Hidden Power Drains:
Residential Heating and Cooling Fan Power Demand

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ABSTRACT

This paper compiles power draw, airflow, and static pressure measurements of residential air handlers taken during nine separate field tests of space conditioning systems in Arizona, California, Florida, Nevada, and Canada. The field tests show that air handler power draws exceed the standard assumptions and that the interactions between airflow and cooling capacity combine to degrade overall system efficiency. The findings support conclusions from previous research in Canada that called for a systems approach to improving air handler efficiency. This study reports that fan power in U.S. air conditioners is about 40% higher than estimates used in the DOE Central AC and Heat Pump Test Procedure when rating air conditioners. Some fan power draws approach 1000 watts, similar to adding a 1000 watt electric resistance heater in the air stream. The low assumed power draw: masks the need for continued improvements in air handler and overall system performance; creates operating cost penalties for customers; and increases utility demand on peak. Application of more effective filters without attention to static pressure considerations would exacerbate these effects by raising air horsepower and power draw.

This paper consists of five sections: measured data, comparison of measured data to standard assumptions, air conditioner performance at low airflow, analysis of approaches to correcting low airflow, and conclusions.

Introduction

In the early 1990s Canadian researchers investigated the influence of residential air handling devices on furnace energy consumption and estimated the potential efficiency improvements available in these devices (CMHC 1992; CMHC 1993). One study (CMHC 1993) concluded that residential furnace air handler efficiencies in terms of air moving load external to the furnace were less than 10%. This poor performance was attributed to a number of causes, including fan inefficiency, motor inefficiency, and poor cabinet airflow design.

Another Canadian study (Phillips 1995) concluded that:

- air handler flow rates on furnaces have increased about 25% in recent years in spite of a general reduction in installed furnace size;
- the increase in flow produces higher efficiency furnaces (when efficiency is measured as heat output per heating fuel input) but duct systems have not been modified to allow for the higher flows;
- the resulting inadequacy of duct design causes an increase in external static pressure that adversely influences fan energy use, airflow, and total system performance;
- while permanent split capacitor motors have improved motor efficiencies, the fan power draw per unit airflow has remained almost constant.

Air Conditioner and Heat Pump efficiencies have also risen substantially in recent years (ARI 1999). The power draw of the compressor has been substantially reduced on these machines (ibid.), so that the air handler fan power draw has become a larger part of the total power draw.

For example, the air handler fan power draw on a three ton 1980 air conditioner with an EER95 of eight is around 525 Watts.[1] The total power draw for the 1980 unit, including the compressor and condenser fan is 4500 Watts. The air handler power draw is 12% of the total.

The air handler power for a three ton 2000 air conditioner with an EER95 of twelve is the same 525 Watts. The total power draw of the 2000 unit is 3000 Watts. The air handler power draw is 18% of the total.

At the same time, more effective air filtration is being added to air handlers. The proposed ASHRAE Standard 62.2P, Ventilation and Acceptable Indoor Air Quality in Low-Rise Residential Buildings (ASHRAE 1999), if adopted, will further the use of more effective filtration devices. These devices could increase external static pressure beyond current levels that already exceed Department of Energy AC and Heat Pump Test Procedure default values. Fan power draw will become a greater contributor to the unit total power draw, degrading air conditioner efficiency from the published values.

The electrical consumption and peak power demand of air handler fans become increasingly important as the penetration of air conditioning into the residential market grows. Market penetration increased by 79% in the Northeast US and by 45% in the Midwest US between 1984 and 1994 (ARI 1999).

Since the CMHC report on furnace air handler inefficiencies, nine additional field tests conducted in the U.S. and Canada have produced corroborating data. These tests extend the significance of the Canadian furnace observations to U.S. air conditioning systems. The field tests were individually sponsored by a variety of utility company and industry research organizations.

This paper consists of five sections: Measured Data, Comparison of Measured Data to Standard Assumptions, Air Conditioner Performance at Low Airflow, Analysis of Approaches to Correcting Low Airflow, and Conclusions.

Measured Indoor Fan Power Draw, External Static Pressures, and Airflow

The nine field tests reported in this paper were conducted from 1994 to 1998 under sponsorship of various utility companies and research organizations. These tests of residential systems include:

- a 1995 test of 40 air conditioner systems in new homes in Las Vegas;
- a 1996 study of 28 air conditioner systems in 22 new houses in Phoenix, Arizona;
- a 1997 study of 9 air conditioner systems in existing residences in central Florida;

[1] EER 95 is the total capacity of the unit in Btu/hr divided by the total power draw of the unit in watts at 95 degrees F outdoor temperature, 80 degrees F dry bulb return temperatures, 67 degrees F wet bulb return temperature. These figures are based on typical measured fan power draws and airflows of 500 watts per 1000 CFM and 350 CFM per ton respectively.

- a 1995 field test of 40 air conditioner systems in existing housing in the Cochella Valley of California;
- a 1996 test of 37 air conditioner systems in new houses in Las Vegas;
- a 1998 study of 5 new evaporative cooled air conditioners installed on existing furnaces and duct systems in houses located in various areas of California;
- a 1998 study of 15 air conditioner systems installed in newly constructed townhouses across New Jersey;
- a 1995 study of 32 near new furnace systems in houses across Canada; and
- a 1995 study of 39 pre-1990 furnace systems in houses across Canada.

The results are replications from three independent organizations in a wide variety of areas. The performance characteristics of these systems are presented in Table 1.

Table 1. Measured Air Handling Equipment Performance Data for North American Installations

Reference	Study Location and Equipment Age	Number of Units in Sample	Average Capacity (tons)	Average Inside Fan Watts	Average CFM	Average Watts per 1000 CFM	Average External Static (IWC) [2]
Air Conditioners							
Blasnik et al. 1995a	Las Vegas, new	40	3.4		1150		.41
Blasnik et al. 1996	Phoenix, new	28	3.6	620	1220	510	.48
Parker 1997	Florida, existing	9	2.5	420	850	490	.55
Proctor et al. 1995	Cochella Valley CA, existing	40	4.0		1240		.53
Proctor et al. 1996a	Las Vegas, new	37	3.5		1320		.50
[3] Proctor and Downey 1998	California, replacement	5	3.4	760	1320	570	
Proctor et al. (unpublished)	New Jersey, new townhouses	15	2.7		1050		.45
Non-AC							
Phillips 1995	Canada post-1990 heating speed	32		510	1120	450	.52
Phillips 1995	Canada pre-1990 heating speed	39		370	860	440	.38

[2] Table 1 external static pressures are for the duct system, registers, and typical filters. The pressure drop given does not include inside coil pressure drop of 0.2 to 0.3 IWC (50 to 75 pascals).

[3] The replacement air conditioners in this study were downsized an average of 20%.

External Static Pressure and Fan Motor Power Draw: Standard Assumptions vs. Field Data

The standard assumption for external static pressure, according to DOE test standards, ranges from 0.1 inches of water column (IWC) for 2-ton residential units to 0.2 IWC for units larger than 3.5 tons. As shown in Table 2, the external static pressure values measured in field tests representing both new and existing construction, are two to four times higher than DOE assumptions. The values for the field-tested units ranged from 0.41 IWC to 0.55 IWC. This is at least twice the value assumed for larger (3.5+ ton) units.

Table 2. Comparison of Static Pressure and Fan Motor Power Test Assumptions with Field Data for Air Conditioners

	External Static Pressure	Fan Motor Power Demand
Standard Assumption	0.1 to 0.2 (IWC)	365 (W per 1000 CFM)
New Construction Single Family Air Conditioner	0.41 to 0.50 (IWC)	**510 (W per 1000 CFM)**
Existing Construction Single Family Air Conditioner	0.53 to 0.55 (IWC)	492 to 574 (W per 1000 CFM)

High static pressures produce reduced airflows and the need for higher horsepower fan motors to approach proper flow. Indoor fan motor power demand is a result of external static pressure, flow, fan efficiency, motor efficiency, as well as cabinet and heat exchanger design. The standard DOE assumption for indoor fan energy consumption is 365 watts per 1000 CFM. As presented in Table 2, fan motor power draw under operating conditions averages 510 watts per 1000 CFM, 40% higher than the assumed value. For a five-ton air conditioner achieving 2000 CFM of airflow, this is equivalent to a one kilowatt electric resistance heater in the air stream.

Air Conditioner Performance at High Static Pressures, Low Airflow, and High Fan Power Demand

Air conditioner performance suffers from the effects of both high fan power demand and low airflow.

Effect of High Fan Power Demand

The discrepancy between assumed and actual fan power draw has deleterious effects. The smallest effect is that the total capacity of the air conditioner is diminished from the specification sheet value by approximately 1.3% due to additional fan heat. Second, the total unit power draw is increased by approximately 4 to 5%.[4] The result from increased fan power alone is an overall efficiency drop of approximately 5 to 6%.

Effect of Low Airflow

Air conditioners are generally designed to have an airflow rate of about 400 CFM per ton across the inside coil. For 3.5- to 4-ton units, the airflow rate should range from 1400 to 1600 CFM. As documented in Table 1, the units tested in these studies did not achieve the design airflow rate even in new construction and even in new townhomes with minimal ductwork.

Low airflow across the inside coil has adverse effects on unit performance. It lowers evaporator temperatures, reduces total capacity, increases latent capacity, and lowers sensible capacity. These effects have been measured in laboratory situations (Parker et al. 1997; Proctor et al. 1996b; Rodriguez et al. 1995). Full sets of data were available to the authors for the Parker and Proctor tests.

Tables 3 and 4 illustrate the effect of low airflow on gross total efficiency, where fan heat and power draw are excluded. These tables also show the effect of low airflow on net sensible efficiency, where only sensible cooling is considered and fan heat and power draw are taken into account.[5] The sensible EER is of high importance in many areas because the thermostat responds only to sensible cooling.[6]

Tables 3 and 4 were built using the assumption that the fan power draw has a constant relationship to the airflow (500 watts per 1000 CFM). This assumption is equivalent to saying that the static pressure does not increase with increased airflow, that is, the duct and air handler system restriction is reduced to allow for higher flow.

Table 3. Air Conditioner Performance with Degraded Airflow Test Unit #1

Proctor (Proctor et al. 1996b)[7]			
Test Case	Airflow (CFM/ton)	Gross Total EER	Net Sensible EER
Avg. 3 Tests	282	11.1	8.0
Avg. 3 Tests	402	12.1	8.8

[4] The effect increases as the efficiency increases. The effect is four percent for an EER 10 unit and five percent for an EER 12 unit.

[5] Laboratory tests are often run without an air handler fan (test equipment provides the airflow across the indoor coil). This analysis uses 500 watts per 1000 CFM as the assumed fan power draw. This is consistent with the field data presented in Table 1.

[6] There are additional complex interactions including the shift of latent capacity to sensible capacity when the evaporator coil entering air is dryer than standard test conditions.

[7] All tests at 90ºF condenser entering temperature, 75ºF dry bulb/59ºF dry bulb. The three test were run with correct charge, 30% overcharge, and 30% undercharge.

Table 4. Air Conditioner Performance with Degraded Airflow Test Unit #2

Parker (Parker et. al. 1997)			
Test Case	Airflow (CFM/ton)	Gross Total EER	Net Sensible EER
Test 1	124	5.8	3.4
Test 2	190	6.9	4.2
Test 3	212	7.1	4.4
Test 4	247	7.5	4.7
Test 5	350	8.1	5.5
Test 6	414	8.4	5.9

Approaches to Correcting Low Airflow

In an actual installation, airflow could be corrected by reducing the duct restriction to obtain higher flows with the same static pressure. On the other hand, the airflow could be corrected by increasing fan size or speed without improving the duct system. In the later case, the watt draw of the fan will increase approximately as the cube of the airflow.

When the airflow is increased and duct sizing is increased sufficiently to maintain the same external static pressure, the net sensible efficiency is also increased.

However if the same duct system is maintained on the air conditioner, the net efficiencies (both total and sensible) drop in cases where the initial airflow is greater than the base test. This is due to fan power draws that increase approximately as the cube of the airflow (for a constant restriction to airflow). This is illustrated in Tables 5 and 6.

Other options to improving airflow without increasing power draw include improving the efficiency of the fan/motor assembly and improving the aerodynamic conditions entering and leaving the fan.

Table 5 Explanation

Column A is the airflow in the test case.

Column B is the Net Sensible EER (Sensible BTU per Watt-hour) when the duct system is increased in size to accomplish the higher airflow and maintain the same static pressure. This figure includes the effect of air handler fan heat and watt draw. When static pressures are held constant, the Net Sensible EER increases as the airflow is increased. The second statistic in this column is the percent change in EER for a 10% increase in airflow. The basis for this statistic is the test with the next lower airflow.

Column C is the Net Sensible EER when the duct system is not changed. Under these conditions the net efficiency drops in cases where the initial airflow is greater than the base test. The second statistic in this column is the percent change in EER for a 10% increase in airflow. The basis for this statistic is the test with the next lower airflow.

Table 5. Air Conditioner Performance with Increased Airflow Test Unit #1

Column	A	B	C
Test Case	Airflow (CFM/ton)	Net Sensible EER w/larger ducts (% per 10% increase in flow)	Net Sensible EER w/same ducts (% per 10% increase in flow)
Proctor (Proctor et al. 1996b)			
Avg. 3 Tests Base[8] of Calculation	282	7.99	7.99
Avg. 3 Tests	402	8.82 (2.4%)	6.80 **(–3.5%)**
Parker (Parker et. al. 1997)			
Test 1	124	3.41	3.65
Test 2	190	4.21 (4.4%)	4.43 (3.9%)
Test 3	212	4.40 (3.9%)	4.56 (2.5%)
Test 4 Base[9] for Calculation	247	4.73 (4.6%)	4.73 (2.4%)
Test 5	350	5.47 (3.7%)	4.55 **(-0.9%)**
Test 6	414	5.92 (4.5%)	4.13 **(-5.2%)**

Conclusions

Reporting on their studies of Canadian furnace performance, CMHC researchers in 1992 observed that a systems approach was needed for optimizing the performance of space conditioning equipment. They concluded their study by issuing a challenge to the industry: “Nobody is looking at the big picture: how can we match the furnace heat exchanger, blower compartment, motor, blower, and controls so as to achieve optimum space heating and ventilation?” (CMHC 1992, 46)

The nine field tests of both U.S. air conditioning and Canadian heating systems reported here:

- substantiate the severity of air handler inefficiencies;
- substantiate the severity of duct system design problems;
- emphasize the need for continued improvement of air handler devices within the framework of the entire HVAC system; and
- quantify the inaccuracies in standard assumptions used when rating residential air conditioners and estimating the demand impacts.

[8] Base test for initial fan power draw calculation (500 watts per 1000 CFM).
[9] Base test for initial fan power draw calculation (500 watts per 1000 CFM).

Air Handler Inefficiencies

Air movement through space conditioning systems is compromised by the non-aerodynamic intake and exit conditions common in cabinet and heat exchanger designs. Improvements are also available in fan and motor efficiencies, but the system as a whole needs to be addressed. The unrealistically low external static pressure assumption promotes use of indoor fan/motor/cabinet designs that often cannot provide the static pressure needed to produce the proper airflow in standard residential installations.

Duct System Design Problems

Duct distribution systems should be more adequately sized. Most residential duct systems are being undersized using duct slide rules with an arbitrary 0.1 IWC/100 ft. input for duct selection without regard for available static pressure, actual duct length, or fittings.

Standard Assumptions

The DOE test specifications used in calculating air handler performance efficiencies misrepresent the actual conditions under which the equipment operates. Utilities and manufacturers alike look to equipment SEER and HSPF ratings when planning residential marketing, peak demand reduction, and energy conservation programs, and customers use these ratings when comparing equipment options. The discrepancy between assumed and actual air handler performance is creating unrealistically high air conditioning and heating systems efficiency ratings which mask the need to improve the efficiency of the air handling equipment.

The unrealistically low external static pressure assumption promotes use of indoor fan/motor/cabinet designs that often cannot provide the static pressure needed to produce the proper airflow.

In negotiating the ISO test procedure, these discrepancies should be addressed.

Design Improvements

Manufacturers aiming to remedy these problems may find benefit through reconfiguring HVAC cabinets and heat exchangers which control the entrance and discharge conditions of the fan. Performance of fans and motors can also be addressed. While a piecemeal approach holds promise for some improvements, the greatest gains would come through the structurally more difficult whole system approach that includes all these items plus the duct system and the refrigerant circuit.

Design improvements should consider the collective impact of each component’s performance on the whole system. Absent that consideration, air handling devices will not necessarily have the ability to keep up with other system improvements, such as high-efficiency filters now entering the market.

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