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Evaluation of a Refrigerant R410A as Substitute for R22 in Window Air-conditioner

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ABSTRACT : *CFCs have been phased out, except for essential users, and HCFCs are to be eliminated by 2020, because of their ozone depletion potential. This generates a need for the investigation of zero ozone depletion potential (ODP) refrigerants or refrigerant blends. R410A is among newer brand of refrigerant blends, with zero ODP. The biggest difference to R22 is the pressure levels generated which are more than 50% higher. The refrigerant R410A operates at higher pressure at the same saturated temperatures than R22, therefore system should be redesigned. The early laboratory trials of R410A in air conditioning systems have showed a significant increase in COP vs. R22. The apparent anomalous behavior of R410A has been shown to be due to its very favorable physical and transport properties. The overall COP of the system is 5 to 6% more than the R22.*

Keywords – *R410A, R22, Micro channel, Condenser, Evaporator, Rating chart.*

I. INTRODUCTION

The depletion of the ozone layer due to the release of chlorine from CFC and HCFC refrigerants has raised serious concerns about using them in vapor compression systems. Therefore according to the amended version of the Montreal protocol, CFCs were phased out by January 1996, except for essential users, and HCFCs are to be phased out by 2020. Hence refrigerants or refrigerant blends with properties similar to CFCs and HCFCs and with zero ozone depletion potential (ODP) must be discovered to be used as replacements in existing systems. Two approaches are used to review the alternatives for R22. The first was to develop a substitute product with similar characteristics to R22. The refrigerant R407C has been accepted worldwide as the replacement with overall attributes best resembling R22. The second approach was to develop a substitute refrigerant, which would give the best performance when applied to the redesigned equipment, which traditionally uses R22. R410A is an azeotropic mixture of equal proportion by mass of HFC refrigerants R32 (CH_2F_2, difluoromethane) and R125 (HF_2C-CF_3, pentafluoroethane) with properties similar to those of R22. R410A has a higher volumetric cooling capacity compared to R22 and has better thermal exchange properties. This results in overall performance gains in terms of system efficiency. The higher density of the vapor in R410A permits higher system velocities, reduces pressure drop losses and allows smaller diameter tubing to be used. In turn a smaller unit can be developed using a smaller displacement compressor, less coil and less refrigerant while maintaining system efficiencies comparable to current day R22 equipment. Therefore we have a low cost solution to meet specific equipment requirements.

II. PROPERTIES OF R410A

2.1 Physical Properties

R410A is a blended refrigerant using R32 and R125 in an equal mix. It is a near azeotropic blend with a negligible glide (0.1%). As a HFC refrigerant, R410A requires the use of polyoester oils (POE). It is non-flammable and non-toxic. But the biggest difference to R22 is the pressure levels generated which are more than 50% higher. Although operating pressures of R410A are significantly higher than those of R22, the R410A system actually runs slightly cooler than a comparable R22 system due to the higher vapour heat capacity of the refrigerant.

2.2 Compatibility with oil

R410A is a blend of HFC refrigerants. The great majority of systems using HFC refrigerants contain polyoester oils (POE). POE oils are required because other oils, like mineral oil, are not miscible with HFC refrigerants. Miscibility is a measure of the ability of a liquid refrigerant to mix with the oil. Care should be taken to avoid exposure of the POE oil to air as it readily absorbs moisture (often referred to as being highly hydroscopic).

Table 1 Comparison of R22 and R410A properties.

Properties	**Units**	**R22**	**R410A**
Components	-	$CHCIF_2$	R32 R125
Composition	% weight	-	50/50
Molecular Weight	g/mol	86.5	72.6
Bubble Temperature (at 1.013 bar)	°C	-40.7	-52.2
Temperature Glide (at 1.013 bar)	K	0	0.1
Liquid Density (at 25°C)	Kg/dm^3	1.194	1.0615
Density of Saturated Vapour (at boiling point)	Kg/m^3	4.70	4.12
Vapour Pressure at: 25°c 50°c	bar bar	10.4 19.4	16.4 30.5
Critical Temperature	°C	96	72.2
Critical Pressure	bar	49.8	49.5
Critical Density	Kg/dm^3	0.525	0.491
ODP	-	0.055	0

III. PERFORMANCE ASSESSMENT

The HFC options for new air conditioning equipment have, by general consensus, been reduced to R407C, R134a and R410A. For refrigeration applications R410A is preferred because it has better low temperature efficiency, and lower discharge temperatures, but in air conditioning R407C has a better efficiency characteristic. A theoretical COP can be calculated for comparison purposes by using the thermodynamic properties of the refrigerant. Fig. 1 shows a comparison of the major refrigerants on this basis; a standard vapour compression cycle with a 100% efficient compressor is taken to make the calculations. This enables comparison of the efficiency effect of the thermodynamic properties of the refrigerant to be made. It is seen from the diagram that none of the replacement HFCs matches R22 in this respect, although R134a comes close. The second point to notice is that R410A is less efficient than R407C. The reason for this is relatively low critical temperature, 71°C, of R410A.

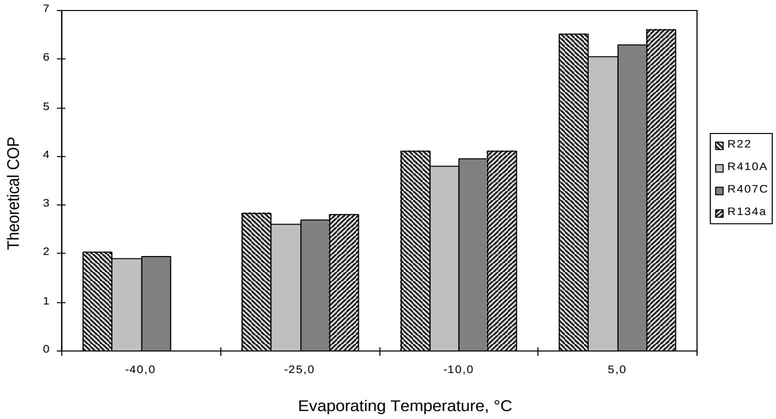

Fig. 1 Comparison of theoretical COP for refrigerants
(Condensing temperature 40°C, suction superheat 20K, zero subcooling.)

There are several other vital properties of a refrigerant which contribute towards the overall system behavior. R410A has a pressure considerably above that of R22, which should tend to reduce system cost. However it has taken time for proven high pressure components suitable for R410A to become available. This combined with the fact that the theoretical COP is poorer, has lead to the extensive adoption of R407C as an R22 replacement. A further benefit of high pressures is that there is a reduction in the effect of pressure drops. This can either result in smaller tubing for equivalent pressure drop effect, or lower losses if the same size tube is used. In order to appreciate why R410A has the potential for improvements over R22 and R407C, it is necessary to consider the relative effects on parameters round the system. Many studies have taken place for evaluating R22 alternatives for both residential and commercial air conditioning applications and the results of those studies is summarized with in Fig 2. The nominal operating conditions for the tests were: evaporating temperature, 7 °C, condensing temperature 40 °C with 11K superheat and 8.3K subcooling. The first parameter is the theoretical efficiency, and this is approximately 4% lower than R22, and is shown as a negative in Fig 2.

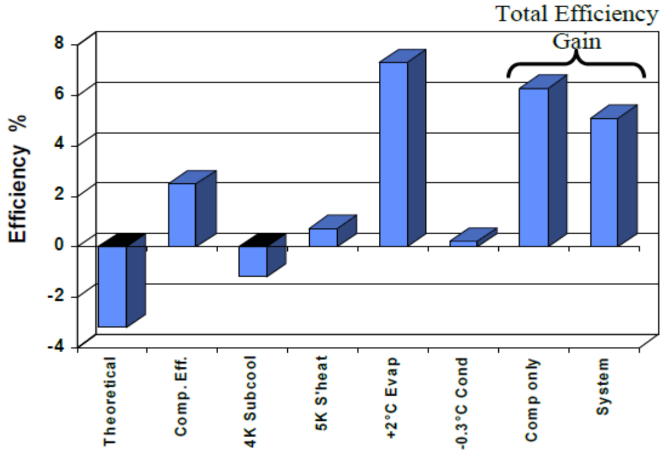

Fig. 2 Percentage efficiency effects for R410a, reference R22.

Compressor testing has shown that there can be a gain of up to 2% in compressor efficiency in the R410A system. This is shown as a positive in Fig 2 and goes some way towards offsetting the negative refrigerant properties effect, although it should still be noted that the COP of the R410A scroll will generally be slightly below that of the R22 equivalent, as shown in Fig. 1. The compressor COP is a combination of the compressor efficiency and the refrigerant properties. Now we move on to other system parameters. The superheat and subcooling will have a small effect as shown, due to refrigerant properties. By far the largest effect is the major gain in performance due to better heat transfer in the evaporator. This gain has the effect of raising the evaporating temperature by 2K. For the same air temperatures, the increased evaporating temperature with the R410A system improves system efficiency and capacity by a significant amount. There was also a small effect due to improved heat transfer in the condenser. The overall COP percentage improvement is shown in Fig 2 as 6% when referenced to the compressor only, or 5% for the system, which takes account of the fan power.

IV. SYSTEM COMPONENTS

There are four important components for any refrigeration system which are as follows: Compressor, Condenser, Expansion device and Evaporator. This section includes a brief description of all the above components that should be used for R410A.

4.1 Compressor

The new model family of "ZP" compressors is designed for high pressure R410A refrigerant. R410A operates at approximately 50 to 70 percent higher pressure at the same saturated temperatures than R22. Several design changes had to be made to the scroll compressors to accommodate the operating differences between R22 and the new refrigerant. The new models include an internal pressure relief valve, an internal discharge gas temperature sensor, and a device to limit the shutdown noise caused by scroll reversal. The compressor shell thickness was increased to handle the higher required pressures. The "P" in the model number designates that this compressor has a housing designed for the higher pressure encountered with R410A. Use of a compressor that is not specifically designed for R410A may cause shell rupture and personal injury. R410A has greater enthalpy per unit volume than R22. For this reason the displacement is smaller vs. motor power in the "ZP" scroll than an equivalent capacity R22 compressor. Using R22 compressor in R410A system may cause the compressor to stall. Conversely using a "ZP" compressor in an R22 system would result in a drastic system capacity reduction. Fig. 3 shows a cross section of the “ZP” or R410A scroll, and the components which differ from the previous R22 model are indicated.

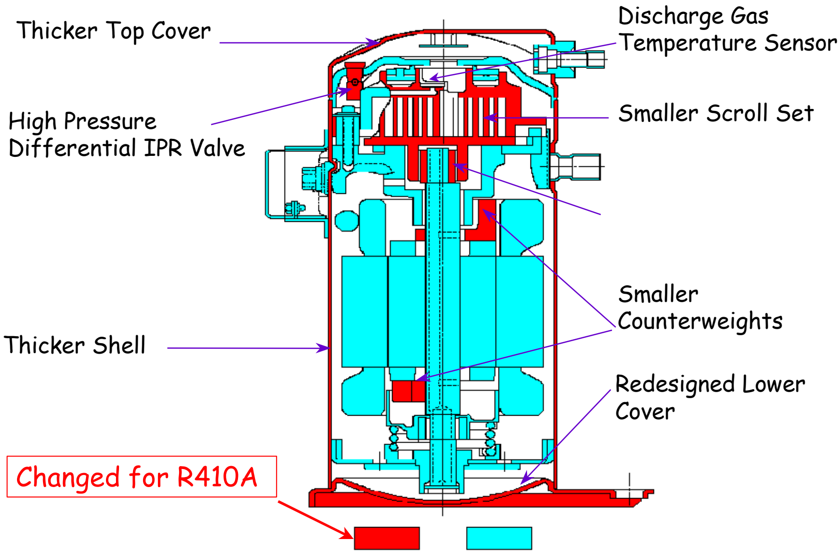

Fig. 3 ZP Scroll Compressor Showing Changes Made for R410A

4.2 Condenser

This section compares the effects of use of microchannel and round tube condenser on the performance of the air conditioning system. Heat exchangers with multi-ported microchannel tubes are already used in mobile air-conditioning systems due to their compactness and high performance. Besides the application of microchannel heat exchangers as gas coolers in CO_2 systems and as condensers in mobile air-conditioning systems, there is a possibility that microchannel heat exchangers can be used as condensers in a residential air-conditioning system instead of the widely used condensers with round tubes. It is widely known that microchannel heat exchangers have an obvious advantage over RTPF (Round Tubes and Plate Fins) heat exchangers in compactness. Consequently, it is inferred that microchannel heat exchangers have a larger capacity than RTPF heat exchangers for an identical heat exchanger package volume. Even though apparently better heat transfer characteristics of microchannel heat exchangers exist, microchannel heat exchangers are not commonly used in residential air-conditioning systems because RTPF heat exchangers have a cost advantage, which is one of the most critical factors of commercial products [1]. Fig. 4 shows schematics for the unfolded microchannel and round-tube condensers. The microchannel and round-tube condensers have almost identical shapes, volumes, face areas, and fin densities as shown in Fig. 4. The microchannel condenser has three passes with vertical headers. The first, second, and third passes have 44, 19, and 11 multi-ported microchannels, respectively. The round-tube condenser has two passes. The first pass consists of two long round tubes and each tube has 10 rows in the condenser. The supplied R410A to the round-tube condenser is divided into two flows; one is upward and the other is downward. The two flows in the first pass are merged at the inlet of the second pass which has a long tube with 10 rows.

The arrangement of the two condensers is presented schematically in Fig. 4. The microchannel condenser has folded louver fins and the round-tube condenser uses offset-strip fins. The fin spacing for the two condensers is identical at 1.06 mm. Fig. 5 (a) and (b) shows the folded louver and offset-strip fins, respectively, along with the micro-channel dimension.

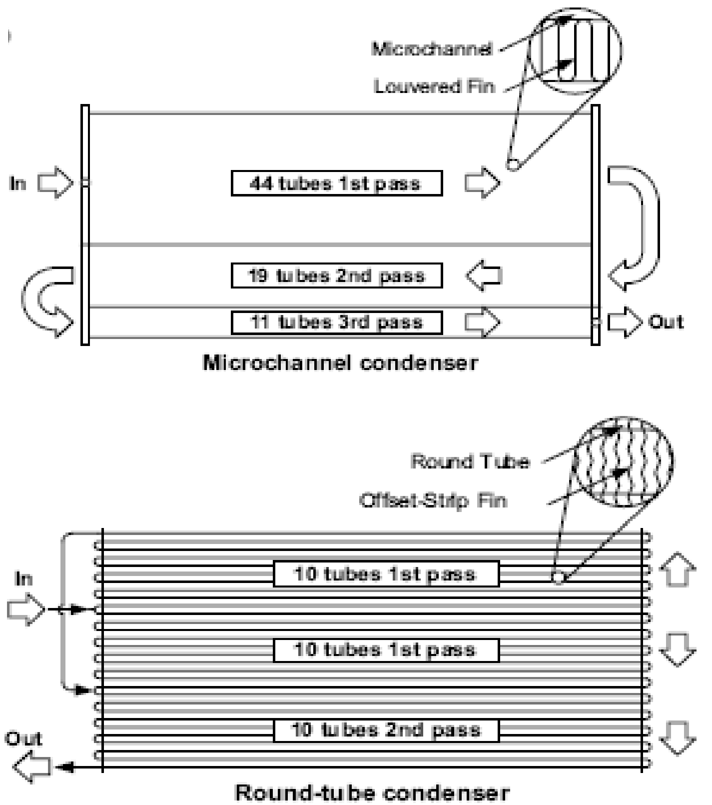

Fig. 4 Schematics of the unfolded microchannel and round-tube condensers

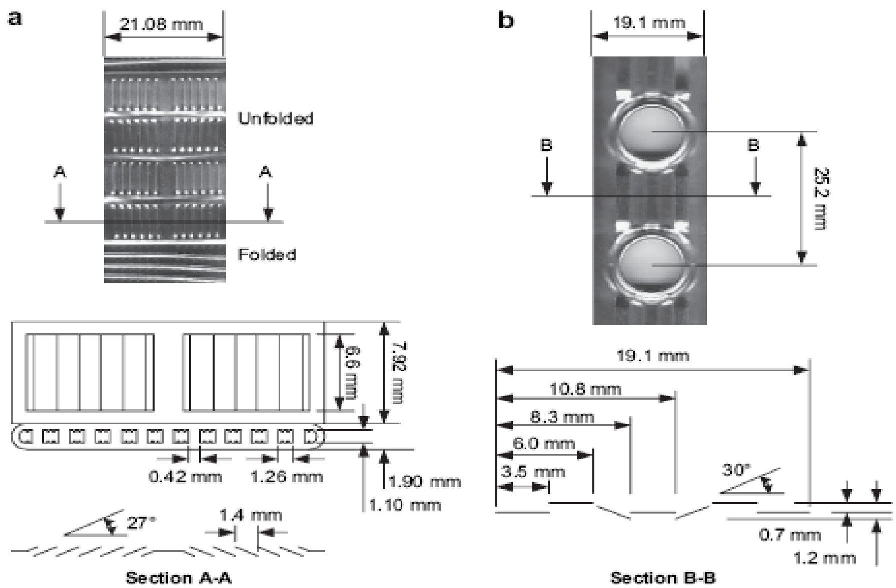

Fig. 5 (a) Louvered folded fins and micro-channel dimensions in the micro-channel condenser and (b) offset-strip plate fins in the round-tube condenser.

Two R410A residential air conditioning systems, one with a micro-channel condenser and the other with a round-tube condenser, were tested experimentally, while the other components of the two systems were identical except the condensers. Two condensers had almost same package volumes. The two systems were operated in separate environmental chambers and their performance was measured [1]. The performance of the system with a round-tube or micro-channel condenser was examined and compared in three standard conditions A, B, and C defined in the ARI (Air Conditioning and Refrigeration Institute) Standard (2003). The air temperatures for each standard test are presented in Table 2.

Table 2 Air entering conditions for standard rating conditions for an air-conditioning system (Source: ARI Standard, 2003) (Unit: 8°C)

Standard rating conditions	Indoor unit		Outdoor unit	
	Dry-bulb (°C)	Wet-bulb (°C)	Dry-bulb (°C)	Wet-bulb (°C)
A	26.7	19.4	35.0	23.9
B	26.7	19.4	27.8	18.3
C	26.7	13.9	27.8	18.3

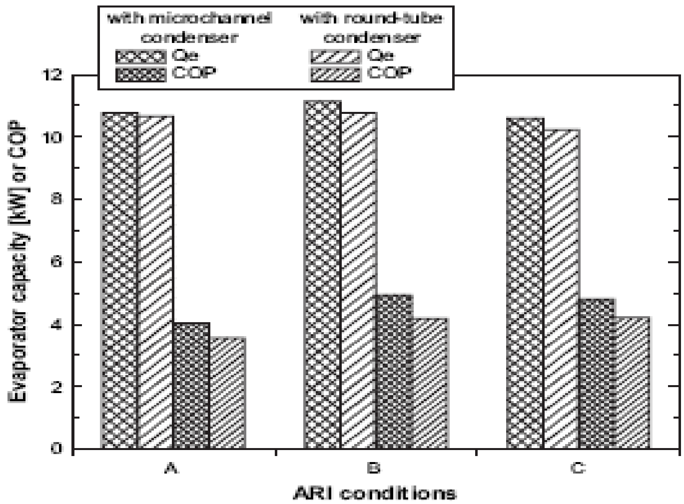

Fig. 6 Comparison of evaporator capacity and COP

Fig. 6 shows the comparison of evaporator capacity and COP (coefficient of performance) for the systems with the round-tube and microchannel condensers in three standard conditions. The evaporator capacity and COP of the system with the microchannel condenser were 3.4 and 13.1% higher, respectively, than those of the system with the round-tube condenser in ARI condition A. Further improvement in system performance using the microchannel condenser was seen in ARI conditions B and C. Also, using a microchannel condenser resulted in a 2.5°C lower condensing temperature and decreased the refrigerant pressure drop from 166 kPa in the round-tube condenser to 57 kPa in the microchannel condenser. The refrigerant charge amount for the system with the microchannel condenser was 9.2% smaller than that with the round-tube condenser. Even though the microchannel condenser showed better heat transfer performance of capacity and system COP than the round-tube condenser, the round-tube condenser has a cost advantage at this point in time.

4.3 Expansion device

A capillary tube has been widely used as an expansion device in small refrigeration and air-conditioning systems due to the advantages of simplicity, low cost, and the requirement of low starting torque of a compressor. The capillary tube size and system charge must be determined to have a compatibility with the compressor and heat exchangers that are redesigned to meet the required design conditions with alternative refrigerants. Therefore, a generalized correlation and rating charts for the prediction of the refrigerant flow rate through adiabatic capillary tubes with new alternative refrigerants must be developed to afford a convenient design tool.

4.3.1 Rating charts

Rating charts for predicting refrigerant mass flow rate through adiabatic capillary tubes are developed based on the present correlation. Two quantities used in the prediction of the refrigerant flow rate are the flow rate for a reference capillary tube (mr) and the geometric correction factor (Φ_1). The actual mass flow rate (ma) through the adiabatic capillary tube is determined by multiplying these two quantities as given by the following equation (1) [2].

$$ma = mr * \Phi_1 \tag{1}$$

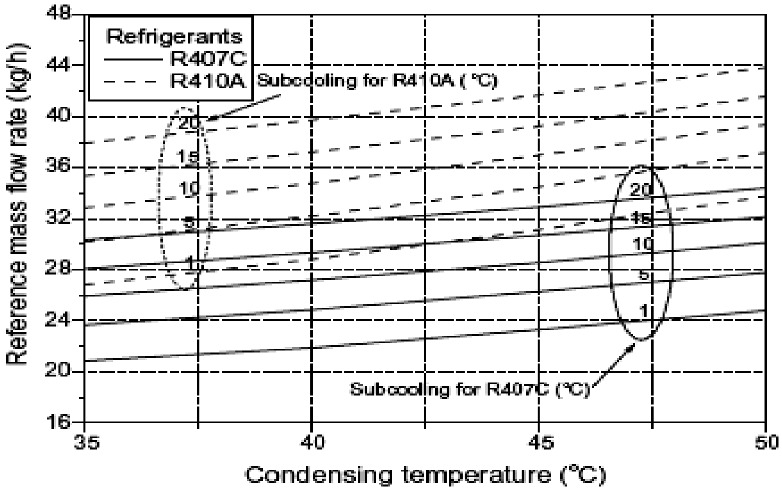

Fig. 7 Reference mass flow rates for R407C and R410A
(Reference capillary tube: L =1500 mm and D =1.21 mm)

Fig. 7 gives reference mass flow rates for R407C/R410A, as a function of condensing temperature and inlet subcooling for a reference capillary tube with a length of 1500 mm and a diameter of 1.21 mm. The x-axis of the charts is expressed in terms of condensing temperature instead of inlet pressure or condensing pressure to retain the same operating range of x-axis for various refrigerants.

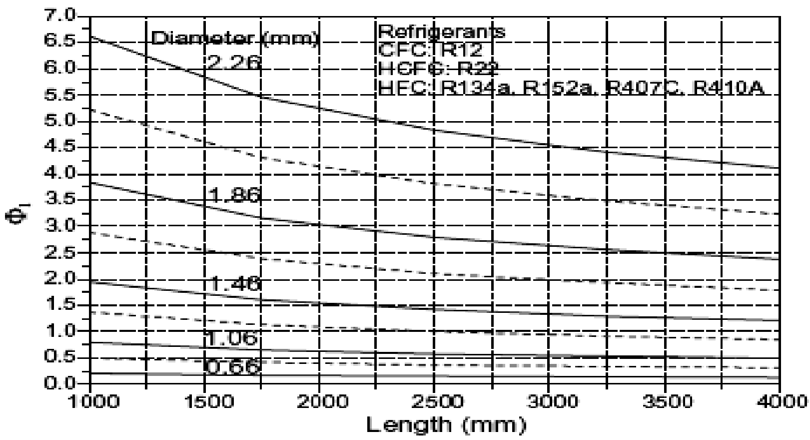

Fig. 8 Correction factor for capillary tube geometry.

Fig. 8 shows a geometric correction factor for various refrigerants. After determining the correction factor from Fig. 10 at a given capillary tube geometry, multiplying this factor with the reference mass flow from Figs. 9 for given inlet conditions yields the actual mass flow rate. The rating charts are only applicable for subcooled inlet conditions. It should be noted that the charts are constructed on the assumption of choking conditions. The accuracy of the charts cannot be guaranteed when the charts are extrapolated beyond the application limits of the correlation.

4.4 Evaporator

This section presents a comparable evaluation of R600a (isobutane), R290 (propane), R134a, R22, R410A, and R32 in an optimized finned-tube evaporator, and analyzes the impact of evaporator effects on the system coefficient of performance (COP). Table 3 shows the evaporator design data that was common for all evaporator simulations in this study. Additionally, the air condition was 26.7 °C dry-bulb temperature and 50% relative humidity. The refrigerant inlet condition was specified in terms of the saturation temperature and subcooling at the inlet to the distributor, which was included in the simulation runs. We used subcooling of 5.0 K in all simulations.

Table 3 Evaporator design information

Items	**Unit**	**Value**
Tube length	mm	500
Tube inside diameter	mm	9.2
Tube outside diameter	mm	10.0
Tube spacing	mm	25.4
Tube row spacing	mm	22.2
Number of tubes per row		12
Number of depth rows		3
Fin thickness	mm	0.2
Fin spacing	mm	2
Tube inner surface		Smooth
Fin geometry		Louver
Air volumetric flow rate	m3 min^{-1}	25.5

Table 4 Performance for the theoretical cycle and the cycle accounting for evaporator effects

Refrigerant	Basic theoretical cycle		Cycle including evaporator effects	
	Tsat (°C)	COP	Tsat (°C)	COP
38.0 °C Condensing temperature				
R600a	7.0	4.103	5.7	3.895
R134a	7.0	3.993	6.4	3.896
R290	7.0	3.929	7.7	4.036
R22	7.0	3.898	7.0	3.898
R410A	7.0	3.703	8.1	3.874
R32	7.0	3.701	8.5	3.926
45.0 °C Condensing temperature				
R600a	7.0	3.237	5.8	3.111
R134a	7.0	3.133	6.4	3.064
R290	7.0	3.074	7.8	3.155
R22	7.0	3.063	7.0	3.063
R410A	7.0	2.869	8.2	2.995
R32	7.0	2.878	8.5	3.073

The first simulation task was to obtain evaporator capacity for each refrigerant at the same exit saturation temperature of 7.0 °C**.** A simulation was performed at two condensing temperatures of 38.0 °C and 45.0 °C. Table 4 presents the obtained results for the two rounds of simulations. The results in the left-hand-side of the table, with T_{sat}=7.0 °C, are from the basic thermodynamic calculations of the cycle. The results located in the right-hand-side of the table, with different values of T_{sat}, account for the impact that the thermodynamic and transport properties have on the cycle through their effect on the performance of the optimized evaporator. Comparable theoretical evaluations of refrigerants in a vapor-compression cycle using thermodynamic properties alone tend to yield a better COP for low-pressure refrigerants (having a high critical temperature) versus high-pressure refrigerants (having a low critical temperature). This is due to smaller irreversibilities realized in a cycle at given evaporating and condensing temperatures when it operates far away from the refrigerant's critical point. The COP advantage shown by such theoretical evaluations for low-pressure refrigerants does not, however, account for the advantage high-pressure fluids have in optimized finned-tube heat exchangers. For the same cooling capacity, high-pressure refrigerants tend to have a higher saturation temperature at the evaporator exit than low-pressure refrigerants, which can compensate for the theoretical cycle inferiority high-pressure fluids may have [3].

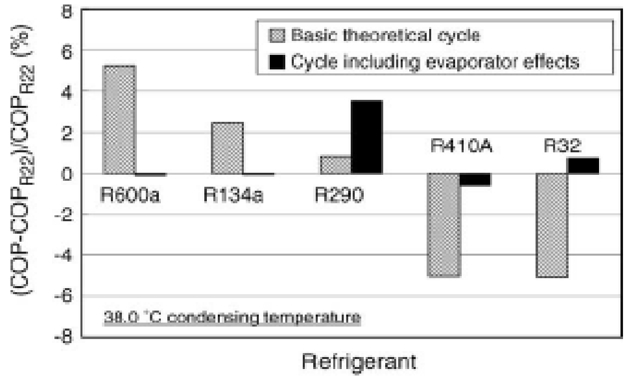

Fig. 9 (a) COPs compared to the COP of R22 for the basic cycle and for the cycle including evaporator effects for 38.0° C

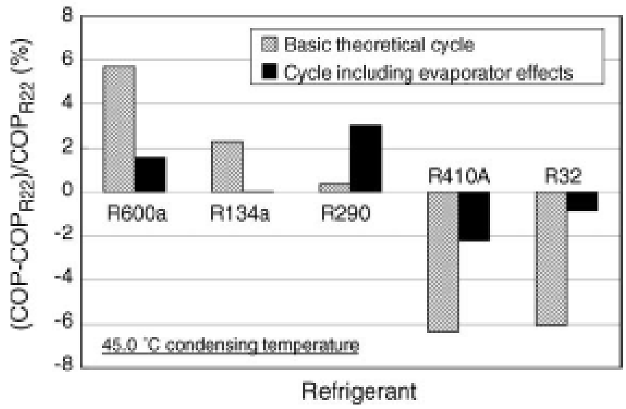

Fig. 9 (b) COPs compared to the COP of R22 for the basic cycle and for the cycle including evaporator effects for 45.0° C

Fig. 9 (a) and (b) presents COP results of R600a, R134a, R290, R22, R410A, and R32, referenced to the COP of the baseline R22 cycle. The high-pressure refrigerants provided higher evaporator capacities than the low pressure refrigerants. For a 7.0°C evaporator exit saturation temperature, and using R22 as a reference, R32, R410A, and R290, had a greater capacity by 14.5, 10.7, and 6.0%, while R134a and R600a had a lower capacity by 5.2 and 9.5%, respectively. The subsequent theoretical cycle simulations with the same 7.0°C evaporator saturation temperature showed the COPs of the studied refrigerants to be in the order of their critical temperatures, i.e. low pressure refrigerants had the best COPs. However, for the cycle simulations including evaporator effects (carried out at a different evaporator saturation temperature for each fluid to match the R22 capacity), the refrigerants performed within approximately a 2% band of the R22 COP baseline for the two condensing temperatures used [3].

V. CONCLUSION

R410A is refrigerant blend, with zero ozone depletion potential (ODP). It has a higher volumetric cooling capacity compared to R22 and has better thermal exchange properties. The overall COP of the system is 5 to 6% more than R22. R410A operates at approximately 50 to 70 percent higher pressure at the same saturated temperatures than R22, therefore system should be redesigned. A specially designed "ZP" scroll compressor must be used for R410A otherwise shell rupture may take place. The evaporator capacity and COP of the system with the microchannel condenser were 3.4 and 13.1% higher, respectively, than those of the system with the round-tube condenser. A microchannel condenser resulted in a 2.5°C lower condensing temperature and decreased the refrigerant pressure to 57 kPa, at the same time it required 9.2% lesser refrigerant charge. For a 7.0°C evaporator exit saturation temperature, R410A had a greater capacity by 10.7% than that of R22. The greater density of the vapour in R410A permits higher system velocities, reduces pressure drop losses and allows smaller diameter tubing to be used. In turn a smaller unit can be developed using a smaller displacement compressor, less coil and less refrigerant while maintaining system efficiencies comparable to current day R22 equipment. Therefore we have a low cost solution to meet specific equipment requirements.

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