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Performance and Reliability Degradation of a Residential Unitary Air Conditioning System Resulting from a Mismatch of a 1992 Air Conditioning Unit with a 15-20 Year Old Evaporator

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PERFORMANCE AND RELIABILITY DEGRADATION OF A RESIDENTIAL UNITARY AIR CONDITIONING SYSTEM RESULTING FROM A MISMATCH OF A 1992 AIR CONDITIONING UNIT WITH A 15-20 YEAR OLD EVAPORATOR

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ABSTRACT

The National Appliance Energy Conservation Act has established a minimum efficiency of 10 SEER for residential split system air conditioners and heat pumps. Approximately 50% of residential split system air conditioners are sold as replacements for units having much lower SEER levels, typically 5-9 Btu/W·h.

This investigation assessed the detrimental affects that could occur when a 1992 vintage 10 SEER air conditioner is used to replace a 15-20 year old outdoor unit without replacing the indoor evaporator section. An evaporator constructed with similar geometry as a 15 year old, 30,000 Btu/h (8,790 W) evaporator was tested with a typical 1992, 10 SEER residential air conditioner. The performance of the 1992 outdoor unit with the old evaporator experienced 5% degradation in capacity and 11% degradation in SEER. The system also operated at compression ratios typically 20% higher than the system with the properly sized evaporator; thus increasing the stresses on the compressor and hence reducing its life.

INTRODUCTION

The National Appliance Energy Conservation Act has mandated that all residential split system (central) air conditioners and heat pumps built on or after January 1, 1992 have seasonal efficiencies of at least 10 SEER. This is a substantial increase from older systems which typically operated between 5 and 9 SEER. By redesigning the three primary components of the system, the compressor, condenser, and evaporator, manufacturers are able to attain minimum efficiency standards.

Although manufacturers focus on capacity and SEER while designing new air conditioners and heat pumps, they are also very aware of design parameters which influence the reliability of their systems. Product and more specifically, compressor reliability is highly dependent upon the proper design and installation of the air conditioning system. Historically, nearly one-half of first year residential split system component failures are compressors. Primary causes of these failures include: (1) liquid entrainment in the suction gas during start-up (slugging), (2) liquid entrainment in the suction gas during steady-state operation (flooding), (3) excessive cycling, (4) oil starvation and dilution, (5) operation at excessive compression ratios, and (6) overheating.

Residential split systems are typically installed in three types of applications: new construction, add-on, and replacement. Of these, the replacement market represents approximately 40-50% of the annual sales volume of residential split systems. Because the replacement market is such a large and growing market segment, dealers (who are the final distributors and installers of split systems) must be careful to avoid misapplication when replacing existing systems, or excessive compressor failure rates can occur.

Dealers frequently replace older air conditioners or heat pumps when the compressor fails. It is common for the compressor to fail during peak summer cooling months when the system is heavily loaded. Unfortunately, this also coincides with peak consumer demand for repair and replacement of split system air conditioners. Surveys have shown that nearly 50% of the time, the dealer will not replace the indoor section when he installs a replacement outdoor unit. While this practice may have been acceptable with older less efficient systems, continuing this practice with the current vintage high efficiency systems will result in serious degradation of both the performance and reliability of the system.

This deterioration in performance and reliability is the result of design differences between the older and current indoor coils. Differences between the two include: coil heat transfer capacity and expansion device types and sizes. Table 1 provides a comparison of a typical 1970's and 1992 evaporator coil. One can see that the typical 1992 coil has greater face area and fins per inch. It also utilizes internally enhanced tubes and externally lanced fins. These differences enable the 1992 coil to transfer the same amount of heat as the 1970's coil at a much lower air-to-refrigerant temperature difference. The table also lists representative saturated condensing temperatures for 1970's and 1992 condensing units. This temperature determines the sizing of the expansion device; systems with higher saturated condensing temperatures will have smaller diameter capillary tubes or orifices.

TABLE 1
Comparison of 1970's and 1992 Indoor Coil

	1970's	1992
Face area, ft^2/ton (m^2/kW)	0.75-1.0 (0.0198-0.0264)	1.25-1.5 (0.033-0.0396)
Fin density, fin/in. (fin/mm)	10-12 (0.394-0.472)	14-16 (0.551-0.630)
Inner tube surface	Smooth	Grooved
Fin type	Corrugated or flat	Lanced or louvered
Expansion device	Capillary tube	Cap. tube & orifice
Saturated cond. temp °F (°C)	130-135 (54.4-57.2)	120-125 (48.9-51.7)

TEST FACILITY

The test facility used for this study was a state of the art automated psychrometric testing laboratory. The facility consisted of four components: (1) indoor room, (2) outdoor room, (3) programmable controller, and (4) data acquisition system. The facility and system were fully instrumented to perform Department of Energy (DOE) capacity and SEER measurements and ARI abnormal tests. The rooms were controlled independently to specified temperature and humidity conditions. The condensing unit was installed in the outdoor room and connected to the indoor coil with 25 ft (7.63 m) of refrigerant line set.

In addition to controlling the evaporator and condenser inlet dry-bulb and wet-bulb air temperatures, the airflow across the evaporator was also controlled. All other parameters were dependent variables. The evaporator inlet and outlet wet-bulb and dry-bulb temperatures and airflow were measured. Capacity was calculated using the Air Enthalpy Method. The energy balance was verified using both the Refrigerant Enthalpy Method and the Condensate Measurement Method (described in ASHRAE Standard 37-88). The system power was measured and used to calculate the energy efficiency ratio (EER). Capacity, EER, and SEER were calculated in accordance with DOE and ARI standards (ARI Standard 210/240-89). Other relevant measurements included the compressor suction and discharge pressures and compressor entering and leaving temperatures. These values were used as reliability indicators. The data acquisition system collected each data point (temperature, pressure, power, etc.) every five seconds.

TEST PROGRAM

The experimental program consisted of both DOE standard and ARI abnormal tests. The standard tests were the DOE A, B, C, and D tests. (All four were required for the calculation of SEER.) The capacity at DOE A test conditions and SEER were the two primary performance criteria in this study. Test D was a cyclic test in which the unit was off for 24 minutes and on for 6 minutes while the other three were steady-state tests run for 30 minutes each at various inlet conditions (Table 2). In addition, the airflow rate across the indoor coil was approximately 1,000 cfm (472 L/s) for all standard DOE tests.

TABLE 2
Standard DOE Test Temperature Conditions

Test	Indoor DB °F (°C)	Indoor WB °F (°C)	Outdoor DB °F (°C)
A	80 (26.7)	67 (19.4)	95 (35)
B	80 (26.7)	67 (19.4)	82 (27.8)
C	80 (26.7)	less than 57 (13.9)	82 (27.8)
D	80 (26.7)	less than 57 (13.9)	82 (27.8)

In addition to the four standard tests, three abnormal or off-design tests were performed: (1) ARI maximum cooling load (MAX), (2) minimum outdoor operation temperature test (MIN), and (3) continuous floodback test (FLOOD). Table 3 outlines the conditions of these three tests. These tests were designed to simulate severe conditions which a unit can occasionally experience during operation. Although performance (capacity and EER) was calculated for these tests, the parameters of greatest importance were the reliability indicators.

TABLE 3
ARI Abnormal Test Conditions

Test	Outdoor DB °F (°C)	Indoor DB °F (°C)	Indoor WB °F (°C)	Voltage (V)	Airflow cfm/ton (L/kW·s)
MAX	115 (46.1)	80 (26.7)	67 (19.4)	197	450 (60.3)
MIN	55 (12.8)	75 (23.9)	57 (13.9)	230	400 (53.6)
FLOOD	67 (19.4)	67 (19.4)	57 (13.9)	230	200 (26.8)

The reliability criteria considered were compressor suction superheat, compressor discharge superheat, discharge pressure, and compression ratio. The suction and discharge superheats were selected because they are strong indicators of compressor liquid flooding. When the suction superheat is nearly as low as the saturation temperature, it is no longer discernable and the compressor is flooding. At this point, the discharge superheat is used as an indicator of flooding severity; the lower the discharge superheat, the lower the vapor quality entering the compressor. Compressor reliability is degraded in proportion to the quantity of liquid refrigerant entering the compressor. When a compressor is exposed to extended periods of liquid flooding imminent failure is likely.

The final criteria, discharge pressure and compression ratio, were indicators of how hard the compressor was working. Compressors are designed to operate within specific compression ratio envelopes; operation outside this envelope can cause excessive stresses and result in premature failure.

A typical 1992, 10 SEER, nominal 30,000 Btu/W·h (8,790 W) condensing unit with a properly matched indoor coil (similar to that outlined in Table 1) served as the baseline for this study. The condensing unit consisted of: (1) a single row coil, (2) a vertical air discharge system with an axial fan, (3) and a reciprocating compressor, while the baseline expansion device was a fixed diameter orifice (0.067 in., 1.702 mm). The system was charged to 10°F (5.56°C) suction superheat temperature at DOE A conditions. Once established, the charge was held constant for all remaining baseline tests.

Since actual older coils were not available for testing, a coil having similar physical dimensions as a 15-20 year old evaporator was used (Table 4). This coil was tested with the baseline condensing unit. To replicate recommended field charging practices, the refrigerant charge level was adjusted in the same fashion as the baseline unit (suction superheat was set at 10°F (5.56°C) at DOE A test conditions). The 1970's vintage evaporator was tested with an 0.049 inch (1.24 mm) diameter orifice. This orifice size yielded a saturated discharge temperature of 137°F (58.3°C) at DOE A test conditions.

Different orifice sizes were selected for the two coils because of different design condensing temperatures between the 1992 and 1970's condensing units. Standard efficiency air conditioners designed during the 1970's typically operated with saturated discharge temperatures between 130°F (54.4°C) and 135°F (57.2°C). Indoor coils designed to operate with these outdoor units therefore had expansion devices (fixed orifice or capillary tubes) sized for these higher condensing temperatures. Testing of the 1970's vintage coil with several different orifice diameters indicated that an 0.049 inch (1.24 mm) orifice yielded saturated condensing temperatures typical of 1970's vintage units. Thus, it was used for all subsequent 1970's coil testing.

TABLE 4
Description of Tested 2½ Ton Indoor Coils

	1970's	1992
Face area, ft^2 (m^2)	3.56 (1.085)	3.11 (0.948)
Fin density, fin/in. (fin/mm)	12 (0.472)	16 (0.630)
Rows	3	3
Inner tube surface	Smooth	Grooved
Fin type	Sine-wave	Sine-wave with lances
Orifice diameter, in. (mm)	0.049 (1.24)	0.067 (1.70)
Saturated cond temp, °F (°C)	137 (58.3)	123 (50.6)

Upon completion of the experimental analysis, a computer simulation was used to verify the trends and accuracy of the test data. System geometry, test conditions, and compressor performance maps were used as simulation inputs. Output parameters included performance (i.e., capacity and EER) and reliability indicators (i.e., suction and discharge pressures and superheat temperatures). The computer simulation was also used to predict the performance of other products not tested during the investigation.

RESULTS

Table 5 is a summary of the performance and reliability measures for the tested coils at DOE A test conditions. Additional refrigerant charge was required to achieve the same 10°F (5.6°C) suction superheat with the smaller orifice 1970's coil. The smaller orifice also increased the restriction in the system which caused the refrigerant flow rate to drop 11.4% for the 1970's coil. The greater restriction in the 1970's coil also resulted in a higher condenser temperature and pressure. The 20.0% increase in discharge pressure caused the compressor to work hard and consequently reduce its life. The increased discharge pressure also increased the total system power with the 1970's coil by 3.0%.

As a result of its greater refrigerant mass flow and enhanced heat transfer surfaces, the 1992 coil had a capacity 5.0% greater than the 1970's coil. The combined effect of lower capacity and higher system power resulted in an 11.3% drop in SEER with the older coil.

Figure 1 demonstrates the reduced capacity for the 1970's coil at standard DOE A, B, and C test conditions. The drop in capacity was primarily due to differences in air and refrigerant side heat transfer surface area and coefficients, and consequently refrigerant mass flow rate and coil geometry. The amount of capacity degradation was dependent upon indoor and outdoor temperatures.

TABLE 5
Experimental Results for DOE A Test

	1970's	1992
Refrigerant flow, lb/h (g/s)	404 (50.9)	456 (57.5)
Refrigerant charge, lb (kg)	6.7 (3.04)	5.0 (2.27)
Saturated discharge temp, °F (°C)	137 (58.3)	123 (50.6)
Cooling capacity, Btu/h (W)	26,400 (7,740)	27,800 (8,150)
System power, W	3,187	3,094
SEER*, Btu/W·h (COP)	8.67 (2.54)	9.78 (2.87)
Discharge superheat, °F (°C)	57.3 (31.8)	53.3 (29.6)
Discharge pressure, psig (kPa)	323.3 (2,227)	269.4 (1,856)

*Calculated from DOE A, B, C, and D tests

FIGURE 1
Capacity Reduction of 1970's Coil

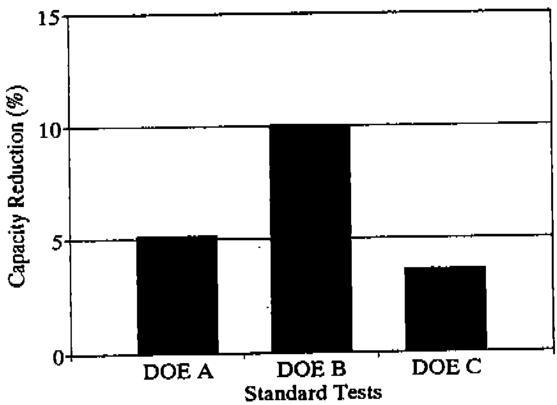
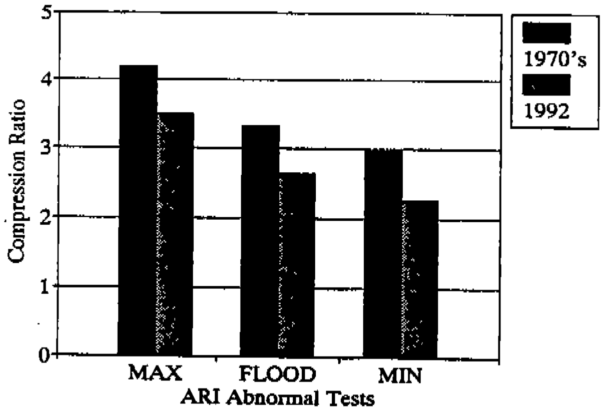


Figure 2 demonstrates substantially higher compression ratios for the 1970's coil at each ARI abnormal test condition; this was an indication of reliability degradation. The trend of lower compression ratio at lower loads was also observed. From the maximum to the minimum load tests, the ambient temperature drop caused the pressure ratios to decrease; however, the difference in pressure ratio between the 1970's and 1992 evaporator coils remained nearly constant.

FIGURE 2
Comparison of 1970's and 1992 Compression Ratio



Another reliability indicator, discharge superheat, is shown in Figure 3 for the ARI abnormal tests. Because the compressor was flooding (as determined by near zero suction superheat) in all but the minimum load test, discharge superheat was used as an indicator of flooding severity. The 1992 coil demonstrated the expected maximum flooding during the floodback test as indicated by 23°F (12.8°C) discharge superheat. In addition, the 1992 coil showed higher discharge superheat for the maximum load test but lower discharge superheat for the floodback and minimum load tests. Thus, the restricted expansion device on the 1970's coil actually increased the discharge superheat on the floodback and minimum operation tests; indicating less flooding. Unfortunately, the restricted expansion device also resulted in higher compression ratios and operating temperatures, both of which can reduce the life of the compressor.

Although it was difficult to determine the net effect of compression ratio and liquid entrainment on reliability, certain operating conditions indicate substantial reliability degradation. Although no flooding was occurring during the minimum load test, the compression ratio was 31% higher for the 1970's coil. Similarly, the compression ratio during the DOE A test was 22% higher for the 1970's coil. In addition, during the ARI maximum load test - for which the compressor was flooding for both coils - the discharge pressure increased from 342 psig (2,356 kPa) to 399 psig (2,749 kPa) when the 1970's coil was used in place of the 1992 coil.

The experimental results were used to verify the trends and accuracy of a computer simulation. Table 6 outlines the results of this simulation for DOE A conditions. Corresponding experimental data is located in Table 5. The last column of Table 6 lists the maximum percent difference between the experimental and simulated data for the two coils.

FIGURE 3
Comparison of 1970's and 1992 Discharge Superheat

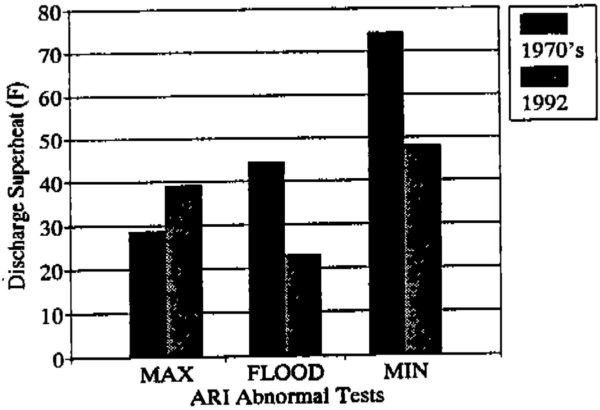


TABLE 6
Comparison of Experimental and Computer Simulated Data for DOE A Test

	Simulated 1970's	Simulated 1992	Max % Diff.
Refrigerant flow, lb/h (g/s)	411 (51.9)	455 (57.3)	1.7
Refrigerant charge, lb (kg)	5.66 (2.57)	4.59 (2.08)	15.5
Saturated disch temp, °F (°C)	139 (59.4)	124 (51.1)	1.4
Cooling capacity, Btu/h (W)	27,000 (7,911)	28,600 (8,380)	2.9
System power, W	3,267	3,081	2.5
SEER*, Btu/W·h (COP)	8.75 (2.56)	10.1 (2.96)	3.3
Discharge superheat, °F (°C)	74.9 (41.6)	66.3 (36.8)	30.7
Discharge pressure, psig (kPa)	334 (2,301)	274 (1,888)	3.3

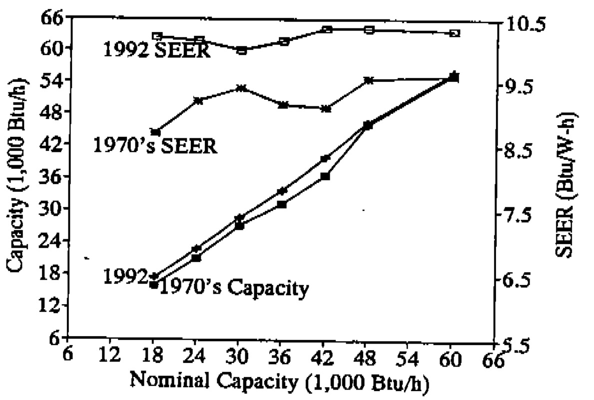
*Calculated from DOE A, B, C, and D tests

All trends were duplicated with extremely close agreement (with few exceptions) between experimental and simulated data. The three exceptions were discharge superheat, suction superheat, and refrigerant charge. Extending the suction and discharge lines to locate the compressor on a remote scale caused the difference in refrigerant charge. The difference in discharge superheat temperatures was probably caused by the method of discharge temperature measurement; it was made by a thermocouple soldered to the outer wall of the discharge tube. Although the immediate vicinity of the thermocouple was insulated, heat conducted axially through the tube wall and convected radially to the ambient, would have caused this temperature and consequently, the discharge superheat temperature to read lower than it actually was.

Since computer simulated capacity and SEER results replicated experimental data extremely well, the computer analysis was extended to include all sizes, 18,000-60,000 Btu/h (5,274-17,580 W), of a 10 SEER air conditioner family. In each case the simulation results examined the performance with both 1970's and 1992 indoor coils.

Similar to the experimental data, capacity and SEER were lower for the 1970's coil (Figure 4). The average drop in capacity from the 1992 to the 1970's coil over the full range of sizes was 5.5% while the average drop in SEER was 9.5%. The drop in SEER roughly translates into 10% higher energy usage. These percentages closely match those calculated for the experimental tested combinations.

FIGURE 4
Computer Simulation Performance Degradation



CONCLUSIONS

This study demonstrates that a mismatch of a 1970's indoor coil with a typical 1992 condensing unit results in 5% reduction in capacity (for DOE A test) and 11% reduction in SEER. Although the former may not drastically affect the homeowner, the latter will result in higher than anticipated energy consumption. In addition, utility companies which are offering rebates for high efficiency systems may find their efforts stymied if the dealer does not replace the indoor section.

The study also determined that a system mismatch causes discharge pressures and compression ratios well in excess of those of a properly matched system. Compression ratio increases of 20% were typically observed. One test, the ARI maximum load test, demonstrated a discharge pressure of 400 psig (2,756 kPa) for the 1970's vintage coil. These results indicate that air conditioners replaced without upgrading the indoor coil will exhibit substantially higher compressor failure rates than those installed with new indoor coils.

The experimental data corroborated computer simulated results extremely well with few exceptions. Thus, the simulation was used to generalize performance and reliability degradation across an entire range of sizes. It demonstrated an average 5% reduction in capacity at DOE A test conditions and an 11% reduction in SEER with the 1970's coil. It also indicated that the new system installed without replacing the old indoor coil will operate at significantly higher discharge pressures than the matched system.

The data presented in this study were for a 1992, 10 SEER air conditioner, which should be considered the best case scenario. If a 1992 vintage 11 or 12 SEER air conditioner were installed with a 1970's vintage indoor coil, the performance and reliability degradation will be significantly worse. It is therefore recommended that dealers and utilities encourage the replacement of the indoor coil in all replacement applications - this will substantially improve efficiency and reliability.

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