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Comparative performance study of vapour compression refrigeration system with R22/R134a/R410A/R407C/M20

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Abstract
Several refrigerants have emerged as substitutes to replace R22, the most widely used fluorocarbon refrigerants in the world. These include the environmentally –friendly hydrocarbon (HFC) refrigerants R134a, R410A, R407C and M20. In the present research study a refrigerant property dependent thermodynamic model of a simple reciprocating system, which can simulate the performance of actual system as closely as possible, has been used to compare the characteristics of various refrigerants [R22, R134a, R410A, R407C and M20] used by world manufacturers to meet the challenges of higher efficiency and environmental responsibility while keeping their system affordable. Considering the recent trends of replacement of ozone depleting refrigerants and improvement in system efficiency, in the present study, R407C can be a potential HFC refrigerant replacement for new and existing systems presently using R22 with minimum investment and efforts.

Keywords: Compression system, Modelling, performance study, Refrigeration, R407C.

1. Introduction
Refrigeration, cooling, and heating processes are important in a variety of everyday situations, including the air conditioning and heating of buildings, hospitals, operation theatres, hotels, restaurants, automobiles and transportation. Refrigeration also finds large-scale industrial applications, especially in the manufacture of ice, dehydration of gases, domestic and commercial refrigerators, large scale warehouses for storage and preservation of foods and beverages and a host of other commercial and industrial services. Applications of refrigerants in the petroleum industry include lubricating-oil purification, low temperature reactions, and the separation of volatile hydrocarbons. Evaporation and condensing processes in refrigeration systems are as a result of the heat transfer occurring by means of phase change in refrigerants. Therefore, the design of a cooling system largely depends on the properties of the refrigerants. 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 to these working fluids gained momentum in the recent years. By international agreement (Montreal Protocol), signed in 1987 and later amended several times, this group of refrigerants, 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 displacement of earlier working fluids, with successive criteria, and remerge interest in some early refrigerants, for example renewed interest in those now identified 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, their harmful effects on the ozone layer and worldwide concern over global warming and, secondly, there is a stringent need for improvement in 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, or even quartet 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 vapour state), was carried out by Monte [2, 3]. Arora et al. [4] did the theoretical analysis of a vapour compression refrigeration system with R502, R404A and R507A. Their work presents a detailed exergy analysis of an actual vapour compression refrigeration (VCR) cycle. The efficiency effect in condenser was highest, and lowest in liquid vapour 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 vapour 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 vapour 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 applied to residential air conditioner. Wu et al. [11] reported a ternery blend 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 vapour compression plant. In this work a simplified steady-state model to predict the energy performance of a single stage vapour compression plant was proposed. This model has been validated using 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 vapour 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. It was concluded that among the mixtures considered M20 (R407C 80% & HC blend 20%) had the optimal composition in respect of better COP and per day energy consumption.

From literature review, several refrigerants have emerged as substitutes to replace R22, the most widely used fluorocarbon refrigerants in the world. These include the environmentally –friendly hydrocarbon (HFC) refrigerants R134a, R410A and R407C and M20. Table 1 shows the physical and environmental characteristics of these refrigerants.

Table 1. 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 (^{0}C)	0		0.08	7.0	

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 characteristics of various refrigerants [R22, R134a, R410A, R407C and M20] used by world manufacturers to meet the challenges of higher efficiency and environmental responsibility while keeping their system affordable.

2. Thermodynamic model for performance comparison

Considering the steady-state cyclic operation of the system shown in Figures 1 and 2, refrigerant vapour 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 Figure 1, using the first law of thermodynamics and the fact that change in internal energy is zero for a cyclic process, we get

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

where Q_{cond} is the rate of heat rejection in condenser (kW), $Q_{loss,cond}$ is the rate of heat leak from the hot refrigerant (kW), Q_{evap} is the rate of heat absorbed by the evaporator (kW), $Q_{loss,evap}$ is the rate of heat leak from the ambient to the cold refrigerant (kW), W is the rate of electrical power input to compressor (kW) and $Q_{loss,W}$ is the rate of heat leak from the compressor shell to ambient (kW).

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

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

where ε is the effectiveness of heat exchanger, C is capacitance rate for the external fluids (kW/K), $T_{in,evap}$ is the evaporator coolant inlet temperature (K), T_{evap} is refrigerant temperature in the evaporator (K), m_{ref} is the mass flow rate of refrigerant (kg/s) and h is specific enthalpy of refrigerant at state point (kJ/kg).

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

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

where, T_{cond} is the refrigerant temperature in the condenser (K) and $T_{in,cond}$ is the condenser coolant inlet temperature (K).

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

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

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

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

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

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$$COP = Q_{evap} / W \tag{7}$$

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

The above equations have been solved numerically by using the thermodynamic property data (using REFPROF) for the five refrigerants (R22, R134a, R410A, R407C and M20). The program gives the COP and all other system parameters for the following set of input data: Evaporator coolant inlet temperature ($T_{in,evap}$), Condenser coolant inlet temperature ($T_{in,cond}$), Rate of heat absorbed by evaporator (Q_{evap}), product of condenser effectiveness and capacitance rate of external fluid [$(\varepsilon C)_{cond}$], product of evaporator effectiveness and capacitance rate of external fluid [$(\varepsilon C)_{evap}$] and efficiency of compressor (η_{comp}).

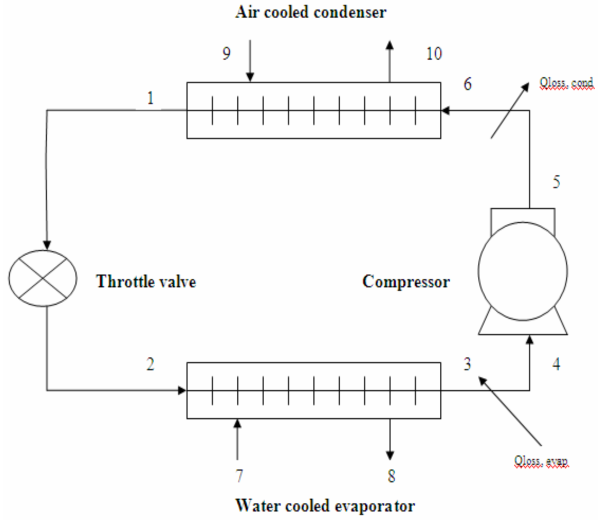


Figure 1. Schematic diagram of a simple refrigeration cycle

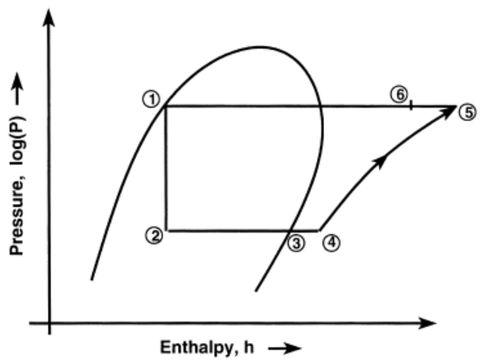


Figure 2. Pressure-Enthalpy diagram for vapour compression cycle

3. Results and discussions

3.1 Design conditions

The input to the thermodynamic model is given below [15]:

Evaporator coolant inlet temperature ($T_{in,evap}$) = 277 K

Condenser coolant inlet temperature ($T_{in,cond}$) = 313 K

Range for rate of heat absorbed by evaporator (Q_{evap}) = 50 to 100 kW [design value = 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 ($(\epsilon C)_{evap}$) = 8.20 kW/K

Efficiency of compressor (η_{comp}) = 0.65

Based on these design conditions, operating parameters, such as COP, compressor work, refrigeration efficiency and mass flow rate of refrigerant are calculated (Table 2). All the alternative refrigerants are having lower values of COP (14.17 % to 0.97 %), refrigeration efficiency (13.23 % to 0.97 %), higher compressor work (14.05 % to 0.97 %) and mass flow rate (14.1 % to 5.05 %) as compared to R22. Based on these criterions, R407C is the nearest substitute for R22 for which COP is 0.97% lower.

Table 2. Comparison of performance parameters for different refrigerants

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
Condenser pressure (MPa)	1.938	1.314	3.091	2.21	2.472
Evaporator pressure (MPa)	0.432	0.25	0.696	0.492	0.608
Heat transfer rate(kW)	94.36	98.12	94.36	94.36	95.77

3.2 Characteristic performance curves

The characteristic performance curves of vapour-compression refrigeration systems are defined as a plot between the inverse coefficient of performance (1/COP) and inverse cooling capacity ($1/Q_{evap}$) of the system. Figure 3 shows the characteristic chiller 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 [$(\epsilon C)_{evap}$] is taken from the actual performance of the system reported by Zubair [15]. It was found that the characteristic performance curve (Figure 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. The model can be used to study the variation of refrigerating efficiency, the effect of subcooling and superheating for the variable speed system.

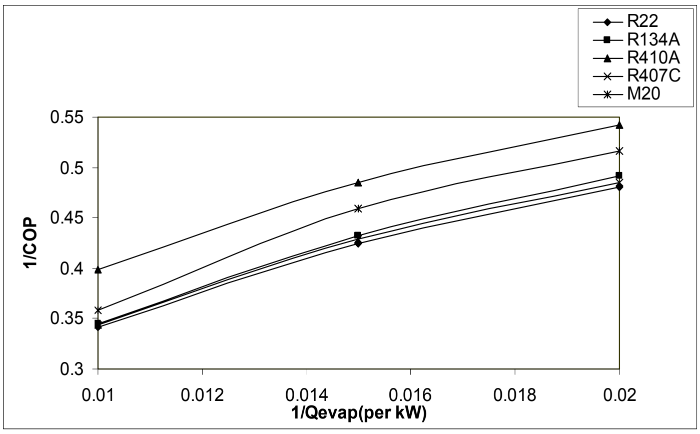
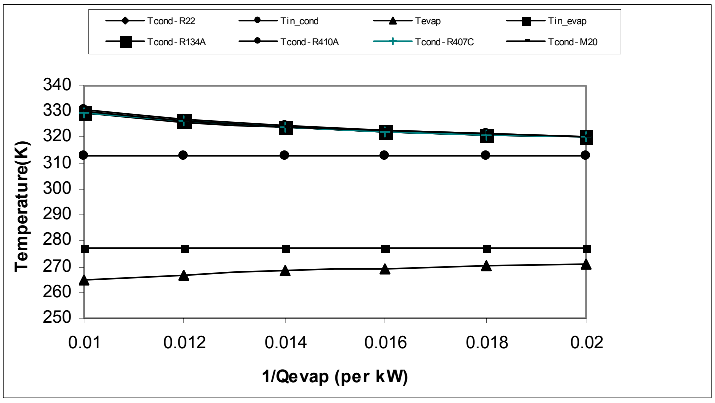


Figure 3. Comparison of performance curves of different refrigerants

In Figure 3, a comparison of the effect of variation of inverse of cooling capacity ($1/Q_{evap}$) with inverse of coefficient of performance (1/COP) are 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 there is an approximate linear relationship exits between (1/COP) and ($1/Q_{evap}$). 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.

3.3 Characteristic performance comparison in a variable speed system

The variation in performance characteristics of a refrigeration system for different refrigerants in which the evaporator capacity is varied by varying the compressor speed is shown in Figures 4 to 7, respectively. The curves shown are plotted for the designed conditions mentioned above. It should be noted that for an actual system, as refrigeration capacity of the system varies, the performance of the compressor and heat exchangers, represented by η_{comp} and ϵ, respectively, will not be constant. However, for the present investigation, these parameters are considered to be constant. Figure 4(a) shows the effect of variation of inverse of cooling capacity on system temperatures. $T_{in,cond}$, $T_{in,evap}$, T_{evap} data are independent of refrigerant temperature in evaporator and condenser. There is slight change in condenser temperature (T_{cond}) for different refrigerants for higher cooling capacity as shown in figure 4(a). The temperature gradient inside the condenser and evaporator increases with increase in cooling capacity.



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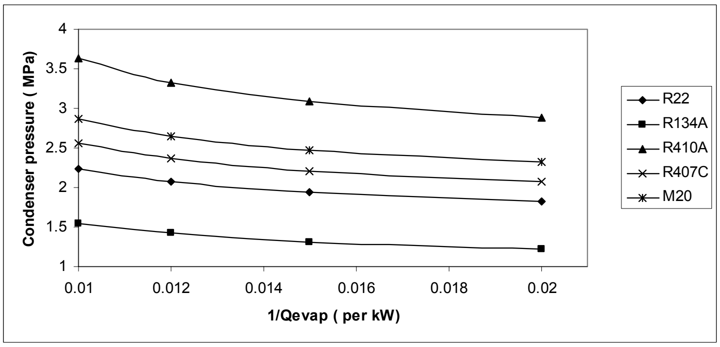

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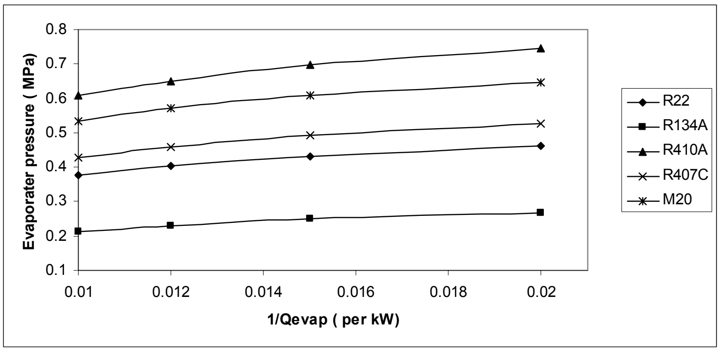

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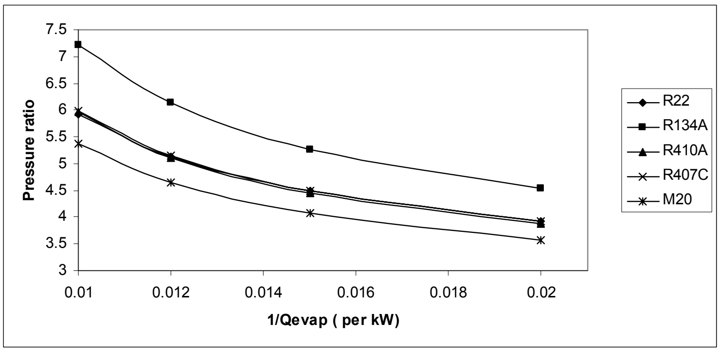


(d)

Figure 4. Variation of (a) operating temperatures (b) condenser pressure (c) evaporator pressure and (d) pressure ratio vs. $1/Q_{evap}$

Variation of condenser and evaporator pressure with inverse of cooling capacity is shown in Figure 4(b) and Figure 4(c) respectively. The condenser pressure increases whereas evaporator pressure decreases with increase in cooling capacity. R410A data shows the highest values for pressures in condenser and evaporator as compared to R134A. As shown in Figure 4(d), pressure ratio increases with increase in cooling capacity. For the entire range of cooling capacity the values for pressure ratio is highest and lowest for R134A and M20 refrigerants, respectively. The variation of pressure ratio is almost same for R22, R134A and R407C.

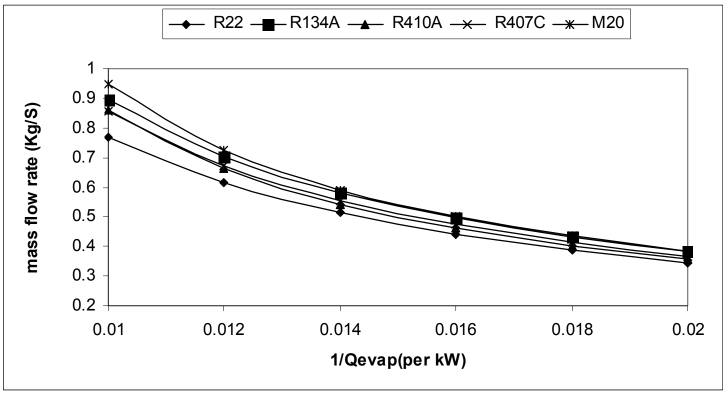


Figure 5. Variation of mass flow rate with $1/Q_{evap}$

It can be seen from Figure 5 that, at high evaporator capacity, the refrigerant mass-flow rate through the system is increased and, thus, the temperature difference in the heat exchangers is also high. Therefore, the losses due to finite-temperature difference in the heat exchangers are also high and, hence, the COP is reduced (Figure 6). 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.

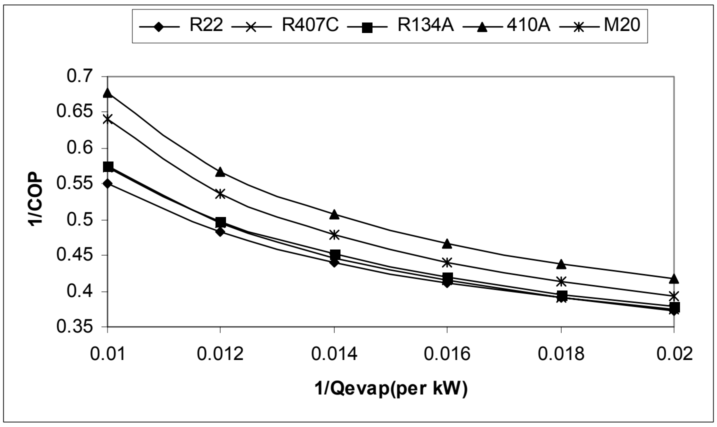


Figure 6. Variation of 1/COP with $1/Q_{evap}$

Figure 7 shows compassion 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 Figures 4 to 6)**.** However, the Figure 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 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.

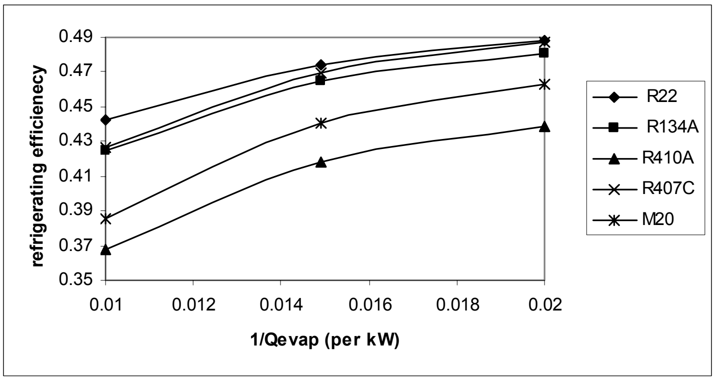
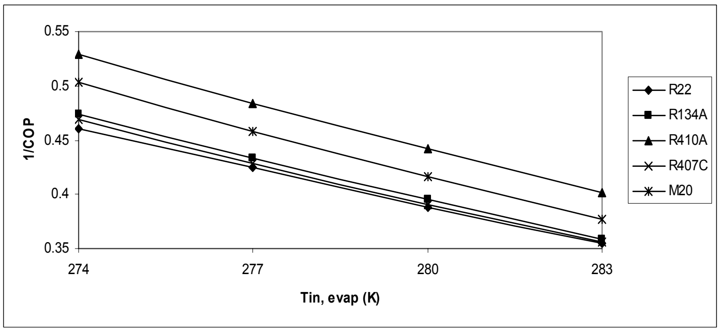


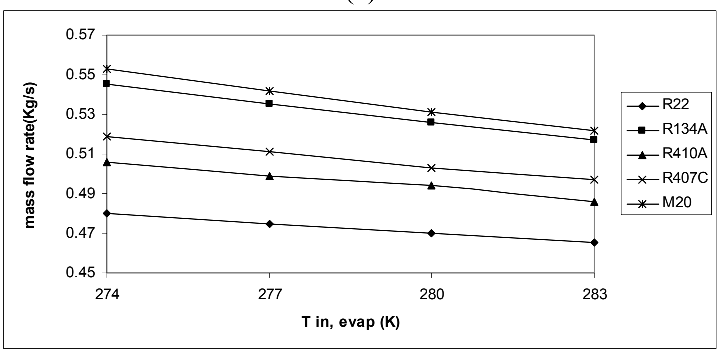
Figure 7. Variation in refrigeration efficiency with $1/Q_{evap}$

3.4 Effect of evaporator and condenser inlet temperature of external fluids

Figure 8 (a) and (b) shows the effect of variation of inlet evaporator temperature of external fluid on COP and refrigerant mass flow rate, respectively. It can be observed that as the inlet temperature of external fluid at evaporator increases, COP of system increases and the corresponding refrigerant mass flow rate decreases. With increase in evaporator temperature of external fluid, the corresponding evaporator temperature also increases due to which the pressure ratio across the compressor reduces causing compressor work to reduce. Further, cooling capacity increases because of increase in specific refrigerating effect which causes reduction in refrigerant mass flow rate. The combined effect of these two factors is to enhance the overall COP. R22 presents maximum COP and minimum mass flow rate among all the refrigerants corresponding to inlet evaporator temperature of the external fluid. The COP of R134A at 274 K is 2.67 % lower as compared to R22. This difference reduces to 1.24 % at 283 K. The COP of R407C lies between R22 and R134A.



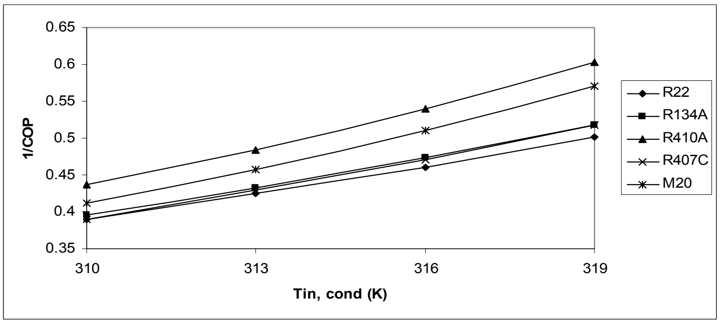
(a)



(b)

Figure 8. Variation of inlet temperature of external fluid in the evaporator ($T_{in,\ evap}$) vs. (a) 1/COP and (b) mass flow rate

Figure 9 (a) and (b) shows the effect of inlet condenser temperature of external fluid on COP and refrigerant mass flow rate. It can be seen that as the inlet temperature of external fluid at condenser increases, COP of system decreases and refrigerant mass flow rate increases. With increase in condenser temperature of external fluid, the corresponding condenser refrigerant temperature also increases due to which the pressure ratio across the compressor increases causing compressor work to increase. Further, total heat rejection capacity increases because of increase in compressor work and mass flow rate. The combined effect of these two factors is to reduce the overall COP. R22 presents maximum COP and minimum mass flow rate among all the refrigerants corresponding to inlet condenser temperature of the external fluid. The COP of R134A at 310 K is 1.83% lower as compared to R22. This difference increases to 3.31% at 319 K. The COP of R407C lies between R22 and R134A.



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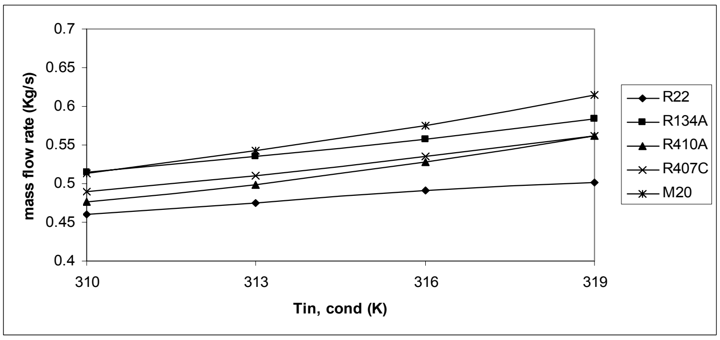


(b)

Figure 9. Variation of inlet temperature of external fluid in the condenser ($T_{in,\ cond}$) vs. (a) 1/COP and (b) mass flow rate

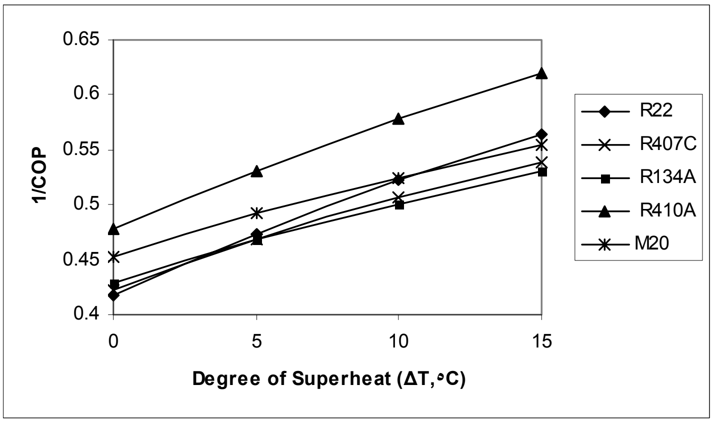
3.5 Effect of subcooling and superheating

The superheating of refrigerant (after exiting the evaporator and before entering the compressor) may occur owing to the heat gain in the line joining the evaporator and compressor. This heat gain process is shown from state (3) to (4) in Figure 2. It is obvious that the specific volume of refrigerant vapour is increased owing to superheating, thus reducing the mass-flow rate through the fixed-displacement compressor. On the other hand, in the subcooling the refrigerant beyond the saturated state after exiting the condenser and before entering the expansion valve normally occurs owing to heat losses in the line joining the condenser and expansion valve. It is expected that subcooling increases the system performance because the specific refrigeration capacity increases with subcooling.

Figure 10 shows the separate effect of superheating, subcooling and both superheating and subcooling for different refrigerants. As expected, the superheating degrades the performance of the system, while subcooling improves the system COP. When we take equal amounts of superheating and subcooling, the performance degrades. Therefore, the Figure 10 shows that, for the given operating condition, the effect of superheating has more influence on the system overall performance.

3.6 Effect of (ϵC) of external fluid in condenser and evaporator

The factor (ϵC) represents multiplication of effectiveness of heat exchanger and heat carrying capacity of external fluid. The effect of variation of (ϵC) of external fluid on COP and refrigerant mass flow rate is shown in Figures 11 and 12 respectively for condenser and evaporator. As the (ϵC) of external fluid reduces, the mass flow rate of the system increases whereas COP reduces.



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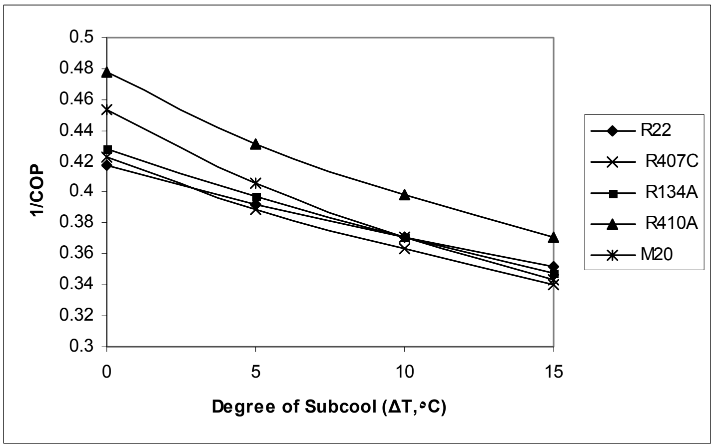

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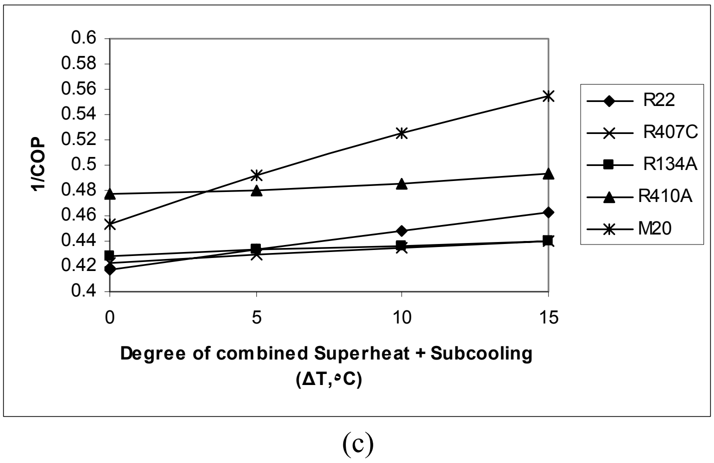
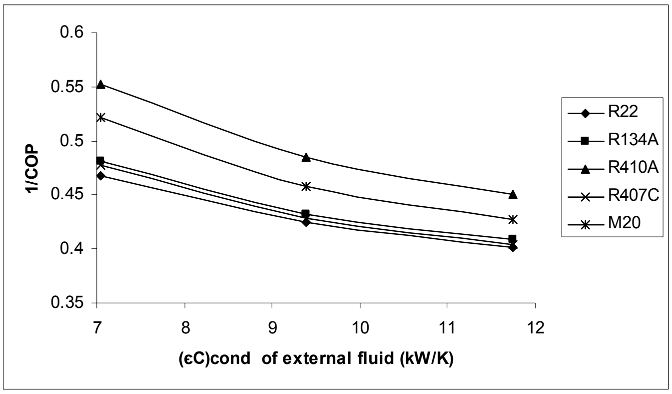


(c)

Figure 10. Effect of (a) superheating (b) subcooling and (c) combined superheating and subcooling on the performance of a simple vapour compression system



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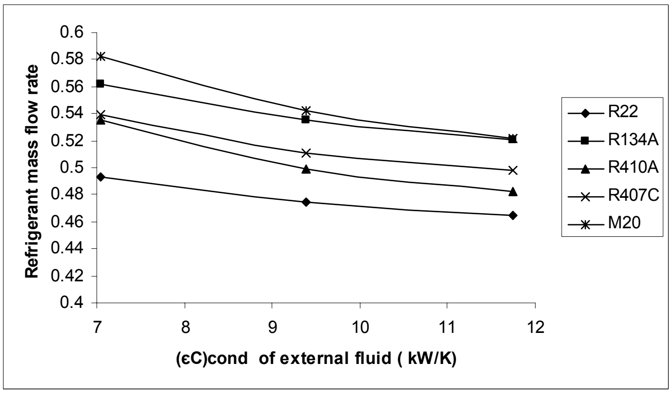
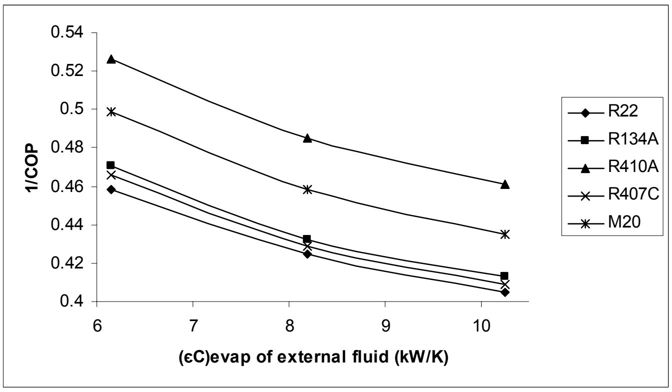


(b)

Figure 11. Effect of *$(\epsilon C)_{cond}$* of external fluid in the condenser vs. (a) 1/COP (b) refrigerant mass flow rate



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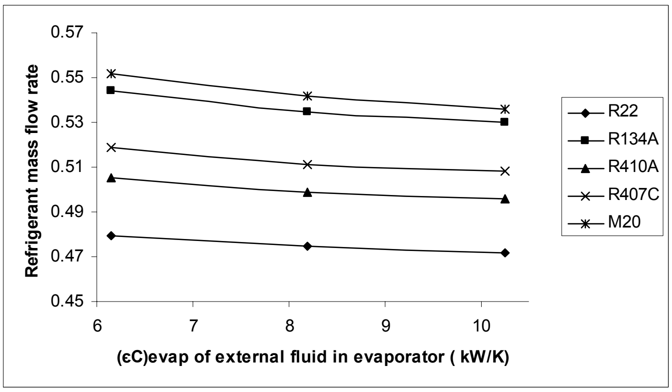

(b)

Figure 12. Effect of *$(\epsilon C)_{evap}$* of external fluid in the evaporator vs. (a) 1/COP (b) refrigerant mass flow rate

4. Summary of relative comparison of refrigerants in reference to R22

Several refrigerants have emerged as candidates to replace R22, the most widely used fluorocarbon refrigerant in the world. These include the environmentally-friendly hydrofluorocarbon (HFC) refrigerants R134a, R410A, R407C and M20. R134a is a pure refrigerant, whereas R407C and 410A are blends of refrigerants. 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. On the basis of above results, R134a, R410A, R407C and M20 are compared with R22 at the designed conditions (Table 3).

Table 3. Summary of relative comparison of refrigerants with R22

FACTORS	R410A	R407C	R134A	M20
Pressure ratio	99.01 %	100.12 %	117.18 %	90.78 %
Refrigerant flow rate	105.05 %	107.57 %	112.63 %	114.1 %
COP	87.67 %	99.02 %	98 %	92.77 %
Compressor Work	114.05 %	100.97 %	102 %	100.97 %
Refrigerating Efficiency	88.3 %	99.02 %	98.01 %	93.03 %
Condenser Heat Transfer	103.98 %	100 %	100%	101.49 %
Redesign Required	significant	Minor	Significant	Significant
System Cost	Lower	Same	Slightly more	Lower

R134a is a lower capacity and lower pressure refrigerant than R22. Because of these characteristics, a system with R134a of the same capacity requires a larger displacement compressor and larger evaporator, condenser, and tubing. The end result is a system which costs more to build and to operate than an equivalent R22 system.

R407C is a potential HFC refrigerant replacement for R22 system such as new or existing residential and commercial air conditioners and heat pumps. A system with R407 C having similar capacity and pressures as R22 can be designed. Because of these features, it can be used as an alternative in R22 systems with a minimum of redesign. System efficiency is slightly lower as compared R22 system due to temperature glide. R407C exhibits a relatively high temperature glide (7 K) compared to the other refrigerants, which have almost no glide. It also offer '0' ODP, low global warming potential. Europeon market embraced R407C and currently offer a wide R407C airconditioner product range. Further, a switch over to polyolester lubricant is also required.

R410A has been in the market place for more than 10 years and is the leading HFC refrigerant for replacing R22 in residential and light commercial air-condiitong and heat pump systems. R410A is having a 50-60% higher pressure refrigerant than R22. As a result of higher pressures and higher gas density, smaller displacement compressors can be used along with smaller diameter tubing and valves and therefore, R410A should only be used in new systems designed for this refrigerant and should not be substituted into existing R22 systems. Greater skill and attention to cleanliness is required during the installation of an R410A system to prevent moisture entering into the system. R410 A requires POE oil which is highly hydroscopic. Further, R410A has reduced environmental footprint as compare to an R22 unit for a comparable size range.

M20 is a highest pressure refrigerant as compare to other refrigerants taken for the analysis. It has low COP as compare to R22 system and has high refrigerant flow rate. So it requires small displacement compressor, large evaporator, condenser and tubing.

5. 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 R407C is a potential HFC refrigerant replacement for new and existing systems presently using R22 with minimum investment and efforts.

References

[1] Calm, J. M. The next generation of refrigerants – Historical review, considerations, and outlook: International Journal of Refrigeration 2008, 31, pp. 1123-1133.

[2] Monte, F. Calculation of thermodynamic properties of R407C and R410A by the Martin–Hou equation of state — part II: Technical interpretation: International Journal of Refrigeration, 2002, 25, pp. 314–329.

[3] Monte, F. Calculation of thermodynamic properties of R407C and R410A by the Martin–Hou equation of state — part I: Theoretical development: International Journal of Refrigeration, 2002, 25, pp. 306–313.

[4] Arora, A., and Kaushik, S. C. Theoretical analysis of a vapour compression refrigeration system with R502, R404A and R507A: International Journal of Refrigeration, 2008, 31, pp. 998-1005 .

[5] Wang, X., Hwang, Y., Radermacher, R. Investigation of potential benefits of compressor cooling. Applied Thermal Engineering , 2008, 28, pp. 1791–1797.

[6] Selbaş, R., Kızılkan, O., and Şencan A. Thermoeconomic optimization of subcooled and superheated vapor compression refrigeration cycle: Energy, 2006, 31, pp.2108–2128.

[7] Kiatsiriroat, T. and Thalang K. Performance Analysis of vapour compression refrigeration with R22/R124/R152a refrigerant: International Journal of Energy Research, 1997 vol. 21, 221-232.

[8] Arcakliogle E., Cavusoglu A., and Erisen A. Thermodynamic analysis of refrigerant mixtures for possible replacements for CFCs by an algorithm compiling property data: Applied Thermal Engineering, 2007, 26, 430-439.

[9] Han, X, H., Wang Q., Zhu Z. W., Chen, G. M. Cycle performance study on R32/R125/R161 as an alternative refrigerant to R407C: Applied Thermal Engineering, 2007, 27, 2559-2365.

[10] Chen J., and Yu J. Performance of a new refrigeration cycle using refrigerant mixture R32/R134a for residential air-conditioner applications: Energy and Buildings, 2008, 40, 2022-2027.

[11] Wu J., Chu Y., Hu J., Liu Z. Performance of mixture refrigerants R152a/125/R32 in domestic air-conditioner: International Journal of Refrigeration, 2009, 32, 1049-1057.

[12] Winkler, J., Aute, V., and Radermacher, R. Comprehensive investigation of numerical methods in simulating a steady-state vapor compression system: International Journal of Refrigeration, 2008, 31, pp. 930-942.

[13] Cabello,R., Navarro, J., and Torrella, E. Simplified steady-state modeling of a single stage vapour compression plant. Model development and validation: Applied Thermal Engineering, 2005, 25, pp. 1740–1752.

[14] Ecir K. E., Selbas, R.,and Sencan, A. Data mining techniques for thermophysical properties of refrigerants: Energy Conversion and Management, 2009, 50 pp. 399-412.

[15] Zubair, S. M. Performance evaluation of Vapour Compression System: International Journal of Refrigeration, 1999, 22, pp 235-243.

[16] Raj, M. H., And Mohan Lal D. On The Performance Of R22 and a HFC/HC Refrigerant Mixture Over A Range Of Charge Quantity In A Window Airconditioner: Acreconf, Feb'09.

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