FAN ENERGY SAVINGS: ANALYSIS OF A VARIABLE SPEED DRIVE RETROFIT

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ABSTRACT

Variable speed drives (VSDs) were installed on two 50 hp and two 20 hp fan motors in a commercial office building with a variable air volume (VAV) distribution system. The drives replaced inefficient variable inlet vanes (VIVs) as a means of controlling the air flow. Fan motor part-load performance has been studied for the periods before and after the retrofit using analysis of fan power as a function of air flow.

The building in which the fans are located uses fan-powered VAV boxes for terminal air distribution and space temperature control—a system commonly found in newer commercial buildings. The requirement that minimum duct static pressure levels be maintained constrains the minimum fan power, thus limiting the energy savings that can be achieved at lower flow rates. Experiments were conducted to determine how the fan systems perform given lower static pressure set points. Energy savings for supply and return fans were evaluated as a function of flow rate distribution at two duct static pressure set points. Based on short-term data, annual estimated savings for the two supply and two return fan motors were 100 MWh/yr (35% of pre-retrofit consumption) and a simple payback of eight years with no change in duct static pressure set point; 150 MWh/yr (52% of pre-retrofit), with a five year payback, if static pressure is decreased for nine months of the year) from 2.5" w.g. to 1.5" w.g. The relatively long payback favors incorporation of VSDs in new construction rather than as a retrofit item.

Decreasing the static pressure set point in a nearly identical system still employing VIV fan control was found to have no significant effect on fan power, in contrast to the results for VSDs, where the magnitude of estimated savings due to decreasing static pressure set point is half of the savings due to the installation of VSDs alone. This suggests that VSDs installed in combination with static pressure reset as a control system function is a desirable configuration, and that digitally controlled terminal boxes, just coming on the market, should be employed if such control is to be implemented.

The discrepancy between actual fan energy use and that obtained using power curves found in the literature and used (by default) by the building simulation program DOE-2 was quantified. These curves, which do not account for maintenance of duct static pressure, resulted in under-prediction of supply fan energy savings by 6% of estimated pre-retrofit supply fan annual energy use.

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INTRODUCTION

Variable speed drives (VSDs) were installed on two supply fan and two return fan motors for variable air volume (VAV) distribution systems in a 12,000 m^2 (130,000 ft^2) commercial office building in New Jersey. The drives, which control airflow by varying the speed of the fan motors, replaced less efficient variable inlet vane (VIV) flow control, which works by throttling the flow at the fan inlet. The purpose of this study is to measure and model part-load fan performance and annual energy use for VIV and VSD flow control, estimate annual energy savings due to the retrofit, and investigate the applicability of these results to fan systems in other buildings.

System Description

There are two independent but identical VAV air distribution systems in this building, each serving one side. Such systems are common in newer office buildings, each consisting of a supply fan, a return fan, and terminal boxes in the offices. The terminal boxes are of four types: VAV with and without reheat coils which are used in corridors and supply only primary air (from the central supply fan); and fan-powered VAV with and without reheat coils, in which the fan serves to induce circulation of secondary, or room air. Three-fourths of the boxes are fan-powered, and the majority (located in perimeter spaces) have reheat coils.

Each box is connected to a local thermostat, and maintains the space temperature by modulating its primary air damper, and in the heating season, modulating its heating coil if the damper is already at a minimum position. The amount of cooling that a box provides is proportional to the product of the flow rate and difference in temperature between supply and room air. Since supply air temperature is relatively constant, cooling capacity is proportional to the rate of air flow through the box (ignoring enthalpy considerations). This flow rate is some function of damper position and duct static pressure (DSP). Since DSP is maintained by the supply fan at a constant set point manually selected by the building operator, the cooling capacity of a box is a function of damper position alone, and it is this variable which the controls at the box change in response to a changing cooling load in the space served.

Given some design cooling load there is a DSP at which the box damper would be wide open while maintaining set point temperature. If the load were to increase without a corresponding increase in DSP, the box would be "starved for air," and the space temperature would increase. Hence, the supply fans must keep DSP at a level such that at any given time, the box with the highest load is not starved for air. Since the DSP set point is set manually and cannot vary automatically in response to changing loads with existing controls, it must be fixed such that terminal boxes can always handle peak loads in the offices, which for perimeter spaces occur during the hottest months of the cooling season[1]. This point is relatively high compared to the actual requirement for much of the time. The set point could be adjusted by the building operator, but in practice has been left at a fixed value.

The pneumatic control system was designed to adjust the supply fan flow to maintain the DSP set point (measured by a sensor located in the duct), and the return fan to maintain flow at 10,000 CFM (4.7 m^3/s)—which is the required outside air flow rate—less than supply flow. The flow rates of the supply and return fans were originally controlled by VIVs, where vanes at the fan inlet vary in position between fully closed (except for some

[1]Terminal boxes can also be starved for air in milder weather if the chillers are not operating and the supply air temperature is too high, or if the office space has large internal loads. High supply air temperatures may occur during the economizer cycle when the mixing damper controls sacrifice low dry-bulb temperature for low enthalpy of the mixed air stream.

by-pass area around the vanes) and fully open. The VIV control was replaced, as mentioned above, and now VSDs control the flow by adjusting the fan motor speed. There is no longer a pressure drop across the inlet vanes, and now the fans do less work for a given flow rate.

APPROACH

Pre- and post-retrofit annual fan energy use were estimated, and in turn, energy savings. Annual fan energy use for each fan was estimated by combining a polynomial fit of power versus flow data with a flow rate histogram. The power and flow data are hourly averages collected with a data acquisition system for a one year period before the retrofit (the "base year") and for shorter (one to two month) periods afterward. Base year flow histograms were used both in pre- and post-retrofit estimates of energy use so as to "normalize" the energy use, thus removing flow rate distribution as a variable in the comparison of pre- and post-retrofit consumption. Actual metered fan motor electricity use for the base year was used to verify the model.

Since DSP set point is a determinant of fan energy use and is a parameter that is specific to particular systems and control schemes, and one that might be varied in buildings utilizing control systems designed for energy management, we attempted to quantify the effect of changing this parameter on fan energy use in both VIV and VSD systems. Since we did not experiment with changing DSP in the pre-retrofit VIV system of the building under study, we attempted to apply results of experiments involving decreasing DSP in a similar building (with identical fans).

The building studied here was modeled in another study (Hsieh, 1988) using DOE-2, a commonly used building energy simulation program. The default power curves DOE-2 uses to model the fan system apparently do not account for maintenance of DSP, and result in an under-prediction of annual fan energy use. We present the DOE-2 default curves for comparison and quantify the discrepancy in predicted fan energy savings.

ANALYSIS AND RESULTS

Fan Power, Pre- and Post-retrofit

The base year for this study was chosen as April 1, 1986 to March 31, 1987. Pre-retrofit estimates of fan power as a function of flow were obtained using base year data. The flow rates for each fan were "binned," resulting in histograms which give the number of hours of operation in the base year as a function of flow rate (Figure 1). Regressions were performed to determine expressions for fan power as a function of flow for both supply fans and both return fans:[2]

$$\text{Supply fans: } P_e = 25.0 - 0.36q + 0.0507q^2 - 0.000587q^3 \qquad R^2 = 0.61$$
$$\text{Return fans: } P_e = 9.0 + 0.0888q - 0.00601q^2 + 0.000425q^3 \qquad R^2 = 0.25$$

where P_e is the electric power used at the motor (kW), q is the flow (m^3/s), and R^2 is the coefficient of determination. A portion of the data and the regression curves are shown in Figures 2 and 3. The poor R^2 in the case of return fans can be explained by the small slope in the data corresponding to relatively constant power with respect to flow.

Hsieh (1988) compared the regression curve (a) in Figure 2 with the default curve DOE-2 uses in its fan system model (b), which apparently does not account for maintenance of static pressure, and found that the latter under-predicts annual fan energy consumption by 14%. While the DOE-2 Engineers Manual (1982) does not explicitly state that DOE-2 ignores static pressure control, the shape of the DOE-2 power versus flow curve is consistent with those endorsed by ASHRAE (1982, 1988), which correspond to an absence of static pressure control. While static pressure need not be maintained in such systems as boiler exhaust fans, constant static pressure control is required for VAV systems, and this control system maintains a DSP of 2.5" w.g. (0.62 kPa).

The DOE-2 curve (b) in Figure 2 originates at the "full power point," 47,000 cfm (22.2 m^3/s) and 42 kW. This point is the maximum flow and power at which 2.5" (0.62 kPa) DSP can be maintained, obtained from field

[2]Fitting both supply fans with one curve was justified by the data (Figure 2), and we know of no physical reason why this should not be the case—the ductwork is identical. The same is true for return fans.

observations (Curtiss, 1987), and confirmed by analysis of the base year data. Nearly all of our data are at flows below this point. The curve is given by

$$C_p = 0.35 + 0.31C_q - 0.54C_q^2 + 0.87C_q^3$$

where

C_p = power / full power
C_q = flow / flow at full power

Using the measured full flow and power, this becomes

$$P_e = 14.7 + 0.586q - 0.0460q^2 + 0.000334q^3$$

This curve drops below most of the data (by 0 to 8 kW) because it does not account for static pressure control, although due to the choice of full power point, it over-predicts the power near peak flow for the bulk of the data in the region.

At a flow higher than that at full power is the "purge point," achieved only when all the terminal boxes are wide open (Figure 4). We observed this point when the ventilation system was in smoke purge operation and measured a DSP of 0.9" (0.22 kPa). For these supply fans, the increase in DSP from 0.9" to 2.5" occurs at nearly constant fan power, as the flow drops. (This is illustrated in Figure 5; the path along the 875 rpm line from the purge point to the full power point is nearly parallel to lines of constant power).[3]

The HVAC controls literature notes that constant static pressure control will cause deviations from the fan performance assumed by DOE-2 (McQuiston and Parker, 1982 and Wood, 1987), however, we know of no field study that has quantified this effect.

Similar analyses are performed on the post-retrofit data, and are shown in Figures 3 and 6. The post-retrofit correlations are

Supply fans: $P_e = -0.91 + 2.1q - 0.0798q^2 + 0.00284q^3 \quad R^2 = 0.93$
Return fans: $P_e = 0.68 - 0.24q + 0.0533q^2 - 0.000332q^3 \quad R^2 = 0.95$

The increase in R^2 is due (physically) to less scatter in the data and (statistically) to greater slope. The greater precision of VSD control results in less scatter;[4] the greater decrease in power as flow decreases yields a larger slope. It is easy to see how these characteristics affect R^2 if we express R^2 as

$$R^2 = 1 - \frac{n \cdot MSE}{\sum (P_i - \bar{P})^2}$$

[3]Researchers performing similar analyses should use caution in the choice of the full power point. The full power point as specified in a mechanical system design is intended as a maximum for use in sizing the fans, and is not an operating point at which DSP can be maintained. For the building studied here, the full power point as specified is close to the purge point, where static pressure is not maintained.

[4]Less scatter may also be due to the fact that the post-retrofit measurement period is shorter and DSP was observed to be fairly constant at 2.5" during this period. Subsequent observation however, has indicated that DSP occasionally deviates from this value, and it is probable that this occurred during the base year. Scatter in these data has also been shown by Curtiss (1987) to be related to outside air volume, as the duct design in this building enables outside air to influence static pressure upstream of the intake, hence affecting fan power.

where

n = number of data points

MSE = mean squared error $= \frac{1}{n}\sum r_i^2$

r_i = residual values

P_i = measured power values

$\bar{P}$ = measured mean power

So a decrease in MSE and increase in $P_i - \bar{P}$ (greater slope) increase R^2.

The DOE-2 curve for VSD control (using the measured full power point) is given by:

$$P_e = 0.643 + 0.00986q + 0.0945q^2 - 0.000445q^3$$

Annual Energy Use and Savings

A useful basis for comparison of pre- and post-retrofit energy use is a "normalized" annual consumption, calculated using flow distributions observed in the base year. Since the actual flow distributions after the retrofit may vary, this method of normalization does not predict an actual post-retrofit consumption, but serves to evaluate the effect of the motor controls on fan energy use. The normalized annual energy use for a given fan is then

$$E = \sum_{i=1}^{n} H_i \cdot P(q)\big|_{q = q_i}$$

where

n = number of bins in flow rate distribution

H_i = number of run-time hours in bin i for pre-retrofit period

P = power as a function of flow determined by regression for given period, evaluated at q_i

q_i = mean flow rate of bin i

As a check on this model, the predicted pre-retrofit consumption for each fan was compared with the actual metered electricity use. The agreement was close, within 3%. Estimated pre- and post-retrofit consumption and energy savings are given in Table I. Savings are taken as the estimated base year consumption minus estimated post-retrofit consumption.

The energy-weighted combined savings is 35% of pre-retrofit consumption. Except for supply fan AC-2, which is using 12% less energy than the pre-retrofit value, the estimated savings range from 45% to 61% across fans. The low savings for AC-2 are elucidated in Figure 2: this fan is operating in the upper range of flow rates most of the time (unlike AC-1), and the motor is usually running near full speed. AC-2 serves the east side of the building which receives higher solar loads. However, it is unlikely that this fact alone could account for the higher flow rates. An explanation of this flow behavior is beyond the scope of this phase of the study, but is a good candidate for further investigation. The economics of the retrofit are discussed below.

Reducing Static Pressure

We recognized that manually reducing the DSP set-point from 2.5" to 1.5" (0.62 kPa to 0.37 kPa) would reduce the supply fan power. The return fans are not controlled to maintain static pressure, and to the first order, we expect no change. We chose 1.5" because this value had been specified at the building design stage and had subsequently been raised by a balancing contractor. We originally hypothesized that the higher value is required only in summer to prevent overheating. However, work by Hsieh (1988) shows high flows occurring throughout the year, suggesting a year-round need for higher static pressure to cool some offices during periods of high loads.

Although only a week of 1.5" data was collected at the time of analysis and not enough consistent data could be obtained for the return fans, good results were achieved for the supply fans, and are shown in Figure 7 (for AC-1) and tabulated in Table I. The annual consumption reflects operation at 1.5" for the whole year. This is not realistic, as discussed below, but provides an indication of the effect on savings of lowering static pressure.

***Table I.** Estimated fan energy savings, normalized by base year flow distribution. Units are MWh/yr.*

	Supply fans		Return fans		Total
	AC-1	AC-2	RF-1	RF-2	
Base year: 4/1/86 – 3/31/87					
Consumption, actual[a]	96.2	118.7	35.4	37.9	288.2
estimated[b]	94.3	118.5	35.8	36.8	285.4
% difference	–2%	0%	1%	–3%	–1%
Post-retrofit, 2.5" DSP[c] 11/10/87–12/22/87					
Consumption, est.[b]	52.0	104.4	11.6	16.1	184.4
Savings, estimated	42.3	14.1	24.1	20.5	101.3
% savings	45%	12%	68%	56%	35%
Post-retrofit, 1.5" DSP[d] 12/23/87–1/7/88					
Consumption, est.[b]	28.8	64.5	—	—	121.0[e]
Savings, estimated	65.5	54.0	—	—	164.1[e]
% savings	69%	46%	—	—	57%[e]

[a]Based on meter readings.

[b]Using model combining measured power and flow data with base year flow distribution.

[c]Duct static pressure set point for this period was 2.5" w.g.

[d]Duct static pressure set point for this period was 1.5" w.g.

[e]Not enough consistent data were available to determine return fan energy consumption at this static pressure; total includes return fan consumption for 2.5".

The reduction in static pressure results in a clear power reduction, which seems to be independent of flow rate, yielding energy savings over the whole range of flow conditions. The savings jump to 69% for AC-1 and to 46% for AC-2, bringing the energy-weighted average to 57%, not accounting for any change in return fan performance. The increase in savings due to less required fan power is the result of the decrease in pressure drop across terminal boxes, which are open wider for a given flow rate. Of course, the 1.5" set point is somewhat arbitrary; the requirement at any given time may be higher or lower than this. However, if an operating strategy were adopted such that the static pressure were reset by the building operator on a daily basis, this value is probably the lowest set point that could safely be used at this resetting interval.

Figure 4 shows system pressure curves, with and without maintenance of a given DSP. The DOE-2 model does not include a static pressure constraint, hence its curve (b) intersects the origin. Both system curves intersect the 875 rpm fan curve at the measured full-flow point. These curves indicate the static pressure rise across the fan as a function of flow rate. We expect the system to follow the upper curve (a), intersecting the vertical axis at 2.5", but we do not have pressure data either accurate enough or over a broad enough range to confirm this.

This requirement to maintain DSP has important energy implications. Figure 5 shows system curves identical to those in Figure 4, but with lines of constant power (in hp) superimposed. It may be useful to illustrate the operation of the system in the following manner: Suppose the flow were controlled by an outlet damper instead of by inlet vanes. The fan power as a function of flow could be determined as the intersection of the 875 rpm curve with the constant horsepower curves. The distance between the 875 rpm curve and the system curve for 2.5" static pressure (a) represents the pressure drop across the damper, and the distance between curve (a) and the horizontal axis is the pressure drop across the system. The difference between the horsepower at the 875 rpm curve and the curve (a) is the energy loss due to the damper. VIVs theoretically do better than outlet dampers due to a swirl imparted to the flow (McQuiston and Parker, 1982), and are usually thought to follow a power curve originating at the full-flow point and travelling somewhere between the 875 rpm and 2.5" lines—but closer to the upper line—yielding a slight improvement in energy performance. If the static pressure constraint is removed, the pressure drop across the system and vanes is less, and the curve (b) intersects constant power lines at a lower level for much of the flow range, corresponding to lower energy use.

A power curve for VSDs, maintaining static pressure, could be expected to track just slightly above curve (a), as there is no pressure drop due to a damper, but a slight energy loss (as heat) due to the efficiency of the controller, which is probably around 98% to 99%. A corresponding curve for no maintenance of static pressure should follow slightly above curve (b) going through zero pressure. Curves such as (b), which can serve as lower bounds on the energy consumption one could expect from installation of a VSD, are often seen in the literature indiscriminately applied to distribution systems such as this one, where DSP must be maintained.

The observed power curves, before and after the retrofit, are also shown in Figure 5. The transformation to the pressure-flow axes is done by relating fan power to electrical power as a function of flow.[5] The relationship is

$$P_e = \frac{0.746\, P_s}{\eta}$$

where

P_e = electrical power measured at the motor (kW)
P_s = shaft power used by the fan (hp)
η = motor efficiency x pulley/belt efficiency = 0.878
0.746 = kW per hp

The efficiency term was measured at the purge point,[6] by relating electrical power to fan shaft power, as determined by the manufacturer's fan curve. The name-plate motor efficiency is 0.902; the overall motor/pulley/belt efficiency of 0.878 corresponds to a measured belt/pulley efficiency of 0.973 (Curtiss, 1987). The power curves shown should not be extrapolated below the flow rates for which they are shown, as they are based on an assumption of constant motor efficiency. This assumption does not hold below about 45% of full load, where the efficiency decreases sharply (Baldwin, 1988).

The curve for VSDs is slightly higher than we expect it to be as flow decreases, but not unreasonable considering the scatter in the data, and some allowance for uncertainty in measurement. The VIV curve, however, behaves quite unexpectedly at lower flows, asymptotic to the 30 hp line and above the 875 rpm curve which corresponds to the power used by an outlet damper system. This would indicate that the pressure drop due to the inlet vanes is greater than that for an outlet damper—something not readily justifiable from a physical standpoint and contrary to our expectation that as flow approaches zero, the power lines for VIV and outlet damper should converge on the same point (at about 15 hp).[7] Two factors that could contribute to the discrepancy are

- The measurement of pulley/belt efficiency is inaccurate.
- The flow at the inlet is extremely complex, and it is conceivable that the vanes could be imparting a net tangential velocity opposite to that of the fan blades. Investigation of such phenomena are, however, beyond the scope of this study.

We have nearly ruled out as a cause for the discrepancy the assumption of constant motor efficiency by applying a typical efficiency curve for this type of motor to the VIV power curve. The constant-efficiency VIV curve shown in Figure 5 comes within a few percent of a variable-efficiency one, obviating the use of the latter. An actual efficiency curve from the manufacturer would, of course, be necessary to ascertain this.

In order to gauge the magnitude of savings that could be achieved by lowering static pressure in a VIV system, we performed tests at two set points in a very similar building with identical fans controlled with VIVs. In a VIV system, when the DSP set point is decreased, the inlet vanes close, and terminal boxes open up to maintain

[5]Pressure can then be found as a function of flow and power, determined by fitting a cubic surface to manufacturers' data.

[6]This point is the same as the peak power point specified by the building designer.

[7]Buffalo Forge Co. (1983) provides a good illustration of this. Compare Figures 15.23 and 15.25 in this reference.

the same flow. The pressure drop across the system is less, but this pressure drop is transferred to the inlet vanes, and a reduction in power should be observed, due to the greater swirl that the more highly angled vanes impart to the inlet air. We had hypothesized, based on published power curves for VIV control (Buffalo Forge Co., 1983) that this reduction in power is perhaps about 15%. The data shown in Figure 8 indicate no significant savings resulting from decreasing DSP from 2.7" to 1.7" in a VIV system. Our expected savings could, however, be obscured by noise in the data (~15% at 25 kW). Comparing published power curves for VIV and VSD leads us to expect varying DSP to have a larger impact on VSD than VIV. In fact, we see a 34% (of post-retrofit annual energy use at 2.5" DSP) reduction for this VSD system.

Comparing Measured Savings with those Predicted Using DOE-2 Fan Curves

Given that the DOE-2 curves do not model constant DSP control, it is useful to compare the savings predicted using DOE-2 curves with those calculated using measured power curves.

The combined savings of 35% is in good agreement with reported results of simulations (which employ the DOE-2 default fan curves) for office building fan systems, which range from 26% to 30% (Eto and Almeida, 1987). Comparisons between buildings are tenuous, however.

Using power curves presented in Buffalo Forge Co. (1983) as a basis, we expect the savings calculated using the DOE-2 curves for VIV and VSD control to be larger than that of the in-situ results. This is not the case—if we apply the DOE-2 curve to the our histogram of measured flows, we obtain estimated retrofit savings of 30% (of estimated pre-retrofit supply fan energy use) for supply fans only, compared to 36% for our model. We believe that the DOE-2 curves predict smaller, rather than larger savings because the measured VIV power curve is higher than expected at flow rates below about 38,000 CFM (18 m^3/s), and does not intersect the measured full-power point as do the DOE-2 curves.

Table II. *Cost, Savings, and Simple Payback for Variable Speed Drive Retrofit, with and without Change in Static Pressure Set Point.*

	Supply Fans	Return Fans	Total
Cost[a]	$25,200	$14,800	$40,000
Savings, 2.5" s.p.			
Energy	$2,900	$2,300	$5,200
Demand	—	—	—
Total	$2,900	$2,300	$5,200
Payback (yrs)	8.7	6.4	7.7
Savings, 1.5" s.p.[b]			
Energy	$5,300	$2,300	$7,600
Demand	$1,100	—	$1,100
Total	$6,400	$2,300	$8,700
Payback (yrs)	3.9	6.4	4.6

[a]Including design

[b]Static pressure at 1.5" for September through May; 2.5" for June through August.

COST EFFECTIVENESS

Our review of fan power before and after the VSD retrofit indicates that there is no demand savings when the static pressure is maintained at 2.5". If static pressure is reduced to 1.5", the demand savings (for supply fans only) is about 20 kW. For the purpose of a simple payback evaluation, we assume that the static pressure set-point can be reduced to 1.5" for nine months (September to May).[8] The cost of energy is $0.051/kWh.[9] The cost of the avoided demand is $8.17 (on peak) + $1.06 (off peak) = $9.23/kW-month for nine months. The results of the

[8]We make simplifying assumption of constant monthly use: the energy savings is simply 9/12 of 164 Mwh/yr plus 3/12 of 101 MWh/yr, or 150 MWh/yr, 52% of estimated pre-retrofit annual fan energy use.

[9]Average rate for consumption only, taken from bills for the period 3/87 through 2/88.

payback calculations are shown in Table II. Since the demand is billed on a "building heating service" rate, which includes a demand credit and minimum *billed* demand, the avoided billed demand is less than 20 kW. For nine months in the period 3/87 to 2/88, this amount would have averaged 13.4 kW, and that is what is used in this calculation.

At 2.5" DSP, a simple payback period of 7.7 years is too long to make VSDs a good investment. Property managers consider two to three years to be attractive. Even if the flow rates for the east side fan were comparable to those for the west side, the payback would still be too long at 6.1 years. At a reduced DSP of 1.5", the savings increases to yield a payback of 4.6 years (or 4.3 years if AC-2 performs as well as AC-1).

If the DSP is set manually over a range between 1.5" and 2.5", depending on load, the savings should fall somewhere in between. If it were controlled automatically by an energy management system using feedback from the terminal boxes, savings could exceed the prediction based on maintaining 2.5" throughout the summer, since peak loads are not seen during much of the cooling season (although the demand portion of the savings would probably not increase). The benefits of such a scheme, of course, would be influenced by occupancy factors that affect internal loads. Consider, for example, an office far from the building perimeter and therefore subject to a nearly constant load, which contains so much electronic equipment that its terminal box is nearly wide open throughout the year at a given static pressure. It would not be possible cool this office at a reduced DSP without first providing additional or larger terminal boxes, or lowering the supply air temperature. Static pressure control should ideally be viewed as part of an overall optimization of ventilation system design and operation.

CONCLUSIONS

One important result of this study is the quantification of the discrepancy in fan energy use as a consequence of applying power curves for systems that do not maintain constant DSP to those that do. Indiscriminate use of such curves may be a natural consequence in a field where even the most authoritative literature (ASHRAE, 1988) and state-of-the-art building simulation model, DOE-2, have been unable to keep up with the rapid advance of technology. This illustration of the technology/information time lag makes a good case for improvements in this area.

The economics of installing VSDs point to their use in new construction rather than as a retrofit item. Their substantial energy savings will not offset the high cost until economies in production can reduce this, which will happen as they become more popular.

For the VSDs, the magnitude of the additional annual savings due to lowering DSP (164 MWh - 101 MWh = 63 MWh) is two thirds of the savings resulting from the variable speed drives alone (101 MWh), while lowering DSP in a VIV system has little benefit, indicating a synergistic relationship between the two measures. Manually resetting DSP, a low-cost measure, is not feasible if frequent adjustments are required. While the combined savings are probably not high enough to justify the cost of a control system retrofit that would enable automatic DSP reset, they do bolster the feasibility of VSDs with DSP reset in new construction where electronic control is used.

We have only cursorily explored the question of how varying DSP set point affects energy use in a VIV system, and have obtained results that we cannot interpret from a physical standpoint. If these results can be generalized, however, they have important implications for ventilation system control—DSP reset in a VIV system is not worth any added expense unless it is expected that VSDs will be installed at a point in the future.

Precise DSP control can not be achieved without the addition of some sort of electronic control system, with communication from each of the terminal boxes to ensure that none of them are starved for air. Such a control system is not at all common in buildings today, and is in fact just becoming possible with the introduction of direct digital control (DDC) terminal boxes (although conceivably a system could simply use switches to determine whether any boxes were starved for air). This leads to many interesting questions involving strategies for such a system, such as the combined optimization of static pressure, air flow rate and supply air temperature, which must be dealt with in further ventilation system research.

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FIGURES

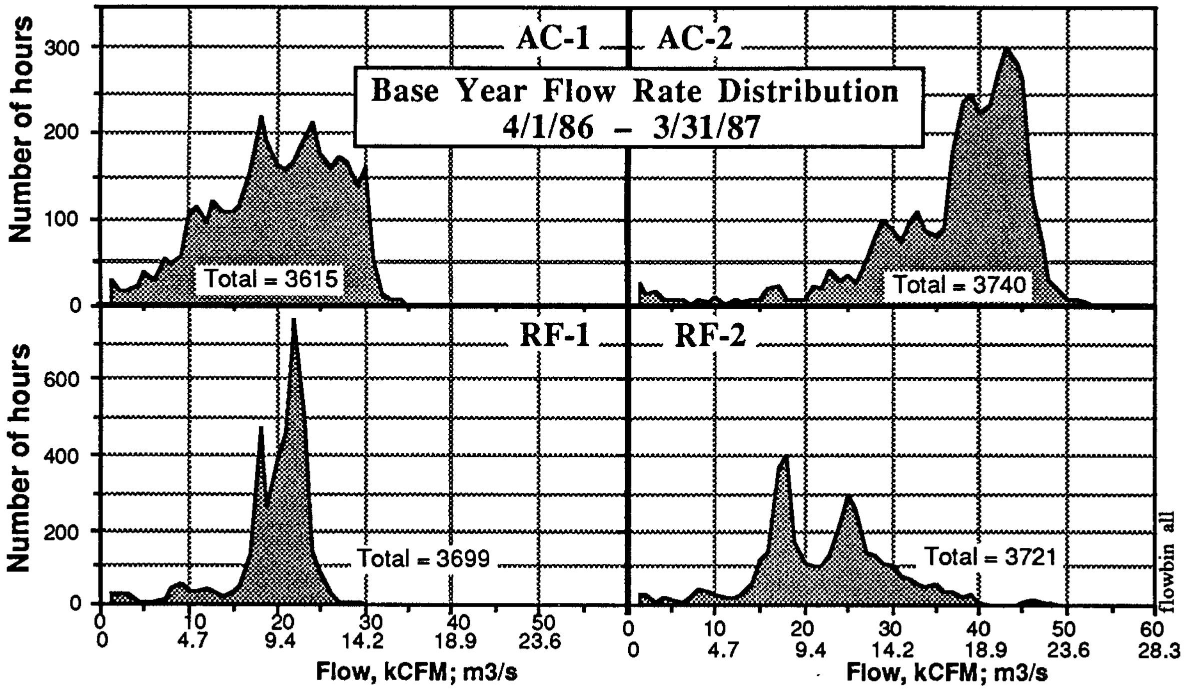

Figure 1. Histograms of air flow rates for supply fans AC-1 and AC-2, and return fans RF-1 and RF-2. Data used are hourly samples taken during the base year. The hours were normalized to account for bad or missing data.

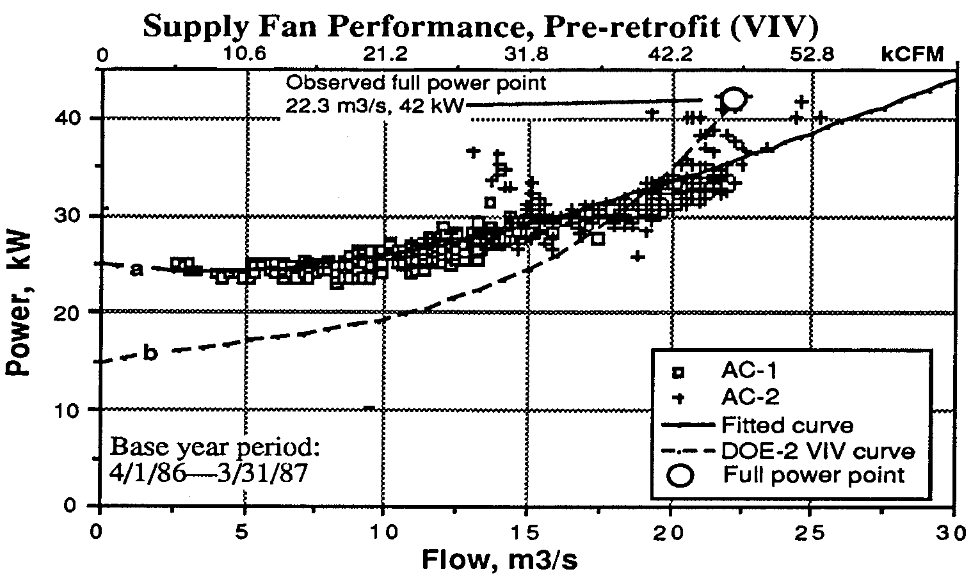

Figure 2. Pre-retrofit supply fan performance, showing a sample of hourly data for the period (curve (a) was fitted using all data in the period). Curve (b) indicates performance predicted by DOE-2, which does not allow for maintenance of duct static pressure The corresponding fan static pressure curves are shown in Figure 4.

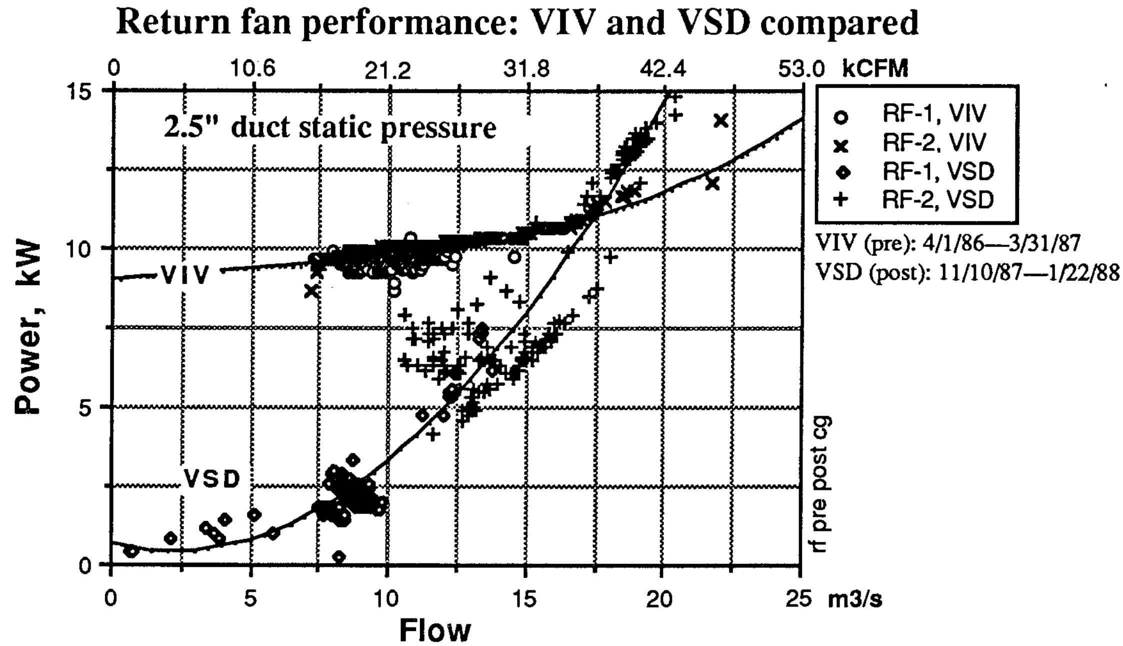

Figure 3. Return fan performance before and after installation of variable-speed drives. Points shown for the pre-retrofit period are a sample of the whole set of hourly data.

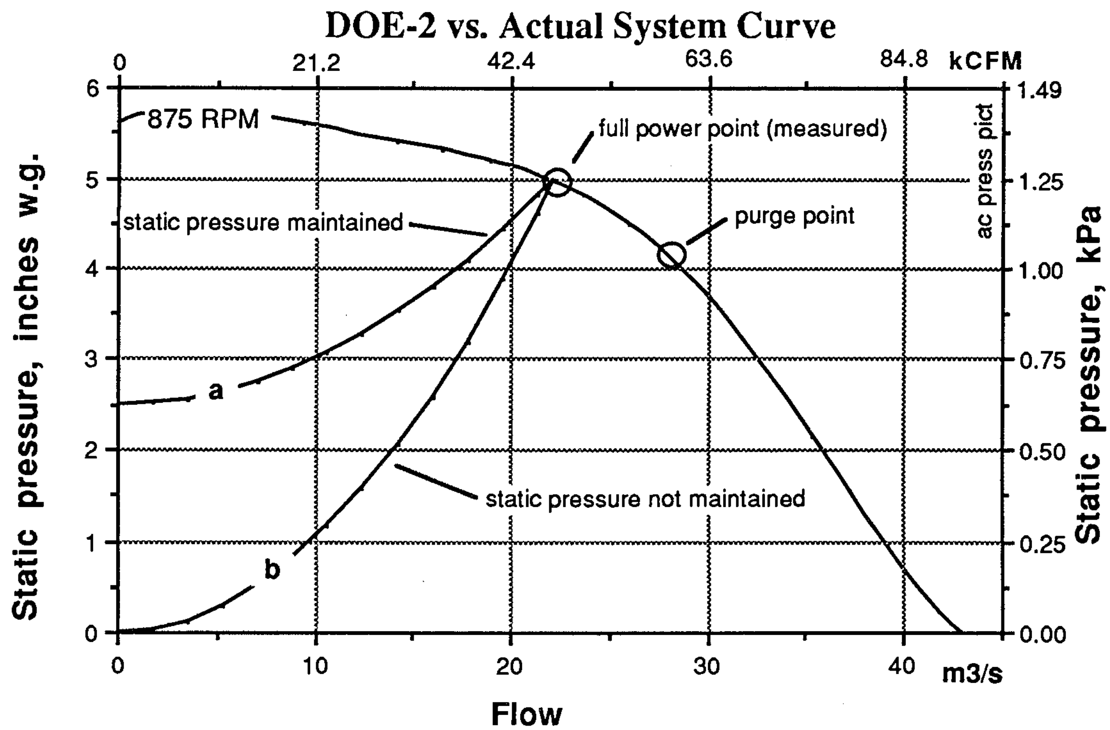

Figure 4. Supply fan system curves for the actual distribution system (a), where static pressure must be maintained, and DOE-2 model (b), which does not allow for constraints on static pressure. The corresponding power curves are shown in Figure 2.

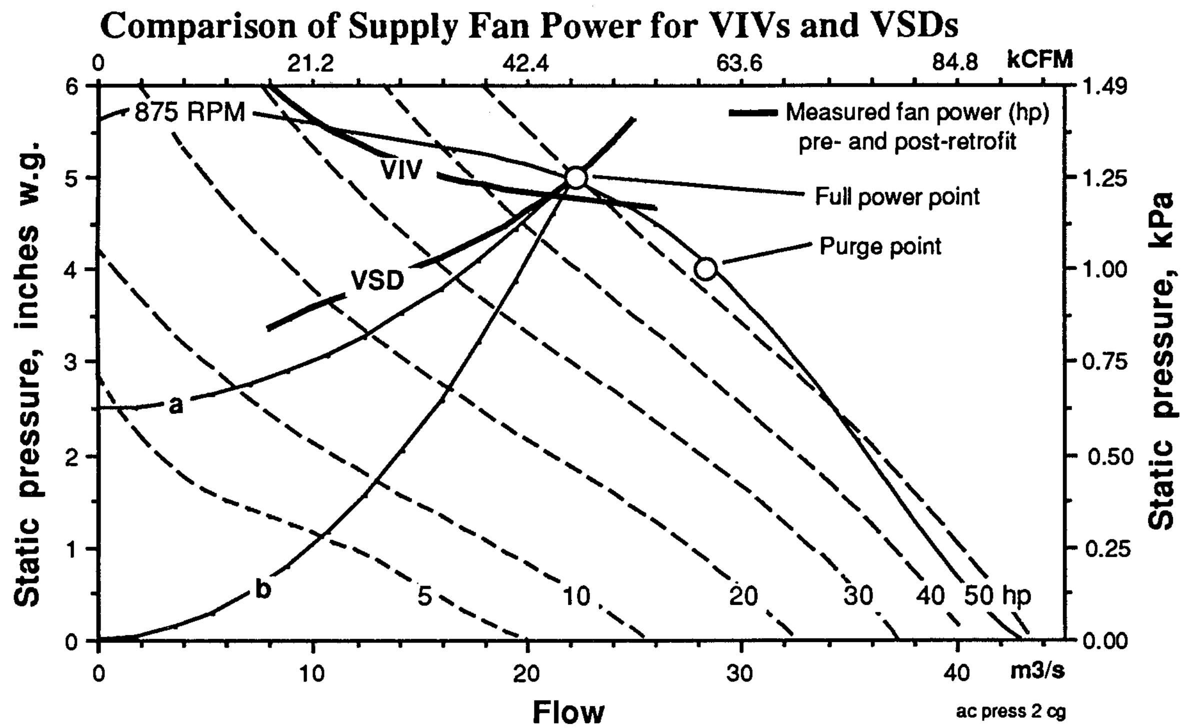

Figure 5. The effect of the installation of the variable speed drives on fan power, shown on pressure-versus-flow axes. The parabolic curves originating at the full-power point are system curves for controlling flow with and without maintenance of duct static pressure at 2.5" (Figure 4). The VIV and VSD curves are the polynomial fits shown in Figure 6. (Source of constant power and rpm lines: Trane Air Conditioning, Inc., fan 44AFDW).

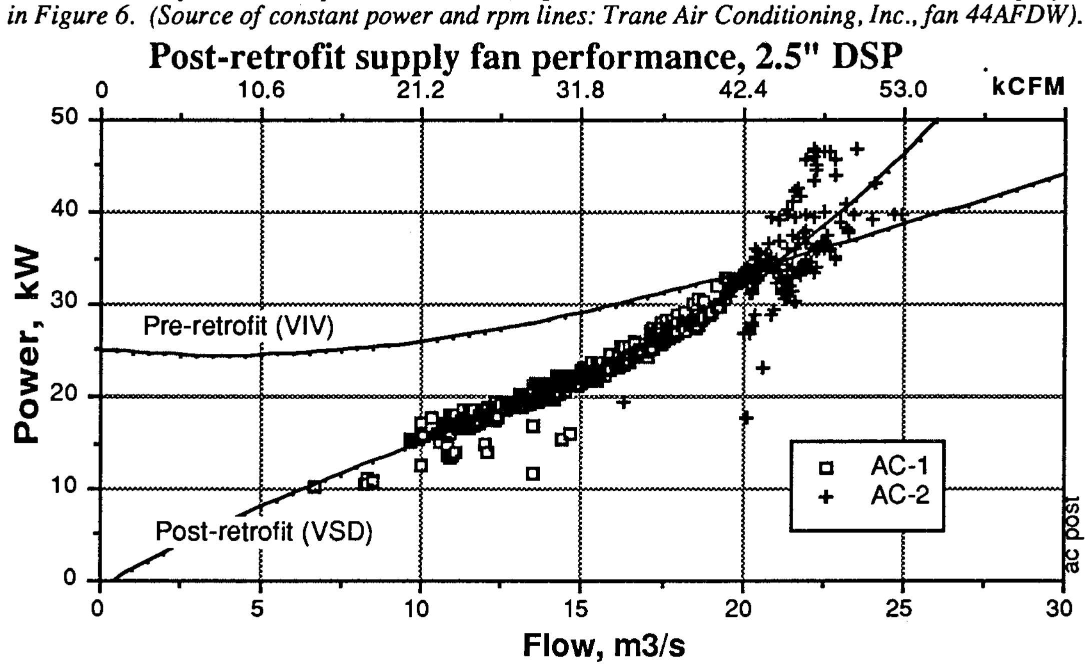

Figure 6. Supply fan performance before and after installation of variable-speed drives. Pre-retrofit curve is a third-degree polynomial fit through a year of hourly data, for the base year period 4/1/86 to 3/31/87. Post-retrofit data are hourly samples for the period 11/10/87 to 12/30/87. The duct static pressure set point was 2.5" WG for data shown.

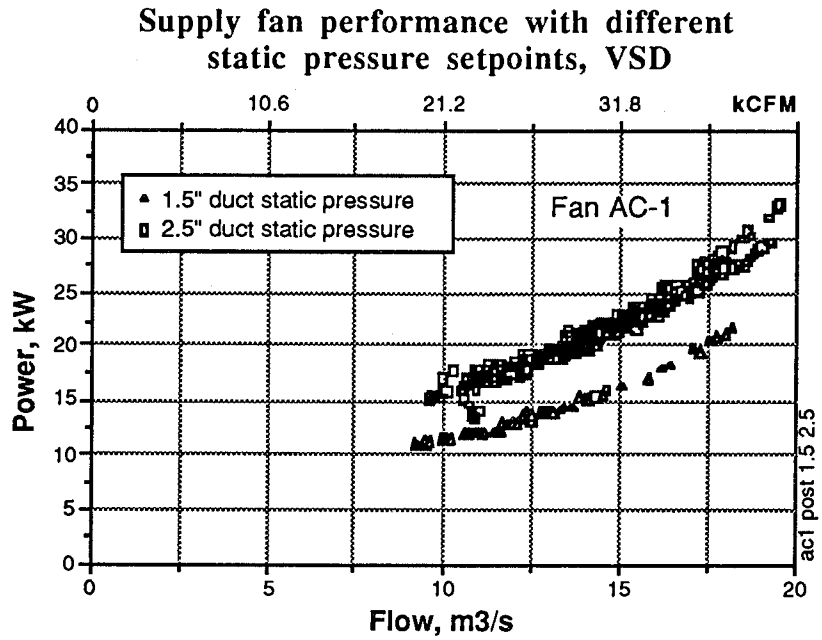

Figure 7. *Comparison of post-retrofit fan performance showing the effect of changing the duct static pressure set point from 2.5" to 1.5". Data for the 2.5" set point are hourly samples from the period 11/10/87 to 12/22/87; data for the 1.5" set point are from the period 12/23/87 to 12/30/87.*

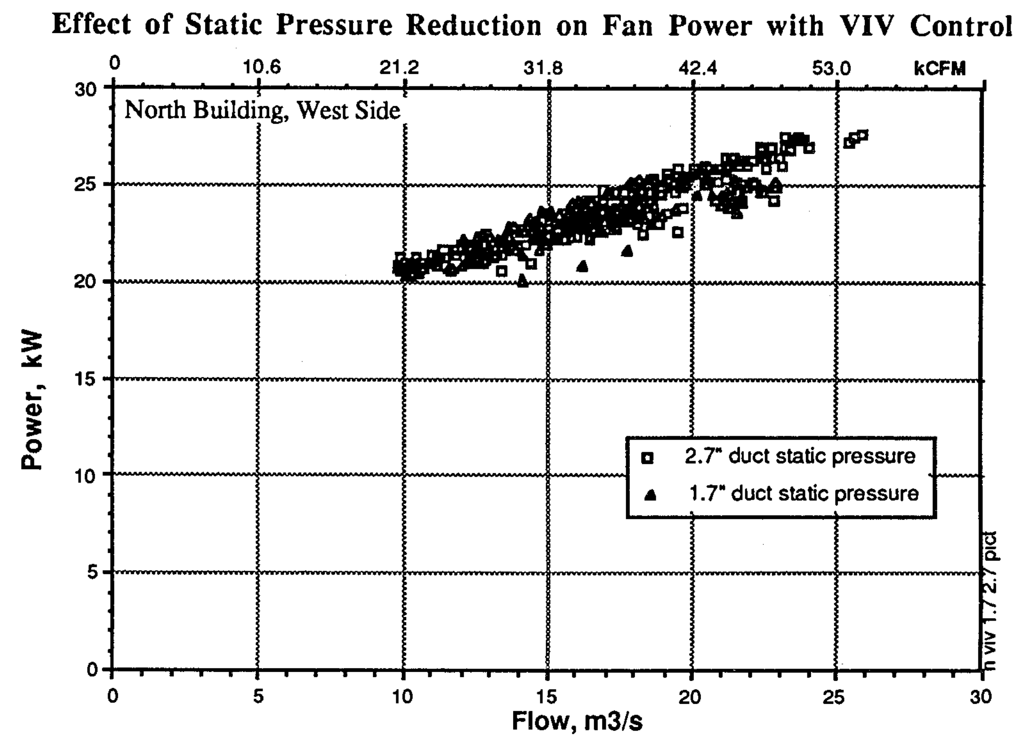

Figure 8. *Comparison of fan power at two duct static pressure set points in a VIV system. Data are hourly samples from a building similar to the retrofit building (with identical fans), for the period 5/2/88 to 6/11/88.*