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Performance Study of a Water-to-Water Heat Pump Using Non-azeotropic Refrigerant Mixtures R407C

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Abstract – *This paper compares the performance of the refrigerant R22 and the non-ozone depleting R407C for high temperature heat pump heat recovery application. While most reported works were conducted on residential heat pumps with evaporating temperature below zero degree Celsius, this study considers higher evaporating temperatures, from 5 to 35 degrees Celsius. The comparison was made by theoretical cycle calculation with variable heating and cooling capacities and at different temperatures. The experimental results of R407C were compared with the existing R22 refrigerant to explore the potential for retrofitting existing installations. It has been found that both the thermodynamic properties and general performance of R407C are comparable with that of R22. The operational behavior of R407C was found better with the increasing condensation and evaporation temperature. It has also been observed that the heating capacity of the refrigerant R407C is 5% higher than that of R22. When the compression ratio exceeds 3, at high evaporating temperatures, the coefficient of performance with R407C is greater than that obtained with R22. Therefore, R407C is a good substitute for R22 in all applications requiring high evaporation temperatures, such as air-conditioning plants and heat pumps for heat recovery.*

Keywords – Heat pump, R22, R407C.

1. INTRODUCTION

Many chlorofluorocarbons and hydro chlorofluorocarbons have been universally used as working fluids in air-conditioning and refrigeration systems for more than five decades and for heat pumps in the past few decades. Heat pump technology is a promising means of promoting efficient use of thermal energy and thus achieving energy saving. It is capable of recovering low-temperature waste heat, which would otherwise be discharged to the air or water, in an efficient way and converting it to high temperature useful energy for various processes. In this regard, a heat pump can be considered as an optimal heat recovery technology.

The refrigeration and heat pump industries are facing great challenges to reduce power consumption while they are also required to adopt new refrigerants that may increase the power consumption. Over the past decades the hydro-chlorofluorocarbon (HCFC) refrigerant R22 has been used as a working fluid in many refrigeration systems. But it will have to be phased out by 2030 in developing countries because of its high ozone depletion potential (ODP) and comparatively high global warming potential (GWP). So, some of the alternative refrigerants such as R410A, R410B, R407C, and R507 have been tried to replace R22. Yet uncertainties in the application of these mixtures exist, hence the provisions to undertake an experimental investigation of heat transfer, pressure drop, and overall system performance is necessary. The main difference between a non-azeotropic mixture and a pure working fluid is that with the mixture, the evaporation temperature rises along the evaporator and the condensation temperature falls along the condenser. Due to this problem, the necessary lift over of the compressor can be decreased [1]. Hence, the coefficient of performance (COP) can be increased. The improvements that are possible to obtain in reality depend on which mixture is used along with the external conditions. A ternary non-azeotropic refrigerant mixture, R407C has been considered as a potential substitute for R22 in air-conditioning and heat pump systems. It may be mentioned that the R407C is a non-azeotropic (zoetrope) blend with a severe temperature glide of around 7K. It consists of 23% R32 (CH_2F_2), 25% of R125 (CF_3CHF_2) and 52% of R134a (CF_3CH_2F) by weight. It has been mentioned that the R407C has very similar properties to that of R22 in terms of operating pressures and performance in dry expansion air conditioning systems. However, it is not identical to R22 in its performance as a refrigerant. This refrigerant offers a close match to R22 in existing equipment with respect to energy efficiency.

Cabello *et al.* [2] performed an analysis of a single-stage vapor compression plant using three different working fluids, R134a, R407C and R22, based on experimental results. Authors reported that the refrigerant mass flow rate has the highest influence on the refrigerating capacity and the plant COP. Hellmann *et al.* [3] presented a comparison of the performance of a refrigeration plant for low evaporating temperature using R410A and R407C without changing any of the plant components. Authors found that the overall energetic performance of R22 is consistently better than that of R407C. A comparative performance study was done by [4], and reported a similar trend with [3]. Authors concluded that R407C can be used as a substitute for R22 in air-conditioning and heat pumps due to its similar vapor pressure curve with R22.

Bansal *et al.* [5] developed a simulation model to estimate the performance of alternative refrigerants in a vapor compression refrigeration heat pump system. The simulation model analyzes the heat transfer behavior of brazed plate heat exchangers by using the NTU-ε method followed by elemental approach separately for single phase

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and two-phase heat transfer regimes in the heat exchangers. This model was tested for pure refrigerants. Another simulation program was developed by [6] for a geothermal heat pump system (GHPS) operating with a Non-Azeotropic Refrigerant Mixture (NARM). The program was validated with experimental results for the NARM, R123/R290 (50/50 by mass %) and concluded that the model could predict the performance of GHPS with a variation of ±12% with the experimental data.

It may be mentioned that the major problems arise when the working-fluid substitution needs to be carried out in existing plants. The only short-term drop-in substitute for R22 is R407C because its characteristics are sufficiently similar to that of R22, to allow immediate replacement in plants designed on the basis of the thermodynamic and transport properties. Its compatibility with plant materials is good, except for the lubricating oil. It may be mentioned that R407C is the only refrigerant available for immediate use in existing R22 plants which has zero ODP and a GWP of 1,700. The use of this refrigerant is increasing, although there are still some engineering difficulties for service companies and manufacturers. The majority of previous studies regarding the replacement of R22 were focused on heat pumps with low evaporating and low condensing temperatures. The objective of this study is to investigate the effects of the high evaporating temperature on the performance of a heat pump using R407C as refrigerant under various operating conditions. A water-to-water heat pump using R407C is tested in steady state, with heating and cooling modes. The performance of the R407C system is compared with the baseline refrigerant R22.

2. THEORETICAL ANALYSIS

Computer simulations have been carried out to compare the thermal performance of the refrigerants R22 and R407C. The two most important parameters for the study of a heat pump cycle are:

1) Coefficient of Performance (COP) $= \dfrac{Q_C}{W_C}$ (1)

2) Volumetric Heating Capacity (VHC) $= \dfrac{Q_C}{\dot{V}_S} \; \left(\dfrac{MJ}{m^3}\right)$ (2)

where Q_C is the heating capacity, W_C is the mechanical work input to the compressor, $\dot{V}_S$ is the swept volume of the compressor.

For an efficient heat pump, both the COP and the VHC should be as high as possible. The high COP minimizes the running cost while the high VHC minimizes the investment cost. The effects of temperature and pressure on the COP, the VHC, the heating capacity and power consumption have been studied for the cycle. The computations have been carried out using the characteristics of a semi-hermetic reciprocating compressor with a constant swept volume of 12.9m^3/hr at 1450 rpm. The volumetric efficiency of the compressor is assumed to be 100% for the simplicity of comparison between these two refrigerants. Although the isentropic efficiency is a function of pressure ratio, an average value of 0.7 is taken for this study.

The refrigerant is considered to be superheated by 5 K at the suction of the compressor but without any subcooling at the condenser outlet. For the refrigerant mixtures, the average temperatures of the evaporator and the condenser are used in all computations for comparison with the pure refrigerant. The pressure drops in the system are neglected. Comparison between these two refrigerants has been carried out at same condensation and evaporation temperatures because of the lack of available heat transfer properties over the whole range of temperature.

It can be stated that the practical limitations on heat pump operating temperatures are imposed by both the refrigerant properties, and the operating pressure of the compressor. The comparison of saturation pressures over the temperature range of interest is shown in Figure 1. The graph shows that R22 and R407C exhibit a very close saturation pressures over the expected operating temperature range. It may be stated that the R407C and R22 are limited to only low to medium temperature application range of 50-70°C at 2 to 3 MPa pressure.

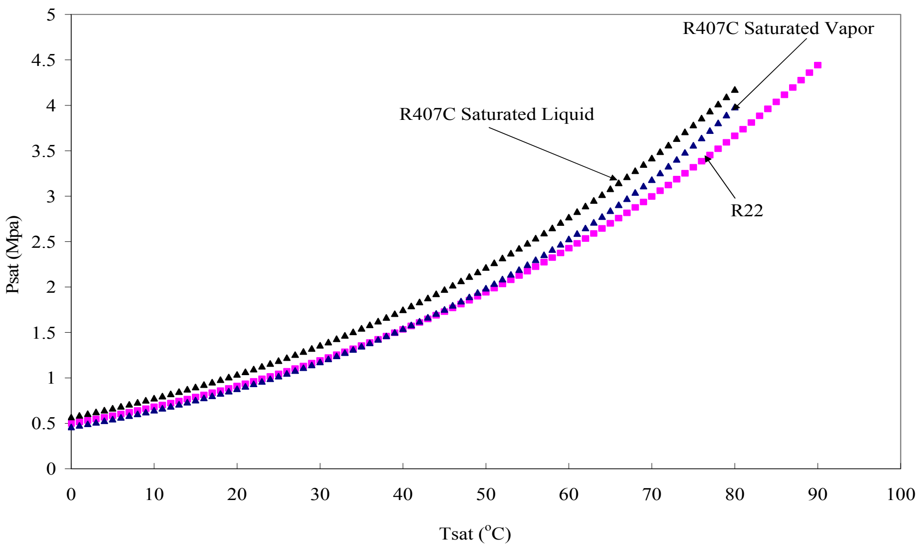

Fig. 1. Saturation pressure vs. saturation temperature for different refrigerants.

The three important characteristics of a heat pump cycle, namely COP, VHC and the pressure ratio are shown in Figures 2, 3, and 4 respectively as a function of ΔT (the temperature difference between the condenser and the evaporator) for two evaporating temperatures (T_{EV} =10°C and 30°C). It may be seen from the Figure 2 that R22 provides higher COP for both of the evaporating temperatures. The COP decreases with the increase in condensing temperature and is lower for lower evaporating temperatures. The same phenomenon is true for VHC, as has been shown in Figure 3. The VHC of R22 is higher compared to the other refrigerant for both of the evaporating temperatures. The VHC variation for the two refrigerants shows a decreasing trend with increasing ΔT. Subsequently, the low VHC values of R407C suggest that for the same condensing temperature, a high swept volume of the compressor is required.

The variation in pressure ratio at a particular value of ΔT (as shown in Figure 4) is very small which proves the hypothesis that the isentropic efficiency may be taken constant for the two refrigerants for the same ΔT. It may also be seen in Figure 4 that the pressure ratio increases with increasing ΔT for both of the refrigerants.

Theoretical cycle performance obtained from the thermodynamic properties of the refrigerants is shown in Table 1.

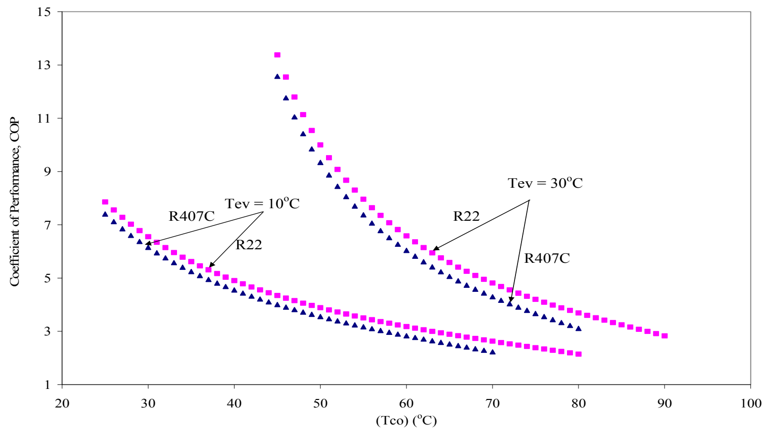

Fig. 2. Variation of COP vs. (T_{co}-T_{ev}) for evaporating temperature, T_{ev}=10°C and T_{ev}=30°C.

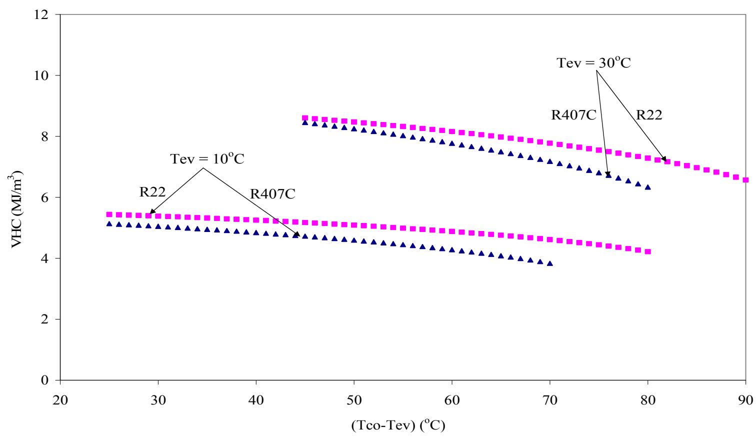

Fig. 3. VHC vs. (T_{CO}-T_{Ev}) for evaporating temperature, T_{Ev}=10°C and T_{Ev}=30°C.

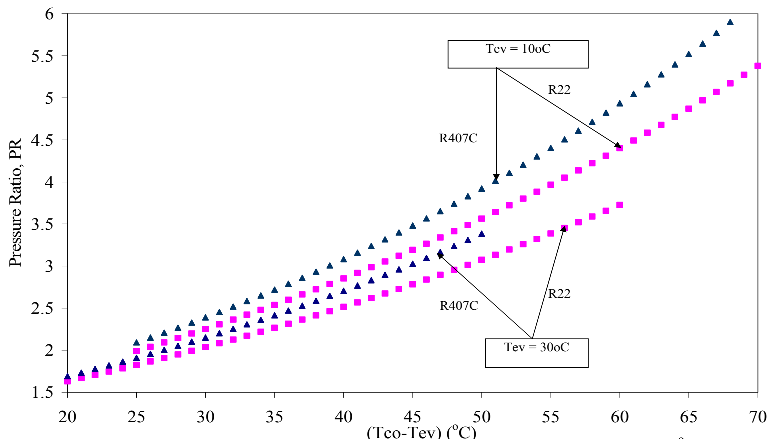

Fig. 4. Pressure ratio vs. (T_{con}–T_{evap}) for evaporating temperature T_{ev}=10°C and T_{ev}=30°C.

Table 1. Theoretical cycle performance calculation conditions: condensing temperature=40°C, evaporating temperature=10°C, subcools 3K, superheat 10K.

Refrigerant		R407C	R22
Composition		HFC-32/125/134a (23/25/52wt %)	HCFC-22
Cooling	Capacity ratio (%)	104.1	100.0
	COP ratio (%)	98.2	100.0
Heating	Capacity ratio (%)	104.3	100.0
	COP ratio (%)	98.0	100.0
Discharge pressure (MPa)		1.65	1.53
Suction pressure (MPa)		0.70	0.68
Temperature glide (K)		6.5	-

The refrigeration cycle performance was calculated assuming the same condensing and evaporating temperature, and compressor capacity for both of the refrigerants. It can be stated that R407C has temperature glides in condensation and evaporation processes due to its zeotropic characteristics. So, it may be mentioned that the average of dew point and boiling point temperatures were used as condensation temperature, and the average of evaporator inlet temperature and dew point temperature were used as the evaporating temperature. The thermodynamic properties from the database REFPROP published by NIST [7] were used. From the Table 2, it is found that R407C has nearly the same suction pressure as R22, but its discharge pressure is about 8% higher. The COP of R407C is lower in cooling and heating operations than that of R22 for this range of application. It is also found that the capacity of R407C is 4% higher than R22.

3. EXPERIMENTAL EQUIPMENT AND PROCEDURE

The test rig has been designed and installed in the thermodynamics laboratory at the University Malaya, consisted of three loops: the refrigerant loop, the condenser water loop and the evaporator water loop. The experimental equipment is shown schematically in Figure 5. It consisted mainly of a 3-phase × 1.7 kW compressor supplying R-134a refrigerant to a plate-type heat exchanger as a condenser and another plate-type heat exchanger as an evaporator. The condenser was cooled using a once-through flow of cold water at ambient temperature. Heat to the evaporator was supplied by circulating hot water from a hot water storage tank. The water in the tank was heated using 3 × 2.5 kW immersion heating elements. Temperature in the tank was controlled using a bimetallic thermostat.

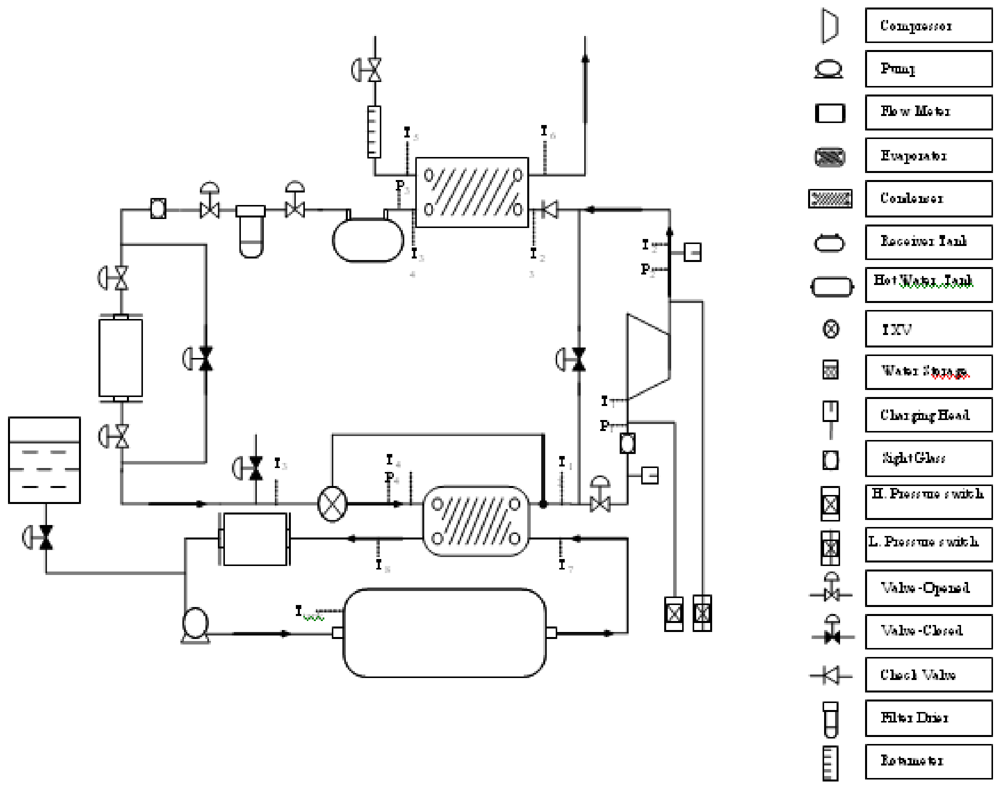

Fig. 5. Schematic diagram of the experimental apparatus.

Two Schlumberger coriolis-type mass flowmeters were employed; one connected to the liquid line after the receiver measured liquid refrigerant flow rate, the other flowmeter measured the water mass flow rate to the evaporator. The condenser water mass flow rate was measured with a manually-read Rotameter. Temperatures were monitored at the selected locations using Cu-Con thermocouples according to ASHRAE Standard 41.1 [8], and refrigerant pressures were also measured using electronic pressure transducer according to ASHRAE Standard 41.3 [9]. The mass flow meter was installed between the condenser and expansion device with a pressure drop of approximately 12 kPa, which was less than 82.7 kPa allowed in ASHRAE Standard 116 [10] at full charge condition. The power consumption was measured with a Dranetz Power Platform. Dynamic data logging was carried out using a SR630 logger and LABVIEW software.

Different experimental conditions were maintained by varying the system operating conditions as presented in Table 2. Experiments were carried out at different heat transfer fluids. For each set of run, the water flow rate in both condenser and evaporator are kept constant. The water inlet temperature at the evaporator was varied by using the electrical heater in the hot water tank (T_{tank}). The refrigerant pressure in the evaporator was thus varied, whereas the condensing pressure was kept almost constant. The compressor ratio was varied as well. The test data were recorded continuously for 10 min with 20-s intervals and repeated for different variables. The test rig can be operated in heat pump, refrigeration and heat pump chiller modes. Different condensing and evaporating temperatures could be achieved by controlling the mass flow rate and inlet temperature of heat transfer fluid at the condenser and evaporator.

Table 2. Experimental test conditions.

Fluid	Water inlet to the evaporator T (˚C)	Evaporator water mass flow rate $m_{w,e}$ (kg/s)	Condenser water mass flow rate $m_{w,c}$ (kg/s), with T=29˚C
R22	17	0.100, 0.150, 0.200, 0.246, 0.256	0.08, 0.17, 0.25, 0.33
	28	0.100, 0.150, 0.200, 0.246, 0.256	0.08, 0.17, 0.25, 0.33
	35	0.100, 0.150, 0.200, 0.246, 0.256	0.08, 0.17, 0.25, 0.33
R407C	17	0.100, 0.150, 0.200, 0.246,0.256	0.08, 0.17, 0.25, 0.33
	28	0.100, 0.150, 0.200, 0.246,0.256	0.08, 0.17, 0.25, 0.33
	35	0.100, 0.150, 0.200, 0.246, 0.256	0.08, 0.17, 0.25, 0.33

4. RESULTS AND DISCUSSION

The heating capacity was calculated and shown in Figure 6. The uncertainties of cooling, heating and COP estimated by the single-sample analysis according to ASHRAE guideline 2 [11] were approximately 4.1% and 4.2% respectively. The effect of presence of lubricant in the refrigerant was neglected.

The coefficient of performance of the heat pump (experimentally evaluated) is calculated as the ratio between heating capacity and electrical power supplied to the compressor.

$$COP_{HP} = \frac{\dot{Q}_C}{\text{Electrical energy supplied to compressor (KW)}} \tag{3}$$

where Q_c is equal to:

$$\dot{Q}_c = \dot{m}_{wc}.c_p.(T_{wc,out} - T_{wc,in}) \tag{4}$$

The effect of pressure ratio on heating capacity is shown in Figure 6. The variation of the heating capacity is mainly due to the changes in refrigerant mass flow rate. The refrigerant mass flow rate decreases when the compression ratio increases. At the same compression ratio, the heating effect of R407C is slightly higher than that of R22. If the capacity of an alternative refrigerant deviates too much from that of the reference fluid, the compressor must be redesigned completely which would be expensive. Therefore, it would be good for the alternative refrigerants to provide a similar capacity to that of the reference fluid. The capacity of the refrigerant R407C is 5% higher than that of R22. Since its COP is slightly lower than R22, this mixture seems to be good alternative for R22 for this application.

Figure 7 shows the electrical power consumption of compressor as a function of the compression ratio. The figure shows that for the same compression ratio and for the tested range of temperature, the electrical power consumption by R407C is around 3-10% more than that of R22.

Figure 8 shows the coefficient of performance as a function of the pressure ratio for R22 and R407C. When the compression ratio exceeds 3, with the high evaporating temperature, the coefficient of performance of R407C is greater than that of R22. A refrigerant with a higher saturation pressure has a higher capacity due to low specific volume at the compressor inlet, which results in a higher mass flow rate. Thus, the heating capacity increases with the increasing evaporating temperature. It has to be mentioned the fact that the refrigeration plant working with R22 shows a higher COP than R407C for low compression ratios. These results agreed with the results presented by [2] and [3]. With reference to the electrical power consumption, the refrigeration power consumption with R22 tends to decrease gradually with the increasing compression ratios than other working fluids. The COP also decreases with the increases of compression ratio for R22 [2].

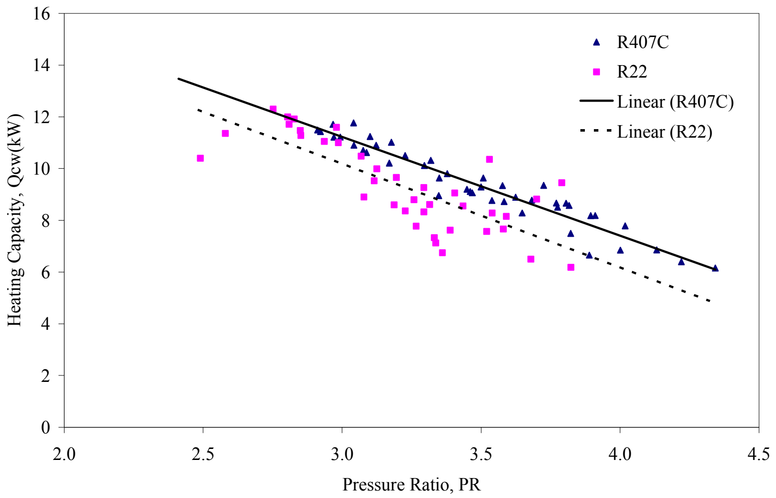

Fig. 6. Heating capacity vs. pressure ratio.

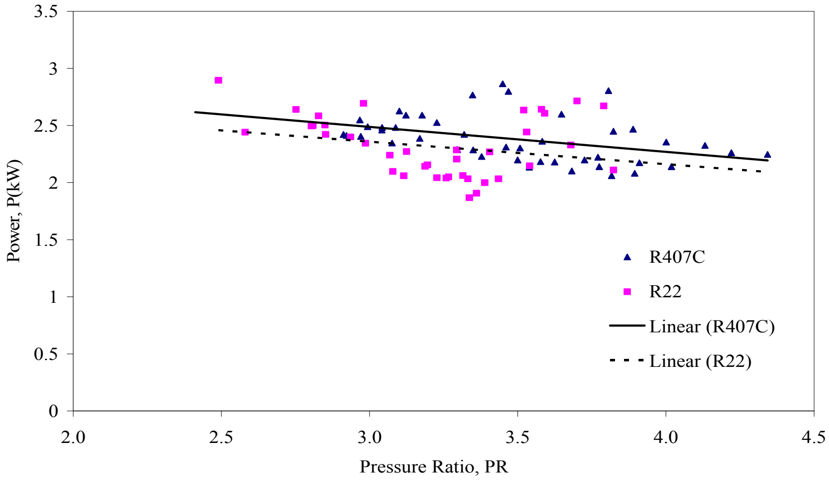

Fig. 7. Electrical power consumption vs. pressure ratio.

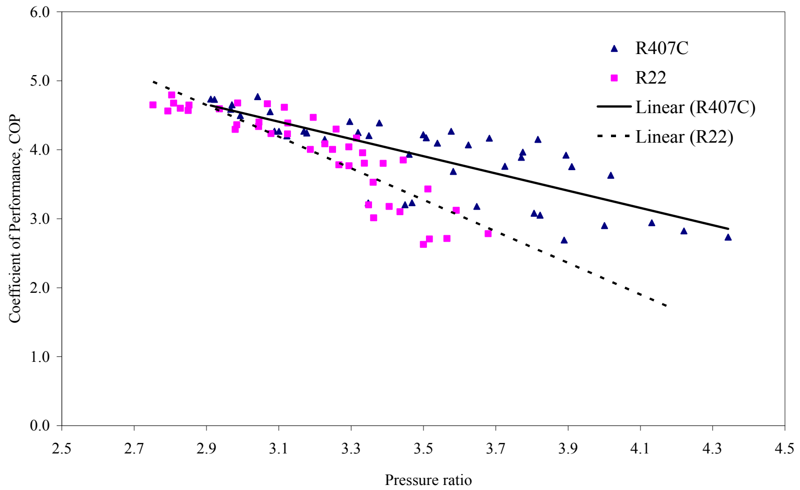

Fig. 8. COP of heating vs. pressure ratio.

Figures 9, 10, 11 show the predicted and experimental results for the COP, pressure ratio and VHC, respectively. The measured COP shows a good agreement with predicted one while the pressure ratio is less than the predicted one because the pressure drops in the system was neglected. The predicted VHC was higher than measured value because the volumetric efficiency of the compressor is assumed to be 100%. From Figure 11, it has been observed that R22 offers higher volumetric capacity than R407C. The volumetric capacity of the refrigerants decreases with the evaporating temperature and condensing temperature.

So far, a general approach for the simulation of a heat pump is presented. With the aid of the computer program developed in this study, one can do a parametric study systematically and narrow down good refrigerants candidates. The method of specifying equal heat transfer area needs to be considered to provide more accurate data for the use of specific refrigerants (mixtures) as mentioned in [12]. In order to perform this kind of analysis, however, one need to know the evaporation, condensation, single-phase (both liquid and vapor) heat transfer coefficients as well as pressure drop of the mixture.

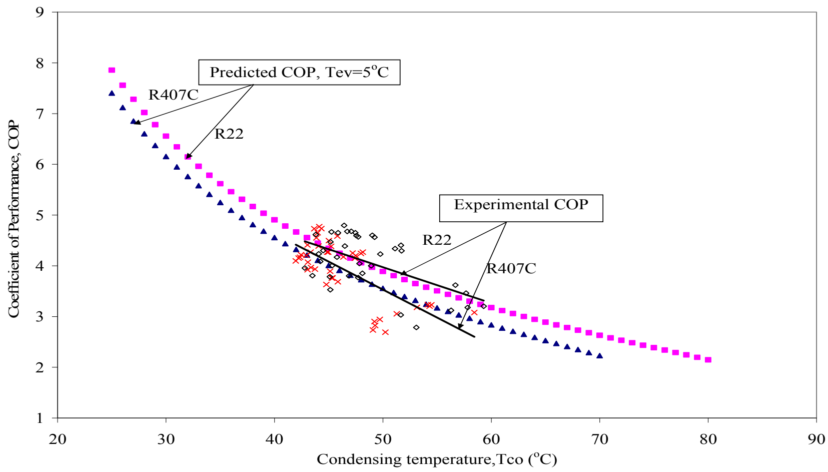

Fig. 9. Predicted and measured COP vs. (T_{co}) for evaporating temperature T_{ev}=5°C

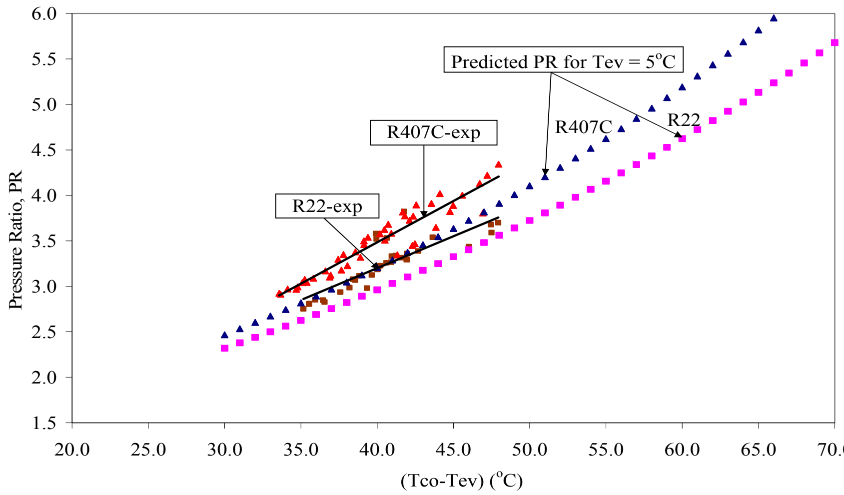

Fig. 10. Predicted and measured pressure ratio vs (T_{con}-T_{evap}) for T_{ev}=5°C.

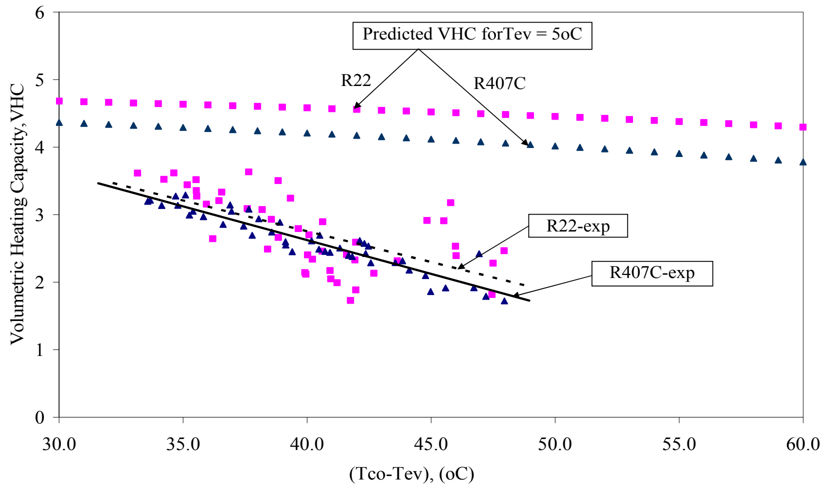

Fig. 11. Predicted and measured VHC vs. (T_{co}-T_{ev}) for T_{ev}=5°C.

5. CONCLUSION

In this study the refrigerant R22 and its substitute R407C were compared by thermodynamic analysis and their performances were compared in a vapour compression refrigeration plant using a semi-hermetic compressor. The capacity of the refrigerant R407C is found to be 5% higher than that of R22. When the compression ratio exceeds 3, with the high evaporating temperature, the coefficient of performance of R407C is greater than that of R22. The present study can be useful to the design engineers to improve specific aspects of the refrigeration and heat pump system when using R407C as a refrigerant.

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