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EVALUATION OF R-22 ALTERNATIVES FOR HEAT PUMPS

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ABSTRACT

This study investigates three different possibilities for replacing refrigerant R-22 with R-407C in a heat pump. The first and simplest scenario is the retrofit with no hardware modification at all. This scenario resulted in cooling and heating capacities of 79 to 103 % and coefficients of performance (COPs) of 76 to 95 % of the R-22 baseline. The second possibility investigated is the path modification that requires altering the refrigerant path to attain a near-counterflow configuration in the indoor coil for the heating mode. The path modification improves the heating capacity by 2 to 16 % and the heating COP by 10 to 18 % as compared to the retrofit case. The third and most complex possibility is soft optimization consisting of maximizing the COPs in the heating and cooling modes by optimizing refrigerant charge and expansion devices. The soft optimized system with R-407C has a -3 to +1% different capacity and a -8 to -4 % different COP in the cooling and heating modes than those of R-22.

INTRODUCTION

The accelerated technical development and economic growth of most countries during the last century have produced severe environmental problems. Many manufactured products contributing to human comfort have side effects threatening our health as a result of harming the environment. The greatest inventions in thermal engineering which contribute to comfortable living are the refrigerator and air-conditioner. The first supplies cold water and ice and preserves food. The second cools and removes moisture in the air and separates indoor space from the hot and humid outdoor environment. CFCs (chlorofluorocarbons) and HCFCs (hydrochlorofluorocarbons) have been used in refrigerators and air-conditioners as working refrigerants and blowing agents in foam. CFCs are being regulated and HCFCs will be regulated because of stratospheric ozone depletion and global warming (Reed, 1993). The refrigerator and car air-conditioner industries have already responded to this challenge. We can now buy refrigerators and car air-conditioners that use non-ozone-depleting hydrofluorocarbon HFC-134a. Within a limited time, the HCFCs, including R-22, also have to be replaced.

Theoretical analysis was used to screen possible replacement mixtures for CFC and HFC refrigerants (Domanski and Didion, 1993; Radermacher and Jung, 1991).

Experimental results showing performance within ± 10% of R-22's performance were presented by refrigerant manufacturers using their proposed refrigerant mixtures (Bivens et al., 1994; Ferrari et al., 1994; Spatz and Zheng,1993). Although much research has already gone into R-22 substitutes, the problem is not solved completely. To contribute to the evaluation of R-22 alternatives, an experimental study on what was seen to be the most probable replacement refrigerant was initiated at the Center for Environmental Energy Engineering (CEEE). In this study, the "retrofit" performance of HFC-32/125/134a (23/25/52 wt. %), R-407C, was evaluated, and system hardware modifications such as "path modification" and "soft optimization" were tested to match the performance of R-407C with that of R-22.

DEFINITIONS

In this study, three options were investigated to evaluate the performance of the conventional refrigerant, R-22, and the refrigerant mixture, R-407C. They are defined as follows.

(1) Retrofit: The refrigerants, R-22 and R-407C were used in an existing system without changing any system hardware. The refrigerant charge was optimized to maximize coefficient of performance (COP) at ASHRAE[1] cooling test A condition.

(2) Path Modification (R-407C only): The refrigerant path within the indoor coil was modified by adding four additional check valves to supply the refrigerant to the indoor coil always in the same direction regardless of the action of the four-way valve. This was done to maintain near-counterflow heat exchange between refrigerant and air.

[1] ASHRAE: American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., Atlanta, GA

TABLE 1 ASHRAE TEST CONDITIONS (ASHRAE, 1983)

Test	*Indoor Dry Bulb [°C (°F)]*	*Indoor Wet Bulb [°C (°F)]*	*Outdoor Dry Bulb [°C (°F)]*	*Outdoor Wet Bulb [°C (°F)]*
A	*26.7 (80)*	*19.4 (67)*	*35.0 (95)*	*23.9 (75)*
B	*26.7 (80)*	*19.4 (67)*	*27.8 (82)*	*18.3 (65)*
High Temp.	*21.1 (70)*	*≤ 15.6 (60)*	*8.3 (47)*	*6.1 (43)*
Low Temp.	*21.1 (70)*	*≤ 15.6 (60)*	*- 8.3 (17)*	*- 9.4 (15)*

(3) Soft Optimization (R-22 and R-407C): The refrigeration cycle was optimized for both cooling and heating performances. The combination of the expansion devices and the refrigerant charge was chosen to maximize both the cooling COP for ASHRAE cooling test A and the heating COP for ASHRAE heating test 47S.

EXPERIMENTAL SETUP AND PROCEDURES

Test Facility

A test facility, called a psychrometric calorimeter, was set up to measure the steady state performance of a air-to-air heat pump. The test chamber could simulate the summer and winter climate conditions defined by ASHRAE Standard 116, Table 1 (ASHRAE, 1983).

The test facility consisted of a closed indoor loop, an environmental chamber, and a data acquisition system. The closed indoor loop was equipped with devices which measured the dry bulb and dew point temperatures of the air, and a nozzle which measured the air flow rate downstream of the test unit. The desired air flow rate for the indoor side was adjusted by an inverter that controls the speed of an additional blower located in the air handler. This blower was used to overcome the additional pressure drop caused by mixing devices, nozzle, and air handler. The frequency setting for the inverter was carefully adjusted to obtain the specific air flow rate of 22.7 m^3/min (800 ft^3/min) for both the cooling and heating modes.

Test Unit

The test unit was a 7.0 kW capacity split heat pump system. The test unit used a reciprocating compressor and two expansion devices. One expansion device was a thermostatic expansion valve (TEV) for the heating mode and the other was a short tube restrictor (S.T.) for the cooling mode. The inside diameter of the S.T. shipped originally was 1.50 mm (0.0590 in). Figure 1 shows the heat exchange configuration for the indoor coil. The indoor unit was installed at an inclination of 20° from the vertical position of the wall with downward air flow as shown in Figure 1.

The refrigerant circuit of the test unit is shown in Figure 2. The refrigerant flow direction can be switched by the action of a four-way valve. The indoor coil piping configuration was changed so that the system could be operated in two modes as shown in Figure 2. The original configuration allows near-counterflow heat exchange between the refrigerant and the air during the cooling mode, but becomes a near-parallelflow heat exchanger during the heating mode because of the action of the four-way valve. With the new arrangement, termed "path modification," the near-counterflow heat exchange can be utilized for both modes.

Capacity Measurement

The experiments to measure the capacity were performed based on ASHRAE Standard 116, and ARI[2] Standard 210/240. The air enthalpy method and the refrigerant enthalpy method were used for the capacity measurement in this study. The air enthalpy method was used as a primary method and the refrigerant enthalpy method was used as a secondary method to confirm the reliability of data. To confirm that the data are reliable, the capacity determined using these two methods should agree within 6% of each other as required by ASHRAE Standard 116. The two methods agreed within 3% for all tests conducted in this study.

In the loop air enthalpy method, the total capacity is calculated as a sum of the sensible and latent capacities. The sensible capacity is determined from measurement of the air flow rate, the humidity ratio, the air temperature entering the indoor unit, and the air temperature exiting the indoor unit. The latent capacity is calculated from the humidity ratio difference between inlet and outlet of the indoor coil. In the refrigerant enthalpy method, the refrigerant mass flow rate and the refrigerant enthalpies at the inlet and outlet of the heat exchanger are used to calculate the refrigerant-side capacity. The mass flow rate is measured by a mass flow meter, and the refrigerant enthalpies are calculated by using the REFPROP database V4.0 (Gallagher et al., 1993) from the temperature and pressure of the refrigerant at the inlet and outlet of the indoor coil. Then the capacity is calculated from the energy balance for the indoor coil.

After the test conditions were set, the test unit and the reconditioning apparatus for the indoor loop and outdoor environmental chamber were operated until equilibrium conditions were attained within the test tolerances specified in ASHRAE Standard 116. An hour after the equilibrium condition was attained, the data acquisition program was started to obtain data at 1 minute intervals for 1 hour. Then the data were averaged for every 5 minutes, and the 12 sets of data were used as an input for an energy balance program.

Test Procedures

The experimental tests were carried out with refrigerants R-22 and R- 407C. First, the R-22 "baseline" test and the "retrofit" test for R-407C were carried out without making modifications except changing the lubricant in the compressor and adding a filter drier for the mixture tests. The optimum charge that gave the maximum COP at ASHRAE Cooling test A condition was found for each refrigerant. After the optimum charge was found, ASHRAE tests were carried out to compare the performance of each refrigerant. Second, the "path modification" was implemented to improve the performance of the system with R-407C. Third, the "soft optimization" was carried out for both R-22 and R-407C. The reported capacity and COP values were based on air-side values. The refrigerant-side values were used only to check the total energy balance and to prove the validity of the test.

[2] ARI: Air- Conditioning and Refrigeration Institute, Arlington, VA

TEST RESULTS AND DISCUSSION

R-22 Charge Optimization Test

The charge optimization test results for R-22 are shown in Figure 3. When increasing the charge, the capacity and COP for R-22 increase to a maximum point and then decrease. COP for R-22 reaches a maximum at approximately 4.0 kg (8.8 lb) charge, and the capacity reaches a maximum at 4.2 kg (9.2 lb) charge. By increasing the charge, the amount of subcooling increases, while the superheat decreases. Throughout the ASHRAE tests, the refrigerant charge was maintained at 4.0 kg (8.8 lb) which corresponded to the maximum COP for ASHRAE Cooling test A.

Figure 4 shows the effect of charge on COP, power input, pressure ratio, and refrigerant mass flow rate. With increasing refrigerant charge, the system pressure increases. This causes a pressure and temperature increase in the evaporator and condenser while the system is running. Moreover the evaporating pressure rises faster than the condensing pressure does, causing a pressure ratio decrease. Then, the compressor can accept and discharge more refrigerant, resulting in a higher mass flow rate. With increased mass flow rate, the compressor has to do more work, resulting in increased energy consumption. With increased mass flow rate, more heat is absorbed by the evaporator, and the temperature distribution of indoor coil is changed. Indoor coil temperature variation with charge for the case of R-22 charge optimization with a S.T. 1.50 mm (0.0590 in) is represented in Figure 5. Approximately half of the evaporator contains two-phase flow until a 3.9 kg (8.5 lb) charge is reached. At 4.1 kg (9.0 lb), roughly 75% of the evaporator contains two-phase flow and, here, exhibits the highest COP. It is evident that the capacity increases with charge increase. The evaporating temperature also increases with charge increase. Increasing the charge beyond 4.3 kg (9.5 lb) has a negative effect because the temperature difference between air and refrigerant has been reduced.

After the charge optimization tests, the ASHRAE tests as given previously in Table 1 were carried out for R-22 with the optimum charge.

Modifications for the Refrigerant Mixture

Before the mixture was tested, the compressor was separated from the cycle, the oil remaining in the compressor was drained, and the compressor was flushed with solvent. The system piping and heat exchangers were also flushed. The refrigerant mixture, R-407C, tested is composed of HFC refrigerants only. In testing HFC refrigerant mixtures, an ester based oil was used. The filter drier was also exchanged for a drier that was originally developed for R-134a. Before charging the system, the concentration of the mixture was checked with a gas chromatograph. The concentration agreed with desired composition within ± 0.8 wt.%. The result of the concentration measurement of R-407C is compared with the desired composition and the manufacturer's data in Table 2.

R-407C Charge Optimization Test

The mixture, R-407C, was first tested as a "retrofit." Only the refrigerant charge was optimized to obtain maximum COP. The same procedure used for R-22 was used for the charge optimization test of R-407C. The results are shown in Figure 3. The COP is maximum at approximately 4.0 kg (8.8 lb) which is the same as that of R-22. After these tests, ASHRAE tests were carried out with the optimum charge. The results for both R-22 and R-407C are listed in Tables 3, 4, and 5.

TABLE 2 COMPARISON OF R-407C CONCENTRATION

Component	Desired Composition	Manufacturer's Data	Data Measured
R-32	0.23 [wt. %]	0.229 [wt. %]	0.226 [wt. %]
R-125	0.25 [wt. %]	0.236 [wt. %]	0.258 [wt. %]
R-134a	0.52 [wt. %]	0.534 [wt. %]	0.517 [wt. %]

TABLE 3 COMPARISON OF ASHRAE COOLING A TEST RESULTS

	R-22 Base	R-407C Retrofit	R-407C Path Modi.	R-22 Soft Opti.	R-407C Soft Opti.
Capacity [kW]	6.95	7.13	6.69	6.82	6.89
COP	3.08	2.94	2.91	3.20	3.06
$T_{Discharge}$ [°C]	85.5	82.0	77.8	76.3	78.1
$T_{suction}$ [°C]	22.6	21.8	19.6	18.0	23.3
P_{cond} [kPa]	1768.7	2124.2	1959.6	1641.8	1825.9
P_{evap} [kPa]	712.7	736.2	699.8	707.5	704.6
Pressure Ratio	2.48	2.89	2.80	2.32	2.59
Subcooling [°C]	9.6	14.0	10.4	4.0	4.8
Superheat [°C]	11.1	11.3	10.8	6.7	14.3

TABLE 4 COMPARISON OF ASHRAE HEATING 47S TEST RESULTS

	R-22 Base	R-407C Retrofit	R-407C Path Modi.	R-22 Soft Opti.	R-407C Soft Opti.
Capacity [kW]	6.24	5.84	5.98	6.13	5.95
COP	3.03	2.53	2.79	3.13	2.96
$T_{Discharge}$ [°C]	83.2	96.5	86.3	81.4	77.1
$T_{suction}$ [°C]	8.7	9.9	9.0	9.2	8.9
P_{cond} [kPa]	1793.9	2678.9	2236.6	1678.5	1860.3
P_{evap} [kPa]	507.4	502.9	483.1	494.6	478.9
Pressure Ratio	3.54	5.33	4.63	3.39	3.88
Subcooling [°C]	21.1	39.9	29.9	20.2	21.4
Superheat [°C]	9.9	11.3	11.6	9.5	11.7

Path Modification Test

The indoor coil of the test unit has a quasi-crossflow heat exchange configuration. In the cooling mode, the refrigerant flows from bottom to top, whereas in the heating mode the refrigerant flows from top to bottom as shown in Figure 1. The heat exchange pattern between air and refrigerant is near-counterflow for the cooling mode, but it is near-parallelflow for heating. The indoor coil path was changed by installing four additional check valves as shown in Figure 2. With this path modification, the refrigerant can be supplied to the indoor coil from bottom to top for both cooling and heating modes to maintain a near-counterflow heat exchange between air and refrigerant.

Comparison of R-22 Baseline, R-407C Retrofit, and Path Modification

The retrofit and path modification test results for R-407C are compared with R-22 baseline test results in Tables 3 and 4 for cooling test A and heating test 47S. In Table 5, the test results for cooling test B and heating test 17L are compared. The retrofit test results for R-407C show a 2.6% higher capacity and a 4.5% lower COP in the cooling A case (Table 3), and a 6.4% lower capacity and a 16.5% lower COP in the heating 47S case (Table 4) as compared to the R-22 baseline. Cooling test B has similar results to cooling A case (Table 5), while heating test 17L has a greater degradation than the heating 47S case (Table 5). The cooling results are similar to the Alternative Refrigerant Evaluation Program (AREP) results shown in Table 6 (Godwin, 1993), but the heating results show more degradation. Although R-407C has similar thermodynamic characteristics to R-22, some key cycle parameters are different. R-407C shows a 16.5% higher pressure ratio compared to that of R-22 for the cooling A case. Also, the cycle shifts to higher evaporating and condensing pressures. The higher pressure ratio lowers the compressor efficiency. Although R-407C has a slightly higher capacity, it has a lower COP.

R-407C retrofit shows more degradation for heating than for cooling. This can be explained by the heat exchange configuration of the indoor coil as shown in Figures 1 and 2. As already mentioned, the heat exchanger configuration contributes to the capacity degradation in the heating mode, especially for mixtures that have a temperature glide.

Path modification results show a 3.7% lower capacity and a 5.5% lower COP for cooling test A as compared to that of R-22. The reason is that the four additional check valves cause additional pressure drop, resulting in a lower mass flow rate and a lower evaporating pressure. But it shows a significant improvement, 10.3%, in the heating COP for heating test 47S, as compared to the retrofit case. The greatest improvement for path modification can be observed in low temperature heating test 17L: 15.6% for capacity and 17.6% for COP. The refrigerant temperatures along the cycle for retrofit and path modification are compared in Figures 6 and 7, for cooling test A and heating test 47S. In these figures, the x-axis is not to scale; it indicates just the sequence of each temperature probe as the refrigerant passes on its way through the cycle. The first point is the compressor discharge and the last point is the compressor inlet. The profile for the cooling of R-407C is similar to that of R-22. The heating case shows a higher compressor discharge temperature and a lower evaporating temperature than that of R-22, resulting in a high pressure ratio for R-407C. The temperature profiles in the indoor coil, the condenser, show large differences. These temperature profiles explain the performance degradation of the retrofit case which had the large temperature difference because of the near-parallelflow configuration. They also explain the performance improvement due to path modification which resulted in a small temperature difference because of the near-counterflow configuration. The path modification can improve the performance by reducing the thermodynamic irreversibility.

TABLE 5 COMPARISON OF ASHRAE COOLING B AND HEATING 17L TEST RESULTS

	R-22 Base	*R-407C Retrofit*	*R-407C Path Modi.*	*R-22 Soft Opti.*	*R-407C Soft Opti.*
"B" Capacity [kW]	*7.12*	*7.13*	*7.15*	*7.34*	*7.16*
"B" COP	*3.52*	*3.22*	*3.40*	*3.78*	*3.49*
"17L" Capacity [kW]	*3.40*	*2.69*	*3.11*	*2.77*	*2.75*
"17L" COP	*2.09*	*1.59*	*1.87*	*1.97*	*1.82*

TABLE 6 COMPARISON OF SOFT OPTIMIZATION RESULTS WITH AREP DATA

	Cooling		*Heating*	
	Capacity	*COP*	*Capacity*	*COP*
Present Study	*0.98 - 1.01*	*0.92 - 0.96*	*0.97 - 0.99*	*0.92 - 0.95*
AREP Result	*0.93 - 1.01*	*0.90 - 0.97*	*0.98 - 1.02*	*0.93 - 1.02*

Note: The above data are the ratios of R-407C performance to R-22 performance.

Although the path modification improved the performance of the R-407C over the retrofit, it still has degradation as compared to R-22, especially in the heating mode: a 4.2% lower capacity and a 7.9% lower COP at heating test 47S. The reason for this degradation is that the retrofit was performed after the charge was optimized at cooling test A condition without optimizing at the heating test condition. So it is natural to degrade the heating performance. These results indicate that more extensive optimization is necessary.

Soft Optimization Test Results

To improve the steady state performance of the mixture, the soft optimization was carried out for R-22 and R-407C, which could reconcile the imbalance between the cooling and heating optimums. At first, heating test 47S was conducted to obtain the optimum charge for heating. The results are shown in Figure 8. The charge that has the maximum COP is 3.6 kg (8.0 lb) for R-22 and 3.4 kg (7.5 lb) for R-407C. To obtain the same optimum charge for cooling, the S.T. diameter was increased. Optimum charge tests for each S.T were carried out with charge increments of 0.2 kg (0.5 lb). Figures 9 and 10 show the change of capacity and COP with refrigerant charge for R-22 and R-407C, respectively. The results show that R-22 has the same optimum charge for the cooling and heating modes with a 3.6 kg (8.0 lb) charge and a 1.65 mm (0.0650 in) S.T., while R-407C has the same optimum charge for both modes with a 3.4 kg (7.5 lb) charge and a 1.70 mm (0.0670 in) S.T. Therefore soft optimization has a 10-15 % less refrigerant charge and a 10-14% larger S.T. than those of the retrofit case.

Another trend observed was the change in the amount of subcooling and superheating. As the charge was increased, the subcooling increased but the superheat decreased. Increasing the size of the S.T. with the same charge decreases both the subcooling and superheat.

Since the refrigerant mass flow rate increases as the charge increases, a greater portion of the evaporator can be used, and the superheat is reduced. The decrease in the superheat at the compressor inlet reduces the superheat of the compressor outlet, and the rise in refrigerant mass flow rate increases the capacity of the condenser. Therefore, the subcooling increases with increasing refrigerant charge because of the increased condenser capacity and the reduced superheat of the compressor outlet. Moreover, these phenomena can be retarded by decreasing the size of the S.T., because of the decreased mass flow rate.

After the charge optimization of the heating and cooling modes, ASHRAE tests were carried out with the optimum S.T. and refrigerant charge. Test results are shown in Tables 3, 4, and 5.

Comparison of Soft Optimization for R-22 and R-407C

As can be seen in Tables 3 and 4, the pressure ratio of R-22 after soft optimization is reduced by 6.5 % for cooling test A and by 4.2% for the heating test 47S as compared to the R-22 baseline. The lower mass flow rate causes a loss of 2% in capacity, but the lower pressure ratio causes a 3-4 % increase in COP. From this result, it can be said that the existing unit can also be improved in terms of COP with a small loss of capacity by soft optimization. The R-407C soft optimization still shows degradation as compared to R-22 by 4-8 % for the cooling and heating cases in terms of COP. But these COP results are better than the retrofit case, by 4-8 % for the cooling mode and 15-17 % for the heating mode in terms of COP. Also, these results agree with the results presented by AREP (Godwin, 1993).

The temperature profiles for the R-22 and R-407C soft optimization are shown in Figures 11 and 12. In these figures, coordinates are the same as those of Figures 6 and 7. The temperature profiles are very close to each other except the temperature glide of R-407C for both modes. This suggests why soft optimization can achieve a capacity and COP that are closer to those of R-22.

R-407C shows a performance degradation when retrofitting, so soft optimization should be used. The soft optimization of R-407C shows a steady state performance (capacity and COP) within ± 8% of the soft optimization performance of R-22. These results are compared with AREP test results (Godwin, 1993) in Table 6.

CONCLUSIONS

The R-22 baseline test and the R-407C retrofit test were carried out using ASHRAE test conditions for both cooling and heating. "Retrofit" was defined as changing only the refrigerant and lubricant with no equipment changes. With this definition, the retrofit tests in this study were conducted with charge optimization only. The results show that R-407C has a 0-3 % higher cooling capacity, a 5-9 % lower cooling COP, a 6-21 % lower heating capacity, and a 17-24 % lower heating COP as compared to the R-22 baseline.

The capacity and COP in the heating mode show a larger degradation than in the cooling mode, so the heating performance should be improved by methods such as hardware modifications or soft optimization. Path modification is one way to improve the heat exchange performance of mixtures by taking advantage of the temperature glide. The advantage of this option was proven by comparing the results with those of the retrofit. The path modification for this study improved the heating capacity by 2-16 % and the heating COP by 10-18 % as compared to the retrofit case. It also changed the cycle temperature profile so that it approached that of R-22.

Soft optimization, balancing cooling and heating performance, requires significant effort to find the appropriate expansion device and the optimum charge. Even with R-22, soft optimization can improve the COP by 3-4 % at the expense of a 2% capacity loss, if this decrease in capacity is acceptable. Relative to the R-22 soft optimization, the R-407C soft optimization shows a capacity of 97-101 % and a COP of 92-96 %.

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NOMENCLATURE

17L	Low Temperature Heating Test based on ASHRAE Standard 116
47S	High Temp Heating Test based on ASHRAE Standard 116
ASHRAE	American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., Atlanta, GA
ARI	Air- Conditioning and Refrigeration Institute, Arlington, VA
AREP	Alternative Refrigerant Evaluation Program
CFCs	Chlorofluorocarbons
COP	Coefficient of Performance
G.W.P.	Global Warming Potential
HCFCs	Hydrochlorofluorocarbons
HFCs	Hydrofluorocarbons
O.D.P.	Ozone Depletion Potential
R-22	Refrigerant 22, Trifluoromethane
R-407C	Refrigerant consists of R-32/R-125/R-134a in composition of 23/25/52 wt.%
P_{cond}	Condensing Pressure
P_{evap}	Evaporating Pressure
Pressure Ratio	Ratio between P_{cond} and P_{evap}
S.T.	Short Tube Restrictor
$T_{discharge}$	Compressor Discharge Temperature
$T_{suction}$	Compressor Suction Temperature

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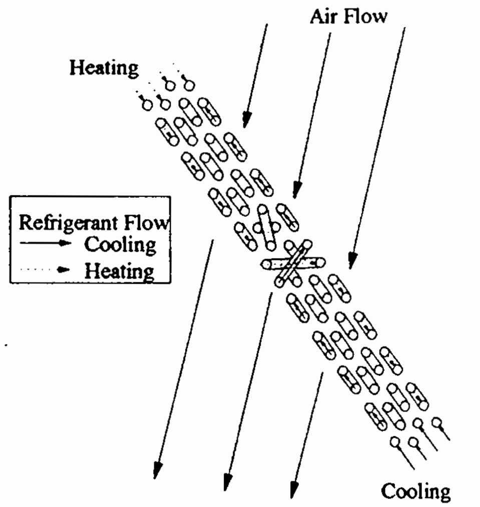

FIGURE 1. INDOOR COIL HEAT EXCHANGER CONFIGURATION

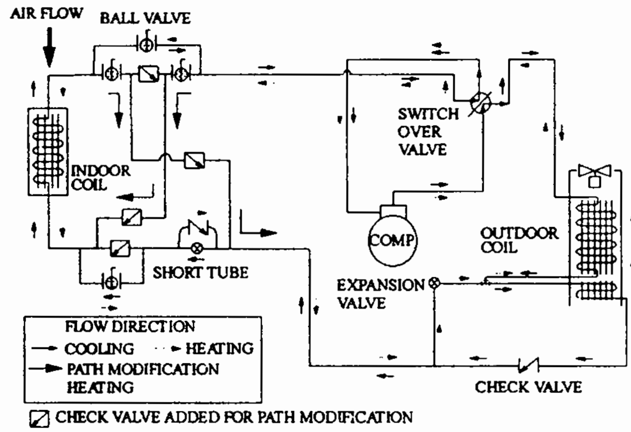

FIGURE 2. REFRIGERANT CIRCUIT DIAGRAM

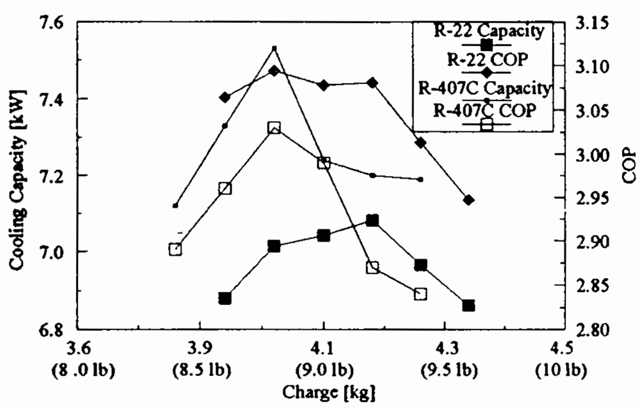

FIGURE 3. R-22 & R-407C CHARGE VS. CAPACITY & COP AT COOLING TEST A (1.50 mm S.T.)

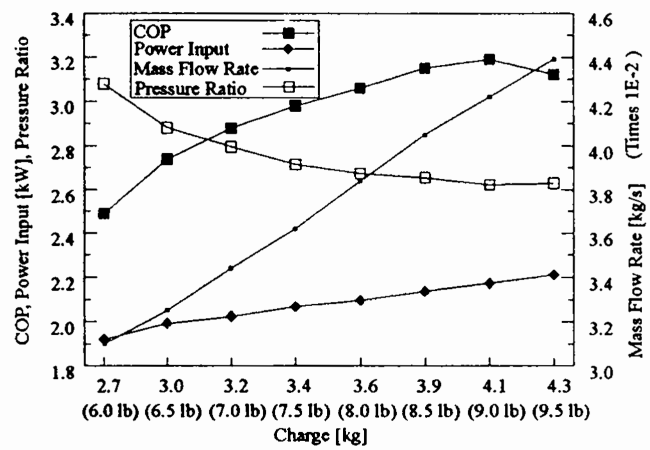

FIGURE 4. CHARGE EFFECT ON CYCLE (R-22, COOLING TEST A, 1.50 mm S.T.)

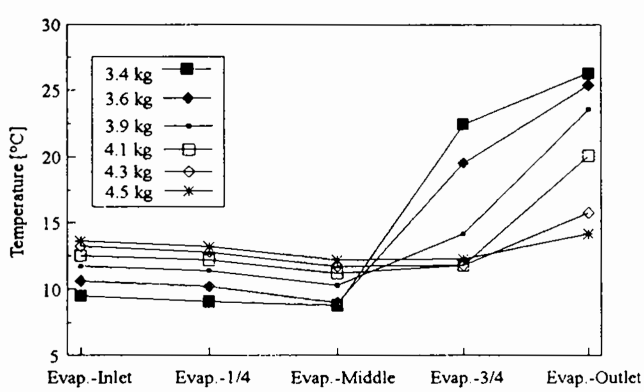

FIGURE 5. CHARGE EFFECT ON INDOOR COIL TEMPERATURE (R-22, 1.50 mm S.T.)

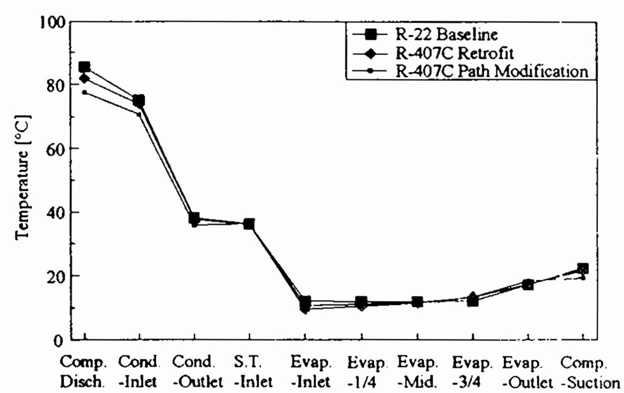

FIGURE 6. TEMPERATURE PROFILES ALONG THE CYCLE AT COOLING TEST A

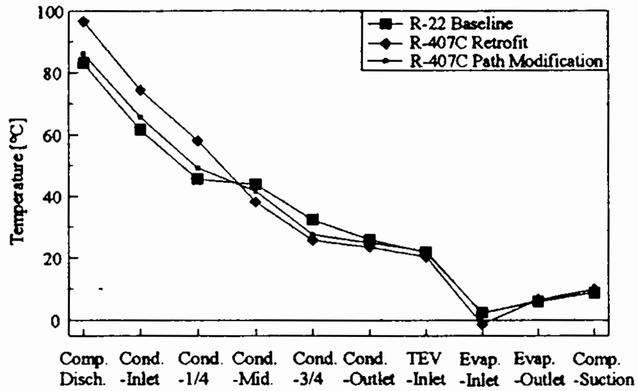

FIGURE 7. TEMPERATURE PROFILES ALONG THE CYCLE AT HEATING TEST 47S

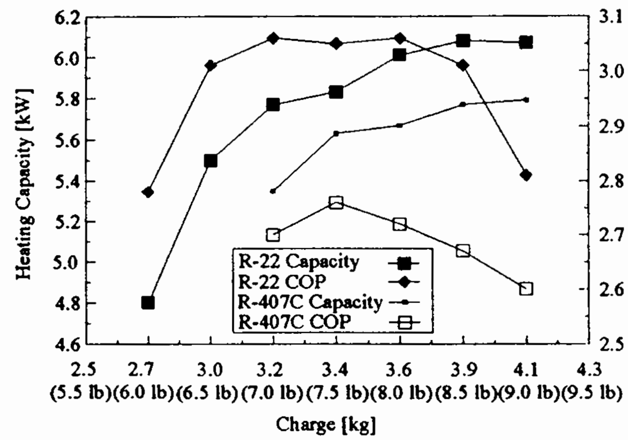

FIGURE 8. R-22 & R-407C CHARGE VS. CAPACITY & COP AT HEATING TEST 47S

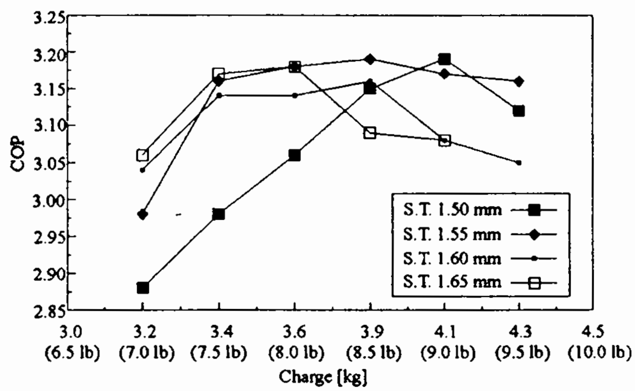

FIGURE 9. R-22 CHARGE VS. COP WITH DIFFERENT SHORT TUBE RESTRICTORS

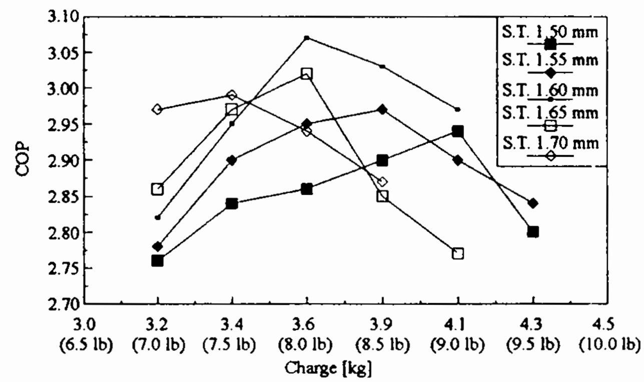

FIGURE 10. R-407C CHARGE VS. COP WITH DIFFERENT SHORT TUBE RESTRICTORS

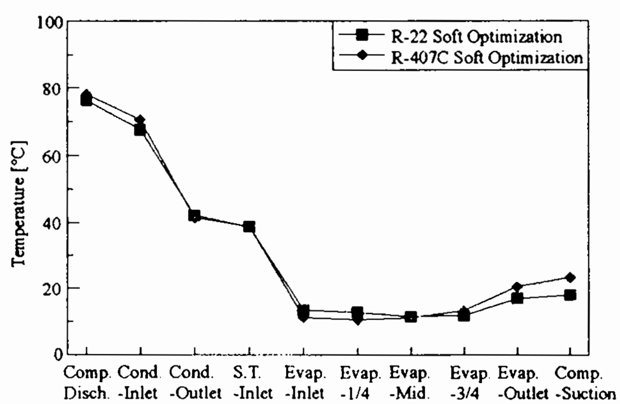

FIGURE 11. TEMPERATURE PROFILES ALONG THE CYCLE AT COOLING TEST A

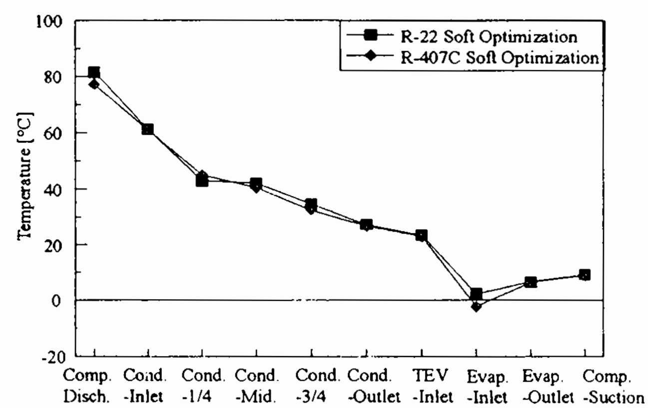

FIGURE 12. TEMPERATURE PROFILES ALONG THE CYCLE AT HEATING TEST 47S

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16. ABSTRACT The paper reports results of a study investigating three different possibilities for replacing refrigerant R-22 with R-407C in a heat pump. The first and simplest scenario was a retrofit without any hardware modifications, resulting in cooling and heating capacities of 79 to 103% and coefficients of performance (COPs) of 76 to 95% of the R-22 baseline. The second possibility was a modification that required altering the refrigerant path to attain a near-counterflow configuration in the indoor coil for the heating mode: this improved the heating capacity by 2 to 16% and the heating COP by 10 to 18%, compared to the retrofit case. The third and most complex possibility was soft optimization, consisting of maximizing the COPs in the heating and cooling modes by optimizing the refrigerant charge and expansion devices: with R-407C, this had a -3 to +1% different capacity and a -8 to -4% different COP in the cooling and heating modes, compared to R-22.

17. KEY WORDS AND DOCUMENT ANALYSIS

a. DESCRIPTORS	b. IDENTIFIERS/OPEN ENDED TERMS	c. COSATI Field/Group
Pollution	Pollution Control	13B
Heat Pumps	Stationary Sources	13A
Refrigerants	Chlorofluorocarbons	
Halohydrocarbons	Hydrochlorofluoro-carbons	07C
Ozone		07B
Greenhouse Effect	Global Warming	04A

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