Modeled and Measured Effects of Compressor Downsizing in an Existing Air Conditioner/Heat Pump in the Cooling Mode

William P. Levins

C. Keith Rice, Ph.D.
Member ASHRAE

Van D. Baxter, P.E.
Member ASHRAE

ABSTRACT

It is not uncommon to find oversized central air conditioners in residences. Heating, ventilating, and air-conditioning (HVAC) contractors sometimes oversize central air conditioners for one reason or another—some to the point that they may be 100% larger than needed to meet the load. Retrofit measures done to improve house envelope and distribution system efficiency also contribute to HVAC oversizing, as they reduce house heating and cooling loads. Proper sizing of an air conditioner or heat pump allows more efficient operation and provides a more comfortable environment than a highly oversized unit. Another factor that lowers operating efficiency is an improper refrigerant charge. Field inspections have revealed that about half of the units checked were not properly charged.

An option available to homeowners with oversized air conditioners is to replace the existing compressor with a smaller, more efficient compressor rather than purchasing a new, smaller unit. Such a retrofit may be economically justified, especially during a compressor failure, provided the oversizing of the existing unit is not too great.

A used, 15-year-old, single-package heat pump with a capillary tube expansion device on the indoor coil was purchased and tested in a set of environmental chambers to determine its cooling performance under various conditions. The system was also modeled to estimate its existing performance and that with two different types of retrofitted state-of-the-art (SOA) efficient compressors with about 30% less capacity than the original compressor. This reduced the overall system cooling capacity by about 20%.

Modeling estimated that the retrofit would increase the system's energy efficiency ratio (EER) at 95°F (35°C) by 30%, increase the seasonal energy efficiency ratio (SEER) by 34%, and reduce power demand by 39% compared to the existing unit. Reduced cycling losses account for the higher increase in SEER.

The proper refrigerant charge of the as-received unit—determined using superheat, operating pressures, and EER as guidelines—was 22% higher than the nameplate charge. After testing, the existing compressor was replaced with one of the 30% smaller SOA compressors that had been modeled. Further testing confirmed that a 33% increase in EER was attained, compared to the predicted 30%. Power demand was reduced 38% compared to the predicted 39%.

The authors found that the surest way to obtain a proper refrigerant charge on the unit was to use a set of gauges coupled with superheat measurements.

INTRODUCTION

Several studies have shown that oversized, contractor-installed residential central air conditioners are not uncommon in the U.S. and that the extent of the oversizing can approach 100% in some cases (Proctor et al. 1995; Neal and O'Neal 1992). Proper sizing allows more efficient operation and attains better moisture removal (provides a more comfortable home environment) than oversizing, which in turn leads to more economical operation for the homeowner and less demand for the electrical utility. Another factor indirectly contributing to the oversizing of an existing HVAC system is any retrofit measure done to improve house envelope and distribution system efficiency, as they reduce house heating and cooling loads from preretrofit values.

Proper sizing is, however, a somewhat nebulous term. Proctor defines a properly sized air conditioner for a dwelling as one that will start to run continuously when at the 2.5% design dry-bulb and mean coincident wet-bulb temperatures of a location from June through September with indoor conditions of 75°F DB/62°F WB (23.8°C/16.6°C). The *Manual J* (ACCA 1986) sizing—probably the most popular of the various analytical sizing methods—uses 95°F (35°C) as the design temperature for the start of continuous unit operation in southern cities.

William P. Levins is a member of the senior development staff, **C. Keith Rice** is a research engineer, and **Van D. Baxter** is a group leader at Oak Ridge National Laboratory, Oak Ridge, Tenn.

However, it predicts a load that is about 25% higher than the true load, according to Neal and O'Neal (1992). Neal and O'Neal (1992) define a properly sized unit as one that does not allow the indoor temperature to exceed 80°F (26.6°C) for more than 1% of the cooling season when the thermostat is set at 78°F (25.5°C). Some contractors recommend installing a unit that is one size larger than *Manual J* estimates, just to be safe, which further increases the oversizing.

All sizing estimates naturally assume a properly operating unit, inferring that the unit contains a proper refrigerant charge. The refrigerant charge does indeed affect air-conditioning performance, and the proper charge for each unit is set by the manufacturer based on the optimum system EER while operating at reasonable system pressures and temperatures. Proctor and Downey (1995b) reported findings from studies concerning refrigerant charges in residential systems in California:

- One study found 31% of the units were undercharged and 69% were either properly charged or overcharged.
- Another study in 1990-1991 found about 60% of the units were undercharged or overcharged (slightly more were overcharged), and 40% were properly charged.
- A third study found 22% undercharged units, 33% overcharged units, and 45% properly charged units.

These studies suggest that there are a significant number of air conditioners in the field that are relatively new (10 years old or less) but are oversized and/or improperly charged. One option available to a homeowner with an oversized system is to replace the existing compressor with a state-of-the-art (SOA) efficient compressor with less capacity instead of replacing the entire existing system. This process may reduce the oversizing, but it raises the question as to how this retrofit affects system performance. If the oversized system in question is a relatively new high-efficiency model, downsizing could adversely affect its moisture-removal capacity.

The purpose of this project was to try to answer the above question as well as to determine the proper charge for a unit that undergoes such a retrofit. Note that English units are used throughout this paper in order to minimize clutter and avoid confusion; however, Table A4 in the appendix contains the necessary factors and algorithms to convert values to SI units.

APPROACH

We modeled an existing system operating in the cooling mode with a heat pump design program developed at a national laboratory and then attempted to verify the modeling predictions experimentally. We purchased a used 15-year-old, nominal 3-ton single-package heat pump from a local HVAC contractor to use as a sample for this work. The modeling work was based on this system both before and after being retrofitted with a new SOA compressor with about 30% smaller capacity than the original.

The experimental testing was conducted in a set of side-by-side climate chambers that can control outdoor and indoor temperature and humidity conditions to within ±1°F and 2% relative humidity (RH), respectively. The authors' computer-based data-acquisition system can continuously monitor refrigerant, component, air conditions, and power consumption and demands of the various system components and store the data in a file for later analysis.

Initial operation of the heat pump was used to calibrate the computer model as well as to measure existing system performance. The existing compressor was then replaced with a new, smaller-capacity SOA compressor, the proper charge was determined, and the system was tested at several ambient temperatures to verify performance.

MODELING PHASE

Existing Equipment

The existing 37-kBtu/h (cooling) single-package heat pump was simulated using manufacturer's component information (GE 1977a) supplemented with direct measurements of heat exchanger face areas and line lengths. A six-coefficient compressor representation (Fischer and Rice 1983) was obtained using the manufacturer's compressor map data, and the manufacturer's charging chart was used to calibrate the heat exchanger and compressor performance for the heat pump design model (HPDM) (Rice 1991). The HPDM calibration was refined using additional suction line pressure drop to account for the accumulator and reversing valve, after which the manufacturer's rated capacity, EER, and power draw were within 1% of modeled predictions.

Analysis indicated that the required rating for a reciprocating compressor to reduce system design capacity 20%, from 37 to 29.6 kBtu/h, was 31.5 kBtu/h—28.4% smaller than the 44-kBtu/h compressor in the original unit. The extra 8.4% compressor size reduction is needed to offset the capacity-beneficial effects of the lower pressure ratio and higher suction pressure in the downsized system. SOA reciprocating and scroll compressors closest to the required capacity at standard ARI 520-90 (ARI 1990) rating conditions were selected from U.S. manufacturers' product information. Rating sheets for SOA scroll and reciprocating models were obtained, as well as performance representations based on ARI 540-91 (ARI 1991). Table 1 shows a comparison of the rated performance and size of the as-built compressor to the two candidate SOA retrofit compressors. The rated EERs of the SOA compressors are 25% and 19% higher than the original equipment.

TABLE 1 Comparison of Compressors Used in Modeling

Compressor Type	Original - Recip.	SOA - Recip.	SOA - Scroll
Rated EER (Btu/W)	9.43	11.8	11.1
Rated Capacity (kBtu/h)	44	31.5	32

Note: ARI 520-90 rating conditions are conducted at conditions of $T_{evap}/T_{cond}/S_{heat}/S_{cool}$ = 45/130/20/15°F, respectively.

Component Performance Comparisons

Performance analyses were done for as-built and downsized retrofit designs at 82°F, 95°F, and 115°F (27.7°C, 35°C, and 46.1°C) ambient temperatures for standard ARI 210/240-89 (ARI 1989) indoor cooling conditions of 80°F/67°F (26.6°C/19.4°C) DB/WB. These ambient conditions correspond to SEER rating, design cooling capacity, and peak cooling conditions, respectively. Table 2 contains the performance comparisons among the three designs.

The HPDM predicted an SHR rise from 0.715 to 0.757 with no reduction in indoor airflow, so the indoor airflow rate was reduced by 20% to provide the same indoor airflow per unit capacity (cfm/ton) to obtain an equivalent sensible-to-total heat ratio (SHR). The airflow reduction was accomplished by adding more pressure drop in the duct system and using the medium-speed tap of the indoor blower motor. This minimized the fan power to 472 W at 960 cfm from the original 550 W at 1,200 cfm.

The analyses showed that condenser saturation temperature at design conditions would drop 8°F and the condenser exit condition would change from 7°F subcooled to 0.5% quality with the existing capillary tubes. The evaporator saturation temperatures would rise only 1.6°F to 2.1°F because reduced indoor airflow was used to maintain equivalent dehumidification. Simulations using more restrictive capillary tubes showed that the EER at 95°F would improve less than 1%, while capacity would increase by about 2.5%.

Compressor efficiency comparisons show that the SOA reciprocating compressor has a higher EER rating than the scroll compressor, 11.8 to 11.1 (Table 1), and also has higher isentropic efficiencies (Table 2). The scroll almost equals the SOA reciprocating compressor at the 82°F (27.7°C) SEER rating point but fares progressively worse at the 95°F (35°C) nominal capacity rating and at extreme ambient conditions, 115°F (46.1°C).

System Modeling

Table 3 compares system performance between the as-built unit and the 20% downsized retrofit designs for the three cooling conditions. The SOA reciprocating design has a slightly higher SEER than the scroll, and power demand of the scroll is 1.5% to 5% higher at the 95°F (35°C) and 115°F (46.1°C) conditions, respectively. Sensible heat ratios are given at all three ambients and the results show predicted dehumidification to be essentially the same as for the original equipment.

SEER values are provided for both the U.S. Department of Energy (DOE) standard procedure (DOE 1979) and the alternative bin approach. The default DOE sizing procedure was used in all but the bin analysis for the as-built unit. The assumed load was reduced by 20% for all bins to approximate the oversizing scenario used for the compressor retrofit analysis. This simulates the as-built unit becoming oversized by an additional 25% above the standard DOE sizing, once the loss and load reduction are made.

The downsized SOA retrofit designs save energy both by reducing cycling losses and by improving steady-state system coefficient of performance (COP). The improvements in steady-

TABLE 2 Modeled Component Cooling Performance Analysis

Compressor Type	As-Received Recip.			SOA Recip.			SOA Scroll		
Cooling Ambient Temp.	82°F	**95°F**	115°F	82°F	**95°F**	115°F	82°F	**95°F**	115°F
System Capacity (kBtu/h)	39.3	**37.0**	32.6	31.4	**29.6**	26.1	30.7	**29.4**	26.5
COMPRESSOR									
Isentropic Efficiency (%)	54.5	**55.2**	55.0	70.7	**71.1**	68.9	69.4	**68.9**	66.2
Volumetric Efficiency (%)	73.3	**70.9**	66.7	80.1	**77.3**	73.3	96.9	**95.7**	93.6
Pressure Ratio	3.24	**3.55**	4.00	2.81	**3.11**	3.53	2.77	**3.11**	3.58
HEAT EXCHANGERS									
Sat. Suction Temp. (°F)	39.7	**43.1**	48.3	41.4	**44.7**	50.4	41.7	**44.9**	50.2
Sat. Disch. Temp. (°F)	118	**130**	148	110	**122**	140	109	**122**	141
Condenser Subcooling (°F)	12.0	**7.1**	(0.2)	3.3	**(.52)**	(5.3)	2.4	**(.65)**	(5.1)
Evaporator Superheat (°F)	20	**14**	5	20	**14**	(99)	20	**14**	(99.4)
INDOOR UNIT									
Sensible Heat Ratio	.691	**.715**	.75	.691	**.716**	.767	.695	**.718**	.762
Airflow (cfm)1200	1200	**1200**	1200	960	**960**	960	960	**960**	960
Airflow (cfm/ton)[1]	389	**389**	389	389	**389**	389	392	**392**	392
Airflow (cfm/ton actual)	366	**389**	441	367	**389**	441	375	**392**	434
Fan Speed Selector	Med.	**Med.**	Med.	Med.	**Med.**	Med.	Med.	**Med.**	Med.
Indoor Fan Power (W)	550	**550**	550	472	**472**	472	472	**472**	472
External ΔP (in. H_2O)	0.3	**0.3**	0.3	0.88	**0.88**	0.88	0.88	**0.88**	0.88

Notes: Numbers in parentheses ( ) refer to refrigerant quality in percent. Bold numbers correspond to design rating temperature @ 95°F.
[1] cfm/ton is usually based on nominal capacity at 95°F.

TABLE 3 Modeled Comparison of As-Built System and Downsized Retrofit System

Compressor Used	As-Built Recip.	SOA Recip.		SOA Scroll	
82°F Cooling	**Value**	**Value**	**% Diff.**	**Value**	**% Diff.**
Capacity (kBtu/h)	39.3	31.4	−20.1%	30.7	−21.9%
EER (Btu/W)	7.59	9.92	30.7%	9.82	29.3%
Sensible Heat Ratio	0.691	0.691		0.695	
SEER-DOE (0.25 C_d)	6.64	8.68		8.59	
SEER-BIN (0.25 C_d)	**6.43**	**8.64**	**34.4%**	**8.58**	33.4%
95°F Cooling					
Capacity (kBtu/h)	**37.0**	**29.6**	−20.0%	**29.4**	−20.5%
EER (Btu/W)	6.69	8.72	30.3%	8.45	26.3%
Power Demand (kW)	5.54	3.39	−38.8%	3.48	−37.2%
Sensible Heat Ratio	0.715	0.716		0.718	
115°F Cooling					
Capacity (kBtu/h)	32.6	26.1	−19.9%	26.5	−18.7%
EER (Btu/W)	5.34	6.93	29.8%	6.51	21.9%
Power Demand (kW)	6.1	3.77	−38.2%	4.07	−33.3%
Sensible Heat Ratio	0.750	0.767		0.762	

Note: % Diff. = 100 · (1 − value/as-built recip. value).

state COP result from the higher compressor efficiencies and from the lower pressure ratios due to heat exchanger unloading. Unit downsizing reduces cycling losses and heat exchanger loading, while the higher EER compressor raises the power efficiency of the compressor.

Retrofitting the original package system with an SOA reciprocating compressor with 28.4% less capacity results in a 34.4% higher SEER than the original unit for meeting the reduced cooling load and lowers the energy required to meet the reduced seasonal load by 25.6%.

Of the 34.4% higher SEER, 3.9 percentage points (11.2%) are from cycling loss reduction, 10.4 percentage points (30.1%) are from heat exchanger unloading, and 20.2 percentage points (58.7%) are from a more efficient compressor. We chose to use the reciprocating compressor for our retrofit on the basis of our modeling.

System gains from a higher efficiency compressor are not as large in percentage as the increases in compressor power (isentropic) efficiency noted earlier because the compressor consumes only part of the total system power. Fan power is 17.4% of the input power for the original equipment, but it increases to 26.2% as the SOA reciprocating compressor power is reduced—the outdoor fan power remains at 415 W and the indoor blower power drops from 550 to 472 W.

Although not evaluated experimentally, the increasing percentage of the total power from the fans in the downsized designs suggests that lower-speed, smaller-horsepower replacement fan motors would operate at higher efficiencies with lower pressure drops. Further modeling of the SOA reciprocating design with resized, higher efficiency (from 55% to 65%) SOA fan motors predicted SEER-DOE and SEER-BIN increases to 9.94 and 9.92, respectively—an additional 19.9% increase in seasonal efficiency over the oversized as-built unit. The analysis suggests that a compressor and fan motors retrofit on a 15-year-old single-package unit could exceed the 9.7 SEER required of packaged units in 1993 by the National Appliance Energy Conservation Act (NAECA 1987). The authors do not recommend such fan motor retrofits at present for many reasons but point out that they might be a source of additional retrofit savings.

Summary of Modeling of Downsized SOA Retrofit Designs

Total energy savings for the modeled compressor retrofit includes savings from both load reduction and more efficient equipment operation. A 20% load reduction does not result in a 20% energy savings with the original heat pump because cycling losses increase as the unit becomes oversized. A bin analysis on the oversized heat pump with a default cycling degradation factor (C_d) of 0.25 (DOE 1979) predicts an energy savings of only 17.4% from the load reduction.

The retrofit SOA compressor lowers energy use by 25.6% (from the improved equipment) relative to the oversized unit *when meeting the reduced seasonal house load*. Relative to *before load reductions*, the system with an efficient downsized SOA reciprocating compressor yields a 21.1% energy savings.

The 17.4% energy savings from the building load reduction and the 21.1% energy savings from unit downsizing with a more efficient compressor combine to give a 38.5% reduction in energy use relative to the original heat pump/building combination. This 38.5% drop in energy use is the net effect of a 20% load reduction, 20% compressor downsizing, and a more efficient compressor. Of the 38.5% energy savings, 45.2% comes from the load reduction and 54.8% from the combined effects of compressor downsizing and efficiency upgrade. Peak power draw at the 95°F (35°C) and 115°F (46.1°C) conditions is also reduced by 38% relative to the original heat pump/building combination.

EXPERIMENTAL TESTING

Test Setup

Figure 1 is a photograph of the original heat pump after it was externally washed and the heat exchangers were cleaned. The unit was instrumented with thermocouples, thermopiles, pressure transducers, and watt transducers so that all points of interest were monitored. Refrigerant temperatures were measured with type-T thermocouples strapped to tube walls and covered with insulating tape. Sensible capacity was calculated from air-side temperature measurements from nine-point type-T thermopiles and an airflow measurement from a parallel-cell, honeycombed grid with a multipoint pitot tube array located on the inlet air side. Latent capacity was measured by collecting and weighing condensate. The monitoring system scanned each channel every five seconds and output averaged one-minute readings to a data file. Pressure transducers and the airflow-measuring array were calibrated by instrument technicians after the heat pump was installed in the climate chambers. The heat pump and the climate chambers were checked to ensure there was no air leakage between indoor and outdoor sections. Figure 2 is a schematic diagram of the unit operating in the cooling mode and shows thermocouple locations.

Initial Testing

After the external unit and heat exchangers were cleaned, the as-received heat pump was pumped down to remove and recover the existing charge, and was then recharged with eight pounds of R-22, the manufacturer's recommendation. It became readily apparent that something was not correct when the unit was started, as the discharge pressure was too high, the suction pressure was too low, and the capacity was too low. Adding more refrigerant raised the head pressure much faster than it raised the suction pressure. We suspected constricted capillary tubes, so they were removed and cleaned. However, they did not appear to be either fouled or bent. Further inspection of the unit by the refrigeration mechanics revealed a partially blocked filter-dryer on the high-pressure (liquid) side. It was replaced, the unit recharged to eight pounds of R-22 again, and testing started.

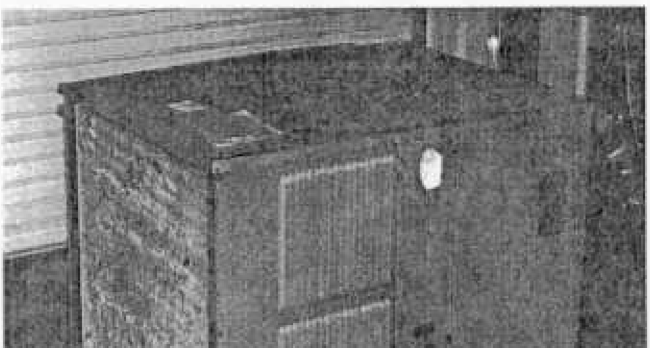


Figure 1 *As-received single-package 15-year-old heat pump.*

Table A1 contains the results of cursory testing to determine system performance at 95°F (35°C) outdoor conditions with different levels of charge. Note that the capacities and performance figures in Table A1 contain sensible numbers only—no latent data were taken for these scoping runs. Service information data at 95°F (35°C) outdoor dry-bulb and 80°F (26.6°C) and 67°F (19.4°C) indoor dry-bulb and wet-bulb temperatures (50% RH) predict suction and discharge pressures of 74 and 295 psig, respectively. Our testing showed that the proper charge was not 8 pounds but somewhere between 9.25 and 10 pounds of R-22. We decided to use 9.75 pounds as the proper charge based on capacity, superheat, and EER results. After a few runs were made, a review of our data revealed that an extra ounce of R-22 had mistakenly been added to the system during charging, so we continued our testing with 9.81 pounds of R-22 instead of the planned 9.75 pounds.

A series of tests was conducted at 82°F (27.7°C) and 95°F (35°C) outdoor air, 80°F (26.6°C) indoor air at 50% RH, and several indoor airflows to determine system performance. Table A3 contains the raw data and the effects of charge on capacity and EER. It shows that the unit was only delivering about 32,000 Btu/h at 95°F (35°C), or 86% of its rated capacity of 37,000 Btu/h, and that it needed substantially more charge than the 8.0-pound nameplate-recommended charge. Before the clogged filter/dryer was replaced, the capacity was substantially lower, the head pressure was about 50 psi higher, and the suction pressure was about 20 psi lower. It is obvious that the original unit was not performing at its original rated levels, but we did our best to optimize the system without doing any excessive rebuilding. Since the unit produced consistent data, we decided to continue with our testing.

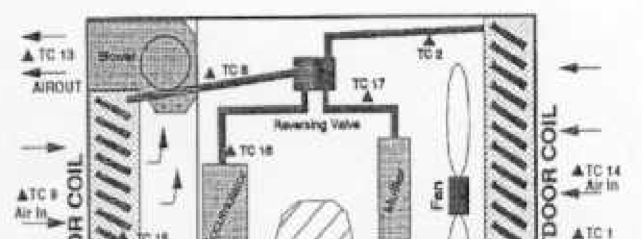


Figure 2 *Schematic diagram of heat pump showing thermocouple locations.*

After the original compressor was removed, the new down-sized SOA reciprocating compressor was installed and another series of short testing was done to determine the correct charge for this system configuration. Note that it is standard practice to replace the liquid-line filter-dryer during a compressor change-out, so our original clogged filter-dryer would have been replaced during this operation. Table A2 contains the results of this testing, from which we concluded that about 8.75 pounds of R-22 is the optimum charge for this combination—one pound less than was used with the original compressor. Note again that this table only contains sensible performance data.

A series of tests followed with 82°F, 95°F, and 115°F (27.7°C, 35°C, and 46.1°C) air entering the condenser, while maintaining 80°F (26.6°C) and 50% RH indoor air for various refrigerant charges and airflow rates. Table A3 contains the results of these tests. The most striking aspect of these data is that the capacity and system EERs both increase with charge at 82°F (27.7°C) and 95°F (35°C) but decrease with charge at 115°F (46.1°C). Perhaps abnormally high vapor velocities are entraining some liquid refrigerant from the evaporator at the higher temperature. Some of the capacity and EER scatter around a given charge and outdoor temperature are the result of differing indoor airflow rates at a given charge, while some scatter is normal experimental error from instrument calibrations, precision, etc.

COMPARISON OF PREDICTED AND MEASURED PERFORMANCE

The task of comparing predicted and measured results seems fairly straightforward, but it turned out to be somewhat difficult because the measured capacity of our original unit was so much lower than its rating. To make a fair comparison of predicted and measured performances, it was necessary to select experimental data from Table A3 that corresponded closely with those actual (not nominal) airflow rates/ton specified in Table 2 for the original and retrofitted SOA reciprocating compressors. Table 4 contains the basic data for this comparison.

TABLE 4 Modeled vs. Measured Performance at 95°F (35°C) Outdoor, 80°F/67°F (26.6°C/19.4°C) Indoor DB/WB

	Original Unit with 9.81 lb R-22		Retrofit Unit with 8.75 lb R-22	
	Predicted	Measured	Predicted	Measured
Capacity (kBtu/h)	37.00	31.92	29.60	26.40
Electric Demand (kW)	5.54	5.49	3.39	3.41
Sensible Heat Ratio	0.72	0.72	0.72	0.75
Airflow (cfm/ton)	389	421	389	407
EER	6.69	5.82	8.72	7.74

A capacity comparison of the measured retrofitted and original systems at 95°F (35°C) with optimum charges of 8.75 and 9.81 pounds R-22, respectively, and indoor flow rates of about 400 cfm/ton actual shows that the retrofitted unit capacity of about 26.4 kBtu/h is 17% less than that of the original unit, 31.9 kBtu/h. An EER comparison shows the retrofitted unit at about 7.74, or 33% higher than that of the original 5.82.

Figures 3 through 5 plot the data for capacity, electrical demand, and EER from Table 4 along with similar predicted and measured data at 82°F (27.7°C) and 115°F (46.1°C). These figures show that the model predicts the relative differences in capacity and EER (i.e., the percentage change) fairly consistently at 95°F (35°C) and even at 82°F (27.7°C) and 115°F (46.1°C). The model predicts the electrical demand to within 1%. This leads one to conclude that had the unit been operating at rated capacity, the modeling results would be close to the measured results. It follows that the predicted savings from the modeling are reasonable estimates of what to expect from retrofitting an existing unit with a smaller, more efficient compressor.

Based on our experience, there are most likely a reasonable number of originally oversized units in the field operating at reduced capacity and efficiency for one reason or another (leaky ducts, fouled heat exchangers, refrigerant leaks, clogged filter-dryers, etc.) that may not be as oversized as they appear. This condition would reduce cycling losses but the demand would essentially not change, especially at higher ambients, because of operation at a lower EER. One would basically be paying the same operating costs for reduced performance.

DETERMINATION OF PROPER CHARGE

A proper charge is necessary to ensure the optimum performance of a system and also ensure that the compressor operates in a safe manner. Too much refrigerant in a system will increase the head pressure and cause the compressor to work too hard. It will also promote supplying liquid to the suction side of the compressor, which is dangerous to reciprocating compressors. This is why manufacturers always recommend that the refrigerant entering the compressor be superheated, that is, contain no liquid refrigerant.

Too little charge will cause a system to operate below rated capacity and can also be dangerous to hermetic compressors, which are in most residential systems. These rely upon the refrigerant for internal cooling of the motor and windings and also to return any oil leaving the compressor. Insufficient refrigerant mass flow through the compressor means increased operating temperatures and less lubrication for the compressor and hence reduced life. The volumetric flow can be high, but the suction gas enters at a lower pressure than normal and is highly superheated, which reduces the mass flow into and also increases the pressure ratio across the compressor—factors that lower efficient operation.

Therefore, the superheat of the suction gas to the compressor is a quantity that can be used to determine the condition of the refrigerant charge—with certain limitations, such as the type of expansion device and the existing load on the system (usually determined by the outdoor temperature).

Procedures

As their first choice to obtain a proper charge in the system, both Proctor and Downey (1995) and manufacturers recommend

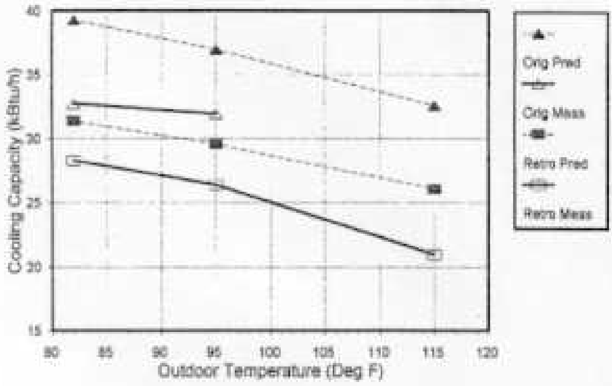


Figure 3 *Measured and predicted capacities.*

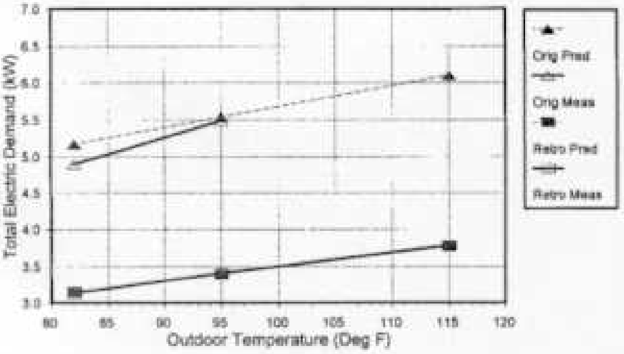


Figure 4 *Measured and predicted electric input.*

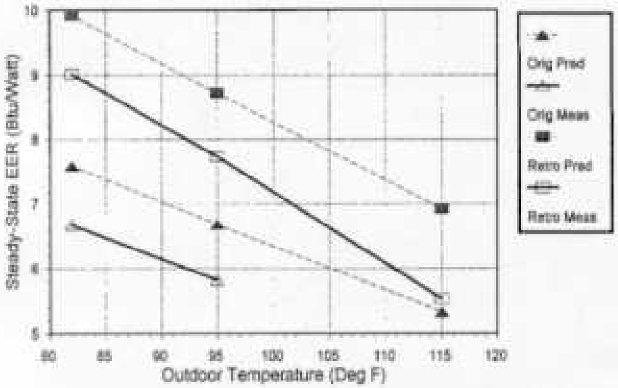


Figure 5 *Measured and predicted system steady-state EER.*

properly evacuating a system and charging it with the charge listed on its nameplate. This can often be time consuming and expensive to follow and is not always feasible for a poorly equipped servicer to perform properly. However, it is usually the recommended method to follow to obtain a proper charge.

Manufacturers usually supply charging charts with their systems that are based on outdoor dry-bulb temperatures and indoor air wet-bulb temperatures for cooling-mode operation. They correlate these parameters with corresponding suction and discharge pressure readings obtained from their units operating with given indoor airflow rates, outdoor temperatures, and indoor humidity (for cooling). Such data are invaluable, especially if the indoor coil has a thermal expansion valve expansion device, but such charts are not always available on-site.

The service information charging chart supplied by the manufacturer of the heat pump used in this testing contains a procedure for charging units containing capillary tube expansion devices on indoor coils in cooling-mode operation (GE 1977b). Essentially the same procedure is being promoted by Proctor and Downey (1995) as an in-the-field method of charging air conditioners/heat pumps with indoor unit capillary tube expansion devices. A summary of this procedure is contained in the appendix. Proctor and Downey recommend not using gauges at all since they extract some charge from the system (newer gauge sets minimize this, and adapters are available for older gauge sets to prevent losses). They recommend measuring the saturation temperature halfway up the indoor coil with a thermocouple, which should be the more accurate method. However, all indoor coils are not readily accessible and not all refrigeration repairpeople have a thermocouple or equivalent temperature-indicating device. They are more likely to have a set of gauges, but not necessarily accurate gauges. Proctor and Downey and the service information chart say to measure the suction temperature where the suction line enters the outside unit housing or just before the suction line accumulator for a single-package unit.

Our Experience

After repairing the original test system, we charged it with the nameplate charge of 8 pounds of R-22 and found the capacity to be much lower than the rated 37 kBtu/h (see Table A3, Test ID's Ret-08, 49, and 50). The expected superheat from a properly charged unit from the charging chart in the appendix (also included in the last column in Table A3) is listed as 14°F (−10°C) for 95°F (35°C). Our measured superheat averaged 23°F (−5°C) for these conditions, indicating that the nameplate charge was too low for the unit in its current condition. The service information charging chart (not included here) said to expect suction/discharge pressures of 73/290 psig, respectively, while we measured 63/276 psig, also indicating that we were low in charge. Table A1 contains some measured pressure and superheat data for the original unit and shows that proper charge for the unit is somewhere between 9.5 and 10 pounds of R-22, based on both superheat and suction/discharge pressures. We decided upon 9.75 pounds of R-22 since we measured a higher sensible capacity there than at 9.5 pounds R-22 and lower pressures than at 10 pounds R-22.

Since no data were available for the retrofit unit, we had to rely upon the superheat chart suggestion—14°F (−10°C)—and our modeled pressures (from the saturation temperatures in Table 3)—75/267 psig. Table A2 contains the results of our testing. We selected 8.75 pounds of R-22 as the optimum charge based on these data.

Conclusions on Charging

We have a sample of one unit on which to base our opinions, but our experiences were real and informative. We had an advantage in our situation in that we had access to two refrigeration mechanics and had guidance from system modeling before working on it. The flow restriction in the filter-dryer was not expected and led us to wrongly suspect clogged capillary tubes as the reason for our initial low-capacity, high-discharge-pressure problem. Without pressure gauges we could easily have added refrigerant blindly, not recognizing the restriction problem. Using pressure gauges on the system was beneficial because they showed us that we had a high discharge pressure as well as a low suction pressure. We therefore feel that initially using a gauge on the suction side, as recommended by the service information charge checkout procedure, is a good idea because it can be a useful diagnostic. However, using a gauge on the discharge side was equally informative in our situation. All gauges used should be of the newer leak-free (almost) design. We also agree that attaching a temperature sensor halfway up the evaporator coil is good to determine the saturation pressure. The temperature difference between the sensor on the evaporator and that at the suction-line accumulator, or the refrigerant superheat, agreed well with the difference between the saturation temperature obtained from the suction-line pressure tap and the temperature reading from the sensor at the suction-line accumulator, especially so when close to the proper charge.

Since it is not always easy to attach a temperature sensor in the middle of the evaporator, using a gauge on the suction-line pressure tap to derive the suction temperature may be the preferred way to obtain this reading. It does, however, require an accurate suction gauge to get the reading. Since a temperature reading is also necessary, the temperature gauge also must be accurate. If two temperature sensors are used to obtain the superheat, the temperature meter reading then need not be as accurate, since the difference between the two readings is used.

We did not get proper refrigerant operating conditions when the nameplate charge was installed in our unit, even after replacing the restricted filter-dryer. We found no other obvious problems, so we continued our experiment with the unit as is, after cleaning the capillary tubes. We followed the manufacturer's charging procedures with pressure gauges attached to the unit. After we obtained the stated pressures, the superheat closely matched the recommended superheat, albeit at a higher-than-recommended nameplate charge.

We had to use gauges on our system as well as the manufacturer's chart of operating pressures to arrive at the proper charge. The superheat method will most likely work well (for units with capillary tube expansion devices on the indoor coils), provided the outdoor temperature is sufficiently high and the unit is working well. Using pressure gauges coupled with superheat measurements appears to be the surest and safest method to obtain a proper refrigerant charge.

CONCLUSIONS

It is feasible to retrofit a 30% smaller SOA compressor to an existing air conditioner (reducing its cooling capacity by about 20%) and obtain improved efficiency and reduced electrical demand yet still meet house load and comfort conditions. Our modeling and experimental testing confirmed this, although not without some initial problems. After replacing a partially clogged filter-dryer, the unit only reached 85% of rated performance and this with a charge 20% greater than that recommended by the manufacturer. Our modeling predicted an efficiency (EER) gain of 30% and we measured a gain of 33% when the original compressor was replaced with a smaller, high-efficiency model. This measured EER increase corresponds to a 38% decrease in electricity demand at 95°F (35°C)—a factor of considerable interest to electrical utilities.

The nameplate refrigerant charge was much too low for our original system—we don't know why. The best way to obtain a proper charge in our original unit with a capillary tube expansion device was by using pressure gauges and a charging chart based on operating pressures. The superheat method also worked well, especially as we neared the proper charge, and was the only method to use for the retrofitted unit, where no pressure/operating data were available. Using both newer leak-free pressure gauges and the superheat method is the safest procedure to follow for general use.

We recommend follow-on compressor downsizing work that models and tests air conditioners using short-orifice tubes and thermal expansion valves on their evaporator coils. Modeling and testing of heat pumps in the heating mode with retrofitted, downsized efficient compressors should also be done. Such modeling and laboratory testing will provide technical answers and generate procedures for retrofit downsizing options. One option involves an existing oversized unit that is a relatively new high-efficiency model in which moisture removal could be a problem. Field testing of the retrofit downsizing concept in different sections of the country should follow to evaluate the feasibility of compressor downsizing, including the practical problems of reliability and service problems. The feasibility of accompanying indoor and outdoor fan motor retrofit downsizing should also be investigated.

REFERENCES

ACCA. 1986. *Manual J residential load calculation*, 7th ed. Washington, D.C.: Air Conditioning Contractors of America.

ARI. 1989. *ARI Standard 210/240-89, Standard for unitary air-conditioning and air-source heat pump equipment*. Arlington, Va.: Air-Conditioning and Refrigeration Institute.

ARI. 1990. *ARI Standard 520-90, Standard for positive displacement refrigerant compressors, compressor units, and condensing units*. Arlington, Va.: Air-Conditioning and Refrigeration Institute.

ARI. 1991. *ARI Standard 540-91, Standard for method of presentation of compressor performance data*. Arlington, Va.: Air-Conditioning and Refrigeration Institute.

DOE. 1979. Test procedures for central air conditioners, including heat pumps. 10CFR Part 430. *Federal Register* 44(249): 76700-76723. Washington, D.C.: Department of Energy.

Fischer, S.K., and C.K. Rice. 1983. The Oak Ridge heat pump models: I. A steady-state computer design model for air-to-air heat pumps. ORNL/CON-80/R1, August. Oak Ridge, Tenn.: Oak Ridge National Laboratory.

GE. 1977a. *GE product data, single package Weathertron heat pumps*. Pub No. 22-1028-6. Tyler, Texas: The Trane Company.

GE. 1977b. *GE service information, single package heat pumps, BWC024,030,036B*. Pub No. 18-BB07E2. Tyler, Texas: The Trane Company.

NAECA. 1987. National appliance energy conservation act. Public Law 100-12, March 17. Washington, D.C.: U.S. Congress.

Neal, L., and D.L. O'Neal. 1992. The impact of residential charging and sizing on peak electrical demand. 8th Symposium on Improving Building Systems in Hot and Humid Climates.

Proctor, J., and T. Downey. 1995. Heat pump and air conditioner performance. Handout from Oral Presentation. Affordable Comfort Conference, Pittsburgh, Pa., March 26-31.

Proctor, J., Z. Katsnelson, and B. Wilson. 1995. Bigger is not better. *Home Energy Magazine* 7(3): 19-26.

Rice, C.K. 1991. Benchmark performance analysis of an ECM-modulated air-to-air heat pump with a reciprocating compressor. *ASHRAE Transactions* 98(1): 430-450.

APPENDIX

TABLE A1 Test of Original Reciprocating Compressor to Determine Proper Charge

OD Temp (°F)	R-22 Charge (lb)	Refrigerant Temp EvMid (°F)	Refrigerant Temp AccIn (°F)	Pressure Suct psig	Pressure Disch psig	Air Temperature EvDBi (°F)	Air Temperature EvDBo (°F)	Airflow (cfm)	Sensible Capacity (Btu/h)	Total Elec In (W)	Sensible Performance COP	Sensible Performance EER	Super heat (°F)
95	8.00	58	81	63	276	79.7	64.3	1342	21492	5264	1.20	4.08	23
94	8.25	57	80	64	276	79.5	64.0	1346	21698	5320	1.19	4.08	24
95	8.50	55	79	66	281	79.2	63.4	1346	22118	5376	1.21	4.11	25
95	8.75	49	78	69	287	79.0	62.5	1342	23027	5492	1.23	4.19	29
95	9.00	49	77	70	289	78.6	61.9	1346	23374	5533	1.24	4.22	28
95	9.25	49	76	72	294	78.5	61.2	1355	24365	5612	1.27	4.34	26
95	9.50	51	65	75	301	78.2	60.7	1316	23945	5707	1.23	4.20	14
95	9.75	51	70	75	303	79.0	61.4	1334	24409	5725	1.25	4.26	19
96	10.00	52	65	77	308	79.3	61.8	1342	24423	5790	1.24	4.22	13

TABLE A2 Test of SOA Reciprocating Compressor to Determine Proper Charge

OD Temp (°F)	R22 Charge (lb)	Refrigerant Temp EvMid (°F)	Refrigerant Temp AccIn (°F)	Pressure Suct psig	Pressure Disch psig	Air Temperature EvDBi (°F)	Air Temperature EvDBo (°F)	Airflow (cfm)	Sensible Capacity (Btu/h)	Total Elec In (W)	Sensible Performance COP	Sensible Performance EER	Super heat (°F)
95	8.00	49	78	73	259	79.2	61.8	1125	20356	3382	1.76	6.02	29
95	8.25	48	77	75	260	78.8	60.9	1125	20941	3415	1.80	6.13	29
95	8.50	50	75	78	266	78.9	60.9	1170	21894	3457	1.86	6.33	26
95	8.75	51	74	79	268	79.4	60.8	1170	22624	3492	1.90	6.48	20
96	9.00	52	60	81	271	79.7	61.5	1170	22138	3548	1.83	6.24	9

TABLE A3 Results of Compressor Exchange Testing

Test ID	Date	Compressor - System	R-22 (lbs)	OD Temp	CAPACITIES (Btu/h) Sens	Latent	Total	Power Input watts	SYSTEM COP	EER	AIRFLOWS cfm	cfm/ton	Superht (°F) Meas	Recm'd
Ret-12	04/20	SOA Recip	8.00	82	21327	5646	26973	3138	2.52	8.60	1150	512	21	22
Ret-44	05/09	SOA Recip	8.00	82	21359	5440	26799	3064	2.56	8.75	1001	448	20	22
Ret-34	04/27	SOA Recip	8.00	82	20232	4738	24970	3075	2.38	8.12	961	462	20	22
Ret-13	04/20	SOA Recip	8.00	95	20594	5337	25931	3398	2.24	7.63	1145	530	28	14
Ret-33	04/27	SOA Recip	8.00	95	19774	5974	25748	3378	2.23	7.62	964	449	28	14
Ret-43	05/09	SOA Recip	8.00	95	19233	6067	25300	3328	2.23	7.60	845	401	29	14
Ret-35	04/27	SOA Recip	8.00	115	18297	4738	23035	3782	1.78	6.09	964	502	29	3
Ret-48b	05/11	SOA Recip	8.00	115	19274	2827	22101	3802	1.70	5.81	976	530	13	3
Ret-48a	05/11	SOA Recip	8.00	115	19248	2744	21993	3806	1.69	5.78	976	533	13	3
Ret-45	05/09	SOA Recip	8.25	82	20877	6423	27300	3091	2.59	8.83	907	399	28	22
Ret-18	04/21	SOA Recip	8.25	82	22639	4283	26922	3158	2.50	8.53	1172	522	20	22
Ret-14	04/20	SOA Recip	8.25	95	21278	5833	27111	3430	2.32	7.90	1149	509	28	14
Ret-42	05/09	SOA Recip	8.25	95	20025	6236	26260	3373	2.28	7.79	893	408	27	14
Ret-46	05/09	SOA Recip	8.50	82	21577	6891	28468	3127	2.67	9.10	935	394	27	22
Ret-19	04/21	SOA Recip	8.50	82	22766	5178	27944	3186	2.57	8.77	1171	503	22	22
Ret-15	04/20	SOA Recip	8.50	95	21722	6151	27873	3475	2.35	8.02	1149	495	26	14
Ret-41	05/09	SOA Recip	8.50	95	20597	6423	27019	3405	2.33	7.94	931	413	25	14
Ret-23	04/21	SOA Recip	8.75	82	24162	5459	29621	3200	2.71	9.26	1079	437	26	22
Ret-20	04/21	SOA Recip	8.75	82	23485	5814	29299	3215	2.67	9.11	1170	479	26	22
Ret-31	04/27	SOA Recip	8.75	82	21476	7667	29143	3175	2.69	9.18	966	398	26	22
Ret-27	04/20	SOA Recip	8.75	82	21110	7190	28301	3129	2.65	9.04	888	376	25	22
Ret-47	05/10	SOA Recip	8.75	82	22706	5552	28259	3194	2.59	8.85	977	415	24	22
Ret-38	04/28	SOA Recip	8.75	82	20034	7490	27524	3124	2.58	8.81	809	353	20	22
Ret-39	05/09	SOA Recip	8.75	95	21709	6722	28431	3504	2.38	8.11	1153	487	25	14
Ret-16	04/20	SOA Recip	8.75	95	21747	6554	28301	3513	2.36	8.06	1150	488	23	14
Ret-24	04/21	SOA Recip	8.75	95	22937	5112	28050	3476	2.36	8.07	1070	458	21	14
Ret-40	05/09	SOA Recip	8.75	95	20978	6966	27944	3433	2.38	8.14	972	417	21	14
Ret-30	04/26	SOA Recip	8.75	95	21497	5206	26703	3433	2.28	7.78	970	436	16	14
Ret-28	04/20	SOA Recip	8.75	95	19700	6629	26328	3409	2.26	7.72	887	404	20	14
Ret-37	04/27	SOA Recip	8.75	95	18712	6208	24920	3386	2.16	7.36	829	399	6	14
Ret-26	04/21	SOA Recip	8.75	115	19123	2660	21784	3899	1.64	5.59	1149	633	1	3
Ret-25	04/21	SOA Recip	8.75	115	19647	2080	21727	3852	1.65	5.64	1070	591	2	3
Ret-32	04/27	SOA Recip	8.75	115	17693	3634	21326	3799	1.64	5.61	963	542	2	3
Ret-29	04/20	SOA Recip	8.75	115	17230	4008	21238	3784	1.64	5.61	892	504	4	3
Ret-36	04/27	SOA Recip	8.75	115	16214	4476	20690	3780	1.60	5.47	839	487	6	3
Ret-21	04/21	SOA Recip	9.00	82	23911	6282	30193	3249	2.72	9.29	1171	465	24	22
Ret-17	04/20	SOA Recip	9.00	95	21650	6629	28279	3542	2.34	7.98	1156	491	13	14
Ret-22	04/21	SOA Recip	9.25	82	24411	6395	30805	3277	2.75	9.40	1170	456	22	22
Ret-07	04/11	Orig. As-Rec	8.00	82	16139	1766	17905	4101	1.28	4.37	1362	913	26	22
Ret-06	04/11	Orig. As-Rec	8.00	95	16337	2014	18351	4511	1.19	4.07	1358	888	25	14

TABLE A3 Results of Compressor Exchange Testing (Continued)

Test ID	Date	Compressor - System	R-22 (lbs)	OD Temp	CAPACITIES (Btu/h)			Power Input watts	SYSTEM		AIRFLOWS		Superht (°F)	
					Sens	Latent	Total		COP	EER	cfm	cfm/ton	Meas	Recm'd
Ret-02	04/05	Orig. As-Rec	13.8	82	24583	6449	31032	5515	1.65	5.63	1371	530	24	22
Ret-03	04/05	Orig. As-Rec	13.8	95	24395	6379	30774	6227	1.45	4.94	1359	530	7	14
Ret-49	05/16	Orig.-Cln	8.00	95	21174	7003	28177	5029	1.64	5.60	1136	484	24	14
Ret-50	05/16	Orig.-Cln	8.00	95	20401	6685	27086	4987	1.59	5.43	929	412	25	14
Ret-08	04/18	Orig.-Cln	8.00	95	21482	5140	26622	5248	1.49	5.07	1345	606	23	14
Ret-11	04/18	Orig.-Cln	9.81	82	25608	8501	34109	5138	1.95	6.64	1349	475	27	22
Ret-53	05/16	Orig.-Cln	9.81	82	23296	9437	32733	4898	1.96	6.68	1137	417	26	22
Ret-51	05/16	Orig.-Cln	9.81	95	23387	8688	32075	5493	1.71	5.84	1130	423	25	14
Ret-52	05/16	Orig.-Cln	9.81	95	23073	8688	31761	5477	1.70	5.80	1112	420	26	14
Ret-09	04/18	Orig.-Cln	9.81	95	24499	6649	31148	5748	1.59	5.42	1340	516	19	14
Ret-10	04/18	Orig.-Cln	10.0	82	25694	8866	34560	5204	1.95	6.64	1348	468	25	22

NOTE: Inlet air to evaporator kept at 80°F DB/67°F WB. Recommended superheat is for an airflow of 400 cfm/ton.

TABLE A4 Conversion from I-P to SI Units

To Convert from	To	Multiply by (or algorithm)
Btu/h	kW	0.000293
kBtu/h	kW	0.293
EER	COP	0.293
lb_m	kg	0.454
cfm	L/s	0.472
cfm/ton	L/s·kW	0.134
°F	°C	(°F + 32)·0.556
F (Temp Difference)	C	0.556
psig	kPa	(psig + 14.7)·6.89
psi (Difference)	kPa	6.89

APPENDIX—SERVICE INFORMATION

Charge Checkout Procedure—Cooling Cycle 60 Hz for All Units Having Indoor Capillary

The following procedures should be used in checking proper refrigerant charge in systems having a capillary tube expansion device on the indoor coil. If equipped with a thermal expansion valve, use performance curves shown on previous pages to check charge.

Charge must be checked with outdoor fan in high-speed operation.

PROCEDURE

Charge checking must be done in cooling operation, with all panels in place and with stabilizing running conditions.

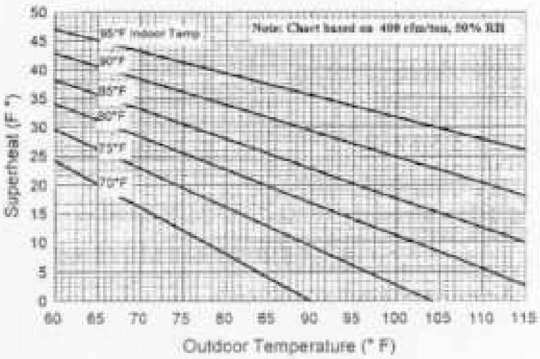
1. Connect suction pressure gauge—do not connect head pressure gauge, as this loses a measurable amount of R-22.
2. Measure suction-line temperature by securing the sensing bulb of a dial-type thermometer to the suction line approximately four inches away from outdoor unit. Insulate the bulb and suction line with a strip of foam rubber.
3. Measure:
   Suction Pressure (SP).
   Suction Line Temperature (ST).
   Outdoor Temperature (ODT).
   Indoor Temperature (IDT).
4. Determine °F superheat from a "temperature/pressure" table and a low-side manifold gauge. Refer to R-22 temperature/pressure table. Determine suction gas temperature at suction gauge pressure reading and subtract from temperature reading of suction line. This is your superheat.

EXAMPLE

Suction pressure = 70 lb (31.7 kg).
Actual temperature reading = 59°F (15°C).
(Minus) Suction temp. (from table) = 41°F (5°C).
Superheat = 18°F (−7.7°C).

5. Place an "X" on charging chart at intersection of OD temperature and ID temperature.
6. Draw horizontal line from X to left side of chart.
7. From determined F superheat:

SUPERHEAT CHARGING CHART



a. If superheat (from step 4) is within 5°F of chart reading, charge is OK.

b. If superheat (from step 4) is more than 5°F above chart reading, add R-22 until within 5°F.

c. If superheat (from step 4) is more than 5°F below chart reading, remove and recover R-22 until within 5°F.

8. If superheat (from step 4) is below the 5°F limit DO NOT ADD R-22.

QUESTIONS AND COMMENTS

Charles W. Frazell, Senior Engineer, TU Electric, Dallas, Texas: What was the effect of a smaller compressor on latent capacity?

William P. Levins: Table 4 in the paper shows that at 95°F the measured total capacity of the original unit was 31,920 Btu/h with a sensible-to-total ratio of 0.72. The retrofitted unit had a measured total capacity of 26,400 Btu/h with a sensible-to-total ratio of 0.75. Therefore, at 95°F the latent capacity of the system with the original compressor was about 35% higher on an absolute basis (8,938 Btu/h vs. 6,600 Btu/h) than the retrofitted system with a 30% smaller compressor. Both sensible-to-total ratios are in the acceptable comfort range for 95°F operation. Table A-4 in the paper contains capacity, power, and airflow data for all runs performed for the paper.