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Optimization Strategies for Unitary Air Conditioners Using R-410A

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Abstract

R-410A (AZ-20) has achieved broad acceptance as the ultimate replacement for HCFC-22 in residential air conditioning systems. This paper examines the design optimization of this type of equipment based on the use of R-410A. The Oak Ridge Heat Pump Model (MARK V or Purez) is utilized in this study for steady-state performance simulation. Efficiency levels corresponding to SEER levels of 10,11,12,13, and 14 are examined for a typical nominal 3 ton split system. This study included an assessment of the cost and equipment size impacts of coil circuiting, alternative tube diameters, and compressor efficiency. The early market price of the HFC alternative refrigerants will likely to be higher than R-22, which will play a larger role in equipment design than has been typical in the recent past. The envelope of conventional design practice is pushed to examine operation at extreme conditions. The potential for supercritical heat rejection and its effect on system operating parameters is evaluated.

INTRODUCTION

In response to the growing concerns and scientific evidence of a decline in the earth's protective ozone layer, an international agreement, the Montreal Protocol on Substances that Deplete the Ozone Layer, was signed in September 1987. Its consequent provisions control the production and consumption of chlorofluorocarbons (CFCs) and hydrochlorofluorocarbons (HCFCs). CFCs such as R-12 were targeted first and their productions ceased at the end of 1995 in developed countries. The most current amendments of the Protocol in Copenhagen and Vienna call for a freeze in production of HCFCs. A 35% reduction to the "cap" value calculated from 1989 consumption data is expected in 2004. R-22 has been the primary refrigerant for air conditioning for many years. This popular refrigerant continues to be used today as replacement for CFCs in refrigeration applications. The U.S. 1990 Clean Air Act in conjunction with Montreal Protocol established January 1, 2010 as the date to ban the manufacturing of products using R-22 in the United States. In fact, the use of R-22 in new equipment has already ceased in some European countries such as Germany.

In the United States, the largest volume use for R-22 is in unitary air conditioning for residential applications. The American 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. Three candidates were identified initially to be the most likely replacement of R-22 in residential application. They were an azeotropic mixture of HFC-32 and 125 (now known as R-410A or AZ-20), a ternary non-azeotropic blend of HFC-32, 125, and 134a (R-407C) and a pure refrigerant R-134a. Progress made since then by manufacturers of air conditioning equipment, compressors, and refrigerant producers indicate R-410A as the dominant choice to replace R-22 in new equipment.

R-410A CHARACTERISTICS

R-410A is non-flammable and very low in toxicity (A1/A1 rating by ASHRAE). This binary mixture of HFC-32 and 125 exhibits azeotropic behavior, having negligible refrigerant component segregation and temperature glide characteristics. R-410A vapor pressure is about 60% higher than R-22, that means R-410A will be used mainly for new equipment. It would not be easily adopted for retrofitting existing unitary equipment in the field such that the UL listing requirement is satisfied.

With its superior transport properties and other advantages, R-410A emerged as the refrigerant of choice in the last few years. With a higher vapor pressure in the application temperature range, a given pressure drop will result in a lower saturation temperature change in the condenser and evaporator. It has also been demonstrated from compressor calorimeter studies that smaller losses within the compressor or better isentropic efficiency can be obtained with R-410A compared to R-22. These favorable characteristics of R-410A will positively impact heat exchanger design, line sizing and compressor efficiency. These attributes will allow overall system design optimization to achieve 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 return to the compressor. Polyol ester lubricants or other soluble lubricants must be used. The combination of this HFC-based refrigerant and its paired lubricant must be considered together when evaluating and selecting elastomers or plastics. The refrigerant also impacts the selection of desiccant material in the filter-drier. These factors along with heat exchangers, lines,, and compressor considerations influence the cost of the system. This paper only addresses cost concerns in coil design and configuration, sheet metal casing, unit packaging and estimated refrigerant costs. The higher cost of new lubricant would be incorporated into the compressor cost.

UNITARY SYSTEM CONFIGURATION

Unitary split air conditioners, most popular in the United States, are typically equipped with a single evaporator finned coil (most commonly "A" coil) which delivers cooling via an air handler (indoor unit) and ducts throughout the living space. The condensing unit, consisting of a condenser finned coil and a compressor, is located outdoors with interconnecting suction and liquid lines to the indoor coil. Lines are varying in length form about 15 to 150 feet with most commonly about 20-30 feet. The basic configuration chosen for this modeling study is an "A" coil indoor unit, compressor with a wrap around condenser coil and squared base pan as outdoor unit, and connecting liquid and vapor lines of 25 feet.

From previous studies, it was learned that the following changes can be made to the R-22 design to allow for proper and efficient operation with R-410A. These changes include a reduction in compressor displacement of about 33%, as well as smaller diameter heat exchanger tubing. Earlier work by the authors has verified experimentally that a finned coil design with the same sized tube (so called soft-optimization) will achieve optimum performance with R-410A when the number of circuits is reduced by one third of the R-22 circuits. Higher refrigerant flow velocity through smaller tubes and better tolerance for pressure drop losses improved refrigerant-side heat transfer of R-410A. An increase in the wall thickness of the compressor shell and refrigerant tubes may be required in order to accommodate the higher pressure of R-410A. The possibility of using smaller tubing for R-410A gives equipment design engineers an opportunity to utilize the increased strength of the smaller tubes while reducing overall material use and cost. It allows for a more compact design of R-410A system for a desired ARI energy efficiency level (commonly referred as SEER). There is also an associated benefit of reduced refrigerant charge for a system of given tonnage.

PERFORMANCE & COST MODELLING

The Oak Ridge Heat Pump model (MARK V or Purez) was used for performance modeling. Purez is a public-domain system design tool specifically for air-to-air air conditioning or heat pump applications. It has evolved over the years and presently has the capability of handling many different refrigerants including R-22 and R-410A. The program predicts the steady-state performance of a given design by using single and two-phase heat transfer and pressure drop models for finned tube, air-to-refrigerant heat exchangers. Simplified parallel refrigerant circuiting is assumed and air-side dehumidification and evaporator sensible-heat-ratio are calculated. Compressors are modeled using the manufacturer's performance maps and curve-fit to ARI ten-coefficient format. The model facilitates performance optimization studies aimed at finding the best balance of heat transfer versus pressure drop for R-22 and R-410A.

The cost model was internally developed. It consists of three modules: coil cost including material, labor and overhead; sheet metal cost which was linked to condenser coil geometry; packaging cost which was proportioned to sheet metal cost. Compressor and refrigerant costs were not included in the cost model. Their impacts can be estimated separately. Additional cost for R-410A compressors compared to R-22 is expected due to several reasons. They include: heavier shell, new lubricant costs, capital investment recovery and developmental expenses. Consideration given to refrigerant charge size can be based on percentage of liquid phase occupied internal volume of the coils and interconnecting tubes. Using smaller tubing for R-410A would reduce internal volumes, lowering the required amount of refrigerant, thereby reducing the total refrigerant cost increase.

The design changes were simulated maintaining equal performance at ARI steady state rating conditions--cooling capacity at 95 °F outdoor temperature and energy efficiency ratio (EER) at 82 °F outdoor temperature. The cost model was used to calculate coil, sheet metal, and packaging direct costs plus a standard overhead cost of any particular design. Iterations between the cost and performance models were necessary to finalize on the optimized coil configuration—minimum coil and packaging cost with equivalent performance at the given SEER level. The study was started with a baseline unit using R-22 at a 10 SEER efficiency rating with a reciprocating compressor. This was chosen for two reasons. First, it represents the current minimum seasonal energy efficiency required under federally mandated standards in the United States. Also, previous experimental studies of a similar 10 SEER air conditioner by the authors had also provided a calibrated performance base of both R-22 and R-410A to gain more confidence in the model prediction. For 11,12,13 and 14 SEER, the units were then modeled by using a map from a commercially available scroll compressor for R-410A at the current efficiency level. Compressor displacement was scaled up or down in the model to get the desirable cooling capacity of 3 tons.

There were no changes for the R-22 evaporator design. Optimization was performed to find the desired "A" coil configuration for R-410A at 10 SEER. No additional changes were made to this design in subsequent modeling efforts in this report. It was indicated that one can save $11.71 with the change in evaporator coil design for R-410A compared to R-22. Table 1 shows the optimized R-410A and baseline R-22 evaporators.

TABLE 1. Optimized R-410A and R-22 Evaporator Coils

For All SEER Levels

		R-410A Optimized	R-22
Copper Tube Internal Surface		Micro-fin (rifled)	Smooth
Finned Height	inch	12	14
Length between Tube Sheets	inch	13	18
Vertical Tube Spacing	inch	0.75	1
Horizontal Tube Spacing	inch	0.625	0.866
Number of Rows		3	3
Number of Slabs		2	2
Copper Tube Diameter	inch	0.2756	0.375
Copper Tube Wall Thickness	inch	0.0118	0.013
Aluminum Fin Type		Enhanced	Enhanced
Aluminum Fin Thickness	inch	0.0047	0.0047
Fin Spacing	fpi	13	14

A matrix of various designs of condensing units was explored. Tube diameters of 3/8", 5/16" and 7 mm with smooth or rifled (enhanced micro-fin) internal surfaces were evaluated for both R-22 and R-410A. Optimized designs of 3 ton condensing units at efficiency levels of 11,12,13 and 14 SEER were developed and summarized. It was noted that the system performance was very sensitive to air flow due to dominant heat transfer resistance on the air-side. A consistant face velocity was assumed during the simulation, so CFM (air flow) and consequently the condenser fan power consumption were varied to take into account their impacts.

EFFECT OF COMPRESSOR EFFICIENCY

It was believed that due to anticipated compressor technology advancement and computerized manufacturing, a compressor efficiency improvement of 5% relative to the current R-410A compressor may be possible without substantial cost penalty in the near future. A typical scroll compressor with 5% improved efficiency compared to the current baseline R-22 compressor was modeled and added to the system simulation. With higher compressor efficiency, cost savings from decreased heat exchanger coil surface can be realized while maintaining the same equipment SEER rating.

MODELLING RESULTS

Table 2 displays baseline R-22 condensing unit coil and optimized R-410A coil designs for a typical 3 Ton, 13 SEER condensing units with a scroll compressor.

TABLE 2. Condensing Unit and Costs (13 SEER)

				Coil Costs:	**R-410A**	**R-22**
Optimized 13 SEER		**R-410A**	**R-22**	Total Copper Cost	$16.69	$ 33.74
Copper Tube Internal Surface		Smooth	Rifled	Total Aluminum Cost	$13.04	$ 20.65
Finned Height	in.	36	36	Misc. Parts Cost	$ 1.37	$ 3.21
Length between Tube Sheets	in.	55.8	75	Labor and Overhead Cost	$15.17	$ 17.80
Vertical Tube Spacing	in.	0.75	1	Subtotal Coil Costs:	$46.27	$ 75.40
Horizontal Tube Spacing	in.	0.625	0.866	Sheet Metal Costs:		
Number of Rows		1	1	Base Pan Cost	$ 2.57	$ 4.31
Copper Tube Diameter	in.	0.2756	0.375	Top Panel Cost	$ 2.57	$ 4.31
Copper Tube Wall Thickness	in.	0.013	0.015	Front Panel Cost	$ 2.60	$ 3.40
Aluminum Fin Type		Enhanced	Enhanced	Other Parts Cost	$ 4.25	$ 4.25
Aluminum Fin Thickness	in.	0.0047	0.0047	Subtotal Sheet Metal Costs:	$11.99	$ 16.27
Fin Spacing	fpi	20	17	Packaging Costs:	$ 3.47	$ 4.72
				Total Unit Savings:	**$34.66**	

Figure 1 illustrates the condensing unit cost savings at different SEER levels. Figure 2 demonstrates the effect of the equivalent number of circuits on performance of an optimized coil.

Figure 1. Condensing Unit Cost Savings vs. SEER, R-410A Compared to R-22

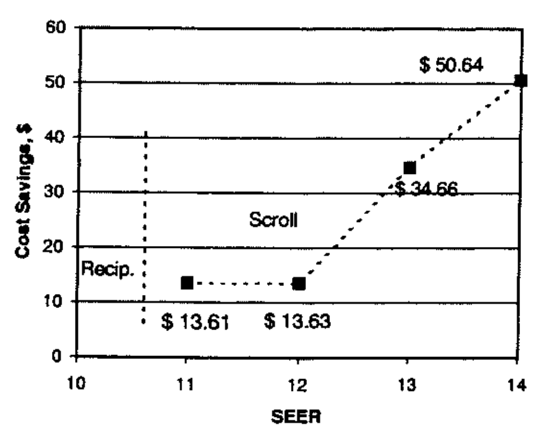

Figure 2. Effect of Circuit Numbers (R-410A)

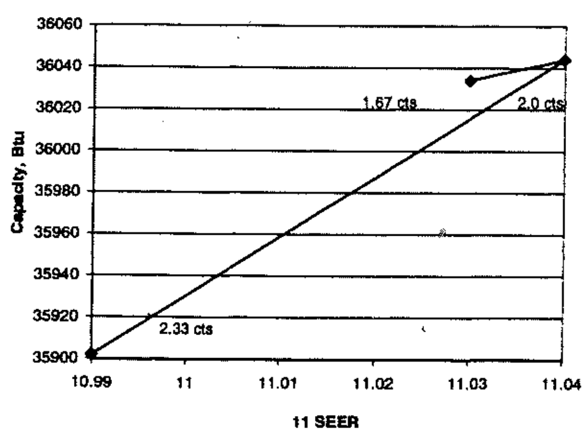

Table 3 summarizes performance and cost optimization results and condensing unit details. It should be noted that the performance model was verified at the 10 SEER level design. When the modeling was extended to higher efficiency levels, especially at 14 SEER, its response to design changes became extremely flat (insensitive). Further experimental verification of high efficiency equipment would be desirable in the future. There are a few alternate designs worth mentioning. For an 11 SEER unit, the fully optimized version calls for 7 mm tubes which would likely require re-tooling capital investment in the coil manufacturing facility (not included in our analysis). The unit savings for this 7 mm design is $13.61. The alternative is to stay with 3/8'' tubing of R-22 design thereby avoiding re-tooling expense, but the unit savings will drop from $13.61 down to $10.84. For the 12 SEER simulation, two very similar cases are presented where cost savings are comparable, both use 7 mm tube for R-410A but one is smooth tube while the other is rifled.

TABLE 3. Summary of Performance & Cost Modeling Results

$ Savings not Reflecting Compressor and Refrigerant Costs' Increases

SEER	Refrigerant	Savings $	Capacity Btu/hr	Air Flow CFM	Fan W	Sens./Lat. load ratio	Condensing Coil Details Tube Dia. (nominal)	Surface type	Dimension inch x inch	fpi
11	22	Baseline	36082	2400	282	0.7628	3/8"	smooth	24 x 70.3	17
	410A	$ 13.61	36044	2117	271	0.7702	7mm	smooth	22.5 x 58	20
	410A alt.	*$ 10.84*	*36112*	*2190*	*274*	0.7682	*3/8"*	smooth	*24 x 52.5*	*17*
	410A w. 5% eff. comp.	$ 15.73	36026	1938	272	0.7726	7mm	smooth	20.25 x 59	20
12	22	Baseline	35950	3000	300	0.7632	3/8"	smooth	30 x 70	17
	410A	$ 13.63	35992	2555	286	0.7707	7mm	smooth	22.5 x 70	20
	410A alt.	*$ 13.24*	*36050*	*2300*	*279*	0.7696	*7mm*	rifled	*22.5 x 63*	*20*
	410A w. 5% eff. comp.	$ 19.22	35950	2100	273	0.7710	7mm	rifled	22.5 x 56	20
13	22	Baseline	36083	3859	324	0.7629	3/8"	rifled	36 x 75	17
	410A	$ 34.66	36034	3261	308	0.7702	7mm	smooth	36 x 55.83	20
	410A w. 5% eff. comp.	$ 39.97	36023	2823	294	0.7703	7mm	smooth	30 x 58	20
14	22	Baseline	36600	4000	320	0.7518	5/16"	rifled	42 x 72, 2 row	17
	410A	$ 50.64	36001	3115	304	0.7706	7mm	smooth	30 x 64, 2 row	20

OPERATION AT EXTREME CONDITIONS

The relatively low critical properties of R-410A (163 °F, 720 psia) is a concern to OEMs as they consider system performance in high ambient conditions. Model predictions were made at 115 °F and 125 °F outdoor air temperatures on the optimized 12 SEER unit. Details of performance simulation are shown in Table 4. The model indicated that R-410A would still operate below critical temperature but its cooling capacity and efficiency would gradually drop as air temperature increases. For R-22 the drop is not as severe. Supercritical heat rejection of R-410A might occur in the "condenser" for even higher air temperature such as in the desert. The system cooling capacity and efficiency corresponding to heat rejection slightly above critical point should not be so different from the performance found for condensing temperatures below the critical. In a transcritical vapor compression system the condenser pressure is independent from its temperature. The term condensing temperature no longer applies. An equipment made to perform cooling duties in these extreme conditions may have to be examined closely to see its effect on other system operating parameters. Certain design or cost trade-off may be made to achieve a balance between rating point performance and acceptable performance during these extremely high ambient operations. Design consideration has to be given to the fact that the equipment operates mostly as a conventional Rankine cycle machine while it still needs to perform reasonably at the extreme conditions in transcritical mode.

Use of a suction line heat exchanger to recover capacity loss of R-410A at these extreme high ambient conditions should be beneficial and fairly easy to accomplish for cooling only applications. Further analysis with this design enhancement is necessary. Studies on other fluid transcritical cycle characteristics indicate that high-side refrigerant gas cooler (condenser in Rankine machine) pressure variations will affect both cooling capacity and efficiency. There is an optimum pressure level which largely depends on exit temperature of the cooler and total internal volume of the air conditioning equipment. Further investigation or experimentation will help to resolve these issues.

TABLE 4. High Ambient Modeling Results Comparison R-410A vs. R-22 (12 SEER)

Outdoor Air Temperature [F]	Refrigerant	Capacity Btu/hr	EER	Condenser Temp. Details [F] Inlet	Saturation	Outlet	Air Temp. [F] Leaving Coil
95	R-410A	35992	10.7	180	122	107	112
	R-22	36047	11.0	187	118	102	109
115	R-410A	31989	7.7	208	141	126	132
	R-22	32999	8.3	218	137	121	129
125	R-410A	29531	6.4	224	151	135	141
	R-22	31386	7.2	236	147	131	139

CONCLUSION

Modeling studies were performed to predict the cost and performance benefits of a new , environmentally friendly refrigerant, R-410A. R-410A units can be successfully designed to replace R-22 in unitary air conditioning equipment with potential savings in direct materials and labor. With current technology, a higher efficiency unit can get a significant benefit from switching R-22 to R-410A while a lower efficiency unit, such as 10 SEER unit may realize less benefit due to estimated higher compressor and refrigerant costs. If the National Energy Efficiency Standard were changed, it might become an additional driver for encouraging OEM toward producing more R-410A units. Further study in high ambient condition might be needed for R-410A equipment.

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