LBNL 53606

ERNEST ORLANDO LAWRENCE
BERKELEY NATIONAL LABORATORY

Improving Air Handler Efficiency in Residential HVAC Applications

I.S. Walker, M.D. Mingee and D.E. Brenner

Environmental Energy Technologies Division

August 2003

This work was supported by the Assistant Secretary for Energy Efficiency and Renewable Energy, Building Technologies Program, of the U.S. Department of Energy under contract No. DE-AC03-76SF00098.

Disclaimer

IMPROVING AIR HANDLER EFFICIENCY IN RESIDENTIAL HVAC APPLICATIONS

I.S. Walker, M.D. Mingee and D.E. Brenner

Energy Performance of Buildings Group
Indoor Environment Department
Lawrence Berkeley National Laboratory
Berkeley, CA. 94720, USA

ABSTRACT

In continuing the development of energy efficiency standards, consideration has turned to air handlers used for heating and air conditioning of consumer residences. These air handlers have typical efficiencies of about 10% to 15% due to poor electric motor performance and aerodynamically poor fans and fan housings. This study was undertaken to examine some of these performance issues, under carefully controlled laboratory conditions, to support potential regulatory changes. In addition, this study examined the performance of a prototype air handler fan assembly that offers the potential for substantial increases in performance. This prototype and a standard production fan were tested in a full-scale duct system and test chamber at LBNL which was specifically designed for testing heating, ventilation, and air conditioning systems. The laboratory tests compared efficiency, total airflow, sensitivity to duct system flow resistance, and the effects of installation in a smaller cabinet. The test results showed that, averaged over a wide range of operating conditions, the prototype air handler had about twice the efficiency of the standard air handler and was less sensitive to duct system flow resistance changes. The performance of both air handlers was significantly reduced by reducing the space between the air handler and the cabinet it was installed in. Therefore any fan rating needs to be performed using the actual cabinet it will be used in.

INTRODUCTION

Increases in the efficiency of forced air heating and cooling equipment have provided considerable reductions in energy use. For example, furnaces have little gains remaining to be realized because high efficiency furnaces already have efficiencies (AFUE ratings) greater than 90%. However, the fans that are used to move the air in forced air systems have not seen significant efficiency improvements and have efficiencies in the 10%-15% range (Phillips 1998). These low efficiencies indicate that there is significant room for improvement of these air handler fans. Part of the reason why there haven't been significant fan efficiency improvements in the past is that air handler fan energy use was not specifically included in Federal ratings. For example, SEER allows the use of a default fan power consumption and furnaces are only rated with AFUE: a measure of gas use efficiency. As furnaces, air conditioners and heat pumps reach their efficiency limits, the air handler fan remains as a device that has room for improvement. In particular, as air conditioners have become more efficient, the fraction of total energy consumption for the HVAC system attributed to the air handler fan has increased, thus making the air handler fan a greater contributor to the overall system energy use. This trend has increased the need to have the air handler fan energy use included in ratings or standards. Before changes are made to ratings and standards it is important to fully understand the performance issues of the devices being regulated. This study was undertaken to examine under carefully controlled laboratory conditions some of these performance issues to support potential regulatory changes. In addition, this study examined the performance of a prototype air handler fan assembly that offers the potential for substantial increases in performance.

A study by CMHC (1993) has shown that residential air handlers are almost an order of magnitude less efficient than large commercial air handlers. The CMHC report indicated the potential air handler efficiency to be as high as 70%, thus using only one fifth of the energy of a typical system. Typical furnace fan efficiencies are on the order of 15%, but poor cabinet and duct design can reduce the efficiency to about 7%. For comparison, individual exhaust fans typically used in bathroom and kitchen vents are even worse at about 2% efficiency. The spread from best to worst systems was on the order of ten to one – so clearly it is possible to have much better systems. Too quote this report directly: "*The potential for energy efficiency improvements of small air handlers is clearly vast*". The cost premiums of replacing low efficiency motors with high efficiency motors were estimated to be about CAN$20 to CAN$100 (in 1993). The cost of other improvements, such as fan blade and cabinet design are hard to quantify, but a reasonable estimate would be on the order of $200 or less. As with most mass produced items, it is expected that this cost would decline as more efficient air handlers become more common and are produced in greater quantities.

Field studies by LBNL, Proctor and Parker (2000) (245 systems) and Phillips (1998) (71 systems) have shown that existing fans in residential air handlers typically consume 500W or more of electricity and supply about 2 cfm/W. This is substantially lower than the 2.8 cfm/W default used in the air conditioner rating test procedure (DOE (1996)). A recent field survey Pigg (2003) of 31 new furnaces in Wisconsin found that "*An ECM air handler substantially reduces electricity consumption; the average ECM furnace in our study used about 0.5 kWh of electricity per therm of gas consumed, which is about half what we measured for the non-ECM furnaces. That translates into about 400 kWh less electricity over the course of an average heating season in Wisconsin.*" Savings were less than indicated in GAMA (GAMA 2003) ratings – mostly due to GAMA assumptions about the fraction of time staged furnaces operated in the low-fire range. This level of energy use corresponds to about 5% of the electricity used in a house. Note that most appliances that have similar or even less significant contributions to total household energy use are regulated by minimum federal efficiency standards. These issues of air handler efficiency become even more important for ventilation systems that utilize the air handler fan and run the air handler for extended hours beyond that needed solely for heating and cooling. The increased operation time therefore leads to greater energy use.

Another issue to consider is the potential of retrofitting more efficient air handlers into existing heating and cooling systems. In addition to providing input to appliance standards, the ability to have standards for the air handler fans separate from those for the heating or cooling equipment they are installed with could yield important energy savings in the retrofit market.

Using a combination of field observations and engineering judgment we can assemble a list of the problems that lead to low air handler efficiency and potential solutions to these problems, as shown in Table 1. None of the problems require exotic or complex solutions and there are no technological barriers to adopting them. Some of the solutions are simple equipment swaps (using ECM motors), others require changes to the way the components are built (tighter tolerances) and others relate to HVAC equipment design (not putting large fans in small cabinets). In this study we examined how much performance improvement can be gained by addressing these problems. As a baseline a standard furnace air handler was tested and its test results were compared to a prototype air handler that incorporated many of the above solutions.

Table 1. Issues for improving air handler performance	
Problem	Solution
Clearances between fan blades and housing (or scroll) are too large as a result of large tolerances in production. Turbulent air flows around blade edges rather than moving into the ducts.	Tolerances should be reduced to less than one eighth of an inch (3 mm).
The blades are fabricated from plain sheet metal.	Use more aerodynamic blades.
Forward curved blades are generally less efficient than backward curved blades (although they have advantages in terms of relatively constant flow over a wide pressure range).	Backward curved blades could be used combined with the control capabilities of an electrically commutated motor (ECM) to maintain flow rates.
Blower inlets have sharp edges, which disrupts airflow into the fan.	Smooth, large radius inlet cones create less noise and a better flow pattern entering the fan.
Electric motors are usually low efficiency.	Use higher efficiency Electrically Commutated Motors (ECM).
Fans are fitted in restrictive furnace cabinets.	Integrate design of furnace cabinets and fan housings to ensure sufficient clearance around fan inlets.

TESTED FANS

Two fans were tested in this study. The standard fan, installed by the manufacturer in the furnace we used in the test apparatus, and a prototype fan, recently developed by a fan manufacturer in collaboration with the US Department of Energy (DOE). The two fans are illustrated in Figure 1.c

The standard fan in a residential forced-air heating and cooling system has a permanent split capacitor (PSC) type motor. PSC motors can be optimized for efficiency and power factor at a rated load. They are considered to be the most reliable of single-phase motors. In residential furnaces, PSC motors usually have between two and four fixed speeds. Different speeds are necessary to match the different airflow requirements for heating and cooling operation (typically cooling air flows are about 25% greater than heating air flows). Due to the way the speed is controlled in a PSC motor, a fan operating at a fractional speed consumes approximately the same power as one operating at full speed, with an accompanying decrease in efficiency. Speed is controlled by jumpers on the control board located on the fan housing. In the standard air handler used in this study, a PSC motor was used to drive a centrifugal fan with forward curved blades and large (1 inch (25mm)) clearances between the fan and its housing.

The prototype air handler used an electronically commutated motor (ECM). Since the speed of an ECM is electronically controlled, it can be set specifically to match the airflow requirements for each application. Furthermore, ECM's are designed to speed up or slow down (maintaining torque) in an attempt to preserve airflow regardless of the static pressure across the fan, e.g. when filters become dirty and restrict airflow. This helps maintain an airflow range through the heat exchanger, close to the optimal flow rate for which they were designed. Also, ECM's use power proportional to their airflow requirements, thereby making them inherently more efficient than PSC motors over a wide range of air flows, particularly at significantly lower airflows. This is particularly important when using the fan in a HVAC system for ventilation, a case where volumetric flows are typically less than a quarter of heating or cooling air flow rates. Another significant feature of the prototype blower is the backward facing aerodynamically shaped blades on the centrifugal fan wheel, where the impeller blades point backward relative to the wheel motion. Unlike forward curved designs, these blades build pressure by compression more than by inertia. Thus, less energy is lost converting dynamic pressure into static pressure when the air moves into the scroll housing and on to the ducts. Lastly, this prototype has significantly tighter tolerances than the standard production fan. The inlet cones end much closer to the fan blades; around 1/8 inch (3mm) compared to 1 inch (25mm) in current production designs. Again, less energy is lost to turbulent recirculation around the blade edges.

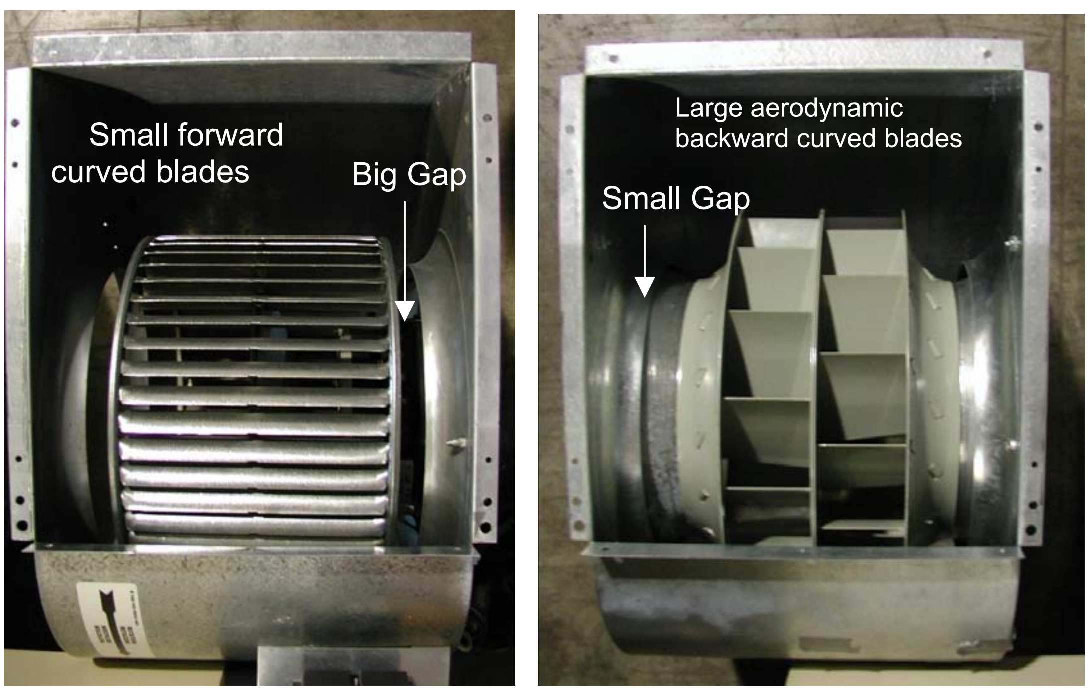

Figure 1. Illustration of different blade design and fan to housing clearances for the standard fan (left) and prototype fan (right).

TEST APPARATUS AND PROCEDURE

There are existing test methods for evaluation the performance of air handler fans (AMCA210/ASHRAE 51 - 1999) that use a test apparatus that uses large plenums and does not include and supply or return ducting and their associated flow restriction. We wanted to use a real duct system and have the fan operate inside an air handler cabinet to make the testing more realistic. This is important because a real system has a higher pressure drop across the fan, and the cabinet restricts the air inlet to the fan. All the tests were conducted using the Energy Performance of Buildings Group full-scale duct system and test chamber (Figure 2). The test chamber (which represents a house) is a 32' long, 8' wide, 8' tall box (9.2 m x 2.4 m x 2.4 m) over a 4' (1.2 m) "crawl-space". The chamber is constructed using standard wood framing materials (two-by-fours and plywood), with all joints taped and sealed to reduce air leakage (chamber background leakage is about 60 cfm25 [100 L/s at 25 Pa]). No windows are installed but there is one door for access to the interior of the house. The test chamber is located inside a warehouse to eliminate the effects of weather on measurements and allow for more controlled experimental conditions. Several deliberate holes are cut into the side of the house and equipped with air tight covers to provide for controlled leakage. For these tests a single 11 inch diameter opening was used in the test chamber envelope. There are ten supply registers, two each of five different styles, and a single return register. The opening and closing of these registers was used to control the system flow resistance and therefore the air flow rates through the system. Figure 3 is a floor plan of the test chamber illustrating the register locations.

Figure 2. Exterior view of test chamber inside warehouse.

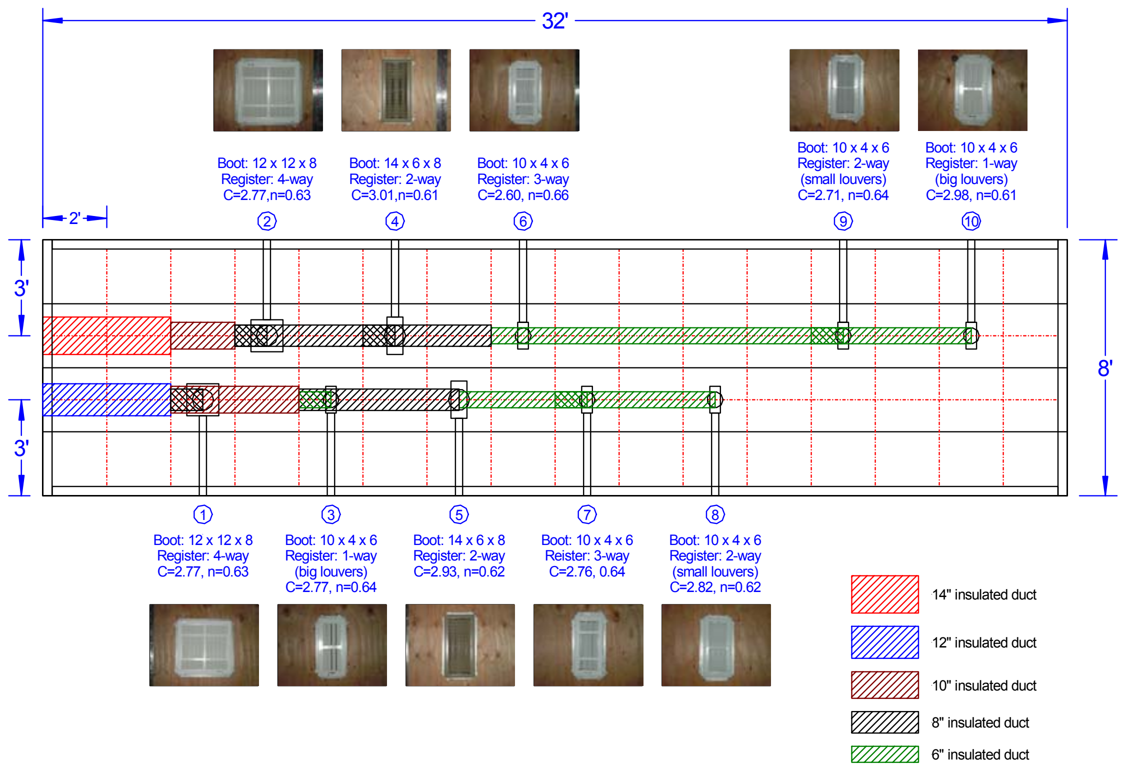

Figure 3. Floor plan of test chamber showing duct system layout, register location and duct leakage location.

The duct system is comprised of the following components (in order of air flow from the chamber): return register, large flow nozzle, return plenum, fan box, furnace, supply plenum, two main ducts, and ten supply registers. The two main supply ducts connected to the supply plenum have diameters of 12 inches and 14 inches (305 mm and 356 mm). These two main ducts branch into ten individual ducts, each connected to a supply register. The supply ducts are made of flexible insulated duct and are mounted on hangers in the crawlspace. In order to maintain a tight system, the duct system has mastic at all connections, the register boots are screwed and taped into the floor, and the air handler cabinet had all openings sealed with tape. These leakage reduction efforts resulted in background leakage of about 5 cfm25 (8 L/s at 25 Pa). It is important to have as little leakage as possible so that we can be sure that the measured air flows and pressures are accurate. A leaky system may allow air to enter or leave the system between the measurement point and the air handler, thus leading to inaccurate estimates of air handler performance. This system normally operates with a system air flow of 1125 CFM at a static pressure of 0.5” wc (0.554 m^3/s at 125 Pa) with the standard air handler and all the registers open.

The duct system had deliberately adjustable leakage. The return plenum was modified to allow the opening of a monitored leak (using an orifice air flow meter) as well as the return duct connection to the test chamber. Similarly, a monitored leak (using a nozzle flow meter in this case) was attached to the supply plenum in addition to the two major supply ducts that supply air to the test chamber. Each supply register boot also has an openable leakage site whose air flow can be measured. These leaks were all sealed for the air handler tests in this study, but were used for other related studies using this same test apparatus.

Total system airflow was measured using a high precision flow nozzle (±0.5% of measured flow) located in the return duct upstream of the return plenum (as illustrated in Figure 4). Fan inlet and exit pressures (±1.5%) were measured upstream and downstream of the fan. The locations for these pressure measurements were carefully chosen after experimenting with several pressure probe placements in order to avoid unstable or extreme results caused by non-uniform flows exiting the air handler. The downstream measurements were taken between the fan outlet and the furnace heat exchanger to eliminate the pressure drop across the heat exchanger that would have been included if supply plenum pressures were used. This makes for a more accurate estimate of the air handler performance, however, it should be noted that the external static pressure quoted in furnace manufacturers' literature is based on supply plenum measurements. Fan electrical power use was measured with a true power meter (±1%), which accounts for the fan power factor. Data were recorded using two automated data acquisition systems, with the exception of the fan speed which was obtained from the fan controller computer interface on a laptop. All the data were recorded using five second time averages after waiting for readings to stabilize (about two to three minutes).

Different test operating points were obtained by systematically closing supply registers to increase the flow resistance of the system. After each register was closed the system was allowed to operate for several minutes to reach a steady operating point. The standard air handler was operated at a single speed. The prototype air handler was operated using a range of six torque settings. This allowed us to evaluate the performance of the prototype at typical furnace air flow rates as well as at reduced rates more suitable for ventilation air flows. The test results for the different settings were numbered from 1 to 6, with 1 being the lowest torque and 6 the highest setting.

The scroll housings of the fans had identical dimensions and mounting flanges. These dimensional and mounting similarities were chosen deliberately because a key application for the prototype air handler is in retrofit applications where it will have to fit in the same space as the fan being removed and connected to the same furnace flanges. The only difference between the two housings was an added pair of inlet cones on the prototype.

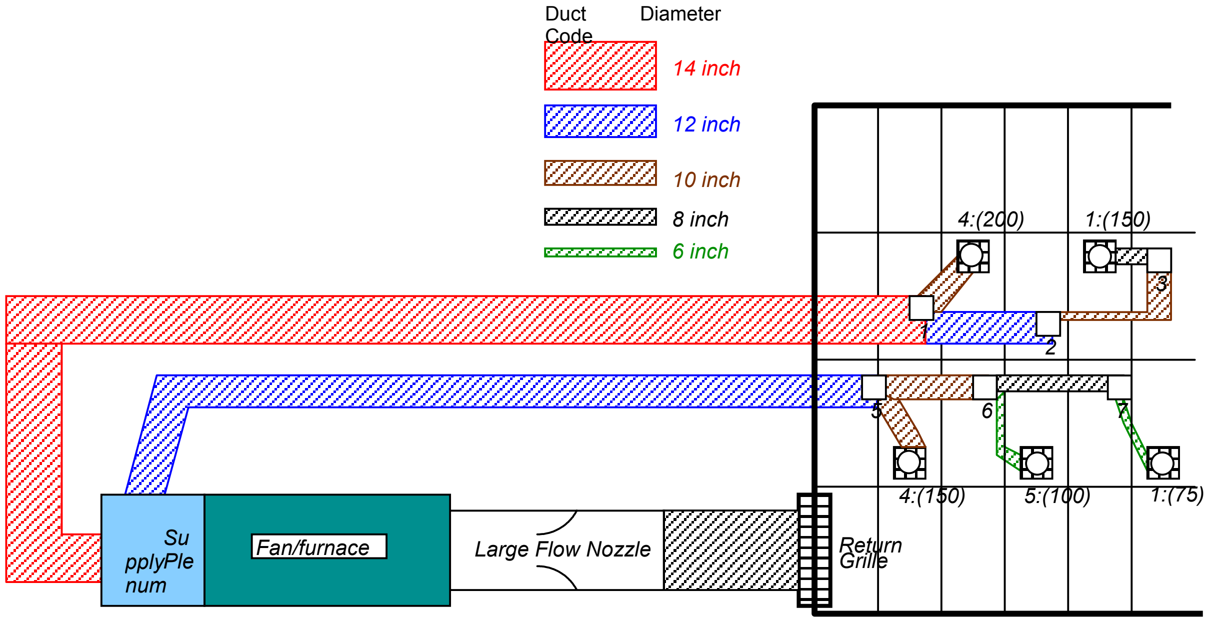

Figure 4 Supply and return duct and air flow meter layout.

The experiments were repeated with blockage installed on the walls of the air handler cabinet to simulate installation in a smaller cabinet. Because HVAC systems are installed in tight spaces, such as attics and closets, manufacturers have made smaller furnace and matching air handler fan cabinets. However, the size of the air handler fans and scroll housings are often unchanged. This results in small clearance around the fan housing and inlet that can significantly reduce the flow through the fan. Although this has been observed in field installations, the field tests were not able to directly determine the effect of this restriction separate from other system effects such as restrictive duct systems. Therefore we performed additional experiments with reduced clearance for the fan inlets for both the standard and prototype motors. The normal clearance was 2 inches (50 mm) between the fan inlets and the cabinet - this clearance was reduced to 1 inch (25 mm) by adding 1 inch (25 mm) thick blocking to the sides of the air handler cabinet. This is illustrated in Figures 5 and 6.

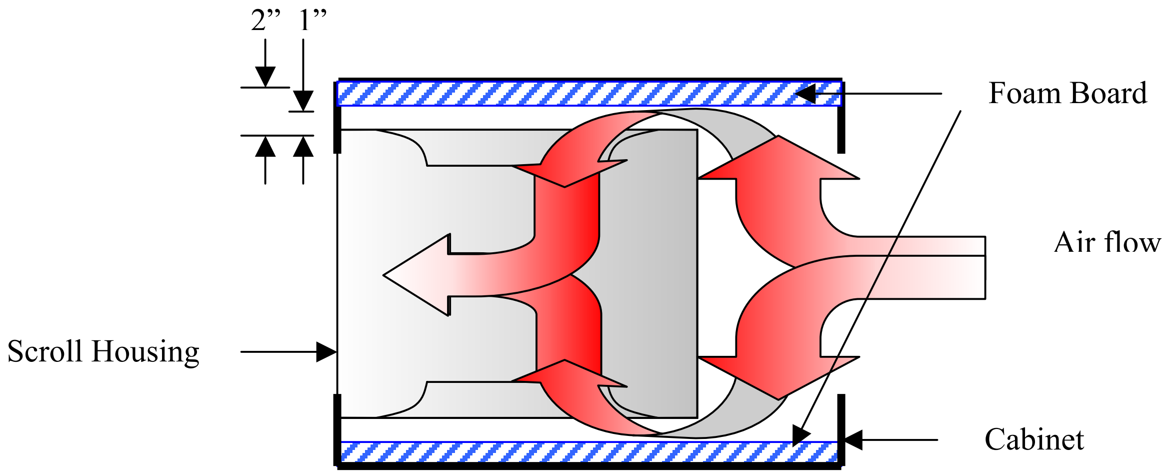

Figure 5. Illustration of foam board added to restrict fan inlet clearance.

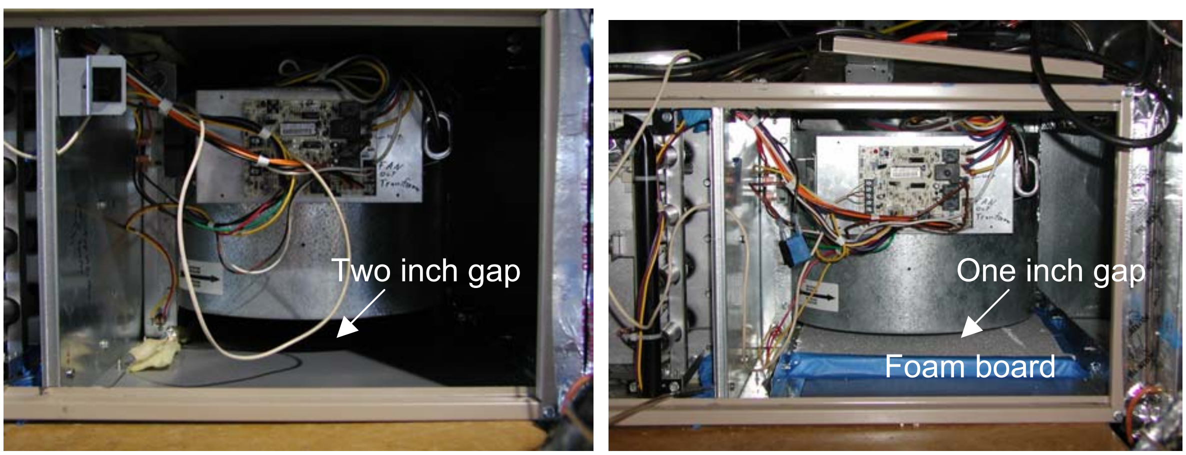

Figure 6. Standard fan in cabinet with no restriction (left) and with added foam board restriction (right).

RESULTS

All the test data are provided in Appendix 1. The following sections discuss the key results from these data. Figure 7 shows how the airflow increased with increasing torque setting and decreased as the system flow resistance was increased. There is still some flow, even with all the register dampers closed because these dampers do not have airtight seals. The constant torque control tends to keep the air flow relatively constant as the flow restriction increases, particularly at the low torque/low air flow settings. This graphically illustrates one of the major advantages of the ECM motor and its controller’s ability to maintain a pre-set flow no matter how the system changes- either due to damper settings for zone control, or with progressive fouling of coils and filters. With the increase in flow resistance as more registers are closed, the pressure difference across the fan increases, as shown in Figure 8. The electric power consumption remains relatively constant for each torque setting no matter how many registers are closed, as shown in Figure 9. The fan efficiency was calculated by first determining the power transferred to the air flowing through the fan: i.e., the product of the volumetric air flow and the pressure difference. This air power was then divided by the electric power consumption. The resulting efficiencies are shown in Figure 10. These results indicate that the prototype fan is most efficient at the higher torque settings and air flow rates. These efficiencies are generally much higher than those reported earlier for standard fans.

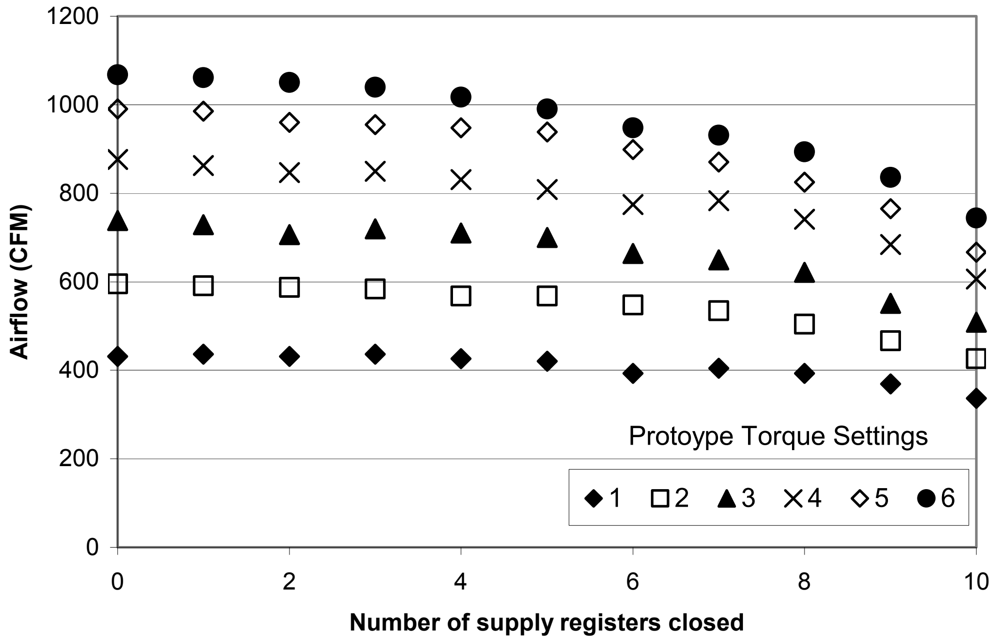

Figure 7. Airflow changes due to torque setting and system flow resistance (number of closed registers)

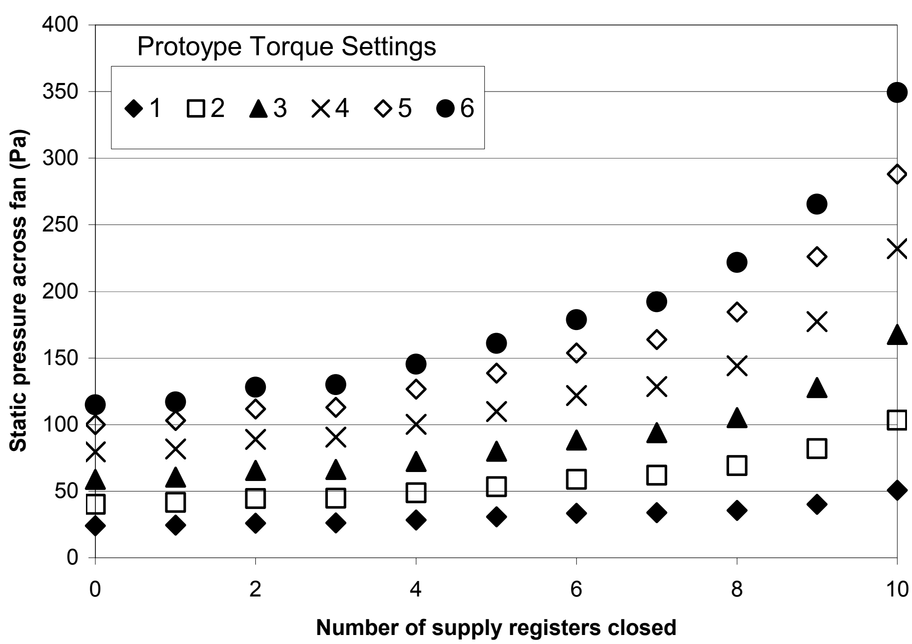

Figure 8. Static Pressure changes due to system flow resistance (number of closed registers)

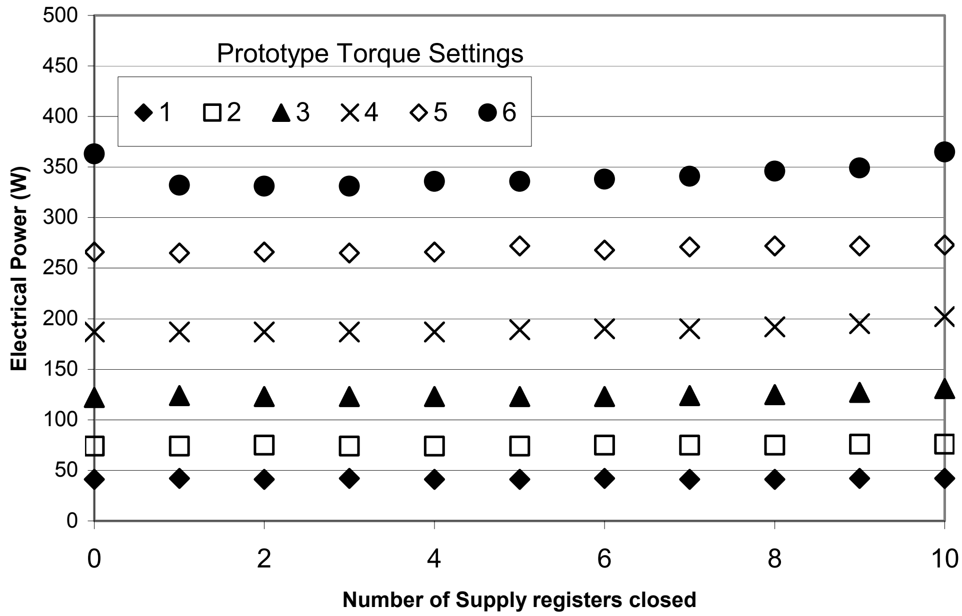

Figure 9. Electrical Power consumption of prototype fan at different torque settings and system flow resistance (number of closed registers)

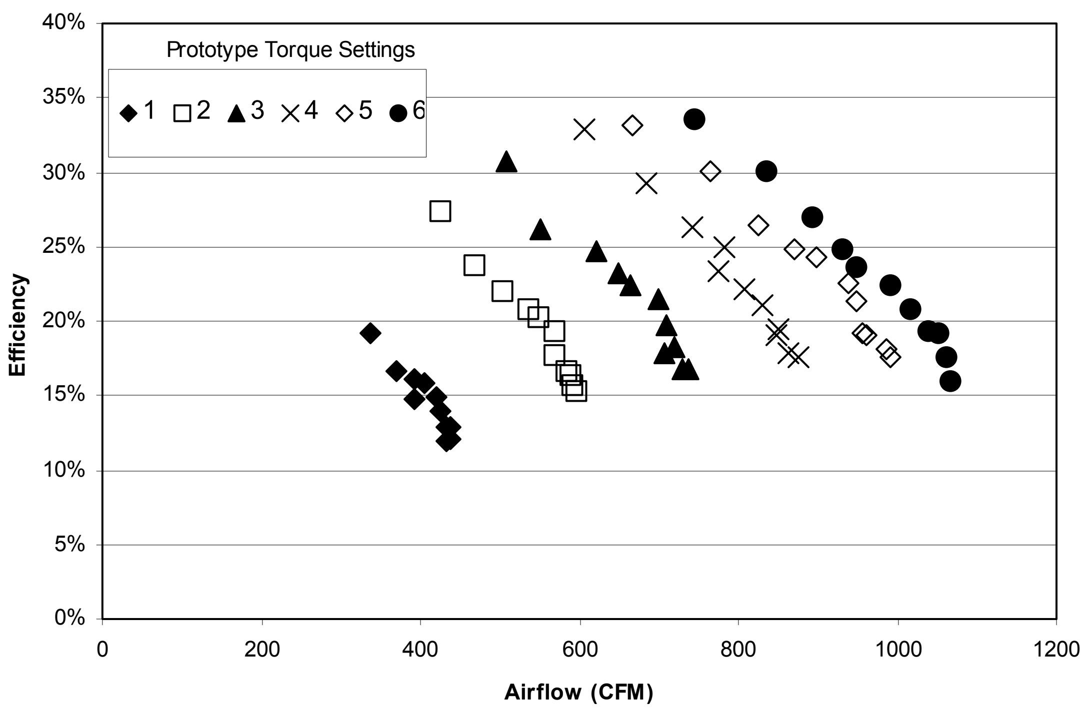

Figure 10. Efficiency and airflow as system flow resistance is increased

There were no multiple experiments for different operating modes for the standard fan. Instead we will compare the standard fan performance to the prototype with the prototype operating at torque setting 6. This highest torque setting was chosen because it came closest to the maximum air flow achieved by the standard fan. Figure 11 compares the fan performance curves for the standard and prototype fans. This shows that the prototype has a much flatter curve and its air flow changes less as the pressure difference across the fan changes. This is a good attribute because it allows the prototype fan to better maintain the air flow as a system (and its filters) becomes dirty with age (increasing its flow resistance), or if it is poorly installed. Poor installation of ducts with added bends, sharp changes in direction and high flow resistance fittings leads to a duct system with higher flow resistance and therefore greater pressure difference.

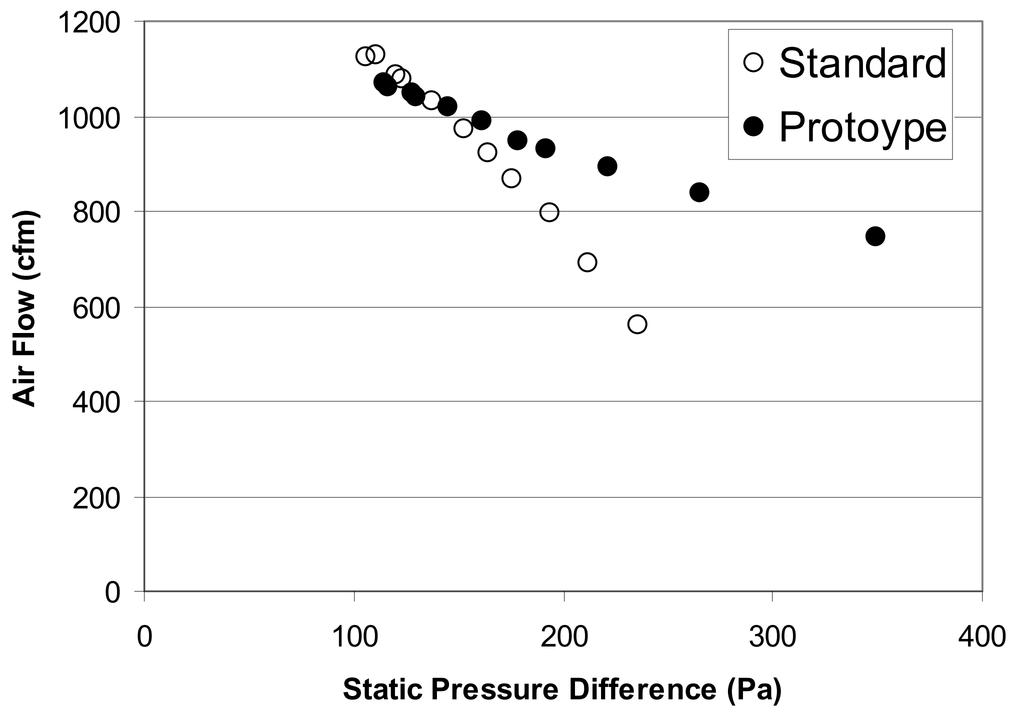

Figure 11. Comparison of standard and prototype fan performance curves.

Electricity and Energy Consumption

The amount of electricity required to provide a certain volume of air flow is an important factor. We can see in Figure 12 that the prototype fan requires only about half the power of the standard fan. Because the number of closed registers and corresponding air flow rates never exactly match for the two fans we need to look at the data in a couple of different ways (high flow rates, and same flow rate and amount of system resistance).

First we simply match high flow rates. At 1050 cfm (496 L/s) the prototype moved 2.9 cfm/W compared to only 1.9 cfm/W for the standard fan. Similarly, the fan efficiencies are much higher for the prototype fan than for the standard fan: averaged over all the tests shown in Figure 10, the prototype efficiency was 23.1% and the standard fan efficiency was 12.5%. Repeating these calculations at a lower torque setting (setting 5) and a slightly lower air flow rates (950 cfm (448 L/s)), the prototype performed even better, and moved 3.7 cfm/W compared to 1.9 cfm/W for the standard fan. These results imply that the operating torque of the prototype fan can be optimized for a particular flow rate and duct system. In this case torque setting 5 gave superior results to torque setting 6.

Although the results discussed above are at the same air flow rate, they are at slightly different system flow resistances, i.e., number of closed registers. The second way to compare the results is to find where the air flow rates and number of closed registers are the same for both fans. For the torque setting 6 case, the match occurs with four closed registers at a flow rate of 1020 cfm (481 L/s). The prototype fan moves 3.0 cfm/W compared to 1.9 cfm/W for the standard fan, and the corresponding efficiencies are 21% and 12% respectively. For the torque setting 5 case, the match occurs with seven closed registers at a flow rate of 870 cfm (481 L/s). The prototype fan moves 3.2 cfm/W compared to 1.7 cfm/W for the standard fan, and the corresponding efficiencies are 25% and 14% respectively. These results are consistent with the first comparison: i.e., the prototype fan is about twice as good as the standard fan both in terms of efficiency and cfm/W rating.

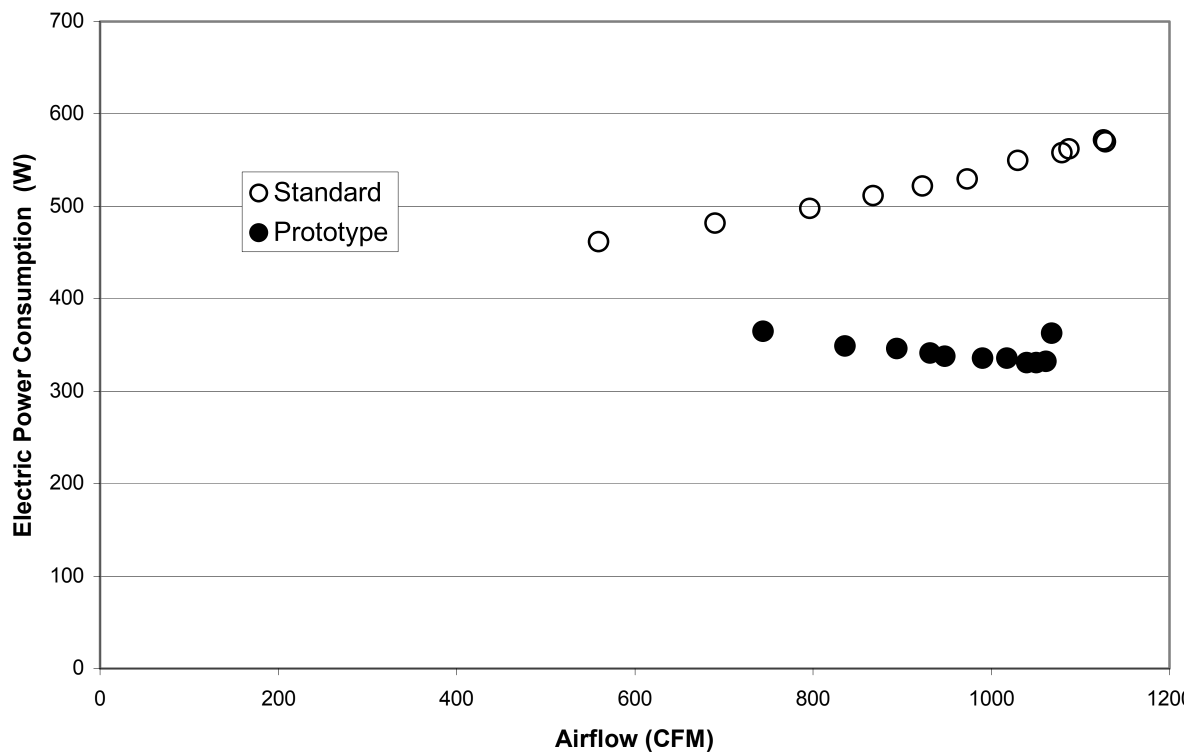

Figure 12. Reduced electrical power consumption for the prototype fan (using torque setting 6 for the prototype)

The effect of the prototype on annual energy consumption can be estimated by taking the standard number of hours of operation used in equipment ratings (2080) and multiplying by the energy savings. Using an operating point of 950 cfm and using torque setting 5, the standard fan consumes 500W and the prototype fan 260W, a saving of 240W (essentially halving the energy use). This is equivalent to 500kWh per year of electricity savings for a heating system. At 10 cents/kWh, this represents $50/year cost savings. As heat is transferred from the air handler fan to the air flowing through the air handler, in heating modes this power difference is the heat lost to the airstream in the furnace. To compensate for lost heat to the airstream in heating mode, this translates to fuel switching from electricity to whatever fuel is used in the furnace. Therefore an additional 500kWh of furnace fuel would be required to meet the same building load. For air conditioners the heat added by the fan is an extra load for the air conditioner. Assuming a COP of about 3 for a typical air conditioner, the 240W of wasted heat requires 80W extra air conditioner capacity. The total power savings is then 320W and assuming a 35% duty cycle for an air conditioner this represents an annual energy savings of almost 1000kWh. At 10 cents/kWh, this represents $100/year cost savings.

Effect of Cabinet Restriction

The tests were repeated with the inlet to the fans restricted as illustrated in Figures 5 and 6. The results (summarized in Figure 13) show that both fans are sensitive to this inlet restriction. The prototype is more sensitive, and its average efficiency dropped from 23.3% to 10.6%. The standard fan was less sensitive, and its average efficiency dropped from 12.5% to 9.1%. Aside from the efficiency changes, the drop in air flow is dramatic. For the prototype, the maximum air flow dropped from 990 cfm to 770 cfm (467 L/s to 363 L/s) - a 22% drop – the same effect as closing nine out of ten registers. Similarly for the standard fan the sair flow dropped from 1120 cfm to 990 cfm (530 L/s to 467 L/s) - a 12 % drop. In terms of cfm/W ratings, the prototype dropped from 3.3 cfm/W to 2.6 cfm/W and the standard fan stayed the same at 1.7 cfm/W.

These results are a dramatic illustration of the sensitivity of air handler performance to cabinet restrictions. If improved motors and fans are to realize their potential they need to be installed in correctly sized cabinets. In addition, when rating air handler fans they should be tested with entering air flow conditions similar to those in field installations. This includes testing them in the cabinets that they are used in and connecting some standardized return plenum or ducting to the air handler entry.

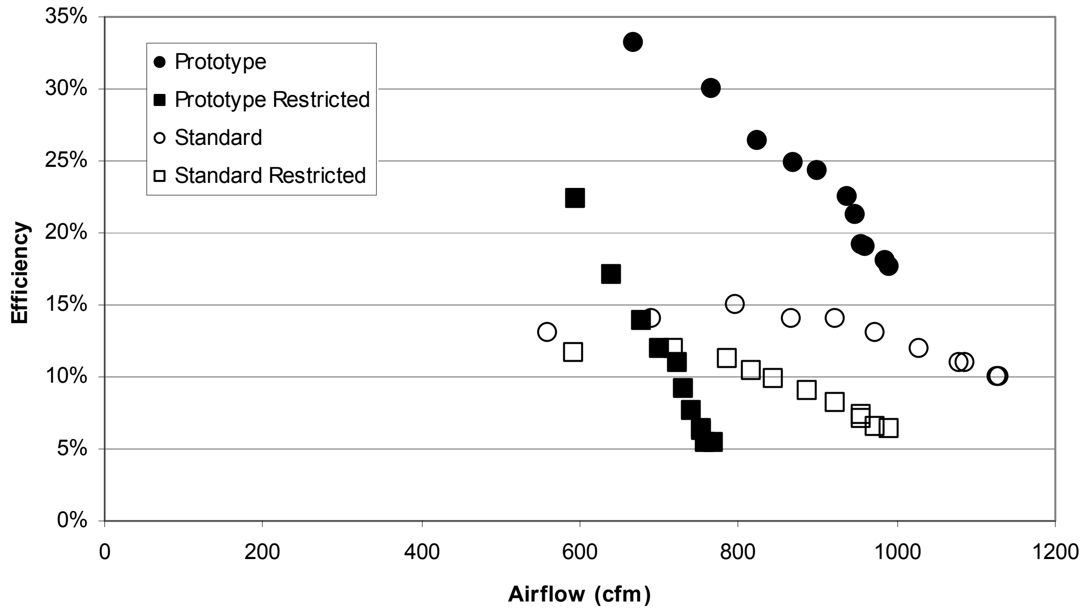

Figure 13. Reduction in air flow and efficiency due to inlet restriction

Separation of motor and fan blade efficiency

Figure 13 showed that the overall efficiency of the prototype air handler (without additional restriction) was almost double the standard fan (23% compared to 12% averaged over all air flows). A key question to be answered is "What fraction of the improved performance of the prototype can be attributed to the electric motor compared to the improved aerodynamic fan blade and housing design?" To investigate this we used a calibration supplied by the prototype manufacturer (Wiegman 2003) that estimated the electric losses from the motor and its controller. These losses are based on the measured rotational speed and power used. The electric motor efficiency was calculated using the measured total power consumption and the calculated losses. Figure 14 shows that the ECM motor efficiency is fairly constant (at about 75%) over a wide range of torque settings and tested air flow rates. The exception is at the lowest torque setting, where the motor is considerable more efficient (greater than 90%).

The aerodynamic (fan blade and housing) efficiency was calculated by taking the output of the electric motor as input power to the fan. The pressure difference and air flow were used to calculated the power put into the air; the ratio of the two is the aerodynamic efficiency. Figure 15 shows how the aerodynamic efficiency is highly variable (between 12% and 45%) depending on the operating condition. This is to be expected because the aerodynamic efficiency depends on the air velocities over the blades and through the housing, and these velocities change over a wide range depending on total airflow and rotational speed of the fan (670 to 1755 rpm for these tests).

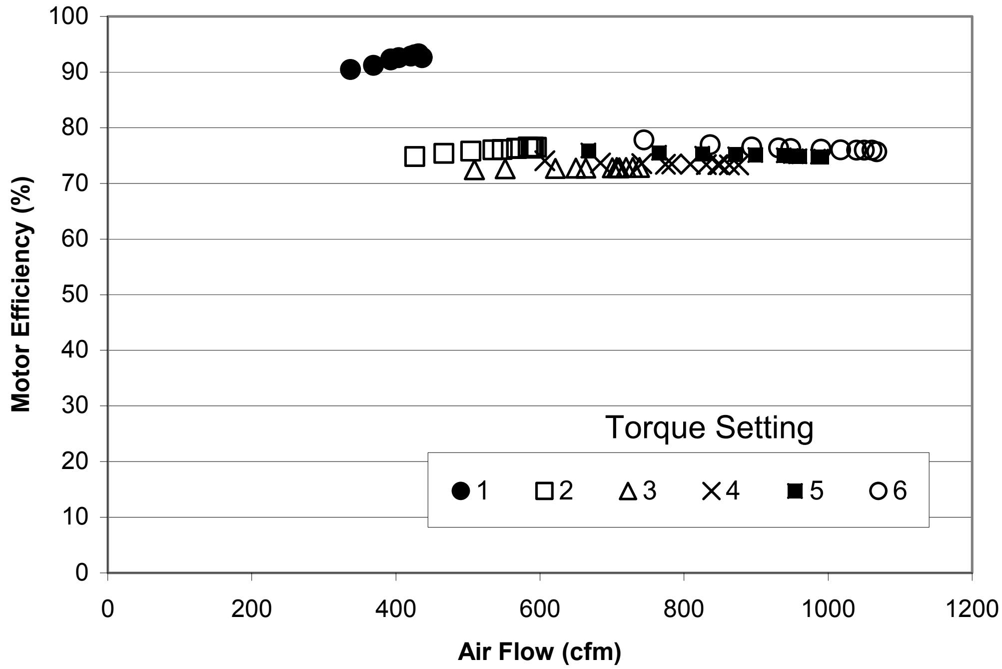

Figure 14. ECM Electric motor efficiency for the Prototype fan.

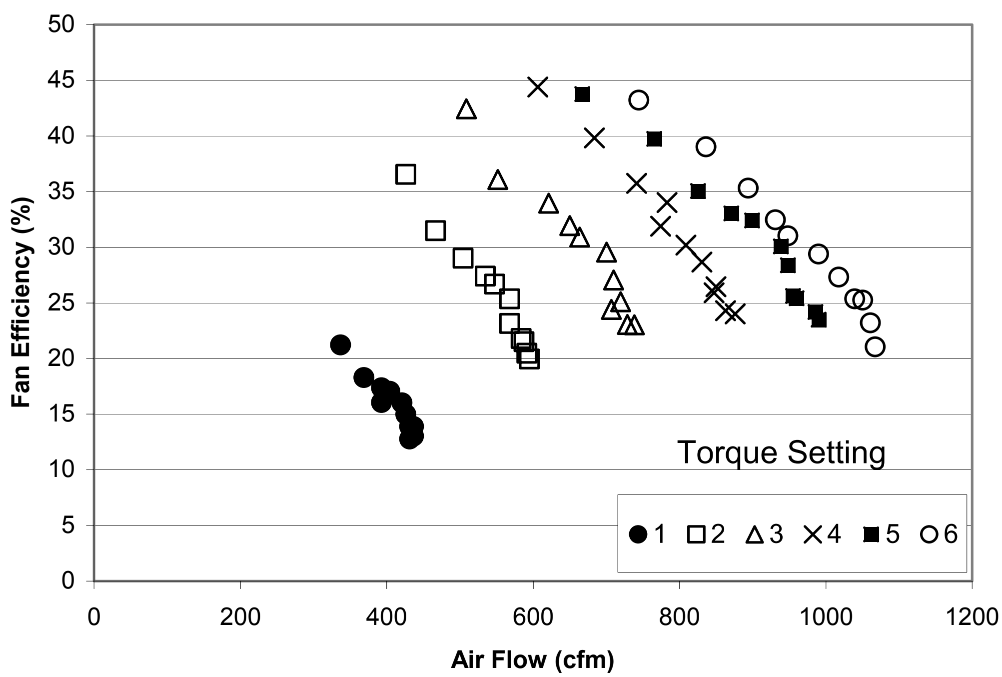

Figure 15. Prototype Fan blade and housing efficiency

System Pressure Differences

The test apparatus uses a typical new duct system made of flexible duct using standard duct fittings. Under normal operating conditions with all the registers open (minimum flow resistance) the pressure difference across the fan was about 125 Pa (0.5 inches of water). Other studies (e.g., Proctor and Parker [2000]) have found similar pressure differences in field testing. The other results reported in the current study show that the fan performance is strongly affected by the system flow resistance. This indicates that pressure difference across fans used in rating procedures (25 to 50Pa [0.1 to 0.2 inches of water]) is too low and should be raised to at least 125 Pa (0.5 inches of water).

CONCLUSIONS

The test results for the prototype air handler show that straightforward engineering changes can significantly increase performance, with the prototype fan being about **twice** as efficient as a standard fan (23% compared to 12%). At lower air flows, such that may be used if the air handler is part of a ventilation system, the prototype fan has an even greater advantage due to its low power consumption – using only 75W at half the maximum fan flow. The protoype fan was less sensitive to increases in system flow resistance than the standard fan – thus it is more likely to maintain air flows as system flow resistance increases due to damper settings for zone control, or with progressive fouling of coils and filters.

The effect of restrictive cabinets was profound. The prototype's efficiency was halved (to 12%) and the standard fan performance was reduced to 9% efficiency. This supports the premise that fans need to be rated in the cabinets that they will be used in. The air flows through the fans were significantly reduced (by up to 22%) by the restriction – it is clear that these restrictions can contribute to the low air flows often found in field installations. If improved motors and fans are to realize their potential they need to be installed in correctly sized cabinets. To improve the representativeness of test conditions compared to operating conditions, the standard pressure difference used across fans during testing should be raised to at least 125 Pa (0.5 inches of water).

Future work should look at a more detailed examination of cabinet restriction effects, and examine a wider range standard and prototype air handlers covering a range of air flow rates

ACKNOWLEDGEMENTS

The authors would like to thank David Bouly for his assistance in the laboratory testing on the air handlers and Duo Wang (Energy Performance of Buildings Group, LBNL) and Jim Lutz (Energy Analysis Depratment, LBNL) for their review of this report.

REFERENCES

ASHRAE Standard 51. 1999. *Laboratory Methods of Testing Fans for Aerodynamic Performance Rating (AMCA Standard 210-99)*, ASHRAE, Atlanta, GA.

Biermayer, P. 2003. *Measurement of Airflow in Residential Furnaces*, Energy Analysis Department, Lawrence Berkeley National Laboratory, Berkeley, CA.

CMHC. 1993. *Efficient and Effective Residential Air Handling Devices*. Research Division, Canada Mortgage and Housing Corporation. Ottawa, Ontario, Canada.

DOE. 1996. *Uniform Test Method for Measuring the Energy Consumption of Central Air Conditioners Appendix M to Subpart B.* US Department of Energy, Washington, DC.

GAMA. 2003. *Consumers Directory of Certified Efficiency Ratings for Heating and Water Heating Equipment.* Gas Appliance Manufacturers Association. Arlington, VA.

Pigg, S. 2003. *The Electric Side of Gas Furnaces*. Home Energy, Vol.20., No.6. Home Energy Magazine, Berkeley, CA. www.homeenergy.org

Phillips, B.G. 1998. *Impact of Blower Performance on Residential Forced-Air Heating System Performance.* ASHRAE Trans. V. 104, Pt.1, American Society of Heating, Refrigeration and Air-conditioning Engineers, Atlanta, GA.

Proctor, J. and Parker, D. 2000. *Hidden Power Drains: Residential Heating and Cooling Fan Power Demand.* Proc. ACEEE Summer Study 2000 pp. 1.225-1.234. American Council for an Energy Efficient Economy, Washington, D.C.

Sachs, H, *Residential HVAC Fans and Motors Are Bigger than Refrigerators*, 2002 Summer Study on Energy Efficiency in Buildings, American Council for an Energy-Efficient Economy, Washington, D.C.

Sachs, H. 2003. *Refurbishing Furnaces*. Home Energy, Vol. 20, no. 6. Home Energy Magazine, Berkeley, CA. www.homeenergy.org

Wiegman, H, 2003. *ECM 4.0 Operation and Control*, GE Research, GE Global Research Center, February 2003

APPENDIX 1: DETAILS OF AIR HANDLER TEST RESULTS

RPM = Motor speed in Revolutions Per Minute
Pelec = electric power consumption, Watts
Pr = return plenum to room pressure difference, Pa
Ps = supply side of fan (measured after fan but before heat exchanger) to room pressure difference, Pa
Pd = pressure difference measured across the flow nozzle (used to determine Q), Pa
ΔP = static pressure across the fan, Pa (Ps-Pr).
P air = power put into the air stream ,i.e., volumetric flowrate multiplied by static pressure difference, W
η = air handler efficiency, i.e., P air divided by Pelec
Q = volumetric air flow rate, cfm
Errors are ECM control module errors reported by the operating software. Appendix 2 discusses these in more detail.

Prototype (1)

Inputs		time	RPM	Pelec (W)	Pr	Ps	Pd
Successive registers closed	0	11:08	669.0	41	-8.0	16.0	4.1
	1	11:12	669.0	42	-8.0	16.6	4.2
	2	11:16	672.3	41	-7.7	18.3	4.1
	3	11:18	670.6	42	-7.6	18.6	4.2
	4	11:26	672.3	41	-7.3	21.1	4.0
	5	11:28	675.7	41	-6.9	23.8	3.9
	6	11:34	677.5	42	-6.6	26.9	3.4
	7	11:45	680.8	41	-6.4	27.5	3.6
	8	11:48	684.2	41	-6.0	29.5	3.4
	9	12:00	694.8	42	-5.5	34.7	3.0
	10	12:06	707.6	42	-4.7	46.0	2.5

Outputs		ΔP (Pa)	P air (W)	η	Q (CFM)		Errors
Successive registers closed	0	24.0	4.9	11.9%	431		
	1	24.6	5.1	12.1%	437		
	2	26.0	5.3	12.9%	431		
	3	26.2	5.4	12.9%	437		1
	4	28.4	5.7	13.9%	426		
	5	30.7	6.1	14.9%	421		
	6	33.5	6.2	14.8%	393		+
	7	33.9	6.5	15.8%	404		
	8	35.5	6.6	16.1%	393		+
	9	40.2	7.0	16.7%	369		1
	10	50.7	8.1	19.2%	337		

Prototype (2)

Inputs		time	RPM	Pelec (W)	Pr	Ps	Pd
Successive registers closed	0	12:15	895.0	74	-14.0	26.2	7.8
	1	12:17	895.0	74	-14.1	27.4	7.7
	2	12:19	898.0	75	-13.7	30.7	7.6
	3	12:21	892.0	74	-13.6	31.2	7.5
	4	12:39	901.0	74	-12.8	36.0	7.1
	5	12:40	904.0	74	-12.2	41.2	7.1
	6	12:43	910.0	75	-11.8	47.1	6.6
	7	12:44	913.1	75	-11.0	50.9	6.3
	8	12:47	922.5	75	-10.4	58.9	5.6
	9	12:49	938.5	76	-9.0	72.9	4.8
	10	12:51	965.3	76	-7.3	96.0	4.0

Outputs		ΔP (Pa)	P air (W)	η	Q (CFM)		Errors
Successive registers closed	0	40.2	11.3	15.3%	595		
	1	41.5	11.6	15.6%	591		
	2	44.4	12.3	16.4%	587		
	3	44.8	12.3	16.7%	584		1
	4	48.8	13.1	17.7%	568		
	5	53.4	14.3	19.3%	568		
	6	58.9	15.2	20.3%	547		
	7	61.9	15.6	20.8%	535		
	8	69.3	16.5	22.0%	504		
	9	81.9	18.0	23.7%	467		
	10	103.3	20.8	27.3%	426		

Prototype (3)

Inputs		time	RPM	Pelec (W)	Pr	Ps	Pd
Successive registers closed	0	12:55	1098.8	122	-20.9	37.9	12.0
	1	12:57	1103.3	124	-20.8	39.8	11.7
	2	1:05	1103.3	123	-20.5	45.1	11.0
	3	1:07	1107.8	123	-20.5	45.7	11.4
	4	1:10	1112.4	123	-19.1	53.2	11.1
	5	1:12	1117.0	123	-18.7	61.4	10.8
	6	1:15	1121.6	123	-17.6	70.7	9.7
	7	1:18	1126.3	124	-16.9	77.1	9.3
	8	1:19	1140.6	125	-14.7	90.5	8.5
	9	1:21	1160.0	127	-12.8	115.1	6.7
	10	1:23	1212.1	131	-10.4	157.4	5.7
Outputs		**ΔP (Pa)**	**P air (W)**	**η**	**Q (CFM)**		**Errors**
Successive registers closed	0	58.8	20.5	16.8%	738		
	1	60.6	20.8	16.8%	729		2
	2	65.6	21.9	17.8%	707		
	3	66.2	22.5	18.3%	719		
	4	72.3	24.2	19.7%	710		
	5	80.1	26.5	21.5%	700		
	6	88.3	27.7	22.5%	664		
	7	94.0	28.8	23.2%	650		
	8	105.2	30.8	24.7%	621		
	9	127.9	33.3	26.2%	552		
	10	167.8	40.3	30.8%	509		

Prototype (4)

Inputs		time	RPM	Pelec (W)	Pr	Ps	Pd
Successive registers closed	0	1:27	1293.3	187	-29.2	50.4	16.9
	1	1:31	1293.3	187	-28.3	53.5	16.4
	2	1:34	1299.5	187	-28.4	60.5	15.8
	3	1:35	1299.5	187	-28.6	61.9	15.9
	4	1:37	1305.8	187	-26.7	73.6	15.2
	5	1:39	1312.1	189	-25.3	84.5	14.4
	6	1:41	1318.5	190	-24.0	97.8	13.2
	7	1:44	1325.0	190	-22.2	106.3	13.5
	8	1:46	1331.6	192	-20.9	123.3	12.1
	9	2:08	1358.3	195	-18.6	158.6	10.3
	10	2:10	1422.7	202	-14.0	218.1	8.1
Outputs		**ΔP (Pa)**	**P air (W)**	**η**	**Q (CFM)**		**Errors**
Successive registers closed	0	79.6	32.9	17.6%	876		
	1	81.8	33.3	17.8%	863		
	2	88.9	35.5	19.0%	847		
	3	90.5	36.3	19.4%	850		
	4	100.3	39.3	21.0%	831		
	5	109.8	41.9	22.2%	809		
	6	121.8	44.5	23.4%	774		
	7	128.5	47.5	25.0%	783		3
	8	144.2	50.4	26.3%	741		+
	9	177.2	57.2	29.3%	684		
	10	232.1	66.4	32.9%	606		

Prototype (5)

Inputs		time	RPM	Pelec (W)	Pr	Ps	Pd
Successive registers closed	0	2:12	1469.0	266	-37.0	63.0	21.6
	1	9:47	1469.0	265	-37.5	65.5	21.4
	2	9:52	1477.0	266	-36.5	75.3	20.3
	3	9:53	1477.0	265	-35.4	77.4	20.1
	4	9:55	1485.1	266	-34.3	92.2	19.8
	5	9:56	1493.5	272	-33.0	105.6	19.4
	6	9:58	1501.7	268	-30.4	123.3	17.8
	7	9:59	1510.0	271	-28.6	135.2	16.7
	8	10:01	1527.1	272	-26.9	157.5	15.0
	9	10:03	1544.6	272	-22.3	203.7	12.9
	10	10:04	1590.0	273	-17.8	270.3	9.8

Outputs		∆P (Pa)	P air (W)	η	Q (CFM)		Errors
Successive registers closed	0	100.0	46.7	17.6%	990		3
	1	103.0	47.9	18.1%	986		+
	2	111.8	50.7	19.0%	960		
	3	112.8	50.9	19.2%	955		
	4	126.5	56.6	21.3%	948		
	5	138.6	61.4	22.6%	939		
	6	153.7	65.2	24.3%	899		
	7	163.8	67.3	24.8%	871		
	8	184.4	71.8	26.4%	825		
	9	226.0	81.6	30.0%	765		
	10	288.1	90.7	33.2%	667		

Prototype (6)

Inputs		time	RPM	Pelec (W)	Pr	Ps	Pd
Successive registers closed	0	10:07	1580.7	363	-43.7	71.1	25.1
	1	10:10	1590.0	332	-42.6	74.3	24.8
	2	10:11	1590.0	331	-42.0	86.0	24.3
	3	10:13	1590.0	331	-42.1	87.9	23.8
	4	10:15	1599.4	336	-39.5	105.8	22.8
	5	10:17	1609.0	336	-36.9	124.0	21.6
	6	10:18	1618.6	338	-34.5	144.2	19.8
	7	10:20	1628.4	341	-33.8	158.5	19.1
	8	10:22	1648.2	346	-30.7	191.0	17.6
	9	10:25	1679.0	349	-26.8	238.7	15.4
	10	10:27	1755.3	365	-21.3	328.0	12.2

Outputs		∆P (Pa)	P air (W)	η	Q (CFM)		Errors
Successive registers closed	0	114.8	57.8	15.9%	1068		
	1	116.9	58.5	17.6%	1061		
	2	128.0	63.4	19.2%	1050		
	3	130.0	63.8	19.3%	1040		
	4	145.3	69.8	20.8%	1017		
	5	160.9	75.2	22.4%	990		
	6	178.7	80.0	23.7%	948		
	7	192.3	84.5	24.8%	931		
	8	221.7	93.5	27.0%	894		
	9	265.5	104.8	30.0%	836		
	10	349.3	122.7	33.6%	744		

Standard Unrestricted

Inputs

Successive registers closed	time	RPM	Pelec (W)	Pr	Ps	Pd
0	2:12	1469.0	266	-37.0	63.0	21.6
1	9:47	1469.0	265	-37.5	65.5	21.4
2	9:52	1477.0	266	-36.5	75.3	20.3
3	9:53	1477.0	265	-35.4	77.4	20.1
4	9:55	1485.1	266	-34.3	92.2	19.8
5	9:56	1493.5	272	-33.0	105.6	19.4
6	9:58	1501.7	268	-30.4	123.3	17.8
7	9:59	1510.0	271	-28.6	135.2	16.7
8	10:01	1527.1	272	-26.9	157.5	15.0
9	10:03	1544.6	272	-22.3	203.7	12.9
10	10:04	1590.0	273	-17.8	270.3	9.8

Outputs

Successive registers closed	ΔP (Pa)	P air (W)	η	Q (CFM)		Errors
0	100.0	46.7	17.6%	990		3
1	103.0	47.9	18.1%	986		+
2	111.8	50.7	19.0%	960		
3	112.8	50.9	19.2%	955		
4	126.5	56.6	21.3%	948		
5	138.6	61.4	22.6%	939		
6	153.7	65.2	24.3%	899		
7	163.8	67.3	24.8%	871		
8	184.4	71.8	26.4%	825		
9	226.0	81.6	30.0%	765		
10	288.1	90.7	33.2%	667		

Standard Restricted

Inputs

Successive registers closed	time	RPM	Pelec (W)	Pr	Ps	Pd
0	2:16	1571.6	275	-24.5	16.6	13.0
1	2:18	1580.7	274	-23.0	18.3	12.7
2	2:20	1580.7	275	-22.4	26.2	12.5
3	2:21	1580.7	275	-22.3	27.3	12.5
4	2:23	1580.7	276	-21.9	38.1	12.1
5	2:25	1590.0	274	-20.9	51.8	11.8
6	2:27	1599.4	277	-20.3	68.7	11.5
7	2:28	1599.4	273	-19.6	79.2	10.8
8	2:31	1599.4	274	-18.2	101.3	10.1
9	2:32	1609.0	274	-16.7	138.3	9.0
10	2:35	1638.3	270	-13.7	201.6	7.8

Outputs

Successive registers closed	ΔP (Pa)	P air (W)	η	Q (CFM)		Errors
0	41.1	14.9	5.42%	768		
1	41.3	14.8	5.40%	759		
2	48.6	17.3	6.28%	753		
3	49.6	17.6	6.41%	753		
4	60.0	21.0	7.60%	741		
5	72.7	25.1	9.17%	732		
6	89.0	30.3	11.0%	723		
7	98.8	32.6	12.0%	700		
8	119.5	38.2	13.9%	677		
9	155.0	46.8	17.1%	639		
10	215.3	60.5	22.4%	595		

Protoype unrestricted

Inputs	Successive registers closed	time	RPM	Pelec (W)	Pr	Ps	Pd
	0	n/a	n/a	572	n/a	n/a	n/a
	1			570			
	2			562			
	3			558			
	4			550			
	5			530			
	6			522			
	7			512			
	8			498			
	9			482			
	10			462			

Outputs	Successive registers closed	ΔP (Pa)	P air (W)	η	Q (CFM)		Errors
	0	106.0	56.3	10.0%	1126		
	1	110.5	58.8	10.0%	1128		
	2	120.1	61.6	11.0%	1087		
	3	123.5	62.9	11.0%	1079		
	4	137.8	67.0	12.0%	1030		
	5	152.7	70.1	13.0%	973		
	6	163.9	71.4	14.0%	923		
	7	175.3	71.8	14.0%	867		
	8	193.6	72.8	15.0%	796		
	9	211.7	68.9	14.0%	690		
	10	235.9	62.3	13.0%	559		

Prototype restricted

Inputs	Successive registers closed	time	RPM	Pelec (W)	Pr	Ps	Pd
	0	10:44	n/a	530	-37.6	35.3	21.6
	1	10:45		533	-37.4	38.7	20.9
	2	10:46		527	-35.2	48.4	20.1
	3	10:47		529	-35.0	50.6	20.1
	4	10:48		524	-33.8	64.5	18.8
	5	10:49		515	-30.8	80.3	17.3
	6	10:50		503	-27.5	97.8	15.7
	7	10:51		500	-26.5	108.6	14.7
	8	10:52		489	-23.5	124.0	13.6
	9	10:53		477	-19.2	148.1	11.4
	10	10:54		469	-14.1	182.9	7.7

Outputs	Successive registers closed	ΔP (Pa)	P air (W)	η	Q (CFM)		Errors
	0	72.9	34.1	6.43%	990		
	1	76.1	35.0	6.56%	974		
	2	83.6	37.7	7.15%	955		
	3	85.6	38.6	7.30%	955		
	4	98.3	42.9	8.18%	924		
	5	111.1	46.5	9.02%	886		
	6	125.3	49.9	9.9%	844		
	7	135.1	52.1	10.4%	817		
	8	147.5	54.7	11.2%	786		
	9	167.3	56.8	11.9%	719		
	10	197.0	55.0	11.7%	591		

Appendix 2. Prototype Operating Problems

The prototype fan used a computer interface to control the fan (this would not be the case in production models where the control would be hard wired). During testing of the prototype this control system generated several errors. All errors prompted the controller to shut down the fan, and only once was the fan able to be restarted without shutting down both the fan and the controller and resetting the system. About half the time there was a response returned from the controller. The errors occurred only when commands where being sent to the controller, and they came up as often with “Set” commands as with “Request” commands.

The computer to motor interface controller program (Comm 1.0) has a “Request history” command to help monitor the state of the fan/controller setup. It seemed the “Over current” error came up the most often at lower torque settings, while “Over speed” error populated the upper torque settings.

Another issue with the ECM was inconsistent returns from the controller. Regularly, the controller’s response would not match the request sent, such as replying that the fan was spinning counter clockwise on a speed request. In addition, with successive speed requests the values returned were often erratic. For example, when the speed was around 1300 rpm, the value returned ranged from 50-2200 rpm. Consequentially, speed data was taken as the most prominent value returned from seven or fourteen requests.

These control issues indicate that more work is required on the control algorithms and hardware before this fan is ready for production.