Methods for Improving First and Second Law Efficiencies of Vapour Compression Refrigeration Systems Using Flash-Intercooler with Ecofriendly Refrigerants

Kapil Chopra *, V. Sahni, R. S. Mishra
Department Of Mechanical Engineering, Delhi Technological University, New Delhi, India

Article Info

Article history:
Received 2 January 2014
Received in revised form
29 January 2014
Accepted 20 February 2014
Available online 15 March 2014

Keywords

Abstract

In the present thermodynamic analysis, the comparison and impact of environmental friendly refrigerants(R410a, R290, R600, R600a, R1234yf, R125, R717 and R134A)on multiple stage vapour compression refrigerator with flash intercooler and individual throttle valves (system-1) and multiple stage vapour compression refrigerator with flash intercooler and multiple throttle valves (system-2) has been carried out on the basis of energetic and exergetic approach. It wasobserved that for all selected ecofriendly refrigerants, energy and exergy efficiency of system-1is lower than sytem-2.For both systems R125 showed poorperformance in terms of energetic efficiency, second law efficiency and irreversibility whereas thermodynamic performances of hydrocarbon isobutene (i.e. R600) and R717 gives better performance in comparison of other selected ecofriendly refrigerants. AsR717 is toxic in nature and restricted to limited applications, and also R600 is flammable in nature, therefore R-134a is suggested better for practical applications for both systemswithout taking of any safety precautions. Although thermodynamic performance of R134a is only 2-3% lesser than R600 and R134A is easily available in the market

1. Introduction

Nowadays most of the energy utilize in cooling and air conditioning in industrial as well as for domestic applications, in addition to energy consumption, using of refrigerants in cooling and air conditioning having high GWP and ODP which are responsible for increasing global warming and ozone depletion. The primary requirements of ideal refrigerants are having good physical and chemical properties. Due to good physical and chemical properties such as non-corrosiveness, non-toxicity, non- flammability, low boiling point, Chlorofluorocarbons (CFCs) have been used over the last many decades, but hydro chlorofluorocarbons (HCFCs) and Chlorofluorocarbons (CFCs) having large amount of chlorine content as well as high global warming potential and ozone depletion potential, so after 90s refrigerants under these categories these kinds of refrigerants are almost prohibited [1].Most of the study has been carried out for the performance evaluation of vapour compression refrigeration system using energetic analysis, but with the help of first law analysis irreversibility destruction or losses in components of system unable to determined [2], so exergetic analysis is the advanced approach for thermodynamic analysis which gives an additional practical view of the processes [3-5]. The utility of second law analysis on vapour compression refrigeration systems is well defined because it gives the idea for improvements in efficiency due to modifications in existing design in terms of reducing exergy destructions in the components. In addition to this second law analysis also provides new thought for development in the existing system [6]. Xuan and Chen [7] presented in this manuscript about the replacement of R502 by mixture of HFC-161.Throug

Corresponding Author,
E-mail address: technokapilchopra@gmail.com

h experimental study it was found that mixture of HFC-161 gives same and higher performance than R404A at lower and higher evaporative temperature respectively on the vapour compression refrigeration system designed for R404A.Cabello et al. [8] effect of condensing pressure, evaporating pressure and degree of superheating was experimentally investigated on single stage vapour compression refrigeration system using R22, R134a and R407C.It was observed that mass flow rate is greatly affected by change in suction conditions of compressor in results on refrigeration capacity because refrigeration capacity depended on mass flow rate through evaporator. It was also found that for higher compression ratio R22 gives lower COP than R407C.Spatz and Motta [9] focused on replacement of R12 with R410a through experimental investigation of medium temperature vapour compression refrigeration cycles. In terms of thermodynamic analysis, comparison of heat transfer and pressure drop characteristics, R410a gives best performance among R12, R404a and R290a.Han et al. [10] under different working conditions experimental results revealed that there could be replacement of R407C in vapour compression refrigeration system having rotor compressor with mixture of R32/R125/R161 showing higher COP, less pressure ratio and slightly high discharge compressor temperature without any modification in the same system. Cabello et al [11] had studied about the effect of operating parameters on COP, work input and cooling capacity of single-stage vapour

Nomenclature

LTE	low temperature evaporator	Φ	specific enthalpy (kJ/kg)
ITE	intermediate temperature evaporator	$\dot{\Psi}$	irreversibility rate(kW)
THE	high temperature evaporator	C	compressor
TR	ton of refrigeration	X	exergy rate of fluid (kW)
P	power (kJ/s)	$\dot{m}$	mass flow rate (kg/s)
F	flash intercooler	S	specific entropy (kJ/kgK)
$\dot{Q}$	rate of heat transfer (kW)	$\dot{EP}$	exergy rate of product (kW)
$\dot{W}$	work rate (kW)		
T	temperature (^{o}C)		
TV	throttle valve		
X	dryness fraction(non-dimensional)		
Subscript			
E	Evaporator	Tv	throttle valve
C	Compressor	Lsc	liquid subcooler
O	dead state	K	kth component
F	flash intercooler	Gen	generation
R	Refrigerant	cond	condenser

Refrigerant	Chemical formula	Molecular mass(g/mol)	NBP (^{o}C)	T_{cri} (^{o}C)	P_{cri} (MPa)	ASHRAE safety code
R410A	R-32/125	72.58	-60.9	72.5	4.95	A1
R290	$CH_3CH_2CH_3$	44.1	-42.2	96.7	4.25	A3
R600A	C_4H_{10}	58.122	-11.74	134.661	3.62	A3
R1234YF	$C_3H_2F_4$	114.04	-29.4	94.85	3.38	A2L
R600	C_4H_{10}	58.122	-0.49	151.98	3.79	A3
R134A	CH_2FCF_3	102.03	-26.1	101.1	4.06	A1
R125	C_2HF_5	120.02	-48.09	66.023	3.61	A1
R717	NH_3	17.03	-33.327	132.25	11.33	B2

Table: 1. Physical and environmental characteristics of considered refrigerants [2, 3]

compression refrigeration system. There is great influence on energetic parameters due change in suction pressure, condensing and evaporating temperatures. Arora and Kaushik [12] developed numerical model of actual vapour compression refrigeration system with liquid vapour heat exchanger and did energy and exergy analysis on the same in the specific temperature range of evaporator and condenser. They concluded that R502 is the best refrigerant compare to R404A and R507A, compressor is the worst and liquid vapour heat exchanger is best component of the system. Getu and Bansal [13] had optimized the design and operating parameters of like condensing temperature, sub cooling temperature, evaporating temperature, superheating temperature and temperature difference in cascade heat exchanger R744-R717 cascade refrigeration system. A regression analysis was also done to obtain optimum thermodynamic parameters of same system. Mohanraj et al [14]through experimental investigation of domestic refrigerator they arrived on conclusions that under different environmental temperatures COP of system using mixture of R290 and R600a in the ratio of 45.2: 54.8 by weight showing up to 3.6% greater than same system using R134a, also discharge temperature of compressor with mixture of R290 and R600a is lower in the range of 8.5-13.4K than same compressor with R134a.Padilla et al [15] exergy analysis of domestic vapour compression refrigeration system with R12 and R413A was done. They concluded that performance in terms of power consumption, irreversibility and exergy efficiency of R413A is better than R12.In this paper great emphasis put on saving of energy and using of ecofriendly refrigerants due to increase of energy crises, global warming and depletion of ozone layer. In this investigation the work input required running the vapour compression refrigeration system reduced by using compound compression and further decreased by flash intercooling between compressors.COP of system can also be enhanced by compressing the refrigerant very close to the saturation line this can be achieved by compressing the refrigerants in more stages with intermediate intercoolers. The refrigeration effect can be increase by maintaining the condition of refrigerants in more liquid stage at the entrance of evaporator which can be achieved by expanding the refrigerant very close to the liquid line. The expansion can be brought close to the liquid line by sub cooling the refrigerant and removing the flashed vapours by incorporating the flash chamber in the working cycle. The evaporator size can be reduced because unwanted vapours formed are removed before the liquid refrigerant enters in the evaporator. Multi-stage vapour compression with flash intercooler and individual throttle valves (system-1) consists of three compressors arranged in compound compression, individual throttle valves, condenser and evaporators as shown in Fig.1.Multiple evaporators at different temperatures with compound compression, flash intercooler and multiple throttle valves (system-2) consists of three compressors arranged in compound compression, multiple throttle valves, condenser and evaporators as shown in Fig.2.

2. Energy and Exergy Analysis

For carrying out energetic and exergetic analysis, computational models of system-1 and system-2 has been developed and impact of chosen refrigerants on these systems has been analyzed using Engineering Equation Solver software[16].In this investigation following assumptions are made:

1. Load on the low, intermediate and high temperature evaporators are 10TR, 20 TR and 30 TR respectively.
2. Dead state temperature (T_0): 25 ^oC
3. Difference between evaporator and space temperature (T_r-T_e):5 ^oC.
4. Adiabatic efficiency of compressor: 76%.
5. Dead state enthalpy (Φ_0) and entropy (s_0) of the refrigerants have been calculated corresponding to the dead state temperature (T_0) of 25 ^oC.
6. Variation in kinetic and potential energy is negligible.
7. Expansion process is adiabatic
8. Temperature of low, intermediate and high temperature evaporators are -10 ^oC, 0 ^oC and 10 ^oC respectively.
9. Condenser temperature : 40 ^oC
10. Degree of sub cooling : 10 ^oC

Exergy at any state is given as

$$X = (\Phi - \Phi_0) - T_0(s - s_0) \quad (1)$$

Energy analysis

First law of thermodynamic gives the idea of energy balance of system.

Mass flow analysis of system-1

$$\dot{m}_{c1} = \dot{m}_{e1} = \frac{\dot{Q}_{e1}}{(\Phi_1 - \Phi_{10})} \quad (2)$$

$$\dot{m}_{e2} = \frac{\dot{Q}_{e2}}{(\Phi_3 - \Phi_9)} \quad (3)$$

$$\dot{m}_{f1} = \frac{\dot{m}_{c1}(\Phi_2 - \Phi_3)}{(\Phi_3 - \Phi_9)} \tag{4}$$

$$\dot{m}_{c2} = \dot{m}_{c1} + \dot{m}_{e2} + \dot{m}_{f1} \tag{5}$$

$$\dot{m}_{e3} = \frac{\dot{Q}_{e3}}{(\Phi_5 - \Phi_8)} \tag{6}$$

$$\dot{m}_{f2} = \frac{\dot{m}_{c2}(\Phi_4 - \Phi_5)}{(\Phi_5 - \Phi_8)} \tag{7}$$

$$\dot{m}_{c3} = \dot{m}_{c2} + \dot{m}_{e3} + \dot{m}_{f2} \tag{8}$$

Energy consumption for sytem-1

$$P_{c1} = \frac{\dot{m}_{c1}(\Phi_2 - \Phi_1)}{60} \tag{9}$$

$$P_{c2} = \frac{\dot{m}_{c2}(\Phi_4 - \Phi_3)}{60} \tag{10}$$

$$P_{c3} = \frac{\dot{m}_{c3}(\Phi_6 - \Phi_5)}{60} \tag{11}$$

Energetic efficiency of system-1

$$\mathrm{COP} = \frac{\dot{Q}_e}{P_c * 60} \tag{12}$$

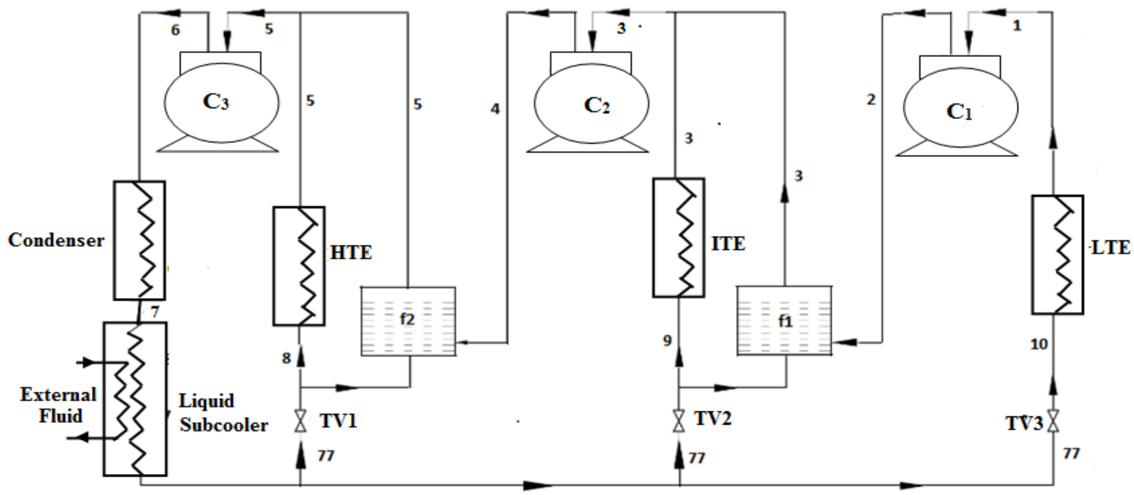

Fig: 1. Schematic diagram of multiple evaporators with compound compression, flash intercooler and individual throttle valves

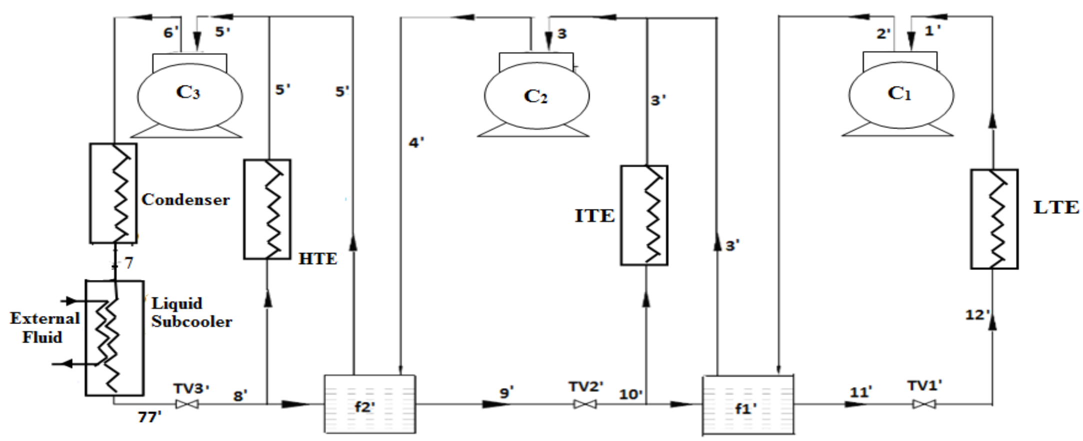

Fig: 2. Schematic diagram of multiple evaporators with compound compression, flash intercooler and multiple throttle valves

Rate of exergy loss due to irreversibility $(T_o\dot{S}_{gen})$ in various components of system-1

The concept of exergy was given by second law of thermodynamics, which always decreases due to thermodynamic irreversibility. Exergy is defined as 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 [12].

Compressors

$$\left(T_o\dot{S}_{gen}\right)_{c1} = \dot{W}_{c1} + m_{c1}(X_2 - X_1) \quad (13)$$

$$(T_o\dot{S}_{gen})_{c2} = \dot{W}_{c2} + m_{c2}(X_4 - X_3) \quad (14)$$

$$(T_o\dot{S}_{gen})_{c3} = \dot{W}_{c3} + m_{c3}(X_6 - X_5) \quad (15)$$

$$\Psi_c = (T_o\dot{S}_{gen})_{c1} + (T_o\dot{S}_{gen})_{c2} + (T_o\dot{S}_{gen})_{c3} \quad (16)$$

Evaporators

$$(T_o\dot{S}_{gen})_{e1} = \dot{m}_{e1}(X_1 - X_{10}) - \dot{Q}_{e1}\left(1 - \frac{T_0}{T_{r1}}\right) \quad (17)$$

$$\left(T_o\dot{S}_{gen}\right)_{e2} = \dot{m}_{e2}(X_3 - X_9) - \dot{Q}_{e2}\left(1 - \frac{T_0}{T_{r2}}\right) \quad (18)$$

$$(T_o\dot{S}_{gen})_{e3} = \dot{m}_{e3}(X_5 - X_8) - \dot{Q}_{e3}\left(1 - \frac{T_0}{T_{r3}}\right) \quad (19)$$

$$\Psi_e = (T_o\dot{S}_{gen})_{e1} + (T_o\dot{S}_{gen})_{e2} + (T_o\dot{S}_{gen})_{e3} \quad (20)$$

Condenser

$$\Psi_{cond} = \left(T_o\dot{S}_{gen}\right)_{cond} = \dot{m}_{c3}(X_6 - X_7) - \dot{Q}_e\left(1 - \frac{T_0}{T_r}\right) \quad (21)$$

Throttle Valves

$$(T_o\dot{S}_{gen})_{tv1} = \dot{m}_{e1}(X_{77} - X_{10}) \quad (22)$$

$$\left(T_o\dot{S}_{gen}\right)_{tv2} = (\dot{m}_{e2} + \dot{m}_{f1})(X_{77} - X_9) \quad (23)$$

$$(T_o\dot{S}_{gen})_{tv3} = (\dot{m}_{e3} + \dot{m}_{f2})(X_{77} - X_8) \quad (24)$$

$$\Psi_{tv} = (T_o\dot{S}_{gen})_{tv1} + (T_o\dot{S}_{gen})_{tv2} + (T_o\dot{S}_{gen})_{tv3} \quad (25)$$

Liquid sub cooler

$$\Psi_{lsc} = (T_o\dot{S}_{gen})_{sc} = \dot{m}_{c3}(X_7 - X_{77}) \quad (26)$$

Flash intercoolers

$$(T_o\dot{S}_{gen})_{f1} = \dot{m}_{f1}(X_9 - X_3) + \dot{m}_{c1}(X_2 - X_3) \quad (27)$$

$$(T_o\dot{S}_{gen})_{f2} = \dot{m}_{f2}(X_8 - X_5) + \dot{m}_{c1}(X_4 - X_5) \quad (28)$$

$$\Psi_f = (T_o\dot{S}_{gen})_{f1} + (T_o\dot{S}_{gen})_{f2} \quad (29)$$

Total irreversibility destruction in system-1

$$\sum \Psi_k = \Psi_e + \Psi_c + \Psi_{cond} + \Psi_{tv} + \Psi_{lsc} + \Psi_f \quad (30)$$

$$\dot{m}_{c1'} = \dot{m}_{e1'} = \frac{\dot{Q}_{e1'}}{(\Phi_{1'} - \Phi_{12'})} \quad (31)$$

$$\dot{m}_{e2'} = \frac{\dot{Q}_{e2'}}{(\Phi_{3'} - \Phi_{10'})} + \dot{m}_{c1'}\left(\frac{x_{10'}}{1 - x_{10'}}\right) \quad (32)$$

$$\dot{m}_{f1'} = \frac{\dot{m}_{c1'}(\Phi_{2'} - \Phi_{3'})}{(\Phi_{3'} - \Phi_{10'})} \quad (33)$$

$$\dot{m}_{c2'} = \dot{m}_{c1'} + \dot{m}_{e2'} + \dot{m}_{f1'} \quad (34)$$

$$\dot{m}_{e3'} = \frac{\dot{Q}_{e3'}}{(\Phi_{5'} - \Phi_{8'})} + \dot{m}_{c2'}\left(\frac{x_{8'}}{1 - x_{8'}}\right) \quad (35)$$

$$\dot{m}_{f2'} = \frac{\dot{m}_{c2'}(\Phi_{4'} - \Phi_{5'})}{(\Phi_{5'} - \Phi_{8'})} \quad (36)$$

Power required for running the compressors

$$P_{c1'} = \frac{\dot{m}_{c1'}(\Phi_{2'} - \Phi_{1'})}{60} \quad (37)$$

$$P_{c2'} = \frac{\dot{m}_{c2'}(\Phi_{4'} - \Phi_{3'})}{60} \quad (38)$$

$$P_{c3'} = \frac{\dot{m}_{c3'}(\Phi_{6'} - \Phi_{5'})}{60} \quad (39)$$

$$\text{Energetic efficiency} = \frac{\dot{Q}_{e'}}{P_{c'} * 60} \quad (40)$$

2.1 Rate of exergy loss due to irreversibilties $(T_o\dot{S}_{gen})$ in various components of system-2

Compressors

$$(T_o\dot{S}_{gen})_{c1'} = \dot{W}_{c1'} + m_{c1'}(X_{2'} - X_{1'}) \quad (41)$$

$$(T_o\dot{S}_{gen})_{c2'} = \dot{W}_{c2'} + m_{c2'}(X_{4'} - X_{3'}) \quad (42)$$

$$(T_o\dot{S}_{gen})_{c3'} = \dot{W}_{c3'} + m_{c3'}(X_{6'} - X_{5'}) \quad (43)$$

$$\Psi_{c'} = (T_o\dot{S}_{gen})_{c1'} + (T_o\dot{S}_{gen})_{c2'} + (T_o\dot{S}_{gen})_{c3'} \quad (44)$$

Evaporators

$$(T_o\dot{S}_{gen})_{e1'} = \dot{m}_{e1'}(X_{1'} - X_{12'} - \dot{Q}_{e1'}\left(1 - \frac{T_0}{T_{r1'}}\right) \quad (45)$$

$$(T_o\dot{S}_{gen})_{e2'} = \dot{m}_{e2'}(X_{3'} - X_{10'}) - \dot{Q}_{e2'}\left(1 - \frac{T_0}{T_{r2'}}\right) \quad (46)$$

$$(T_o\dot{S}_{gen})_{e3'} = \dot{m}_{e3'}(X_{5'} - X_{8'}) - \dot{Q}_{e3'}\left(1 - \frac{T_0}{T_{r3'}}\right) \quad (47)$$

$$\Psi_{e'} = (T_o\dot{S}_{gen})_{e1'} + (T_o\dot{S}_{gen})_{e2'} + (T_o\dot{S}_{gen})_{e3'} \quad (48)$$

Condenser

$$\Psi_{cond'} = (T_o\dot{S}_{gen})_{cond'}$$

$$= \dot{m}_{c3'}(X_{6'} - X_{7'}) - \dot{Q}_{e'}\left(1 - \frac{T_0}{T_{r'}}\right) \quad (49)$$

Throttle Valves

$$(T_o\dot{S}_{gen})_{tv1'} = \dot{m}_{e1'}(X_{11'} - X_{12'}) \quad (50)$$

$$(T_o\dot{S}_{gen})_{tv2'} = \dot{m}_{c2'}(X_{9'} - X_{10'}) \quad (51)$$

$$(T_o\dot{S}_{gen})_{tv3'} = \dot{m}_{c3'}(X_{77'} - X_{8'}) \quad (52)$$

$$\Psi_{tv'} = (T_o\dot{S}_{gen})_{tv1'} + (T_o\dot{S}_{gen})_{tv2'} + (T_o\dot{S}_{gen})_{tv3'} \quad (53)$$

Liquid sub cooler

$$\Psi_{lsc'} = (T_o\dot{S}_{gen})_{lsc'} = \dot{m}_{c3'}(X_{7'} - X_{77'}) \quad (54)$$

Flash intercoolers

$$(T_o\dot{S}_{gen})_{f1'} = \dot{m}_{f1'}(X_{10'} - X_{3'}) + \dot{m}_{c1'}(X_{2'} - X_{3'}) \quad (55)$$

$$(T_o\dot{S}_{gen})_{f2'} = \dot{m}_{f2'}(X_{8'} - X_{5'}) + \dot{m}_{c2'}(X_{4'} - X_{5'}) \quad (56)$$

$$\Psi_{f'} = (T_o\dot{S}_{gen})_{f1'} + (T_o\dot{S}_{gen})_{f2'} \quad (57)$$

Total irreversibility destruction in system-1

$$\sum \Psi_{k'} = \Psi_{e'} + \Psi_{c'} + \Psi_{cond'} + \Psi_{tv'} + \Psi_{lsc'} + \Psi_{f'} \quad (58)$$

Exergetic efficiency

$$\text{Exergetic efficiency} = \frac{\text{Exergy of cooling load of}}{\text{Compressors wo}} \quad (59)$$

$$\text{Exergetic efficiency of system} - 1 = \frac{(\dot{Q}_{e1} + \dot{Q}_{e2} + \dot{Q}_{e3}) - T_0\left(\frac{\dot{Q}_{e1}}{T_{r1}} + \frac{\dot{Q}_{e2}}{T_{r2}} + \frac{\dot{Q}_{e3}}{T_{r3}}\right)}{P_c * 60} \quad (60)$$

$$\text{Rational efficiency of system} - 2 = \frac{(\dot{Q}_{e1'} + \dot{Q}_{e2'} + \dot{Q}_{e3'}) - T_0\left(\frac{\dot{Q}_{e1'}}{T_{r1'}} + \frac{\dot{Q}_{e2'}}{T_{r2'}} + \frac{\dot{Q}_{e3'}}{T_{r3'}}\right)}{P_{c'} * 60} \quad (61)$$

3. Results and discussions

Variation in low, intermediate and high temperature evaporator with coefficient of performance for considered refrigerants of system-1 and system-2 is shown by Fig.3-5 and Fig.6-8 respectively. Both systems (system-1& system-2) were analytically analyzed and it was observed that COP (energetic efficiency) of system-2 is higher than system-1.

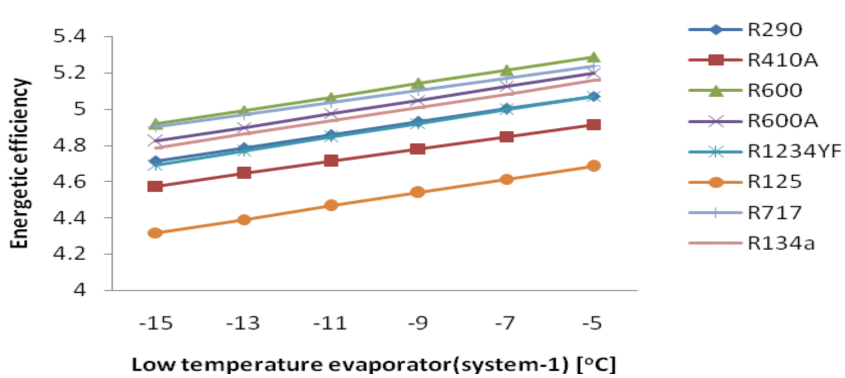

Fig: 3. Variation of low temperature evaporator of system-1with energetic efficiency

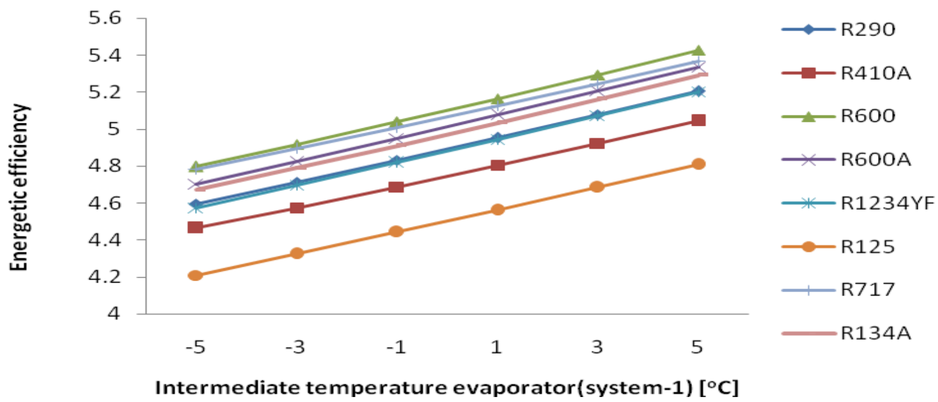

Fig: 4. Variation of intermediate temperature evaporator of system-1with energetic efficiency

The COP of both system-1 and system-2 increase with increase in evaporator temperature for chosen refrigerants. It was also observed that R600 and R717 show better performance and R125 gives poor performance in term of energetic efficiency than other refrigerants for both systems. The maximum percentage difference of COP was observed in high temperature evaporator of system-2 and system-1 is 9.59% for R125 at 15 ^{o}C,

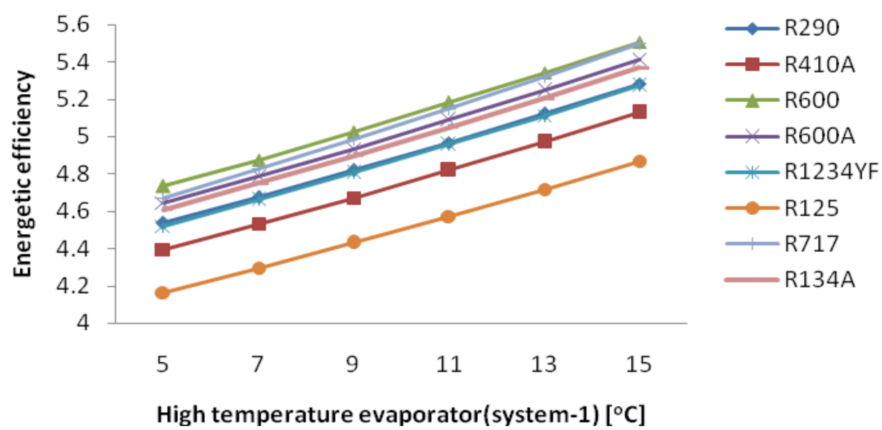

Fig: 5. Variation of high temperature evaporator of system-1with energetic efficiency

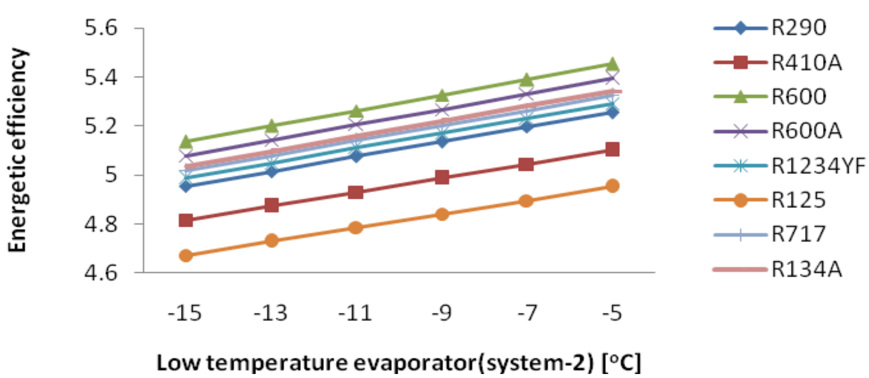

Fig: 6. Variation of low temperature evaporator of system-2 with energetic efficiency

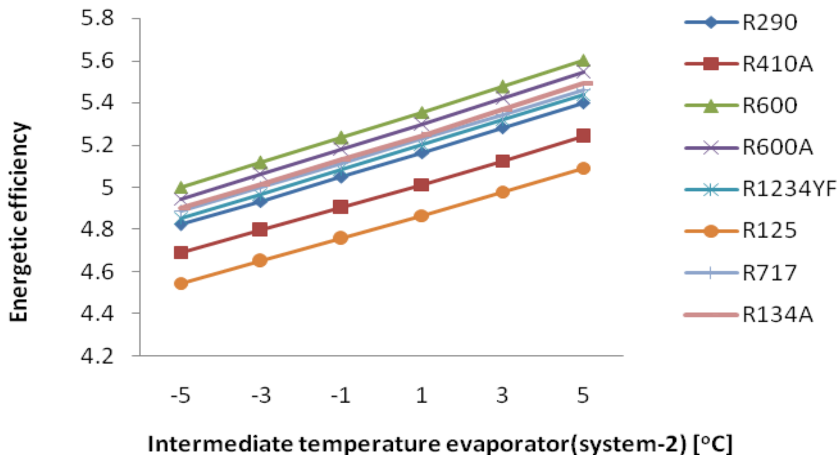

Fig: 7. Variation of intermediate temperature evaporator of system-2 with energetic efficiency

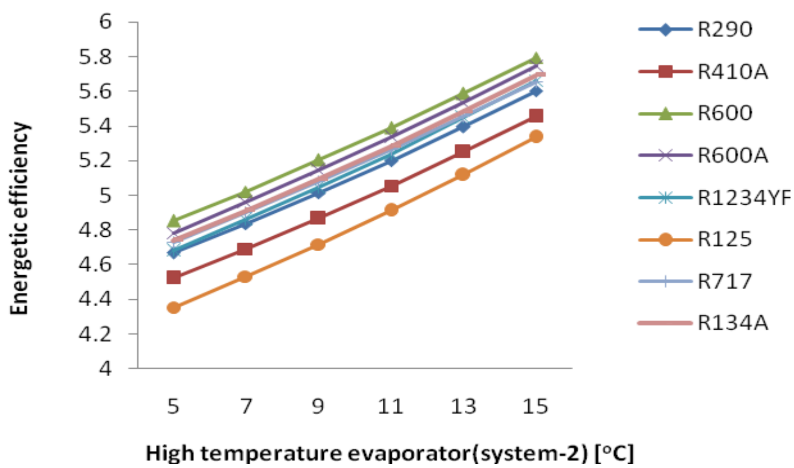

Fig: 8. Variation of high temperature evaporator of system-2 with energetic efficiency

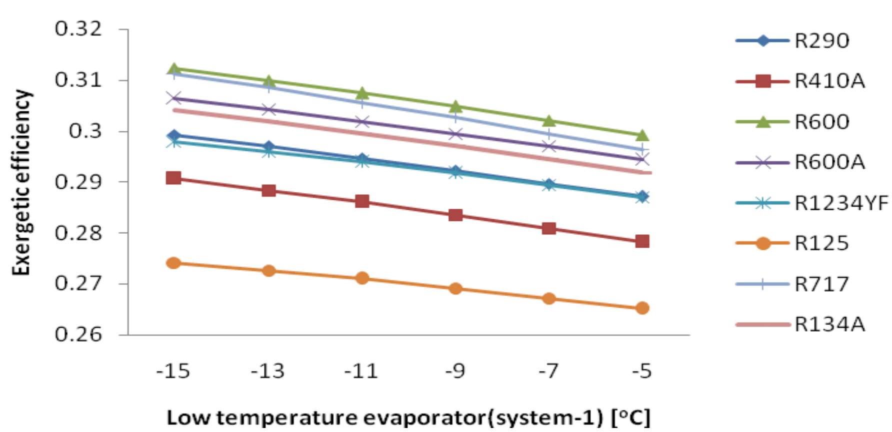

Fig: 9. Variation of low temperature evaporator of system-1with exergetic efficiency

The impact on second law efficiency (exergetic efficiency) with change in temperature of low, intermediate and high temperature evaporator of system-1 and system-2 shown by Fig.9-11and Fig.11-14 respectively. As variation in second law efficiency is reciprocal to coefficient of performance. It is also observed that second law efficiency decrease with increase in evaporator temperature. R600 and R125 have maximum and minimum second law efficiency for both systems similar to performance evaluation in terms of energetic efficiency. It was also found that temperature variation in low and intermediate evaporator put great impact on second law efficiency in comparison with high temperature evaporator, for both systems

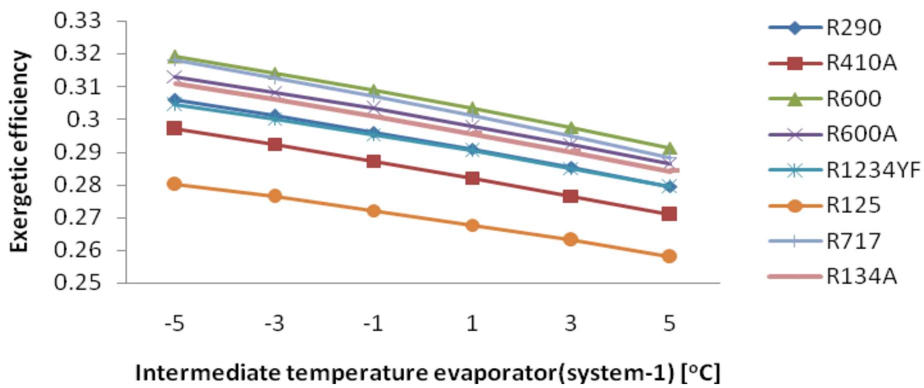

Fig: 10. Variation of intermediate temperature evaporator of system-1with exergetic efficiency

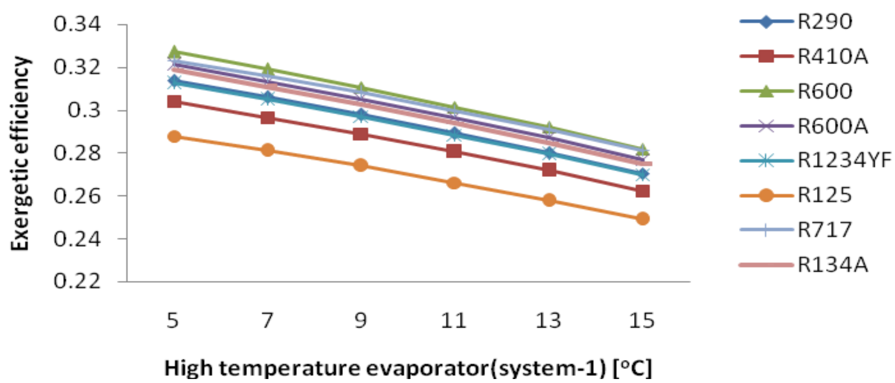

Fig: 11. Variation of high temperature evaporator of system-1with exergetic efficiency

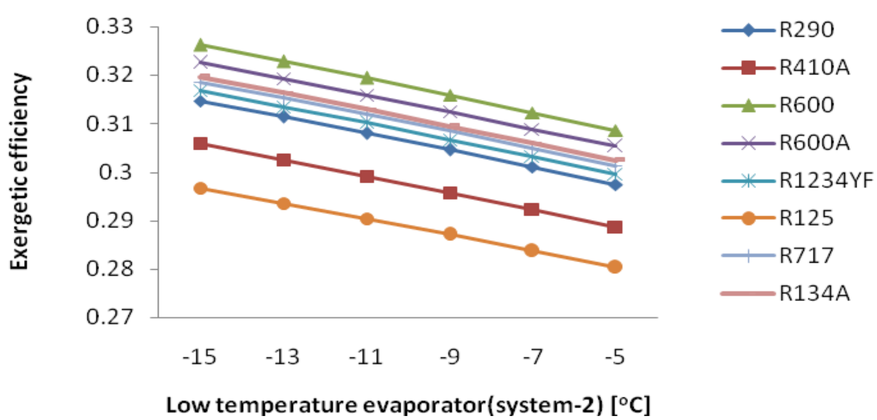

Fig: 12. Variation of low temperature evaporator of system-2 with exergetic efficiency

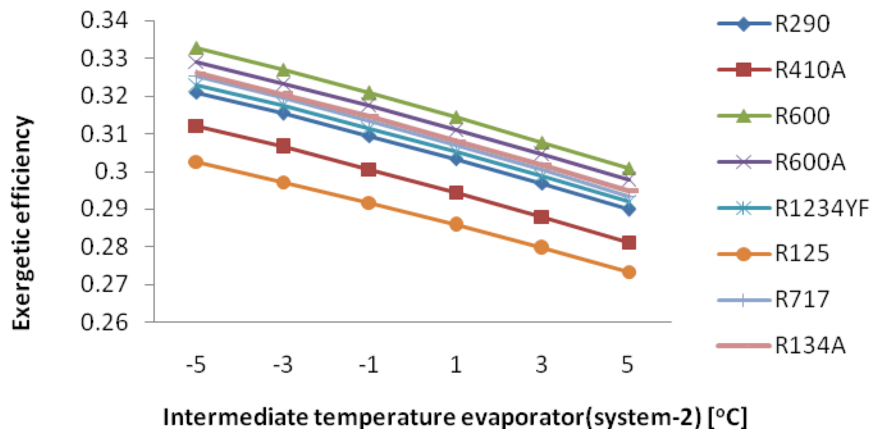

Fig: 13. Variation of intermediate temperature evaporator of system-2 with exergetic efficiency

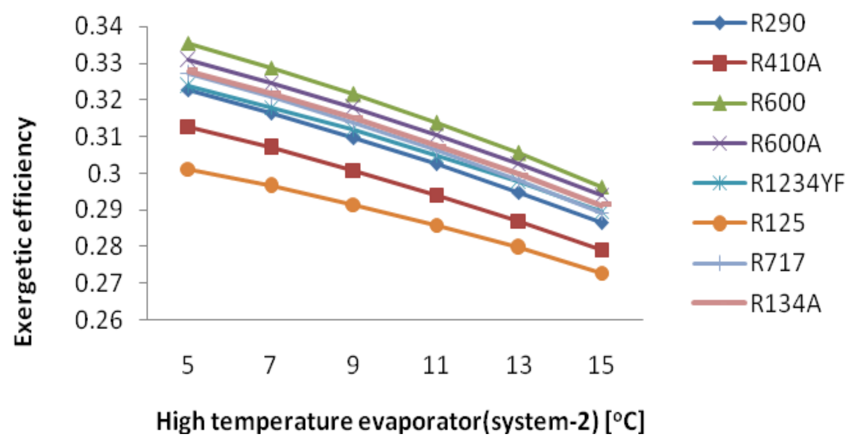

Fig: 14. Variation of high temperature evaporator of system-2 with exergetic efficiency

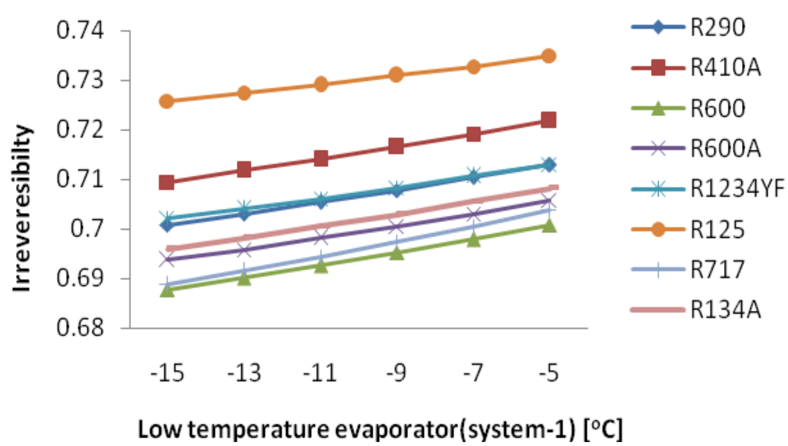

Fig: 15. Variation of low temperature evaporator with irreversibility of sytem-1

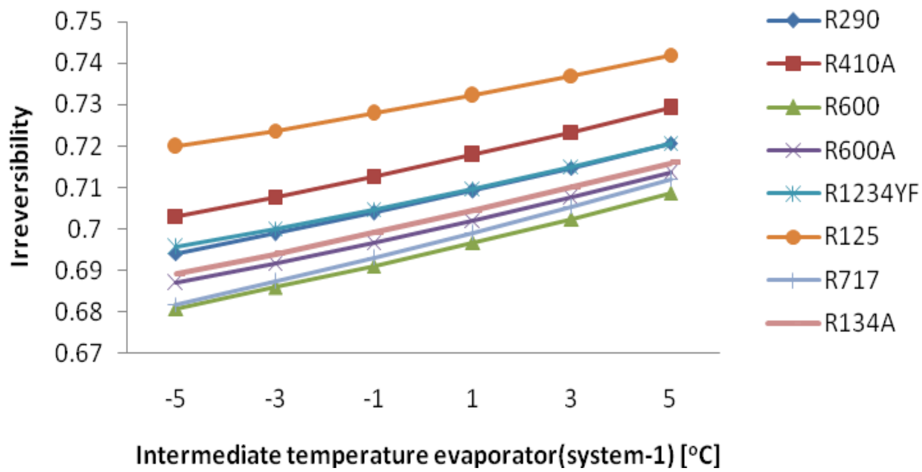

Fig: 16. Variation of intermediate temperature evaporator with irreversibility of sytem-1

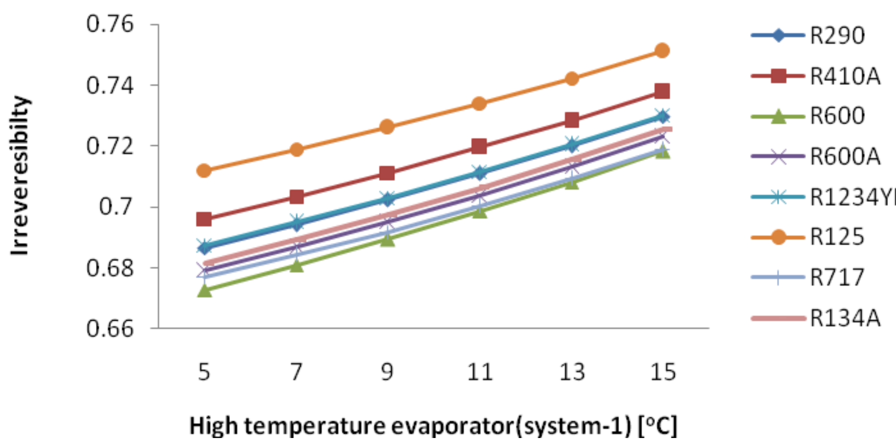

Fig: 17. Variation of high temperature evaporator with irreversibility of sytem-1

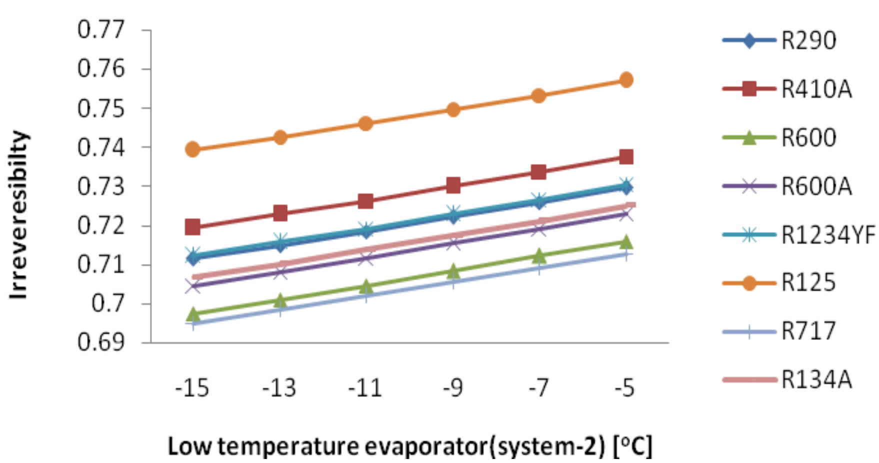

Fig: 18. Variation of low temperature evaporator with irreversibility of sytem-2

Irreversibility in system is work required to displace the atmosphere or lost work during the process. The irreversibility analysis of system-1 and system-2 is presented by Figs.15-17 and Figs.18-20 respectively. It was experienced that irreversibility of both system-1 and system-2 increase with increase in temperature of evaporator.R125 shows maximum irreversibility, on the other hand R600 and R717 show minimum irreversibility in systems compared with another selected refrigerants. It is also observed that irreversibility in system-1 is 1.4-2.1%, 1.3-2.2% and 1.6-2.0% using R600 and 1.8-3%, 1.7-3.1%, 2.2-2.7% using R125 is lower than system-2 for low, intermediate and high temperature evaporator respectively. This marginal irreversibility differences between system-1 and system-2 could be neglected.

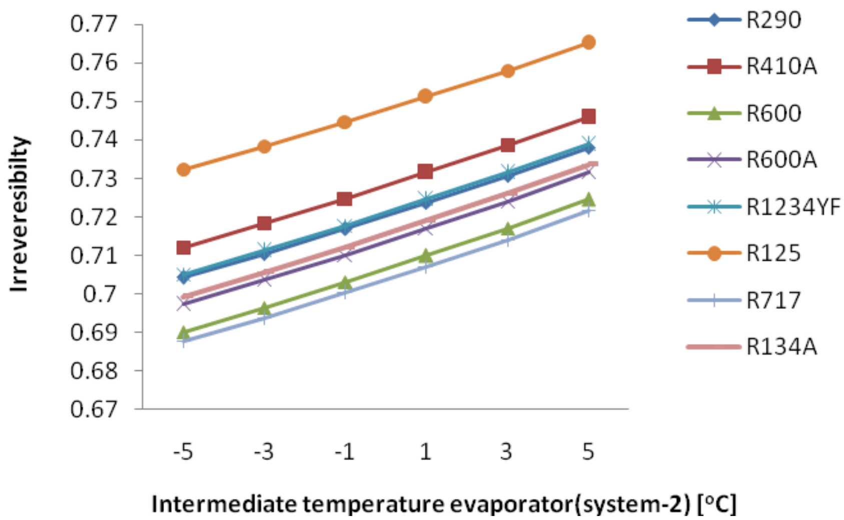

Fig: 19. Variation of intermediate temperature evaporator with irreversibility of sytem-2

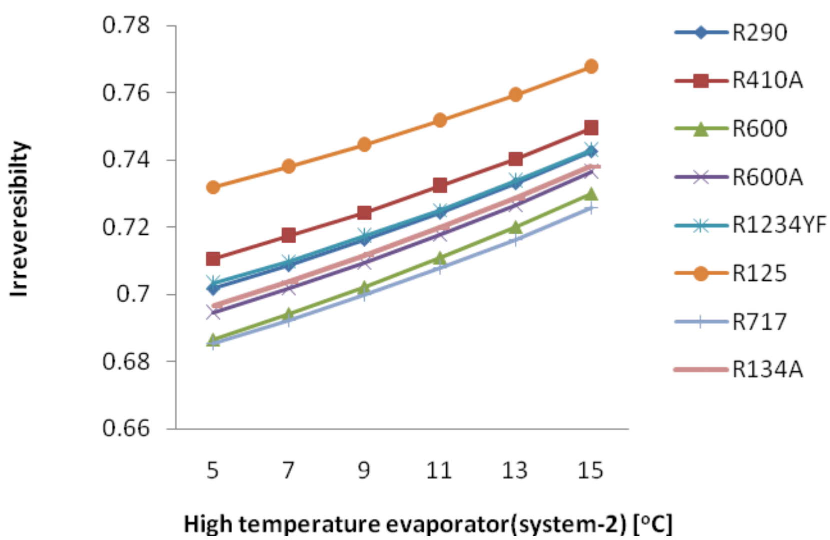

Fig: 20. Variation of high temperature evaporator with irreversibility of sytem-2

The impact of change in condenser temperature in range of 25 °C to 45 on coefficient of performance, second law efficiency and system irreversibility is shown in Figs.21-26 for sytem-1 and system-2 using ecofriendly refrigerants. This analysis reveals that COP and second law efficiency decreases with increase in condenser temperature on the other hand exergy destruction (system irreversibility) increase with increase in condenser temperature for system-1 & sytem-2

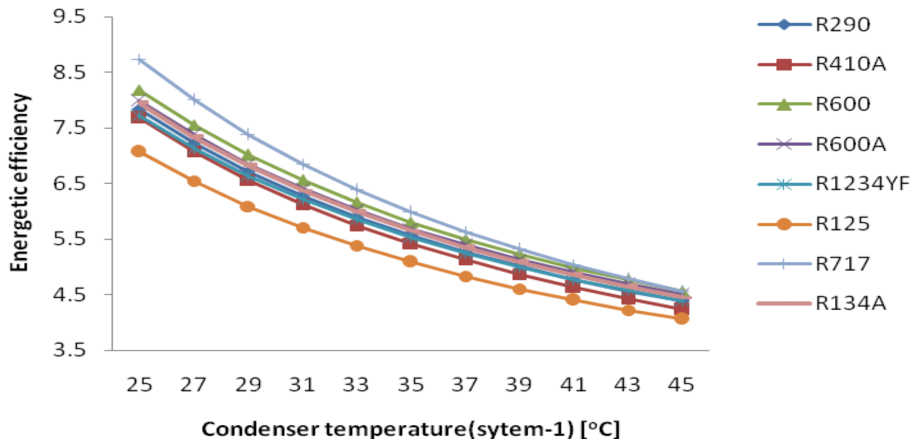

Fig: 21. Variation of condenser temperature with energetic efficiency of sytem-1

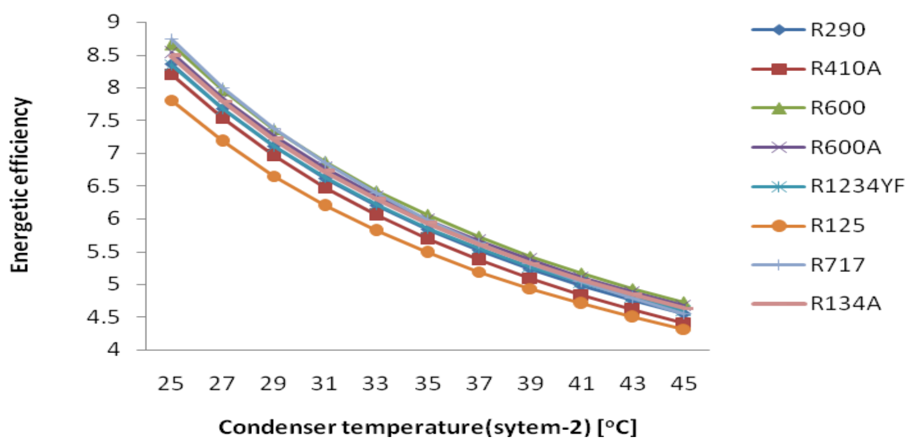

Fig: 22. Variation of condenser temperature with energetic efficiency of sytem-2

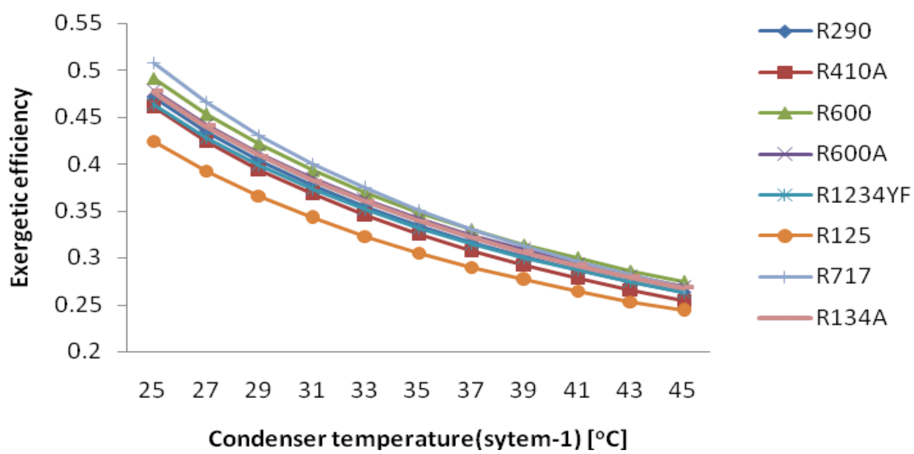

Fig: 23. Variation of condenser temperature with exergetic efficiency of sytem-1

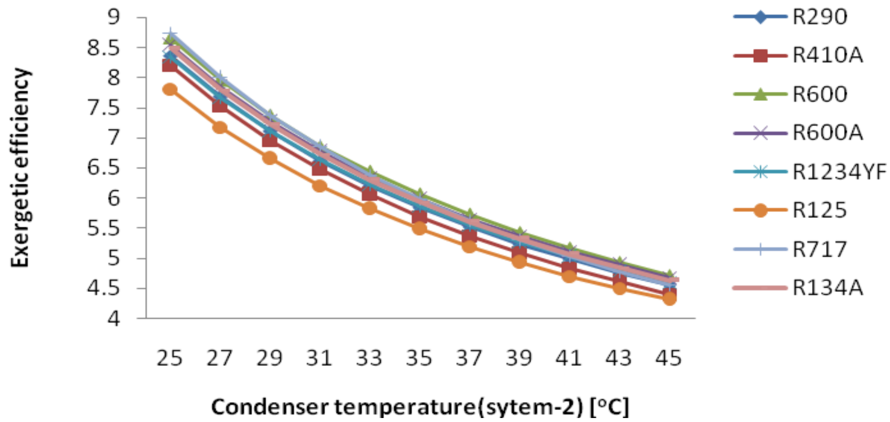

Fig: 24. Variation of condenser temperature with exergetic efficiency of sytem-2

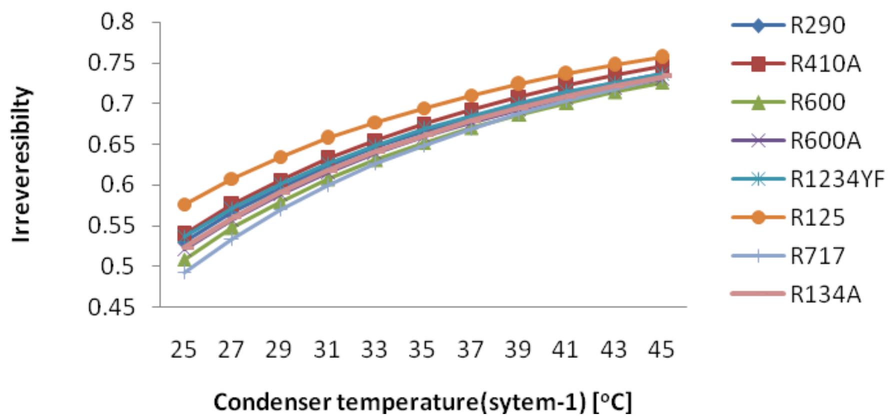

Fig: 25. Variation of condenser temperature with irreversibility of sytem-1

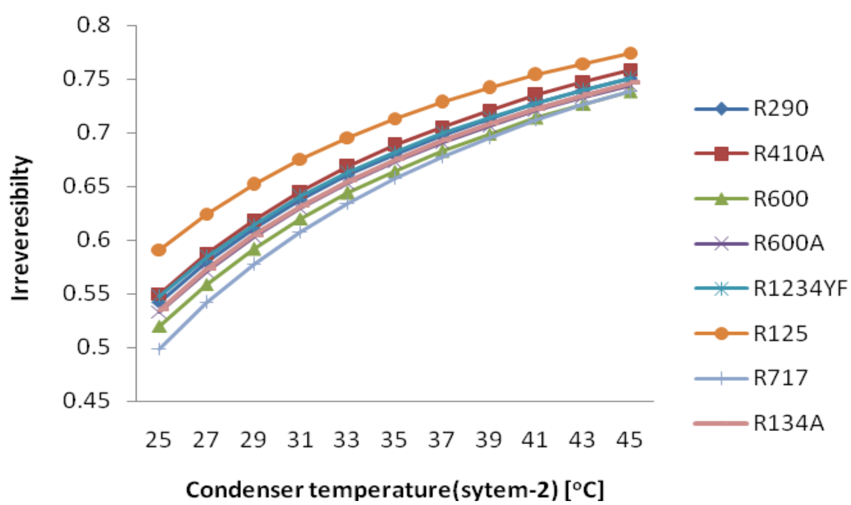

Fig: 26. Variation of condenser temperature with irreversibility of sytem-2

4. Conclusion

Thermodynamic analysis of multi-stage vapour compression refrigerator and flash intercooler with individual or multiple throttle valves has been carried out and numerical computation was done in terms of COP, second law efficiency and irreversibility destruction and following conclusions were made:

1. First law performance (Energetic) and second law performance (exergetic performance) of system-2 is higher than system-1 for selected temperature range of condenser and evaporators with chosen ecofriendly refrigerants.
2. For both systemsR125 shows minimum thermodynamic performance in terms of COP, second law efficiency and irreversibility in terms of exergy destruction in the components as well as in the both systems
3. Thermodynamic Performances in terms of COP and second law efficiency of R600and R717 better in comparison of other selected ecofriendly refrigerants for system-1 and system-2. ASR717 is toxic and limited to industrial applications, therefore R600 is recommended for both systems by taking safety precautions. Performance of R134a is slighly lesser than R600, therefore R134a can also be used for practical applications without taking of any safety precautions.
4. The maximum percentage difference of COP between system-2 and system-1 is 9.59% for R125 at 15 ^{o}C, high temperature evaporator. Irreversibility in system-1 is 1.4-2.1%, 1.3-2.2% and 1.6-2.0% using R600 and 1.8-3%, 1.7-3.1%, 2.2-2.7% using R125 is lower than system-2 for low, intermediate and high temperature evaporator respectively. This marginal irreversibility differences between system-1 and system-2 can be neglected.

References

[1] Camelia Stanciu, Adina Gheorghian, Dorin Stanciu, Alexandru Dobrovicescu, “Exergy analysis and refrigerant effect on the operation and performance limits of a one stage vapour compression refrigeration system, Termotehnia”, 36-42, 2011

[2] V. Siva Reddy, N. L. Panwar, S. C. Kaushik Exergetic analysis of a vapour compression refrigeration system with R134a, R143a, R152a, R404A, R407C, R410A, R502 and R507A,Clean Techn Environ Policy,14:47–53, 2011

[3] J. U. Ahamed , R. Saidur, H. H. Masjuki, “A review on exergy analysis of vapor compression refrigeration system, Renewable and Sustainable Energy Reviews”,15: 1593–1600

[4] D. Szargut, R. Petela, Egzergia(1965),WNT

[5] J. Szargut, D. Morris, F. Steward, “Exergy analysis of thermal, chemical and metallurgical processes”. New York: Hemisphere Publishing Corporation. 1998

[6] R. Saidur, HH Masjuki, MY Jamaluddin, “An application of energy and exergy analysis in residential sector in Malaysia”, Energy Policy, 35: 1050–63

[7] Yongmei Xuan, Guangming Chen, “Experimental study on HFC-161 mixture as an alternative refrigerant to R502”, Int J Refrigeration, Article in Press

[8] R. Cabello, E. Torrella, J. Navarro, “Esbr, Experimental evaluation of a vapour compression plant performance using R134a, R407C and R22 as working fluids”, Int J Applied Thermal Engineering.2004; 24:1905-1917

[9] Mark W. Spatz, Samuel F. Yana Motta, “An evaluation of options for replacing HCFC-22 in medium temperature refrigeration systems”, Int J Refrigeration.2004; 27:475-483

[10] X. H. Han, Q. Wang, Z. W. Zhu, G. M. Chen, “Cycle performance study on R32/R125/R161 as an alternative refrigerant to R407C”, Int J Applied Thermal Engineering.2007; 27:2559-2565

[11] R. Cabello, J. Navarro-Esbrı, R. Llopis, E. Torrella, “Analysis of the variation mechanism in the main energetic parameters in a single-stage vapour compression plant”, Int J Applied Thermal Engineering.2007; 27:167-176

[12] Akhilesh Arora, S. C. Kaushik, “Theoretical analysis of a vapour compression refrigeration system with R502, R404A and R507A”, Int J Refrigeration.2008; 31:998-1005

[13] H. M Getu, P.K Bansal, “Thermodynamic analysis of an R744-R717 cascade refrigeration system”, Int J Refrigeration.2008; 31:45-54

[14] M. Mohanraj, S. Jayaraj, C. Muraleedharan, P. Chandrasekar, “Experimental investigation of R290/R600a mixture as an alternative to R134a in a domestic refrigerator”, Int J Thermal Sciences.2009; 48:1036-1042

[15] M. Padilla, R. Revellin, J. Bonjour, “Exergy analysis of R413A as replacement of R12 in a domestic refrigeration system”, Int J Energy Conversion and Management.2010; 51:2195-2201

[16] S. A. Klein, F. Alvarado, “Engineering Equation Solver, Version 7.441. F Chart Software, Middleton, WI, 2005