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ERNEST ORLANDO LAWRENCE BERKELEY NATIONAL LABORATORY

Residential Furnace Blower Performance

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Disclaimer

Executive Summary

The objective of this study was to assess the performance of furnace blowers and the potential cost-effectiveness of setting performance standards and changing motor technologies.

In this study, a testing program was undertaken at Lawrence Berkeley National Laboratory and the PG&E test laboratory in San Ramon to compare the performance of furnace blowers. Five different combinations of blowers and residential furnaces were tested for air moving performance. The laboratory test results for blower power and air flow were combined with DOE2 models of building loads, models of air conditioner performance[1], standby power, as well as igniter and combustion air blower power.

Energy savings and peak demand reductions in this study are based on replacing a Permanent Split Capacitor (PSC) blower, dominant in the market, with a Brushless Permanent Magnet (BPM) blower[2]. Annual energy savings for a typical three-and-a-half ton air conditioner with typical California ducts are 45 kWh. Peak demand reductions are 50 W per system, or about 0.13 GW statewide if all blowers were replaced. The numbers improve significantly for duct systems that match manufacturers' rating points. For such systems, annual energy savings increase to 153 kWh, and peak demand reductions to 70W per system, or about 0.18 GW statewide when all blowers are replaced. The payback time assuming no utility rebates for typical system is 13.5 years and for an improved system that meets manufacturer's specifications is reduced to 4 years. If fans were operated continuously—for mixing, filtration or ventilation purposes, the potential savings for BPMs improve even more, about 1800 kWh per year and the payback time is less about 4 months.

However, the benefits of the variable speed BPM motors were found to depend strongly on interactions with the rest of the thermal distribution system. The high air flow resistance of typical California duct systems almost eliminates the advantages of the BPM motors. As a result, BPM blowers may not be cost-effective because the potential improvements blower performance are restricted by the static pressures prevalent in residential thermal distribution systems in California.

It is possible to obtain better performance from BPMs and interactions with the rest of the HVAC system were found to be very important. Key areas for improvement include:

- Use of larger or multiple returns
- Use of low pressure filters – promote the use of 4 inch deep pleated filters
- Use of larger air conditioning coils to reduce coil pressure drop[3]
- Use of more compact duct systems with shorter duct runs

[1] To account for reduced air conditioner capacity at lower air flows for PSC motors.

[2] The furnace has two-stage heating and single speed cooling

[3] Need to be careful about maintaining a good match with the rest of the cooling system.

- Careful installation of ducts to reduce number of elbows and make duct runs as straight as possible
- Encourage use of sheet metal duct instead of flexible duct

In conclusion:

- BPM and PSC blowers have distinctly different performance characteristics that must be accounted for when proposing performance specifications
- A performance specification should include both an air flow specification and a cfm/W specification.
- One important utility issue with BPMs is their lower power factor. For PSC motors, the power factor ranged from 0.7 to 0.9, with the lower power factors at high-speed settings and high pressures. For BPMs the power factors range from 0.53 to 0.62 with higher power factors at higher speed.
- As well as the external static pressure effects, the air flow patterns within blower door cabinets also affect performance.

Background

The blowers in residential furnaces typically move heated air through a duct system that distributes heat and then returns it to the furnace. Usually the blowers are double inlet models with air entering the centrifugal blower at both sides. The motor mounts inside one side of the blower (see Figure 2). If a house has central air-conditioning, the same blower is usually used to move the cooled air throughout the house. Some systems also use the central blower to distribute ventilation air or to mix air to improve comfort and reduce stratification.

Although furnaces, air conditioners and heat pumps have become significantly more efficient over the last couple of decades, residential forced air system blowers have not experienced similar improvement. The most common blowers have been shown by in-field testing to have efficiencies of only 10% to 15% (Phillips 1998 & 1995 Gusdorf et al. 2002). These low efficiencies indicate that there is significant room for improvement of both electric motor and the aerodynamic performance of furnace blowers.

An important consideration in analyzing forced air system blowers is that essentially all of the wasted electricity is manifested as heat. This extra heat reduces air conditioning cooling and dehumidification performance and effectively acts as a low-efficiency electric supplement to fossil-fueled furnaces. For heat pumps, this heat substitutes for vapor compression-based high COP heating and effectively reduces the COP of the heat pump.

The California Energy Commission (CEC) requested laboratory testing of furnace blowers to provide technical background support for potential changes to the 2008 California Building Energy Efficiency Standards. Proposed changes include minimum ratings (in cfm/W) for efficient blower credits that require field measurement of blower power.

In the CEC tests, blowers were compared according to their air flow and power consumption characteristics over a wide range of operating pressures. Criteria for evaluation included:

- How air flow rates change with system air flow resistance
- Performance changes with changes in operating speed
- Comparing air flow and power consumption at typical operating conditions
- Comparing blower efficiency (output air power divided by electricity input)
- Comparing cfm/W ratings (the performance metric most likely to be used by rating schemes)

Characteristics of Residential Blowers

The following alternative blowers were evaluated in this study.

1. PSC motors with forward inclined blade blower wheels

Permanent split capacitor motors are by far (>90% of the residential market) the dominant air handler used today. The single-phase PSC motors are six-pole induction motors with a synchronous rotation speed of 1200 rpm. They can operate at several fixed speeds over a range of air flow rates, with highest air flows about 1.5 times the lowest air flows. The speed is set by using different electric current taps that result in different slip, or lag from synchronous speed, of the rotor. Different speeds are necessary to match the different air flow requirements for heating and cooling operation, and allow a single blower to have a wider range of applications than if it operated at a single speed.

Speed is controlled by changing jumpers on the control board located on the fan housing and/or spade-lug connectors on the motor. The relatively constant rotational speed at each tap means that the air flow is highly variable with static pressure. As shown in Figure 1, the blower wheel has many narrow chord forward curved bent sheet metal blades, with large gaps between the wheel and housing. The housing has one opening on each side with the direct drive motor located in one of these openings (Figure 2), and a rectangular discharge. This side entry means that the air flow pattern inside the air handler cabinet is fairly convoluted as air typically enters the bottom of the cabinet, flows around the housing, then changes direction 90° to enter the blower wheel. Also, unlike older belt-drive blowers, the mounting of the electric motor in the inlet restricts the flow on that side of the fan.

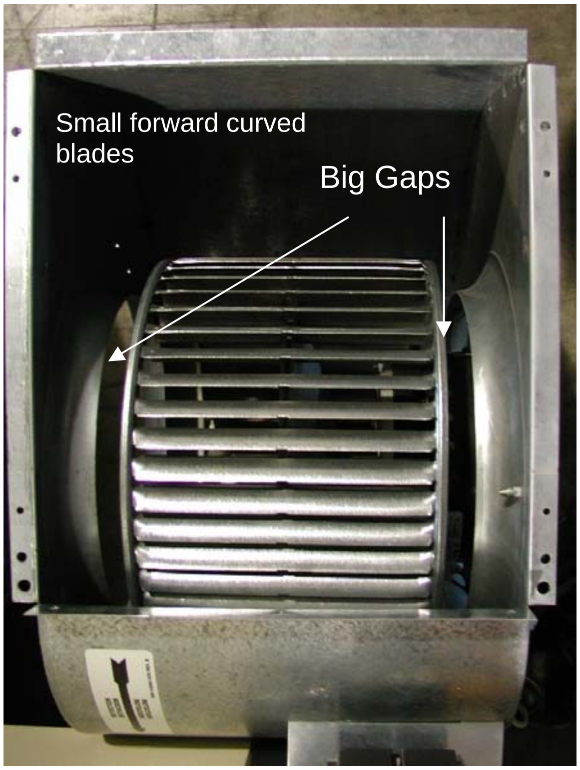


Figure 1. Blower viewed from the air exit.

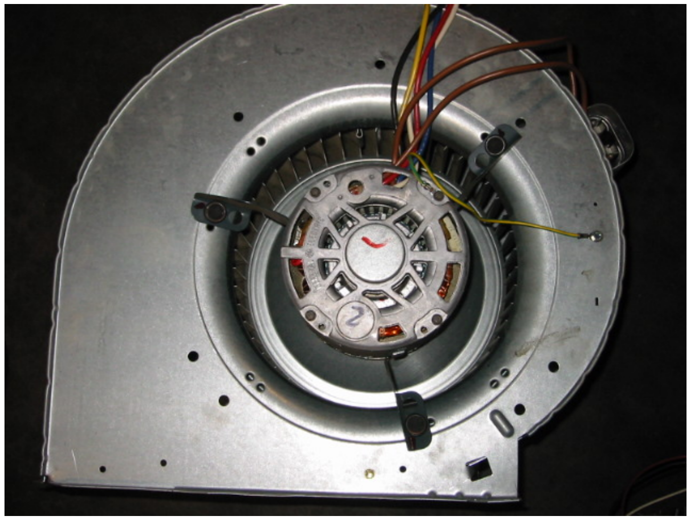


Figure 2. Fan housing inlet with direct drive PSC motor

2. Variable Speed BPM motors and forward inclined blade blower wheels

The second motor type tested was a brushless permanent magnet motor. These motors electronically control the rotating stator field by shifting the field to different coils in the windings. The rotor consists of permanent magnets directly mounted on the shaft of the motor. By varying the voltage and frequency of the electrical current to the stator coils, the motor can be made to rotate over a wide range of speeds and torques. The blower and motor combination can provide a constant air flow across a wide range of static pressures through programming controls based on the performance of the blower.

Because the magnetic field of the rotor is provided by permanent magnets, this type of motor is more efficient than a PSC induction motor, where the field in the rotor is induced by the current caused by the rotating magnetic field of the stator coils. A key characteristic related to the wide speed range of BPM motors is their ability to operate at much lower air flow rates, making them more suitable for continuous fan operation used for mixing and/or distribution of ventilation air. The ability to operate at much lower air flows (usually about 2.5 times less than the maximum air flow) results in the use of considerably less power at low fan speeds. The blower wheel and housing are the same as those used with PSC motors.

3. Prototype Blower

Although the prototype blower is not available in the market, it is included in this report as an example of an emerging technology that PG&E may want to encourage. In 2000, the U.S. Department of Energy funded development of a prototype blower with backward inclined impellor blades (shown in Figure 3). To achieve the same airflows, the prototype's impellor rotates faster than impellors with forward curved blades. Because of the narrower inner diameter and the higher rotational speed of the impellor (up to 2000 rpm compared to a maximum of 1200 rpm for the forward inclined blower), a new generation BPM was developed for this blower. The higher rotational speeds needed to achieve the required air flow means that the impellor cannot be used with a PSC motor.

A report by Weigman et al. (2003) discusses the development of the prototype air handler in detail. This report disaggregated the efficiency increases for different technical aspects of the prototype air handler: the use of rear (or backward) inclined blades (5 to 10% improvement), using inlet cones to condition the flow (2 to 4% improvement), increasing the outlet area (principally height) of the blower housing (5 to 12% improvement), and cabinet effects (inlet cones increased efficiency from 6% to 1%).

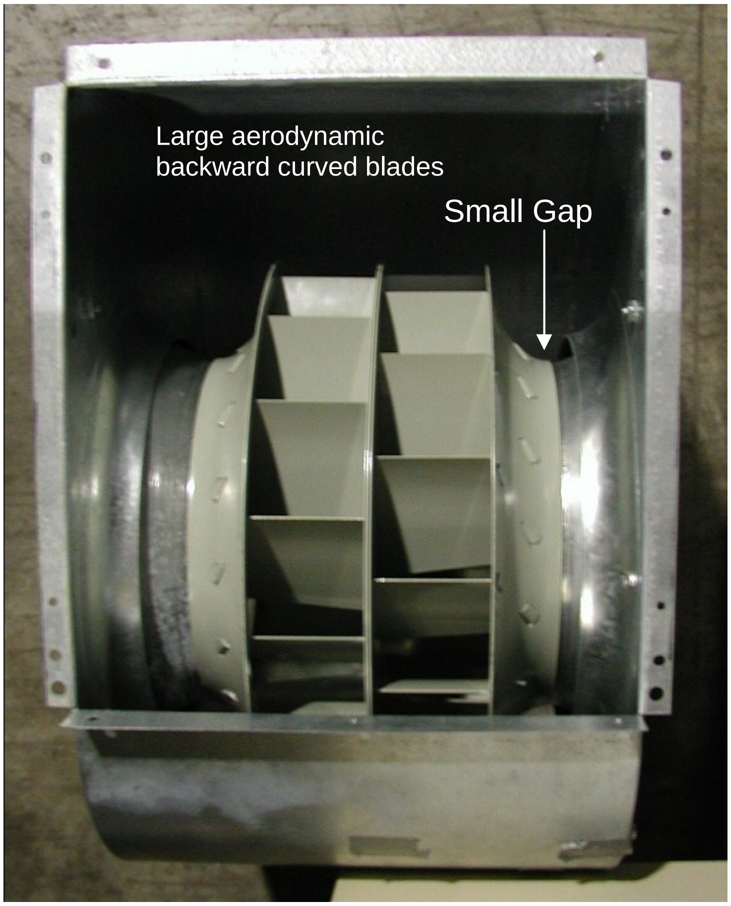


Figure 3. Prototype blower showing different blade design and fan to housing clearances compared to the standard blower in Figure 1.

Objective

The aim of this Residential Furnace Blower study is to characterize the performance of residential furnace blowers to determine typical energy consumption, how performance differs between standard laboratory rating conditions and real life applications, and what attributes of blowers contribute significantly to their performance. The results will help PG&E assess the potential cost-effectiveness of related rebate programs.

The basis for this study are laboratory tests of several furnace blowers over a wide range of conditions combined with the results of other studies that include recent field surveys of blowers in California houses.

Method

Blower Performance Metrics

The baseline engineering metric for blower performance is efficiency. Blower efficiency is defined as the air power divided by the electric power consumption of the electric motor and includes both electric and aerodynamic effects. The air power is defined as the volumetric air flow through the blower multiplied by the pressure difference across the blower.

$$\eta = \frac{Q\Delta P}{W}$$

Where η is the efficiency, Q the air flow (m^3/s), ΔP the pressure difference (Pa) and W is the electric power (Watts).

The pressure difference is the difference in total pressure (static plus velocity) between the blower inlet and exit. In practice, static pressure is often used because the velocity pressures are low. To determine the efficiency of the blower the pressure difference needs to be measured at the inlet to the blower wheel and at the blower exit. These locations are difficult to access and the complexity of the flow field at these locations adds considerable difficulty and uncertainty to these measurements.

In field applications, “external static” pressure is used. That is the pressure difference between the return plenum/blower cabinet and the supply plenum. This pressure difference includes pressure drops in the cabinet itself and across the furnace heat exchanger. For systems including cooling coils, this external static pressure will also include pressure drop across the coil.

In this study, the external static (plenum-to-plenum) pressure difference was used as this is the pressure difference that will be used in field tests. The systems tested for this study had no cooling coils between the furnace exit and the supply duct plenum. In field testing of systems with cooling coils it is important to note that the coils have significant static pressure drops and that when evaluating the system static pressure for blower performance this needs to be included in the measurement. In which case, the external static pressure is measured between the return plenum and a point between the furnace exit and the entry to the cooling coil. The applicability of the test results will be discussed in terms of possible future field testing requirements in building codes.

The efficiency metric is difficult to use in energy and peak demand calculations, so most applications use ratios of air flow (cfm) and power consumption (W). Two combinations are common: cfm/W or Watts/1000 cfm. When interpreting cfm/W results, it is important to realize that there is a limit on cfm/W ratings for 100% efficient operation:

- For static pressure in inches of water, the limit is 8.5 cfm/W per inch of water, so at half an inch of water the limit is 17 cfm/W. At lower pressures, the limit increases and at higher pressures it decreases.
- For static pressure in Pa, the limit is 2120 cfm/W per Pascal. At 125 Pa, the limit is 17 cfm/W. This dependence of the cfm/W limit on pressure is illustrated graphically in Figure 4. Clearly, there are large potential benefits for low static pressure systems.

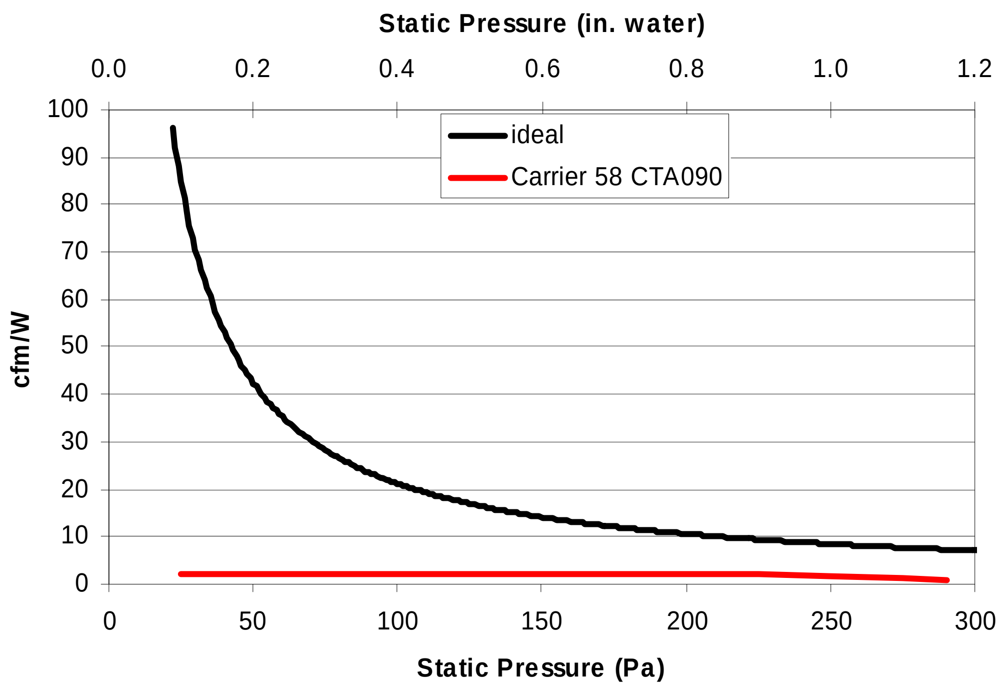


Figure 4. Maximum theoretical cfm/W for different static pressures compared to a typical residential unit

Previous Field and Laboratory Studies

Field studies by many researchers (see Field Testing Bibliography) have shown that existing fans in residential air handlers typically consume about 500W, supply about 2 cfm/W, and have efficiencies on the order of 10% to 15%. In particular, California homes showed a higher than average consumption of about 570W (Proctor and Parker 2000 and Wilcox 2004) and use 510W/1000 cfm or about 2 cfm/W. The results of a recently completed California Energy Commission field survey that focuses on new construction in California show similar results, with an average of about 700W per system and 2 cfm/W.

A Canada Mortgage and Housing Corporation (CMHC 1993) study showed that typical furnace fan efficiencies are on the order of 15%, but poor cabinet and duct design can reduce this to about 7%. The spread from best to worst systems was on the order of ten to one indicating that it is possible to have much better performance using existing technologies. Another Canadian study by Phillips (1998 & 1995) performed field tests on 71 houses and found air handler efficiencies in the range of 10-15%.

More recently, the Energy Center of Wisconsin (Pigg (2003) and Pigg and Talerico (2004)) tested 31 houses with new (less than three years old) furnaces during the heating season. Almost all the BPM furnaces used more electricity in these real installations than their DOE test procedure ratings suggest: with a median of 82% above rated values. This was attributed to the static pressures in these field installations being much higher than those used in rating procedures. The external static pressures used in test procedures are typically 0.20 or 0.23 inches of water (50 or 57.5 Pa) depending on the capacity (DOE Furnace Test procedure[4] and ARI, 2003). The measured field data showed a range of 0.24 to 1.9 inches of water (60 to 475 Pa) with an average of 0.5 inches of water (125 Pa) at the high fire rate.

Natural Resources Canada (Gusdorf et al. (2002 and 2003)) tested two side-by-side calibrated test houses to evaluate the change in energy for using a BPM rather than a PSC motor for continuous fan operation (as required in many Canadian houses). The air handlers used in the study showed PSC efficiencies in the range of 10 to 15% with BPM efficiencies of 17 to 18% over the range of flows used for heating and cooling. The biggest differences were for continuous operation, where the BPM was six times more efficient than the PSC by being able to operate at about half the flow rate of the PSC. The results of this study showed that, for a continuously operating fan in the heating season, there was a 74% reduction in electricity use from a BPM (26% of the whole-house electricity use). There is a corresponding increase in natural gas usage in the heating season of 14% to account for the reduction in waste heat from the electric motor. For cooling, the savings were 48% of fan energy and 21% of all air conditioner use.

Furnace Blowers Tested in the Current Study

[4] Code of Federal Regulations, Title 10, Part 430, Subpart B, Appendix N, Uniform Test Method for Measuring the Energy Consumption of Furnaces and Boilers.

Table 1 summarizes key characteristics of the furnaces and blowers tested in this study. A total of five combinations of furnaces and blowers were evaluated. Furnace #2 and #3 are the same furnace with different blowers. The current study combines measurements made by PG&E using a standard AMCA 210 (Laboratory Method of Testing Fans for Aerodynamic Rating Performance) type air handler test and LBNL Duct Laboratory tests. The furnaces were tested in a horizontal configuration. The burners did not operate during the test and no gas was connected. No cooling coils or filters were installed. The air flow was controlled by an auxiliary fan on the PG&E test apparatus and duct dampers in the LBNL Test Apparatus.

<table>
<tr><th colspan="3">Table 1. Blower and Furnace Characteristics</th></tr>
<tr><th>Furnace</th><th>Blower & Blower motor</th><th>Controls</th></tr>
<tr><td rowspan="2">Furnace # 1
80 kBtu/h
2.5 – 3 Ton AC</td><td>Forward curved blades (10x7 Blower) with PSC motor (1/3 hp)</td><td>Speed taps on motor</td></tr>
<tr><td>Prototype backward inclined with prototype BPM motor</td><td>Software on laptop</td></tr>
<tr><td>Furnace # 2
88 kBtuh
3.5 ton AC</td><td>Forward curved blades (10x8 Blower Size) with PSC motor (1/3 hp)</td><td>Speed taps on motor</td></tr>
<tr><td rowspan="2">Furnace # 3
2-stage, 88 kBtuh
1.5- 4 Ton AC</td><td>Forward curved blades
(10x8 Blower)
with commercial BPM motor (½ hp)</td><td>Circuit board in furnace</td></tr>
<tr><td>Prototype backward inclined with prototype BPM motor</td><td>Software on laptop</td></tr>
</table>

LBNL Laboratory testing

All the tests were conducted using the Energy Performance of Buildings Group full-scale duct system and test chamber (Figure 5). 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 [30 L/s at 25 Pa]). The furnaces were placed on a stand outside the chamber and connected with insulated flexible ducting to the test chamber, as illustrated in Figure 6.

Figure 5. Exterior view of test chamber.

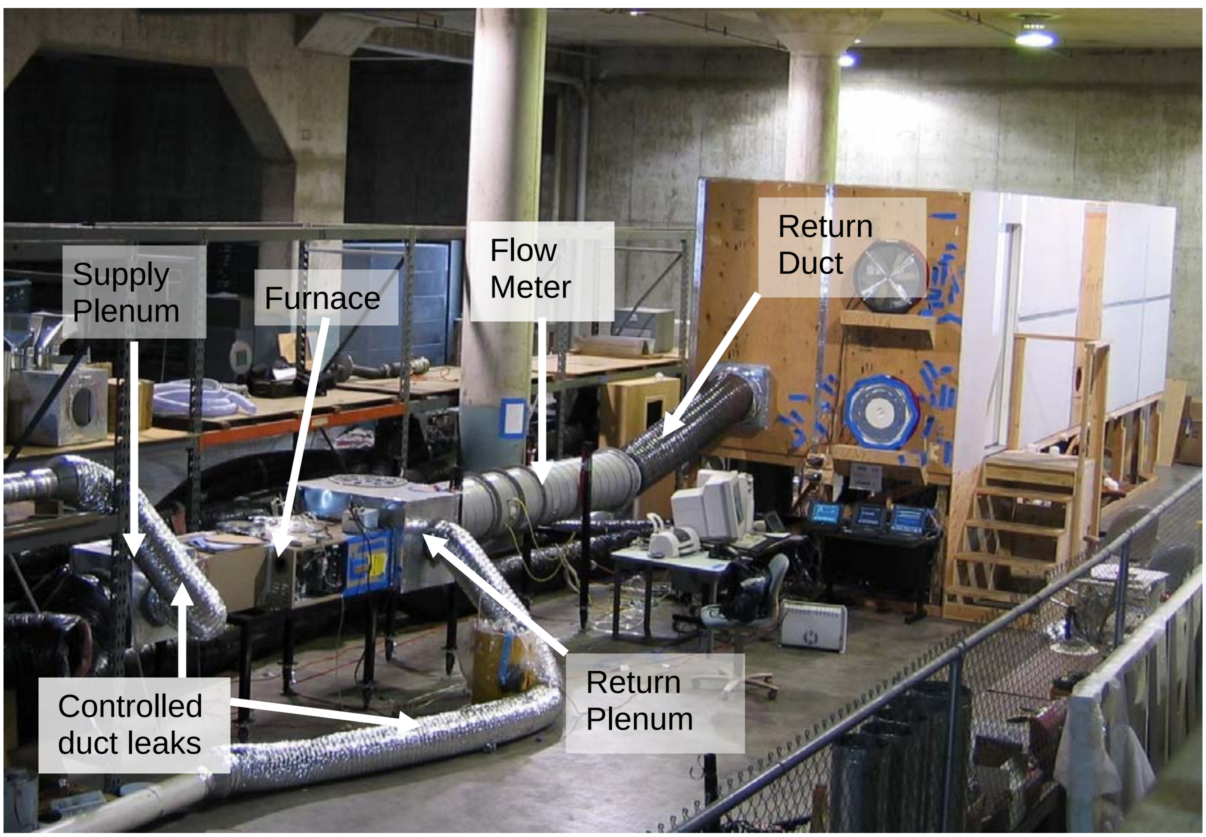


Figure 6. Furnace Blower Test Apparatus.

The duct system is comprised of the following components (in order of air flow from the chamber): return register, return flexible duct, large flow nozzle, return plenum, fan box, furnace, supply plenum, two main ducts, and ten supply registers. The supply ducts contain dampers that are used to modulate the air flow and pressures in the system. 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.

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 6). Fan inlet and exit pressures were measured upstream and downstream of the fan using electronic pressure sensors (±1.5% accuracy). 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. Downstream pressure measurements were taken between the fan outlet and the furnace heat exchanger (see Figure 7) and in the supply plenum. The fan outlet pressures were measured using a total pressure probe. The supply plenum pressures were determined by averaging together four static pressure probes in four corners of the plenum. The results given later use the supply plenum pressures.

A photo-optical sensor was used to measure fan rotational speed as shown in Figure 8. Fan electrical power use was measured with a true power meter (±1%), which also gives details of power factor and an harmonic analysis. All the data were recorded using five-second time averages after waiting for readings to stabilize for about one minute.

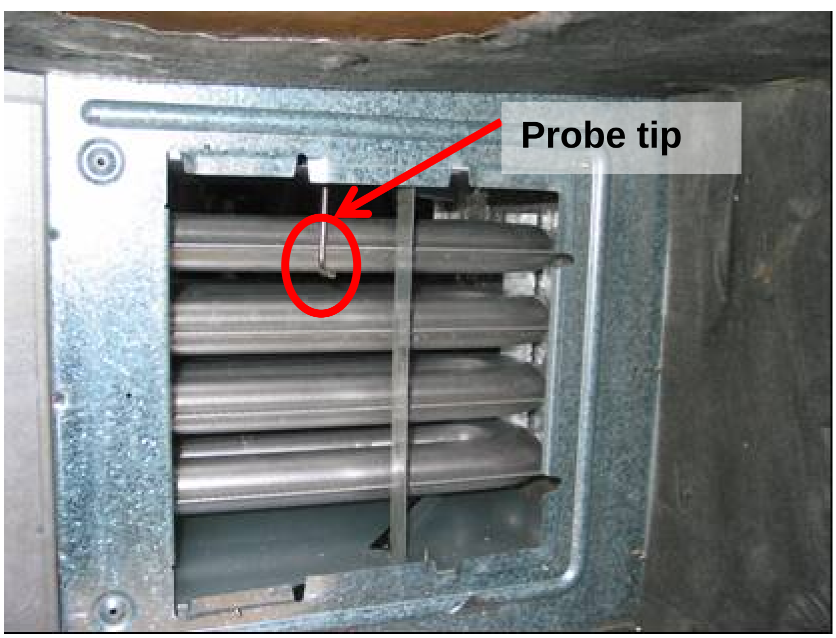


Figure 7. Entry to Furnace #1 heat exchanger showing pressure probe location.

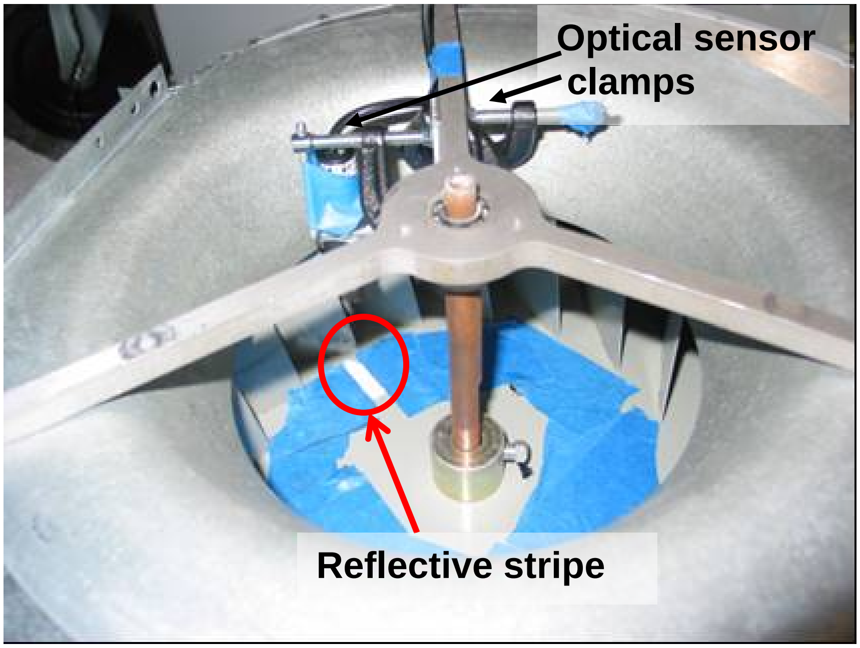


Figure 8. Rotational Speed Sensor Installation showing reflective stripe to generate pulse and photo sensor mounting

Air inlet size and location

The standard or baseline performance was determined with the air entering each furnace through the bottom of the furnace. Return duct design and furnace placement often mean that air enters through the sides of the cabinet or through multiple locations. In the laboratory testing, air was ducted from the return plenum to the side entry via flexible duct as shown in Figure 9. To investigate the effects of different air entries, the following tests were performed:

For Furnace #1, with the both the prototype and PSC motor the air entered through:

- Furnace bottom (Normal operation)
- One 14 inch diameter duct on the motor side of the cabinet
- One 14 inch diameter duct opposite the motor
- Both 14 inch ducts – one on each side
- Two 10 inch diameter ducts (one each side) plus the open bottom

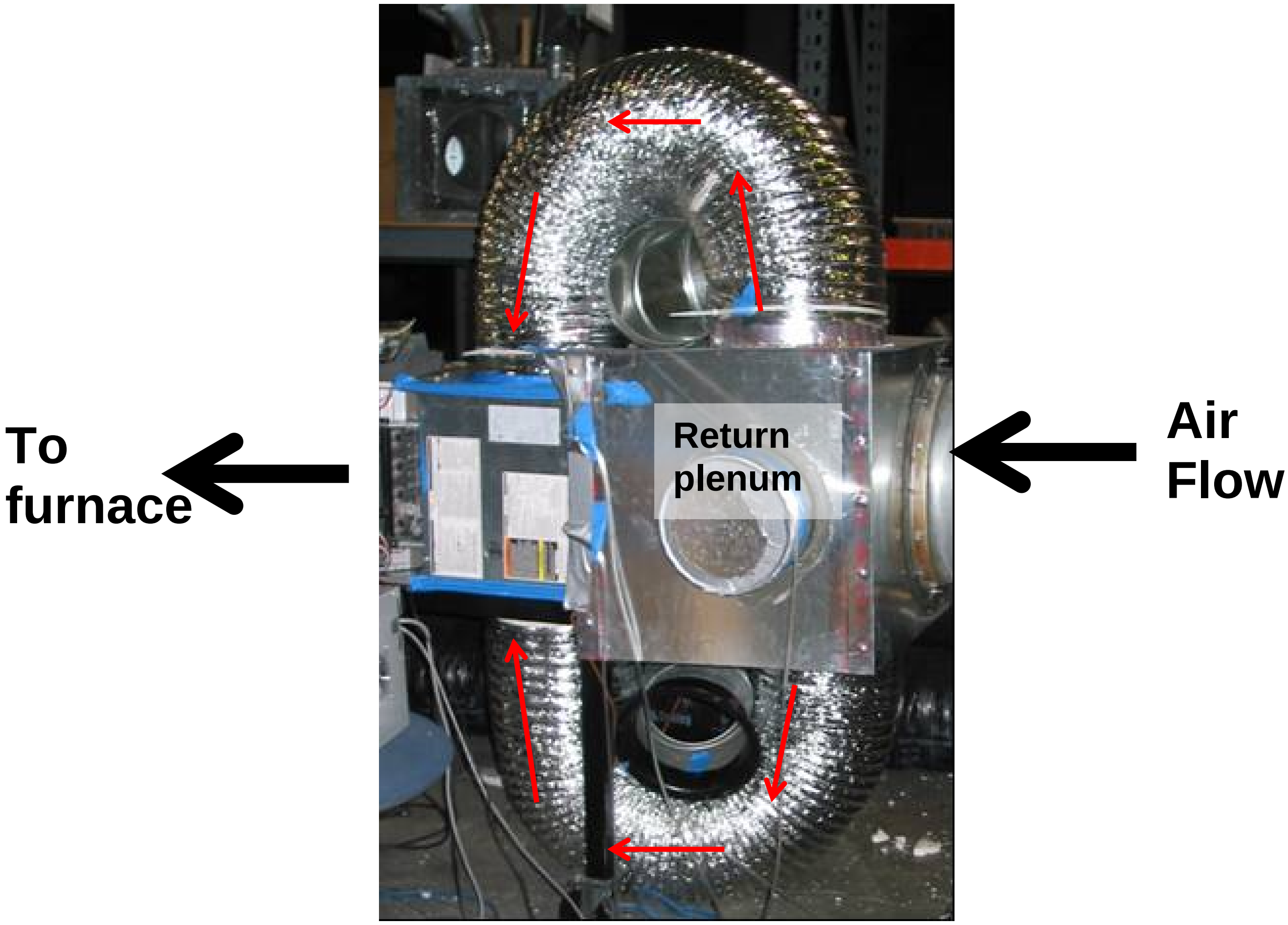


Figure 9. Furnace #3 with two 14 inch diameter side entry ducts. The furnace is mounted horizontally so the side entry is from the top and bottom in this photograph.

Figure 10. Fabricated Motor Side entry for Furnace #1

For furnace #3 BPM motor (at both 400 cfm/ton and 350 cfm/ton) air entered through:

- Furnace bottom (Normal operation)
- One 14 inch diameter duct on the motor side of the cabinet
- One 14 inch diameter duct opposite the motor
- Both 14 inch ducts – one on each side

For furnace #3 with prototype motor and blower air entered through:

- Furnace bottom (Normal operation)
- One 14 inch diameter duct on the motor side of the cabinet
- One 14 inch diameter duct opposite the motor
- Both 14 inch ducts – one on each side
- Both 14 inch ducts plus the open bottom

For furnace #2 (PSC motor) all the tests were performed in the normal configuration.

The side entry openings were fabricated by cutting holes of the appropriate diameter and inserting mounting flanges as illustrated in Figure 10. For the side entry only conditions the normal bottom opening was blocked from the inside.

Restrictive Cabinets

The effect of cabinet restrictions was evaluated by inserting rigid materials (either wall board or rigid insulation foam) against the walls of the cabinets that face the blower openings. Several thicknesses of materials were used. All the tests had the air entering the bottom of the cabinet (the normal configuration). The three furnace cabinets had the following clearances before any blocking:

- Furnace #1: two inches top and bottom (the furnaces were tested on their sides so the blower entries are at the top and bottom as viewed from the side)
- Furnace #2: three inches top and bottom
- Furnace #3: three inches top (motor side) and two inches bottom

The following blocking geometries were evaluated:

- 0.5 inch
- 1 inch
- 1.5 inch (for Furnace #2 only)
- For Furnace #3 the non-blocked clearance was asymmetric, with 3 inches on one side and 2 inches on the other. For this furnace we examined having different blocking on the two sides, with 1 inch an 1.5 inch blocking. This left gaps of 2 inches and 0.5 inches respectively.

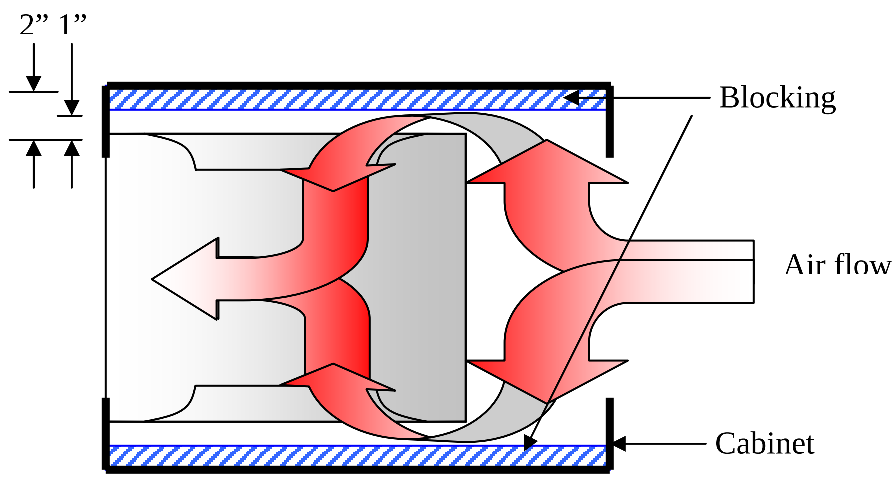


Figure 11. Illustration of blocking added to restrict fan inlet clearance.

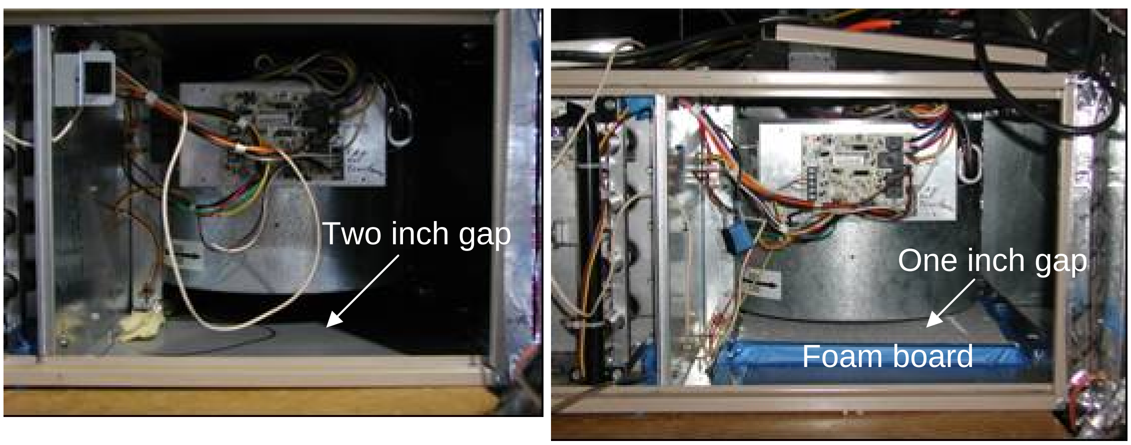


Figure 12. Standard fan in cabinet with no restriction (left) and with added foam board restriction (right).

PG&E Laboratory Tests

The tests performed at PG&E‘s San Ramon test facility used an apparatus similar to those illustrated in AMCA 210 (1999). The furnaces were mounted horizontally, with the bottom of the furnaces open to the room (Figure 13). Air exiting the furnace traveled through a duct system to an array of flow nozzles (Figure 14) that were used to measure the air flow rate. An auxiliary fan at the exit of the apparatus was used to control the air flows and system pressures. The use of this auxiliary fan allowed these tests to achieve lower pressure differences than the LBNL tests whose minimum air flow was set by the resistance of the duct system.

Static pressure differences were measured relative to the room from which the furnace drew air. The static pressure was measured downstream of the furnace exit using a tubing ring connecting four duct taps. Thus, the utility test results are comparable to the LBNL test results using the plenum pressures (rather than the pressures measured between the fan outlet before the furnace heat exchanger). Other information, such as air temperatures, barometric pressure, motor power consumption and rotational speed were recorded together with the air flows and pressures.

The same five furnace/blower combinations as used in the LBNL study were tested. All the tests were with air entering through the bottom of the furnace with no attached return plenum.

Figure 13. Furnace mounted in electric utility test apparatus

Figure 14. Flow nozzles in electric utility test apparatus

Results

LBNL Laboratory Results

Pressure Performance and Power Consumption

For furnace #1 the test results in Figure 15 show that the prototype with its BPM is better at maintaining air flow as pressure differences increase. The PSC motor data show a distinct pressure performance limit of about 300 Pa (1.2 in. water), whereas the prototype can operate at up to 500 Pa. The prototype shows a gradual decrease in flow with increased pressure difference due to being operated in a constant torque mode.

Usually BPM blowers operate to maintain a constant flow (see Figure 16 for furnaces 2 and 3). This maintenance of air flow at higher pressures is reflected in the power consumption data that shows a gradual increase with pressure for the prototype, but a reduction with increasing pressure for the PSC. The power consumption results for the prototype with the BPM shows how low flow operation can be done with very low power consumption (less than 100W) compared to the PSC.

Because high cfm/W ratings and low power consumption can be obtained for a BPM motor by operating at low air flows (and correspondingly low system pressures) any performance specification should include a minimum air flow rate. This is because air conditioner (and to a much lesser extent furnace) performance decreases as air flow is lowered.

In addition, flow reductions for PSC blowers at high operating pressures could result in lower air conditioner performance that needs to be avoided. The peak efficiency for the PSC occurs at about 250 Pa (1 in. water) for all the speed settings. For the prototype, the pressure corresponding to peak efficiency increases at higher speeds/torque settings, and is generally about double the PSC efficiency.

For furnace #1, the PSC shows little pressure-related variation in cfm/W, and is close to 2 cfm/W for all fan speeds until about 200 Pa (0.8 in. water). Above this pressure the cfm/W declines rapidly. For the prototype, the cfm/W rating is close to 10 at low speeds and pressures, and declines to about 2 cfm/W (same as the low pressure difference PSC values) as the pressure increases to 300 Pa.

The results for Furnace #2 in Figure 17 show that the two cfm/ton settings for the BPM result in very similar results, with the 400 cfm/ton setting providing higher air flow. For Furnaces 2 and 3 the PSC results are similar to Furnace #1, with cfm/W ratings between 2 and 2.5 up to 200 Pa (0.8 in. water) then declining as pressures increase. There is a large (30%) reduction in air flow at typical operating pressure differences of 200 Pa (0.8 in. water). The BPM results show almost constant flow up to about 300 Pa (1.2 in. water) at higher speeds. At 200 Pa (0.8 in. water) the BPM operates at about 3 cfm/W, increasing at lower pressures to about 7 cfm/W at typical rating conditions (about 50 Pa (0.2 in. water)). At pressures greater than 300 Pa (1.2 in. water) the BPM drops below 2 cfm/W.

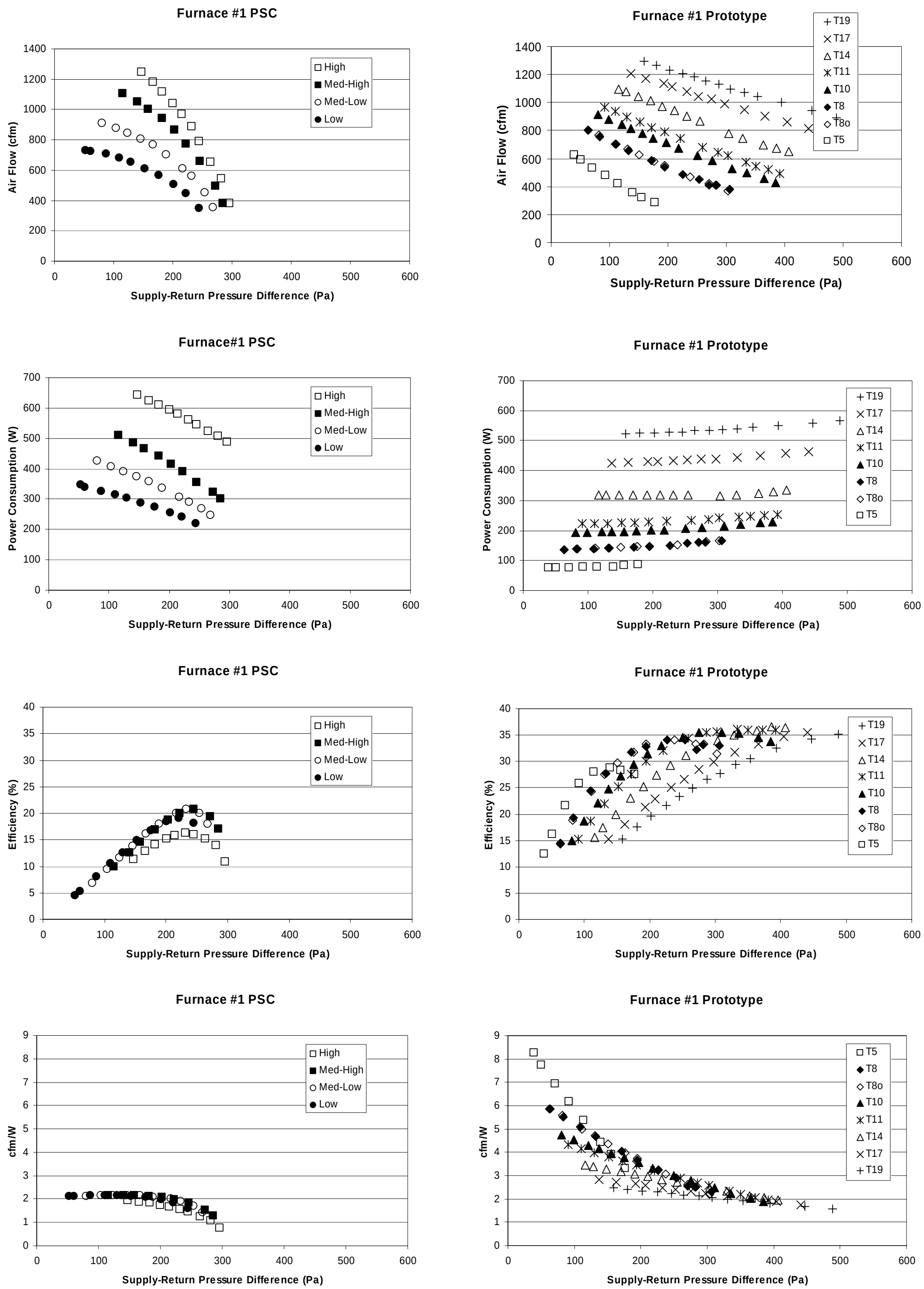


Figure 15. Furnace #1 test results. T numbers are torque settings (5 low to 19 high).

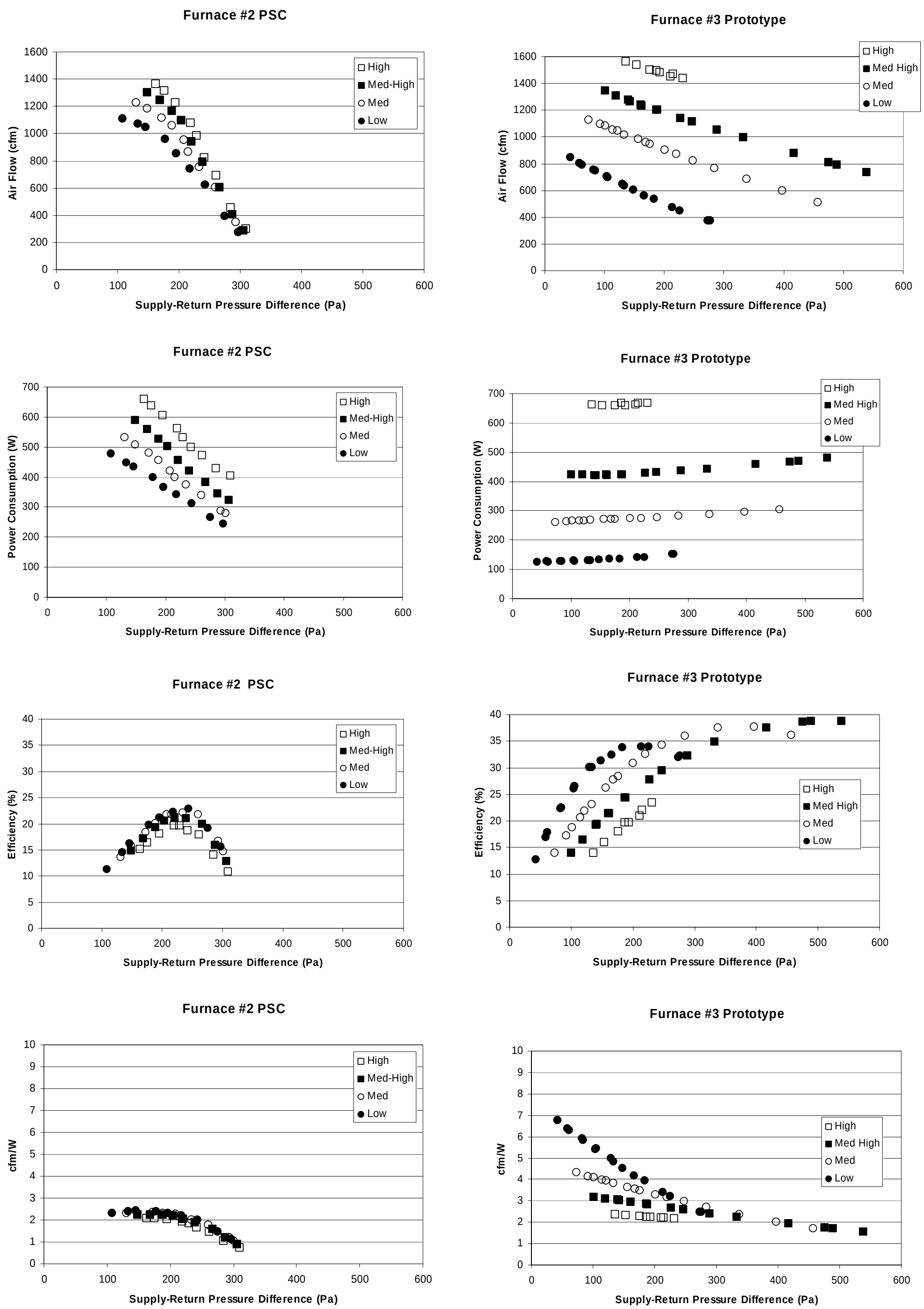


Figure 16. Furnace #2 and #3 test results.

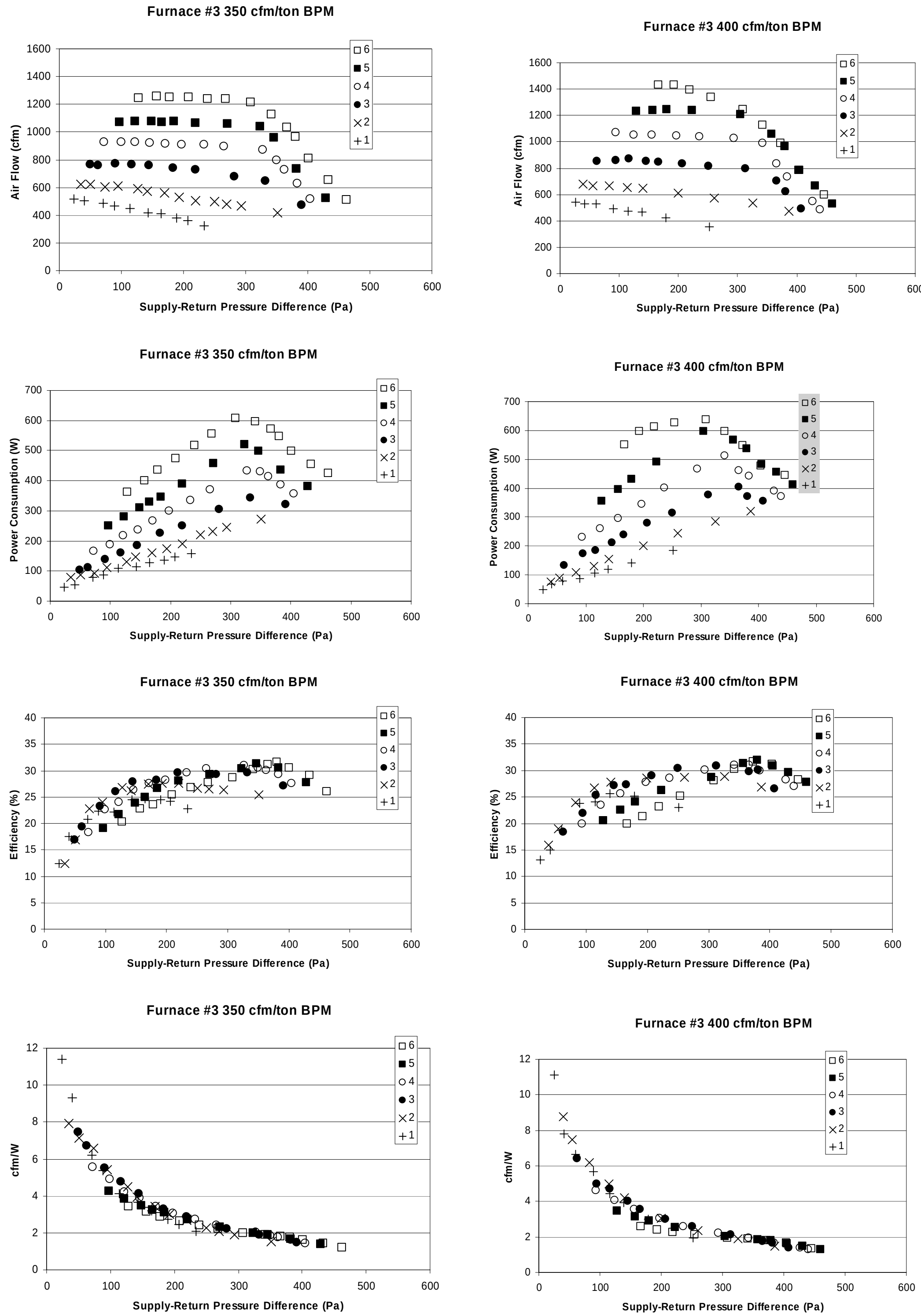


Figure 17. Furnace #3 test results at two different cfm/ton ratings. T numbers are torque settings (5 high to 1 low).

Air Inlet Size and Location Effects

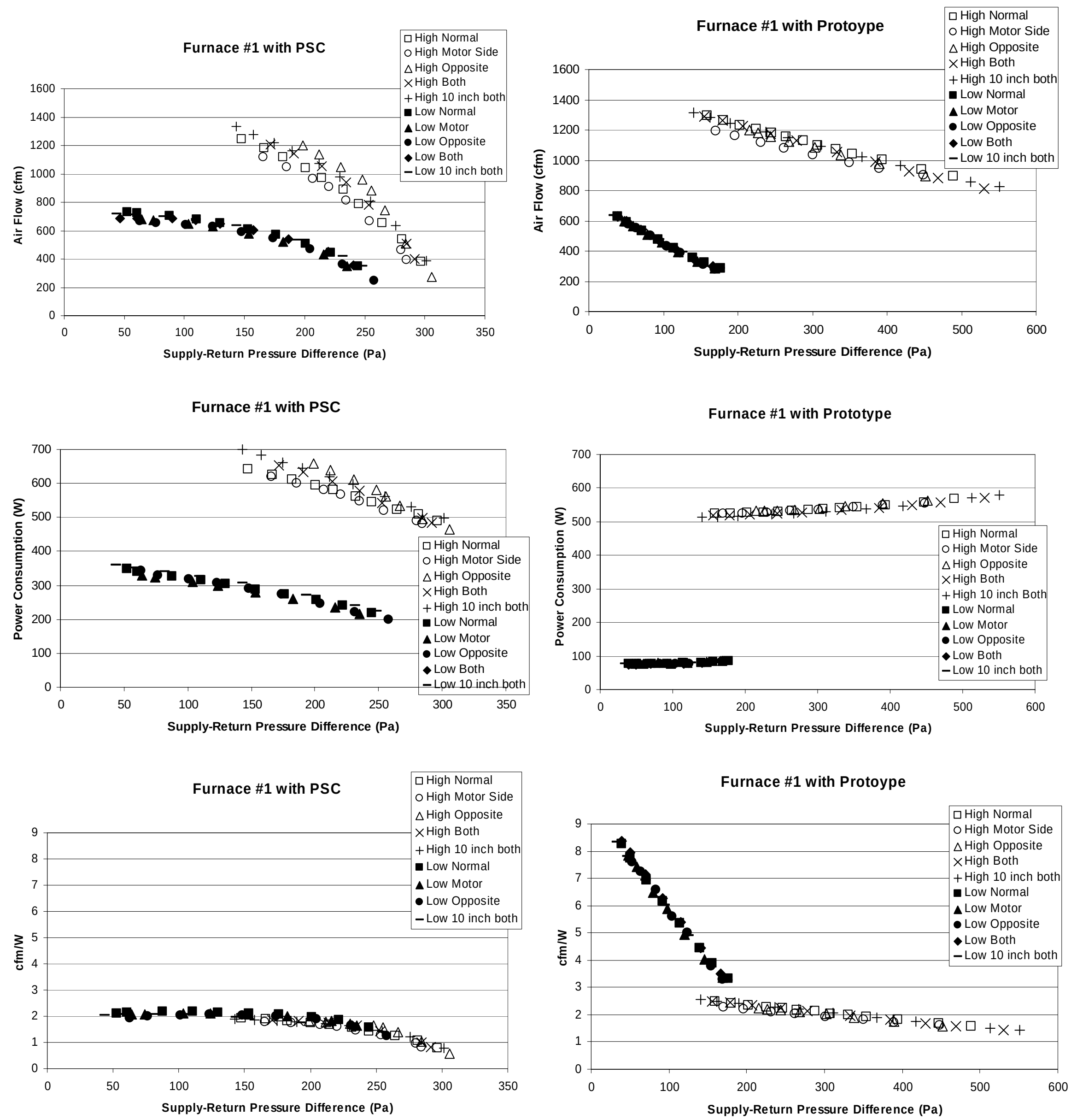


Figure 18. Effect of air inlet location for Furnace #1 with both PSC and prototype motors.

For Furnace #1, the results illustrated in Figure 18 show that the standard bottom entry gives the best performance, with the exception of the prototype with the open bottom and dual side entries. The air flow and cfm/W changes are on the order of 5 to 10 % for the different openings. The biggest effect on air flows is a reduction of about 10% for an opening on the motor side of the blower only. This is probably because the motor presents a significant blockage to air flow from this side.

The results in Figure 18 are for the two extremes of operation – highest and lowest speeds. The results for intermediate speeds fall between these two with no extreme anomalies. The effects are greatest at higher speeds. Figure 19 shows the high speed results only on an expanded scale to better illustrate the differences.

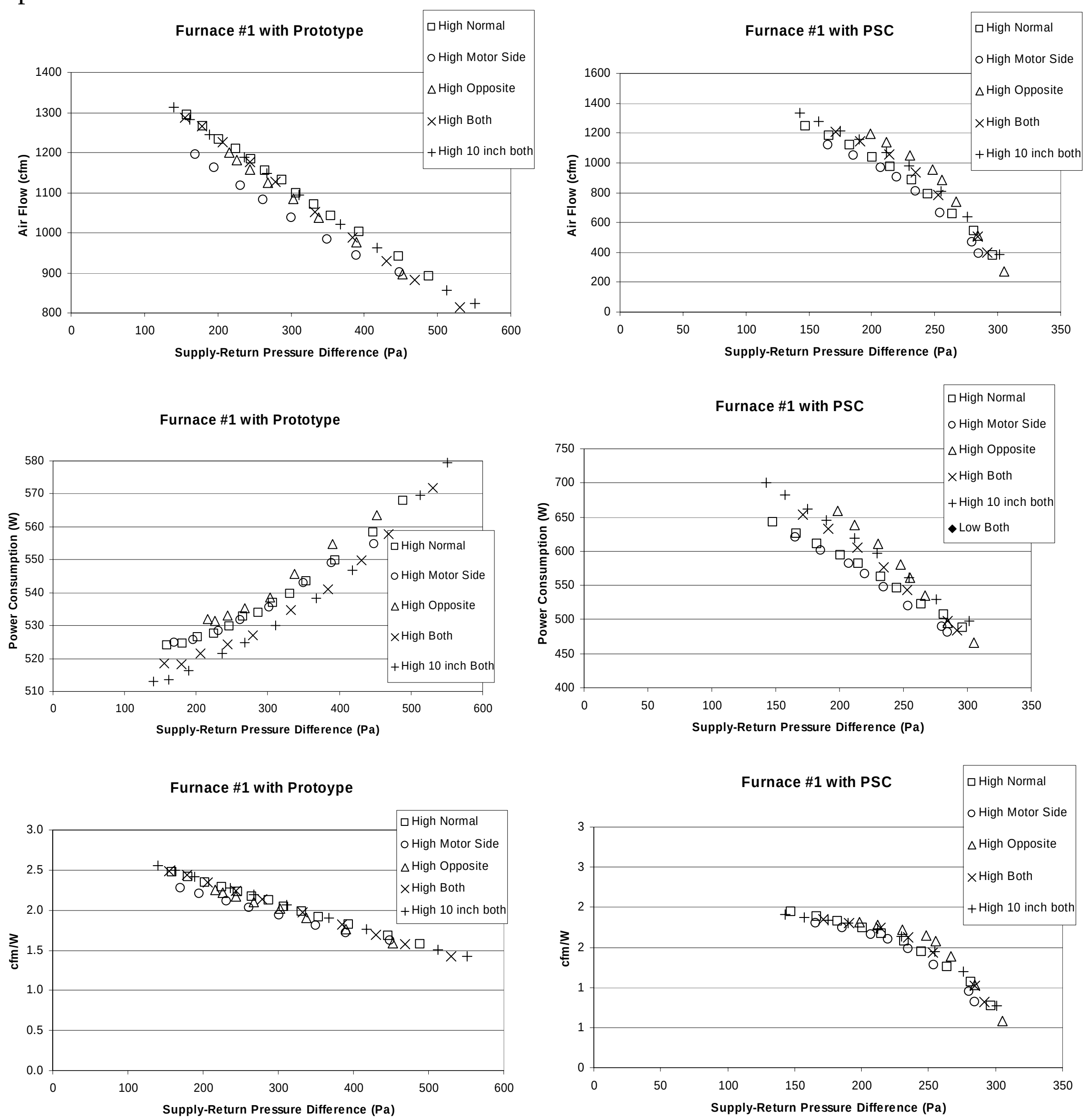


Figure 19. Effect of air inlet location for Furnace #1 with both PSC and prototype motors on high speed.

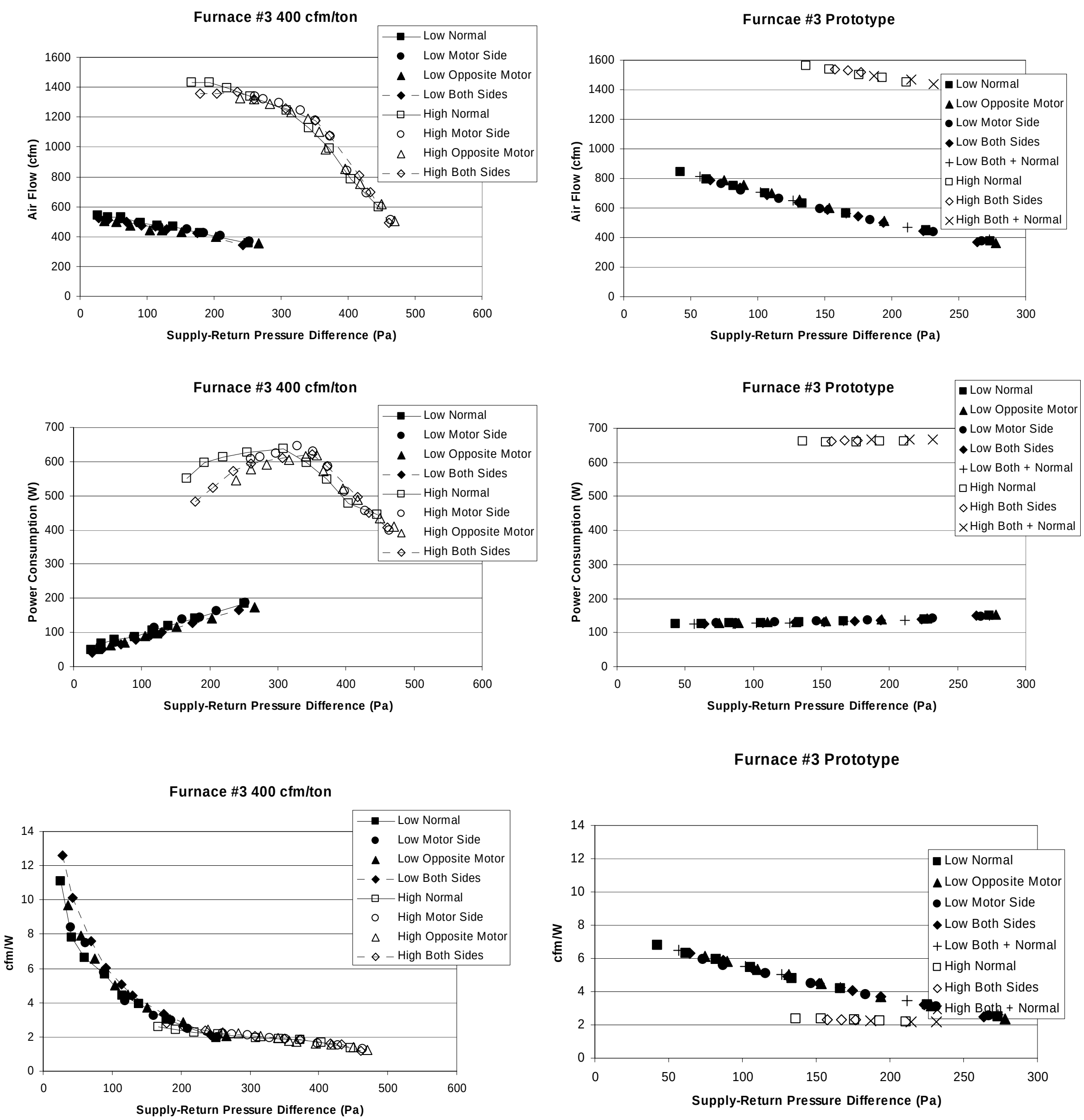


Figure 20. Effect of air inlet location for Furnace #3

The results in Figures 20-22 are for the two extremes of operation – highest and lowest speeds. The results for intermediate speeds fall between these two with no extreme anomalies.

For Furnace #3, the change from bottom entry to other geometries reduced the air flow and cfm/W ratings. The biggest change was for the most restrictive entry: motor side only, where the cfm/W rating dropped by 24% for the 400 cfm/ton BPM motor. The effects were less severe for the prototype motor.

These changes in cfm/W and air flow imply that an installation that was acceptable for obtaining proposed California Building Energy Code credits using bottom entry may not be acceptable if side entry is used. For the PSC results in Figure 21, the fractional changes in air flow are greater than the cfm/W ratings because the PSC uses proportionally less power as air flows decrease. This means that inlet location is more important for making a furnace blower meet air flow requirements than a cfm/W requirement.

Cabinet Restriction

For Furnace #1, Figure 22 shows how decreasing clearance from 2 in. (50 mm) to 1 in. (25 mm) reduces air flow by about 7% for the prototype and about 15% for the PSC, with the PSC showing greater sensitivity at higher speed settings. The corresponding changes in cfm/W of about 0.25 for both PSC and prototype at typical operating conditions would be critical for a system attempting to meet a performance specification of 2 cfm/W.

In Figure 23 the BPM shows less sensitivity to restriction, possibly due its control systems. Table 1 summarizes the results of the cabinet restriction testing.

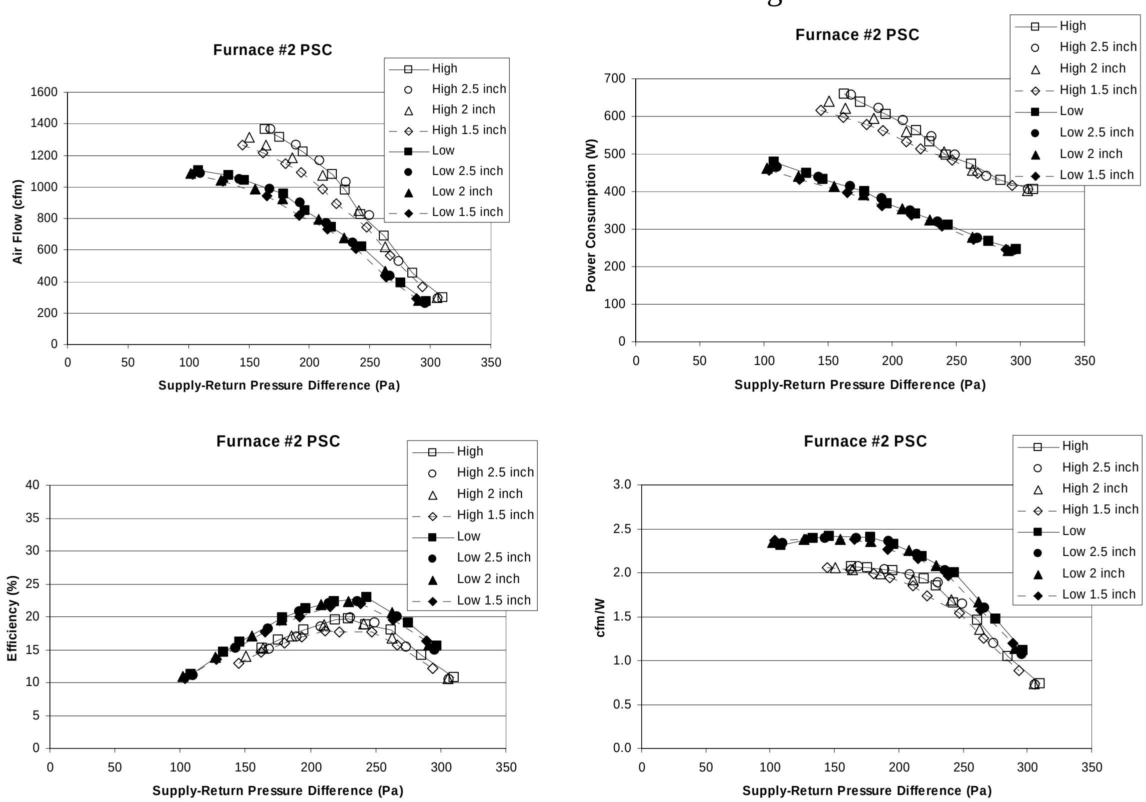


Figure 21. Furnace #2 sensitivity to cabinet restriction for high and low speeds with no restriction (about 3 inches clearance), 2.5 inch, 2 inch and 1.5 inch clearance between cabinet walls and fan housing.

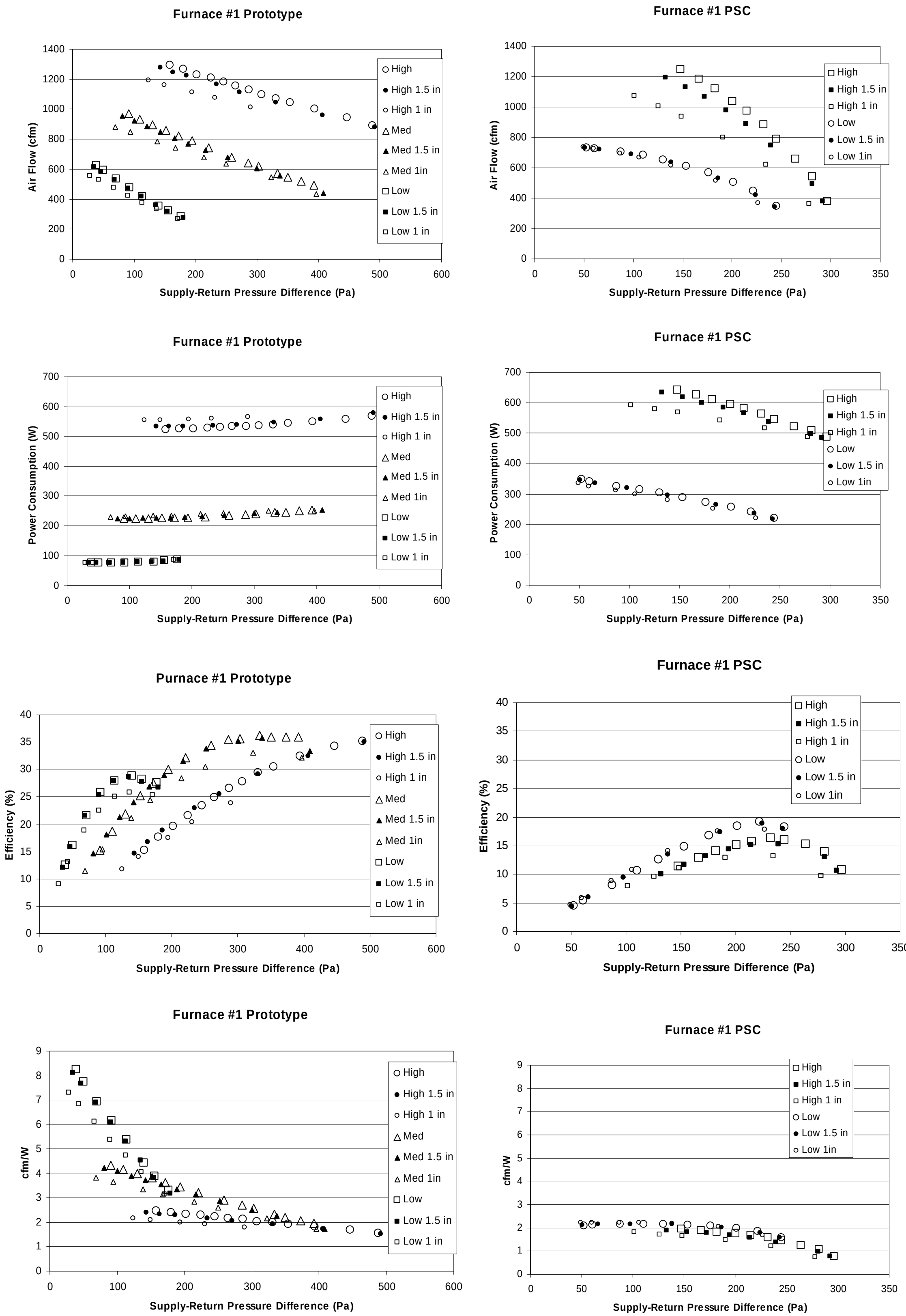


Figure 22. Furnace #1 sensitivity to cabinet restriction for high and low speeds with no restriction, 1.5 inch, and 1 inch clearance between cabinet walls and the fan housing.

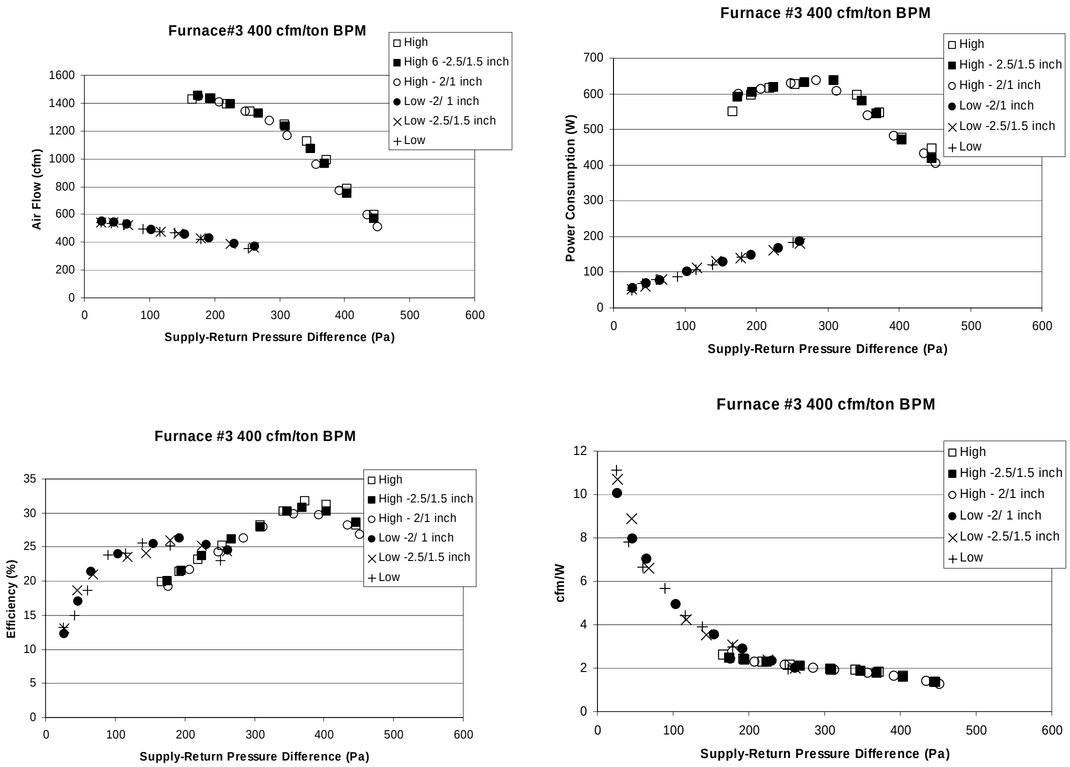


Figure 23. Furnace #3 sensitivity to cabinet restriction for high and low speeds with no restriction, 2.5/1.5 inch, 2/1 inch clearance between upper and lower cabinet walls respectively and the fan housing.

Table 1. Effects of cabinet restriction on furnace blower performance									
		Flow (cfm)	Flow (cfm)	Flow Change	Flow Change	cfm/W	cfm/W	Change cfm/W	Change cfm/W
Furnace	Speed	No Blockage	Max. Blockage	cfm	%	No Blockage	Max. Blockage		%
#2 PSC	High	1362	1268	-94	-7	2.1	2.1	0	0
	Low	1106	1078	-28	-3	2.3	2.4	0.1	4
#1 PSC	High	1246	1075	-171	-14	2.1	2.2	0.1	5
	Low	732	738	6	1	1.9	1.8	-0.1	-5
#1 Prototype	High	1294	1192	-102	-8	2.5	2.1	-0.4	-16
	Low	626	557	-69	-11	8.2	7.5	-0.7	-9
#3 BPM	High	1429	1444	15	1	2.6	2.4	-0.2	-8
	Low	542	548	6	1	11.1	10	-1.1	-10
#3 Prototype	High	1561	1516	-45	-3	2.4	2.3	-0.1	-4
	Low	805	774	-31	-4	6.4	6.2	-0.2	-3
Average					-4.6			-0.3	-4.6

Power factor

Table 2 summarizes the results of the power factor measurements. Power factors were recorded for all flow and pressure difference operating points, and Table 2 gives typical values.

In general, the Power Factor is greater than 0.85 for the PSC motors and ranges from 0.5 to 0.6 for the BPMs. For the two prototypes, the power factor did not change more than a few percent over the tested pressure and flow range. This is probably due to operating in a constant torque mode.

The BPM results are more complex and showed a gradual increase in power factor to a peak at about 250 to 350 Pa pressure difference then decreasing after this point. The PSC motors had power factors that typically decreased by about 5% to 10% as pressure difference increased. The biggest changes were for Furnace #2 with the PSC motor, where the power factor dropped from 0.94 at high flow and low pressure (1300 cfm and 160 Pa) to 0.78 at 299 cfm and 310 Pa. Figures 24-28 show how the BPMs generate large odd order harmonics due to the current being highly non-sinusoidal. For the PSC motor in Furnace #1, the PF ranged from 0.68 to 0.92 depending on the motor operating point and the pressure across the fan.

Two different pressure differences were used to investigate the effect of fan loading on electric motor performance and therefore power factor. Figure 24 shows an example of the power, voltage and current waveforms where the power factor is high (0.92). Figure 25 shows the results of an harmonic analysis that show that for this high power factor almost all the energy is in the first harmonic.

Table 2. Results of power factor measurements from LBNL testing	
Furnace #1 Prototype BPM	
Speed Setting[5]	Power Factor
Low	0.53
Medium	0.55
Medium High	0.59
High	0.63
Furnace #1 PSC motor	
Speed Setting	Power Factor
Low speed low plenum pressure	0.86
Low speed high plenum pressure	0.92
High speed low plenum pressure	0.83
High speed high plenum pressure	0.68
Furnace #3 Prototype BPM	
Speed Setting	Power Factor
Low	0.56
Medium	0.62
Furnace #3 Standard BPM	
Speed Setting	Power Factor
Low	0.53
Medium	0.56
High	0.62
Furnace #2 PSC motor	
Speed setting	Power Factor
High	0.95
Low	0.92

[5] For the prototype the control was more of a torque than speed setting

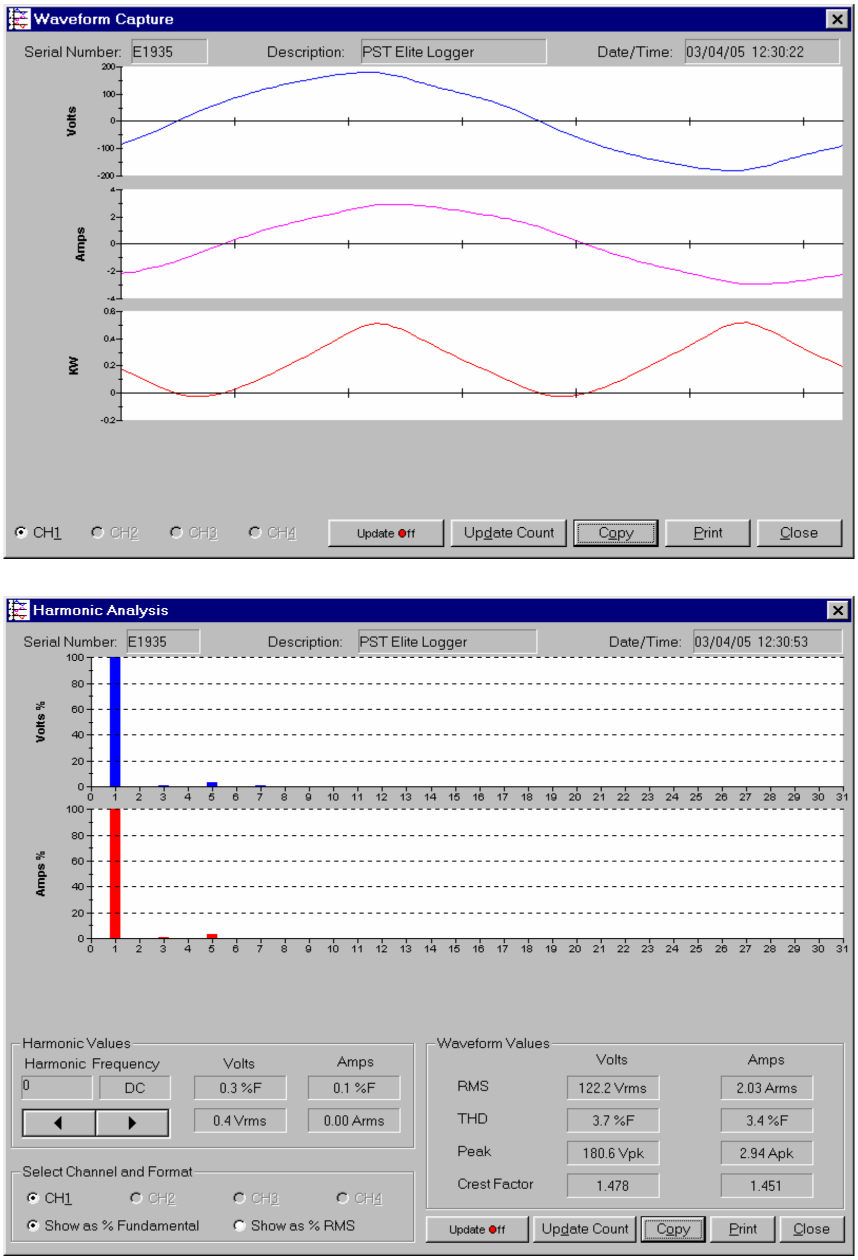


Figure 24. Wave form and Harmonic analysis for Furnace #1 PSC motor, low speed, high plenum pressures.

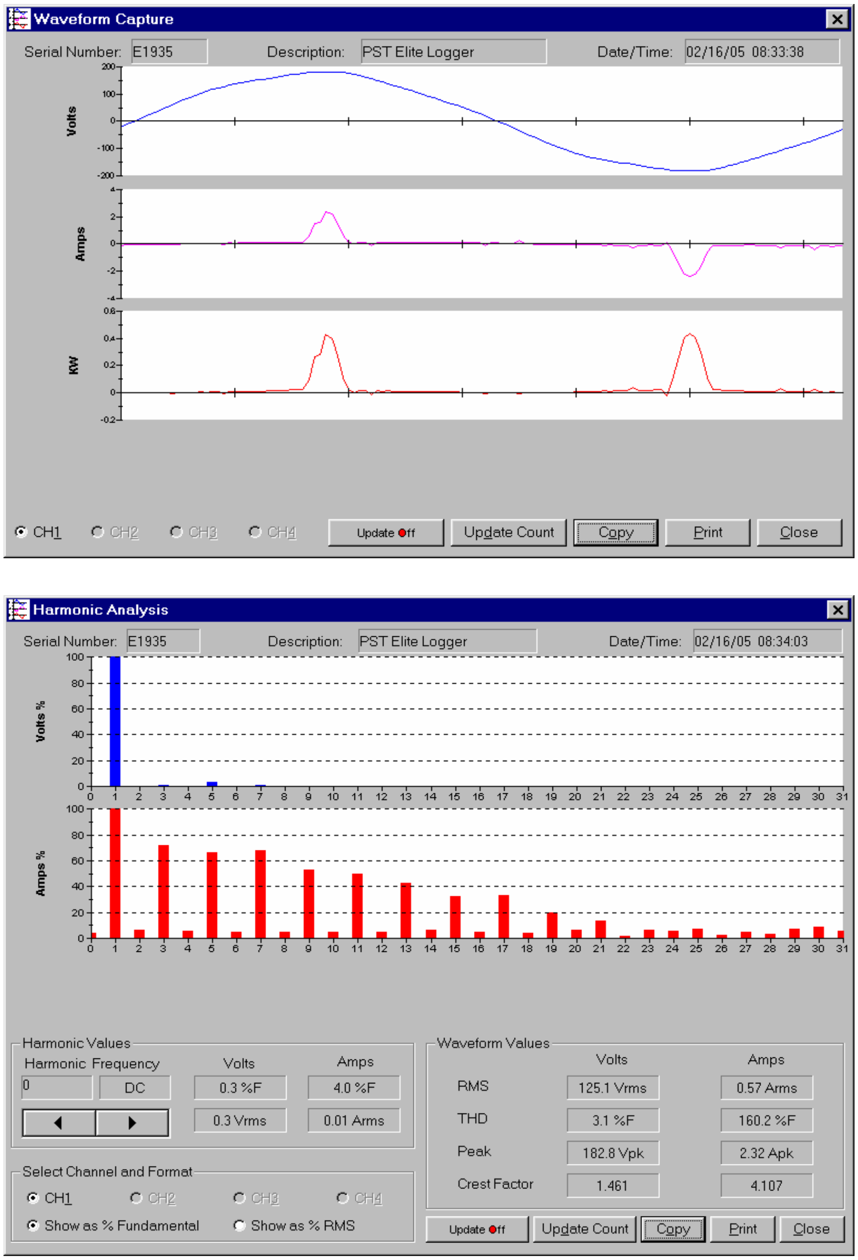
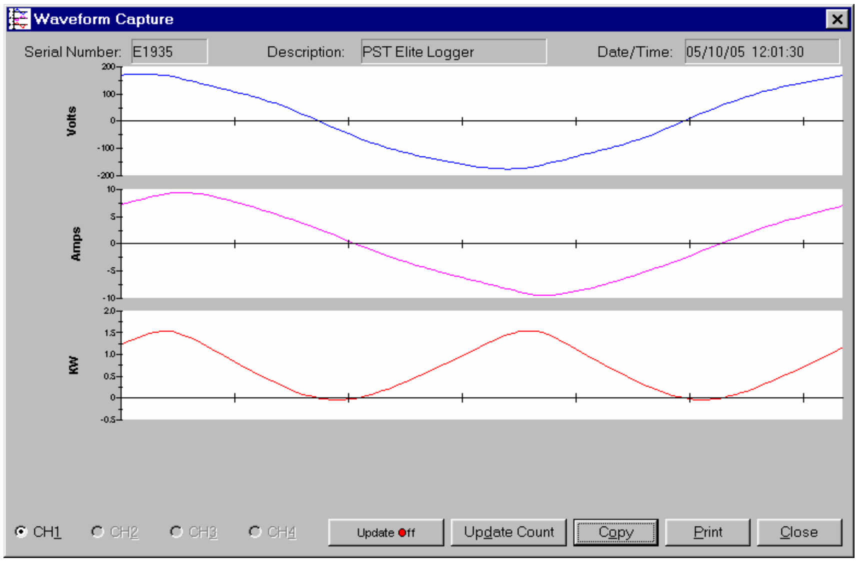


Figure 25. Wave form and harmonic analysis for Furnace #1 PSC Prototype BPM motor, torque setting 2.



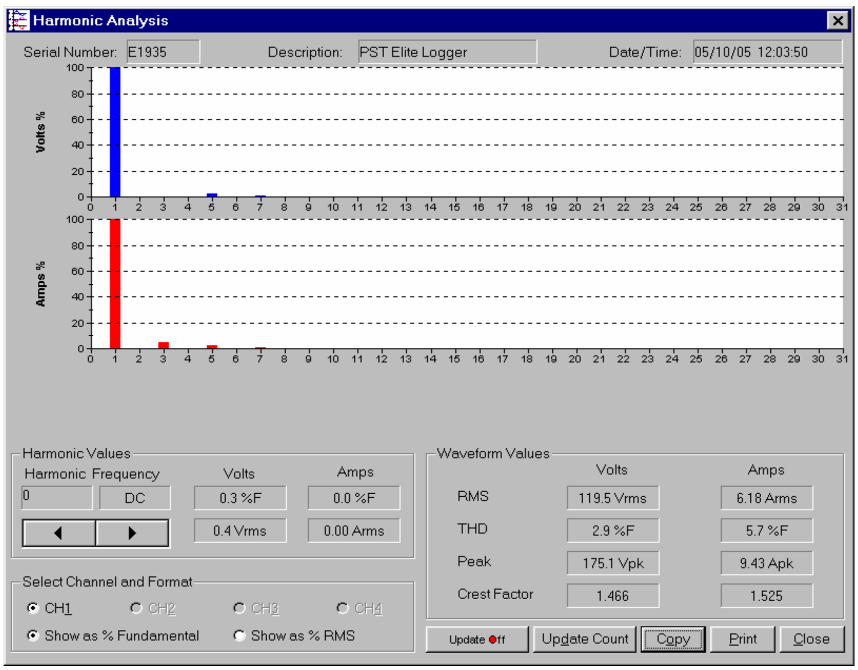
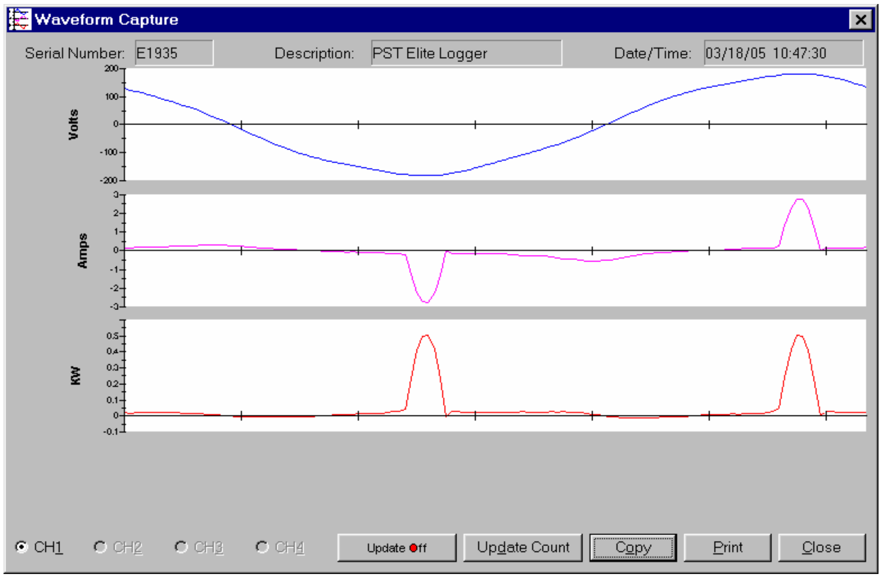


Figure 26. Wave form and harmonic analysis for Furnace #3 PSC motor, high speed setting.



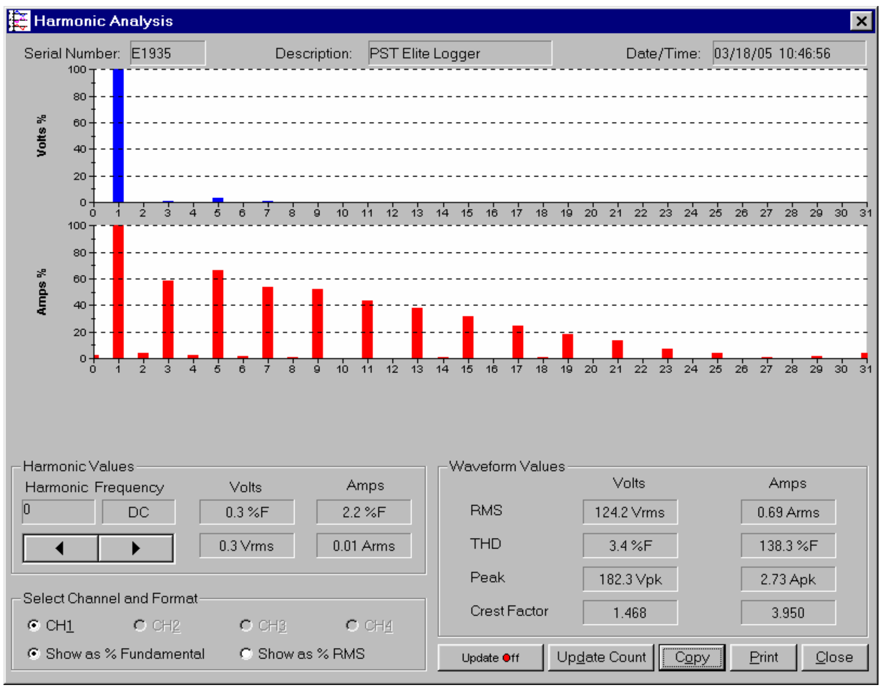
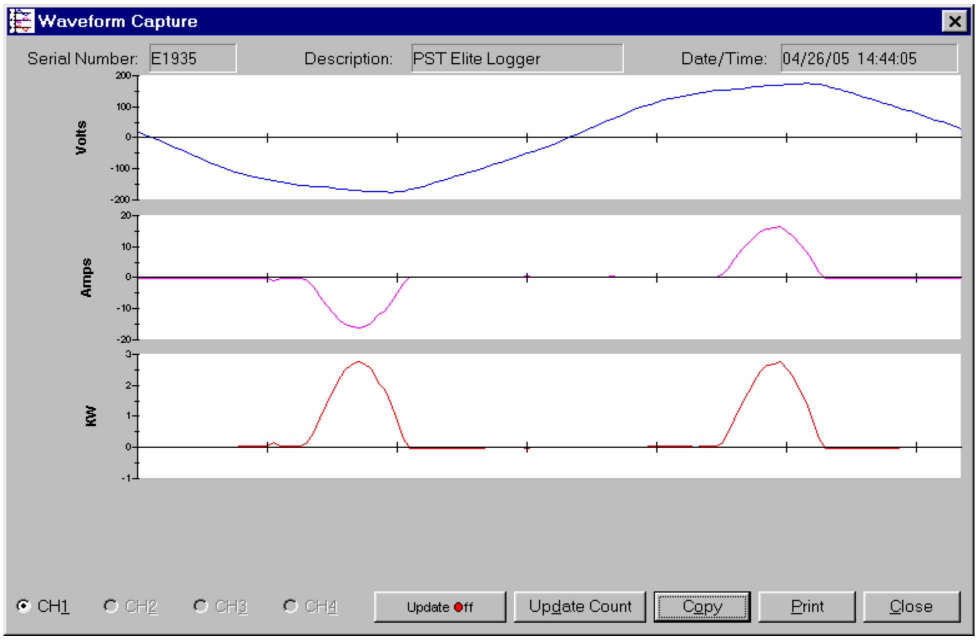


Figure 27. Wave form and harmonic analysis for Furnace #2 BPM motor, lowest speed setting.



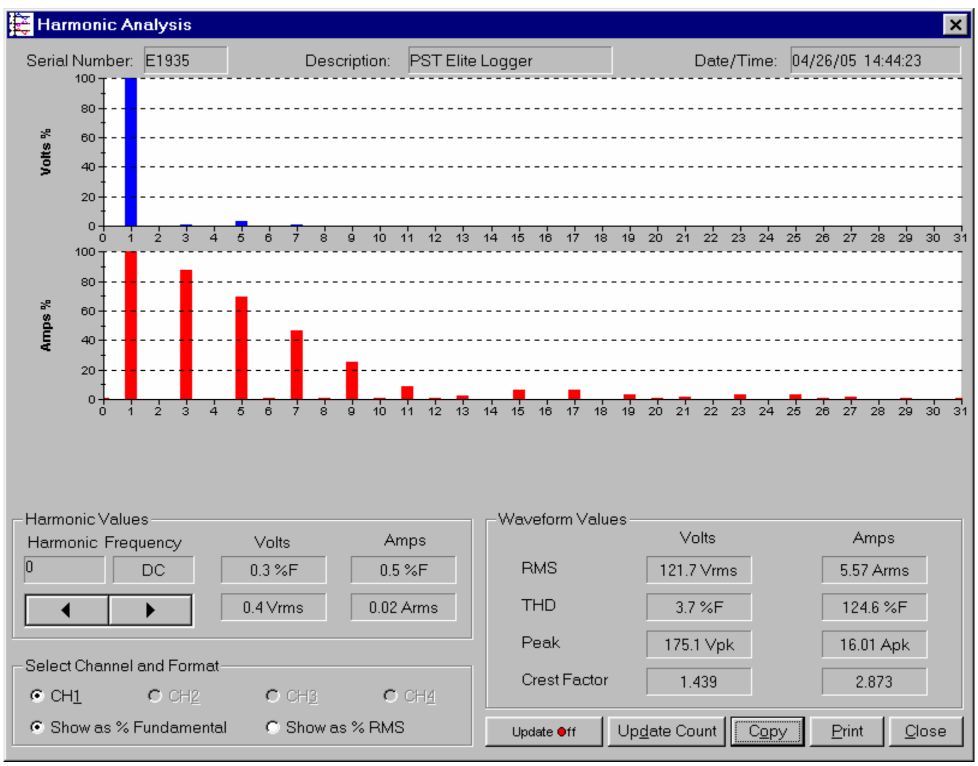


Figure 28. Wave form and harmonic analysis for Furnace #2 prototype BPM motor, torque setting 11.

PG&E Test Results

The tests in the PG&E test facility summarized in Figures 29-32 showed the same key characteristics for the different fan/motor combinations as for the LBNL testing:

- **BPM motors are better at maintaining airflow as static pressures increase.** This means that they are more tolerant to high-resistance duct installations in the sense that they are better able to maintain the air flow across heating and cooling heat exchangers so that they operate efficiently. The flip side of maintaining airflow into increasing pressure differences is the corresponding increase in power use. Therefore, there is a balance between maintaining heat exchanger effectiveness and the extra fan power requirements.

- **Peak overall (motor and aerodynamic) efficiencies vary according to prevailing operating pressures.** They are about 40% for the prototypes, about 30% for current production BPMs and about 25% for PSC motors. For the PSC motors, these peak efficiencies occur at static pressures of about 0.8 in. water (200 Pa). This is a fortunate coincidence as field data show that 0.8 in. water is the typical pressure difference seen by a residential air handler in California duct systems. At lower operating pressures of 0.5 in. water (125 Pa), the efficiencies drop to 12% to 15%. At typical rating points (about 0.2 in. water (100 Pa)) the efficiencies are even lower at 5% or less.

  For the current equipment BPM, the efficiency peak is spread out from about 0.5 in water (125 Pa) to greater than 1.0 in. water (250 Pa), with higher efficiencies at lower operating settings. At typical operating pressures there is a broad range from 15% to 30% efficiency depending on the operating setting. This shows that it is important for installers to choose the correct operating settings when installing these air handlers and that different settings may be appropriate depending on the specific installation. The prototype showed similar efficiency value as the current equipment BPM at normal operating conditions.

- **In terms of cfm/W, the PSC air handlers are fairly constant at about 2 to 2.5 cfm/W for all speeds, with slightly lower cfm/W rating for lower speeds.** The performance falls off sharply above about 1.0 inch of water static pressure (250 Pa). The BPM devices have significantly higher cfm/W ratings as pressures are reduced, approaching values greater than 15 cfm/W for low pressures, but at pressures above 0.8 in. water their cfm/W performance is similar to a PSC.

- The Furnace #3 results for 350cfm/ton and 400 cfm/ton tests show essentially the same results, just with lower operating points for the 350 cfm/ton case.

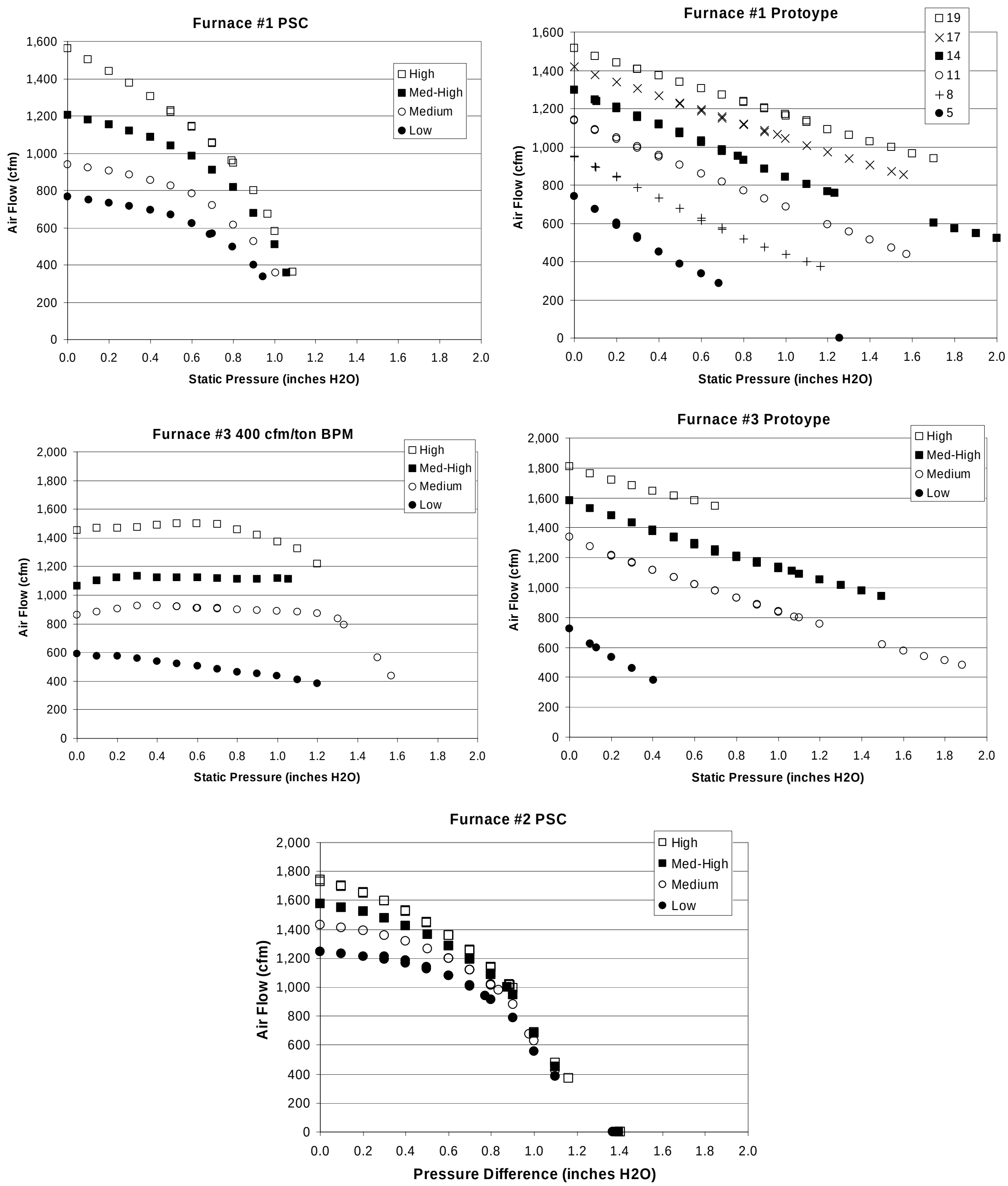


Figure 29. PG&E Testing results for furnace blower air flow

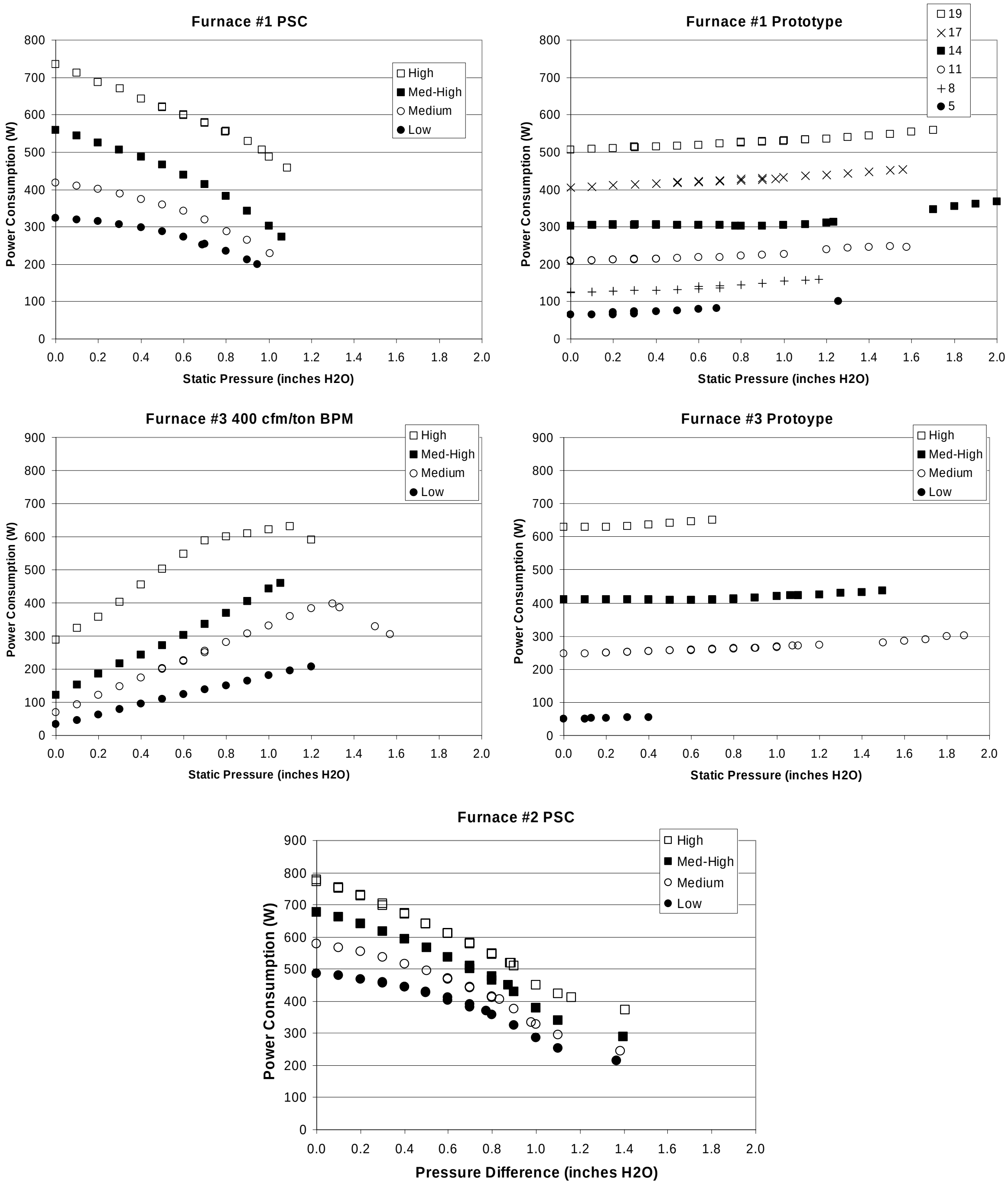


Figure 30. PG&E Testing results for furnace blower power consumption

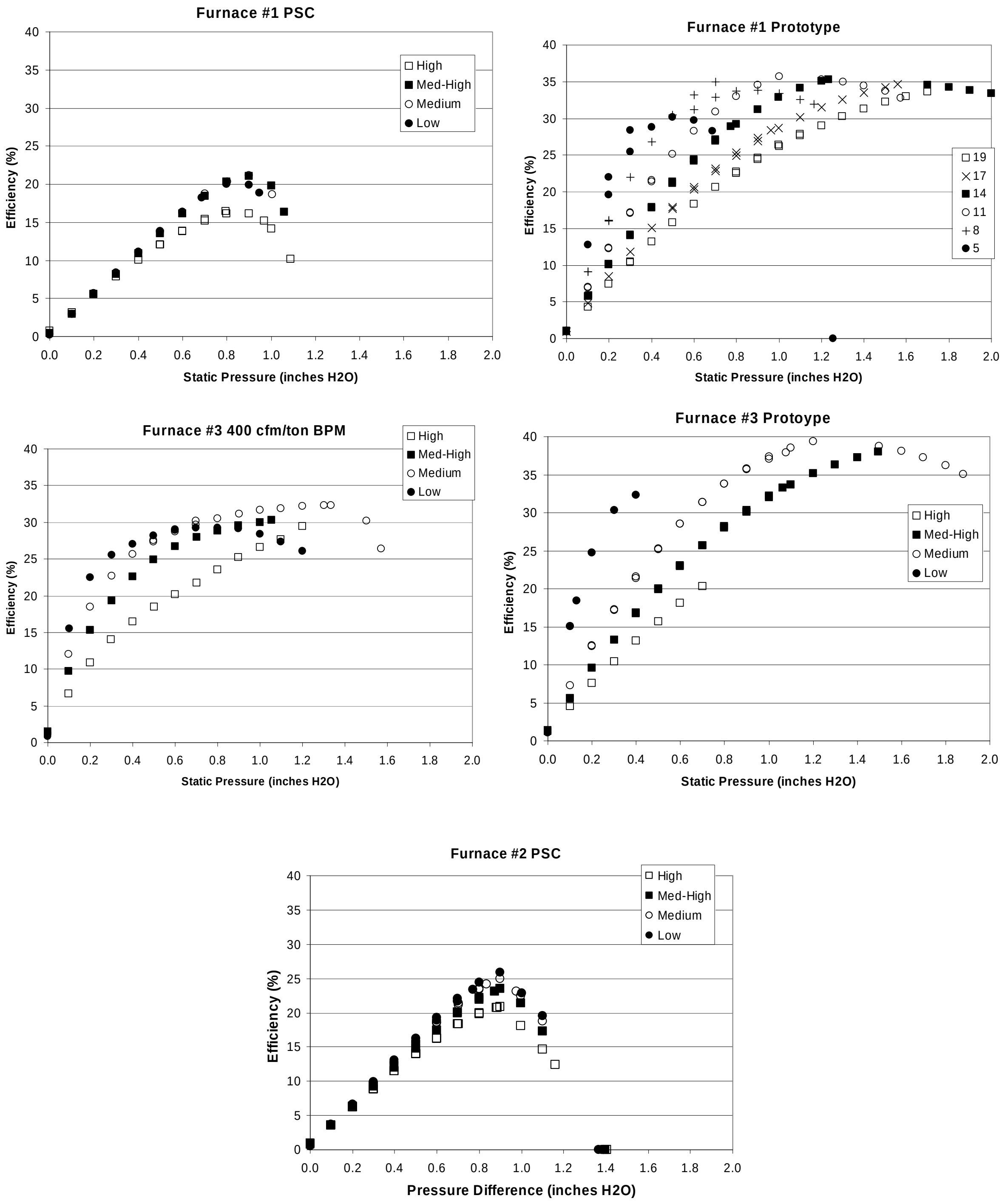


Figure 31. PG&E Testing results for furnace Blower efficiency

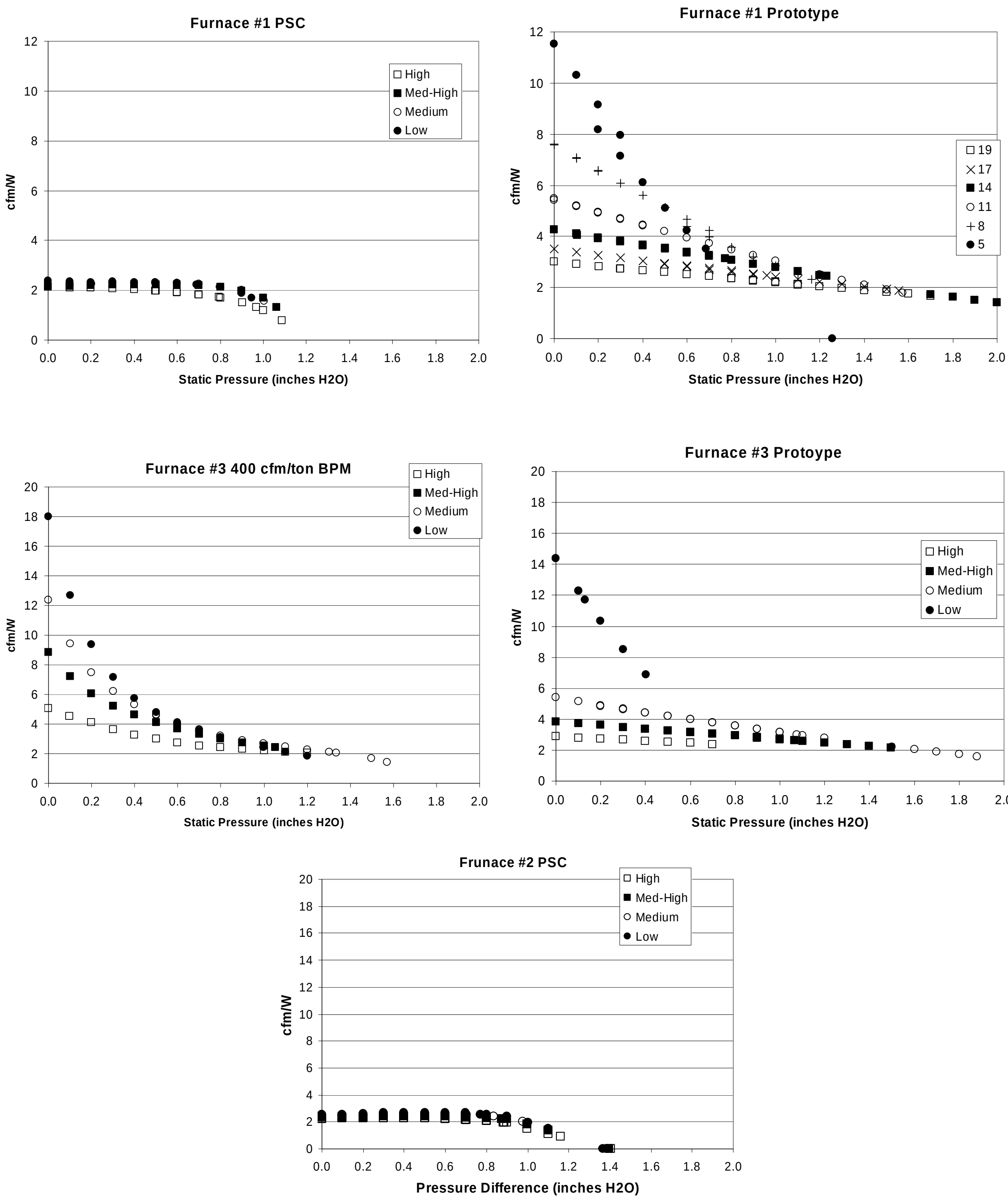


Figure 32. PG&E Testing results for furnace blower cfm/W ratings

Comparing Laboratory test data to Manufacturer's Published Data

Figure 33 compares the measured test results for Furnace #1 to data taken from the manufacturer's specifications for this furnace. At high speed, the manufacturer's rating overpredicts the air flow by about 100 cfm. For other speeds, it tends to underpredict by about 100 cfm, the exception being at medium-high speed where there is little difference above 0.5 in. water pressure difference.

Tests by other researchers (Proctor Engineering – Personal communication) have shown similar differences between laboratory test data and manufacturer's specifications: on the order of 10% of flow. This amount of variability is probably small enough that it would be acceptable to use manufacturer's data to determine energy use and flow for systems of known flow resistance.

In other words, when selecting a furnace, the manufacturer's data can be used with some confidence that the predicted energy savings will be realized. Of course, this conclusion is predicated on using the right system flow resistance and corresponding static pressure.

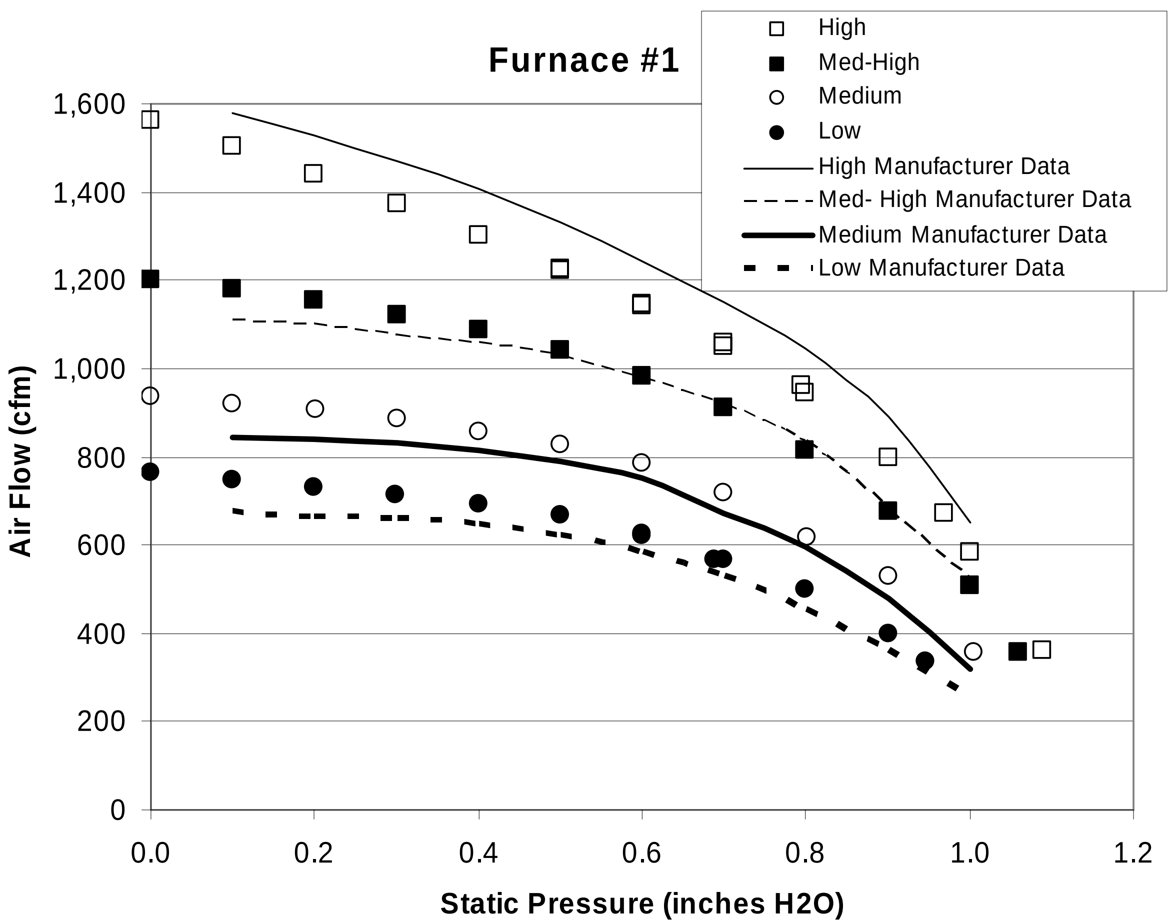


Figure 33. Comparing Laboratory Test Data to Manufacturer's Published Data

Comparing PSC and BPM performance

Because PSC and BPM motors have different performance characteristics, the relative performance depends on the system curve. Figure 34 illustrates three system curves. The one with the lowest flow resistance is that assumed in the DOE test procedure (0.23 in. water at 1200 cfm). The manufacturer's rating point of 0.5 in water at 1400 cfm gives the middle curve. The last curve is that found in field tests (0.8 in. water at 1200 cfm).

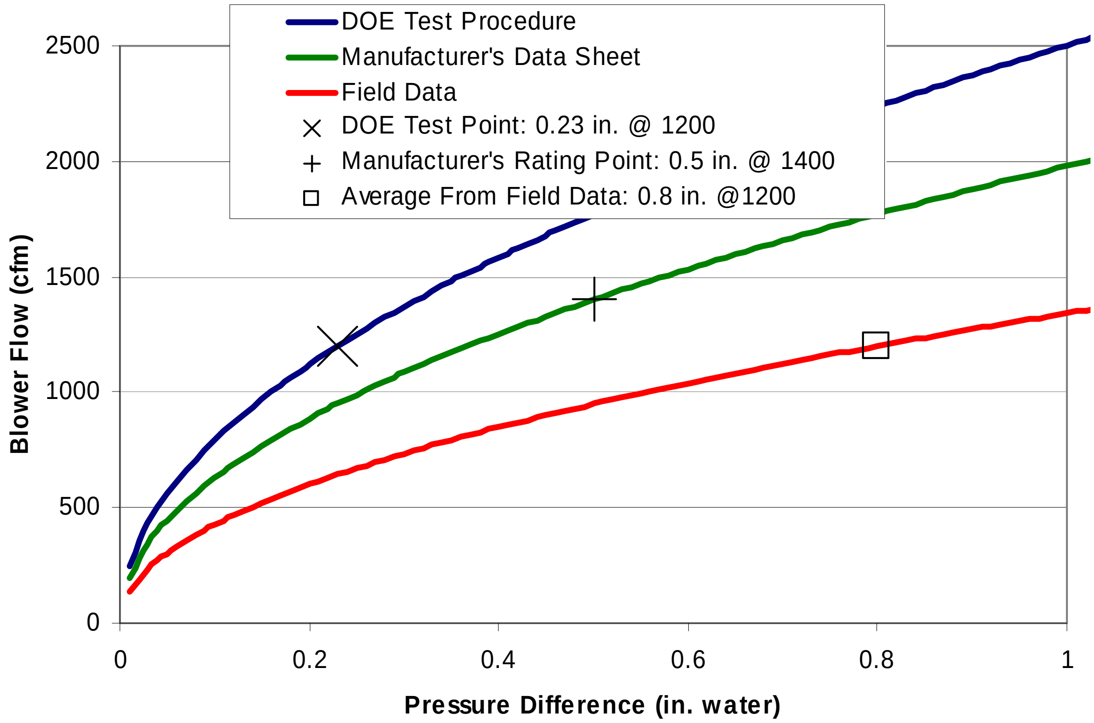


Figure 34. Possible system curves for a residential furnace blower.

Determining Operating points

By superimposing the PSC and BPM fan curves for Furnaces #2 and #3 (identical furnaces with PSC and BPM motors respectively), we get Figure 35. Figure 35 includes both heating and cooling blower speeds. From this figure, we can identify the operating points for the two blowers for heating and cooling.

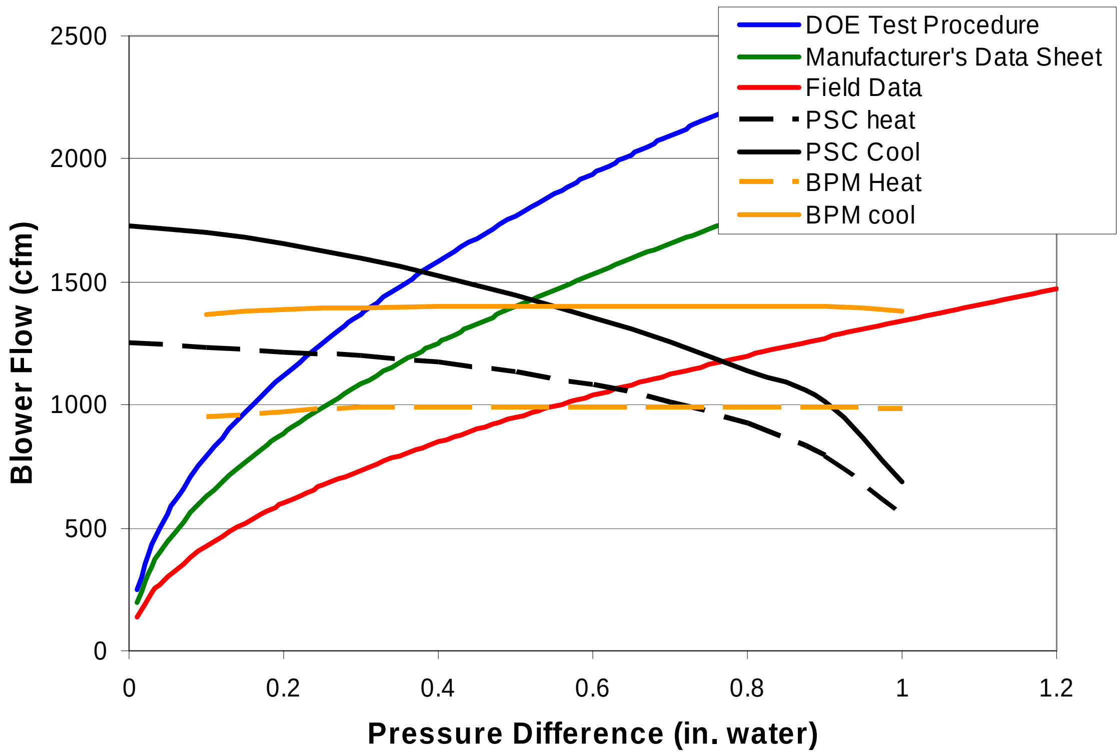


Figure 35. Superposition of blower performance and system curves.

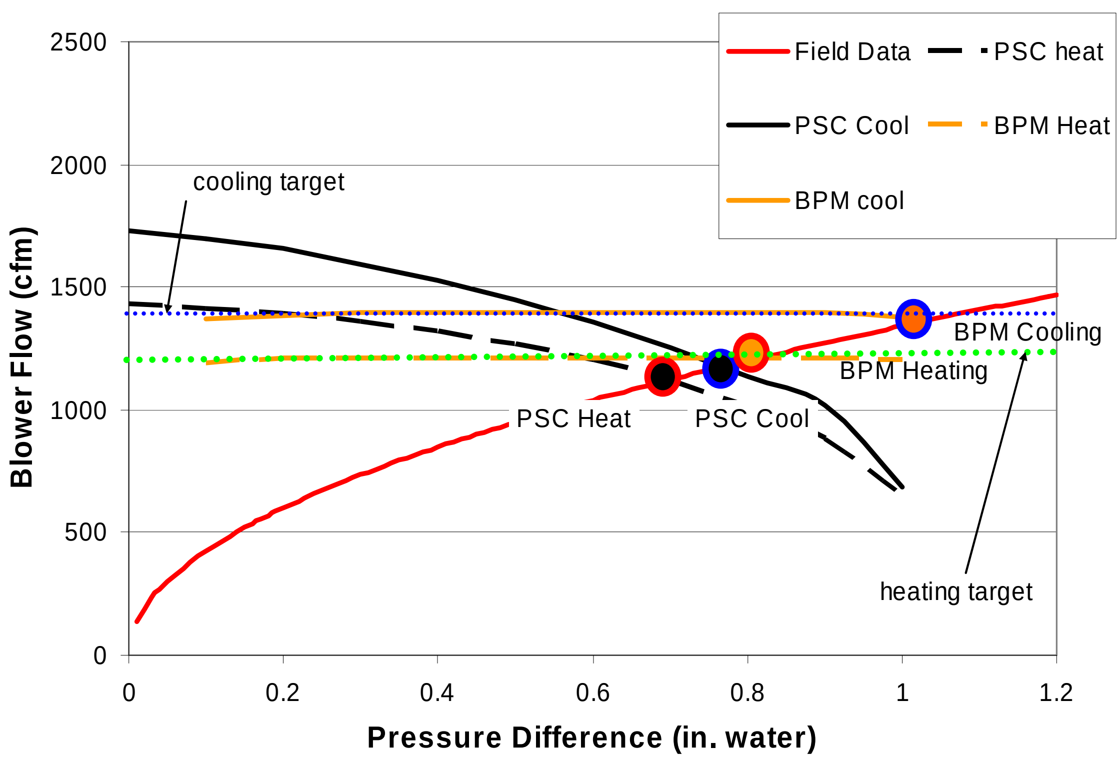


Figure 36. Identification of operating points for a system curve based on field data.

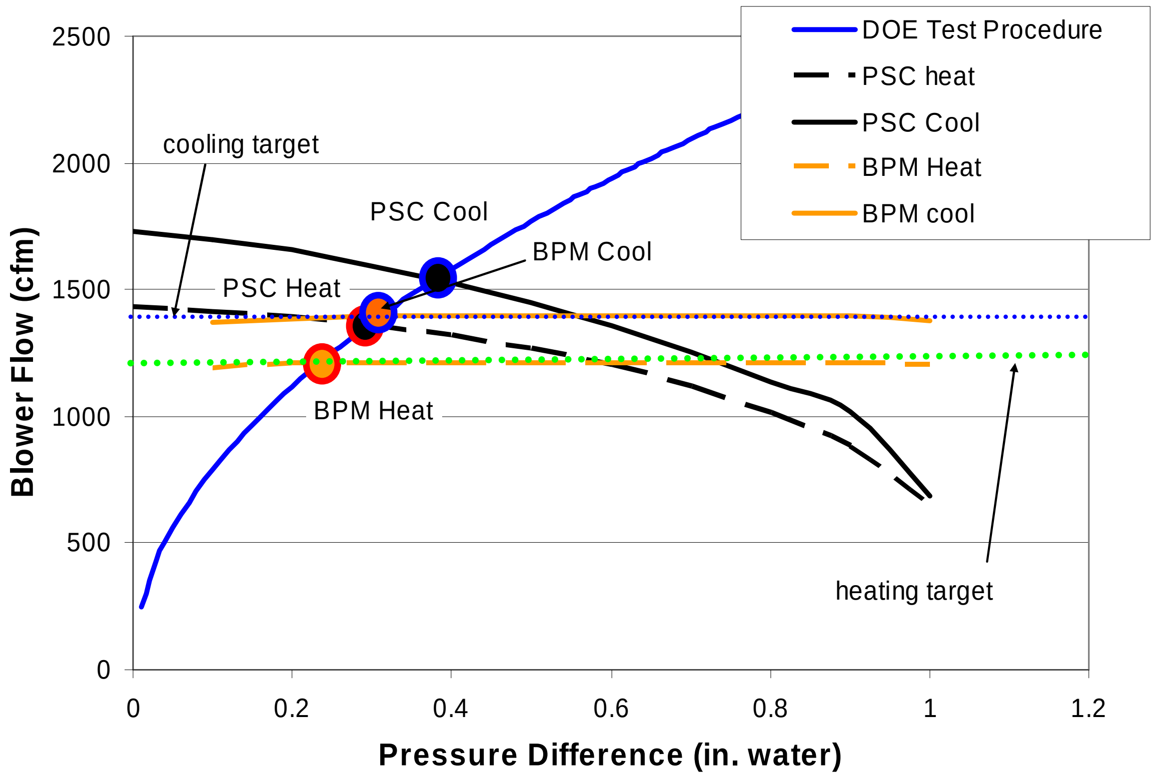


Figure 37. Identification of operating points for a system curve based on the DOE/AFUE test procedure system curve.

Figures 36 and 37 illustrate the flows that would occur for the two extremes of the DOE/AFUE and field measured system curves. Figures 38 and 39 show the corresponding power consumption for four cases looking at the two extremes of system curves: the DOE/AFUE test procedure and the median of field testing of actual systems.

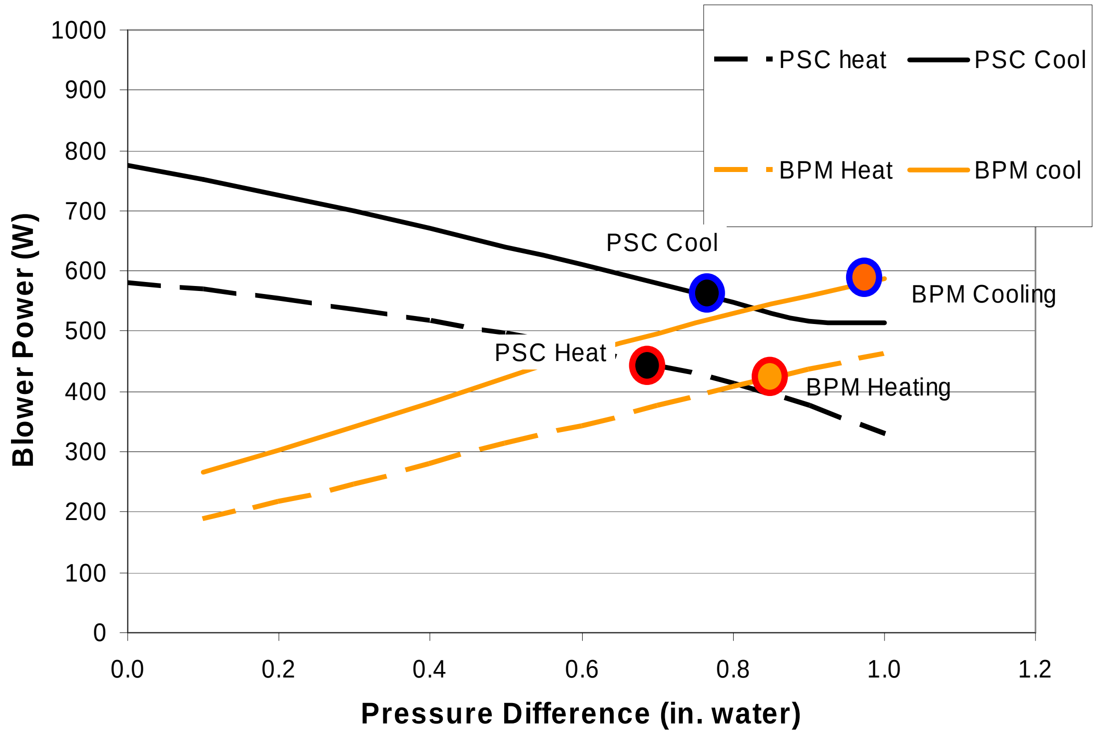


Figure 38. Blower power consumption for typical California ducts.

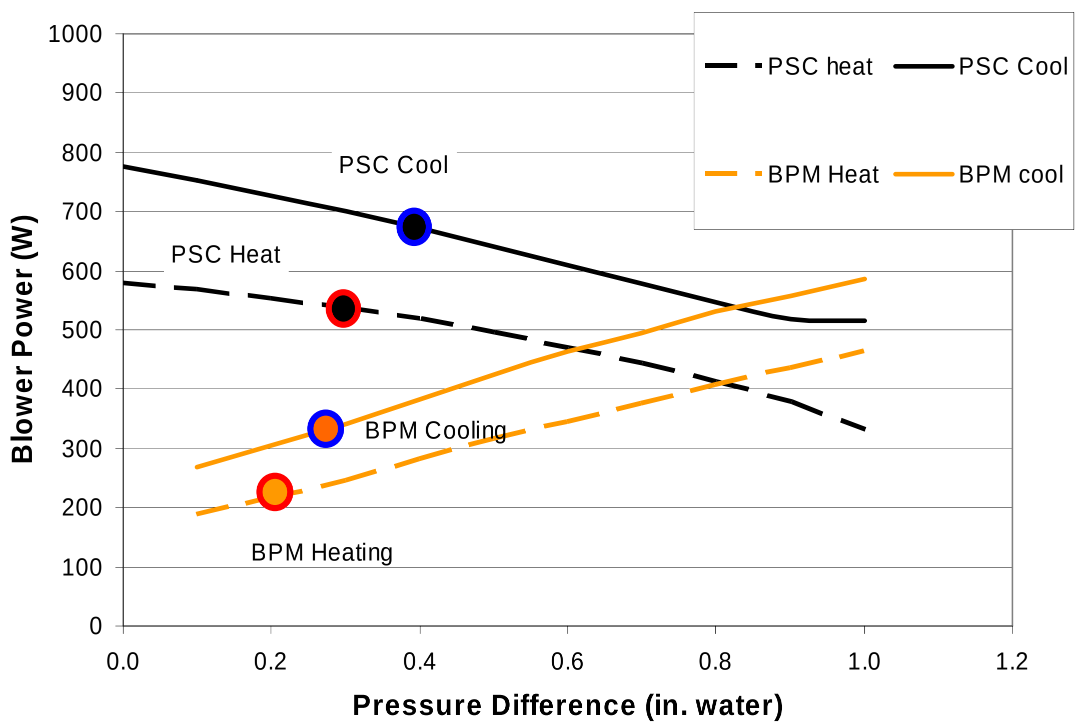


Figure 39. Blower power consumption using DOE test pressures.

Table 3. Watt-draw for three different design and three duct system options			
	DOE Test Procedure	Manufacturer's Rating	Typical California System
Heating, High Fire			
PSC	540	505	450
BPM	227	272	378
Heating, Low Fire			
PSC	477	452	407
BPM	129	154	217
Cooling			
PSC	671	629	570
BPM	350	424	571
Cooling airflow, cfm			
PSC	1554	1434	1214
BPM	1393	1402	1390

Figures 38 and 39 show how using the results from the DOE test procedure will significantly over-estimate the power savings for the BPM blower during furnace operation (heating). The blower power requirements at each operating point illustrated in Figures 38 and 39 are summarized in Table 3. Table 3 also includes:

- the blower power requirements at the Manufacturers' Rating point of 0.5 in. water external static pressure that were excluded form Figures 38 and 39 for clarity,
- the corresponding cooling air flows that shows how the BPM air flow changes very little unlike the BPM, and
- the blower power requirements for a two speed system that can operate at lower blower flows at low fire conditions.

The blower power savings for typical California systems are about 70W (16%) for heating and zero at cooling design conditions. At lower flow resistance using the DOE Test procedure external static pressures, the savings are between 300W and 350W for the BPM blower. Unfortunately, it would be very difficult for a real system to operate at this low system flow resistance[6]. However, two-speed furnaces that can operate in low-fire mode show significant savings potential (of 200W (50% of high fire power) or greater) because they operate at power air flows and system pressures where BPMs have significant advantages. The advantages of multi-speed systems are investigate in more detail in the next section.

[6] A system with no cooling coil, minimal duct system, oversized returns with 4 inch pleated filters could have this low flow resistance – but it would still be difficult to achieve.

Comparing Performance for multi-speed systems

Because multi-speed systems have the capability to operate at lower air flows and therefore lower pressure differences there is the potential to same fan power and energy consumption for these systems. The following analysis compares single speed and two-speed blowers for both PSC and BPM motors. More details can be found in Lutz et al. (2006).

The blower motor power consumption for the two-stage furnace used in this analysis is taken from Table 3. An 80% AFUE furnace was used with a capacity of 88kBtu/h and 58 kBtu/h in high fire and low fire respectively. As an air conditioner, it operates at a 3.5 ton nominal capacity at a single high speed.

Three system curves were used:

1. One based on the DOE test procedure pressures,
2. One based on the manufacturers rating and
3. One for a typical California duct system.

The manufacturer's rating point corresponds to an ideal duct system where the coil, duct, and filter pressures have all been minimized.

A DOE2 simulation was used to calculate hourly heating and cooling loads for a house in the California Central Valley. The furnace cycling rate and occurrence of high-fire operation were based on field studies (Pigg (2003), Pigg and Talerico (2004)) that have shown that staged gas furnaces typically operate such that high-fire mode is only used to recover from setback, and that, for cold climate houses with sheet metal ducts in basements[7], the majority of operation is in low-fire mode[8]. The model of the furnace and air conditioner included the following:

- Standby energy use of 5W for the PSC and 9W for the BPM
- Energy used by the draft inducer including pre and post purge for every cycle
- On and off blower delays and igniter energy use for each cycle
- Effect of changing air flow and outdoor temperature on air conditioning efficiency
- Reduction in natural gas use due to blower motor heating
- Increase in air conditioner operation due to blower motor heating

Including all these interactions is important for making realistic predictions of energy savings for both gas and electricity.

Four design options were examined:

1. Single stage PSC
2. Single stage BPM

[7] This field study was performed in Wisconsin where basement ducts are typical.

[8] For houses more typical of California, with duct systems outside conditioned space (with their increased losses particularly in fan-only mode) we expect the fan-only operation to increase the building load and require more furnace operation and possibly change the high-fire to low-fire ratio.

3. Two-stage PSC
4. Two-stage BPM

Figures 40 through 42 show the annual energy use of the furnace for the four design options and the three duct system options. These figures show the contributions of individual components of energy consumption.

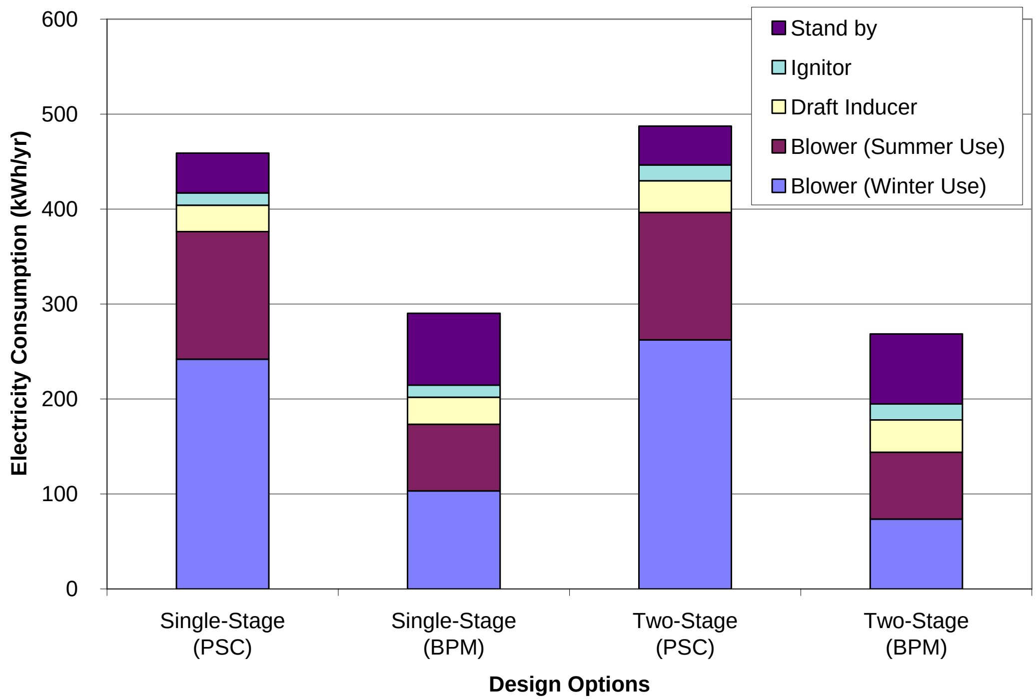


Figure 40. Energy use for four furnace design options using DOE test procedure duct system assumptions

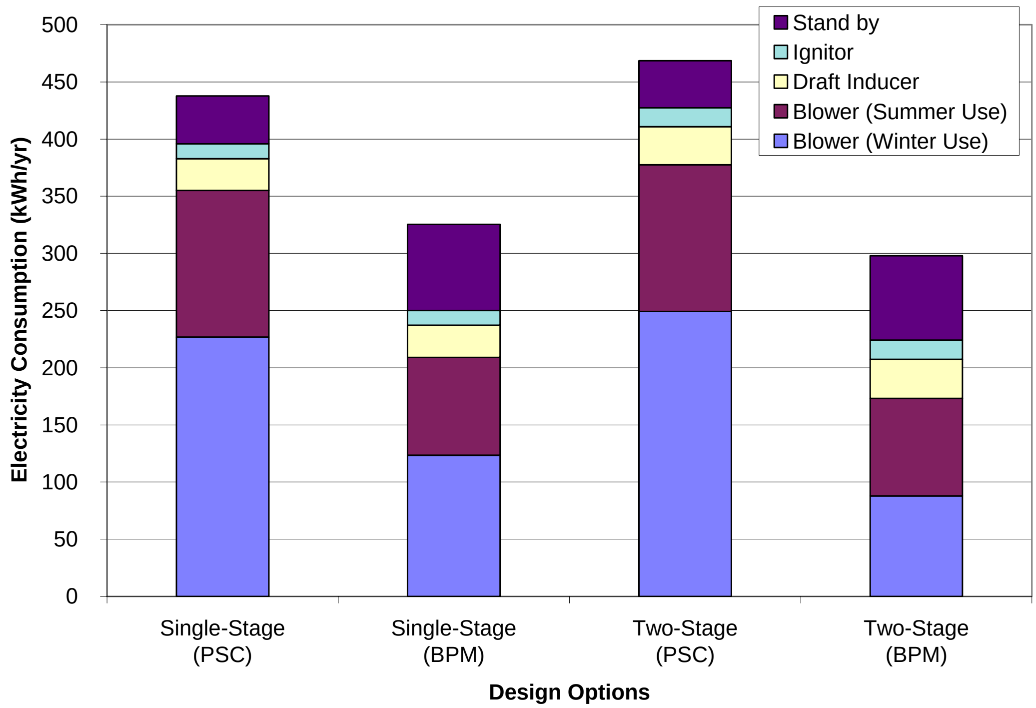


Figure 41. Energy use for four furnace design options using manufacturer's design conditions corresponding to a good duct system

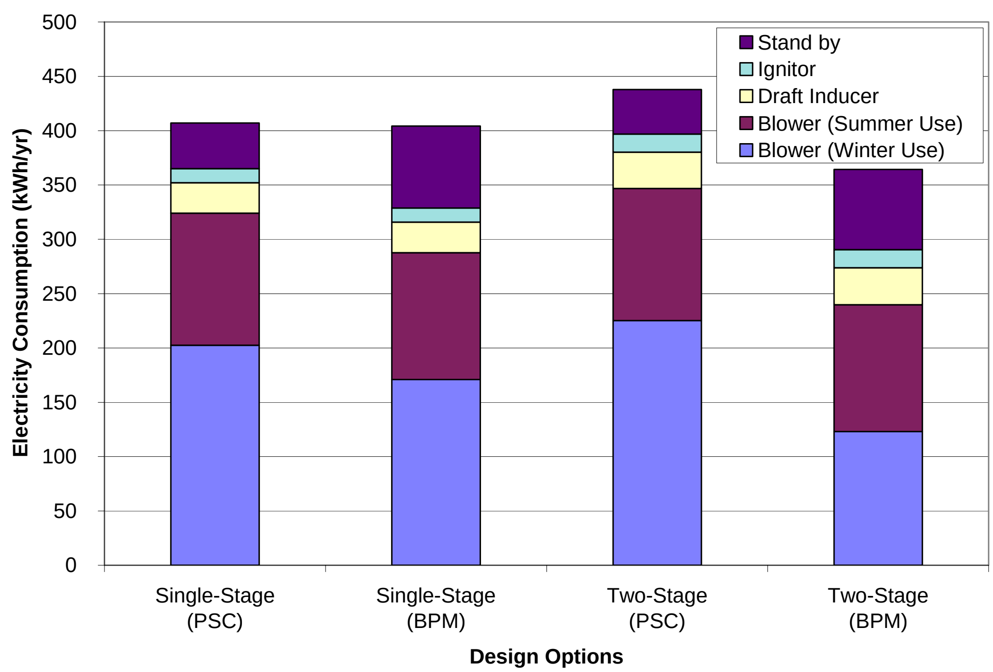


Figure 42. Energy use for four furnace design options using a typical California duct system

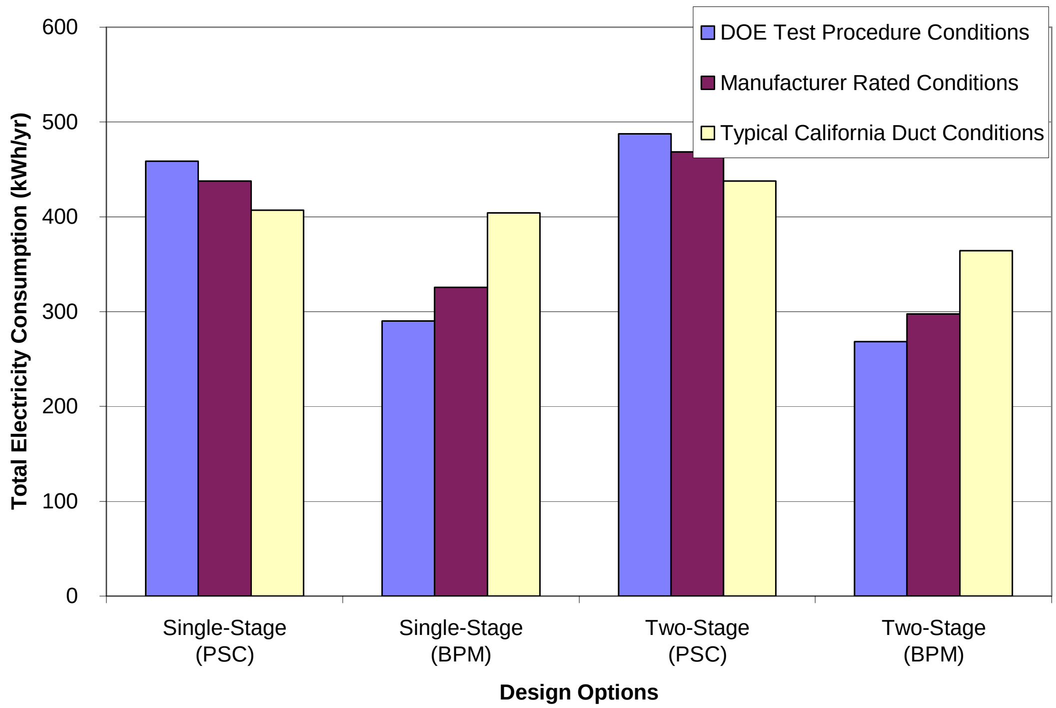


Figure 43. Summary of total energy consumption for four blower motor furnace design options for three duct systems design conditions

It is clear from these test results that care must be taken when determining blower motor energy use and energy savings because the system effects (i.e., the duct system the furnace is connected to) are the same magnitude as the differences between blower motors.

For PG&E the most important result is for the typical California duct system. For this case, the energy savings are minimal for a single-stage furnace. The two-stage PSC motor uses more energy than the single-stage PSC and Variable Speed BPM motors. This is because the PSC motor has lower efficiency at lower flows, and the two-stage furnace operates more at lower speed.

The two-stage BPM uses the least energy. This result means that the variable speed motor rebate should be restricted to BPMs. When considering the PG&E rebate, the most likely decision to be made is between a single-stage PSC and a two-stage BPM (and the associated furnace operation changes).

Under DOE test conditions, the electricity savings are 191 kWh/year (42% or $25 at $0.13/kWh[9]). Using manufacturer's design conditions (representing a good duct system for a California house), the electricity savings are reduced to 140 kWh/year (32% or $18). For a typical California duct system, the electricity savings are further reduced to 43 kWh/year (11% or $6).

[9] PG&E statewide retail average from: http://www.energy.ca.gov/electricity/weighted_avg_retail_prices.html

In winter, the more efficient BPM supplied less heat to the air in the furnace. This must be offset by additional natural gas use. For typical California duct systems the two-stage BPM used 80kWh less electricity than the single stage PSC. This is equivalent to about 2.5 Therms. At $1.40/therm the extra cost of natural gas is $3.50. This offsets the $6 savings for a net savings of $2.50.

Similarly, for the good duct system that meets manufacturer's design specifications, the two-stage BPM used 139kWh less electricity than the single stage PSC. This is equivalent to about 4.7 Therms and the extra cost of the natural gas is $6.50. This offsets the $18 savings for a net savings of $11.50.

These results show that the effectiveness of rebating variable speed motors depends strongly on the duct system the furnace is attached to.

The above results did not include the change in energy use due to the air conditioner necessitated by switching from a PSC to BPM. This can be estimated by looking at the fan energy used in summer. For a typical system, the PSC used 122 kWh and the BPM 117 kWh. The difference of 5 kWh is heat is additional load that the air conditioner does not need to meet. For an EER 11 (COP 3.2) air conditioner, the air conditioner energy reduction is 1.6 kWh, for a total of 6.6 kWh. For a good duct system, the PSC used 128 kWh and the BPM 85 kWh. The difference of 43 kWh for an EER 11 (COP 3.2) air conditioner reduces air conditioner energy by 13 kWh, for a total of 56 kWh.

Combining the additional air conditioner savings with the blower energy savings, the total savings for changing from single-speed PSC to variable speed BPM for a typical duct system is about 45 kWh and for a good duct system 153 kWh.

Peak Savings

The definition of peak demand period used here comes from the CPUC Energy Efficiency Policy Manual Version 2, and is noon to 7:00 p.m. Monday through Friday, June 1 through September 30. The peak demand saving is therefore not just the change in power consumption during system operation but includes the cycling effects of air conditioning at part load. To obtain peak savings estimates, the power savings for an individual system need to be multiplied by the number of houses with air conditioning and the fraction of time the air conditioning operates during the peak period.

There are roughly 8 million homes in California with about 1/3 of them (or 2.67 million) having central air conditioning that would operate during the peak demand period. The fractional operating time in the peak period is highly variable depending on the climate. Recent work by LBNL for the Energy Commission (Walker and Sherman 2006) included sophisticated modeling of energy use of houses in California[10]. Data from the modeling

[10] This simulation effort focused on new, Title 24 compliant homes, with air conditioner sizing based on field summaries performance by other researchers (Chitwood and Wilcox) for the Energy Commission.

shows that in central valley (Climate Zones 11,12 and 13) that fractional operating time for the peak demand period varies from 13% in CZ 12 to 25% in CZ 13.

The question, then is what geographical areas are of interest for PG&E? If it is the whole state, then a lower fractional operating time might be appropriate. If PG&E wishes to focus on high-use areas with rapid growth, then the Central Valley results would be appropriate. If some sort of PG&E service territory weighted value were required, then the power and energy savings for an individual system would need to be weighted by the houses in the PG&E service territory. For the purposes of this report, a fractional operating time of 25% will be used, recognizing that this will be on the high-end of potential impacts. The 25% is not a measure of the number of hours of operation per year – but is the fractional ontime at peak demand conditions. The CPUC peak definition covers three months totaling 121 days. Because the peak only applies on weekdays the total number there are 87 peak days. At 7 hours per day this is 610 hours a year of peak. The 25% value indicates that air conditioners operate for 152.5 hours at CPUC defined peak conditions.

The definition of peak demand period restrains the peak electricity reductions to those during air conditioning operation. This means that the blowers are most likely to be at high speed. The peak demand implications are discussed here for two scenarios:

1. The first assumes that variable speed BPMs are used together with typical California duct systems.
2. The second scenario assumes that duct systems are improved to be as good as we can reasonably expect for California systems with filters and cooling coils. This corresponds to the electricity use of systems that operate at manufacturers rating points.

The improvement to reach 0.5 in. water external static pressure that meets the manufacturers' rating point is possible in California with some relatively simple changes to the way duct systems are installed. Table 4 breaks down the pressure drop by duct system components based on a recent field survey and presents potential improvements (Wilcox 2006). The improved system does not change the supply ducts[11], puts a five ton coil in a three and a half ton system[12], increased return diameter or adding return ducts, and using a 25% greater filter area (or possibly using four-inch pleated filters).

Table 4. California Duct System Pressure Component Breakdown

	Median of Field Survey (in. water)	Improved System (in. water)
Supply Duct	0.18	0.18
Cooling Coil	0.27	0.20
Return Duct	0.15	0.05
Filter	0.15	0.07
Total	**0.75**	**0.50**

Peak Demand Savings for Typical Duct Systems

For a typical California system of 3.5 Tons, Table 3 shows that the variable speed BPM uses the same electric power as the single speed PSC (only one Watt difference). The differences in air flow between the variable speed motor and the PSC change the efficiency of the air conditioning equipment.

Algorithms given in ASHRAE Standard 152[13] (ASHRAE 2003) show that the airflow reduction from 1400 cfm to 1214 cfm reduces a typical air conditioner capacity by about 5%[14]. Although the power used by the air conditioner does not change, this means that the air conditioner will run 5% more to meet the same building load. For a 3.5 ton air conditioner with an EER of 11 (COP=3.2) the air conditioner power is 3820 W. From Table 3 the power consumption of the PSC blower for a typical California duct system is 570 W. The total watt draw including the blower is therefore 4390 W. Because the power consumption of the BPM is essentially the same (571 W) the savings can be estimated directly from the reduction in runtime. Reducing the runtime by 5% by using a BPM therefore results in a 191W savings[15]. Using a fractional on-time of 25% during the peak demand period results in a net peak demand reduction for a variable speed motor of 48 W per house. If this were applied to all 2.7 million systems in the State it would be a peak demand reduction of 0.13 GW. This

[11] Although in new construction revised shortened duct layouts, particularly in high performance houses, could reduce the supply duct pressures

[12] This must be done carefully to ensure good matching to outdoor unit

[13] Based on laboratory and field test data

[14] Power consumption remains the same for typical residential air conditioners

[15] The actual air conditioner will have 5% more runtime. Because the definition of peak is over several hours and many systems then the savings can be expressed as the average change in power for convenience.

probably represents an upper bound as some climates will have less fractional on-time on peak, and other studies have shown that not all air conditioning systems are operational.

To obtain a breakdown for PG&E peak demand savings the number of systems can be estimated from the 4.4 million total electric customers (according to PG&E communications department[16]) and the Residential Appliance Saturation Survey database that indicates that 39% of these customers have central air conditioning. This implies that about 1.7 million air conditioners are in the PG&E service territory, therefore the maximum peak power savings for PG&E is about 0.08 GW. This is the upper bound that assumes are Air Conditioning systems would be operational. Actual peak demand savings will be less than this.

Peak Demand Savings for Good Duct Systems

For a typical California system of 3.5 tons, Table 3 shows that the variable speed BPM uses 424 W compared to 629 W for the single speed PSC, for a difference of 205W. The differences in air flow between the variable speed motor and the PSC are negligible, so there is no change the efficiency of the air conditioning equipment. However, the extra heat from the PSC does contribute to building load.

For an air conditioner with an EER of 11 and a COP of 3.2, the added load results in additional air conditioner consumption of 64W on averaging during peak demand times. The total addition peak power consumption is therefore 269W. Using a fractional on-time of 25% during the peak demand period results in a net peak demand reduction for a variable speed motor of 67 W per house. If this were applied to all 2.7million systems in the State, it would be a peak demand reduction of 0.18 GW. For the 1.7 million systems in PG&E service territory the demand reduction is 0.11 GW. Again, this represents an upper bound.

Cost-Effectiveness of Variable Speed Motors

The costs associated with furnace blowers are notoriously difficult to estimate given the economies of scale that result from transferring technologies from niche markets to mass markets. The factors influencing the costs are:

1. The cost of a BPM motor. There is a range of cost estimates from different sources (e.g., Sachs and Smith (2003)) from about $25 (assuming a fully mature market) to over $100. A CMHC (1993) study stated cost premiums of replacing low-efficiency motors with high-efficiency motors of about C$20 to C$100 (in 1993). General Electric Co. product literature shows that the retail cost (motor only) of BPMs for residential air handler applications is $90 to $100. Gusdorf et al. (2003) reported Canadian wholesale process of $170 and customer costs of $300.
2. The incremental cost over existing PSC motors. Assuming a furnace will have some sort of blower motor, then the cost to switch to a BPM is the cost of upgrading from the PSC

[16] PG&E communications department indicate that there are 4.4 million residential electric customers, 4.0 million gas customers and 3.2 dual customers. The total number of residences serviced is 5.2 million. The residential average bills are 540 kWh/month and 45 Therms/month.

and the cost of the PSC needs to be subtracted from the BPM cost. This was studied in detail by Sachs and Smith (2003) who estimated incremental costs of $80 to $90.

3. Variable speed motors are not offered as a direct swap out in all furnaces. Instead, variable speed motors are usually offered in high-end more expensive furnaces. The real incremental cost to a consumer therefore includes the additional features of the high-end furnaces as well as the increased motor cost. This is offset by the additional features of the high-end furnaces (e.g., slow start-up that utilizes the ability of a BPM to operate at lower speeds) that have comfort, noise and other performance issues.

Cost-effectiveness estimates in previous studies used higher estimates of energy use than discussed in this study, and therefore different life-cycle cost estimates. This is partially due to assumptions about energy savings that are not achieved with typical high system pressures in California systems, use of continuous fan operation, and assumptions about the number of operating hours[17].

For example, a PG&E CASE study (2003) reported positive life-cycle costs (a net present value of $245) for non-continuous operation of the air handler. Other studies have included continuous low-speed operation where the BPM has significant advantages over PSC. For example, a recent CMHC study (2005) field tested a PSC and BPM in the same furnace in the same house and concluded that the PSC saved 74% of electricity use (but used 14% more natural gas for heating). Similar results are reported by Pigg and Talerico (2004) in field monitoring of 31 houses. In cooling operation the fan power savings were 48% with a total energy savings including air conditioner operation of 14%. For non-continuous operation in the CMHC study there was small increase in the cost of operation and a small increase in energy savings.

If we use the Sachs estimate of $80 for incremental cost, then the payback time assuming no PG&E rebate for typical system that saves 45 kWh/year is 13.5 years (at 0.13/ kWh). For an improved system that meets manufacturer's specifications that saves 153 kWh/yr this is reduced to 4 years. If fans were operated continuously—for mixing, filtration or ventilation purposes, the potential savings for BPMs improve even more, about 1800 kWh per year and the payback time is less about 4 months.

Conclusion

Analysis of the detailed laboratory investigations and energy use analysis in this study has shown that:

- BPM and PSC blowers have distinctly different performance characteristics that must be accounted for when proposing performance specifications.
  - BPM blowers have better performance in terms of maintaining air flow at typical system pressures and reduced power consumption compared to PSC

[17] The systems in these simulations operated for about 1000 hours/year for heating and cooling.

blowers. However, the advantage for BPM blowers is marginal at high pressures above about 0.8 in. water (200 Pa).
    - As system pressures increase, the PSC blower power consumption decreases as it is unable to maintain air flow.
    - PSC blowers have cfm/W ratings that are very close to proposed performance specifications, while the generally higher cfm/W ratings for BPM blowers will make it easier for them to meet these specifications.
- A performance specification should include both an air flow specification and a cfm/W specification.
- One important utility issue with BPMs is their lower power factor. For PSC motors, the power factor ranged from 0.7 to 0.9, with the lower power factors at high-speed settings and high pressures. For BPMs the power factors range from 0.53 to 0.62 with higher power factors at higher speed.
- As well as the external static pressure effects, the air flow patterns within blower door cabinets also affect performance. Side entry, particularly on the motor side of the blower should be avoided. Restrictive cabinets that have little clearance around the blower also reduce blower performance. Air flow reductions are about 10%, leading to reduced cfm/W ratings. This may be critical if one is trying to meet a performance specification for energy efficient blower credit such as that proposed for 2008 California Building Energy Standards. Because cabinet size is restricted by the size of attic access, systems with higher air conditioner capacity and larger blower requirements are most likely to have these problems. Field studies have shown that five-ton systems tend to be most affected.

The laboratory test results for blower power and air flow were combined with DOE2 models of building loads, models of air conditioner performance[18], standby power, as well as igniter and combustion air blower power. This sophisticated modeling allowed for the interactions of blower performance with the HVAC equipment[19] to give better estimates of annual energy consumption than a simple power difference and runtime hours analysis. The furnaces used in this modeling were two-stage with single speed air conditioning - as typically found in furnaces with variable speed motors.

The following energy savings and peak demand reductions are based on replacing a PSC blower with a BPM blower to evaluate the effect of the current PG&E rebate program.

- Energy savings for a typical 3.5 ton California system are about 50 kWh. This could be increased to 150 kWh for lower flow resistance systems.
- Peak demand reductions are about 50 W for a typical California system or about 0.13 GW maximum statewide reduction. This could be increased to 70 W, or about 0.18 GW statewide reduction, for low flow resistance systems.

The above results are all for systems whose blowers only operate during heating or cooling. Continuous fan operation used for mixing, filtration or ventilation purposes significantly increases the potential savings for BPMs. BPMs have greater performance advantages at low

[18] To account for reduced air conditioner capacity at lower air flows for PSC motors.

[19] Blower waste heat leads to reduced gas use for heating and increased air conditioner load.

external static pressures associated with the low flows used in continuous fan operation. BPMs also have the advantage of operating over a wider flow range and allowing the use of lower air flows that further contributes to lower power consumption.

- At low flow, the BPM uses about 100W compared to 400W for the PSC. Over a year of 6000 non heating or cooling hours, this adds up to 1800 kWh savings for a BPM.

The conclusion of this study is that potential energy savings for BPM blowers may not be realized because the improvements blower performance are restricted by the static pressures that are prevalent in residential thermal distribution systems in California. These pressures are difficult to reduce due to the presence of filters and cooling coils that account for more than half of the system static pressure. It is possible to extract improved performance from BPMs but good duct design and installation are required. This is easier to accomplish for heating only systems that have no cooling coil. For both heating and cooling, pressure reductions can be achieved by increasing return duct sizing (or number of returns), return grille area (often accomplished by having multiple returns) and for four-inch pleated filters that have reduced pressure drop.

These laboratory tests have examined only a small sample of furnaces; however, these general conclusions can be broadly applied based on input from furnace manufacturers and the test results from other unpublished studies (e.g., Proctor 2005 – personal communication). Specific furnace/blower combinations will have different air flow characteristics, cfm/W ratings and pressure difference sensitivity, but the general trends and observations will still apply.

Recommendations for future work

Only a small sample of furnaces were evaluated in the laboratory tests. To be more definitive, it would be a good idea to test more furnaces from a range of manufacturers. In addition, furnaces over a range of capacities (including extreme high (5 ton units) or low capacity should be tested to see if the general results from the testing described in this report are applicable in all cases.

The benefits of the variable speed BPM motors depend strongly on interactions with the rest of the thermal distribution system. The high air flow resistance of typical California duct systems almost eliminates the advantages of the BPM motors. Future work should investigate how to reduce pressures in residential duct systems in conjunction with use of variable speed BPM motors. Key areas are:

- Use of larger or multiple returns
- Use of low pressure filters – promote the use of 4 inch deep pleated filters
- Use of larger air conditioning coils to reduce coil pressure drop[20]
- Use of more compact duct systems with shorter duct runs

[20] Need to be careful about maintaining a good match with the rest of the cooling system.

- Careful installation of ducts to reduce number of elbows and make duct runs as straight as possible
- Encourage use of sheet metal duct instead of flexible duct

The system interactions between blowers and the rest of the HVAC system have been shown to be very important. More study needs to be performed in this area, e.g., look at the blower power effects of using more efficient coils with higher flow resistance (due to smaller fin spacing). Also, some ventilation systems use the furnace blower to distribute ventilation air. This results in more operation hours for the blower that can have a significant impact on energy use and peak demand (since it reduces the diversity of blower operation at peak). These interactions require further investigation, in particular because these systems are becoming more common.

Another avenue for future work is the investigation of improved blower wheels that can give further performance improvements.

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