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ABSTRACT

This paper will focus on the application issues of HFC replacements for HCFC-22 in air conditioning and heat pump systems. Issues such as energy efficiency, cooling and heating capacity, system design issues, and the total environmental impact of representative systems will be presented. This paper will focus on the use of HFC refrigerants since this allows the continued use of the infrastructure now in place to distribute, install, and service existing air conditioning and heat pump systems. If refrigerants with more restrictive safety classifications such as flammable refrigerants were to be used, changes to this infrastructure will be needed.

INTRODUCTION

In response to the growing concerns and scientific evidence of the thinner earth's ozone layer, an international agreement, the Montreal Protocol on Substances that Deplete the Ozone Layer, was signed in September 1987. Its provisions control the production and consumption of chlorofluorocarbons (CFCs) and hydrochlorofluorocarbons (HCFCs). CFCs such as R-12 were targeted first and their production ceased in 1996 in developed countries. The most current amendments of the Protocol in Copenhagen and Vienna called for a consumption "cap" or freeze of HCFCs production in 1996. This consumption "cap" value calculated from 1989 consumption data is scheduled to be reduced by 35% in 2004, with further cutbacks in later years until a 100% phase-out of all HCFCs occurs in 2030. HCFCs, such as R-22 that has been the primary refrigerant for air conditioning and refrigeration for many years, continues to be used today as a replacement for CFCs in many applications. Many countries will phase-out of R-22 in new equipment by the end of the next decade. The U.S. established January 1, 2010 as the date to ban the manufacturing of new equipment using R-22. The European Union countries will phase-out in the next several years and some countries have already eliminated new equipment using R-22.

The Air-conditioning and Refrigeration Institute, recognizing the need to prepare its member companies for replacement of R-22, established the Alternative Refrigerant Evaluation Program (AREP) in February, 1992[1] and became an international effort to identify and characterize the leading candidates to replace R-22. Three candidates emerged as the most likely replacement of R-22. They were a mixture of HFC-32 and 125[2] (now known as R-410A) that exhibited azeotropic behavior, a ternary zeotropic blend of HFC-32, 125, and 134a (R-407C), and a pure refrigerant R-134a. R-410A recently has emerged as the leading long-term choice to replace R-22 in new residential and light commercial equipment. R-407C is currently being used in regions of the world where an accelerated phase-out of R-22 does not allow sufficient time to redesign equipment for R-410A. R-134a use as an air conditioning refrigerant is restricted to larger chillers and mobile air-conditioning applications and is not used for unitary equipment. This paper will focus on the performance of R-410A and R-407C.

R-410A CHARACTERISTICS

R-410A is non-flammable and very low in toxicity (classified A1/A1 by ASHRAE). This binary mixture of HFC-32 and 125 exhibits azeotropic behavior, having negligible refrigerant component segregation and temperature glide characteristics. R-410A's vapor pressure is about 60% higher than R-22 making R-410A a choice mainly for new equipment. It is not easily adopted either for retrofitting existing equipment in the field or for equipment designed for R-22.

R-410A thermodynamic cycle efficiency is somewhat lower than R-22 as well as some other alternatives. However, its superior transport properties and other advantages results in superior performance at most conventional operating conditions. With a higher vapor pressure, system performance is less sensitive to pressure drop losses which allows for higher refrigerant velocities in heat exchangers resulting in additional heat transfer gains. It has also been demonstrated from compressor calorimeter studies that better isentropic efficiency compared to R-22 can be obtained at low and moderate condensing temperatures. These favorable characteristics of R-410A can impact the design of the heat exchangers, compressors, and line size allowing for better system energy efficiency.

R-410A consists of only hydrofluorocarbons (HFC) and as a result, mineral oil and alkylbenzene lubricants do not have sufficient solubility with the refrigerant to ensure reliable oil returns to the compressor. Polyol ester lubricants or other soluble lubricants are typically used. The combination of this HFC-based refrigerant and its paired lubricant must be considered together when evaluating and selecting elastomer or plastic materials. The refrigerant can also impact the selection of desiccant material in the filter-drier. These factors along with the performance of the heat exchangers and compressor all influence the design and cost of the system.

R-407C CHARACTERISTICS

R-407C also is non-flammable and very low in toxicity (classified A1/A1 by ASHRAE). This ternary mixture of HFC-32, 125, and 134a exhibits zeotropic behavior with potential refrigerant component segregation issues and temperature glide characteristics. R-407C's vapor pressure is only 10% higher than R-22 making R-407C a potential choice either for retrofitting existing equipment in the field or for equipment designed for R-22.

R-407C thermodynamic cycle efficiency is also somewhat lower than R-22. However, heat transfer studies of this refrigerant have revealed poorer heat transfer than R-22. With a similar vapor pressure, system performance is equally sensitive to pressure drop losses so no additional benefit is seen from higher refrigerant velocities in heat exchangers. As will be demonstrated in this paper, use of R-407C will allow systems that have been designed for R-22 to use a chlorine-free refrigerant, but will do so at a lower system efficiency.

HEAT TRANSFER AND PRESSURE DROP CHARACTERISTICS

Conducting a performance comparison of refrigerants based solely on thermodynamic cycle calculations or even compressor data alone assumes that the saturated suction temperature and saturated discharge temperature are identical for all the refrigerants evaluated. With changes in heat exchanger performance due to differences in the transport properties and pressure, this is not the case in real systems. In order to quantify these differences, information on the heat transfer and pressure drop performance of all refrigerants are necessary.

For R-410A data from three studies[3,4,5] on evaporative heat transfer and pressure drop and four studies[3,4,5,6] on condensing heat transfer and pressure drop was summarized. All these studies utilized 9.5mm (3/8") O.D. tubes without internal enhancements and covered similar ranges of mass fluxes. The data was averaged and is presented in Figures 1 and 2. In the case of R-407C there was insufficient data on smooth 9.5 mm tubes available so a different approach was taken. Relative performance of R-407C vs. R-22 smooth and enhanced, 6.5 though 15.9 mm tubes was utilized[6,7,8,9,10,11]. Average values of the heat transfer and pressure drop was used and is shown on Figures 1 and 2.

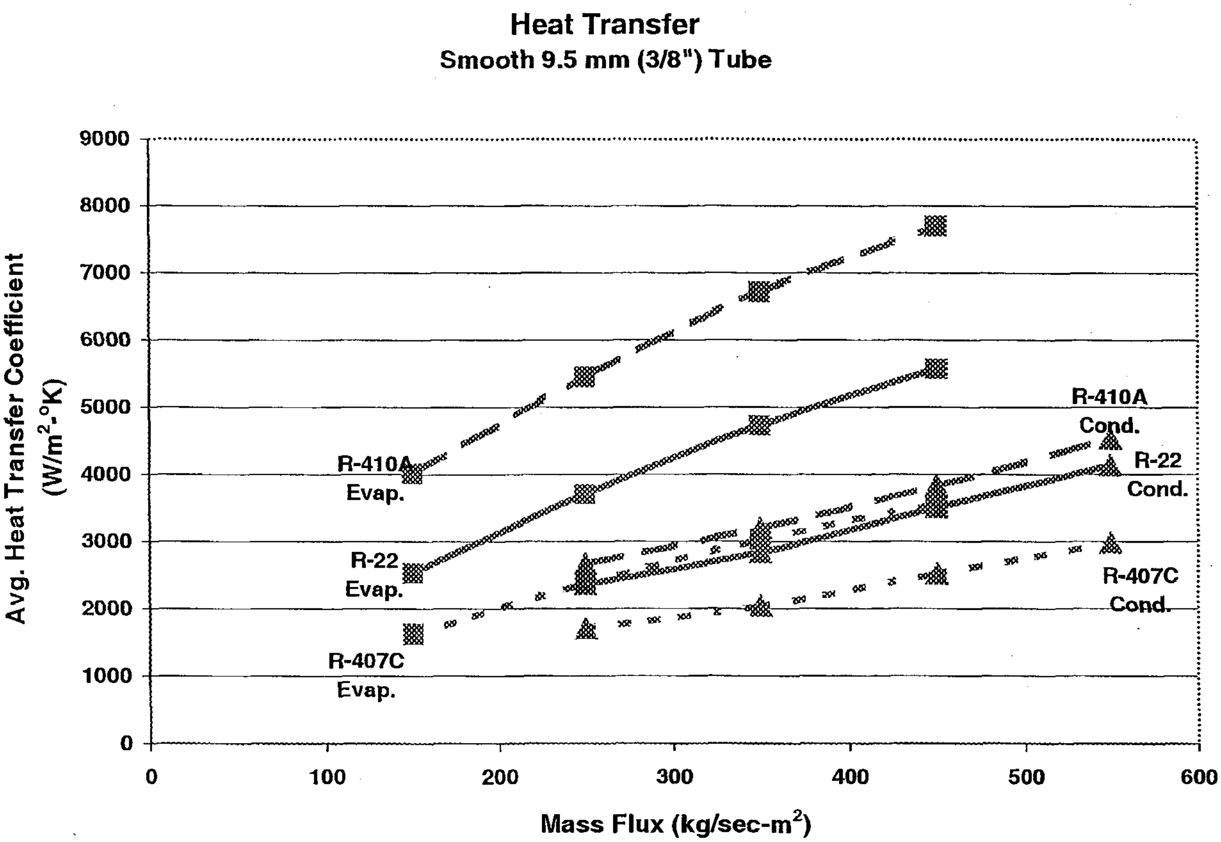

Figure 1

The pressure drop per equivalent length is less for R-410A than R-22 at a given mass flux. Using a representative evaporator mass flux for an R-22 system of 200 kg/sec-m^2, an increase to ~280 kg/sec-m^2 will result in the equivalent pressure drop for R-410A. Since the pressure is higher for R-410A, it will require a 50% increase in pressure drop to get the equivalent saturation temperature change as R-22. Therefore, the mass flux could increase to ~340 kg/sec-m^2 in an R-410A system and result in the same impact on saturation temperature as the original 200 kg/sec-m^2 did for R-22. Examining Figure 1 shows the impact of the higher mass fluxes on the

Pressure Drop
Smooth 9.5 mm (3/8") Tubes

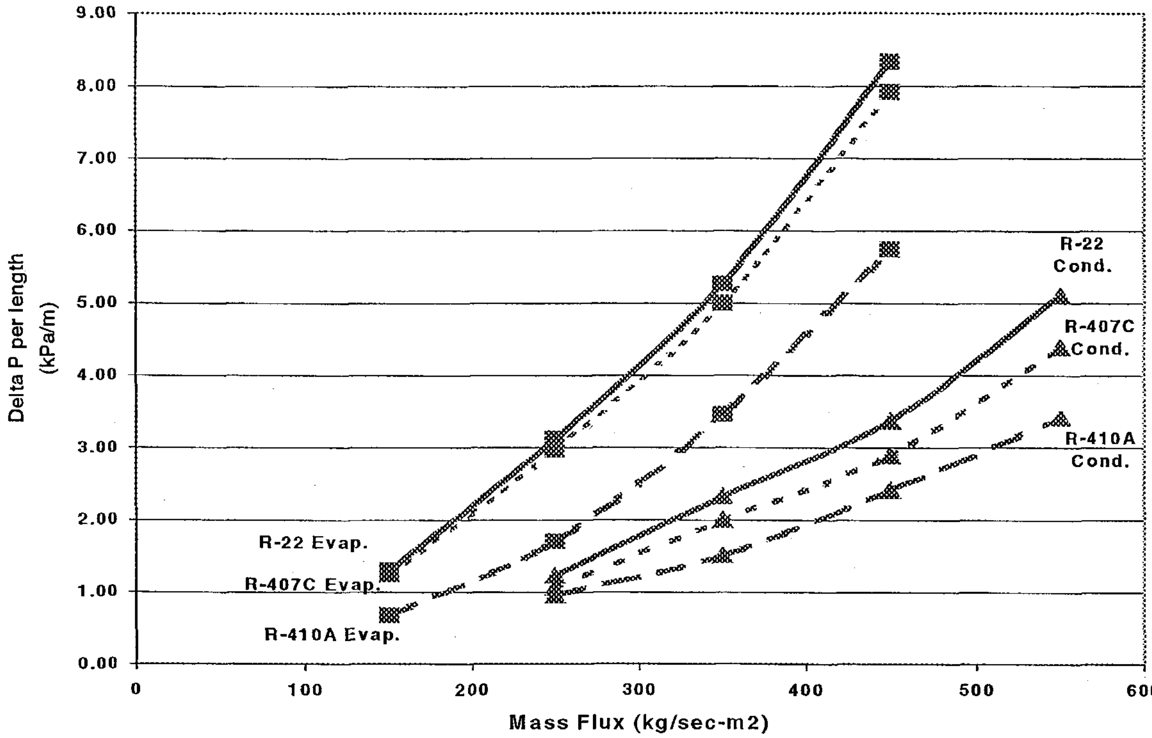

Figure 2

evaporation heat transfer coefficient. At the same mass flux there is an increase of 55% in the refrigerant side heat transfer. At the equivalent pressure drop there is a 90% increase in heat transfer and at the equivalent saturation temperature drop there is a 115% increase in heat transfer. Although the condensing heat transfer for R-410A is only slightly higher than R-22 at the same mass flux (~15%), at the mass flux associated with the equivalent pressure drop it is 35% higher and at the mass flux associated with the equivalent temperature drop it is 65% higher. The average heat transfer of R-407C is ~65% of R-22 for evaporation and ~72% for condensation. The pressure drop is 95% of R-22 for evaporation and 86% for condensation.

5. SYSTEM PERFORMANCE

To determine the impact of the heat transfer and pressure drop characteristics on system performance, a representative 12 kW split-system air conditioner was modeled. Heat exchanger geometry and airside performance were taken from computer runs of the PUREZ heat pump model program[12]. The overall heat transfer coefficient was determined from the airside coefficient produced by the PUREZ model and the information on refrigerant side discussed in the previous section. The NTU and the effectiveness of both heat exchangers were then calculated assuming cross-flow heat exchange. Optimum mass flow rates that produced the highest evaporator outlet temperature and the lowest condenser inlet temperature were determined (Figures 3 & 4). The results of the heat exchanger analyses are shown in Tables 1 and 2. Note that the temperatures shown represent the mid-point of any temperature glide that exists and the change in temperature indicated represents the saturation temperature change due to pressure drop only. The saturated temperature leaving the evaporator is 1.4°C higher for R-410A and 1.0°C lower for R-407C. In the condenser the saturated inlet temperature for R-410A is 1.1°C lower and 0.7°C higher for R-407C.

Heat Exchanger Performance - Evaporator

Parameter	Units	Original R-22	Optimized R-22	R-410A	R-407C
Mass Flow	g/sec	9.95	8.19	9.32	10.08
Flow Area	m^2	0.000048	0.000048	0.000048	0.000048
Mass flux	kg/sec-m^2	206	170	193	209
Delta P	kPa	53.4	39.1	25.3	51.7
Equiv T	°C	2.7	2.0	0.8	2.5
Equiv L	m	24	24	24	24
$UA_{Refrigerant}$	kW/°C	0.624	0.539	0.906	0.403
NTU	-	0.966	0.940	1.019	0.884
Effect.	-	0.619	0.610	0.639	0.587
T_{avg}	°C	9.6	9.4	10.2	8.7
T_{out}	°C	8.3	8.4	9.8	7.4
T_{in}	°C	11.0	10.3	10.5	9.9

Other HX Parameters:		
$(mc_p)_{air}$	kW/°C	0.109
$UA_{airside}$	kW/°C	0.127
UA_{R-22}	kW/°C	0.624
Tube ID	mm	7.8
Air In	°C	26.7
Air Out	°C	16.1

Note: These numbers are based on a typical North American split-system unitary a/c with a 12 SEER (3.5 Seasonal COP) rating.

Table 1

Heat Exchanger Performance - Condenser

Parameter	Units	Original R-22	Optimized R-22	R-410A	R-407C
Mass Flow	g/sec	29.85	25.20	35.28	32.76
Flow Area	m^2	0.000070	0.000070	0.000070	0.000070
Mass flux	kg/sec-m^2	428	361	506	470
Delta P	kPa	112.3	82.8	103.9	114.7
Equiv T	^oC	3.0	2.2	1.8	3.0
Equiv L	m	35	35	35	35
$UA_{Refrigerant}$	kW/^oC	2.100	1.851	2.659	1.633
NTU	-	0.882	0.848	0.940	0.814
Effect.	-	0.586	0.572	0.609	0.557
T_{avg}	^oC	41.0	41.3	40.4	41.6
T_{out}	^oC	39.5	40.2	39.6	40.2
T_{in}	^oC	42.5	42.4	41.3	43.1

Other HX Parameters:		
$(mc_p)_{air}$	kW/^oC	0.700
$UA_{airside}$	kW/^oC	0.874
UA_{R-22}	kW/^oC	2.100
Tube ID	mm	9.4
Air In	^oC	27.8
Air Out	^oC	35.5

Note: These numbers are based on a typical North American split-system unitary a/c with a 12 SEER (3.5 Seasonal COP) rating.

Table 2

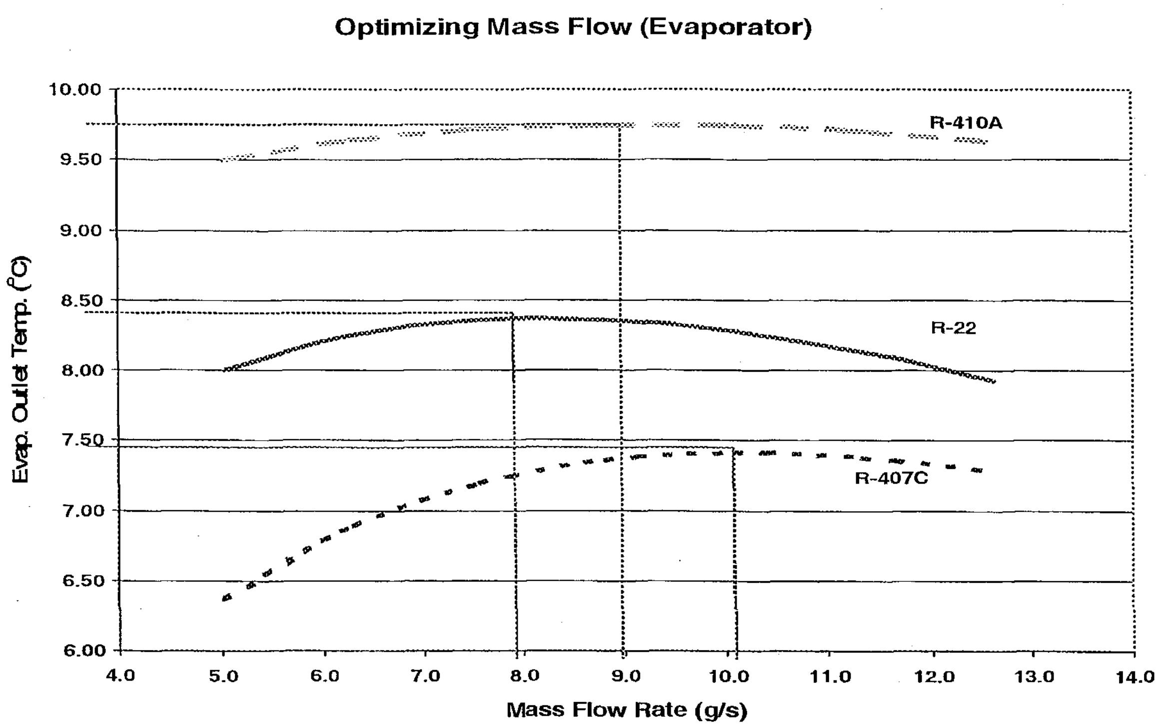

Figure 3

Optimizing Mass Flow (Condenser)

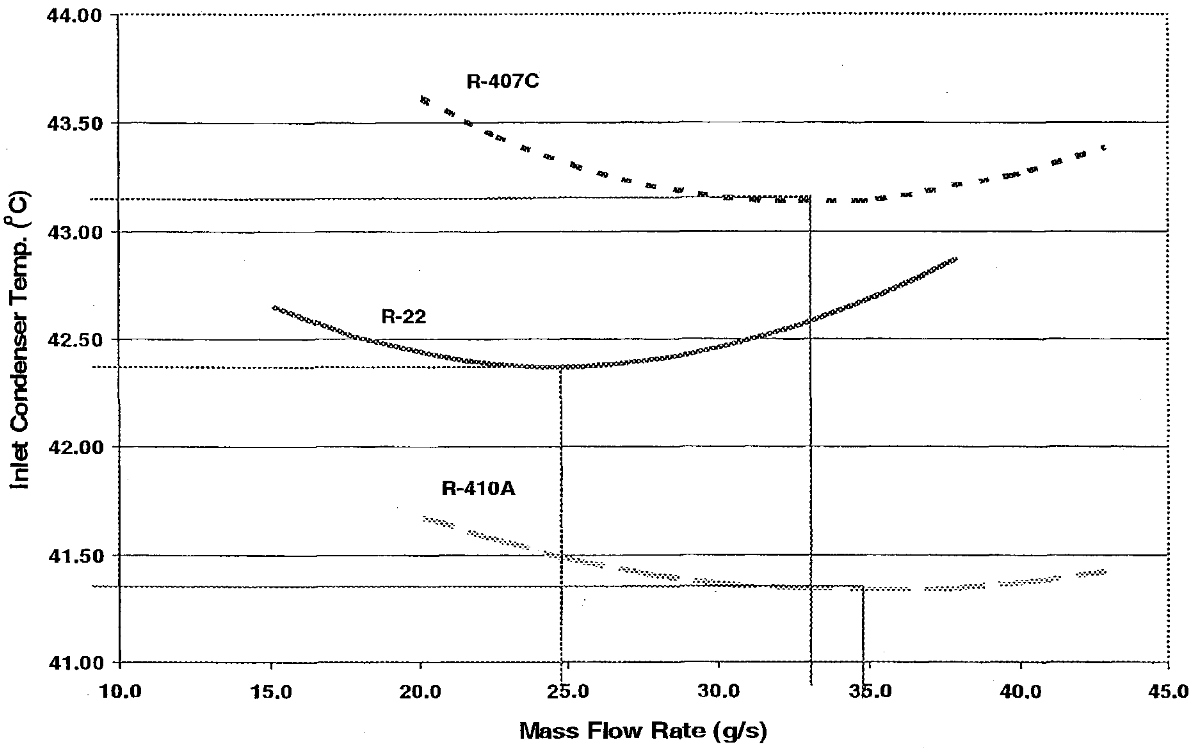

Figure 4

Overall system performance (shown on Table 3) was then determined by reducing the evaporator outlet temperature and increasing the condenser inlet temperature by ~1°C for R-22 and R-407C to take into account the pressure drop in the inter-connecting lines (the line loss was 0.7°C for R-410A due to the lower impact of pressure drop). This resulted in the determination of the saturated suction and saturated discharge temperatures of the R-22, R-407C, and R-410A system compressor. Using the appropriate 10 point performance curve fit, compressor performance[13] was determined for each system. To get system capacity, the compressor capacity was reduced by 300 watts due to the heat added by the indoor air blower. To get system power consumption, the compressor power was increased by 600 watts due to the blower and fan power consumption. System COP was determined by dividing the system capacity by the system power consumption.

The 5% increase in system efficiency for R-410A and 5% decrease for R-407C is consistent with tests of soft-optimized systems[14]. This analysis was repeated for higher ambient temperatures and Table 4 shows the results of these analyses for a range of ambient air temperatures. The efficiency and capacity does fall off at high ambient temperatures. However, even at an ambient temperature of 57°C, the saturated discharge temperature remains below the critical. The reduced capacity at extreme conditions needs to be considered if applying systems in regions where these high ambient temperatures are encountered. These systems must be sized appropriately for these regions.

Compressor/System Performance

	R-22/R-407C	R-410A
Suction line equivalent temperature drop (°C)	1.1	0.7
Discharge line equivalent temperature drop (°C):	1.1	0.7

R-22 ZR40K3-PFV compressor

Sat. Suct. T	Sat. Disch.. T	Capacity (kW)	Power	COP
7.3	43.5	13.1	2790	4.71

R-410A ZP36K3E-PFV compressor

Sat. Suct. T	Sat. Disch.. T	Capacity (kW)	Power	COP
9.0	42.1	13.3	2660	5.0

R-407C ZR40K3E-PFV Compressor

Sat. Suct. T	Sat. Disch.. T	Capacity (kW)	Power	COP
6.3	44.2	13.1	2950	4.4

Capacity is reduced by : 293 Watts due to indoor air fan

Power is increased by: 600 Watts (total parasitic power: fan & blower)

For R-22:	System Capacity (kW) is	12.84		
	System COP is	3.79	3.5 Seasonal COP (12 SEER) Unit	
For R-410A	System Capacity (kW) is	13.01	1.3% Relative to R-22	
	System COP is	3.99	5.3% Relative to R-22	
For R-407C	System Capacity (kW) is	12.82	-0.2% Relative to R-22	-1.4% Relative to R-410A
	System COP is	3.61	-4.7% Relative to R-22	-9.5% Relative to R-410A

Table 3

System Performance as a Function of Ambient Temperature

Ambient Air In		27.8	35.0	46.1	51.7	57.2
Cond. Inlet T	R-22	42.2	49.2	60.2	65.7	71.2
(°C)	R-407C	41.4	50.2	60.9	66.4	71.7
	R-410A	41.4	48.3	59.1	64.5	69.8
Discharge Press.	R-22	1517	1806	2337	2641	2972
(kPag)	R-407C	1675	2006	2599	2951	3316
	R-410A	2399	2834	3654	4123	4689
Capacity	R-22	12.95	12.43	11.45	10.92	10.41
	R-407C	12.89	12.22	11.02	10.33	9.58
	Rel. (%)	-0.5%	-1.7%	-3.7%	-5.4%	-7.9%
(kW)	R-410A	13.00	12.29	11.06	10.39	9.67
	Rel. (%)	0.4%	-1.1%	-3.4%	-4.9%	-7.1%
COP	R-22	3.83	3.23	2.41	2.07	1.77
	R-407C	3.63	3.00	2.17	1.81	1.50
	Rel. (%)	-5.1%	-7.0%	-10.0%	-12.6%	-15.5%
(W/W)	R-410A	3.98	3.28	2.36	1.97	1.63
	Rel. (%)	4.0%	1.7%	-2.3%	-5.0%	-8.1%
	R-22	3381	3849	4741	5273	5871
Power	R-407C	3548	4069	5071	5707	6402
(Watts)	Rel. (%)	4.9%	5.7%	7.0%	8.2%	9.0%
	R-410A	3266	3743	4689	5278	5935
	Rel. (%)	-3.4%	-2.8%	-1.1%	0.1%	1.1%

Table 4

GLOBAL WARMING ISSUES

In order to accurately evaluate the total impact of an air conditioning or refrigeration system on global warming, the impact of energy consumed by the system must be included. Since most energy production results in the emission of CO_2, this can make a significant impact. A comprehensive study of this issue was conducted by the Oak Ridge National Laboratory[15]. This study presented values for the total impact on global warming for unitary air conditioners as well as many other applications. Figure 5 is based on information contained in this report. This figure shows the direct effect of the refrigerant based on assumed leak rate of 4% and an end-of-life loss of 15% for systems. The original graph was based on using R-22 as the refrigerant. This graph assumes the use of R-410A instead of R-22 and assumes a 25% reduction in refrigerant charge, which could be realized in systems optimized for this refrigerant. These systems are designed for the North American market and the energy consumed is based on Pittsburgh, Pennsylvania weather. The current mandated efficiency level is 10 SEER (Seasonal Energy Efficiency Rating). It should be noted that 75% or more of residential a/c systems sold in the US are at this minimum.

It is clear that the indirect effect dominates the total impact on global warming. The direct effect generally represents from 2 to 3% of the total impact. If a 12 SEER unit was used instead of the typical 10 SEER unit, a reduction of ~12% in total warming can be obtained. This represents more than four times the direct effect of the refrigerant. If a 10 SEER system were re-designed to operate with a flammable refrigerant such as propane, there would only be a 2 to 3% decrease in total global warming. However, in order to do so, changes to the system would have to be made so the system could operate safely. Estimates of the cost of these changes are on the order of 30% of the original cost of the system. If these costs were applied to the non-flammable fluorocarbon system, efficiency increases to at least the 12 SEER level could be attained, thereby achieving a 12% reduction in global warming.

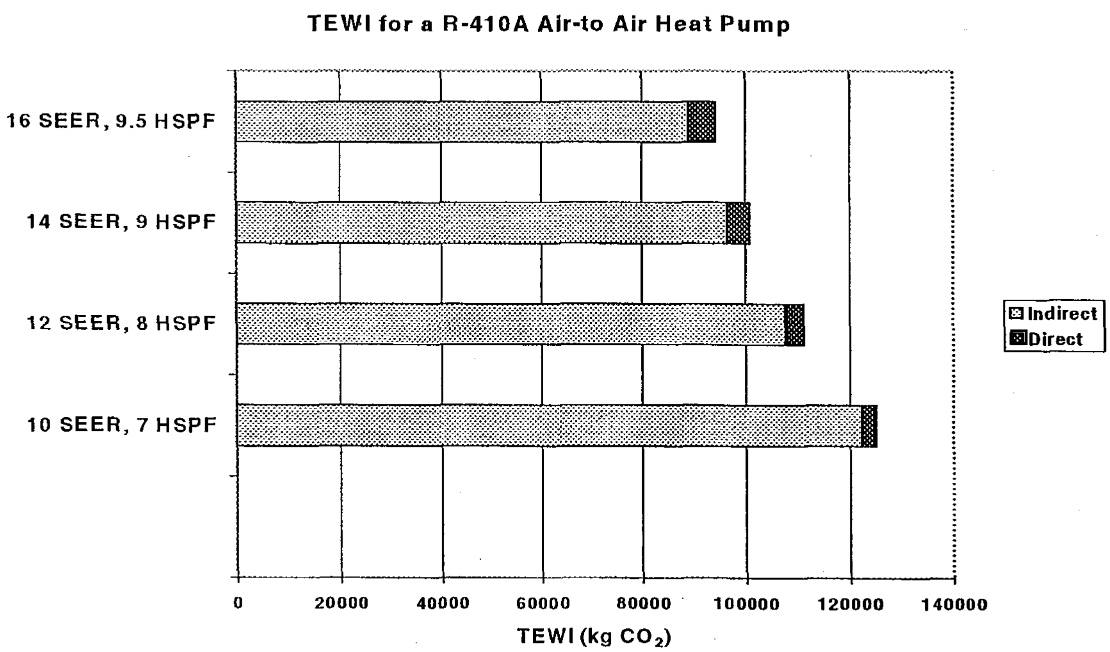

Figure 5

CONCLUSIONS

This paper evaluated the performance of the most common R-22 alternatives, R-410A and R-407C. This evaluation included the impact of the heat transfer and pressure drop characteristics of the refrigerants. One of the issues discussed was the differences in heat exchanger performance that generally leads to heat exchangers with higher mass fluxes (velocities), higher saturated evaporator outlet temperatures, and lower saturated condenser temperatures for R-410A. It was shown that the performance of the R-410A system is superior to the comparable R-22 system when operating either at average or at design conditions. R-407C heat exchanger performance was poorer than R-22 due to lower heat transfer coefficients. This resulted in system efficiency performance inferior to R-22 (~5% lower than R-22 and ~10% lower than R-410A at the average operating condition). When operating in regions of the world where ambient temperatures exceed 40°C, there is a loss in performance relative to R-22 for both refrigerants. It was also shown that energy consumption played the dominant role in determining the contribution of unitary air conditioning and heat pump systems towards global warming. Significant decreases in global warming could only be achieved by increasing the energy efficiency of these systems.

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