_{11}) - \dot{m}_{RII}(h_{12} - T_0 s_{12}) \tag{13}$$

Since the enthalpy is constant during the expansion process, we know that $h_{11} = h_{12}$, the above equation can be written as:

$$I_{TEV-II} = \dot{m}_{RII} T_0 (s_{12} - s_{11}) \tag{14}$$

g) For Thermostatic Expansion Valve-III:

The exergy loss (due to irreversibility) in the Thermostatic Expansion valve-I:

$$I_{TEV-III} = \dot{m}_{RIII}(h_7 - T_0 s_7) - \dot{m}_{RIII}(h_{15} - T_0 s_{15}) \tag{15}$$

Since the enthalpy is constant during the expansion process, we know that h7 = h15, the above equation can be written as:

$$I_{TEV-III} = \dot{m}_{RIII} T_0 (s_{15} - s_7) \tag{16}$$

h) For Evaporator-III:

Heat addition in Evaporator-III:

$$Q_{evaporator-III} = \dot{m}_{RIII}\left(h_{14} - h_{15}\right) \tag{17}$$

The exergy loss (due to irreversibility) in the Evaporator – III:

$$\begin{aligned} I_{evaporator-III} &= \dot{m}_{RIII}\left(h_{15} - T_0 s_{15}\right) + \\ &+ Q_{evaporator-III}\left(\frac{T_0}{T_{evaporator-III}}\right) + \\ &- \dot{m}_{RIII}\left(h_{14} - T_0 s_{14}\right) \end{aligned} \tag{18}$$

i) For Evaporator-II (Cascade Condenser-II: Cold Fluid Flow):
Heat addition in Evaporator-II:

$$Q_{evaporator-II} = \dot{m}_{RIII}\left(h_{13} - h_{12}\right) \tag{19}$$

The exergy loss (due to irreversibility) in the Evaporator-II:

$$\begin{aligned} I_{evaporator-II} &= \dot{m}_{RII}\left(h_{12} - T_0 s_{12}\right) + \\ &+ Q_{evaporator-II}\left(\frac{T_0}{T_{evaporator-II}}\right) + \\ &- \dot{m}_{RII}\left(h_{13} - T_0 s_{13}\right) \end{aligned} \tag{20}$$

j) For Evaporator-I (Cascade Condenser-I: Cold Fluid Flow):
Heat addition in Evaporator-I:

$$Q_{evaporator-I} = \dot{m}_{RI}\left(h_{10} - h_9\right) \tag{21}$$

The exergy loss (due to irreversibility) in the Evaporator-I:

$$\begin{aligned} I_{evaporator-I} &= \dot{m}_{RI}\left(h_9 - T_0 s_9\right) + \\ &+ Q_{evaporator-I}\left(\frac{T_0}{T_{evaporator-I}}\right) \\ &- \dot{m}_{RI}\left(h_{10} - T_0 s_{10}\right) \end{aligned} \tag{22}$$

The total Exergy loss of the system is given by the correlation:

$$\begin{aligned} I_{Total} &= I_{comp} + I_{ACC} + I_{condenser-I} + \\ &+ I_{condenser-II} + I_{condenser-III} + I_{evaporator-I} + \\ &+ I_{evaporator-II} + I_{evaporator-III} + I_{TEV-I} + \\ &+ I_{TEV-II} + I_{TEV-III} \end{aligned} \tag{23}$$

The exergy efficiency is given by:

$$\delta_i = \frac{\Psi_1 - \Psi_{14}}{w_c} = \frac{Q_{evaporator-III}\left[1 - \dfrac{T_0}{T_{evaporator-III}}\right]}{w_c} \tag{24}$$

For the Three stage Auto Refrigerating Cascade system the component wise efficiency defect (δ_i) considering the ratio of exergy used in the corresponding component (ψ_i) to the exergy required to sustain the process (exergy through the compressor, w_c):

$$\delta_i = \frac{\Psi_i}{w_c} \tag{25}$$

The overall performance of the 3 stage ARC System is determined by evaluating its COP and is calculated as the ratio between the refrigerating capacity ($Q_{evaporator-III}$) and the electrical power supplied to the compressor (w_c):

$$COP = \frac{Q_{evaporator-III}}{w_c} \tag{26}$$

IV. Results and Discussion

The values of COP, Exergy Lost, Exergic Efficiency and Efficiency defect readings in individual components were calculated using the readings obtained by the experimental setup. These Parameters were discussed in this section for better understanding of the 3 stage ARC System.

Fig. 3 shows the variation of Exergic Efficiency of the ARC system with the variation of mass fraction of R1270 and R14 in the Zeotropic Mixture of R1270, R170 and R14.

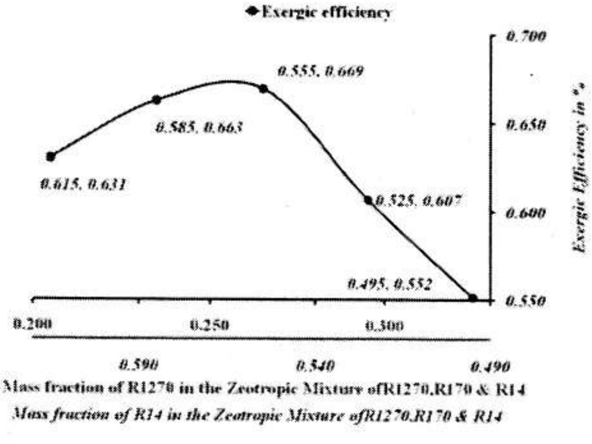


Fig. 3. Variation of Exergic Efficiency for Different Mass Fraction of R1270 & R14 in the Zeotropic Mixture of R1270, R170 & R14

The maximum value of Exergic efficiency from the Fig. 3 was 66.9% which correspond to the mixture of R1270, R170 and R14 with the mass fraction of 0.265:0.18:0.555 and the lowest among the values obtained is 55.2% which correspond to the mixture of mass fraction of 0.325:0.18:0.495.

The gradual increase in Exergic Efficiency of the mixture is noted with increase in quantity of R1270 and drastically decreases after the optimum ratio along the course of operation. The trend of Exergy efficiency may be noted due to the proper heat transfer between refrigerants in cascade condensers and thus the increase in refrigerating effect which intern gives the better efficiencies and it drastically reduces due to the excess amount of refrigerant flow with higher value of terminal velocity which will not be able to efficiently transfer the heat between the refrigerants in cascade condensers.

Fig. 4 shows the Variation in COP with the variation of mass fraction of R1270 and R14 in the Zeotropic Mixture of R1270, R170 and R14.

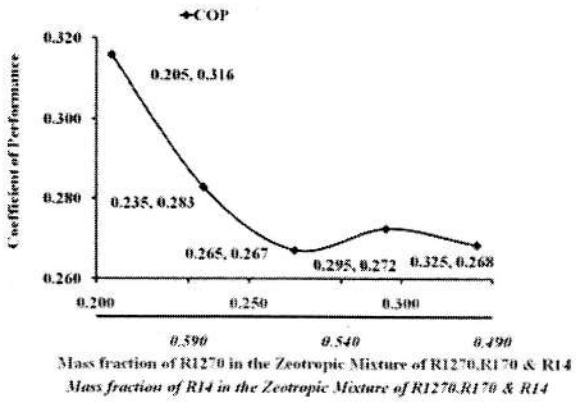


Fig. 4. Variation of COP for Different Mass Fraction of R1270& R14 in the Zeotropic Mixture of R1270, R170 & R14

It is observed that the decrease in COP is due to the increase in mass flow through the system and through the compressor which increases the compressor work and which in turn reduces the COP. The Fig. 4 also explains the concept of slight increase in COP for the higher mass ratio of R1270 and Lower mass ratio of R14 due to the higher flow rate of high boiling refrigerant which ensures the proper cooling and effectiveness in the first stage itself which in turn adds a slight increment in refrigerating effect and thus the COP. The maximum and minimum value of COP observed are 0.316 for the mixture of R1270, R170 & R14 with the mass fraction of 0.205:0.18:0.615 and 0.267 for the mass fraction of 0.265:0.18:0.555.

Even though the COP is higher in the Zeotropic Mixture of mass fraction 0.205:0.18:0.615, it operates with lesser Exergic efficiency, the interpretation should be between the optimum values of COP, and Exergic efficiency will be the solution and thus recommendations can be made for the Zeotropic Mixture of R1270, R170 & R14 with the mass fraction of 0.265:0.18:0.555 as an alternative refrigerant for Three stage Auto Refrigerating Cascade (3 stage ARC) System which has lesser COP and higher Exergic Efficiency.

. Fig. 5 and Fig. 6 show the variation of Evaporating temperature and Refrigerating effect with the variation of mass fraction of R1270 and R14 in the Zeotropic Mixture of R1270, R170 and R14.

Even though the higher mass flow of high boiling refrigerant enable sufficient cooling for medium and low boiling refrigerant in condenser -1, the decreased mass flow rate of the low boiling refrigerant at evaporator-3 with higher terminal velocity reduces the refrigerating effect at lower evaporating temperatures.

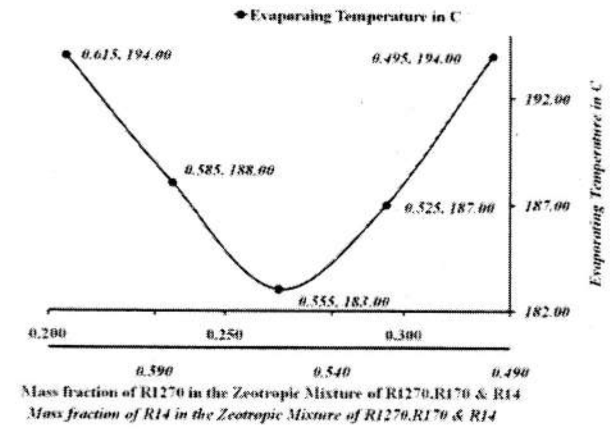


Fig. 5. Variation of Evaporating Temperature for Different Mass Fraction of R1270 & R14 in the Zeotropic Mixture of R1270, R170 & R14

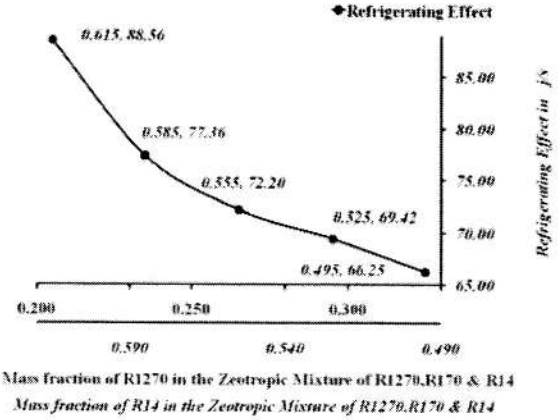


Fig. 6. Variation of Refrigerating Effect for Different Mass Fraction of R1270 & R14 in the Zeotropic Mixture of R1270, R170 & R14

The higher values of refrigerating effect is due to the increased mass flow of low boiling refrigerant assisting with less amount of compressor work and thus the higher COP.

The higher and lower values of refrigerating effect achieved during the trials are 88.56 W and 66.25 W which correspond to the Zeotropic mixtures of mass fractions 0.205:0.18:0.615 and 0.325:0.18:0.495.

Fig. 7 shows the variation of compressor work input with the variation of mass fraction of R1270 and R14 in the Zeotropic Mixture of R1270, R170 and R14.

The maximum and minimum values of these trials are 280.42 W and 246.95 W corresponds to the mixtures of mass fractions 0.205:0.18:0.615 and 0.325:0.18:0.495.

The trend of reduction of compressor work is observed during the trials due to the reduced mass flow of low boiling high density gas which has greater influence on compressor work but with better COP due to the increased mass flow of high boiling refrigerant which can be easily compressed and thus Compressor work.

Since the Compressor work need to be minimum at lowest evaporating temperatures, the Fig. 7 do not suggest the trial 5 with mass fraction 0.325:0.18:0.495.

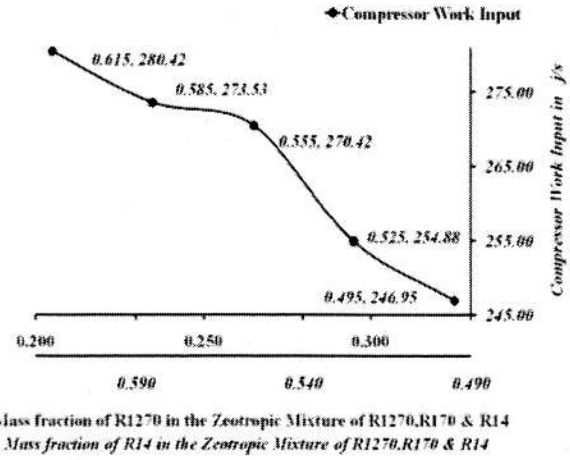


Fig. 7. Variation of Compressor Work Input for Different Mass Fraction of R1270 & R14 in the Zeotropic Mixture of R1270, R170 & R14

So it can be interpreted that the optimum between evaporating temperature and compressor work which results in the recommendation of the Zeotropic mixture of R1270, R170 & R14 of mass fraction 0.265:0.18:0.555 as an alternative refrigerant for Three stage Auto Refrigerating Cascade (3 stage ARC) System.

Fig. 8 show the variation of Exergy lost at compressor with the variation of mass fraction of R1270 and R14 in the Zeotropic Mixture of R1270, R170 and R14. The maximum and minimum values of Exergy lost at compressor are 199.51 W and 167.46 W corresponds to the mixture of mass fraction 0.265:0.18:0.555 having the value of 270.42W of compressor work input and 0.325:0.18:0.495 having the value of 246.95 W of compressor work input.

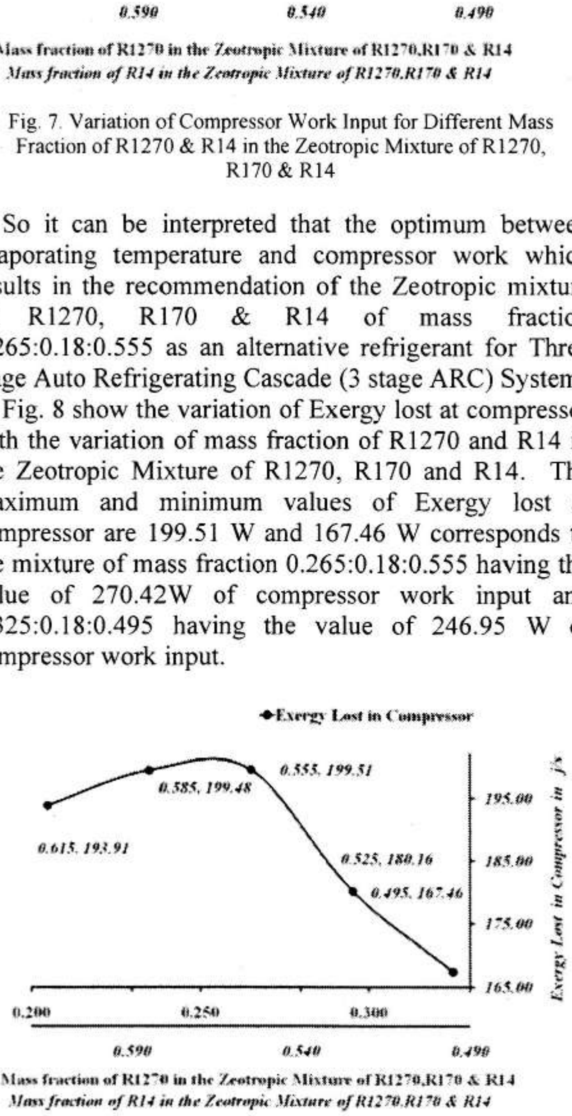


Fig. 8. Variation of Exergy Lost in Compressor for Different Mass Fraction of R1270 & R14 in the Zeotropic Mixture of R1270, R170 & R14

It is evident that whenever the temperature difference between the two state points is high, the exergy lost (Change in entropy is also high which intern the exergy or less availability) is also high.

Even though High compressor work leads to lesser COP and low compressor work to higher COP through the linear variation of exergy loss, this system performs little different due to the mixing of low boiling high density and high boiling low density gases which will increase the COP at both the cases.

The optimum of compressor work and exergy lost can be the solution and the yields through Fig. 8 and Fig. 7 is the Zeotropic mixture of R1270, R170 & R14 of mass fraction 0.265:0.18:0.555 as an alternative refrigerant for Three stage Auto Refrigerating Cascade (3 stage ARC) System with 270.42 W compressor work input and 199.51 W Exergy lost at compressor.

Fig. 9 shows the variation of Efficiency defect in each component of the ARC system with the variation of mass fraction of R1270 and R14 in the Zeotropic Mixture of R1270, R170 and R14.

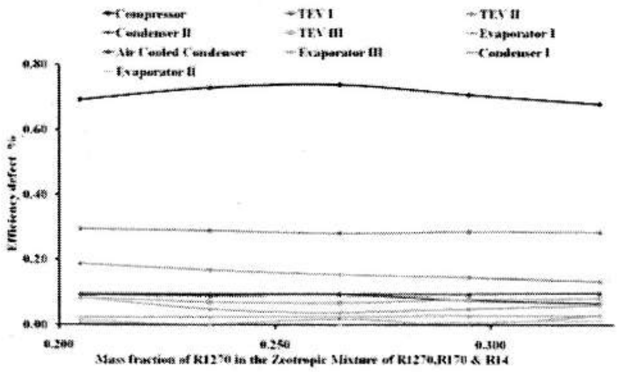


Fig. 9. Variation of Efficiency Defect in Each Component of ARC System for Different Mass Fraction of R1270 & R14 in the Zeotropic Mixture of R1270, R170 & R14

The maximum and minimum exergy defect are found at compressor (0.69, 0.73, 0.74, 0.71 and 0.68 correspond to trials 1, 2, 3, 4 and 5) and at Evaporator 1 (0.01, 0.02, 0.001, 0.01and 0.03 correspond to trials 1, 2, 3, 4 and 5). The phenomenon of increase in Efficiency defect is observed due to greater losses in compressor during trials 1, 2 and 3 with increased mass flow of high boiling refrigerant which has the tendency to be easily compressed and the excess energy available creates excess temperature difference and thus the Exergy loss and also the efficiency defect.

On the other hand the excess amount of work input during the trials 4 and 5 with the mass fraction of 0.295:0.18:0.525 and 0.325:0.18:0.495 are utilized to compress the low boiling high density gases which make sure less amount of temperature raise during operation and thus the lesser exergy losses compared to the trials of 1, 2 and 3 in each and every component. Since the results of the Fig. 9 follows the same phenomenon of ordinary Vapour Compression Refrigerating (VCR) system with the exergy losses and efficiency defect, the three stage (3 stage ARC) Auto Refrigerating Cascade system is efficient in dealing with the low temperature at the range of 183K (-90^{0}C).

The interpretations can reveal the concept of area of improvement or concentration should be on the compressor in the aspect of exergy loss and performance improvement of any Auto Refrigerating Cascade (ARC) System.

Fig. 10 shows the variation of Exergy destroyed in each component of the ARC system with the variation of mass fraction of R1270 and R14 in the Zeotropic Mixture of R1270, R170 and R14. The maximum and minimum exergy destructed are found at compressor (193.91, 199.48, 199.51, 180.16 and 167.46 correspond to trials 1, 2, 3, 4 and 5) and at Evaporator 2 (2.45, 0.86, 4.76, 1.01 and 7.34 correspond to trials 1, 2, 3, 4 and 5).

Since the same phenomenon of exergy destroyed as per the Fig. 10 can be utilized to calculate the efficiency defect of each and every component of the ARC system represented by Fig. 9, the interpretations holds good for the effective and efficient working of Three stage Auto Refrigerating Cascade (3 stage ARC) System.

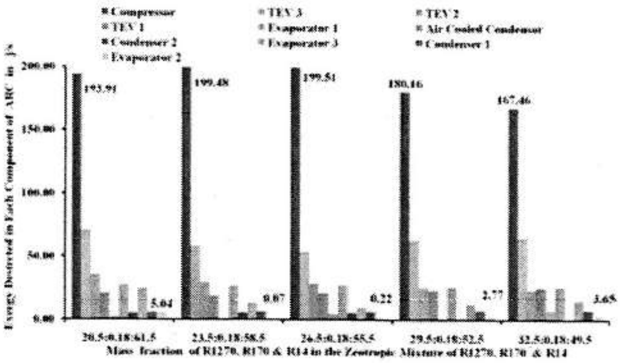


Fig. 10. Exergy Destroyed in Each Component of ARC System for the Different Mass Fraction of R1270 & R14 in the Mixture of R1270, R170 & R14

V. Conclusion

Exergic Analysis and performance analysis of Three Stage Auto Refrigerating Cascade (3 stage ARC) system was conducted on the setup operating between the evaporation and condensation temperatures of 183K (-90^{0}C) and 301K (30^{0}C) or over the temperature range of 190^{0}C:

- The Zeotropic Mixture of R1270/R170/R14 with the mass fraction of 0.265:0.18:0.555 having COP of 0.267 & 66.9% of Exergic Efficiency was recommended as an alternative refrigerant for Three stage Auto Refrigerating Cascade System working at the temperature range of 183K (-90^{0}C).
- The overall Efficiency defect is found to be at the range of 60's and 70's for the mixture of R1270/R170/R14 with the mass fraction of 0.265:0.18:0.555.
- The highest Efficiency defect was found to be at compressor and thus area of improvement lies in Compressor. Following is the order of components which has larger Efficiency defect. They are Compressor TEV's and Condensers.
- The highest Exergic Efficiency was found to be 66.9% for the Zeotropic mixture of R1270/R170/R14 with the mass fraction of 0.265:0.18:0.555 operating at the temperature range of 183K (-90^{0}C).
- In general the Zeotropic Mixture of R1270/R170/R14 with the mass fraction of 0.265:0.18:0.555 having COP of 0.267 & 66.9% of Exergic Efficiency is performing well in the Three Stage ARC System.

Thus it can be concluded that the Zeotropic mixture of R1270/R170/R14 with the mass fraction 0.265:0.18:0.555 as an alternative refrigerant for Three stage Auto Refrigerating Cascade (3 stage ARC) System with slight modification of the system.

References

[1] Eric W. Lemmon, Richard T Jacobsen, Equations of State for Mixtures of R-32, R-125, R-134a, R-143a, and R-152a, *Journal of Physical Chemical Reference Data*, Vol. 33, pp. 592 – 620, 2004.

[2] Ciro Aprea, Angelo Maiorino, Heat rejection pressure optimization for a carbon dioxide split system: An experimental study, *Applied Energy*, Vol.86, pp. 2373-2380, 2009.

[3] Michael Uhlmann, Stefan S. Bertsch, Theoretical and experimental investigation of start-up and shutdown behavior of Residential Heat Pumps, *International Journal of Refrigeration*, Vol.35, pp 2138-2149, 2012.

[4] J.H. Lee, S.W. Bae, K.H. Bang, M.H. Kim, Experimental and numerical research on condenser performance for R-22 and R-407C refrigerants, *International Journal of Refrigeration*, Volume: *25*, Page No: 372-382, 2002.

[5] Zanjun Gao, Yingjie Xu, Peng Li, Xiaolong Cui, Xiaohong Han, Qin Wang, Guangming Chen, Solubility of refrigerant trifluoromethane inN,N-dimethyl formamide in the temperature range from 283.15 K to 363.15 K, *International Journal of Refrigeration*, Vol.35, pp.1372-1376, 2012.

[6] Pradeep Bansal, Edward Vineyard, Omar Abdelaziz, Status of not-in-kind refrigeration technologies for household space conditioning, water heating and food refrigeration, International Journal of Sustainable Built Environment, Vol.1, pp. 85-101, 2012.

[7] X. Boissieux, M.R. Heikal, R.A. Johns, Two-phase heat transfer coefficients of three HFC refrigerants Inside a Horizontal Smooth Tube, part II: Condensation, *International Journal of Refrigeration*, Vol. 23, pp. 345-352, 2000.

[8] Ruixiang Wang, Qingping Wu, Yezheng Wu, Use of nano particles to make mineral oil lubricants feasible for use in a residential air conditioner employing hydro-fluorocarbons refrigerants, *Energy and Buildings*, Vol. 42, pp. 2111-2117, 2010.

[9] Jianyong Chen, Jianlin Yu, Performance of a New Refrigeration cycle using Refrigerant Mixture R32/R134a for residential air-conditioner Applications, *Energy and Buildings*, Vol.40, pp. 2022-2027, 2008.

[10] D.X. Jin, J.T. Kwon, M.H. Kim, Prediction of in-tube condensation heat transfer characteristics of Binary Refrigerant Mixtures, *International Journal of Refrigeration*, Vol.26, pp. 593-600, 2003.

[11] Mark O McLinden, Sanford A. Klein, Richard A. Perkins, An extended corresponding states model for the thermal conductivity of Refrigerants and refrigerants mixtures, *International Journal of Refrigeration*, Vol. 23, pp. 43 – 63, 2000.

[12] Jianlin Yu, Hua Zhao, Yanzhong Li, Application of an Ejector in Autocascade Refrigeration Cycle, for the Performance Improvement, *International Journal of Refrigeration*, Vol. 31, pp. 279-286, 2008.

[13] K. Comakli, F. Simsek, O. Comakli, B. Sahin, Determination of optimum working conditions R22 and R404A refrigerant mixtures in heat-pumps using Taguchi method, *Applied Energy*, Vol. 86, pp. 2451-2458, 2009.

[14] Primal Fernando, Bjorn Palm, Tim Ameel, Per Lundqvist, Eric Granryd, A Mini channel Aluminium tube heat Exchanger PART - I Evaporator Performance with Propane, *International Journal of Refrigeration*, Vol. 31, pp. 681 – 695, 2008.

[15] Ki-Jung Park, Dongsoo Jung, Performance of heat pumps charged with R170/R290 mixture, *Applied Energy*, Vol. 86, pp. 2598-

2603, 2009.

[16] Primal Fernando, Bjorn Palm, Per Lundqvist, Eric Granryd, Propane heat Pump with low Refrigerant charge: Design and Laboratory tests, *International Journal of Refrigeration*, Vol. 27, pp. 761-773, 2004.

[17] Dongsoo Jung , Chong-Bo Kim, Kilhong Song, Byoungjin Park, Testing of Propane/Isobutane Mixture in Domestic Refrigerators, *International Journal of Refrigeration*, Vol.23, pp. 517-527, 2000.

[18] D.J. Cleland, R.W. Keedwell, S.R. Adams, Use of hydrocarbons as drop-in Replacements for HCFC-22 in on-farm Milk Cooling Equipment, *International Journal of Refrigeration*, Vol. 32, pp.1403-1411, 2009.

[19] S. Wongwiscs, T. Songnetichaovalit, N. Lokathada, P. Kritsadathikarn, M. Suchatawat and W. Pirompak, A Comparison of the flow characteristics of Refrigerants Flowing through Adiabatic Capillary Tubes, *International Comm. Heat Mass Transfer*, Vol. 27, pp. 611-621, 2000.

[20] Jianfeng Wu, Quiwei Cheng, Zhaohu Sun, Jiayong Liu, Qinguo Hu, Maoqiong Gong, Development of a -186^{0}C cryogenic preservation chamber based on a dual mixed-gases Joule Thomson refrigeration cycle, *Applied Thermal Engineering*, Vol. 36, pp. 188-192, 2012.

[21] Liu Jie, Pei Nian-qiang, Guo Kai-hua, He Zhen-hui, Li Ting-xuen, Gu Jian-ming, Experimental investigation on start up of a novel two-phase cooling loop, *Experimental Thermal and Fluid Science*, Vol. 32, pp. 939-946, 2008.

[22] Andrey Rozhentsev, Vjacheslav Naer, Investigation of the Starting modes of the low-temperature Refrigerating Machines working on the Mixtures of Refrigerants, *International Journal of Refrigeration*, Vol. 32, pp. 901-910, 2009.

[23] D.-Y. Lee, Y. Ahn, Y. Kim, Y. Kim, Y.-S. Chang, L. Nam, Experimental Investigation on the drop-in performance of R407C as a substitute for R22 in a screw Chiller with shell-and-tube heat exchangers, *International Journal of Refrigeration*, Vol. 25, pp. 575-585, 2002.

[24] Dave Sajjan, Tord Karlsson, Lennart Vamling, Reasons for drop in shell - and - tube Condenser Performance when Replacing R22 with Zeotropic Mixtures. Part 1: Analysis of Experimental Findings, *International Journal of Refrigeration*, Vol. 27, pp. 552-560, 2004.

[25] Andrey Rozhentsev, Refrigerating machine operating characteristics under various mixed refrigerant mass charges, *International Journal of Refrigeration*, Vol. 31, pp. 1145-1155, 2008.

[26] A. Johansson, P. Lundqvist, A method to Estimate the Circulated Composition in Refrigeration and Heat Pump systems using Zeotropic Refrigerant Mixtures, *International Journal of Refrigeration*, Vol. 24, pp. 798 – 808, 2001.

[27] Kai Du, Shaoquian Zhang, Weirong Xu, Xiaofeng Niu, A Study on the Cycle Characteristics of an Auto-Cascade Refrigeration System, *Experimental Thermal and Fluid Science*, Vol. 33, pp. 240-245, 2009.

[28] Yijian He, Guangming Chen, Experimental study on an Absorption Refrigeration system at low Temperatures, *International Journal of Thermal Sciences*, Vol. 46, pp. 294-299, 2007.

[29] Giovanni Di Nicola, Giuliano Giuliani, Fabio Polonara, Roman Stryjek, Blends of carbon dioxide and HFCs as working fluids for the low-temperature circuit in cascade refrigerating systems, *International Journal of Refrigeration*, Vol. 28, pp. 130-140, 2005.

[30] Thangavel, P., Somasundaram, P., Navaneethakrishnan, P., Simulation analysis on thermo physical properties of hydrocarbon refrigerants in vapour compression refrigeration system, (2013) *International Review of Mechanical Engineering (IREME)*, 7 (4), pp. 646-651.

[31] S. Frikha, M. S. Abid, Performance Analysis of Irreversible Combined Refrigeration Cycles Based on Finite Time Thermodynamic Theory, (2008) *International Review of Mechanical Engineering (IREME)*, 2 (2), pp. 325 - 335.

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