Air Handlers: An Appliance of Airtight Defiance?

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ABSTRACT

Many studies have been performed around the U.S. that quantify the air tightness of entire air distribution systems; however, there is very little published data on air handler tightness. Even relatively small holes in or near the air handler can be critical since the static pressures during system operation are greater in this area than in any other part of the air distribution system.

A study of air handler tightness was conducted using 69 heat pump, gas heat, and hydronic heat systems. Air handlers in Florida single-family homes built after January 2001 were tested during the period from June 2001 through June 2002. Testing occurred in 23 units in attics, 23 units in garages, and 23 units indoors.

The study finds that leakage in the air handler cabinet alone is sufficient to disqualify some air distribution systems from receiving "leak free" credit in energy codes. On average the air handler and duct connections to it have a $Q_{25,\ total}$ that is 2% of total system rated airflow, allowing only 3% more to be spread over the hundreds of square feet of the entire duct system under 1998 IECC standards. This paper will discuss the tested air leakage rate, $Q_{25,\ total}$, of air handlers and the significance of operational leakage rate, Q, at the positive and negative pressure regions. Characterization of air handler leaks will explain why the air handler and duct connections to it have an operational leakage that is about 5% of the total rated air flow on average, and what can be done about it.

Introduction

A significant amount of research has been done since the late 1980's on residential air distribution tightness throughout the United States. It has been well established that duct systems are often very leaky (Cummings and Tooley1989; Davis 1991; Modera 1989; Parker 1989; Proctor et al.1990). National residential energy rating programs take this into account. Duct leakage is significant enough that one of the top recommendations to meet an elevated energy efficiency goal, such as Energy Star Homes or other utility programs, is to have a sealed and tested duct system. In 1997 this author found that air distribution systems constructed to receive tight duct credit became 2.7 times more leaky, on average, after the air handler and grille registers were installed. In these cases, the ducts went from being substantially tight, to ones disqualified as a sealed system. A significant part of this added leakage was due to the grille and boot connections. But, simply sealing the return duct to the air handler and sealing air handler leakage was enough to receive the tight duct credit.

Background

A change was proposed to the Florida Building Code in 2000 that would have disallowed air handlers to be located in attic locations for new installations. The intent was to minimize efficiency losses associated with conduction and air leakage. The opposition to this proposed code change requested justification. However, due to a lack of published data on air handler leakage, little was known about energy impacts related to air leakage and location. It was decided to continue to allow air handler installations in attics, but with updated heating and cooling code multipliers. (The multipliers are used to calculate a cumulative point score to determine if a proposed house meets Florida energy code.)

A study was conducted to measure air handler cabinet leakage and cabinet operating static pressures. A detailed inspection was also conducted to evaluate the nature of cabinet leakage. Using the field data as inputs, energy simulations were performed for air handler locations indoors, in garages, and in attics for north, central and south Florida climates. This work was then used to modify Florida Energy Code multipliers.

In order to determine the impact of air handler location upon heating and cooling energy use, the amount of air leakage occurring at the air handler cabinet is required. Leakage at the connections between the air handler cabinet and the return and supply plenums is also required, because these connections are part of the appliance installation. Direct measurement of the actual operating air leakage, Q, is not feasible since much of it occurs through small holes and cracks often located in tight spaces. Therefore, it is necessary to know the size of the holes and the pressure differential operating across each hole.

Test Methods

The study began in 2001 with the goal of evaluating the air tightness of 69 newly installed air handlers with 23 units in the attic, 23 units in the garage, and 23 units indoors. Public records were used to randomly select houses built since January 1, 2001. In most cases, houses tested were about 4 months old and the oldest was about 1 year old by the time the study was completed in 2002. Once the random sample was collected, homeowners were asked to participate in the study and the study sample was screened so that no more than four of the same builder or AC contractor would be used.

Four systems were tested in north Florida and the rest were tested in six central Florida counties. It is worth noting the difficulty in finding new air handler attic installations that could be tested in Florida. For example, out of 186 houses initially identified as potential study homes in three counties, only eight were located in the attic. We were able to gain access to only one of those eight for testing. Of the twenty-three attic units tested, seventeen were located in one county, and four in north Florida. So, while attic installations are permitted, they represent the minority of all locations with most new installations in garages and in indoor closets. Package air conditioning units are located only outdoors and were not studied.

Air leakage ($Q_{25,\ total}$) at the air handler and two adjacent connections was measured. $Q_{25,\ total}$ is the amount of air leakage in cubic feet per minute that occurs when the ductwork (or isolated portions of it) is depressurized 25 pascals (Pa) (0.100 In WC) static pressure with respect to its surrounding environment. The $Q_{25,\ total}$ can also be considered a surrogate measurement of hole size. In order to obtain actual air leakage that occurs while the system operates, it is necessary to measure the operating pressure differential between inside and outside the air handler and adjacent connections. In other words, knowing both Q_{25} and operational pressure differentials, actual air leakage into or out of the system can be calculated such as is done in the ASHRAE test standard 152 on distribution system efficiency. There are two different duct test procedures. One is designed to measure all the leakage in the duct system, known as $Q_{25,\ total}$. The second procedure is designed to only measure the leakage that occurs in unconditioned space outside the primary air barrier of the building. This is known as Q_{25}. Throughout this paper, most of the reported air tightness testing is $Q_{25,\ total}$.

Test equipment was calibrated at a range of airflows with special attention to very low flows prior to test measurements since it was anticipated that leakage flows could be quite low. Calibration results show differences between the Duct Blaster ™ test fan and a TSI wind tunnel model 8390 that range from 1.9 cubic feet per minute (cfm) to 2.1 cfm at flow rates between 10 cfm to 40 cfm. The offset was still only about 2 cfm when calibrated at about 200 cfm.

The test method used for determining $Q_{25,\ total}$ in the air handler (AH) and at adjacent connections involved several steps. First, the calibrated blower was attached to the largest return closest to the air handler. Often there was only one return grille located very close to the air handler. Next, a portion of the air distribution system containing the air handler was isolated, when possible, to maximize accuracy of measurement, however it was only reasonable to do this in 36 tests. Typically this involved cutting through the main supply plenum, placing a thin air barrier through the supply plenum, and then sealing this air barrier to the exterior surface of the supply plenum (Figure 1).

Figure 1. Supply Duct Blocked From Air Handler and Return Duct

It was not appropriate to cut through the supply plenum in a number of homes, either because there was insufficient space around the plenum, or because the supply plenum was a flex duct. In those cases, the entire duct system was tested. When the entire duct system had to be tested, test pressures were measured in parts of the supply and return ducts as well as in the air handler.

The leakage of the three locations (return plenum connection, AH cabinet, and supply plenum connection) was determined by a series of tests after each test item was sealed. The leakage of each item was then calculated by subtracting the pre-seal $Q_{25,\ total}$ value from the post-seal $Q_{25,\ total}$ value. Leaks at the air handler-to-supply plenum connection were sealed, and then the $Q_{25,\ total}$ test was repeated. Next, leaks at the air handler-to-return plenum connection were sealed followed by another $Q_{25,\ total}$ test. Then, leaks in the air handler cabinet were sealed followed by one last $Q_{25,\ total}$ measurement. Air handler leakage was sealed using tape on seams and putty around more difficult geometries.

Finally, the static operating pressures of different sections of the air handler were measured, and surface areas of each pressure section of the air handler were measured. Since appliances were tested as found, those that had been sealed in some way by the installer would be tighter than if they had done nothing at all. Therefore an estimate was made of the percentage of potential leakage (by approximate area) that had been sealed by the installer at the cabinet. This information was not used to make any adjustments to the measured leakage.

Figure 2 shows an electric heat air handler with open panels to help illustrate why pressures vary significantly at the air handler depending upon location of fan, coil and filter. Figure 2 has been edited, using an inserted line, to show a region before the coil that is not as depressurized as the region above where the fan is located. An "A" coil design (not shown) will result in almost the entire exterior surface of the air handler at the "after coil" pressure, whereas the slant coil design often found in units less than 3 tons cooling capacity will have about 33% exterior surface area before the coil. An airflow grid can be seen at the bottom of the air handler where an air filter can be installed. Air filters with higher MERV ratings provide better filtration, but also increase static pressure down stream (closer to the fan). Unfortunately, higher static pressure increases air leakage into the cabinet or other leak locations, resulting in more unfiltered air into the return air stream.

The gas furnace shown in Figure 3 has also been edited with an inserted line to show the bottom portion of the appliance that operates at negative pressure since the fan intake is located here. Everything above this area operates at positive pressure with different pressure regimes between the heat exchanger and coiling coil. All gas furnaces tested had this design.

Figure 2. Heat Pump Coil Design in Air Handler Affects Pressure Distribution

Figure 3. Gas Furnace with Cooling Coil Box on Top of Air Handler

While not within the focus of this paper, several other measurements and inspections were made of the air distribution system such as: heat and cooling system make and model, register and air handler air flow rates, air filter type and location, and $Q_{25,\ total}$ of the entire duct system measured in 26 homes. Q_{25} to outside and $Q_{25,\ total}$ on supply and return were measured separately as well as house tightness (CFM50) in 20 homes.

Test Results

A total of 69 air handlers were tested in 64 houses. Nine appliances were gas furnaces, two were hydronic heat, fifty-seven were heat pumps and one was electric strip heat only. All systems were split DX cooling. The average air tightness of 69 air handlers is 20.4 $Q_{25,\ total}$ in the air handler cabinets, 3.9 $Q_{25,\ total}$ at the return connection, and 1.6 $Q_{25,\ total}$ at the supply connection. The variability in air handler cabinet tightness is indicated in Table 1. There was no significant variability in connection leakage.

Table 1. $Q_{25,\ total}$ Statistics for 69 AH Cabinets

$Q_{25,\ total}$	# AHs	Mean	Median	Minimum	Maximum	Stand. Dev.
Non-Gas	60	19.2	15.3	5.4	46.9	10.4
Gas	9	28.3	25.3	0	74	23.4

These results fit within the range of air handler leakage test results from two groups of tests reported by Wastchak and Ueno (2002). A sample of 253 tests and another 31 more-detailed field tests performed in homes built by three builders claimed an average of 31 $Q_{25,\ total}$ and 17 $Q_{25,\ total}$ in the air handler cabinets respectively. The Wastchak and Ueno studies had a wide variability in air handler tightness; however, no explanation was reported for the variance.

The leakage of the Florida study (Cummings et. al 2002) was measured “as found”. The one exception to this was if the filter access door was off or ajar, then it was placed in its proper position. The filter access door was found removed or ajar in two homes that fortunately had interior located air handlers. In one case, a missing filter access door represented 189 $Q_{25,\ total}$. In the other case, an ajar filter access door represented 37 $Q_{25,\ total}$. One filter access panel design in a very common brand does not close well and can come loose quite easily.

Gas furnaces had significantly more leakage than non-gas heating appliances on average, so results are reported separately by heating type. Operating pressures, tested leakage ($Q_{25,\ total}$), and calculated operational leakage (Q) are summarized in Table 2 for non-gas furnace and Table 3 for gas furnaces.

Calculating $Q_{25,\ total}$ and Q for Positive and Negative Regions of Air Handler

Air handlers can have more than one positive pressure region and more than one negative pressure region. This makes estimating Q more difficult, since there are different holes at different pressures. The exact distribution of leakage among different pressure regions of the appliance was not directly measured (and would be quite time consuming), so estimation was used based upon the percentage of surface area for a given pressure region. The general procedure was as follows: 1) Calculate the weighted average pressure for negative and positive pressure regions using measured surface areas of each region. This results in one weighted positive pressure representing the positive pressure regions and one weighted negative pressure for the negative regions. 2) Estimate $Q_{25,\ total}$ for positive regions and negative regions using total measured surface areas of positive region and total for negative region. 3) Calculate the operational leakage, Q, for the positive and negative regions.

ASHRAE Standard 152 offers a procedure to calculate Q from $Q_{25,\ total}$, but the exact equation would not be appropriate to use in this particular case. This is because standard 152 was designed to evaluate entire duct systems. The air handler leakage measured is in an isolated area that operates at significantly higher pressures than the rest of the distribution system and may or may not be in conditioned space. Equation (C-1) from ASHRAE Standard 152 assumes half of the total leakage measured is from outside the conditioned space and divides $Q_{25,\ total}$ in half. This assumption would not hold true in cases where the air handler is located outside conditioned space, therefore, $Q_{25,\ total}$ is not divided by 2 in the results reported here. The weighted operating pressures in the air handler cabinet were used with the $Q_{25,\ total}$ results to calculate Q for the positive and negative regions using Equation 1.

$$\text{Equation 1: } Q_{neg} = Q_{25,\ total\ neg} * (\Delta P_{neg} / 25)^{0.6}$$

Where Q_{neg} is the operational leakage flow for all negative pressure regions (cfm); $Q_{25,\ total\ neg}$ is the estimated test leakage in the negative portion (cfm); and ΔP_{neg} is the weighted operating pressure for negative pressure regions in the air handler (Pa). The same equation is used for Q_{pos}.

An example calculation is offered here to help elucidate this calculation process. Consider a gas furnace with 20 $Q_{25,\ total}$ and having two positive pressure regions and one negative pressure region. One positive region has 1775 in^2 surface area and is at +115 Pa, a coil box with 1248 in^2 at +103 Pa, and another region with 1890 in^2 at –124 Pa. The first positive region represents 58.7% (1775 in^2 / 3023 in^2) of the total positive area and coil box represents 41.3%.

- STEP1: The positive weighted pressure (ΔP_{pos}) is calculated by:
  (0.587*115) + (0.413*103) = 110 Pa.
  There is only one negative pressure region, so ΔP_{neg} is represented by |-124 Pa.
- STEP 2: Next, $Q_{25,\ total\ neg}$ is estimated where 20 * (1890 in^2 / 4913 in^2) = 7.7 $Q_{25,\ total\ neg}$
  Then, $Q_{25,\ total\ pos}$ is estimated where 20 * (3023 in^2 / 4913 in^2) = 12.3 $Q_{25,\ total\ pos.}$
- STEP 3: Finally Q_{neg} and Q_{pos} can be calculated using Equation 1.
- $Q_{neg} = 7.7 * (124 / 25)^{0.6}$; $Q_{pos} = 12.3 * (110 / 25)^{0.6}$
- This results in 20.1Q_{neg} and 29.9Q_{pos}.

$Q_{25,\ total}$ and pressures at the return and supply connections were measured separately and are shown along with Q in Tables 2 and 3. Electric heat furnaces and heat pumps did not have a positive region and accounted for 58 of the 60 appliances shown in Table 2. One hydronic heat furnace located in an attic and another in a garage account for the two systems with positive pressure regions in the AH.

Table 2. Operating Pressures, $Q_{25,\ total}$, and Q for 60 Tested Non-Gas Furnace Units

	22 Attic	17 Garage	21 Indoors	60 Total
Pressure return connection (Pa)	-69.1	-114.4	-79.4	-85.5
Pressure AH (-) region (Pa)	-125.5	-181.2	-160.1	-153.4
Pressure AH (+) region (Pa)	121.0	36.0	NA	78.5
Pressure supply connection (Pa)	52.8	51.2	48.3	50.8
$Q_{25,\ total}$ return connection	2.1	2.4	3.8	2.8
$Q_{25,\ total\ neg}$ AH (-) region	19.1	18.5	19	19.0
$Q_{25,\ total\ pos}$ AH (+) region	0.6	0.2	0.0	0.2
$Q_{25,\ total}$ supply connection	1.7	1.3	0.7	1.2
$Q_{25,\ total}$ summation	21.7	21.1	22.8	21.9
Q return connection (cfm)	3.9	6.0	7.6	5.9
Q_{neg} AH (-) region (cfm)	50.3	60.7	57.9	56.4
Q_{pos} AH (+) region (cfm)	1.5	0.3	0.0	0.4
Q supply connection (cfm)	2.7	2.0	1.0	1.8
Q AH plus both connections (cfm)	58.4	68.7	66.5	62.6

Table 3. Operating Pressures, Q_{25}, and Q for 9 Tested Gas Furnace Units

	1 Attic	6 Garage	2 Indoors	9 Total
Pressure return connection (Pa)	-50.0	-98.3	-94.5	-92.1
Pressure AH (-) region (Pa)	-66.5	-100.6	-93.0	-95.1
Pressure AH (+) region (Pa)	75.0	120.9	113.8	114.2
Pressure supply connection (Pa)	74	100.7	96.5	96.8
$Q_{25,\ total}$ return connection	1.2	15.8	3.8	11.4
$Q_{25,\ total\ neg}$ AH (-) region	0.0	17.2	3.7	11.6
$Q_{25,\ total\ pos}$ AH (+) region	0.0	21.8	6.8	16.7
$Q_{25,\ total}$ supply connection	0.0	4.7	4.7	4.3
$Q_{25,\ total}$ summation	1.2	59.5	18.9	44.0
Q return connection (cfm)	1.8	35.9	8.4	24.9
Q_{neg} AH (-) region (cfm)	0.0	39.7	8.1	25.9
Q_{pos} AH (+) region (cfm)	0.0	56.1	16.9	41.5
Q supply connection (cfm)	0.0	10.8	10.6	9.7
Q AH plus both connections (cfm)	1.8	142.5	44.0	102.0

There is no significant difference in AH $Q_{25,\ total}$ based upon where the unit is installed; Q however, is greater for air handlers located in garages. While filter media will affect static pressure, it is more likely that greater operating pressures are related to poorer duct design issues such as sharp duct bends made next to the air handler or ducts that are too small. Attic installations were likely to have longer straight duct runs leading up to the air handler. About 17% of the attic systems had filter media, such as pleated textile, and 83% had less efficient filtration media such as economical fibrous woven textile. Garage-located air handler systems had 21% pleated type and 79% fibrous textile. Indoor located systems had only 9% pleated type and 91% fibrous. Only one filter was considered excessively dirty.

Air Handler Leakage and Airtight Code Credit

It is troubling to consider that the amount of $Q_{25,\ total}$ located in approximately 5% of the overall air distribution surface area may be high enough to disqualify a system from meeting a "leak free" goal. The IECC 1998 code describes a "substantially leak free" system as one that does not leak more than 5% of the air handler rated flow when the entire system is at a test pressure of 25 Pa (0.100 In WG). The 2004 Florida Building Code specifies that duct testing measures leakage to outside of the conditioned space. Q_{25} should be no more than 3 cfm per 100 ft^2 of the floor area served by the system to be considered "substantially leak free". There is also a second requirement in Florida code that $Q_{25,\ total,}$ to both outdoors and indoors, be no more than 9cfm per 100 ft^2 to be considered "substantially leak free".

The Florida study found that 4 out of 69 (6%) of all types of units had enough leakage in just the air handler so that the entire duct system would not qualify as "substantially leak free". Two out of nine (22%) gas furnaces (including those with a cooling coil box) and two out of sixty (3%) electric heat air handlers had enough leakage to disqualify the entire duct system as a "substantially leak free".

The average of all systems $Q_{25,\ total}$ and Q are shown in Table 4 as leakage per 100 ft^2 of conditioned floor area, and leakage per rated system flow.

Table 4. Leakage Per 100 ft^2 of Floor Area, and Leakage Per Rated System Flow

	Non-Gas	Gas
$Q_{25,\ total}$ AH	19.2	28.3
$Q_{25,\ total}$ AH+connections	21.9	44.0
$Q_{25,\ total}$ AH / 100 ft^2 floor area	0.9	1.2
$Q_{25,\ total}$ AH+connections / 100 ft^2 floor area	1.0	1.8
$Q_{25,\ total}$ AH / rated flow	1.6	1.9
$Q_{25,\ total}$ AH+connections / rated flow	1.8	2.9
QAH	56.8	67.4
QAH +connections	62.6	102.0
QAH / rated flow	0.046 (4.6%)	0.045 (4.5%)
QAH+connections / rated flow	0.051 (5.1%)	0.068 (6.8%)

Non-gas appliances would have 30% of the allowable leakage (0.9 / 3.0) according to the Florida standard if the all air handlers were located outside the conditioned space. Non-gas appliances had 32% (1.6 / 5.0) of the allowable leakage according to the IECC standard. Gas appliances, on average, had 40% of the allowable leakage per Florida standard and 38% by the IECC standard. The actual leakage is of real consequence as seen in Table 4 where operational non-gas air handler leakage is 4.6% of rated system flow on average and gas furnace air handlers are 4.5% of rated flow on average. This is a concern, when one considers that a 4.6% return leak from a hot attic (peak conditions of 120^{o}F and 30% RH) can produce a 16% reduction in cooling output and 20% increase in cooling energy use (Cummings and Tooley, 1989).

Simulation of energy impacts using the measured $Q_{25,\ total}$ results were conducted for north, central, and south Florida homes and are reported in Cummings et.al (2002). The simulations found that moving the AH from indoors to the attic causes a17% increase in cooling energy for central and south Florida and 15% increase for north Florida. Moving the air handler from indoors to the garage and outdoor locations predicts 9% and 10% increases, respectively, for all regions. Heating degree days can vary significantly from one Florida region to another. The predicted increase in heating energy is 6% for garage locations in all regions, and ranges from 9%-13% for outdoor locations, and 10%-19% for attic locations. South region winters are milder and represent the lowest impact while the north region has the greatest winter impact.

Figures 4 and 5 show a distribution of the number of homes within a range of air handler cabinet leakage, Q, as a % of the system rated flow. Seventy-five percent of the air handlers had estimated operational leakage between 2.1%-6.0 % of the rated system flow. This means that in 75% of all AHs, actual leakage ranges from about 23 cfm to 86 cfm.

Figure 4. Distribution of Q / Rated Flow for 69 Air Handler Cabinets

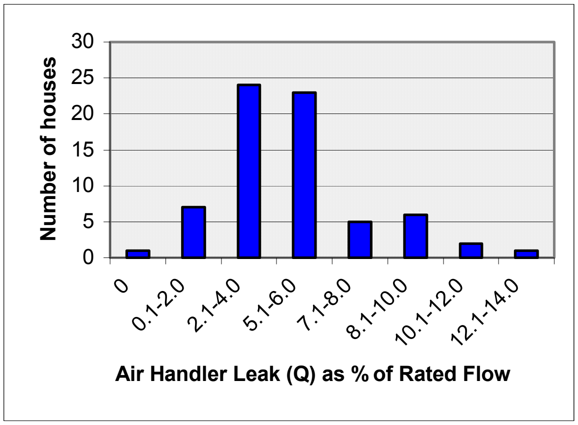
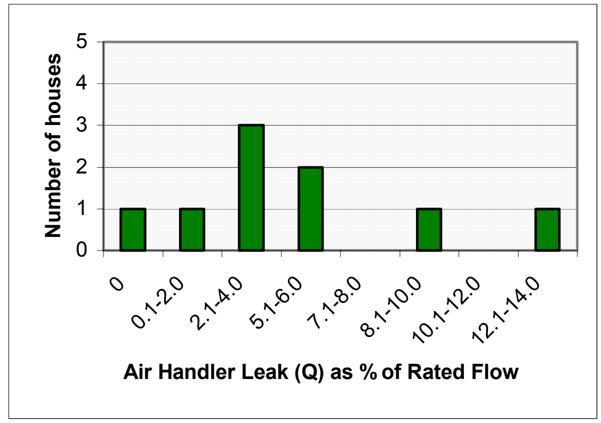


Figure 5. Distribution of Q / Rated Flow for 9 Gas Furnace Cabinets and Cooling Coil Boxes



Characterizing Air Handler Leakage

Air handler leakage, in terms of cumulative hole size, represents a significant portion of the entire system. Air tightness of the entire duct system (including return, supply and air handler) was measured for twenty-six systems. Total duct tightness was found to be166 $Q_{25,\ total}$, on average, while the air handlers had an average of 17.9 $Q_{25,\ total}$, or 10.8% of the entire system. The installers, in this subset of 26 systems, had sealed less than 20% of what could have been sealed at the air handler. This indicates that at least 11% of the entire duct system leakage area (hole size) is at the air handler and as much as 13% if nothing had been done at all to seal the unit!

$Q_{25,\ total}$ varied significantly among different units; however, there was no significant correlation between the physical size of the unit and leakage (R^2=0.19). Wastchak and Ueno (2002) also report that leakage did not change much as it related to system size and that the average air handler leakage for 31 systems was 10% of the total duct system leakage.

The variability in tightness is mostly due to the amount of effort the installer puts into sealing penetrations. It is estimated in the Florida study that of all the things the installers could have sealed at the air handler, only 16% were sealed on average. Statistical analysis indicates that while the highest amount sealed was 90% (occurring once), the median and mode reflect significantly less effort at 10 and 0 respectively. In fact, it was rare to find refrigerant gaskets installed at all, and often they were still packaged inside the air handler.

Detailed visual inspections made on each system found certain types of leaks to be very common. Although subjective in nature, estimation based upon limited tests and detailed inspection is offered here to better understand where the majority of leakage was found and what could be done to eliminate it. The $Q_{25,\ total}$ test did not distinguish how much leakage occurred at specific locations in air handlers, but values were obtained from a few tests after sealing one specific type of leak at a time.

Table 5. Approximate Distribution of Leakage in Air Handler as % of $Q_{25, total}$ AH

	Non-Gas	Gas
Panel seams	45	79
Large refrigerant line	30	NA
Small refrigerant line	20	NA
Wire penetrations	5	7
AC coil box	Does not apply	14
Total	100%	100%

One small and three large refrigerant line penetrations, panel leakage in one popular brand of cabinet, and three coil boxes were tested individually. Table 5 offers an approximation of the percent leakage (hole size) that occurs at different locations in air handlers. About half of the leakage in electric heat furnaces (including heat pumps) is related to refrigerant line penetrations; however, panel leakage accounts for almost the rest. Some air handler models have manufactured holes at the condensate drain connection (Figure 8). This represents about 2 $Q_{25, total}$ of cabinet leakage, but is not represented in Table 5 since it applies more to a specific manufacturer model. While the cooling/heating coil is located within the air handler of an all electric furnace, a cooling coil box is installed on top of gas furnaces and is represented separately in Table 5 as "AC coil box" which includes refrigerant line and coil box panel leakage.

Figure 6. Severe Panel Seam Leakage

Figure 7. Leaky Wire Penetration

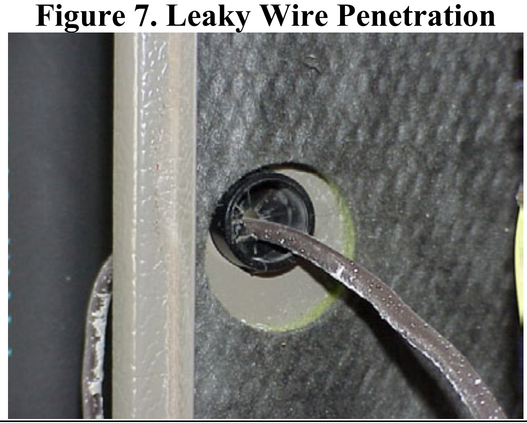


Figure 8. Condensate Drain Penetrations

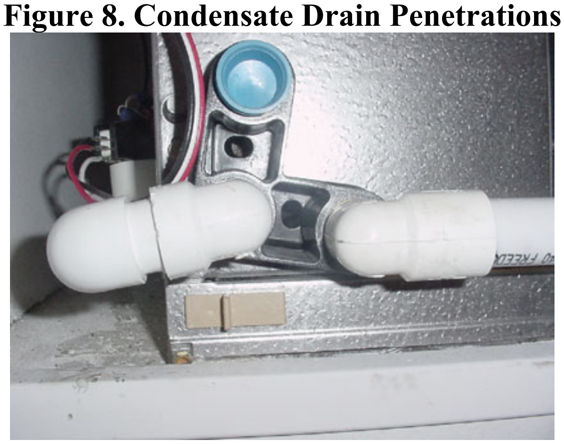
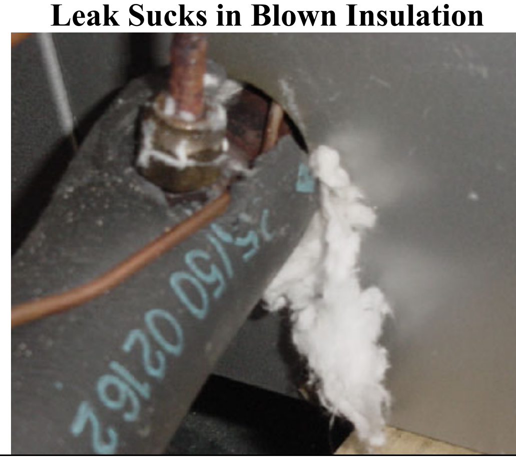


Figure 9. Refrigerant Line Penetration Leak Sucks in Blown Insulation



Conclusions

New air handler installations of this study are not reasonably airtight in most cases. The average air handler leakage, 20.4 $Q_{25,\ total}$, is great enough to result in significant energy penalties and contribute to poor air quality. Most of the leak locations in the air handler occur after the filter, thereby resulting in unfiltered air being pulled through the fan to be distributed around the home. A few situations were documented where blown insulation had been sucked into return leaks in air handlers in attics such as seen in Figure 9. Aerosol pollutant transport is also a concern for air handlers in garages and has been documented as a contributing factor in lethal accidental carbon monoxide poisonings. Space depressurization is yet another potential consequence where dominant return leakage in the air handler may be great enough to interfere with atmospherically vented combustion appliances located in the same space.

In some cases, leakage was great enough in the air handler to disqualify an air distribution system from receiving airtight credit. Consider a worst-case scenario where a "substantially leak free" system has almost all of its leakage in an air handler with rated air flow of 1400 cfm. Five percent of the rated flow would allow 70 $Q_{25,\ total}$. So what if all of the "permitted" leakage is located at the part of the system with the highest pressures? If the average pressure at the leaks is –146 Pa, then the actual leakage will be 201 cfm (nearly 15%) into this "substantially leak free" system! All this is not to say that the 5% test standard is not acceptable, but rather, whatever any standard allows, leakage should not be concentrated at the air handler or connections to it.

At current practice, air handler tightness depends upon two primary things: 1) Manufacturer design, which has the most influence upon panel leakage; 2) Installer effort, which has the most influence upon penetration leakage. Better effort by the installers to use refrigerant line penetration gaskets would have likely reduced the leakage in electric furnaces by half. However, improving the design and application of penetration gaskets would make installing them easier and more likely to be done correctly.

Four air handlers had $Q_{25,\ total}$ less than 7.5 and required 0% added installer effort to achieve this level of air tightness. Three of these had refrigerant penetration areas manufactured with an integrated compressive gasket that the installer simply slides the copper pipe through without extra effort. The panel seams had effective gaskets also.

It was found that some units with inadequate panel gaskets would still leak. Relying on service personnel to seal panel seams is not an effective way to assure long term seals since tape will get torn off whenever service is performed and may not get re-applied. Manufacturers could design air handlers that would eliminate most of the burden (and uncertainty) of sealing the unit by the installer. Using better-designed panel construction, particularly at the filter access location, improved quality gaskets, and manufactured penetration points with integrated seals could be a very effective way to improve air quality and conserve energy in buildings.

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