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Improving Air Handler Efficiency in Houses

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ABSTRACT

Although furnaces, air conditioners and heat pumps have become significantly more efficient over the last couple of decades, residential air handlers have typical efficiencies of only 10% to 15% due to poor electric motor performance and aerodynamically poor fans and fan housings. Substantial increases in performance could be obtained through improved air handler design and construction. A prototype residential air handler intended to address these issues has recently been developed. The prototype and a standard production fan were tested in a full-scale duct system and test chamber at LBNL 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 the prototype air handler had about twice the efficiency of the standard air handler (averaged over a wide range of operating conditions) and was less sensitive to duct system flow resistance changes. The performance of both air handlers was significantly reduced by reducing the clearance between the air handler and cabinet it was placed in. These test results showed that in addition to the large scope for performance improvement, air handler fans need to be tested in the cabinets they operate 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, Gusdorf et al. 2002). These low efficiencies indicate that there is significant room for improvement of these air handler fans. 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. 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. This has increased the need to have the air handler fan energy use included in ratings or standards. Part of the reason why there has been little fan efficiency improvement is that air handler fan energy use has not specifically been 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. 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.

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 – higher static pressure in the field compared to the GAMA test procedure. 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.

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 better electric 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.	Improve manufacturing tolerances
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 a Brushless Permanent Magnet (BPM) motor 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 motors, e.g., BPM.
Fans are fitted in restrictive furnace cabinets.	Integrate design of furnace cabinets and fan housings to ensure sufficient clearance around fan inlets.

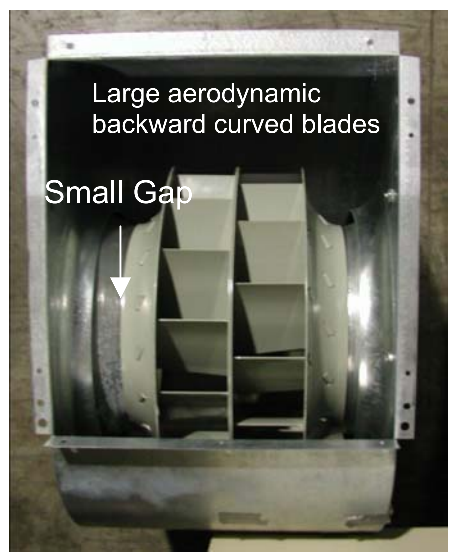
Tested fans

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. 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.

The standard fan in a residential forced-air heating and cooling system has a permanent split capacitor (PSC) type motor. 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). Speed is controlled by jumpers on the control board located on the fan housing. 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. 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 a Brushless Permanent Magnet (BPM). Since the speed of a BPM is electronically controlled, it can be set specifically to match the airflow requirements for each application. Furthermore, BPM's can be operated in a mode that changes speed 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, BPM's use power approximately proportional to their airflow requirements, thereby making them inherently more efficient than PSC motors over a wide range of air flows. 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. 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 the standard fan we used for comparison. The better tolerances should lead to 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

All the tests were conducted using the Energy Performance of Buildings Group full-scale duct system and test chamber. The test chamber 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]). 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. The ducts were carefully sealed and have a total leakage of about 5 cfm25 (8 L/s at 25 Pa). This system normally operates with a system air flow of 1125 CFM at a static pressure of 0.5 in. water (0.554 m^3/s at 125 Pa) with the standard air handler and all the registers open.

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. Fan inlet and exit static pressures 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. Electrical power consumption was measured with a true power meter which accounts for the fan power factor. All the data were recorded using five-second time averages using an automated data acquisition system.

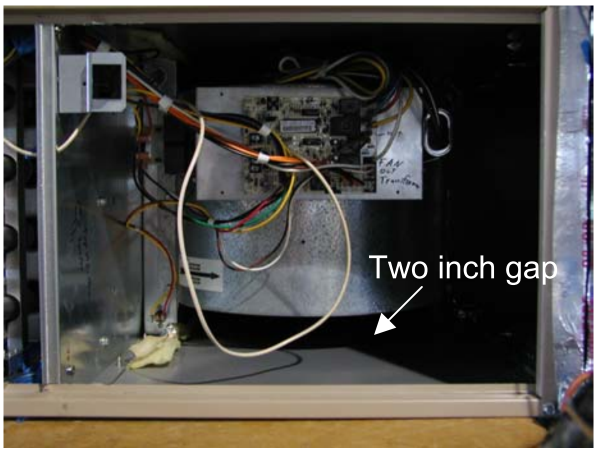
Different test operating points were obtained by systematically closing supply registers to increase the flow resistance of the system. After each register was closed the system was allowed to operate for several minutes to reach a steady operating point. The standard air handler was operated at a single 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 experiments were repeated with blockage installed on the walls of the air handler cabinet to simulate installation in a smaller cabinet. 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 (Figure 2).

Results

Figure 3 shows how the prototype fan 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. With the increase in flow resistance as more registers are closed, the pressure difference across the fan increases (Figure 4). The electric power consumption remains relatively constant for each torque setting no matter how many registers are closed. 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 5. 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, and are higher than the almost 20% efficiency reported by Gusdorf et al. 2002 for a BPM motor with a standard fan assembly.

Figure 2. Standard fan in cabinet with no restriction (left) and with added foam board restriction (right).



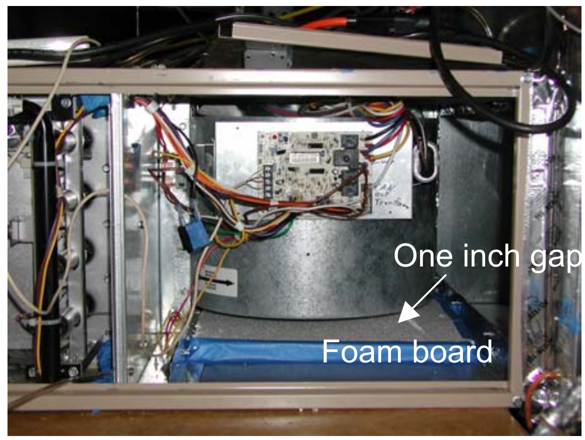


Figure 3. Airflow changes due to torque setting and system flow resistance (number of closed registers)

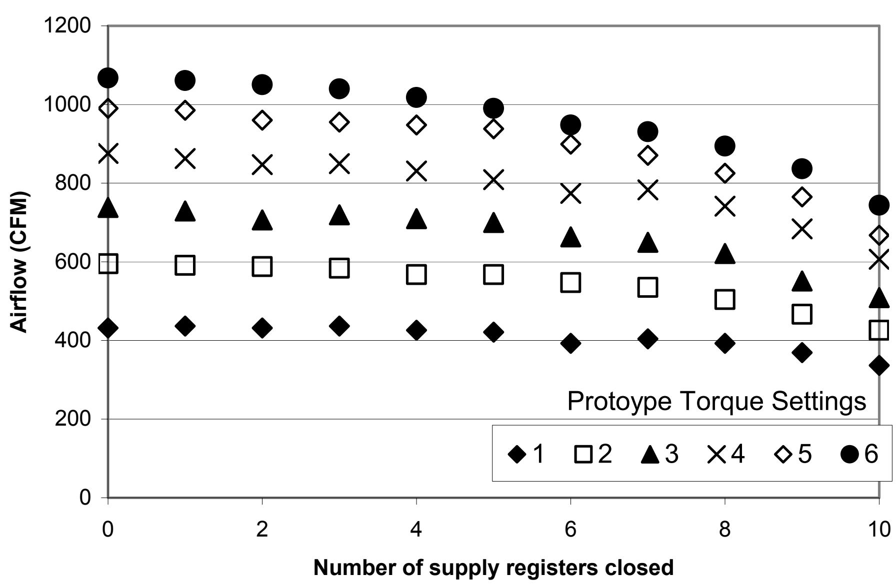


Figure 4. Static Pressure changes due to system flow resistance (number of closed registers)

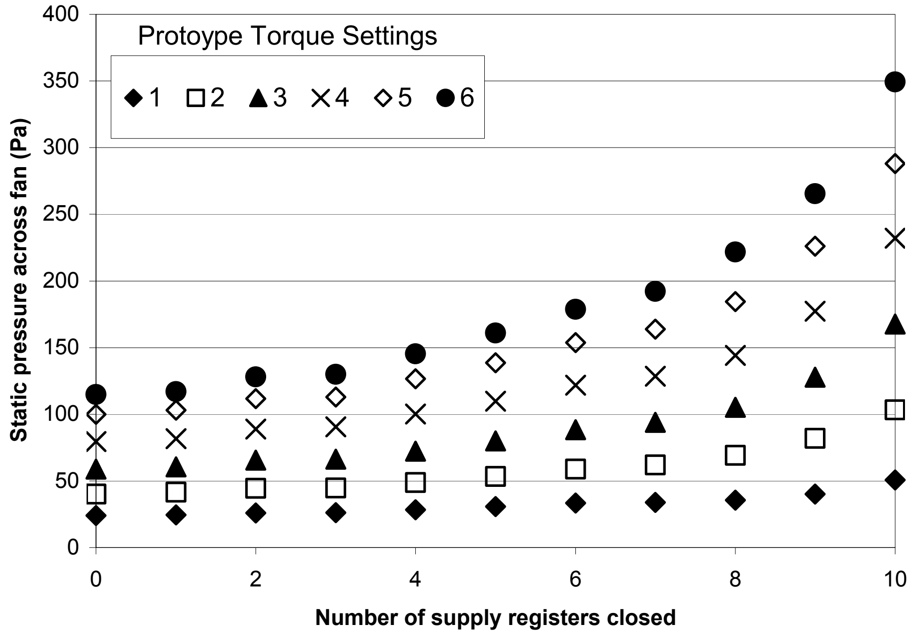
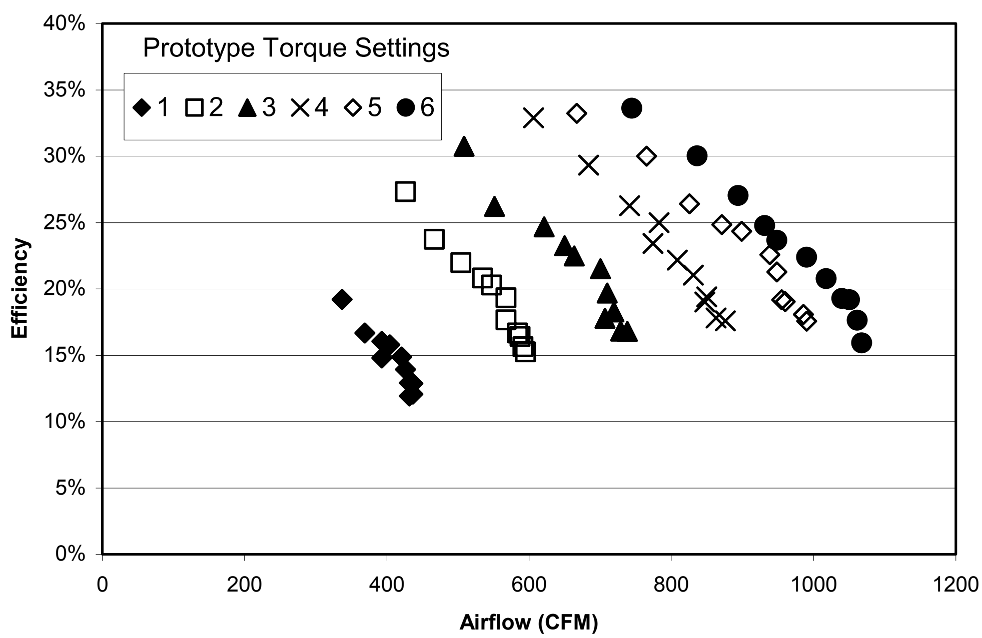
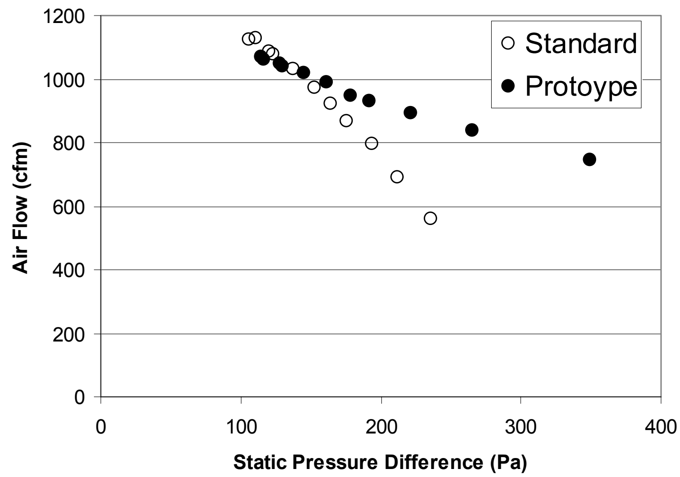


Figure 5. Efficiency and airflow as system flow resistance is increased



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 6 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 6. Comparison of standard and prototype fan performance curves.



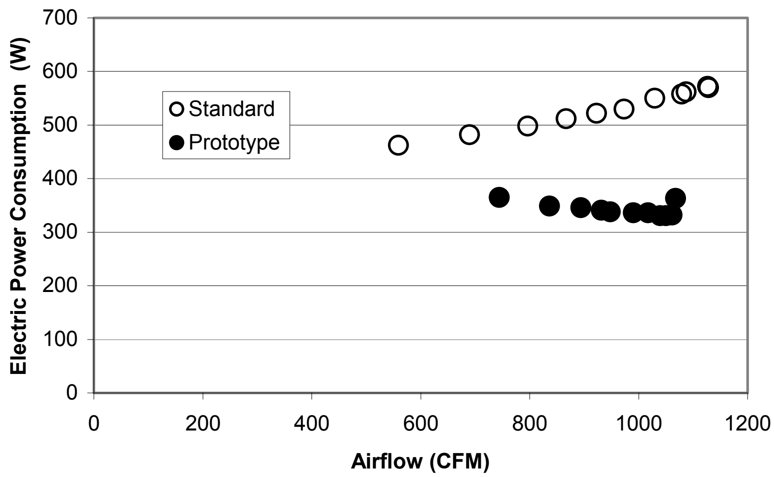
Electricity and Energy Consumption

Figure 7 shows that the prototype fan requires only about half the power of the standard fan. Because the air flow rates and corresponding number of closed registers never exactly match for the two fans we need to look at the data in a couple of different ways: 1). high flow rates, and 2.) same flow rate and amount of system resistance.

1. High flow rate. 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 7, 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.

2. Same flow rate and system flow resistance. 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 7. 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 number of hours of operation and multiplying by the energy savings. Pigg (2003) indicate that the typical number of hours of operation for a furnace is about 960 hours based on observations from rating procedures and measured field data. Using an operating point of 950 cfm and using torque setting 5, the standard fan consumes 500 W and the prototype fan 260 W, a saving of 240 W. This is equivalent to 230 kWh per year of electricity savings for a heating system. At 10 cents/kWh, this represents $23/year cost savings. There will also be a corresponding 230 kWh/year increase in furnace fuel consumption. This is equivalent to about 8 therms of natural gas. At 70 cents/therm, the additional fuel cost is about $5.5 – so the net saving is about $17.5. For heating, the air handler waste heat essentially represents fuel switching to electricity. In source terms this represents about 690 kWh/year of electricity.

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. Sachs and Smith (2003) report that the typical number of operating hours is 900 for air conditioners based on rating procedures (ARI 1994). Therefore, the annual energy savings would be about 290 kWh (870 kWh/year in source energy). At 10 cents/kWh, this represents $29/year savings.

If the fan operates continuously for ventilation/filtration/air distribution purposes, the savings are obviously much greater. Subtracting the combined 1860 hours of heating and cooling operation from 8760 total hours gives 6900 hours of continuous fan operation. In this mode, the required air flow rate is lower than for heating or cooling – for the purposes of these calculations we will use 400 cfm. The standard fan consumes about 450 W a this low air flow, but the prototype can be operated at a lower torque setting and requires only 41 W (this compares to a 500W/100W ratio for PSC vs. BPM motors (without the aerodynamic improvements of this prototype) given by Pigg (2003)). The resulting savings are over 2800 kWh/year and are close to those reported by Pigg (2003). This result also agrees well with the analysis performed by Gusdorf et al. (2002) who used measured data in side-by-side houses (one with a BPM one with a PSC motor) to estimate heating season (with continuous fan operation) savings of 1800 kWh/year.

Effect of Cabinet Restriction

The results of the restricted cabinet tests (Figure 8) 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 air 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.

Separation of motor and fan blade efficiency

To separate the effect of the electric motor and improved aerodynamics 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. The prototype 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 considerably 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 calculate the power put into the air; the ratio of the two is the aerodynamic efficiency. Figure 9 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 8. Reduction in air flow and efficiency due to inlet restriction

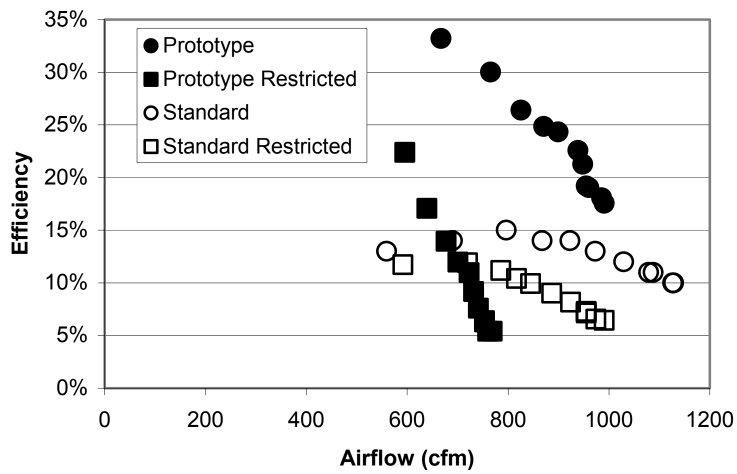
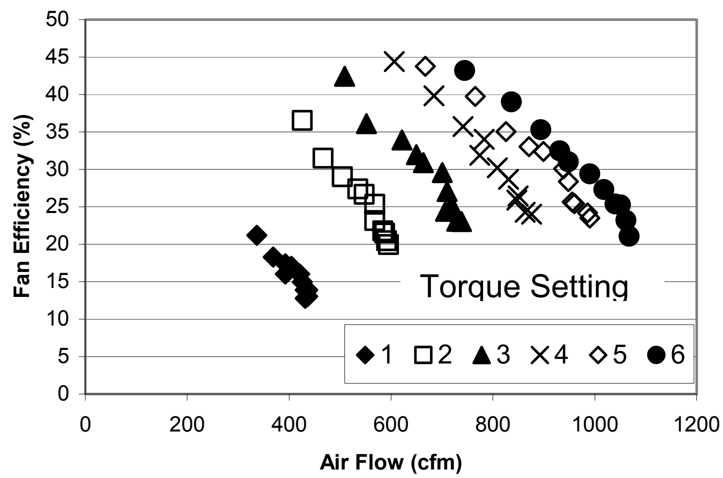


Figure 9. Prototype fan aerodynamic efficiency



Summary

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 prototype 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 significant. 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.

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