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Comparative Exergy Analysis of Vapor Compression Refrigeration System Using Alternative Refrigerants

Gulshan Sachdeva, Vaibhav Jain

***Abstract*—**In present paper, the performance of various alternative refrigerants is compared to find the substitute of R22, the widely used hydrochlorofluorocarbon refrigerant in developing countries. These include the environmentally friendly hydrofluorocarbon (HFC) refrigerants such as R134A, R410A, R407C and M20. In the present study, a steady state thermodynamic model (includes both first and second law analysis) which simulates the working of an actual vapor-compression system is developed. The model predicts the performance of system with alternative refrigerants. Considering the recent trends of replacement of ozone depleting refrigerants and improvement in system efficiency, R407C is found to be potential candidate to replace R22 refrigerant in the present study.

***Keywords*—**Refrigeration, compression system, performance study, modeling, R407C.

I. Introduction

REFRIGERATION, cooling, and heating processes are essential in a variety of everyday situations e.g. air-conditioning and heating of buildings, hospitals, operation theatres, restaurants, hotels, automobiles, and transportation. Evaporation and condensing processes in refrigeration systems are as a result of the heat transfer occurring due to phase change in refrigerants. Therefore, the design and performance of any cooling system depends on the thermodynamic properties of the refrigerant used. For many years, CFCs and HCFCs have been used successfully as refrigerants, blowing agents, cleaning solvents, and aerosol propellants. CFCs seem to be an ideal choice due to their unique combination of properties. However, after the discovery of the harmful effects of CFC based refrigerants on the ozone layer, search to find new alternative refrigerants as working fluids gained momentum in the recent years. By international agreement (Montreal Protocol), signed in 1987 and later amended several times, these were scheduled to be phased out by 1st January 1996, in the developed countries and by the year 2000 in the developing countries. Calm [1] reviewed the progression of refrigerants, from early uses to the present, and also addressed future directions and substitutes. According to this study, the history of refrigerants can be classified into four generations based on defining selection criteria. It discusses the displacement of working fluids, with successive criteria, and remerges interest in old refrigerants, now known as natural refrigerants. This study further examines the outlook for current options in the contexts of existing international agreements, including the Montreal and Kyoto Protocols to avert stratospheric ozone depletion and global climate change, respectively.

Finding drop-in replacements for CFC based working fluids is important due to two main reasons: Firstly, the harmful effects on the ozone layer and worldwide concern over global warming. Secondly, the stringent need for improvement in the system efficiency to conserve resources. Due to the reasons listed above, the researchers prompted with the alternatives, which can be used instead of CFCs. In finding the alternatives to the CFC based cooling refrigerants often, mixtures of binary, ternary are suggested. Mixing two or more refrigerants gives us a chance to obtain the desired thermodynamic properties (i.e. often closing to CFC based ones for current systems) of the refrigerants by changing the mixture ratios.

A theoretical development of the thermodynamic properties of two mixtures of hydrofluoro-carbon (HFC) refrigerants, i.e. R407C and R410A (in the superheated vapor state), was carried out by Monte [2], [3]. Arora et al. [4] did the theoretical analysis of a vapor compression refrigeration system with R502, R404A and R507A. Their work presents a detailed exergy analysis of an actual vapor compression refrigeration (VCR) cycle. The efficiency effect in condenser was highest, and lowest in liquid vapor heat exchanger for the refrigerants considered. Wang et al. [5] investigated the potential benefits and performance improving options of compressor cooling. Selbas et al. [6] performed the exergy based thermoeconomic optimization of subcooled and superheated vapor compression refrigeration cycle for three refrigerants: R22, R134a, and R407C. Thermodynamic properties of refrigerants were formulated using the Artificial Neural Network methodology. Kiatsiiroat and Thalang [7] proposed a blend of R22/R124/R152 as an alternative and easy retrofit for R12. Arcaklioglu et al. [8] developed an algorithm to find refrigerant mixtures of equal volumetric cooling capacity when compared to CFC based refrigerants in vapor compression refrigeration systems. Han et al. [9] presented the new ternary non-azeotropic mixture of R32/R125/R161 as an alternative refrigerant to R407C. A new refrigeration cycle (NRC) using the binary non-azeotrpic refrigerant mixture R32/R134a was developed by Chen and Yu [10] which can be an alternative refrigeration cycle for the residential air conditioner. Wu et al. [11] reported a ternary blend

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R152a/R125/R32 with a mass ratio of 48/18/34 as a potential alternative to R22. The development of refrigeration system model which simulates the actual working of a reciprocating chiller has been the goal of many researchers. Winkler et al. [12] did the comprehensive investigation of numerical methods in simulating a steady-state vapor compression system. The purpose of his work was to describe and investigate the robustness and efficiency of three unique algorithms used to simulate a modular/component-based vapor compression system. Cabello et al. [13] made a simplified steady-state modelling of a single stage vapor compression plant. In this work a simplified steady-state model to predict the energy performance of a single stage vapor compression plant was proposed. This model has been validated through experimental data obtained from a test bench using three working fluids (R134A, R407C and R22). Ecir et al. [14] used ten different modeling techniques within data mining process for the prediction of thermophysical properties of refrigerants (R134A, R404A, R407C and R410A). Relations depending on temperature and pressure were carried out for the determination of thermophysical properties of the refrigerants. Khan and Zubair [15] evaluated the performance of vapor compression system by developing a finite- time thermodynamic model. The model can be used to study the performance of a variable-speed refrigeration system in which the evaporator capacity is varied by changing the mass-flow rate of the refrigerant, while keeping the inlet chilled-water temperature as constant. The model can also be used for predicting an optimum distribution of heat-exchanger areas between the evaporator and condenser for a given total heat exchanger area. Lal et al. [16] give experimental investigation on the performance of a window air-conditioner operated with R22 and M20 refrigerant mixture tested at different refrigerant charge levels. From literature review, many potential refrigerants are found to replace R22. These include the environmentally-friendly HFC refrigerants R134A, R410A and R407C and M20. Table I shows the physical and environmental characteristics of these refrigerants.

TABLE I
PHYSICAL AND ENVIRONMENTAL CHARACTERISTICS OF SELECTED REFRIGERANTS [10]

Properties	R22	R134A	R410A	R407C	M20
Molecular Weight (kg/Kmol)	86.47	102	72.58	86.20	76.665
B.P. at 1.013 bar [°C]	-40.8	-26.1	-51.4	-43.6	-51.15
Critical temperature [°C]	96.1	101.1	70.5	85.8	84.727
Critical pressure [kPa]	4990	4060	4810	4600	4834.9
ODP	0.050	0	0	0	0
GWP_{100}	1810	1300	2100	1800	1292
Temperature glide at NBP (^0C)	0	0	0.08	7.0	0

In the present research study, a refrigerant property dependent thermodynamic model [15] of a simple variable speed reciprocating system, which can simulate the performance of actual system as closely as possible, has been used to compare the performance of alternative refrigerants using energy and exergy analysis.

II. THERMODYNAMIC MODEL OF VCRS

Considering the steady-state cyclic operation of the system shown in Figs. 1 and 2, refrigerant vapor enters the compressor at state 4 and saturated liquid exits the condenser at state 1. The refrigerant then flows through the expansion valve to the evaporator. Referring to Fig. 1, using the first law of thermodynamics and the fact that change in internal energy is zero for a cyclic process, we get:

$$Q_{cond} + Q_{loss,cond} - (Q_{evap} + Q_{loss,evap}) - (W - Q_{loss,W}) = 0 \quad (1)$$

Heat transfer to and from the cycle occurs by convection to flowing fluid streams with finite mass flow rates and specific heats. Therefore, the heat-transfer rate to the cycle in the evaporator becomes:

$$Q_{evap} = (\varepsilon C)_{evap} (T_{in,evap} - T_{evap}) = m_{ref} (h_3 - h_2) \quad (2)$$

Similarly, the heat-transfer rate between the refrigeration cycle and the sink in the condenser is:

$$Q_{cond} = (\varepsilon C)_{cond} (T_{cond} - T_{in,cond}) = m_{ref} (h_6 - h_1) \quad (3)$$

The power required by the compressor, described in terms of an isentropic efficiency, is given by:

$$W = m_{ref} (h_5 - h_4) \quad (4)$$

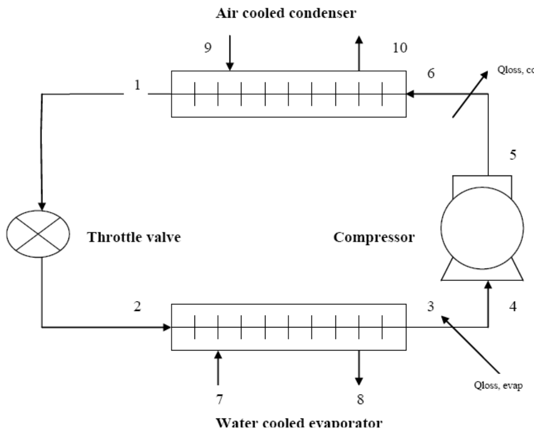

Fig. 1 Schematic diagram of a cascade system

We assume that the heat leaking into the suction line is:

$$Q_{loss,evap} = m_{ref} (h_4 - h_3) \quad (5)$$

Similarly, the heat leakage from the discharge can be expressed as:

$$Q_{loss,cond} + Q_{loss,W} = m_{ref} (h_5 - h_6) \quad (6)$$

The COP is defined as the refrigerating effect over the net work input, i.e.

$$COP = Q_{evap} / W \quad (7)$$

$$\text{Refrigeration efficiency} = COP/(COP)_{carnot} \quad (8)$$

The exit temperature of external fluid at evaporator can be found from:

$$Q_{evap} = C_{evap}\,(T_{in,\,evap} - T_{out,\,evap}) \quad (9a)$$

The exit temperature of external fluid at condenser can be found from:

$$Q_{cond} = C_{cond}\,(T_{out,\,cond} - T_{in,\,cond}) \quad (9b)$$

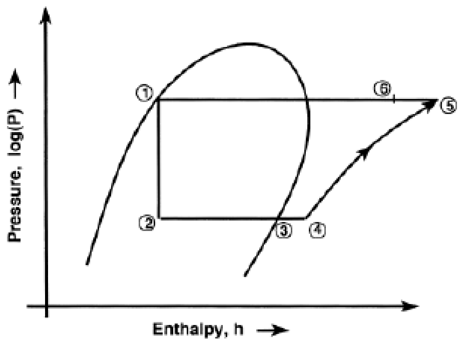

Fig. 2 Pressure-Enthalpy diagram for vapor compression cycle

Exergy analysis is a powerful tool in the design, optimization, and performance evaluation of energy systems. The principles and methodologies of exergy analysis are well established. An exergy analysis is usually aimed to determine the maximum performance of the system and identify the sites of exergy destruction. Exergy analysis of a complex system can be performed by analyzing the components of the system separately. Identifying the main sites of exergy destruction shows the direction for potential improvements. An important objective of exergy analysis for system that consume work such as refrigeration, liquefaction of gases, and distillation of water is finding the minimum work required for a certain desired result. Exergy analysis is applied to a system which describes all loses both in the various components of the system and in the system as a whole. With the help of this analysis the magnitude of these losses or irreversibilities and their order of importance can be understood. With the use of irreversibility, which is a measure of process imperfection, the optimum operating conditions can be easily determined. It can be said that exergy analysis can indicate the possibilities of thermodynamic improvement for the process under consideration.

The difference of the flow availability of a stream and that of the same stream at dead state is called flow exergy (e) and by ignoring chemical exergy terms, is given by:

$$e = (h - T_0 s) + 0.5\,V^2 + gZ - (h_0 - T_0 s_0) \quad (10)$$

Ignoring the potential and kinematic energy terms, (10) becomes:

$$e = (h - T_0 s) - (h_0 - T_0 s_0) \quad (11)$$

The energy balance equation is given by:

$$E_w = \sum E_Q + \sum(me)_i - \sum(me)_o + T_0 S_{gen} \quad (12)$$

In (12), the term T_0S_{gen} is defined as the irreversibility and can be written as:

$$I = T_0 S_{gen} \quad (13)$$

If the above exergy analysis, formulations are performed on each component of the simple vapor compression system shown in Fig. 1, the exergy destruction of each element can be found. Thus: Exergy destruction in compressor:

$$X_{comp} = m_{ref}\,(e_4 - e_5) - W \quad (14)$$

Exergy destruction in condenser:

$$X_{cond} = m_{ref}\,(e_6 - e_1) + m_{ef,\,cond}\,(e_9 - e_{10}) \quad (15)$$

Exergy destruction in evaporator:

$$X_{evap} = m_{ref}\,(e_2 - e_3) + m_{ef,\,evap}\,(e_7 - e_8) \quad (16)$$

Exergy destruction in throttle valve:

$$X_{tv} = m_{ref}\,(e_1 - e_2) \quad (17)$$

Total exergy destruction of the system is given by:

$$X_{total} = (X_{tv}) + (X_{evap}) + (X_{cond}) + (X_{comp}) \quad (18)$$

Exergitic efficiency is given by [4]:

$$\eta_{ex} = \text{Exergy of product/ Exergy of fuel} \quad (19)$$

For the vapor compression refrigeration system, exergy of product is the exergy of the heat abstracted in to the evaporator from the space to be cooled and exergy of fuel is the actual compressor work input.

The exergy destruction ratio is defined as:

$$EDR = \text{Total Exergy Destruction / Exergy of product} \quad (20)$$

The above equations have been solved numerically by using the thermodynamic property data for five refrigerants (R22, R134A, R410A, M20 and R407C) in EES [17]. The program gives the COP and all other system parameters (compressor work, enthalpy and entropy at different state points, exergy destruction in different components, refrigerant mass flow rate, EDR, exergetic efficiency, temperature and pressures at different state points etc.) for the following set of input data: evaporator coolant inlet temperature (Tin, evap in K), condenser coolant inlet temperature (Tin, cond in K), rate of heat absorbed by evaporator (Q_{evap} in kW), product of condenser effectiveness and capacitance rate of external fluid [$(\epsilon C)_{cond}$, kW/K], product of evaporator effectiveness and capacitance rate of external fluid [$(\epsilon C)_{evap}$, kW/K] and efficiency of compressor (η_{isen}).

III. Results and Discussion

A. Input Conditions

It should be noted that for an actual vapor compression system the efficiency of the reciprocating compressor and effectiveness of the heat exchangers do not remain constant with the variation in refrigeration capacity. However, for the present investigation, we have considered these parameters to be constant. The values of inputs at design condition are given in Table II.

Based on these design conditions, operating parameters, such as COP, compressor work, refrigeration efficiency, mass flow rate of refrigerant, exergy destruction in various components, exergetic efficiency and EDR are calculated (Table III).

TABLE II
Values of Inputs at Design Point

Parameters	Values
Evaporator coolant inlet temperature ($T_{in, evap}$)	277 K
Condenser coolant inlet temperature ($T_{in, cond}$)	313 K
Rate of heat absorbed by evaporator (Q_{evap})	66.67 kW
Product of condenser effectiveness and capacitance rate of external fluid [$(\epsilon C)_{cond}$]	9.39 kW/K
Product of evaporator effectiveness and capacitance rate of external fluid [$(\epsilon C)_{evap}$]	8.2 kW/K
Efficiency of compressors (η_{isen})	0.65

TABLE III
Values of Performance Parameter at Design Point

Parameters	R22	R134A	R410A	R407C	M20
COP	2.352	2.31	2.06	2.329	2.182
Compressor work (kW)	28.352	28.921	32.337	28.628	30.544
% Refrigerating efficiency	47.39	46.45	41.85	46.93	44.09
Mass flow rate (kg/s)	0.475	0.535	0.499	0.511	0.542
Total exergy destruction (kW)	16.65	17.38	20.4	17.05	18.91
% Exergy destruction in compressor	48.13	50.18	45.41	49.24	48.19
% Exergy destruction in condenser	18.73	13.76	17.5	19.11	17.34
% Exergy destruction in evaporater	10.22	9.81	8.31	3.37	1.83
% Exergy destruction in throttle valve	22.92	26.25	28.78	28.27	32.64
% Exergitic efficiency	15.1	14.79	13.25	14.95	14.02
EDR	3.89	4.06	4.76	3.98	4.415

B. Characteristic Performance Curves

The characteristic performance curves of vapor-compression refrigeration systems are defined as a plot between the inverse coefficient of performance (1/COP) and inverse cooling capacity (1/Qevap) of the system. Fig. 3 shows the performance curve obtained by using the R22 thermodynamic model for the above mentioned design conditions. Product of evaporator effectiveness and capacitance rate of external fluid [(ϵC)evap] is taken from the actual performance of the system reported by Zubair [15]. It was found that the characteristic performance curve (Fig. 3) for the thermodynamic model of R22 is nearly same, as obtained for the actual system [15] indicating the validity of the thermodynamic model applicable for system design and performance evaluation purpose. Presently the model has been used to study the performance of a variable-speed refrigeration system in which the evaporator capacity is varied by changing the mass-flow rate of the refrigerant, while keeping the inlet chilled-water temperature as constant.

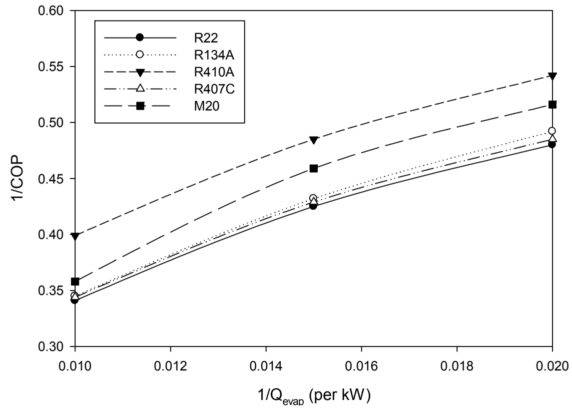

Fig. 3 Comparison of performance curves of different refrigerants

In Fig. 3, a comparison of the effect of variation of inverse of cooling capacity ($1/Q_{evap}$) with inverse of coefficient of performance (1/COP) is also shown for R134A, R410A, M20 and R407C respectively in comparison to R22. COP of the system increases with increase in cooling capacity. From the graphs it can be seen that an approximate linear relationship exists between (1/COP) and (1/Qevap). For refrigeration capacity variation from 50 kW to 100 kW the corresponding variation of COP for R22 is 2.083 to 2.936. For the entire range of cooling capacity, COP for R22 is higher as compared to all the substitutes. COP variation for R407C and R134a is varying close to the R22. The COP of R407C is 0.88 % lower at 100 kW cooling capacity which further reduced to 1.01 % at 50 kW cooling capacity.

It can be seen from Fig. 4 that, at high evaporator capacity, the refrigerant mass-flow rate through the system is increased.

This increases the temperature difference in the heat exchangers. Therefore, the losses due to finite-temperature difference in the heat exchangers are also high and, hence, the COP is reduced (Fig. 5). But as the capacity is decreased, the temperature difference in heat exchangers also decreases, therefore the losses due to the finite rate of heat transfer also decreases and therefore COP of the system increases. At maximum cooling capacity, the mass flow rate is minimum for R22 and maximum for M20. The difference among the mass flow rates reduces with decrease in cooling capacity.

Fig. 6 shows comparison of the variations in refrigerating efficiency for the variable speed system for R22, R134A, R410A, R407C and M20 respectively.

The refrigerating efficiency decreases with increase in refrigeration capacity owing to increased irreversible losses in the heat exchangers at high evaporator capacity (refer Figs. 4-6). However, Fig. 6 shows that, for refrigerating capacity less than the design point value, the efficiency of a variable speed system is high. It should be emphasized that the chilled water inlet temperature is kept constant with evaporator capacity for a variable-speed system which makes the refrigerating efficiency greater than the fixed-speed system at low

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refrigerating capacity. The refrigerating efficiency is maximum for R22 and is minimum for R410A at the designed point as well as for the entire range of evaporation capacity.

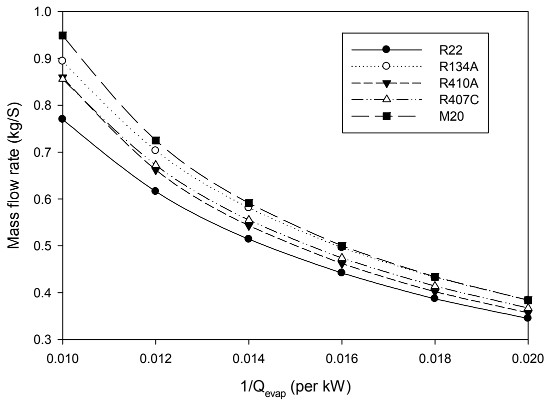


Fig. 4 Variation of mass flow rate with 1/Q_{evap}

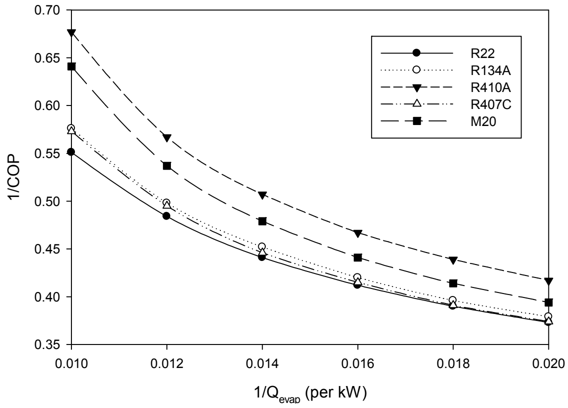


Fig. 5 Variation of 1/COP with 1/Q_{evap}

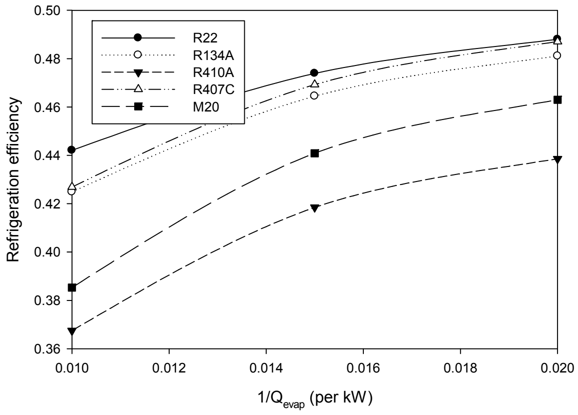


Fig. 6 Variation in refrigeration efficiency with 1/Q_{evap}

It is found that as the inlet temperature of external fluid of evaporator temperature increases, the exergy destruction increases in compressor, condenser, evaporator while decreases in throttle valve for all the five refrigerants. At the design point the exergy destruction is maximum for R134A and is minimum for R410A in compressor and the exergy destruction is maximum for R407C and is minimum for R134A in condenser. At the design point the exergy destruction in evaporator is maximum for R22 and is minimum for M20. In throttle valve exergy destruction is maximum for M20 and is minimum for R22 at the design point. From Fig. 8, it can be seen that the maximum exergy destruction in the system takes place in compressor followed by throttle valve which is followed by condenser and then evaporator for any refrigerant at the given evaporator temperature.

Fig. 8 (a) shows the variation of total exergy destruction in system with inlet temperature of external fluid at evaporator. It can be observed that as the inlet temperature of coolant increases, total exergy destruction in the system decreases for all refrigerants.

Total exergy destruction is maximum for R410A and is minimum for R22 for a given inlet evaporator temperature of external fluid.

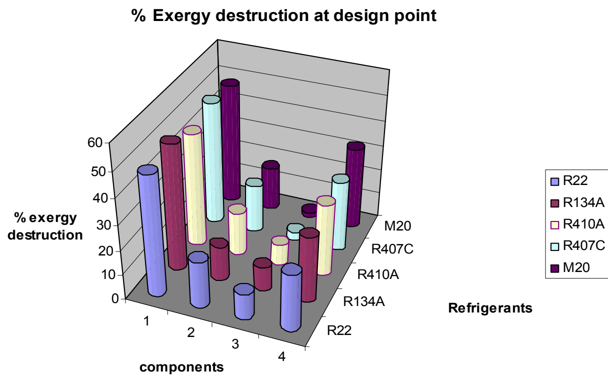


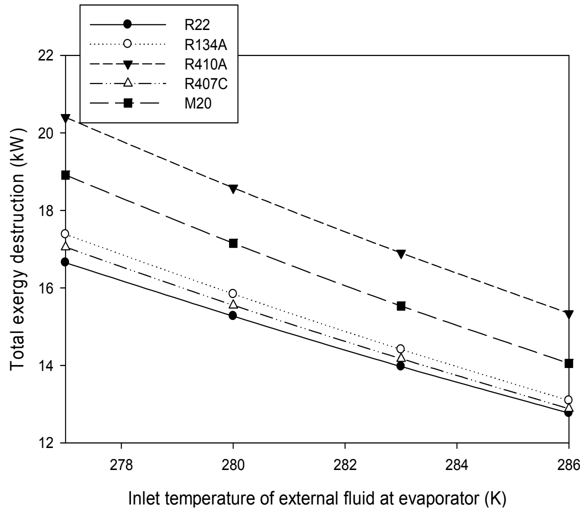
Fig. 7 Percentage exergy destruction at design point (1. Compressor, 2. Condenser, 3. Evaporator, 4. Throttle valve)

Fig. 8 (b) shows the variation of exergetic efficiency of system with inlet temperature of external fluid at evaporator. As the inlet temperature of coolant increases, exergetic efficiency in the system decreases. Exergetic efficiency is maximum for R22 and is minimum for R410A for a given evaporator inlet temperature.

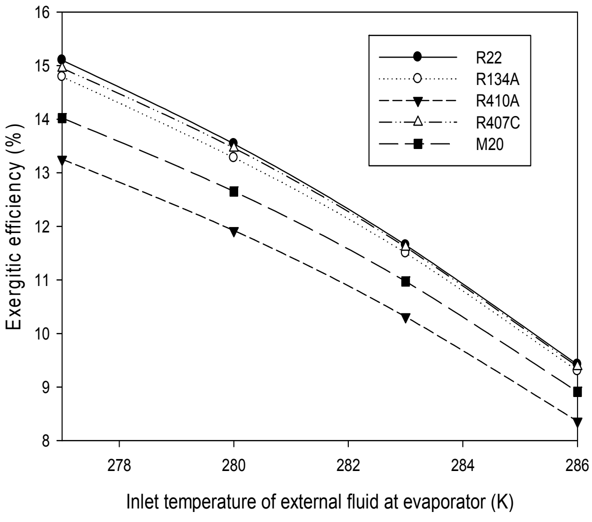
Fig. 8 (c) shows the variation of exergy destruction ratio of system with inlet temperature of external fluid at evaporator. It can be observed that as the inlet temperature of coolant increases, EDR of the system increases for all refrigerants. System EDR is maximum for R410A and is minimum for R22 for a given inlet evaporator temperature.

It is found that as the condenser temperature increases, the exergy destruction in compressor, condenser and evaporator decreases for all the five refrigerants. In throttle valve, exergy destruction increases with increase in inlet temperature of external fluid for all five refrigerants. At the design point the exergy destruction is maximum for R134A and is minimum for R410A in compressor. At the design point the exergy destruction is maximum for R407C and is minimum for R134A in condenser. At the design point the exergy destruction in evaporator is maximum for R22 and is minimum for M20. In throttle valve exergy destruction is maximum for M20 and is minimum for R22 at the design point

Fig. 9 (a) shows the variation of total exergy destruction in system with inlet temperature of external fluid at condenser. As the inlet temperature of coolant increases, total exergy destruction in the system increaser for all refrigerants. Total exergy destruction is maximum for R410A and is minimum for R22 for a given inlet condenser temperature.



(a)



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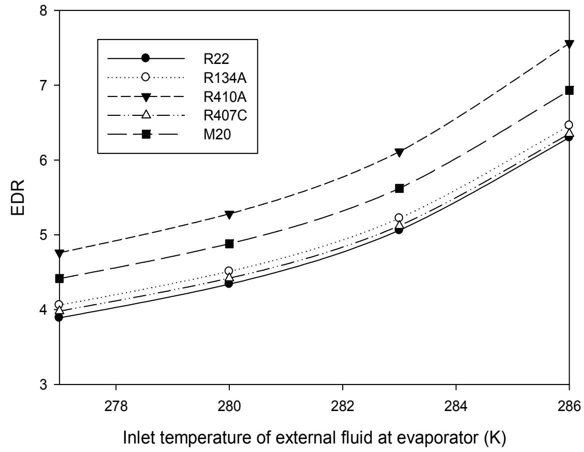
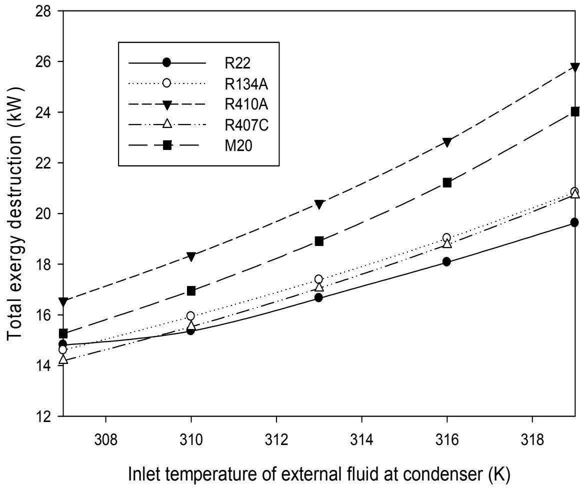
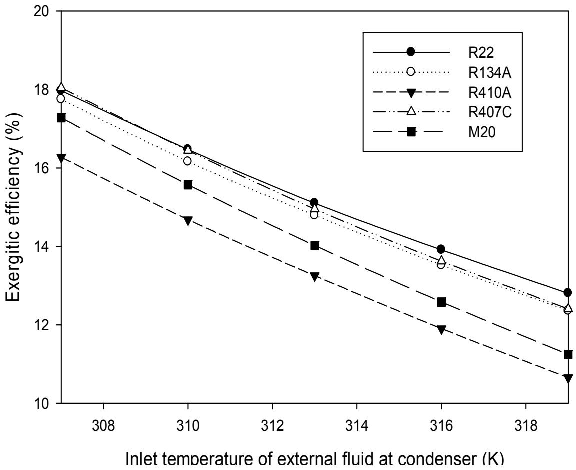


(c)

Fig. 8 (a) Variation of total exergy destruction with inlet temperature of external fluid at evaporator, (b) Variation of exergetic efficiency with inlet temperature of external fluid at evaporator, (c) Variation of EDR with inlet temperature of external fluid at evaporator



(a)



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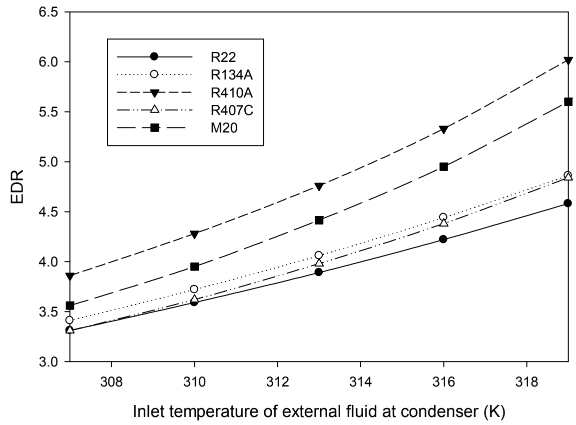

(c)

Fig. 9 (a) Variation of total exergy destruction with inlet temperature of external fluid at condenser, (b) Variation of exergetic efficiency with inlet temperature of external fluid at condenser, (c) Variation of EDR with inlet temperature of external fluid at condenser

Fig. 9 (b) shows the variation of exergetic efficiency of system with inlet temperature of external fluid of condenser. As the inlet temperature of coolant increases, exergetic efficiency in the system decreases for all refrigerants. Exergetic efficiency is maximum for R22 and is minimum for R410A for any inlet evaporator temperature range.

Fig. 9 (c) shows the variation of exergy destruction ratio of system with inlet temperature of external fluid at condenser. As the inlet temperature of coolant increases, EDR of the system increases for all refrigerants. EDR is maximum for R410A and is minimum for R22 for a given inlet condenser temperature.

IV. Summary of Relative Comparison of Refrigerants with R22 System

Several refrigerants have emerged as candidates to replace R22, the most widely used fluorocarbon refrigerant in the world. These include the environmentally-friendly HFC refrigerants R134A, R410A, R407C and M20. R134A is a pure refrigerant, whereas R407C and 410A are blends of refrigerants.

TABLE IV
Summary of Relative Comparison of Various Refrigerants with R22

Factors	R410A	R407C	R134A	M20
Pressure ratio	99.01 %	100.1 %	117.18 %	90.78 %
Refrigerant charge	105.05 %	107.5 %	112.63 %	114.1 %
COP	87.67 %	99.02 %	98 %	92.77 %
Compressure work	114.05 %	100.9 %	102 %	100.97 %
Refrigerating efficiency	88.3 %	99.02 %	98.01 %	93.03 %
Condenser heat transfer	103.98 %	100 %	100%	101.49 %
Total exergy destruction	122.52 %	102.4 %	104.38 %	113.57 %
Exergitic efficiency	87.74 %	99 %	97.94 %	92.84 %
EDR	122.36 %	102.3 %	104.37 %	113.49 %
Redesign required	Less	Minor	Significant	Significant
System cost	Lower	Same	Slightly	Lower

R410A is a mixture of R32 and R125, while R407C is a blend of R32, R125 and R134A. The advantages of blending refrigerants are that properties such as flammability, capacity, discharge temperature and efficiency can be tailored for specific applications. There are many considerations in selecting a refrigerant, and each has an impact on the overall performance, reliability, cost and market acceptance of a manufacturer's system. (Table IV).

V. Conclusions

In this communication, an extensive thermodynamic analysis of R134A, R410A, R407C and M20 in comparison to R22 have been presented. From the comparison of performance parameters, it can be concluded that the performance of R407C is nearly same as R22. Hence, it has the potential to replace R22 system with minimum investment and efforts.

Nomenclature

COP	Coefficient of performance
C	Capacitance rate for the external fluids (kW/K)
h_j	Specific enthalpy of refrigerant at state point j (KJ/Kg)
m	Mass flow rate (kg/S)
Q_{cond}	Rate of heat rejection in condenser (kW)
$Q_{loss, cond}$	Rate of heat leak from the hot refrigerant (kW)
Q_{evap}	Rate of heat absorbed by the evaporator (kW)
$Q_{loss, evap}$	Rate of heat leak from the ambient to the cold refrigerant (kW)t
W	Rate of electrical power input to compressor (kW)
T_{cond}	Refrigerant temperature in the condenser (K)
$T_{in, cond}$	Condenser coolant inlet temperature (K)
$T_{in, evap}$	Evaporator coolant inlet temperature (K)
T_{evap}	Refrigerant temperature in the evaporator (K)
Ta	Ambient Temperature (K)
X	Exergy destruction (kW)
e	Exergy (KJ/kg)

Greek Symbols

ε	Effectiveness of Heat exchanger

Subscripts

1,2,3,…	state points
comp	compressor
cond	condenser
evap	evaporator
tv	throttle valve
ref	refrigerant
ef	external fluid
isen	Isentropic
ex	exergetic

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