Exergy Analysis of Vapour Compression Refrigeration System with Using R-407C and R-410A

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A B S T R A C T:

This paper presents a theoretical performance study of a vapour compression refrigeration system with refrigerants R-407C and R-410A. A computational model based on energy and exergy analysis is presented for the investigation of the effects of evaporating temperatures, degree of subcooling, dead state temperatures and effectiveness of the liquid vapour heat exchanger on the coefficient of performance, second law efficiency and exergy destruction ratio of the vapour compression refrigeration cycle. A theoretical investigation showed that better performances of R-407C in comparisons with R-410A.

Keywords:-Refrigeration System, Computational Model, exergetic Analysis, R-407C and R-410A

1. Introduction

Recently, ozone layer depletion (ODP) and global warming potential have become one of the most important global issues researchers are proposed to resolve this issue. The Montreal protocol (UNEP, 1997) states the phasing out of CFC's and HCFC's as refrigerants that deplete the ozone layer (ODP) [1]. The Kyoto protocol (UNFCC, 2011) encouraged promotion of plans for sustainable development and reduction of global warming potential (GWP) including the regulations of HCFC's [2].

The refrigerant hydro chlorofluorocarbon (HCFC) chlorodifluoromethane ($CHClF_2$), known from its ASHRAE classification as R-22, is a most widely used refrigerant in residential, commercial, industrial and transport systems. It was initially recognized in 1928 and commercialization in 1936. R-22 has been applied in systems ranging from the smallest window air conditioners to the largest chillers and heat pumps, including those for district cooling and heating. However, R-22 is one of a class of chemicals, hydrochloflourocarbons, being phase out, because of its high ozone depleting potential (ODP) and global warming potential (GWP). The issue of the uses of substance R-223 that deplete the ozone layer has led to a search for environmentally friendly alternatives. Some R-22 refrigerant substitutes that meet this requirement are a key process in this research.

Many investigations have been conducted in the research into substitutes for R-22. Douglas et al. [3] presents a computer model based on cost-based method for evaluates the performance of several leading R-22 replacement candidates for window air conditioners. The result showed that the two leading R-22 replacement candidate4s R-407C and R-410A had optimal costs that were nearly identical to R-22. Aprea et al. [5] presented an experimental study for the substitutions of R-22 in vapour compression plant with most widely used drop in substitute i.e. R407C. This experimental analysis was carried out with exergetic approach. The result showed that overall exergetic performance of the vapour compression plant working with R-22 is consistently better than R-407C. Aprea et al. [6] presented an experimental study to R-22 phase out in vapour compression plant with reciprocating compressor. The result showed that R-22 better than R-407C because of because of better compression process due to

Nomenclature

COP	Coefficient of performance (non-dimensional)
E_d	Exergy destruction rate (kW)
EF	Exergy rate of fuel (kW)
EP	Exergy rate of product (kW)
E_L	Thermal exergy loss rate (kW)
E_x	Exergy rate of refrigerant (kW)
W	Work rate (kW)
h	Enthalpy (kJ/kg)
EDR	Exergy destruction ratio (non-dimensional)
T	Temperature (K)
s	Entropy (kJ/kg K)
Q	Heat transfer rate (kW)
ΔP	Pressure drop (bar)
m	Mass flow rate (kg/s)

Greek Notations

η_{ex}	Exergetic Efficiency
ε	Effectiveness of the liquid vapour heat exchanger
ΔT_{sub}	Degree of subcooling of liquid refrigerant in lvhe (K)
ΔT_{sup}	Degree of superheating of vapour refrigerant in lvhe (K)

Subscripts

c	Condenser
r	Space temperature
e	Evaporator temperature
comp	Compressor
t	Refrigerant throttle valve
o	Dead state

numbers of factors. But the isentropic and volumetric efficiencies of the semi hermetic compressor are better than that of R-407C. Aprea et al. [7] investigated the performance of a vapour compression plant working both as a water chiller and as a heat pump. The refrigerants R-22 and its substitutes R-417A are used as a working fluid. The investigation has revealed that the coefficient of performance and exergetic efficiency of the plant working both as water chiller and as a heat pump, when the R-22 is used as working fluid, are higher than R-407A is used as working fluid. K. Comakli et al. [8] experimentally investigated the effects of gas mixture rat, evaporator air inlet temperature, evaporator air mass flow rate, condenser air inlet temperature, and condenser air mass flow rate on the coefficient of performance and exergetic efficiency of vapour compression heat pump systems. The investigation has shown that R-22/404A mixtures can be used in replacement for R-22 or R-407A in vapour compression heat pump systems. V.P. Venkataramanamurthy et al. [9] performed the experimental comparison of energy, exergy flow and second law efficiency of R-22 and its substitutes R-436b in vapour compression refrigeration cycles. The analysis was presents investigation of the effects of the evaporating temperatures on the exergy flow losses and second law efficiency and coefficient of performance of a vapour compression refrigeration cycle. Dalkilic et al. [10] performed the theoretical analysis of a traditional vapour compression refrigeration system with mixture based on R-134a, R-152a, R-32, R-290, R-1270, R-600 and R-600a for various ratios and their results compared with R-12, R-22 and R-134a as possible alternative replacements ranging from -30 ℃ to 10 ℃. Vincenzo et al. [11] conducted an experimental analysis for comparing the performance of a vapour compression refrigerating unit with R-22 and its substitutes R-417A, R422A and R-422D. The results revealed that R-22 was energetically more efficient than the other fluids. The refrigerants R-417A, R422A and R-422D has zero ozone depletion potential and they can easily replace the R-22 in exiting system without having to change the lubricant or renewing the refrigerating unit and its accessories. This replacement accepted for particular easy operation and at a very low cost. Aprea et al. [12] presented an experimental investigation to study the environmental impact of R-22 retrofit with R-422D and to draw possible eco-friendly scenarios. The experimental analysis presents in terms of TEWI aimed to identify the global environment impact of R-22 systems retrofitted with R-422D. the results confirmed that the system lead to an increase of TEWI when retrofitted with R422D.

The literature survey is based on the study of R-22 replacement and exergetic analysis. Most of the performance analysis of refrigeration system is investigated using an energy approach based on the first law of thermodynamics (i.e. by means of coefficient of performance). The energy analysis deals with only quantity of energy and it does not give the information that how, where and how much the

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performance of the system degrade. Thus, modern approach to process analysis required to use the exergy analysis which provides the more realistic view of the process.

In this paper a computational model has been developed to calculate the coefficient of performance, exergetic efficiency and exergy destruction ratio for R-407C and R-410A based on energy and exergy concept. It is also studies the effects of evaporating temperatures, degree of subcooling, dead state temperature and effectiveness of liquid vapour heat exchanger on coefficient of performance (COP), exergetic efficiency and exergy destruction ratio. The main characteristics of the tested refrigerants as shown in table 1.

Table 1
The main characteristics of the tested refrigerants [13]

Refrigerant	R-407C	R-410A
Molar mass	86.20	72.58
Boiling point (℃)	-43.56	-51.6
Critical temperature (℃)	86.74	72.5
Critical pressure (bar)	4.6191	4.95
Critical density ($\mathbf{kg/m^3}$)	527.30	500
Critical volume ($\mathbf{m^3/kg}$)	0.00190	0.00205
Ozone depletion potential	0	0
Global warming potential	1526	1725
Certainty Class	A1	A1

2. Cycle Description and Model

A vapour compression refrigeration system consists of five components such as evaporator, liquid-vapour heat exchanger, compressor, condenser and expansion valve. These components connected in a closed loop through piping that has heat transfer with the surrounding as shown in fig.1. At state 11, refrigerant leaves the evaporator at a low pressure, low temperature, saturated vapour and enter the liquid vapour heat exchanger where it absorbs the heat from high pressure- temperature refrigerant flows from condenser. The refrigerant from the liquid-vapour heat exchanger enter into compressor through the suction line in which both temperature and pressure increased at state 1. This process can be shown in fig.2. At state 2, it leaves the compressor as a high pressure, high temperature, superheated vapour and enter the condenser where it reject heat to surrounding medium at constant pressure after undergoing heat transfer in the discharge line. Refrigerant leaves the condenser at state 3, as high pressure, medium temperature, saturated liquid and enters the liquid-vapour heat exchanger at state 33. The expansion valve allows to flowing the high pressure liquid at constant enthalpy from high pressure to low pressure. At state 4, it leaves the expansion valve as a low temperature, low pressure, and liquid-vapour mixture and enters the evaporator where it absorbs the heat at constant pressure, changed into saturated vapour and cycle is completed.

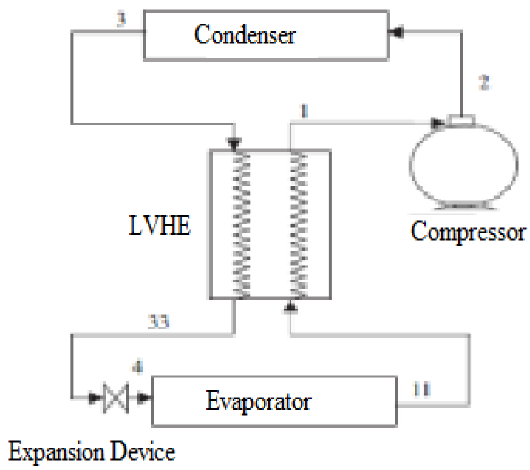

Fig. (1) Schematic of typical vapour compression refrigeration system with a liquid –vapour heat exchanger

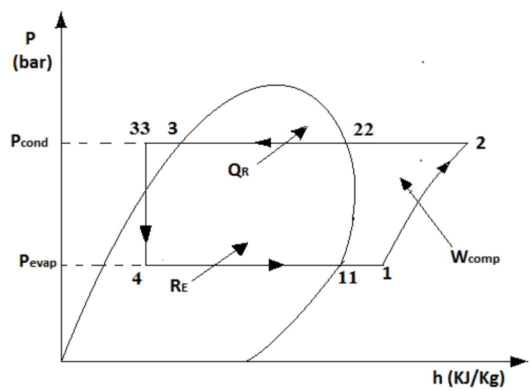

Fig. (2) Pressure-Enthalpy Diagram showing effect of an idealized liquid-vapour heat exchanger

For the analysis following assumption are made:

1. Degree of subcooling of liquid refrigerant in lvhe(ΔT_{sub}) = 5°C.
2. Isentropic efficiency of compressor$\left(\eta_{comp}\right) = 75\%$.
3. Mechanical efficiency of compressor$(\eta_{mec\,h}) = 75\%$.
4. Electrical efficiency of compressor (η_{el})=75%.
5. Difference between evaporator and space temperature $(T_r - T_e) = 20$°C.
6. Effectiveness of liquid vapour heat exchanger(ε)=0.8.
7. Evaporator temperature T_{evap} $(in$ °C) ranging from= -50°C to 0°C.
8. Condenser temperature T_{cond} $(in$ °C) =40°C.
9. Mass flow rate of refrigerant$(m_r) = 1\,kg/s$
10. Surrounding temperature (T_0)=30°C.
11. Pressure losses in pipelines are neglected.
12. Steady state operations are considered in all components.

The energy analysis based on first law of thermodynamic, the performance of vapour compression refrigeration system can be predicted in terms of coefficient of performance (COP), which is defined as the ratio of net refrigerating effect produced by the refrigerator to the work done by the compressor. It is expressed as

$$COP = \frac{Q_e}{W}$$

$$COP = \frac{h_1 - h_4}{h_2 - h_2} \tag{1}$$

The modern approach based on second law of thermodynamic i.e. exergy analysis can be used to measures the performance of the vapour compression refrigeration system. This analysis derives the concept of exergy, which is always decreasing due to thermodynamic irreversibilities. Exergy is the maximum useful work that could be obtained from the system at a given state in a specified environment. Exergy balance for a control volume undergoing steady state process is expressed as

$$E_{d_i} = \sum(me_x)_{in} - \sum(me_x)_{out} + [\sum(Q(1 - T_0/T)_{in} - \sum(Q(1 - T_0/T)_{out}] \pm \sum W \tag{2}$$

Exergy Destruction (ED) in the system components

Exergy destruction in each component of the cycle is calculated as

A. Exergy destruction in Evaporator

$$E_{d_e} = E_{X_4} + Q_e\left(1 - \frac{T_0}{T_r}\right) - E_{X_{11}}$$
$$= m_r(h_4 - T_oS_4) + Q_e\left(1 - \frac{T_0}{T_r}\right) - m_r(h_{11} - T_oS_{11}) \tag{3}$$

B. Exergy destruction in Compressor

$$E_{d\,comp} = E_{X_1} + W - E_{X_2}$$
$$= m_r(h_1 - T_oS_1) + \frac{W}{\eta_{mec\,h} * \eta_{el}} - m_r(h_2 - T_oS_2) \tag{4}$$

C. Exergy destruction in Condenser

$$E_{d\,c} = E_{X_2} - E_{X_3}$$
$$= m_r(h_2 - T_oS_2) - m_r(h_3 - T_oS_3) \tag{5}$$

D. Exergy destruction in Throttle valve

$$E_{d\,t} = E_{X_{33}} - E_{X_4}$$
$$= m_r(h_{33} - T_oS_{33}) - m_r(h_4 - T_oS_4) \tag{6}$$

E. Exergy destruction in liquid vapour heat exchanger

$$E_{d\,lvhe} = E_{X_3} - E_{X_{33}} + E_{X_{11}} - E_{X_1}$$
$$= m_r(h_3 - h_{33} + h_{11} - h_1) - T_0(S_3 - S_{33} + S_{11} - S_1) \tag{7}$$

Total Exergy destruction

Total exergy destruction in the system is the sum of the exergy destruction in different components of the system and is given by

$$\sum E_{d_i} = E_{d_e} + E_{d_{comp}} + E_{d_c} + E_{d_t} + E_{d\,lvhe} \tag{8}$$

Thermal Exergy Loss

Thermal exergy loss rate in a component is given by

$$E_{L_i} = Q_i\left(1 - \frac{T_0}{T_i}\right) \tag{9}$$

The term thermal exergy loss rate is related to external irreversibility which takes place because of temperature difference between the control volume and the immediate surroundings. It depends upon how the boundary of the system is selected. If the system includes the immediate surroundings then the boundary of the system is at the same temperature as the temperature of the immediate surroundings, hence the value of the thermal exergy loss become zero. If the system boundary does not include the immediate surroundings, the temperature difference between the system boundary and immediate surroundings exits. In vapour compression refrigeration system condenser is the component where heat is rejected. Then the equation (9) becomes:

$$E_{L_c} = Q_c\left(1 - \frac{T_0}{T_c}\right) \tag{10}$$

When consider the thermal exergy loss total exergy destruction in the system is given as:

$$\sum E_{d_i} + \sum E_{L_i} = E_{d_e} + E_{d_{comp}} + E_{d_c} + E_{d_t} + E_{d_{lvhe}} - E_{L_c} \tag{11}$$

Now, total exergy supplied is given by:

$$EF = EP + \sum E_{d_i} + \sum E_{L_i} \tag{12}$$

For refrigeration system, product is the exergy of the heat abstracted into the evaporator from the space to be cooled at temperature T_r.

$$EP = Q_e\left|1 - \frac{T_0}{T_r}\right|$$

Exergetic Efficiency (η_{ex})

$$\eta_{ex} = 1 - \frac{Total\ Exergy\ Destruction}{Total\ Exergy\ Supplied}$$

$$\eta_{ex} = 1 - \frac{\sum E_{d_i} + \sum E_{L_i}}{EF} \tag{13}$$

Exergy Destruction Ratio (EDR)

Exergy destruction ratio is the ratio of the total exergy destruction in the system to the exergy in the product and it is given by

$$EDR = \frac{ED_{total}}{EP}$$

Also, in terms of second law efficiency

$$EDR = \frac{1}{\eta_{ex}} - 1 \tag{14}$$

3. Result and discussion

Fig. 3 shows the effects of evaporator temperatures on coefficient of performance. The pressure ratio across the compressor decreases, with increase in evaporator temperature causing work required by the compressor decrease and cooling capacity increases due to increase in refrigerating effect. Hence, the combined effects of these two factors increase the coefficient of performance (COP). R-407C shows better COP than R-410A at condenser temperature 40℃ and evaporator temperature ranging from$-50℃$ to $0℃$. The maximum difference measured between R-407C and R-410A is 10.29% higher end of evaporator temperatures. Fig 4-5 shows the effect of evaporator temperatures on exergetic efficiency (η_{ex}) and exergy destruction ratio (EDR). With increase in evaporator temperatures exergetic efficiency increases till the optimum evaporator temperature and beyond the optimum temperature it decrease. The optimum evaporator is the temperature at which maximum exergetic efficiency is achieved. The curves trend for EDR almost reverses to curves of exergetic efficiency. The increasing and decreasing of exergetic efficiency depends upon the two factors, first factor is the exergy of cooling effects i.e. $Q_e\left|1-\frac{T_0}{T_r}\right|$. With increase in evaporator temperatures, Q_e increases while the term $\left|1-\frac{T_0}{T_r}\right|$ reduces

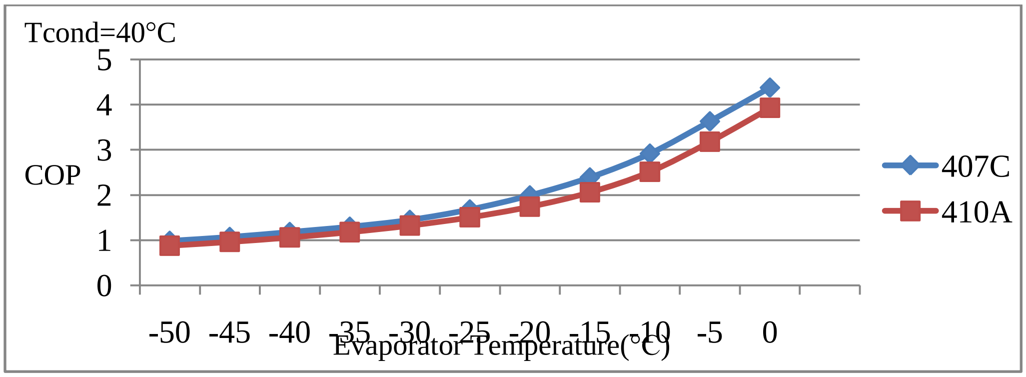

Fig. 3.Effect of evaporating temperatures on coefficient of performance

Second factor is compressor work required by compressor W which decreases with increase in evaporator temperature. The term Q_e and W have positive effect on increase of exergetic efficiency while the term$\left|1-\frac{T_0}{T_r}\right|$ have negative effect on increase of exergetic efficiency. The combined effect of these two factors, increases exergetic efficiency increases till the optimum evaporator temperature and beyond the optimum temperature decrease. The curves trend for EDR almost reverses to curves of exergetic efficiency because of exergetic efficiency is inversely proportional to exergy destruction ratio (EDR). The EDR decreases with increases in evaporator temperatures till the optimum evaporator temperature and beyond the optimum temperature it increase. The optimum evaporator is the temperature at which minimum EDR is achieved. The exergetic efficiency of R-407C is 14-19% is higher than R-410A. Thus, of R-407C is better than R-410A.

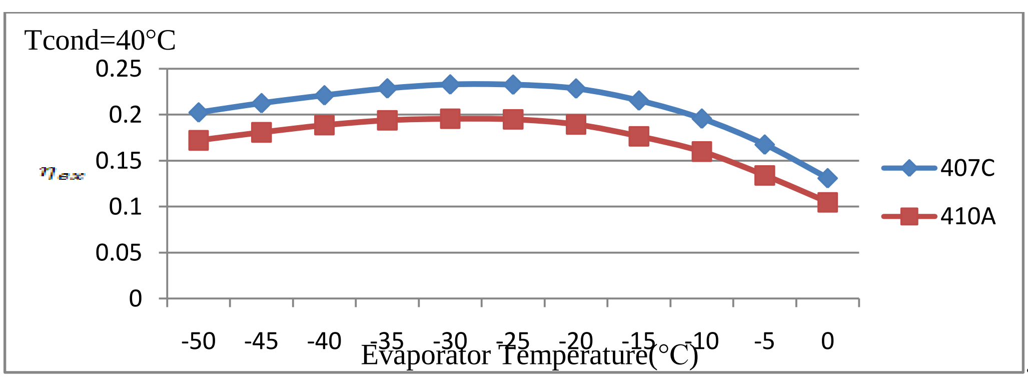

Fig. 4.Effect of evaporating temperatures on exergetic efficiency

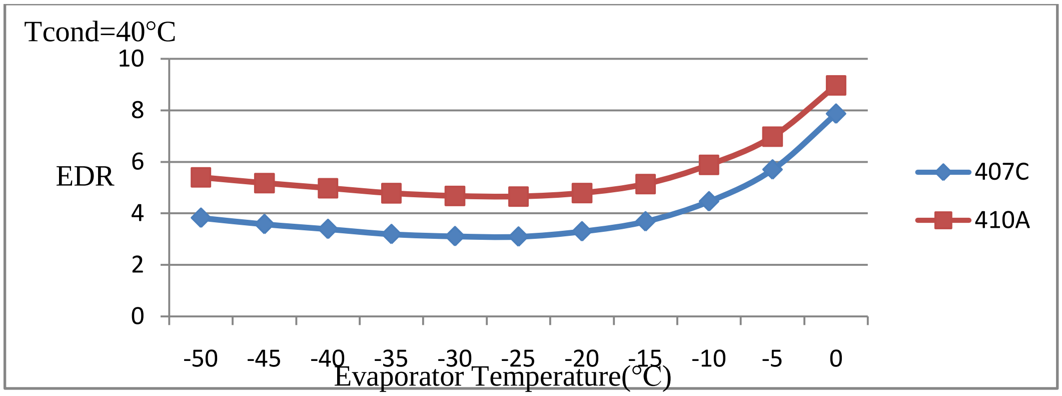

Fig. 5.Effect of evaporating temperatures on exergy destruction ratio (EDR)

Fig. 6-8 presents the effect of degree of subcooling on coefficient of performance (COP), exergetic efficiency and exergy destruction ratio (EDR). With increase in degree of subcooling cooling capacity increase because of increase in refrigerating effect and there is no change in compressor work, hence COP increases. From the study of equation (13)-(14) it is evident that increase in COP increases the exergetic efficiency and decreases exergy destruction ratio (EDR). The rate of increase in COP is

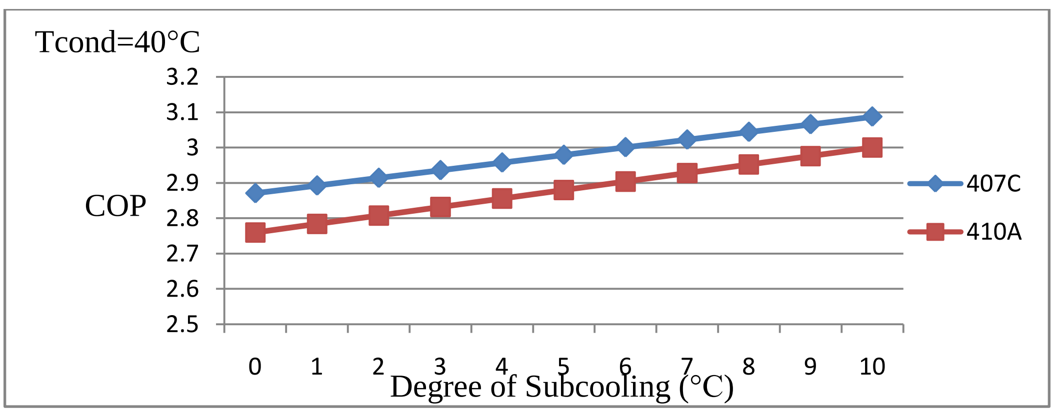

Fig. 6.Effect of degree of subcooling on coefficient of performance

approximately 0.75%/°Cof subcooling in case of R-407C. The rate increase in COP is approximately 0.81%/°Cof subcooling in case of R-410A. The total increase in exergetic efficiency for R-407C is 7.02% for 10°C subcooling and for R-410A is 8.01% for 10°C subcooling.

Fig. 9-10 presents the effects of dead state temperature on exergetic efficiency and EDR. With increase in dead state temperature refrigerating effect and compressor work remains constant; hence COP remains constant while the term $\left|1-\frac{T_0}{T_r}\right|$ increases, hence exergetic efficiency increase and EDR decrease. The curves trends of both R-407C and R-410A are identical and their curves for both exergetic efficiency and EDR are nearly overlapping. The exergetic efficiency of R-407C is 2.5-5.1% higher than R410A for considered range of dead state temperatures.

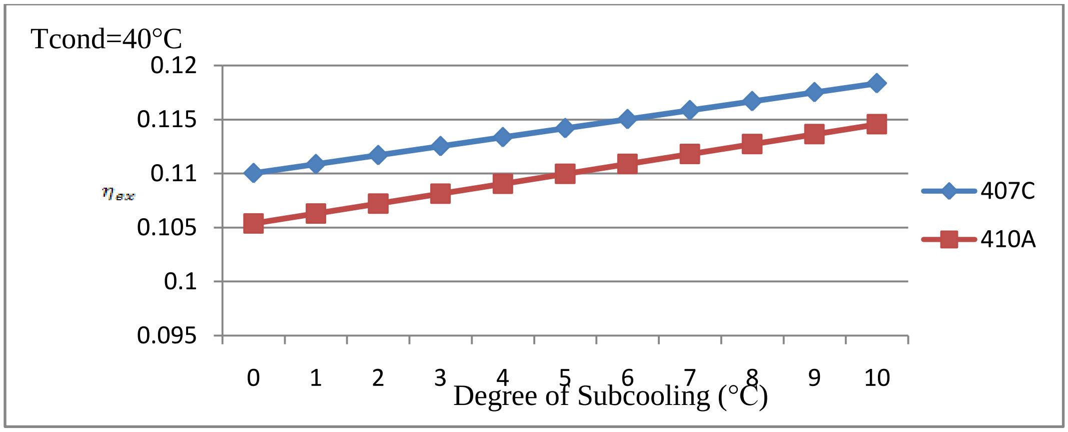

Fig. 7.Effect of degree of subcooling on exergetic efficiency

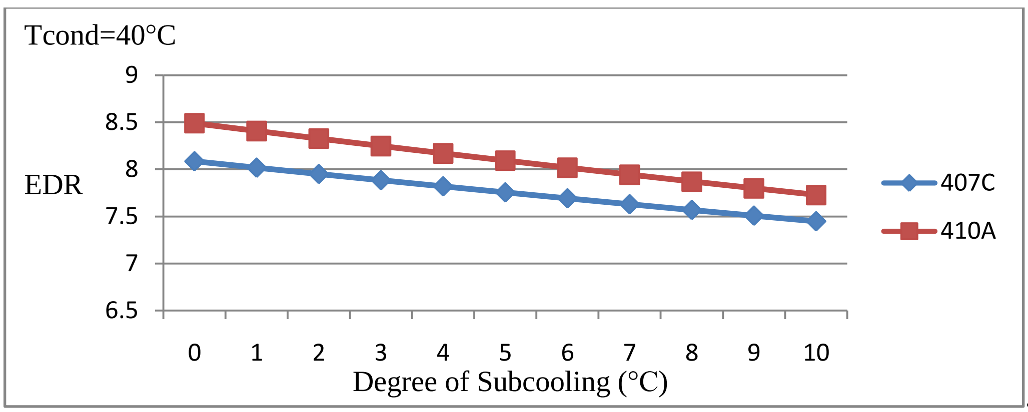

Fig. 8.Effect of degree of subcooling on exergy destruction ratio (EDR)

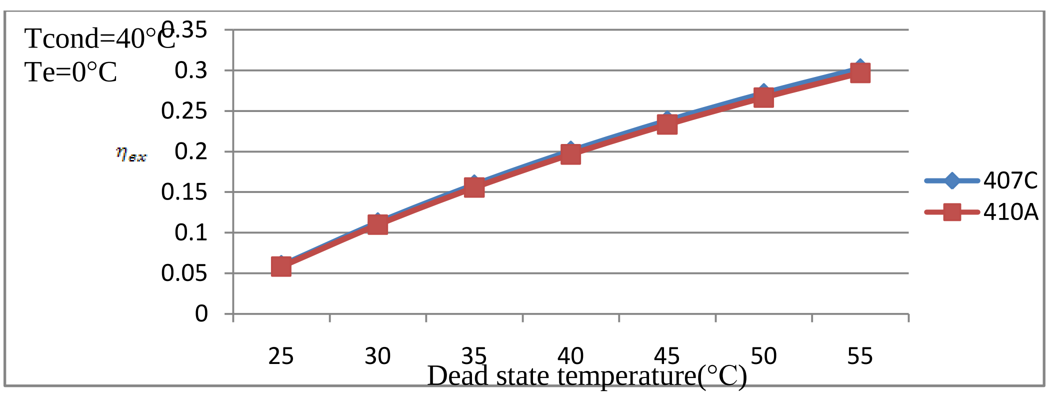

Fig. 9.Effect of dead state temperatures on exergetic efficiency

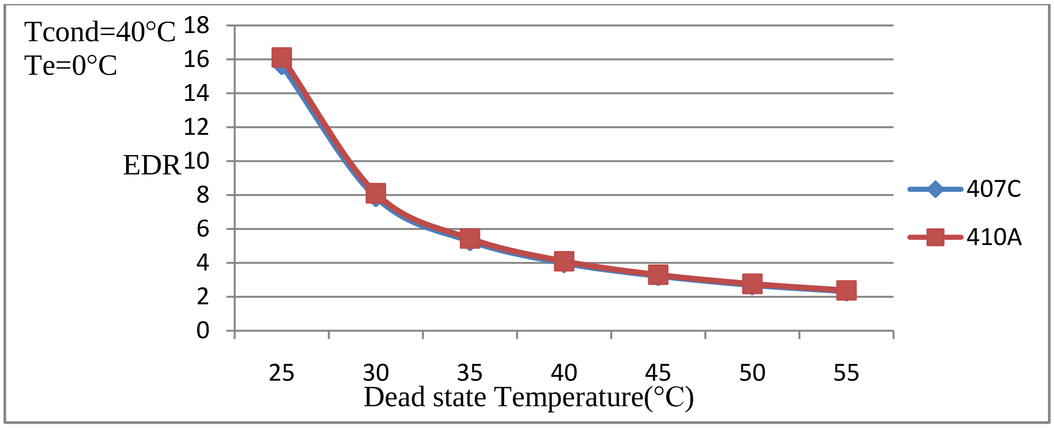

Fig. 10.Effect of dead state temperatures on exergy destruction ratio (EDR)

Fig. 11-13 shows the effect of effectiveness of liquid-vapour heat exchanger on coefficient of performance (COP), exergetic efficiency and exergy destruction ratio (EDR). With increase in effectiveness of liquid-vapour heat exchanger COP and exergetic efficiency decreases while EDR increases. The total COP decrease by15.50%, exergetic efficiency decrease by 9.8% and EDR increases by 18.56% for R-407C. The total COP decrease by22.82%, exergetic efficiency decrease by 14.5% and EDR increases by 24.61% for R-410A.

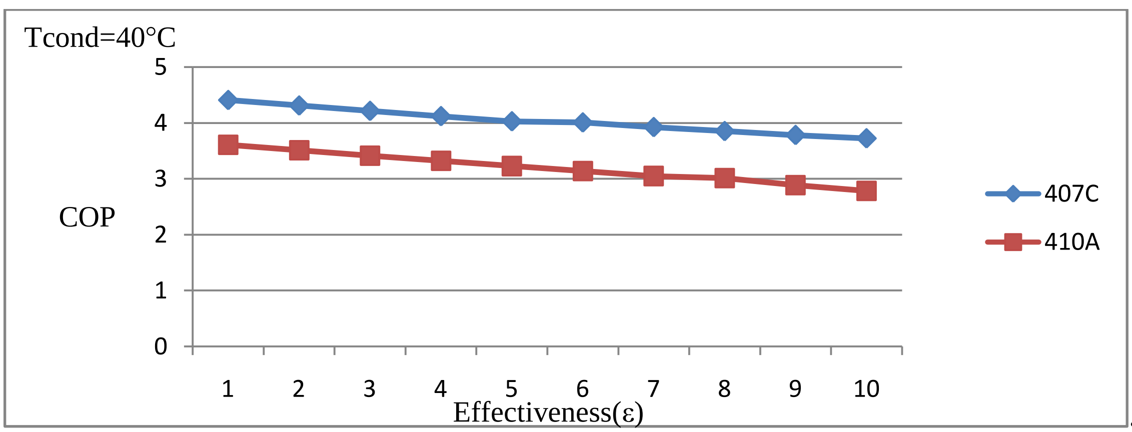

Fig. 11.Effect of effectiveness of liquid-vapour heat exchanger on coefficient of performance

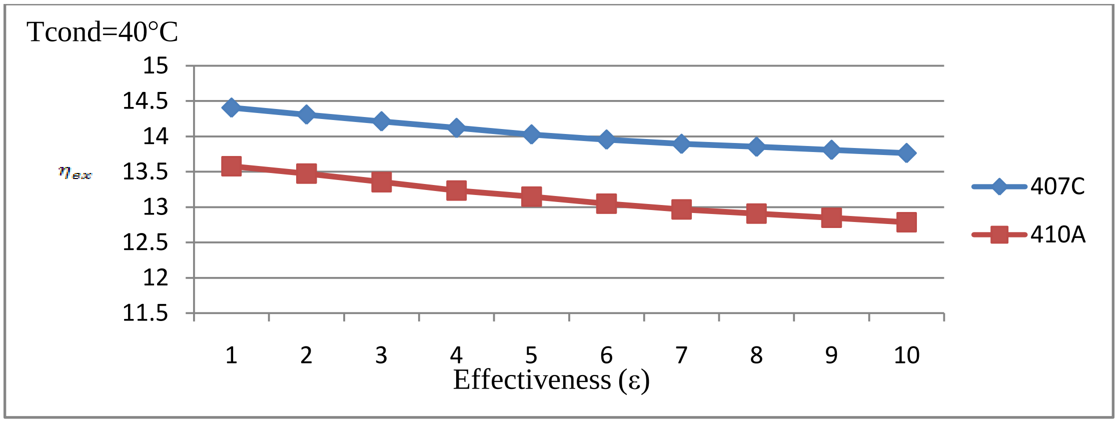

Fig. 12.Effect of effectiveness of liquid-vapour heat exchanger on exergetic efficiency

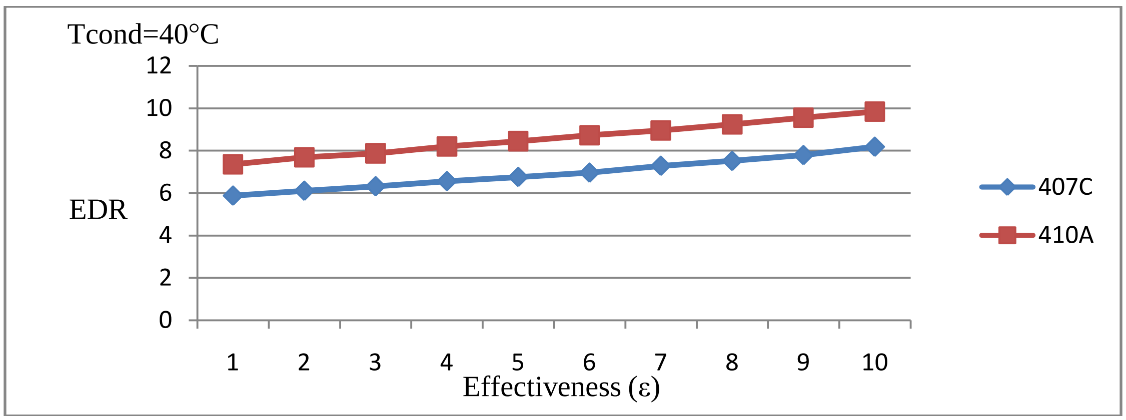

Fig. 13.Effect of effectiveness of liquid-vapour heat exchanger on exergy destruction ratio (EDR)

4. Conclusion

A computational model based on energy and exergy analysis is presented for the investigation of the effects of evaporating temperatures, degree of subcooling, dead state temperatures and effectiveness of the liquid vapour heat exchanger on the COP, second law efficiency and EDR of the vapour compression refrigeration cycle for R-407C and R-410A. The conclusions present in this analysis are given as follows.

1. The COP and exergetic efficiency of R-407C are better than that of R-410A. The EDR of R-410A are higher than that of R-407C. This analysis performed at condenser temperature 40°C and evaporator temperature ranging from $-50°C\ to\ 0°C$.
2. For both refrigerants i.e. R-407C and R-410A, COP and exergy efficiency improve by subcooling of high pressure condensed liquid refrigerant. The total increase in exergetic efficiency for R-407C is 7.02% for 10℃ subcooling and for R-410A is 8.01% for 10℃ subcooling.
3. With increase in dead state temperatures exergetic efficiency increases and EDR reduces while coefficient of performance remains constant. The curves trends of both R-407C and R-410A are identical and their curves for both exergetic efficiency and EDR are nearly overlapping. The

exergetic efficiency of R-407C is 2.5-5.1% higher than R410A for considered range of dead state temperatures.

4. With increase in effectiveness of liquid-vapour heat exchanger COP and exergetic efficiency decreases while EDR increases. The total COP decrease by15.50%, exergetic efficiency decrease by 9.8% and EDR increases by 18.56% for R-407C. The total COP decrease by22.82%, exergetic efficiency decrease by 14.5% and EDR increases by 24.61% for R-410A.

5. References

1. United Nation Environment Programme (UNEPS). The Montreal protocol on substances that deplete the ozone layer 1997.
2. United Nations 2011 Kyoto protocol to the United Nations Framework Convention on Climate Change. http://untccc.int//resource/docs/convkp/kpeng.pdf.
3. J.D. Douglas, J.E. Braun, E.A. Groll, D.R. Tree, 1999. A cost method comparing alternative refrigerant applied to R-22 system. International Journal of Refrigeration 22,107-125.
4. S.A. Klein, D. T. Reindl, K. Brownell, 2000. Refrigeration System Performance using Liquid-Suction Heat Exchanger. International Journal of Refrigeration Vol.23, Part 8, pp. 588-596.
5. C. Aprea, A. Greco, 2002. An exergetic analysis of R22 substitution. Applied Thermal Engineering 22, 1455-1469.
6. C. Aprea, A. Greco, 2003. Performance evaluation of R22 and R407C in a vapour compression plant with reciprocating compressor. Applies Thermal Engineering 23, 215-227.
7. C. Aprea, R. Mastrullo, C. Renno, 2004. An analysis of the performance of a vapour compression refrigeration plant working both as a water chiller and a heat pump using R-22 and 417A. Applied Thermal Engineering 24, 487-499.
8. K. Comakli, F. simsek, O. Comakli, B.Sahin, 2009. Determination of optimum conditions R-22 and R404a refrigerant Mixtures in heat pumps using Taguchi method. Applied Energy 86, 2451-2458.
9. V.P. Venkataramanamurthy, Dr. P. Senthil Kumar, 2010. Experimental Comparative energy, exergy flow and second law efficiency analysis of R22, R436b vapour compression refrigeration cycles. International Journal of science and Technology 2(5), 1399-1412.
10. A.S. Dalkilic, S. Wongwises, 2010.A performance comparison of vapour compression refrigeration system using various alternative refrigerants. International Communications in Heat and Mass Transfer 37, 1340-1349.
11. Vincenzo La Rocca, Giuseppe Panno, 2011. Experimental performance evaluation of a vapour compression refrigerating plant when replacing R22 with alternative refrigerants. Applied energy 88, 2809-2815.
12. Ciro Aprea, Angelo maiorino, 2011. An experimental investigation of the global environmental impact of the R22 retrofit with R422D. Energy 36, 1161-1170.
13. M. Mohanraj, S. Jayaraj, C. Muraleedharan, 2009. Environment friendly alternatives to halogenated refrigerants-A review. International Journal of Greenhouse Gas control 3, 108-119.