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FIELD MEASUREMENTS OF NEW RESIDENTIAL AIR CONDITIONERS IN PHOENIX, ARIZONA

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ABSTRACT

Numerous field studies show that residential air conditioners are not properly installed and as a result do not perform at their design capability. This investigation studied air-conditioning systems installed in newly constructed homes in Phoenix, Arizona. It involved measuring the airflow and charge of the air-conditioning units as well as the leakage of the ducts and building shell in a sample of 28 systems. The investigation found that newly constructed homes in Phoenix have substantial deficiencies in their air-conditioning systems, similar to those found in studies from other parts of the country. Improvements can be made to provide lower energy use and reduced peak electrical demand while improving occupant comfort and satisfaction.

Significant problems were found. Airflow across the inside coil averaged 14% below specification and only 18% of the units were correctly charged. New homes in this sample were very airtight, with up to 82% not meeting ASHRAE ventilation standards with the windows closed. The measured supply duct leakage averaged 9% of the air handler flow. Return leakage was less, on the average, at 5% of flow. Four systems with platform returns had very high return leakage.

These results are supported by extensive information gathered in this project as well as data from projects in other climates.

BACKGROUND

The Phoenix metropolitan area is one of the fastest growing markets for new residential air-conditioning units in the nation. This study was conducted for a local utility to assess the energy savings and peak demand reductions achievable from a heating, ventilating, and air-conditioning (HVAC) efficiency program. Assessment involved the following:

- detailed field testing of a sample of 22 newly built homes (28 HVAC systems) in the Phoenix area to identify problems with current HVAC system installations,
- a three-level nested monitoring of the 22 homes,
- a determination of achievable improvements to current practice and the costs of those improvements,
- analysis using a calibrated simulation based on field and monitored data to estimate the impacts of potential improvements in energy use and peak demand, and
- use of an electronically controlled duct leakage mechanism to assess the impacts of both supply and return system leakage.

This paper describes the results from the field tests.

PRIOR RESEARCH

The author's prior experience and the findings of other research projects have found that typical air-conditioning system installations have numerous problems that adversely impact efficiency, peak electric demand, and comfort. The primary problems include

- excessive duct leakage in unconditioned spaces leading to substantial loss of conditioned air, heated return system air, and increased house infiltration (Cummings et al. 1990; Modera 1989; Palmiter and Bond 1992);
- insufficient airflow through the indoor coil (often caused by restrictive duct design) (Proctor 1991; Proctor and Pernick 1992);
- incorrect refrigerant charge (Neal and Conlin 1988; Hammerlund et al. 1990); and
- excessive air-conditioning system sizing (Cummings et al. 1994; Cavalli and Wyatt 1993; Blasnik et al. 1995a).

In prior studies, these problems were found to be common. Duct leakage has become a significant concern in the recent past (Penn 1993). Studies from California, Florida, Nevada, and the Pacific Northwest have consistently found large efficiency losses due to typical levels of duct leakage and duct conduction

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losses (Jump and Modera 1994; Tooley and Moyer 1989; Blasnik et al. 1995a; Palmiter and Bond 1992; Parker 1989).

SAMPLE

This study utilized a three-level nested sample of 22 homes containing 28 air-conditioning systems. The breakdown of the sample is as follows:

Level 1—Field measurements and air conditioner submetering.
Level 2—Subset of level 1, temperature monitoring—18 systems, 15 homes.
Level 3—Subset of level 2, intensive monitoring—six systems, five homes.

Nine of the houses were unoccupied when tested but were ready for occupancy (that is, fully drywalled with operating central air-conditioning systems). The remainder were occupied and less than one year old. The local utility arranged scheduling and provided contacts with local builders and/or homeowners. The 22 houses came from 19 developments built by 11 general contractors. They are believed to be representative of typical new construction in the area.

FIELD DATA COLLECTION PROTOCOL

Investigators designed the field investigation to examine a wide variety of potential HVAC problem areas and to collect information needed to assess summer design cooling loads and overall building shell thermal integrity. The field procedures included many recently developed state-of-the-art diagnostic tests (particularly for assessing the duct systems). The field testing protocol is summarized in Table 1.

IMPLEMENTATION

Technicians previously trained and experienced with these procedures were used to perform the field work. All technicians were trained in data collection by the field manager to ensure quality. The two-person teams required an average of half a day per house. Scheduling began at the end of June 1995, and all field work was completed by mid-July.

TABLE 1
Summary of Field Test Data Collection Procedures

Parameter	Tests	Description/Use
Duct Leakage	Total leakage	Pressurize ducts to 0.10 in. water (25 pa) with fan/measurement device mounted at the air handler, registers sealed; measure fan flow, check pressures in other parts of duct system
	Exterior leakage	Repeat above test while blower door pressurizes house to the duct test pressure, eliminating pressure difference between ducts and house
	Supply only exterior leakage	Repeat above test after the return system has been separated from the supply system by installing a blockage at the air-handler blower compartment
	Pressure pan - leakage location indicator	Measure pressures at individual registers with blower door pressurizing house to 0.20 in. water (50 pa)
Air Handler Flow	Operating Static Pressures	Measure static pressures in supply and return plenums—used for reference point when measuring airflow with fan/measurement device, also used to determine system flow resistance
	Fan/measurement flow test procedure	Duplicate the supply side pressures after blocking the return and installing the fan/measurement device at the air handler
AC Charge	Weighing of refrigerant	Use recovery equipment to recover and weigh the refrigerant charge in the system and compare to the factory nameplate rating and actual refrigerant line set lengths
AC Input	Wattage input	Use house electric meter to measure actual electric input for a one-time test of input for both the outdoor condensing unit and air handler
AC other	Miscellaneous	Collect nameplate information from indoor and outdoor units, assess potential outdoor unit radiant gain in afternoon
Duct Conduction	Duct system diameters and lengths	Measure individual duct run lengths, record diameter, and draw a diagram of the duct system layout
	Duct system location	Record percentage of supply and return ducts in various locations (attic, garage, inside, etc.), used to estimate ambient conditions around ducts for modeling conduction and leakage
Design Cooling Load	Building dimensions, materials, R-values, shading/exposures	Calculate design cooling loads and proper AC size using enhanced load calculations (Rutkowski 1986)
Building Airtightness	Blower door test	Measure house leakage at a standard test pressure, also measure pressures developed in key building zones such as attics

The field manager reviewed all data daily. The data were entered into spreadsheets along with supplementary information from published air conditioner manufacturer ratings. The raw data were further analyzed for quality, and calculations were performed to derive the system parameters of interest.

FINDINGS—GENERAL CHARACTERISTICS

The typical house in the study was a slab-on-grade home with three bedrooms, about 2,100 ft^2 (195 m^2) of living space, a volume of about 19,500 ft^3 (552 m^3), gas heat (10 of the single-system houses were equipped with heat pumps), double-glazed windows, and 30 $h \cdot ft^2 \cdot °F/Btu$ (5.3 $m^2 \cdot K/W$) attic insulation with a tile roof. Thirteen of the houses had tinted glass to help lower the cooling load, and six of the houses were equipped with external shade screens. There were 18 one-story and 4 two-story houses. Six of the houses had two air-conditioning systems, but only two of the two-story houses had two systems; the remaining two-system houses had one story.

All of the single-AC houses had the air handler located in the attic (one had a roof-mounted package unit). The attic location exacerbates the impacts of return system leakage and increases conductive heat gains.

The houses were tight, with an average blower door measured air leakage of 1,959 cfm (925 L/s) at 0.20 in. H_2O (50 Pa) pressure. This level of airtightness lowers the cooling and heating load of the house and saves energy.

Blower door measurements and an infiltration model[1] (Sherman 1987) were used to estimate the natural infiltration rate for these homes. More than three-fourths of the houses have modeled infiltration less than the minimum ventilation criteria of ASHRAE Standard 62-1989 (ASHRAE 1989). Standard 62-1989 specifies that residential structures must have 0.35 air changes per hour (ACH) or 15 cfm (7.5 L/s) per person, whichever is greater. The modeled natural ACH of the homes in the project averaged 0.29 (with the windows closed).[2] These data are presented in Table 2.

TABLE 2
Modeled Infiltration Failing to Meet ASHRAE Standard

	.35 ACH	15 cfm (7.5 L/s) per person
Failing to Meet Standard	18	6
% Failing to Meet Standard	82%	27%

1. Calculated using wind speeds published in the *1993 ASHRAE Fundamentals* (ASHRAE 1993) and bin weather data published in Rutkowski (1986). Based on an indoor temperature of 70°F (21°C) in winter and 75°F (24°C) in summer.
2. The ASHRAE standard assumes that adequate ventilation can be accomplished by opening windows. Since the lowest ventilation rates will occur when the indoor to outdoor temperature difference is small, opening windows for ventilation may be a viable option (ASHRAE 1989).

FINDINGS—DUCT CHARACTERISTICS

The supply systems commonly consisted of a rigid metal supply plenum with 10 in. (25 cm) and 12 in. (30 cm) diameter helix core flex duct take-offs reduced at rigid sheet metal wyes to 6 in. (15 cm) and 8 in. (20 cm) diameter runs to the registers. The average supply system had about 110 ft (24 m) of supply duct with an average surface area of 250 ft^2 (23 m^2). Most of the supplies were located in the attic (23 of the 28 systems inspected had 100% of the supply duct system located in the attic).

Most of the return systems consisted of helix core flex duct connected directly to the air handler without a return system plenum. The average return system was located in the attic and consisted of a 13-ft (4-m) run of 18-in. (46-cm) flex duct with an average surface area of 58 ft^2 (5 m^2). Five systems used platform returns either with a grille mounted directly on the platform or a ducted return run connected to the platform.

All 28 of the systems examined had the typical 4 $h \cdot ft^2 \cdot °F/Btu$ (0.7 $m^2 \cdot K/W$) insulation value that is common with flex duct systems. Most supply plenums were wrapped with 1.0 in. (2.5 cm) foil-scrim-kraft-faced fiberglass duct wrap. The return system platform plenums had no insulation even though three of them were located in garages.

FINDINGS—DUCT LEAKAGE

Detailed duct leakage measurements were used to quantify the magnitude and impact of the existing leakage problems and the opportunities for improvement. Duct leakage can be measured in several different ways (Proctor et al. 1994). All duct leakage measurements were performed with the fan/measurement device mounted at the air handler's blower compartment opening.[3] Three measures of duct leakage are summarized in this paper: total leakage, leakage to outside, and normal operating leakage split between supply and return.

During the testing, the technicians noted that most of the duct systems had obvious and easily eliminated leakage at the plenums, boot connections, and air handler. For example, it was common to find large leaks at the joint between the supply plenum and the take-offs or starter collars. They also noted that most connections in the duct system may be subject to future failure because they were made with duct tape. One of the HVAC contractors used mastic on some of the joints on the systems he installed. The application of mastic was spotty and only installed where it was easily visible (for example, a common application was at the return grille, while connections in the attic were taped). The systems tested were as tight as they will ever be. They can be expected to leak more over time due to tape failure and disturbances (for example, disconnections and tears) caused by service personnel working in the attics.

The total duct leakage test establishes the total leakage when all the registers are sealed and the ducts are pressurized to

3. This is accomplished by mounting the fan/measurement device to a piece of cardboard cut to fit the opening of the blower compartment door and temporarily attaching the cardboard in place of the blower compartment door.

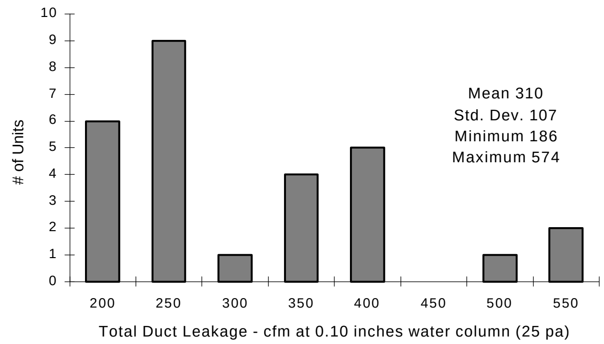


Figure 1 *Histogram of total duct leakage.*

a test pressure of 0.10 in. H_2O (25 Pa). This test measures both leakage to the inside and the outside of the house. Total duct leakage is a fast and accurate test method that is easily applied to new construction even before the drywall is installed. The average total leakage rate was 310 cfm (146 L/s). The distribution of total duct leakage is shown in Figure 1.

The three leakiest duct systems all had major return system leakage. Two of these systems had platform return plenums located in the garage, while the third had a partially disconnected return duct at the air handler in the attic.

Duct leakage communicating with the outside was measured using a blower door and a fan/measurement device pressurizing both the building and the ducts simultaneously. Having the house and the ducts at the same pressure reduces the duct leakage to the inside to a minimum and thus measures the duct leakage to the exterior. The distribution of exterior duct leakage is shown in Figure 2.

The two systems with the highest duct leakage to the outside are two of the systems with the highest leakage in Figure 1. One of the systems has a platform return plenum in the garage and the other has the partially disconnected return duct at the air handler in the attic.

The average duct leakage to the outside was 193 cfm (91 L/s). This is similar to that seen in recent studies of newly constructed houses. Recent studies found duct leakage to the outside in newly constructed homes of 161 cfm (76 L/s) in

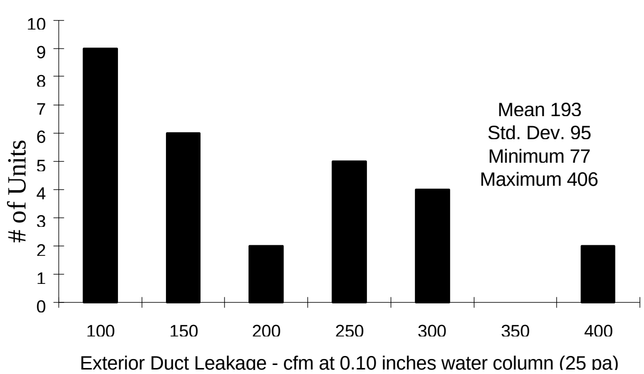


Figure 2 *Histogram of duct leakage to the exterior.*

Nevada (Blasnik et al. 1995a) and 186 cfm (88 L/s) in southern California (Blasnik et al. 1995b).[4]

Both the test for duct leakage to the outside and the test for total duct leakage are useful in estimating the size of the holes in the duct system. The key quantities that affect energy use, however, are the leakage in the supply and return systems under operating conditions (as a percentage of the airflow through the indoor coil). These key duct leakage quantities were determined in the following manner:

- A blockage was installed at the air handler's blower compartment opening to the return system, isolating the supply system. The supply leakage to the exterior was then tested as previously described.[5]
- The return system leakage was calculated as the difference between the total system leakage to the outside and the supply system leakage to the outside.
- The operating leakage for each side was estimated by adjusting the leakage rate to the average pressure in that side of the duct system.[6]
- The operating leakage estimates were divided by the total operating airflow through the indoor coil.

The operating duct leakage split between supply and return is summarized in Figure 3. The flow rates averaged 9% of the air handler flow on the supply side and 5% of the airflow on the return side.

Leaky return systems were concentrated in four of the systems with platform returns and the system with the partially disconnected return duct. Return leakage to the outside on those units was more than three times that of the other return systems. The low return system leakage rates for systems without plat-

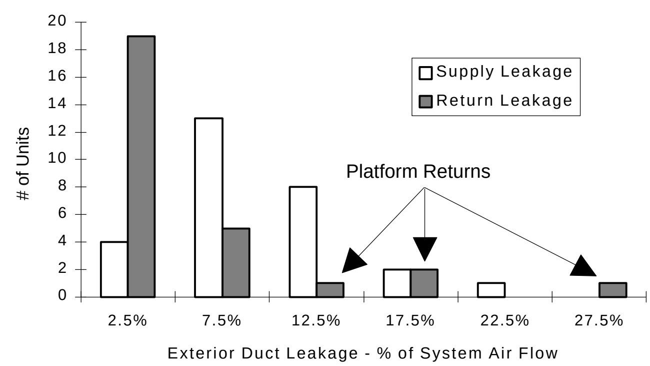


Figure 3 *Histogram of supply and return leakage as a percentage of flow.*

4. The leakage in the Nevada and Southern California reports is listed as cfm at 50 Pa test pressure. The cfm at 25 Pa is estimated by (cfm at 50 Pa)·$(25/50)^{0.65}$.
5. This testing procedure attributes cabinet leakage (other than at the door) to the supply system.
6. The flow exponent was assumed to be 0.65. The leakage at operating conditions, therefore, was calculated as Test Flow·$(\text{operating pressure}/\text{test pressure})^{0.65}$.

form returns are attributed to the short return runs with only two joints sealed with duct tape that had not yet failed.

Duct Leakage Repair

Six of the duct systems tested in this study were randomly selected to receive duct sealing after the initial field tests. This duct sealing was undertaken for two purposes. First, duct sealing helped determine the potential duct integrity for new Phoenix ducts. Second, duct sealing on a few systems widened the range of leakage values, providing additional information for simulation model verification.

On average, the duct sealing required less than four person hours of labor and $50 of materials. It should be noted that sealing took place on a retrofit basis after the system was installed. Repairs were in attics, and most required the removal of duct tape or strapping on the vapor barrier to access the inner liner. The inner liner was then sealed to the take-offs using mastic and mesh. Registers had to be removed to access the boot/drywall connection.

The labor requirements at installation are lower (the only addition beyond current installation methods is the application of mastic rather than tape). Sealing the system at the time of installation also lowers leakage rates because all joints are accessible.

FINDINGS—AIR-CONDITIONING SYSTEMS

The houses had a wide variety of air-conditioning system makes and models. Air conditioners serving an entire house were typically 3½ to 4 tons. Houses with two systems usually had one large system for the main living area (typically 3½ to 4 tons) and a smaller unit for cooling the bedroom areas (typically 2½ tons). The typical air conditioner was a split system with the air handler located in the attic (only one of the systems was a rooftop-mounted package system). The systems examined had properly sized indoor air handlers/coils for the size of the outdoor unit (the exception was a house with a 3-ton outdoor unit mistakenly connected to the 4-ton indoor coil/air handler and vice versa). Only two of the air conditioners had “upsized” indoor coils (both coils were rated one ton larger than the outdoor unit; contractors often do this to get an increased SEER rating). Rated SEER values ranged from 10 to 12, while rated EERs (at 80°F/67°F [27°C/19°C] inside and 95°F [35°C] outside) ranged from 8.7 to 10.9 and averaged 9.9. Twenty-six of the systems had orifice-type refrigerant metering expansion devices (mostly capillary tubes) and the remaining two systems had thermostatic expansion valves (TXVs).

Air Handler Flow Rate

The proper operation of an air-conditioning system depends on providing the correct airflow rate through the indoor coil—usually listed by the manufacturer as 400 cfm (189 L/s) per ton of nominal capacity (Carrier 1978). Low airflow has been a common problem found in other studies of air conditioner performance (Proctor 1991; Neal and Conlin 1988). In addition to potentially shortening equipment life, incorrect airflow renders most standard tests for proper refrigerant charge invalid (Trane n.d.). In a hot, dry climate such as that in Phoenix, where sensible cooling is the major concern, a national air-conditioning contractors’ association and most manufacturers recommend airflows of 525 cfm per ton (248 L/s per ton) and higher (Rutkowski 1995; Carrier 1978).

All systems were tested for airflow with a clean filter in place and operating at the cooling mode blower speed. The fan/measurement device test method was used because of its accuracy and reliability. The procedure involves these steps:

- The supply system static pressures are measured in two duct locations while the system is running at steady state. The static pressures are measured using a static pressure probe and a digital manometer.
- The return system is blocked at the air handler.
- The fan/measurement device is installed in the blower compartment opening. All airflow through the air handler fan must then come through the fan/measurement device.
- The supply system static pressures measured in the first step are duplicated by turning the air handler fan on and adjusting the speed of the fan/measurement device fan.
- The measured flow rate duplicates the operating flow rate of the system.

Figure 4 shows the distribution of measured flow rates compared to manufacturers’ specifications. The average measured flow rate was 344 cfm (162 L/s) per ton, 14% below the target value. More than half of the units were below 350 cfm (165 L/s) per ton (often used as a level requiring corrective action). It should be noted that these units have the highest airflow they will ever experience. As the units get older, the blower and indoor coil will become dirty and the airflow will decrease.

The potential causes of low airflow were investigated. Each air handler’s make and model number were used to access the fan curve information (airflow vs. external static). All of the systems had an air handler capable of delivering the necessary flow against well-designed duct systems. External static pressure consists of the evaporator coil, the filter, the ductwork, and registers. In many cases the measured static pressure of the ducts

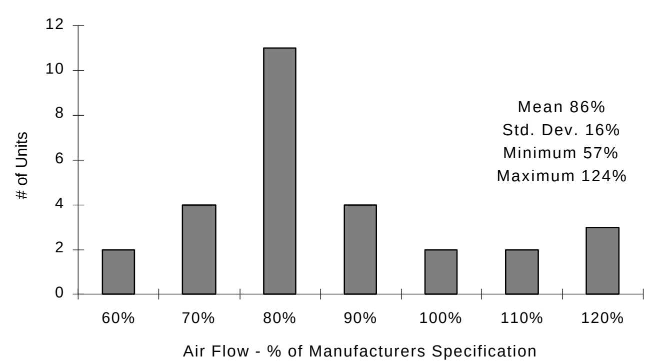
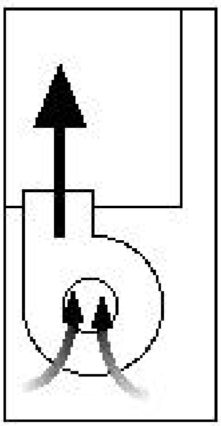


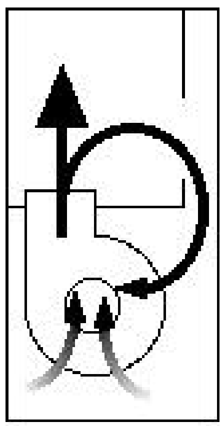
Figure 4 *Histogram of air handler flow.*

alone precluded adequate flow. With the filter and coil in place, airflow was further reduced.

An additional cause of low airflow was found at houses with heat pumps. Heat pump systems had an average airflow that was 9% less than air conditioners with gas heat. Further examination found that it is common practice in Phoenix for the heat pump to be installed without backup electric heat strips. The cabinet opening provided for the heat strip insertion was left open. This opening allowed air recirculation from the supply plenum back to the blower inlet. This problem is illustrated in Figure 5.



Furnace or Heat Pump with all panels in place



Heat Pump with panel missing

Figure 5 *Air handler flow bypass-heat pumps.*

Checking Refrigerant Charge

Manufacturers of residential air-conditioning systems recommend various methodologies for determining proper system charge. The most common noninvasive method for air conditioners with fixed metering devices (cap tube or fixed orifice) is evaporator superheat. For systems with TXVs the subcooling method is suggested. The most accurate (and most time consuming) method is recovery and weighing of the refrigerant.

For accuracy when using superheat or subcooling methods, two items must be within proper range during the test:

- Airflow through the indoor coil must be within ±50 cfm (24 L/s) of the manufacturer's suggested flow (400 cfm [189 L/s] per ton/wet coil)[7] (Trane n.d.).
- For superheat (the method used in the majority of the systems), the indoor and outdoor temperatures must be within a specified range (Trane n.d.).

Considering past experience with new construction testing (Blasnik et al. 1995a), it was anticipated that a large portion of the systems would not have adequate airflow through the indoor coil (64% of the units in this study had airflow less than 350 cfm [165 L/s] per ton). It was also anticipated that the indoor wet-bulb temperature in relationship to the outdoor dry-bulb temperatures would be outside the acceptable range. For these reasons, the refrigerant was recovered and weighed to assess charge.

The field technicians used a step-by-step procedure to lead them through recovery of the refrigerant. Precautions were taken to ensure that no contaminants would be introduced to the system and that all refrigerant would be recovered. The key points of the procedure include the following:

- The use of a vacuum pump and micron gauge to evacuate the service manifold line sets and recovery cylinder prior to recovery of the refrigerant (to keep from introducing noncondensables into the refrigerant).
- The use of a recovery device to evacuate the air conditioner to a minimum vacuum of 15 in. Hg (52 kPa) to ensure that all refrigerant has been recovered from the system.
- The use of a precision scale to weigh the cylinder before and after recovery.

Refrigerant Charge

Incorrect refrigerant charge is a common problem with air-conditioning systems (Proctor 1991). It is an expectation that newly installed systems would be properly charged. Unfortunately, new systems appear to suffer from incorrect charge as often as older systems (Hammerlund et al. 1990; Blasnik et al. 1995a).

Most installation technicians are under demanding time constraints when installing systems. In order to reduce installation time, many technicians rely on shortcuts, rules of thumb, and guesswork rather than adhering to the manufacturer's installation instructions. Most air conditioners come from the factory charged with enough refrigerant to accommodate a 25-ft (8-m) line set. If the installed line set is less than or more than the manufacturer's standard length, the charge must be adjusted to compensate for the difference (if the line set is shorter, charge must be removed or, if longer, charge must be added). Most installation technicians rely on refrigerant system pressures to indicate if the charge is correct.

There are many incorrect and inaccurate rules of thumb for assessing the charge in air conditioners. One of the most common methods used is looking at the refrigerant gauge pressures to see if they are in the "correct" range for the presumed indoor and outdoor conditions. The correct range is often interpreted as low-side pressure near 70 to 80 psi (480 to 550 kPa) or condenser saturation temperature of approximately 25°F (14°C) hotter than ambient.[8] If the pressure/temperature is in the "correct" range, the system is assumed to be charged properly.

This is one of the first studies of new construction that has weighed the refrigerant charge of the air conditioners. Previous studies of new construction used superheat or subcooling in

7. The need for 400 cfm/ton (189 L/s per ton) for charge testing is generally accepted but not universal. Wheeler (1988) states: "Even when the evaporator airflow is low, you can use this method to check for correct charge, before diagnosis."

8. Wheeler (1988) cites 35°F (19°C) as the common but incorrect charging method.

conjunction with measured kW input and system capacity to determine if the air conditioner's charge was correct (Hammerlund et al. 1990; Blasnik et al. 1995a). Most studies of existing construction have found overcharge and undercharge at about the same rate, 25% to 35% of each (Proctor 1991; Proctor and Pernick 1992).

Twenty-seven of the twenty-eight units in the project were tested for refrigerant charge. The one rooftop package unit was not tested because there was no stable, level space for the refrigerant scale. This system is not included in the summary. The results of the charge assessment are summarized in Table 3.

TABLE 3
Air Conditioner Refrigerant Charge

Charge	# of Units	% of Units
Within 5% of Correct Charge	5	18%
Undercharged	21	78%
Overcharged	1	4%

Recovering and weighing of the refrigerant indicated that only 5 of the 27 units tested were correctly charged. Figure 6 displays the distribution of refrigerant charge.

A number of possibilities exist for the predominance of undercharged systems. The most likely is that the system is installed and the charge in the outdoor unit (correct for a 25-ft [8-m] line set) is released into the system. This results in the unit being low on charge by the amount necessitated by the line set length beyond 25 ft (8 m). Sixteen of the systems tested had refrigerant line sets with less than 10 ft (3 m) of deviation from the manufacturer's standard (these systems require less than 8 oz [225 g] of charge adjustment). These units averaged 11% undercharge, while those requiring larger adjustments averaged 33% undercharge.

Another potential cause for the high occurrence of undercharged systems is inadequate evacuation of the systems by the installation technician. Incomplete evacuation would result in the technician reading pressures that are inflated by the air left in the system and drawing conclusions based on the corresponding saturation temperatures.

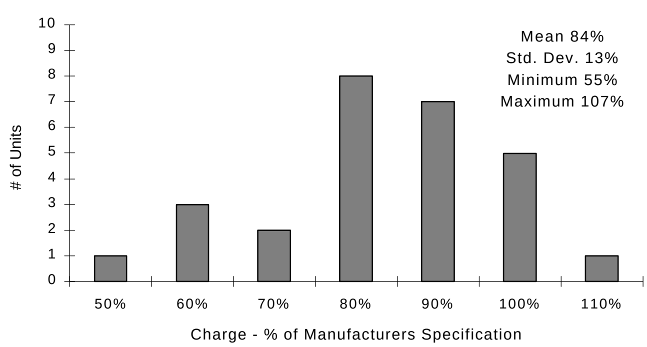


Figure 6 *Histogram of charge as a percentage of manufacturer's specification.*

Installation technicians frequently do not properly evacuate the refrigerant lines and indoor coil prior to releasing the refrigerant from the outdoor unit. The author's staff has yet to observe an installation technician use a micron gauge in evacuation. A common error is the use of a compound gauge to determine vacuum. If the technician had a properly calibrated gauge and if the vacuum pump was able to pull the system down to 29 in. Hg (98 kPa), the vacuum would be 25,400 microns (3,374 Pa) absolute. Most manufacturers recommend a vacuum of 1,000 microns (133 Pa) absolute or less (Carrier 1989). Ensuring this depth of depressurization with a compound gauge is impossible because the technician would have to confirm that the compound gauge read 0.039 in. Hg (0.133 kPa).

The effect of incorrect charge is shown in Figure 7 (Farazad and O'Neal 1988). Incorrect charge reduces both capacity and efficiency. As little as 10% undercharge will reduce capacity as much as 14% (for 82°F [28°C] outside temperature).

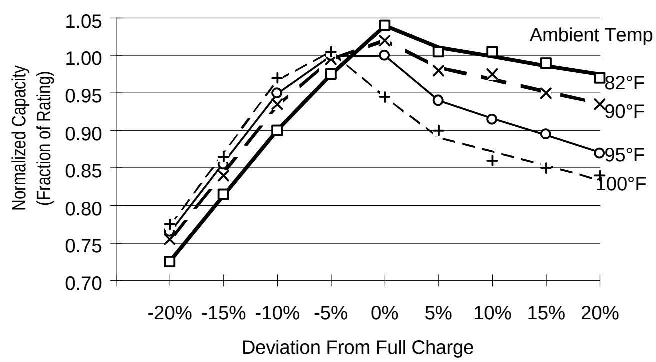


Figure 7 *Capacity vs. charge.*

A dramatic example of the effect of incorrect charge and airflow was encountered during the course of this study. An occupant of one of the houses complained several times to the general contractor that the system was not cooling the house. Eventually the contractor came out and told the homeowner that the attic was not properly insulated (the insulation contractors only added 1 in. (2.5 cm) of cellulose when they were sent back). Approximately two months after the customer moved into the house, the compressor failed on the air conditioner. Testing indicated the system only had slightly more than 60% of the manufacturer's recommended charge, and the airflow was less than three-fourths that recommended by the manufacturer.

Air Conditioner Sizing

A standard reference (Rutkowski 1986) was used to estimate the design load for the study houses. This standard method was enhanced by an infiltration estimate based on blower door testing of each home. These calculations estimated cooling loads at design conditions ranging from 17,500 to 50,400 Btu/h (5,100 to 14,800 W) with an average of 30,251 Btu/h (8,867 W). Slightly less than half of the design load came from heat gains through windows and glass doors. The next highest contributor to the gain was attic and wall conduction, with the remainder of the gains evenly dispersed among infiltration, duct conduction, and internal gains.

The 97.5% design conditions for Phoenix are 107°F (42°C) dry bulb and 71°F (22°C) wet bulb outdoors (about 0.008 humidity ratio) and 75°F (24°C) dry bulb indoors. The capacity of the installed equipment at design conditions was estimated from manufacturers' data corrected to these conditions. The distribution of installed capacity vs. design load is shown in Figure 8.

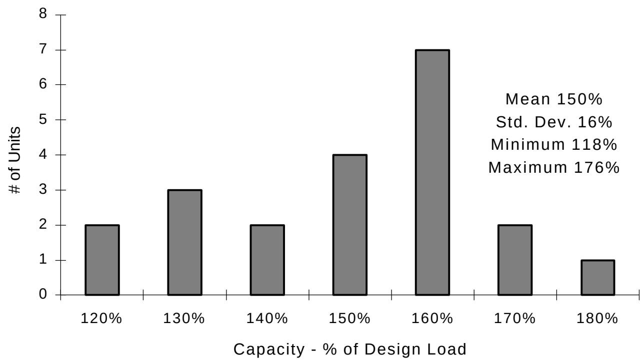


Figure 8 *Histogram of installed capacity as a percent of design load.*

The average design capacity of the equipment installed per house is 44,736 Btu/h (13,112 W). This capacity represents an average of 48% oversizing when compared to the calculated design loads.

Two-system houses were not sized any closer to design than single-system houses. The average two-system house had a calculated heat gain of 42,455 Btu/h (12,444 W) and was equipped with an average total design capacity of 63,729 Btu/h (18,579 W) (50% oversize). The average single-system house had a calculated heat gain of 25,675 Btu/h (7,525 W) and was equipped with an air conditioner with a design capacity of 37,614 Btu/h (11,025 W) (47% oversize). Table 4 summarizes the oversizing issue.

Not only are these units oversized compared to the estimated design load, the estimated design load overestimates the actual cooling load. These issues are detailed in Blasnik et al. (1996).

CONCLUSIONS

New homes in this sample were extremely airtight, with up to 82% that may not meet ASHRAE ventilation standards with the windows closed. The measured supply duct leakage averaged 9% of the air handler flow. Return leakage was 5%. Significant problems were found with low flow across the inside coil and incorrect charge. These findings are consistent with those of similar investigations (Blasnik et al. 1995a; Blasnik et al. 1995b; Hammerlund et al. 1990; Neal and Conlin 1988; Proctor 1991; Proctor and Pernick 1992). Table 5 summarizes the key results from the field investigation.

TABLE 4
Sizing Comparisons

	Single-System Homes Btu/h (W)	Two-System Homes Btu/h (W)	All Homes Btu/h (W)
Rated Capacity@ ARI Std. Conditions	46,600 (13,658)	78,950 (23,433)	55,423 (16,244)
Modeled Capacity @ 107°F out, 75/62°F in	37,614 (11,025)	63,729 (18,579)	44,736 (13,112)
Estimated Design Load	25,675 (7,525)	42,455 (12,444)	30,251 (8,867)
Modeled Capacity Oversize (% of Estimated Design Load)	47%	50%	48%

RECOMMENDATION

This study shows the need for improved control over the installation and design of residential air-conditioning systems. The current system does not ensure that air-conditioning equipment and systems will achieve their designed performance. To achieve a satisfactory level, the infrastructure and market need to be changed. Building the infrastructure (transforming the market) includes

- providing an economic incentive for quality work,
- providing a system that accomplishes the tasks,
- training on that system,
- providing adequate time to accomplish the tasks, and
- holding everyone accountable to follow the system.

One of the qualities necessary in the system is immediate feedback to the technicians on their work. Holding everyone accountable implies certification of participating technicians *based on their field work*.

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TABLE 5
Summary of Field Findings

	Shell	Ducts		Air Conditioner			
	Leakage at 0.20 in. H_2O (50 Pa)	Operating Leakage (% of flow)		AC Sizing (% Over)	Airflow per nom. ton	Charge	
	cfm (L/s)	Supply	Return		cfm (L/s)		
Unit Mean		9%	5%		345 (163)	Correct	18%
House Mean	1959 (925)			148%		Under	78%
Std. Deviation	804 (369)	4%	6%	16%	65 (31)	Over	4%
Median	1634 (771)	8%	4%	147%	329 (155)		
Minimum	956 (451)	3%	0%	118%	229 (108)		
Maximum	3554 (1677)	21%	27%	176%	497 (235)		

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