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Impact of Evaporator Coil Air Flow in Residential Air Conditioning Systems

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Abstract

The performance of conventional split system residential air conditioners is highly dependent on adequate air flow across the evaporator coil. Sufficient air flow is necessary to achieve a proper balance between sensible and latent cooling capacity. Typical target air flow rates are approximately 350 - 450 cubic feet per minute per ton (47.0 - 60.5 L/S per kW) of cooling capacity. The authors have measured the air flow across the coil in 27 installations in Florida. Both flow hood and strip heat resistance methods were used to measure air flow with an established protocol. The installations measured ranged in capacity from 2 to 4 tons (7 - 14 kW). Measured air flows ranged from 130 to 510 cfm per ton (17.5 - 68.5 L/S per kW) with an mean of 320 cfm/ton (43.0 L/S per kW). Reasons for inadequate flows included undersized return ducts and grills, improper fan speed settings, fouled filters and cooling coils. High distribution system static pressures were due to long, circuitous runs and pinched or constricted ducts. Recommendations are made to improve current practice.

Introduction

The performance of a conventional split system residential air conditioner (AC) is dependent on adequate air flow across the evaporator coil to achieve a balance between sensible heat transfer and moisture removal. The Air Conditioning Contractor's Association of America recommends selecting cooling equipment (Manual S) based on its stated sensible and latent performance (from Manual J), designing ducts to accommodate the necessary air flow (Manual D) and adjusting air handler fan speed to match loads (Rutkowski and Healy, 1990; ACCA, 1995a,b,c). However, a problem with this approach is that contractors seldom check in the field to determine if design flow rates correspond with what is achieved.

If coil air flow is too high, air moisture removal is compromised and fan power may be elevated. However if flow is too low, sensible cooling is reduced with degradation of cooling system energy efficiency ratio (EER). Very low air flow may lead to evaporator coil icing, refrigerant flood back and eventual compressor failure.

Manufacturer recommended air flow rates for residential split systems are typically 350 - 450 cubic feet per minute (cfm) per ton (47.0 - 60.5 L/S per kW) of cooling capacity. An airflow of 425 - 450 cfm per ton (57.1 - 60.5 L/S per kW) through a dry coil usually will be needed to achieve 400 cfm/ton (664 L/S per kW) when the AC is operating with a wet coil. These rates are also vital to proper check out of unit performance on installation since superheat and subcooling values in refrigerant charging tables are usually tied to air flows being within a 350 - 450 cfm range (165 - 212 L/S) (Air Conditioning and Refrigeration News, 1989). Assessment of charge when air flows are outside the stated range are invalid.

The air flow produced by an air handler is governed by the indoor unit's fan performance characteristics against the duct system's frictional air flow resistance. The blower fan curve (typically available as tabular data) describes a system pressure versus flow relationship in which air flow increases as the external air flow resistance is reduced. The duct system's performance is characterized by a resistance versus flow relationship; generally resistance through a duct system increases rapidly as more air is forced through the duct. If test and balance data on duct air flow and external static pressure is available for a single point, the entire duct resistance curve is readily derived:

$$R_n = P_1\left(\frac{cfm_n}{cfm_1}\right)^2$$

Where:

R = Duct resistance inches of water column (IWC or Pa) at flow "n"
P1 = External static pressure at test point (IWC)
cfm1 = cfm flow at test point
cfmn = cfm flow "n"

The two curves can be plotted against each other as shown in Figure 1 to determine the system operating point and the corresponding cfm (L/S) of achieved air flow.

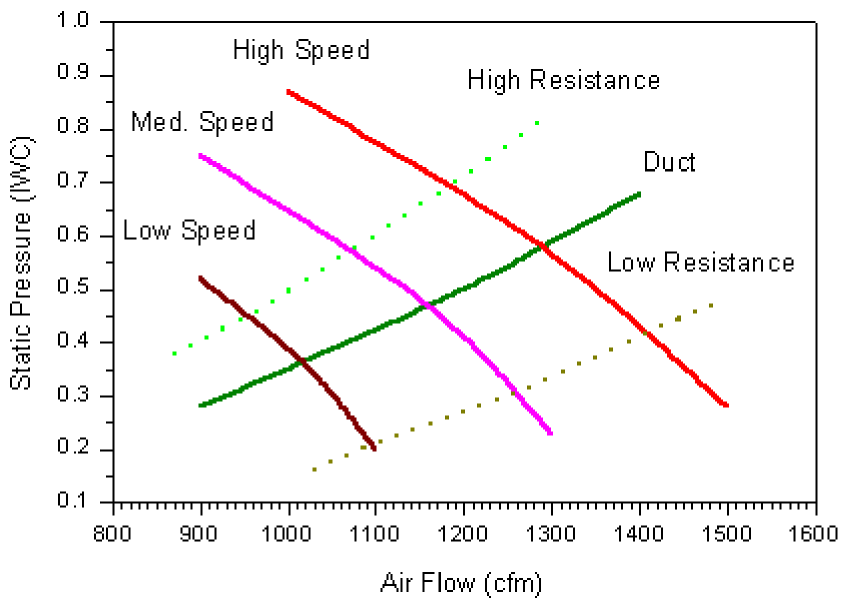


Figure 1. Influence of fan performance and duct flow resistance on system operating point.

The operating point, where the fan curve intersects with the duct system resistance curve, is important since it represents the only operating condition obtained when a single blower is mated to a given duct system. Since often there are three or more speeds available to the blowers, there are a corresponding number of operating points as shown in the example. As illustrated, air side performance is also strongly influenced by the duct system flow resistance.

Establishing the operational duct system resistance prior to mating with a fan is difficult since most systems are site built. The design aspects of such systems are covered in detail in Manual D (ACCA, 1995). However, the resistance of the duct system may be affected by unintentional aspects of the duct system installation and operation. Installation irregularities include site compromises to duct system design, constricted or pinched and collapsed ducts.(2)

Also, use of popular duct slide rules may be misleading if other sources of non-duct pressure drop (such as coils, dampers, filters, grills, resistance heaters) are not subtracted from the available fan static pressure. On the other hand, homeowners can reduce air flow by adding high efficiency filters, or allowing filters and/or coils to become soiled. They can also close off supply registers in an attempt to zone spaces or control room temperature distribution.

Background

A survey of 492 heat pump and air conditioner contractors showed that, on average, residential air conditioning systems are replaced every 12 - 14 years (Lewis, 1987). Moreover, a significant portion indicated that consumers decided to replace their existing system before the end of its useful life, based on a perception of inadequate cooling capacity and/or excess operating costs. Thus, system performance quality emerges as a potential factor in the useful life of AC equipment.

Past research has shown an inconsistent record with respect to installed performance of residential air conditioning (AC) installations. A field assessment of 27 central air conditioners in central California found field-measured EERs to be only 79% of rated values (Sherman et al., 1987). Field tests on 70 AC systems in Arizona discovered that "as found" EER was 40% lower than rated (Kuenzi, 1988). A field evaluation in North Carolina concluded that three of ten examined air conditioning systems had low evaporator air flow (Neal, 1987) and seven of the random sample had improper refrigerant charge.

Proctor has tested many residential air conditioning systems for California utilities. In an evaluation of 15 sites with high bill complaints in Fresno, he found that 10 (67%) had low evaporator air flow and improper charge was discovered in 53% of installations (Proctor, 1991). The largest study for PG&E is one of 999 randomly selected homes in which air flow was measured (Kinert et al., 1992). A total of 44% of the homes were found to have evaporator air flow less than 350 cfm/ton (47.0 L/S per kW) (typically seen as a level below which corrective action is necessary). A detailed study of 37 existing installations for Southern California Edison found an average evaporator flow rate of 300 cfm/ton (40.3 L/S per kW); 80% of the systems were below the 350 cfm/ton (47.0 L/S per kW) level (Proctor et al., 1995). In this study, repairs were effected to increase flow. This involved opening or enlarging grills, replacing dirty filters, cleaning evaporator coil and increasing blower speed. The average post-repair flow rate increased to 322 cfm/ton (43.3 L/S per kW). The study also noted that HVAC contractors who had been previously called out to the homes had not identified or solved the actual problems with the systems.

Similar research was conducted by the same firm for utilities to examine new air conditioner installations. Testing of 37 new residential AC systems for Nevada Power Company in the Las Vegas area found an average flow of 345 cfm/ton (77.0 L/S per kW) (Blasnik et al., 1995a). Half of the units were below 350 cfm/ton (47.0 L/S per kW) and 30% were below 300 cfm/ton (40.3 L/S per kW). Similarly, the average measured flow in 28 new installations tested for Arizona Public Service Company was 344 cfm/ton (46.2 L/S per kW) with more than half the unit below 350 cfm per ton (47.0 L/S per kW) (Blasnik et al., 1995b). Testing of ten new installations for Southern California Edison averaged only 319 cfm/ton (42.9 L/S per kW), with all but one unit below the 350 cfm/ton (47.0 L/S per kW) action level (Blasnik et al., 1995c). In nearly all cases, the new system had air handlers that were capable of delivering the necessary cfm. However, manometer measurements revealed high external static pressure of the duct system (averaging 145 Pa). Undersized returns and filters were identified as the main culprits responsible for the low air flows.

Objective

The intent of our research was to extend the work performed elsewhere to examine conditions prevalent in Florida where residential air conditioning is extremely widespread. Our goals were:

- To measure the adequacy of evaporator air flow rates in field installations
- To evaluate the potential impact of the conditions encountered
- To examine the impact of remedial measures taken on some of the installations.

Measurements

Project technicians visited a total of 27 residential sites in the summer of 1996. The sites were divided between Central and South Florida. The sample was selected and was not intended as a true statistical representation of AC installations, but rather as a series of case studies. Several installations were chosen to allow examination of particular aspects. Two of these were installations where special care had been taken to obtain a good air conditioner installation and another was a case with known problems (coil icing and high bill complaints). Some effort was made to obtain a mix of vintages (both old and new construction) as well as household demographics and system type.

At all sites the air handler return air flow at the evaporator was measured using one of two methods with an established protocol:(3)

1) A calibrated flow hood which was mounted directly over the return grill

2) Electric resistance heat elements used to estimate air flow

The flow hood is a commercially available model with digital readout and air density correction and a specified accuracy of 5% + 5 cfm (2.4 L/S) from 0 to 2500 cfm (1180 L/S). For systems with a single return, the flow hood was placed over the grill and cubic feet per minute (cfm) or liters per second (L/S) measurements taken under standard operation (wet coil). One problem is that the flow hood will under estimate actual coil air flow if significant leakage exists between the return inlet and the coil. Consequently, this method was only used where inspection suggested that return air leaks between the return grill and the coil were minimal.

The second way of determining in situ coil air flow is know as the "resistance heat method." In this procedure resistance heat elements for heating systems or heat pump emergency heat coils are used to derive air flow given known physical properties. Holes are prepared in the supply and return ducts to insert calibrated temperature probes as close to the evaporator coil as possible, but not so near as to be within the line of sight of the resistance heating elements. The air handler fan was activated and run until all evidence of moisture accumulation in the coil was eliminated.(4) Circuit breakers were then turned off for the entire house, except for the heating system and the thermostat was set to maximum heat. The system was then allowed to come into equilibrium for five minutes after which the temperature difference before and after the coil is recorded along with the power demand on the utility meter. The air flow rate in cfm is then determined:

$$cfm = Watts * C / (Ts - Tr)$$

Where:

cfm = cubic feet per minute
C = 3.16 or 3.413 Btu/W/ (60 min/h * 0.24 Btu/lb. * 0.075 lb/ft3)
Ts = supply air temperature after coil (oF)
Tr = return air temperature before coil (oF)

and:

$$Watts = 10 * kh * 3600 / t$$

Where:

kh = the meter watt-hours per revolution (typically 7.2)
3600 = seconds per hour
t = time for ten revolutions (sec)

At one site, both methods were used to measure air flow to verify their approximate repeatability. The results showed agreement of the two methods to within 5% (40 cfm) on dry coil air flow. It is noteworthy, however, that this data also showed that dry coil air flow was approximately 10% greater than a wet coil under standard operation.

Where possible, duct system external static pressure was obtained by differencing the return and supply side duct pressures. This was accomplished using a digital manometer with static pressure probes before and after the system air handler.

Results

The results from our tests of the existing AC installations are presented in Table 1. All supply ducts were left in their current configuration and filters were not changed prior to making the tests.

Table 1

Measured Evaporator Air Flow in Existing Installations

House Designation	Age of AC	Cool Cap.	Test cfm	cfm/ ton	Post Audit cfm	Post Audit cfm/ ton	Post Audit Ext. Static Press. (IWC)	Comments
SC1	16	3.0	945	315	----	----	----	old unit
WD1	8	2.5	865	346	1011	404	0.91	small return, increase fan speed
PF1	5	2.0	666	333	690	345	0.58	replace dirty filter
PF2	5	2.0	644	322	----	----	0.61	----
DBF1	8*	2.0	405	203	650	325	----	old system, clean filter, adjust drive belt
MF1	15	2.5	918	367	1061	424	0.45	increase fan speed, low charge
CH1	2*	3.0	754	251	1091	364	0.78	increase fan speed, filter, clean coil
PJ1	10*	3.5	592	169	----	----	0.68	undersized return, constricted ducts, mismatch unit

MM1	2*	4.0	520	130	----	----	----	oversized outdoor unit, small returns, iced over coil
RS1	6	3.0	755	251	----	----	----	undersized return, dirty coil
DS1	7	2.0	972	486	1011	505	0.75	change filter, leaky return, oversized indoor unit
RV1	5*	2.5	745	298	760	304	0.36	change filter, refrigerant restriction
Average	5	2.47	732	289	896	382	0.63	

* Indoor unit is older.

The group of 12 existing installations had an evaporator air flow of 289 cfm/ton (38.8 L/S per kW). All but two of the measurements were made with a dry coil, so that wet coil performance is likely less than 275 cfm/ton (37.0 L/S per kW). Five of the systems (42%) had flows of less than 260 cfm/ton (34.9 L/S per kW). On seven of the units repairs of various types were performed. These included changing filters, fan speeds and cleaning coils. The average air flow increased from 289 to 382 cfm/ton (38.8 - 51.3 L/S per kW) -- a 24% increase.

A second set of existing AC installations were tested in Homestead, Florida that are part of a project where the energy end uses of ten low income homes are being extensively monitored (Parker et al., 1996). All of the homes were four years old of similar construction with identical central air conditioning equipment. The testing found that evaporator air flows were very low due to clogged filters. Also, installers had located two separate filters on the systems leading to an even higher static pressure resistance on the return side of the air handler fan. In most cases this additional filter was removed and a new primary filter was installed.

Table 2

Measured Evaporator Air Flow in Low Income Homes

House Designation	Cool Cap.	Test cfm	cfm/ ton	Post Audit cfm	Post Audit cfm/ton	Comments
WG1	2.0	340	170	420	210	Two filters, one removed, filter dirty
CS1	2.0	350	175	380	190	Two filters, one removed, filter dirty
AG1	2.0	360	180	440	220	Two filters, one removed, filter dirty
TM1	2.5	460	184	580	232	Two filters, one removed, filter very dirty
KT1	2.0	400	200	380	190	No filter, primary filter added
WB1	2.5	580	232	610	244	Two filters, one removed, filter dirty
TM1	2.5	390	156	620	248	Two filters, one removed, filter very dirty
MJ1	2.0	380	190	440	220	Two filters, one removed, filter dirty
DF1	2.0	370	185	410	205	Two filters, one removed, filter dirty
Averages	2.16	403	186	476	218	

The measured flows for the group of low income homes were extremely low and demonstrate the need for improved maintenance. Our findings illustrate that in many cases, the home owners were unaware of system filters or that they needed to be cleaned. Removing the second filter and changing the primary filter was shown to improve flow by an average of 15%. Flow was still deficient, however, and future work may examine the potential of a fan speed increase.

As shown in Table 3, six of the systems inspected were new installations less than one year old.

Table 3

Measured Evaporator Air Flow in New Installations

House Designation	Cool Cap.	Test cfm	cfm/ ton	Ext. Static Pressure (IWC)	Comments
MB1	3.0	1010	337	0.49	Existing ducts, new indoor and outdoor unit
PJ2*	2.0	1019	510*	0.58	Variable speed system, ducts modified
AM1	3.0	1105	368	0.49	Existing ducts, two filters on return; 378 cfm/ton with one
DP1	2.0	620	310	0.53	Existing ducts, two filters; 335 cfm/ton with one
JS1	2.0	602	301	0.27	Existing ducts, new indoor and outdoor unit
JS2	2.0	645	323	0.35	Existing ducts, new indoor and outdoor unit
Average	2.47	883	328	0.45	

* Not included in average.

The average indoor unit fan flow in the new installations was 328 cfm/ton (73.2 L/S per kW). One installation is not included in this average. This indoor unit had a dry coil air flow of 510 cfm (241 L/S) and was performed by a technician who knew that air flow would be tested in the project. In this installation the existing return duct system was modified to minimize pressure drop and optimize performance. The air conditioner was a variable speed indoor unit which frequently operates at lower flow rates to provide improved moisture removal.

The average air flows for all three groups of 27 tested homes ranged from 130 to 510 cfm per ton (17.5 - 68.5 L/S per kW) with an average of 270 cfm/ton (36.3 L/S per kW). A frequency distribution of measured air flows from our study is shown as Figure 2.

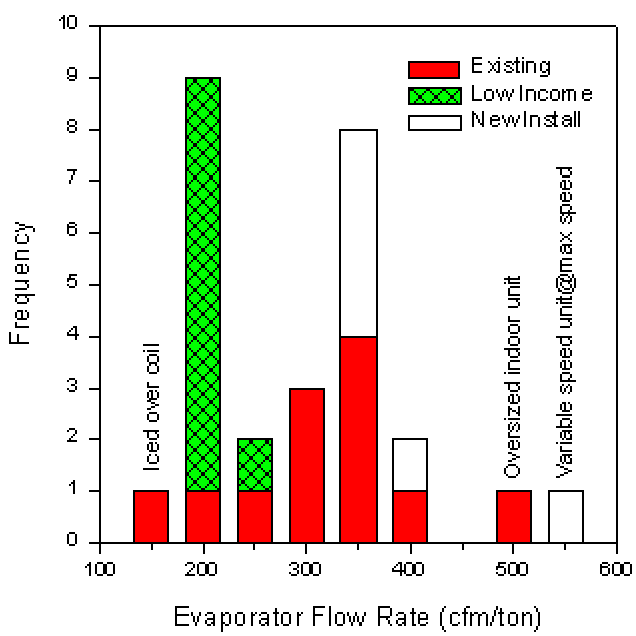


Figure 2 Histogram of evaporator airflow rates

Performance Assessment

Several recurring factors were found to account for the inadequate flows:

- Return ducts and return grills were often undersized
- Fans were set to medium rather than high speed for cooling operation
- Filters and cooling coils were dirty with high flow resistance
- Duct system static pressures were elevated due to circuitous runs, pinched ducts etc.
- Larger outdoor units were installed without changing the indoor unit.
- Devices had been added which increased system static pressures.

Typical static pressure difference before the fan to after the coil in existing installations averaged 0.54 inches of water column (134 Pa). The Air Conditioning Contractor's Association of America (ACCA) duct design manual (Manual D) suggests that typical static pressure difference before the fan to after the coil should be approximately 0.40 inches of water (100 Pa) and AC systems are rated at ARI condition of 0.2 IWC (49.8 Pa). This suggests that some of the problems were due to duct system sizing and restrictions that caused low air flow. In other cases the air flow problem was exaggerated due to changing the outside condensing unit to one of larger capacity, while not replacing the undersized indoor unit.

We also observed the impact of newer higher efficiency pleated filters. These filters often have higher pressure drop than standard "disposable" models. In one case, flow was observed to drop by 25 cfm (12 L/S) (4%) when substituting for a new conventional filter. In a second test at another home the change was a 30 cfm (14 L/S) (5%) flow reduction from adding a higher efficiency filter.

Occupants can also increase system static pressure in ways that are difficult to anticipate. For instance, heavily soiled filters were observed in the project to add up to 0.23 IWC (58 Pa) to system static pressure. Also, a duct system with a measured external static pressure of 0.50 IWC (124.5 Pa) @ 670 cfm (316 L/S) was shown to increase to 0.54 IWC (134 Pa) by closing a single ceiling supply register.

A survey of AC contractors in Florida (Vieira et.al., 1996) showed that 88% used duct slide rules to design duct systems. Further discussions revealed that a friction loss rate of 0.1 IWC per 100 feet (24.9 Pa per 30 meters) equivalent duct length is almost universally used in spite of ACCA recommendations condemning such a rule of thumb.(5) However, the problems observed in our assessment suggest that if such an approximation was to be used, one based on 0.05 IWC per 100 feet (12 Pa per 30 meters) would result in more adequate performance under typical operating conditions.

Impact of Low Air Flow: Simulation

We used a computer simulation to better understand the impact of low AC evaporator air flow on cooling performance. To simulate AC performance under degraded evaporator air flow in a humid climate, it is important to accurately predict SHR (sensible heat ratio) and EER (COP) at off-design conditions. The AC model used for our study was developed based on the DOE-2 default performance curves combined with the apparatus dew point/bypass factor approach to predict off-design sensible and latent performance. A more complete description of the model and its development is contained in work by Henderson et al. (1992). The main advantage is that any system can be simulated with minimal information; only total capacity, EER and SHR at rated conditions are required to produce the entire machine performance map over a range of conditions. However, the performance map for degraded air flow is only valid over a restricted range. Generally the model is reliable for evaporator air flows between 500 and 300 cfm per ton (67.2 - 40.3 L/S per kW).

Figure 3 shows the simulated impact of evaporator air flow from the model.

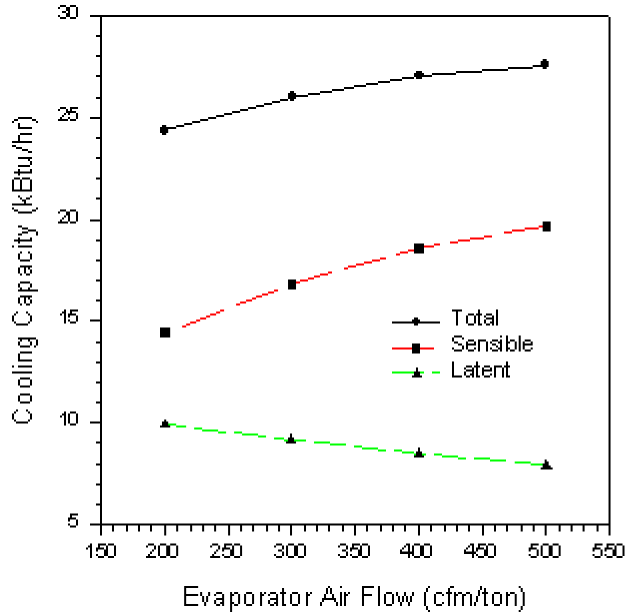


Figure 3 AC capacity vs. evaporator airflow.
Simulated results: 75^0 IDB, 60% RH, 95^0 ODB

A standard air conditioner is shown with an EER of 9.0 Btu/W (9.5 kJ/W) and a 0.80 SHR at standard ARI conditions. The air conditioner is simulated with a 75oF (24oC) indoor temperature and 60% RH and a 95oF (35oC) outdoor condition. Total, sensible and latent cooling capacities are plotted for air flows between 500 and 200 cfm (236 and 94 L/S). As expected, latent capacity is increased by reduced coil air flows, although both sensible and total cooling capacity are adversely impacted. The results predicted that a reduction in coil air flow from 400 to 300 cfm (189 - 142 L/S) would increase latent capacity by 720 Btu/hr (211 W) (8%), but would lower sensible cooling capacity by 1,730 Btu/hr (507 W) (10%). Sensible cooling EER, which governs residential air conditioning consumption in thermostatically-controlled systems, is reduced by a similar amount.

Past Laboratory Testing

Significant work in determining the potential impact of air flow on performance was performed in a series of bench tests by Palani et al. (1992) on a three-ton air conditioner. In the test with a 95oF (35oC) outdoor condition, the evaporator air flow was arbitrary decreased by obstructing supply from a standard rate of 1135 cfm (536 L/S) to values representing a 25% (843 cfm; 398 L/S), 50% (544 cfm; 257 L/S), 75% (254 cfm; 120 L/S) and 90% (109 cfm; 51 L/S) reductions. Although power demand dropped slightly with decreased air flow, EER dropped rapidly beyond a 50% air flow reduction. A 25% reduction in air flow produced a 4.2% decrease in the nominal EER (9.53 Btu/W; 10 kJ/W). The reductions to EER were 6.5%, 34.6% and 71.1% for 50%, 75% and 90% drops in air flow, respectively. Figure 4 illustrates their data.

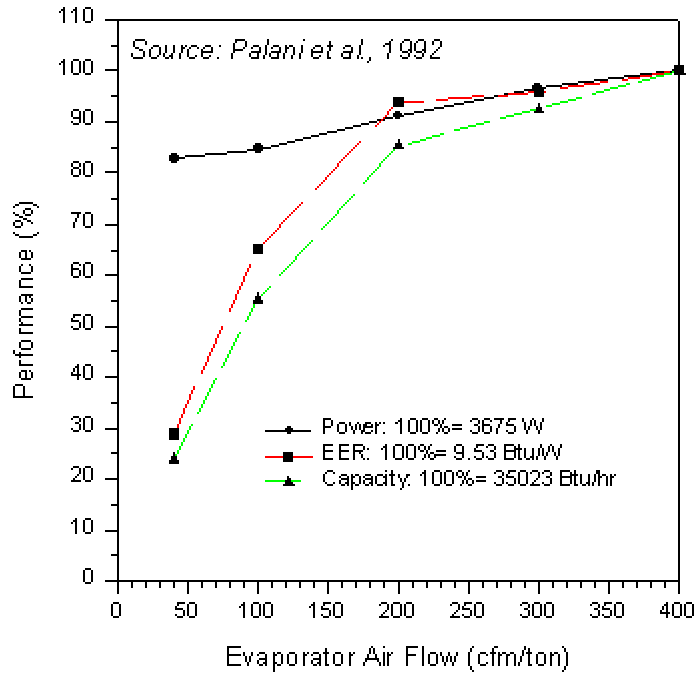
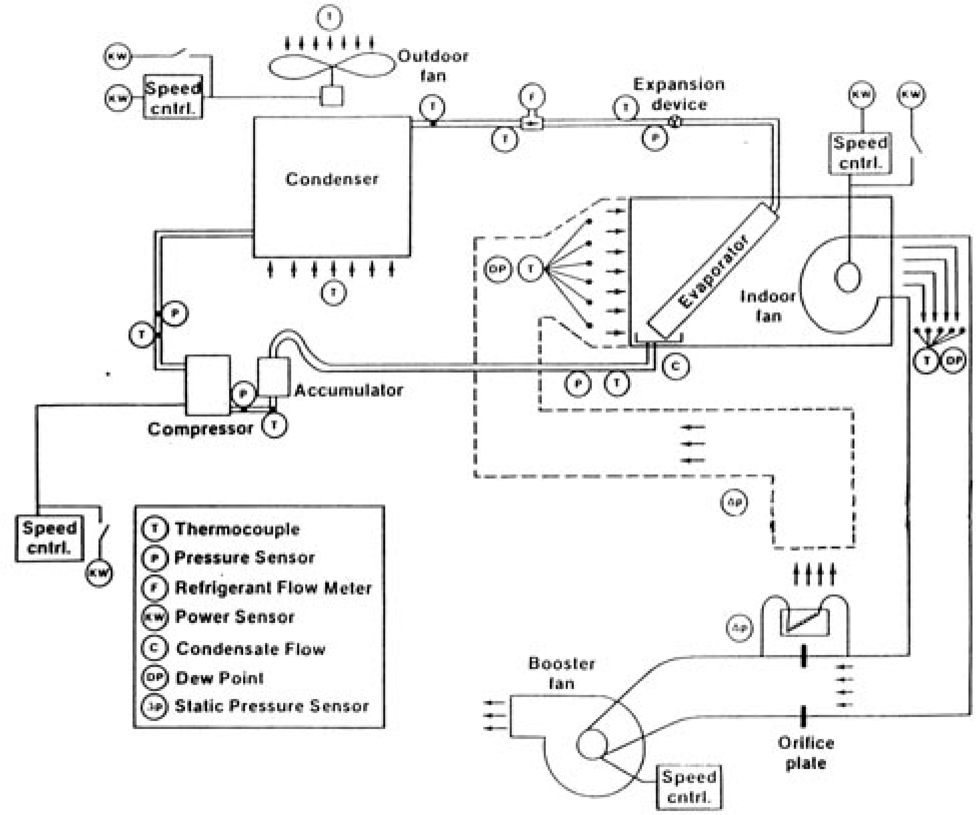


Figure 4 Impact of reduced airflow on AC performance

Since most residential systems operate by thermostat control, sensible cooling capacity more closely represents the potential impact of reduced evaporator air flow on site cooling energy consumption. Sensible EERs were more strongly affected since an increasing fraction of the cooling became latent at the lower air flows. Within the tests, sensible EER dropped by 10.7% for a 25% reduction in air flow and 21.3%, 48.2% and 86.9% for reductions in air flows of 50%, 75% and 90%.

Laboratory Test of Impact of Low Air Flow

To expand on previous work, a test bench was established to evaluate the impact of reduced evaporator air flow in a controlled environment. The testing took place in a laboratory for testing residential air conditioners with cooling capacities up to 3.5 tons (12.32 kW). The laboratory consists of two environmental chambers: one controlled at the desired indoor conditions and the other controlled at the desired outdoor conditions. The chamber conditions are maintained automatically with a data acquisition and control system (DAS). The configuration of the laboratory is illustrated in Figure 5.



A more complete description of the test apparatus, instrumentation and data acquisition system is available elsewhere (Hancock, 1989).

A total of 21 quantities were measured by the data acquisition system. These include refrigerant temperatures and pressures to provide refrigerant side performance data, as well as dry-bulb temperatures and relative humidities to obtain air-side data. Copper constantan thermocouple were used to take dry-bulb temperatures and chilled mirror hygrometers were used to take dew point measurements. Refrigerant mass flow was measured at the liquid line before the expansion device. Air flow rate was measured using the pressure drop across a square edged orifice plate designed according to ASME MFC-3M-1984. A speed controlled booster fan is used to make up for the static pressure loss through the orifice. Electrical power for the compressor, condenser fan and indoor fan unit are individually metered. External static pressure was measured with a differential pressure transducer.

The air conditioner is variable speed unit rated at 35.2 kBtuh at 84 hz. However, for the purposes of the desired tests, we ran the compressor at a constant 60 hz to mimic the performance of a single speed unit. The condensing unit houses a scroll compressor with a circular spine-fin single circuit condenser coil with a one row 16 fins per inch (6.3 fins/cm) configuration. The total surface area of the condenser coil measures 17.1 ft2 (1.6 m2). A slab type three row, 13 fins per inch (5.1 fins/cm) evaporator coil (4.4 ft2; 0.41 m2) is mounted within the indoor air handler. Refrigerant control is provided by an electronic metered system which we set up to mimic the performance of a thermostatic expansion valve.

The tests were conducted over a 30 minute period according to the DOE/ARI specification for steady state testing of unitary air conditioners. This did limit, however, our ability to test very low air flows where damage to the compressor unit was possible. The reduction of evaporator air flow was simulated using a restriction plate on the supply duct outlet to increase the fan external static pressure.

The outdoor portion of the environmental chamber was maintained at a 95oF (35oC) condition, the indoor chamber was maintained at a 75oF (24oC) dry bulb with a 60% relative humidity to approximate the typical residential conditions seen in the field. Control was very good with maximum deviations less than 0.3oF (0.17oC) from either condition for the duration of the test. One test was performed at an 80oF (27oC) indoor condition with a 67oF (20oC) wet bulb to obtain performance at ARI conditions. For each test, data were taken every 52 seconds on available refrigerant and air side parameters. Sampling began after steady state conditions were reached 20 minutes into the test. Air and refrigerant side enthalpy, humidity and volumetric measurements were made using an embedded program with the data acquisition system. The final data were then output from the valid measurement period.

The machine performance at the ARI test condition showed a capacity of 29,254 Btu/hr (8571 W) thermal with a 3888 W electric power draw for an EER of 7.52 Btu/W (59.7 kJ/W) at a coil air flow of 425 cfm/ton (57.1 L/S per kW). The SHR was 0.79 at the ARI condition. Table 4 and Figures 6 and 7 summarize the results from the tests conducted to examine sensitivity to air flow.

Table 4

Laboratory Test Results of Reduced Evaporator Air Flow on AC Performance
(All tests at 75oIDB, 60% RH, 95oF. ODB) Nominal capacity is 2.44 tons

Air Flow (cfm)	Flow (cfm/ ton)	Leaving Evap. Air Temp. (oF)	Discharge Press. (psia)	Suction Press. (psia)	Entering Air-Side Enthalpy (Btu/lb)	Leaving Air-Side Enthalpy (Btu/lb)	Power (W)	Indoor Fan Power (W)	Ext. Static Press. (IWC)	EER (Btu/w)	Sens. EER (Btu/w)
1036	425	58.8	194	94	30.20	24.35	3872	433	0.70	7.04	4.67
876	359	57.5	192	92	30.21	23.54	3804	386	0.83	6.91	4.35
617	253	54.1	186	88	30.16	21.56	3673	324	1.06	6.50	3.77
531	218	52.8	184	86	30.19	20.77	3616	307	1.15	6.22	3.52
475	195	51.5	182	84	30.20	20.05	3565	289	1.21	6.09	3.38
309	127	47.0	175	78	30.22	17.58	3426	251	1.30	5.13	2.73

The results complement the findings of Palani et al. (1992). Figure 6 clearly shows the sensitivity of sensible capacity to flow: capacity drops by approximately 15% from a drop from 425 to 300 cfm/ton (57.1 - 40.3 L/S per kW) whereas latent capacity is increased by roughly 7%.

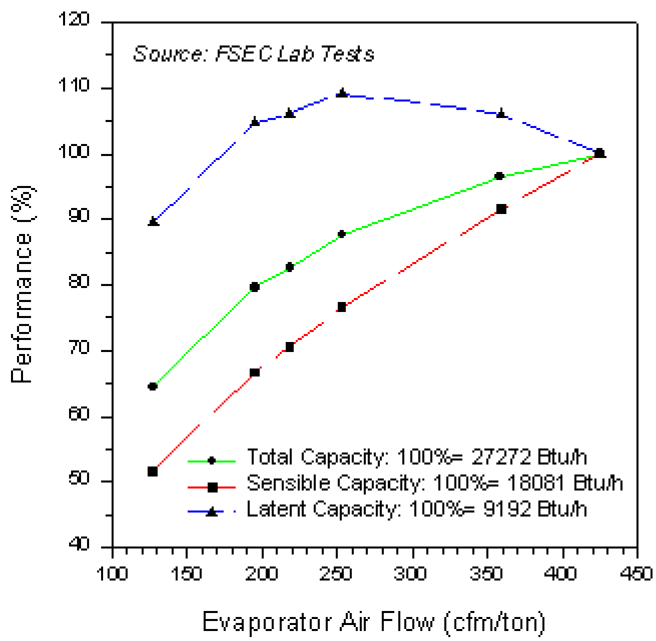


Figure 6 Impact of reduced airflow on cooling capacity

By the point that 200 cfm/ton (332 L/S per kW) is reached, sensible capacity has dropped by nearly 35% and total capacity by over 20%. Interestingly, however, the data show that humidity removal peaks at approximately 250 cfm/ton (415 L/S per kW) and falls rapidly below 200 cfm/ton (26.9 L/S per kW). Generally, the data show the wisdom of the recommendation not to allow air flow to drop below 350 cfm/ton (581 L/S per kW). Ninety-five percent of potential latent capacity is achieved at this point at a cost of only about 10% in sensible capacity.

Figure 7 presents the same data when electrical consumption is considered.

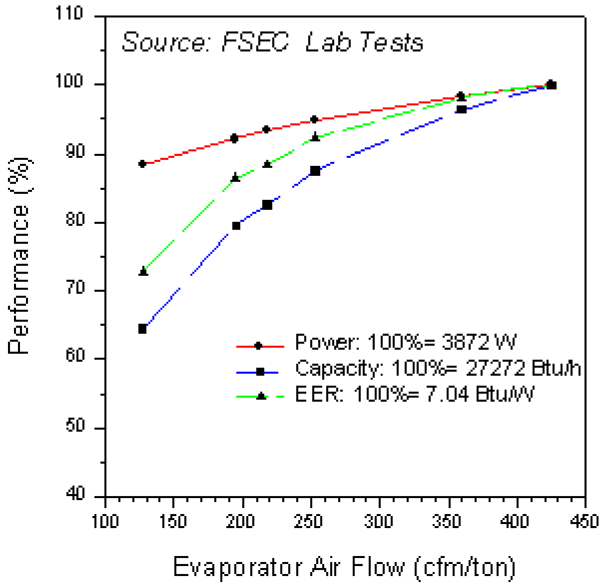


Figure 7 Impact of reduced airflow on cooling performance.

Power demand falls off in a nearly linear fashion between 425 and 125 cfm (201 - 59 L/S) as compressor head pressures diminish. Total capacity and EER drop in a non-linear fashion below 300 cfm/ton (498 L/S per kW) with sharp reductions below 200 cfm/ton (26.9 L/S per kW). If sensible EER is considered (which is most representative of the impact of coil air flow on residential cooling energy), the results suggest a 20% increase in site cooling energy consumption from a flow of 425 - 250 cfm/ton (57.5 - 33.6 L/S per kW).

Discussion

Both the simulation and empirical results from two laboratory test suggest that the lower evaporator air flow rates observed in our field measurements (300 vs. 400 cfm; 142 vs. 189 L/S) might produce a 10% increase in residential cooling energy use over what would be expected based on rated performance. However, increasing cooling system sensible performance by increasing air flow will reduce latent cooling. Warmer coil surface temperatures will result in lower rates of moisture removal and also, the improved sensible cooling performance will shorten duty cycle run-times.

It is noteworthy that consumers are very concerned with energy costs in home comfort systems. Responding to a survey of 80,000 households, 46% of potential buyers indicated the low energy costs was their most important factor in choosing a new air conditioning system (Contracting Business, 1995). However, those responding to the same survey indicated that better humidity control was the characteristic in their systems most in need of improvement (41%). Obviously, consumers desire both low energy costs and improved moisture removal. Proper equipment sizing to produce suitably long duty cycle run-times for effective moisture removal, choice of equipment with low SHR and proper evaporator air flow to produce rated efficiency can potentially address both consumer issues.(6)

Need for Low Flow Resistance Residential Duct Systems

Beyond improving evaporator airflow, reducing fan power and duct leakage are two further reasons to promote duct design with lower external pressure drop than those encountered in our study. With reduced air flow resistance comes improved energy efficiency due to decreased fan power. Since fan energy is related to the work performed (product of air pressure and volume over efficiency) the relationship below can be used to estimate fan power for specific conditions:

W = 0.11755 cfm(delta)P/ (Nu)fan (Nu)motor

Where:

W = fan power in watts
cfm = cubic feet per minute at system operating point
P = total system external static pressure including that of the evaporator coil
(Nu)fan = fan efficiency (typically ~45% for forward-curved centrifugal blowers)
(Nu)motor = motor efficiency (permanent split capacitive motor ~ 50%)

For instance a duct system moving 800 cfm (378 L/S) with a pressure drop similar to that measured in our study (0.63 IWC or 157 Pa without coil and 0.83 with coil) would result in a power draw of 347 W. However, a duct system with a total pressure drop of only 0.2 IWC (50 Pa) (0.4 IWC or 100 Pa with coil) would produce a power demand of only 167 W -- a fan power reduction of 52%. If the compressor electrical demand was 1800 W to produce 24,000 Btu/hr (7032 W) of cooling at the coil (not including fan energy), the improvement would alter EER from 10.63 to 11.91 Btu/W (12.56 kJ/W) -- an 10% net increase in cooling efficiency and capacity.

The second reason to reduce duct air pressures is to minimize the impacts of duct leakage. The pervasive problems associated with duct leakage in residential AC systems are well documented and will not be reported here. Although a leak free air distribution system is desirable, the reality of current duct system fabrication and assembly makes leaks inevitable. Leakage from specific sites is strongly related to pressure difference. For instance, Figure 12-9 of ACCA's Manual D shows that even sealed flex duct will still experience a leakage rate of 6.5 cfm/100 square feet (.33 L/S per m2) of duct at an average duct static pressure (along its length) of 0.25 IWC (62.25 Pa) (ACCA, 1995b). This falls to only 2.7 cfm/100 (0.11 L/S/m2) at 0.1 IWC (24.9 Pa) for a 60% reduction in duct leakage.

Conclusions

A field study was completed to evaluate the adequacy of air handler air flow in 27 residential air conditioner installations. The data presented in this study, as well as others previously cited, suggest that evaporator air flow in residential air conditioning systems is often deficient relative to manufacturer guidelines which typically call for 400 cfm/ton (53.8 L/S per kW). That our tests showed an average flow 317 cfm per ton (42.6 L/S per kW) for our base sample of existing installations with two thirds of the sites below the 350 cfm/ton (47.0 L/S per kW) action level. Flow varied from 130 - 510 cfm/ton (17.5 - 68.5 L/S per kW). New installations generally had greater flow. Results were even worse for a group of nine low income houses which had an average "as found" air flow of only 184 cfm/ton (24.7 L/S per kW). Air flows below 350 cfm/ton (47.0 L/S per kW) render invalid most standard tests for determining refrigerant charge and can lead to improper charging by service personnel who often do not check air handler flow.

Low evaporator air flows also have energy-efficiency implications. Test data taken by Palani et al. (1992) and simulation and test bench data produced by this study suggest that a 25% reduction in air flow from 400 to 300 cfm/ton (53.8 - 40.3 L/S per kW) can reduce typical AC system EER by approximately 4%. However, sensible EER, which controls cooling system energy use under thermostat control, is reduced by about 10%. Reduction to evaporator air flow below 200 cfm per ton (26.9 L/S per kW) can lead to coil icing and greatly shorten compressor life.

We conclude that improving evaporator air flow to rated values (often 400 cfm/ton or 53.8 L/S per kW) in residential air conditioning systems has the potential to reduce average residential cooling energy use by approximately 10%. This will be best accomplished in new installations by properly sizing duct systems and return grills to reduce duct system static pressure. It was also shown that proper duct design aimed at reducing system static pressure has the potential to reduce fan energy by half and to improve overall system EER by 12%. Over sizing of AC system capacity should be avoided to provide adequate dehumidification through long duty cycle run-times.

Another reason to provide emphasis to low flow resistance duct design is the tendency in modern residential AC systems to add increased air filtration. Measurements within the project showed that substitution of pleated "high efficiency" filters typically reduce system air flow by 5%.

In existing installations, constrictions which increase return or supply duct pressure drop should be addressed and fan speeds set according to measured return air flow. In instances where existing air flow is deficient, installation of a larger indoor unit or modification to the duct system may be necessary. We conclude that most installation-related problems can be avoided by standard test and balance of the air side residential AC systems. Maintenance issues, such as encouraging filter changes and coil cleaning may also be useful to improve field performance.

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1. Danny S. Parker, John R. Sherwin, Richard A. Raustad and Don B. Shirey, III are researchers at the Florida Solar Energy Center.

2. Based on the authors' observations, and considerable discussion with AC contractors, site installation frequently involves some compromise in the planned duct design-- often smaller than intended return grills.

3. Other methods, such as use of measured pressure drop across the fan or cooling cool were contemplated, but not used due to the need of manufacturer data which was not readily available. We attempted to use the anemometer method, where return grill cross sectional air velocities are measured, although accuracy and repeatability were unacceptable.

4. Until all moisture is evaporated from the coil, the after coil thermocouple will read lower than the return side, whereas after the moisture is removed, the supply will read slightly higher due to sensible fan heat. We found that it would often take 20 - 30 minutes to completely dry a wet coil at full fan flow.

5. The ACCA recommended procedure calculates the friction rate (FR) based on the fan available static pressure (ASP) and the duct system "total effective length" (TEL) which is based on the largest run length and its geometry:

$$FR = ASP(100)/ TEL$$

6. ACCA's Manual S clearly indicates that cooling equipment should not be oversized by more than 15%. In spite of this caution 38% of surveyed Florida contractors (Vieira, et.al., 1995) admitted intentionally over sizing equipment, frequently

at the customer's request.

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