Variable Speed Drives: Improving Energy Consumption Modeling and Savings Analysis Techniques

Scott L. Englander, New England Power Service Company
Leslie K. Norford, Massachusetts Institute of Technology and Tabors, Caramanis & Associates

A new basis is presented for performance evaluation of variable speed drives (VSDs). A major New England utility initiated this study in order to improve estimates of demand and energy savings achieved by VSDs installed through its conservation programs. Estimates of energy consumed by an electric motor are comprised of two major components: the load profile of the particular application and power vs. load curves. A review of EPRI's and manufacturers' savings estimation software showed that the power curves used are often erroneous or not sufficiently application-specific to yield accurate results. In addition, no guidance is provided for estimating site-specific load profiles, which strongly affect energy calculations.

We propose building a new energy-based taxonomy which groups VSD applications according to power-curve characteristics rather than end use. Equations for power vs. load profiles (using constant-speed and VSD control) are presented for each category in the taxonomy. These curves agree closely with measured data. To predict savings for building VAV systems (a common application of VSDs), we model load profiles as a function of outside temperature. We show how model parameters can be determined for specific installations.

Introduction

A New England electric utility and its retail operating companies provide electricity to customers in Massachusetts, Rhode Island, and New Hampshire. One of the utility's strategies for meeting increasing needs for power is to provide technical and financial assistance to its commercial and industrial customers for equipment and building energy efficiency improvements. The utility currently provides financial incentives to customers for the installation of VSDs. To determine whether and how to continue to include VSDs as a standard measure in its commercial and industrial conservation programs, set appropriate financial incentive levels and criteria, accurately predict energy and demand savings, and demonstrate savings to regulators and other interested parties, the utility needs to better quantify the energy and demand savings and costs involved in an application-specific manner.

The overall objectives of the work described in this paper were:

- to provide a new basis and procedure for cost-effectiveness screening and performance evaluation of VSDs installed through the utility's programs
- to be able to provide customers with a reliable assessment of the energy benefits of installing VSDs, using a simplified procedure for predicting installation-specific savings based on limited information
- to help the utility differentiate its prescriptive incentives for VSDs according to application type
- to develop a better understanding of how motor, drive, and system efficiencies interact, and examine the potential benefits of integrating the utility's approach to drive train and system efficiency improvement
- to identify applications for which further study, including instrumented monitoring, is needed.

An Energy-Based Taxonomy of VSD Applications

The energy consumed by a motor can be estimated by combining the load profile of the particular application with power vs. load curves, with and without the VSD. It is well known that power curves differ considerably among applications, but there is a lack of good information characterizing the curves. Applications are typically grouped by end use (e.g., HVAC fans, HVAC pumps, industrial processes) without regard to power curve characteristics. For example, although VAV supply and return fans are often lumped together for analysis purposes, the energy use characteristics of the supply fan are perhaps closer to those of a building hot water pump (both are controlled to regulate pressure), while the energy use signature of the return fan may be more similar to that of a boiler forced draft fan (both regulate flow). Other researchers have already initiated efforts at differentiating analysis among different types of centrifugal pumping applications (Armintor and Connors 1987). We propose widespread adoption of a new energy-based taxonomy of VSD applications in order to reduce uncertainty in energy use and savings estimates for individual categories.

Three features distinguish different energy-based categories of VSD applications: the type of device driven by the motor, system type, and the controlled variable. The first feature, device type, separates centrifugal machinery from all other devices, on the basis of torque variations with load (EPRI 1991). Traction motors, cranes and lathes require constant power over all loads. Conveyors require constant torque: motor shaft power, the product of torque and speed, varies linearly with motor speed when a VSD is installed. Torque for positive displacement pumps, hoist motors and winches varies linearly with flow; power therefore varies quadratically with flow. Energy savings for VSDs installed on centrifugal devices are substantially larger, because for centrifugal devices torque varies as the square of flow and power as the cube of flow (absent pressure constraints, a key point that is the basis of the second feature, discussed below), resulting in better part-load efficiencies under variable-speed operation. Centrifugal applications of VSDs are more common as well. Larson and Nilsson (1990) cite a U.S. Department of Energy study (1980) which estimates that three-quarters of all pumps in the U.S. are centrifugal and that this type of pump accounts for 90 percent of all pumping energy; positive displacement pumps make up the remainder. For these reasons, this discussion will deal largely with centrifugal devices.

System type, i.e., the characterizing relationship between pressure and flow in the system, is the key element of the taxonomy; our breakdown builds on the classification of Armintor and Connors (1987). Mechanical power can be determined from knowledge of how pressure and flow vary. Clearly, if flow is constrained, the cubic dependence of power on flow cannot be realized in practice, affecting VSD savings. Less universally appreciated are pressure constraints which also restrict VSD savings. For both of these reasons, all centrifugal devices do not exhibit identical power versus load curves. Table 1 distinguishes variable-flow and constant-flow systems; for each type, pressure can in principle be related solely to flow, include a flow-independent offset, vary independently, or take a constant value. Within each of the five categories for which examples are provided in Table 1, pressure or flow may be the controlled variable, the third feature of interest. Knowledge of the controlled variable is often useful in helping to place a system into the correct category.

We describe each of the five possibilities as follows:

1. **Variable-flow, pressure determined solely by flow.** As shown in Figure 1, this category is defined by a pressure versus flow system characteristic that is fixed (i.e., the coefficient of resistance is constant), quadratic in shape and, crucially, includes the zero-flow zero-pressure point. For applications in this category, flow is the controlled variable and is adjusted to regulate such variables as temperature (hydraulic system cooling pump) or oxygen content of the exhaust from a combustion chamber (boiler forced draft fan). Pressure is not constrained in any way and depends on flow via the fluid dynamics associated with frictional pressure losses for turbulent flow in ducts or pipes, i.e., pressure is proportional to the square of flow.

2. **Variable flow, with constant pressure offset.** In many applications, a fan or pump is constrained to overcome a constant pressure at all flow rates, unlike the situation described in Figure 1. Total pressure across the fan or pump includes both the constant component and a contribution from pressure losses due to friction, proportional to the square of flow. Applications can be distinguished by controlled variable.

   Pressure-regulated systems include HVAC supply fans, chilled-water or hot-water circulating pumps, and some city water distribution pumps. In each case the prime mover serves multiple end uses, some or all with variable flow demands, which have minimum pressure requirements at the inlet to throttling equipment (valves or dampers) used to regulate flow. Such pressure requirements also exist for boiler feedwater pumps, which must pressurize the feedwater header to a point where a large quantity of water could rapidly flow into the boiler if there were a pressure failure to the atmosphere.

   The hydraulic pressure of a vertical column of fluid is negligible for air-distribution systems but often significant for water-pumping systems. This hydraulic pressure, known as static head, must be overcome by a pump before flow can be established, in open piping

Table 1. An Energy-Based Taxonomy of VSD Applications

Pressure Characteristic	Flow Characteristic: Variable Flow	Flow Characteristic: Constant Flow
Pressure ~ flow^2 (flow resistance coefficient is constant)	• VAV return fan • boiler forced draft fan • hydraulic system cooling pump • some process pumps	
Constant pressure, plus flow-dependent term	• VAV supply fan • chilled or hot water circulation pump with two-way valve terminals • municipal water pump • boiler feedwater pump • wastewater pump • boiler induced draft fan • dust collector fan • generalized process pump	
Variable pressure	• pump drawing from tank of varying level	• flow-regulated pump drawing from tank of varying level
Constant pressure		• chilled or hot water circulation pump with bypass • pump with recirculation loops

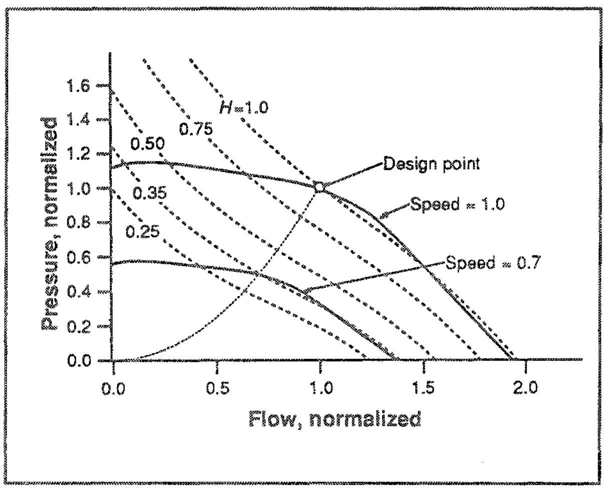

Figure 1. Generalized Fan Curve, in Dimensionless Form. The system characteristic (dotted) intersects the origin, indicative of a fixed resistance coefficient (absence of pressure regulation).

systems. For closed systems the static pressures associated with vertical piping balance (fluid goes up and down any changes in elevation) and the pump must only overcome frictional pressure drops. Flow-regulated systems subject to fixed or nearly fixed hydraulic pressures include process pumps moving liquids between tanks at different elevations, at flow rates that change with the process. Sewage pumps that maintain wet-well water level also fall into this category, as do municipal water supply pumps that regulate water level in storage tanks.

3. **Variable flow, variable pressure.** An example in this category is a pump providing a variable flow rate, drawing from a tank in which the water level fluctuates independently of the flow rate.

4. **Constant flow, variable pressure.** The pump in category 3 may be regulated by a flow sensor connected to a throttle valve in a bypass loop, or to a VSD, to maintain flow at a prescribed quantity.

5. **Constant flow, with constant pressure.** In this category, for example, a bypass loop with a diverting valve varies the flow to a single heating or cooling coil. With the total system pressure drop nearly

constant throughout the operating range of the diverting valve, the pump circulates a constant flow of water. Application of a VSD in this case requires that the diverting valve block any flow to the bypass loop and that flow regulation to the coil be accomplished by motor speed adjustment. The system then becomes a variable-flow type, with pressure determined by flow alone unless there is significant static pressure to overcome.

This category also includes pressure regulation via a bypass loop at the pump, with a throttling valve in the bypass piping. The valve is controlled to maintain a constant pressure at the pump discharge, as downstream valves are adjusted. As with the previous example, the pump operates at a single point, because pressure, flow and speed are all constant. With a VSD installed in lieu of the recirculation loop, the system changes into the variable-flow type.

The proposed taxonomy has reduced variations among nearly all of the utility's current VSD applications to five categories of centrifugal machinery: variable flow, with pressure determined by flow alone, or by flow and a no-flow offset as well, or varying independently; and constant flow, with pressure determined by flow or held to a constant value. For the last case, the electrical power takes on a fixed value and the retrofit must include not only the VSD itself but system changes required to block flow to bypass loops. The problem of estimating VSD retrofit savings for variable-flow, variable-pressure systems boils down to the following:

1. Establish two general power versus flow relationships for VSD control, one for the case where pressure is determined by flow alone and one where there is a pressure offset. The latter will be shown to include the special case of variable pressure offset, a control strategy for minimizing power required by VAV supply fans.

2. Establish power versus flow relationships for constant-speed control, including throttling valves or discharge dampers, inlet vanes, and bypass or recirculation loops.

3. Establish flow, or load, distributions, which will strongly depend on application.

4. Combine the power-flow relationships and the flow distributions to determine energy consumption and peak power.

The taxonomy does not address load distribution but has simplified and organized the process of defining power versus load relationships.

Existing Software for Estimating VSD Savings

Computer software is available to perform the calculations associated with combining load profiles with power vs. load curves. To determine electrical energy consumption, these packages include default or user-assignable values for efficiencies of drive trains (belts and pulleys, for example), VSDs, and the motor itself; these efficiencies may be single values or functions of load. We reviewed software packages from a fan manufacturer, the Electric Power Research Institute (EPRI), and two manufacturers of VSDs. Table 2 summarizes features of these packages to provide a clear basis for comparison. We highlight three key differences:

1. Pressure offset for fans
2. Treatment of efficiencies
3. Load distribution correlations with process loads

Pressure Offset for Fans

Of the packages reviewed, only the fan manufacturer's software accounted for fan systems using pressure regulation (i.e., variable-flow, pressure-offset). For building-ventilation systems, previous work has highlighted the energy impact of static-pressure set point (Englander and Norford 1992). The fan manufacturer's package allows the user to vary the pressure set point associated with pressure regulation systems; this parameter can assume the null value (no pressure regulation), permitting analysis of variable-flow systems for which pressure depends on the square of flow, the sole category for which the remaining three software packages are suitable.

Fan shaft power versus load curves for VSDs, shown in Figure 2, closely match when there is no pressure set point. Load-dependent motor and drive efficiencies have been excluded from the fan manufacturer's data, to highlight differences in shaft power. Manufacturer B's power curve is the cubic relationship dependence on flow that is based on constant fan efficiency under all loads; this load curve is matched precisely by the zero-pressure-offset curve from the fan manufacturer. Estimates from the second VSD manufacturer are slightly higher, particularly at low flows, the result of load-dependent efficiencies that could not be removed from the software. EPRI (1989) specifies a pressure versus flow relationship

Table 2. Key Features of VSD Savings-Estimation Programs

Features	EPRI-I	EPRI-II	Mfgr. A (fans)	Mfgr. B (VSDs)	Mfgr. C (VSDs)
Application					
Fan	•	•	•	•	•
Pump	•	•		•	
Pressure offset for fans			•		
Power vs. load					
User	•	•	•	•	
Default				•	•
Motor/drive efficiencies					
Fixed				•	•
Load-dependent	•	•	•		
Load distribution					
User	•		•	•	•
Correlated with Process		•			

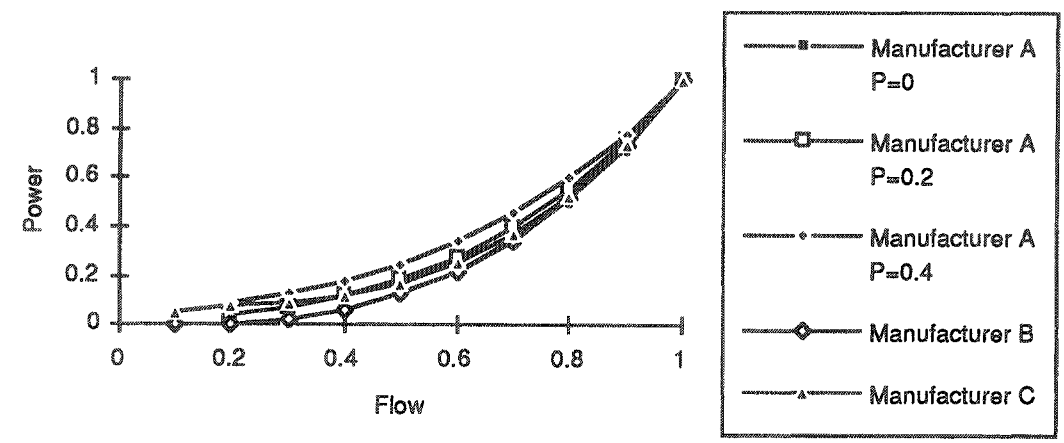

Figure 2. Manufacturer's Estimates of Fan Power as a Function of Flow for VSD Control. Manufacturer A's estimate when there is no pressure set point equals that of Manufacturer B. Manufacturer C's estimate shows the influence of motor and drive efficiencies that could not be separated. Pressure set points of 0.2 and 0.4 produce small increased in power.

appropriate for no pressure regulation and requires the user to provide fan power from manufacturer's data; this approach would yield a curve equal to one of those shown for no pressure regulation. As calculated by the fan manufacturer, shaft power for a given flow rises noticeably as pressure set point increases. The effect of pressure set point will be accentuated in the savings estimates, indicating the importance of distinguishing variable pressure and constant pressure applications. Fan manufacturer's data were taken from a specified fan (airfoil blades, backward inclined); the data are typical of this class of fan but differ markedly from data for fans with forward curved blades, making it important to note blade type when estimating part-load performance.

Figure 3 shows estimated power for inlet-vane control, a common flow control used with constant-speed fans. For no pressure set point, data from the fan manufacturer and VSD manufacturer C agree closely, while manufacturer B has an erroneously simplified linear relationship. Pressure set point has less effect on inlet-vane performance than on VSDs, as previously noted by Englander and Norford (1992). More telling than estimations for inlet vanes or VSDs individually are the differences between inlet vanes and VSDs which represent the energy savings associated with a given load. Figure 4 shows these differences. For no pressure set point, the fan manufacturer's estimate and that of VSD manufacturer C are remarkably close, while manufacturer B's savings estimates favor higher flow rates. Pressure set point makes a significant difference, decreasing savings by about 25 percent at 50 percent flow.

Software provided by one of the VSD manufacturers distinguishes fans from pumps, with the latter subject to substantial static pressures due to the weight of vertical columns of water. It might appear appropriate to take advantage of the taxonomy and use pump software for pressure-regulated fans, given that both are examples of variable-flow systems with a pressure offset. The performance of discharge valves matches that of discharge dampers, used in some packaged air handlers, but pumps lack a control mode analogous to the inlet-vane control typical of many constant-speed fan systems. The pump software allows the user to specify static pressure, due either to hydraulic pressure or control set point or both. The effect of this set point is shown in Figure 5, which includes the part-load performance of discharge valves characteristic of pre-retrofit conditions. At any flow, the savings relative to throttling valve control are strongly influenced by static pressure. As will be seen later, the performance of throttling valves shown in Figure 5 differs from that calculated by Armintor and Connors or via a simplified pump curve developed by the authors.

The differences among software packages in their treatment of power versus load curves points to a need for savings estimation software that:

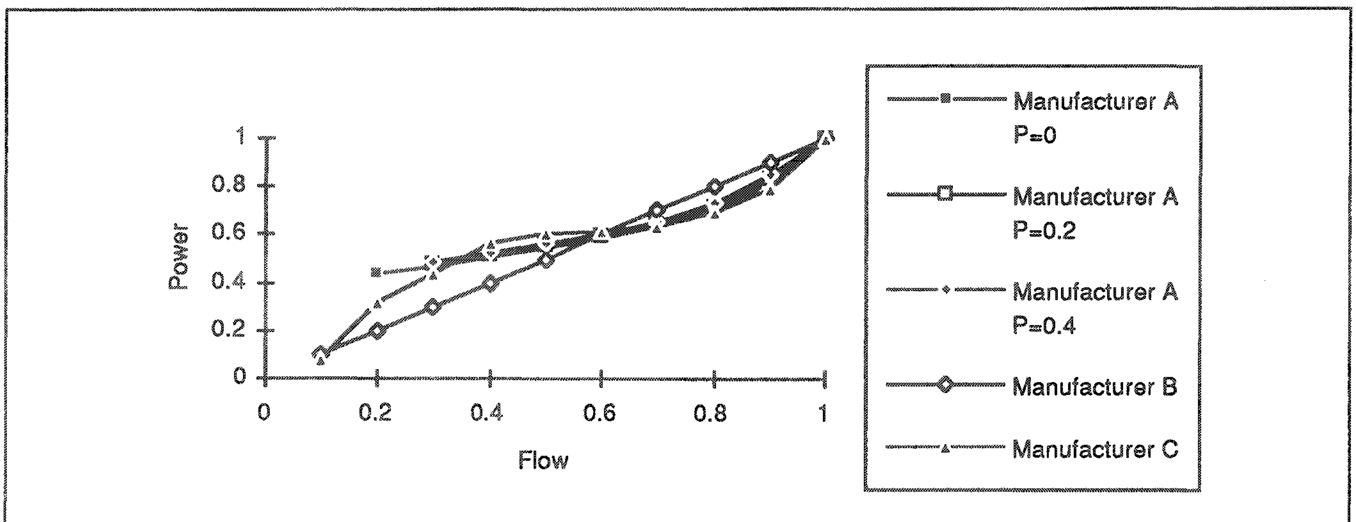

Figure 3. Manufacturers' Estimates of Fan Power as a Function of Flow for VIV Control. For no pressure set point, manufacturers A and C give nearly identical results, while manufacturer B employs a simplified, linear power vs. flow relationship. Again, increases in pressure set point produce small changes in power.

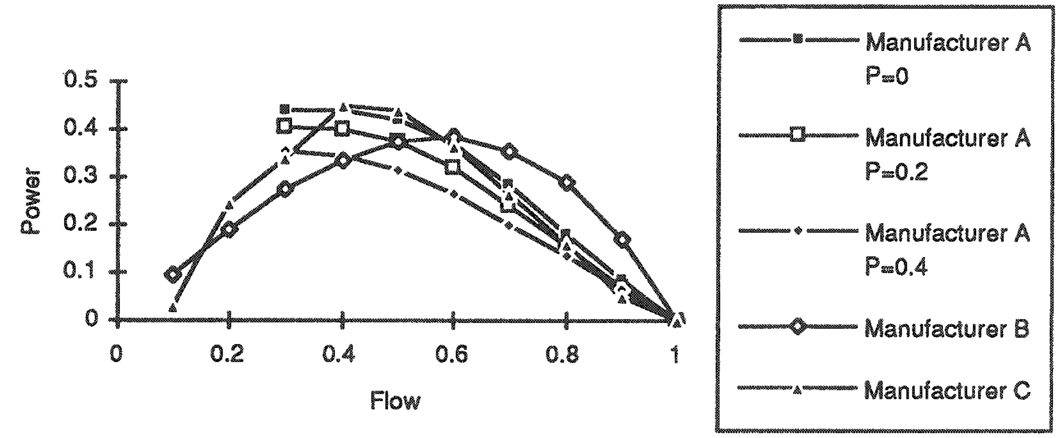

Figure 4. Manufacturers' Estimates of Fan Power Savings, as a Function of Flow, for Replacing VIV with VSD Control. For no pressure set point, manufacturers A and C again agree closely, with the estimate of manufacturer B shifted toward higher savings at higher flow rates.

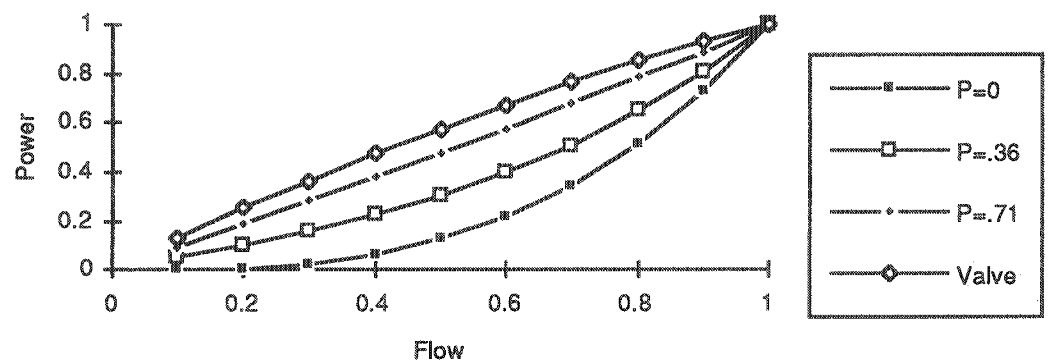

Figure 5. Pump Performance Under VSD Control, with Static Pressures of 0, 0.36 and 0.71 of the Maximum, and with Throttling Valve Control

1. does not artificially separate pumps and fans but rather distinguishes devices on the basis of common power versus load curves; and

2. substantiates the form of the curves with references to solid experimental work.

Motor and Drive Efficiencies

The electrical power required by a motor exceeds the shaft power at the fan, pump or other device due to losses in the drive train, the motor itself, and, if present, the VSD. Drive train losses depend on application and are non-existent when motors are directly coupled to pumps or fans (particularly axial). For centrifugal fans connected to motors via sheaves and belts, the fan manufacturer suggests a loss that remains at 3 percent until loads drop below about 40 percent, at which point the efficiency drops drastically.

EPRI and the fan manufacturer use similar curves for motor and VSD efficiency, with EPRI values shown in Figure 6. EPRI also provides guidance in selecting a motor efficiency at full load, as a function of motor size. The dimensionless part-load data are used to derate the full-load value. Motor efficiency is given as a function of

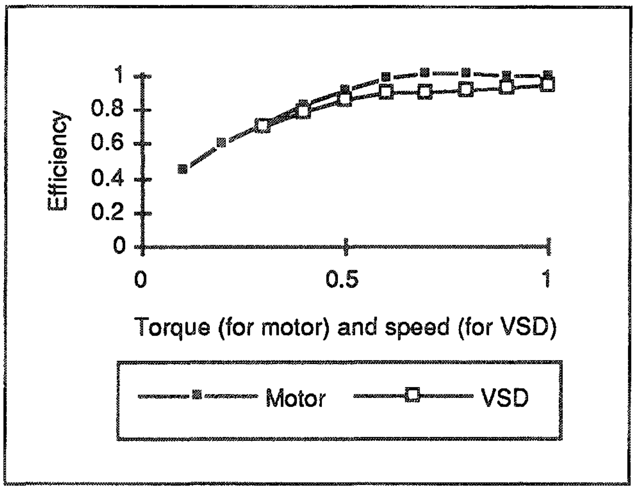

Figure 6. Motor Efficiency as a Function of Torque and VSD Efficiency as a Function of Motor Speed, as Presented in EPRI's ASCON-I VSD Savings Estimation Program

torque alone. The fan manufacturer requires rated motor horsepower, to calculate inefficiencies due to motor oversizing, and also provides motor efficiency curves for standard and high-efficiency motors.

It appears that only one of two VSD manufacturers accounts for motor and drive efficiencies that vary with load, as previously noted. It is curious that the other manufacturer does not address the topic at all, save for required single point efficiencies for motor and VSDs; there is no good reason why any software should not account explicitly for efficiency variations.

It is also worth pointing out the practical cases where assumed constant values for efficiencies may cause minimal degradation in the accuracy of estimated energy consumption. These cases include, first, applications where torque and motor speed remain high over most of the load distribution and, second, applications where power is so low when torque and speeds are low that efficiency corrections make little impact on annual energy calculations. Constant-speed control eliminates the issue of VSD efficiency; for the motor, torques of at least 50 percent of the full-load value will yield a nearly constant efficiency. Torque varies with shaft power for constant speed operation and, as will be apparent later, is typically below 50 percent only at flows so low (less than 30 percent) as to be rarely achieved in practice.

For variable-speed operation, torque and speed can both fall below 50 percent of rated values, particularly for applications where pressure depends on the square of flow. But shaft power is only 12.5 percent of maximum at 50 percent flow and energy calculations with a flow distribution centered at 50 percent will be weighted strongly toward the higher flow rates, where efficiencies are nearly at their full-load values.

Airflow Distributions

Default airflow histograms for the two VSD manufacturers are plotted in Figure 7. Manufacturer B's histogram is narrowly distributed about 70 percent of full flow while manufacturer C's is more broadly centered at 55 percent of full flow. The VSD manufacturers provide no justification for their distributions; users may easily customize them and manufacturer B no longer provides a default distribution. The fan manufacturer, by contrast, supplies several distributions with explanations for each. Figure 8 shows the three distributions for variable-air-volume building ventilation systems. More than 5000 hours per year of operation, with no supply-air temperature reset, produces the lowest distribution, because prolonged operation reduces the need for large flows during start-up periods. Fewer hours of operation increases the number of hours at higher flows. Supply-air-temperature adjustments produce the distribution with the most hours at the highest flow rates.

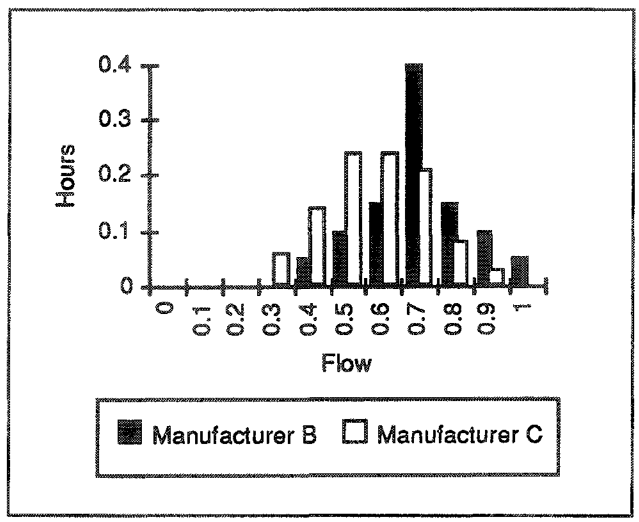

Figure 7. VSD Manufacturers' Default Flow Histograms for Computing Savings due to VSD Controllers. Manufacturer B provides no default histogram with its latest software and neither manufacturer provides the user with guidance in constructing an application-specific flow distribution.

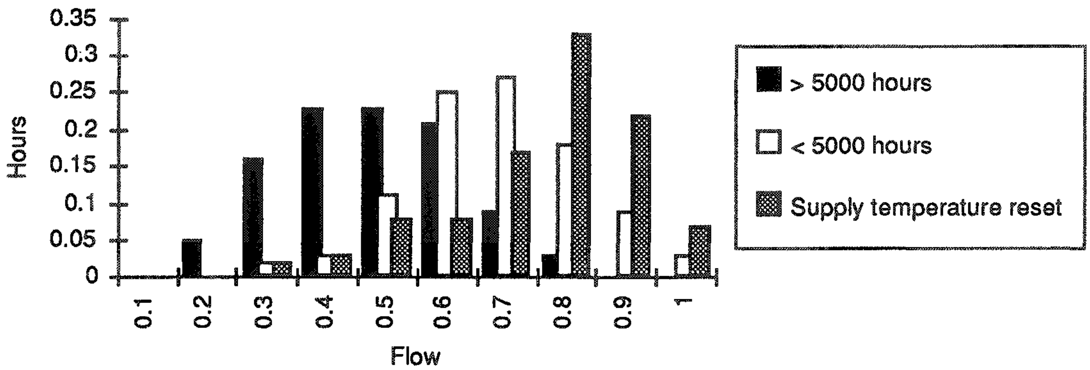

Figure 8. Default Flow Distributions for Variable-Air-Volume Building Ventilation Systems, Taken from a Fan Manufacturer. Shorter hours of operation include more start-up hours with higher airflows, while supply-air temperature reset control causes airflows to increase further.

Using these flow histograms, savings estimates can now be made for the retrofit of inlet vanes with VSDs, using power versus load curves for the two VSD manufacturers that are appropriate for those applications where pressure depends solely on the square of flow. Savings, shown in Table 3, are substantially higher with lower flow distributions because speed under VSD control approaches the constant speed of VIV control and fan power curves converge for the two control methods. Note that the savings estimates from VSD manufacturer B consistently exceed those of manufacturer C, by 0.05-0.08, or 8-29 percent.

Table 3. Manufacturers' Computed Fractional Energy Savings, for Replacement of Variable Inlet Vanes with VSDs

Flow Distribution by Manufacturer	Fractional Energy Savings	
	Mfgr. B	Mfgr. C
A-1	0.71	0.66
A-2	0.47	0.41
A-3	0.36	0.28
B	0.45	0.38
C	0.61	0.56

The sensitivity of the savings to flow distribution makes site-specific flow information of more concern than part-load motor and drive efficiencies.

For site-specific estimation of VSD savings, either by the customer (prior to VSD application) or by the utility (after VSD installation), site-specific load distribution data would, if easily obtainable, improve the accuracy of the calculations. EPRI's software for use at utility generating stations (EPRI 1990) bases its calculations on readily accessible plant output distributions, and user-supplied relations between plant output and flows through fans or pumps. If flow data are not available, electrical power data for a given pump or fan, as a function of plant output, can be related to flow by going to the pump or fan curve after converting to shaft power via an estimated motor efficiency. In a later section of this paper we propose a similar procedure for building ventilation fans.

Non-Dimensional Representation of Pressure and Power As a Function of Flow or Motor Speed for Centrifugal Machinery

Applications of VSDs to power centrifugal machinery involve motors of varying size and static pressures and flow rates of varying magnitude. To establish the relationship of power to load from a minimal amount of information, it is desirable to develop dimensionless relationships among power, pressure and flow that can be scaled from a few site-specific parameters. Our effort to do so is designed to provide a model comparable to that used by the VSD manufacturer that provided savings-estimation software for pumps, but with data requirements that both customers and utilities can easily meet.

VSD Power as a Function of Flow

Taking a fan manufacturer's fan curves, Figure 1, we simplified the graphical relationships and created the dimensionless form shown in Figure 9. Linear simplifications appear visually to be reasonable surrogates for actual pressure, flow and power relationships and make it possible to derive analytic expressions for power. The intersection of the two lines forming a constant-speed fan or pump curve defines the design point. The dimensionless fan or pump curve serves as a guide in developing analytic rather than graphical expressions for dimensionless fan power as a function of either the flow variable or motor speed. Both cases require as a parameter the static pressure under no-flow conditions. Each dimensionless variable is defined as the variable divided by its value at the design point, e.g., $H = H_{actual}/H_{design}$. Note that in scaling up these equations to actual values, design values of all variables should reference the same point on the fan or pump diagram.

For power as a function of flow, f, the starting point is the parabolic system curve that can be expressed as

$$p = (1 - p_o)f^2 + p_o \tag{1}$$

where p is static pressure and p_o is the pressure set point or static head.

Fan power H has a similar form:

$$H = (1 - H_o)pf + H_o \tag{2}$$

where H_o is the power at $p = p_o$ and $f = 0$.

Note that the product of power and flow could be divided by a dimensionless fan or pump efficiency but we assume that the efficiency is constant and retains its value at the design point, where power, flow, pressure and efficiency all assume the dimensionless value of 1.0. The geometrically simple forms in Figure 9 include lines of constant shaft power that yield slightly higher powers at low flows that the curves in a typical fan or pump curve, as shown in Figure 1. The higher powers produce an effect similar to the more exact description of lower powers and lower efficiencies.

From Figure 9, the power H_o under no-flow conditions and a dimensionless pressure of 1.0 equals the power at a point on the parabola where the pressure is 0.5 and the flow equals $(0.5)^{0.5}$. The more general form of this relationship, applicable for any pressure set point, is

$$H_o = \left(\frac{P_o}{2}\right)^{1.5} \tag{3}$$

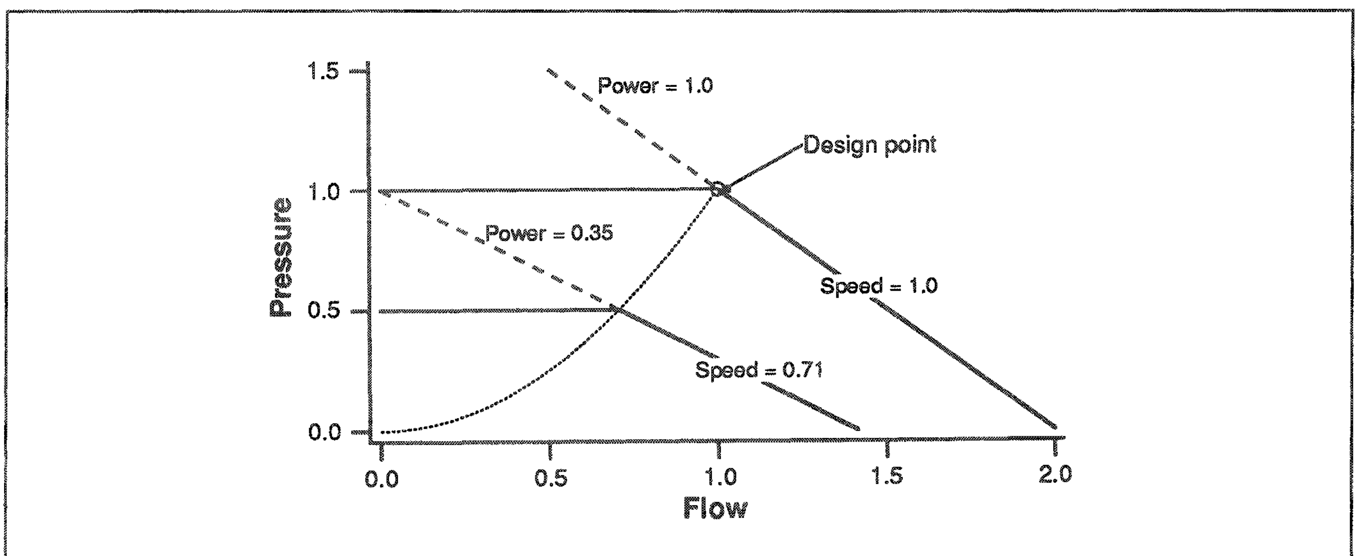

Figure 9. Simplified, Non-Dimensional Relations that Show How Pressure for a Centrifugal Machine Varies with Flow, for Two Constant Rotational Speeds (solid) and Two Constant Powers (dotted). Pressure-flow lines for which power is constant coincide with the sloped portion of the pressure-flow relation for constant speed.

Inserting this expression and Equation (1) into Equation (2), we obtain the desired result:

$$H = \left[1 - \left(\frac{p_o}{2}\right)^{1.5}\right] \cdot \left[(1 - p_o)f^2 + p_o\right] \cdot f + \left(\frac{p_o}{2}\right)^{1.5} \qquad (4)$$

or

$$H = a + bf + df^3$$

where

$$a = \left(\frac{p_o}{2}\right)^{1.5}$$

$$b = p_o(1 - a)$$

$$d = 1 - a - b$$

Note that this equation yields, as expected:

$$H = f^3 \; for \; p_o = 0$$
$$H = 1 \; for \; f = 1$$

In practice, the pressure set point need not remain fixed at a single value. Englander and Norford (1992) described the energy savings that could have been achieved for a building ventilation fan if it were possible to reduce the pressure set point to a lower value. Digital controls now permit automatic adjustment of the pressure set point, to maintain it at the lowest possible value consistent with providing adequate airflow (and cooling) to all areas served by the ventilation system or, in the case of a pump, adequate pressure to all valves downstream. If we assume that the pressure set point varies smoothly from a maximum value at design airflows to a minimum value under no-flow conditions (a trend that is subject to perturbations due to varying building thermal loads) then Equation (4) predicts fan power when p_o is interpreted as the *minimum* static pressure set point and design conditions apply to the *maximum* pressure set point.

VSD Power as a Function of Motor Speed

It is often useful to have power as a function of motor speed when assessing the energy consumption of devices controlled by VSDs. For example, the operator may have no information about flow rates but may know the range of speeds over which the device operates, by observing the display on the VSD housing. For power as a function of speed, ω, recognize from the fan curve that

$$p_o = \omega_o 2 \qquad (5)$$

and hence

$$H_o = \left(\frac{\omega_o}{\sqrt{2}}\right)^3 \qquad (6)$$

Flow is linearly related to fan speed and, in dimensionless form, equals fan speed when there is no static pressure offset. By applying the boundary conditions we find

$$f = \frac{\omega - \omega_o}{1 - \omega_o} \qquad (7)$$

Substituting into the expression for power as a function of flow we determine that

$$H = \left[1 - \left(\frac{\omega_o}{\sqrt{2}}\right)^3\right] \cdot \left[\frac{\left(1 - \omega_o^2\right)\left(\omega - \omega_o\right)^2}{\left(1 - \omega_o\right)^2 + \omega_o^2}\right] \cdot \frac{\omega - \omega_o}{\left(1 - \omega_o\right) + \left(\frac{\omega_o}{\sqrt{2}}\right)^3} \qquad (8)$$

Throttling Power as a Function of Flow

The model for VSD operation applies to throttling valves or discharge dampers when the pressure set point p_o is set equal to 1.0, keeping the speed at a fixed value. In this case, from Equation (4),

$$H = 0.646f + 0.354 \qquad (9)$$

Inlet-Vane Power as a Function of Flow

A model for inlet-vane control of fans starts with the common assertion that inlet vanes at a constant angle produce a new relationship of fan pressure and flow. A family of curves, each pinned at the zero-flow point, results from variation of the vane angle. We model these curves with the straight-line approximations shown in Figure 10.

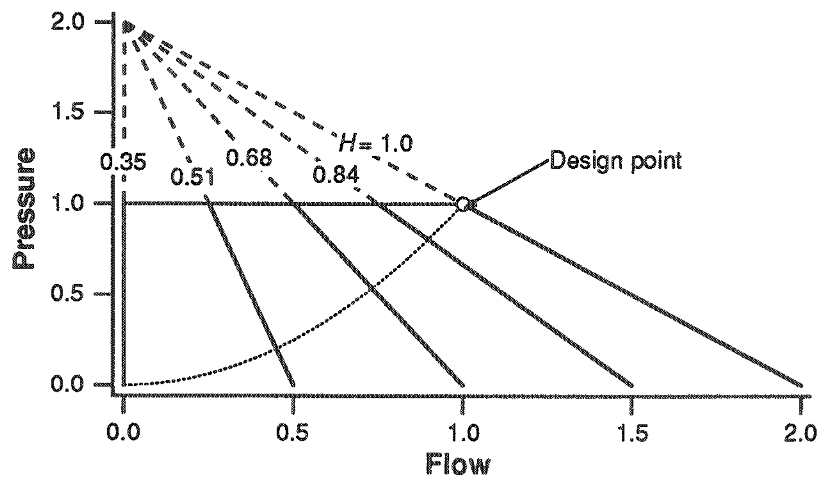

Figure 10. Variable-Inlet-Vane Fan Curves. Each pressure-flow line, along which power is taken as constant, corresponds to a fixed inlet-vane position. Extension of the constant-power lines is used to establish the functional form of the lines and determine the fan power at any point along a system curve.

For any fixed vane position, the pressure-flow relationship is

$$p = \frac{2(1-f)}{f_o} \tag{10}$$

where f_o is the zero-pressure intercept. The intersection of the fan curve described by Equation (10) and the system curve described by Equation (1) determines f_o for any flow f and static pressure p_o:

$$f_o = \frac{2f}{2-\left[(1-p_o)\,f^2 + p_o\right]} \tag{11}$$

The shaft power along the constant-pressure part of the curve for wide-open vanes determines the power associated with each curve for fixed vane position:

$$H = (1-H_o)\,f + H_o \tag{12}$$

From the geometric similitude shown in Figure 10, $f = f_o / 2$ and H_o is set by its value of 0.35 for p_o=,1.0 the no-flow intersection of all inlet-vane fan curves. These relationships and Equation (11), when substituted into Equation (12), yield

$$H = \frac{0.646f}{2-\left[(1-p_o)f^2 + p_o\right]} + 0.354 \tag{13}$$

or

$$h = a + \frac{f}{b + cf^2}$$

where

$$a = 0.354$$

$$b = \frac{2-p_o}{0.646}$$

$$c = \frac{p_o - 1}{0.646}$$

A family of power curves, for varying pressure set point, is shown in Figure 11.

Restrictions Due to Load-Dependent Efficiencies

Load-dependent motor and VSD efficiencies will influence the conversion of shaft power, given in the presented equations, to electrical power when, based on Figure 6, motor torque or, for VSD operation, motor speed, drops below 50 percent. While, as previously argued, the impact

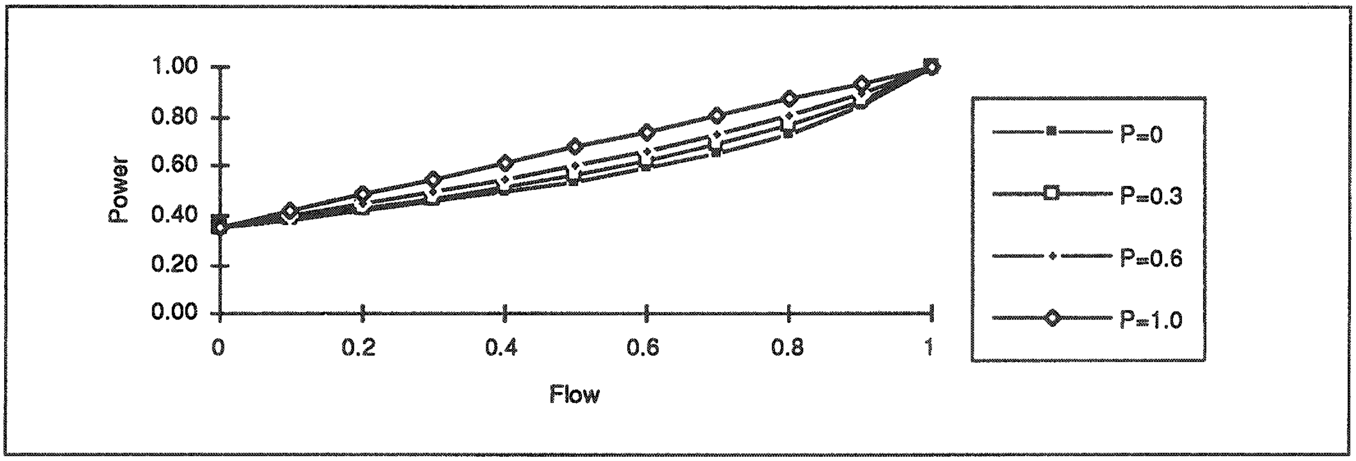

Figure 11. Estimated Power for Variable-Inlet-Vane Control, as a Function of Static Pressure Set Point. When compared against data from the fan manufacturer for pressure set points of 0, 0.2, and 0.4, the model deviated by no more than 5 percent.

of decreasing efficiencies is lessened in cases where there is little operation at low flow rates, applications with flow distributions dominated by low-flow operation may require that shaft power equations be corrected to account for load-dependent efficiencies.

The thresholds of applicability for using the above equations without correction for load-dependent efficiencies can be determined by substituting the torque and speed requirements into the above equations for shaft power and solving for flow. For throttling control, the dimensionless flow rate must exceed 0.23. For inlet vanes, the minimum flow rate depends on pressure set point, from 0.23 for p_o = 1.0 to 0.41 for p_o = 0. For VSDs, in cases with no pressure offset, flow must exceed 0.7; the constant-efficiency requirement is satisfied at lower flows when there is a pressure offset, ranging down to 0.23 at p_o = 1.0.

Comparison with Calculated and Measured Results

Power estimates as a function of flow and static pressure can be compared with data presented in Armintor and Connors for pumps. The latter data are also calculations rather than measured results, but they are based on an actual rather than simplified pump curve and account for pump, motor and VSD efficiencies, all of which vary with load. Figure 12, for VSD control of a pump with static head, show near-perfect agreement. In this case, the product of motor and VSD efficiency calculated by Armintor and Connors varied by about 4 percent over the range of flows, confirming the earlier statement that efficiencies may in practice be of slight importance.

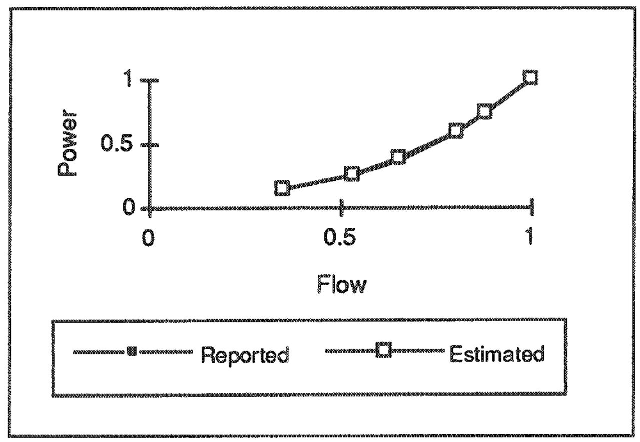

Figure 12. Electrical Power as Estimated by the Simplified Model and as Calculated by Armintor and Connors (1987), for VSD Control of a Pump with Static Pressure. Data points are identical at all flow rates.

Figure 13, for throttle-valve control of the same pump, shows a small offset, with Armintor and Connors' calculated values exceeding our estimation. Armintor and Connors' motor efficiency varied by about 1 percent over the flow range. The reason for the discrepancy in powers is clear: in our model, power decreases for any decrease in flow below the design point, which is at the terminus of the surge line; in practice, the design point is typically somewhat down the fan curve, such that the decrease in flow required to reach the surge point involves no change in shaft power. Our model could separate the design point from the surge line but would require an additional parameter to do so. Figure 13 includes power estimations for two such separations, of 10 and 20 percent flow; the former case matches the reported results nearly exactly.

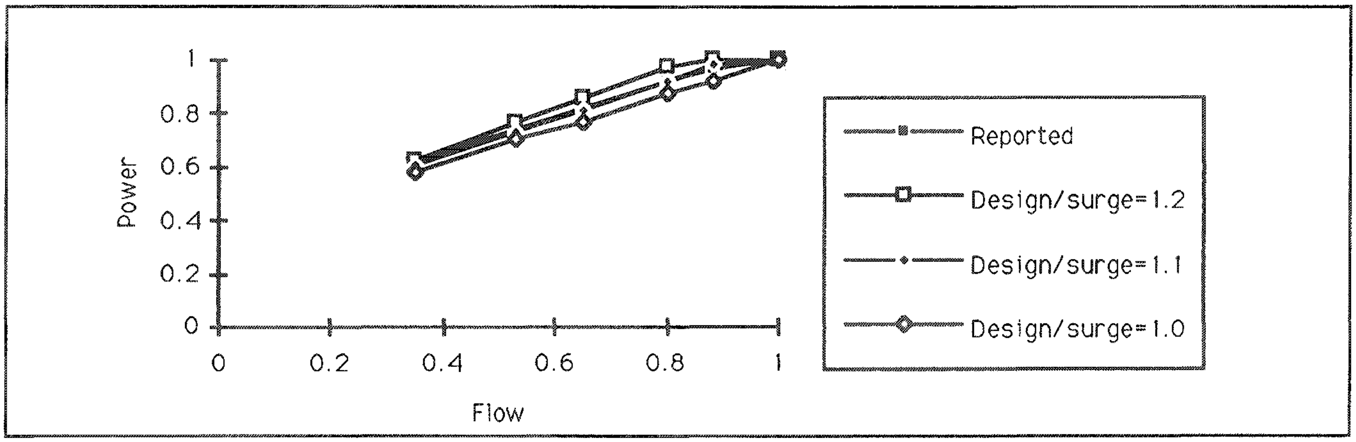

Figure 13. Electrical Power as Estimated by the Simplified Model and as Calculated by Armintor and Connors (1987), for Throttle-Valve Control of a Pump with Static Pressure. The lowest power estimate is derived from equations presented in the text, while the highest estimate is based on a design flow that exceeds that associated with the surge line by 20%. These two estimates bracket the reported data and the estimate based on a 10 percent flow offset, which are nearly coincident.

Building ventilation fan power measured by Englander and Norford (1992) provides a basis for further evaluation of the model. For a supply fan subject to a static pressure set point, the model provides excellent accuracy, as shown in Figure 14. A return fan with no static pressure set point should, in principle, show power that depends on the cube of flow, but the measured data exceed the estimate, as shown in Figure 15.

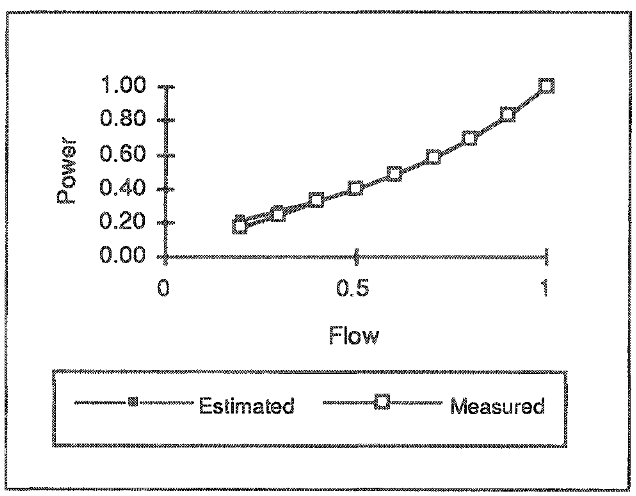

Figure 14. Electrical Power as Estimated by the Simplified Model and as Measured by Englander and Norford (1992), for VSD Control of a Building Ventilation Supply Fan with Static Pressure Set Point.

The inlet-vane model is less successful than the models for VSDs and either discharge dampers or throttling valves. Most notably, there are significant discrepancies among reported results when comparing power at no-flow or low-flow conditions, as shown in Figure 16. In addition, the shape of our model's curves does not quite match reported data. It would be desirable for fan and inlet-vane manufacturers to develop a set of standard algebraic forms for power as a function of flow and static pressure, distinguishing different vane-fan blade combinations if necessary. Absent this information, there appears to be a need, when assessing inlet-vane performance, to measure the low-flow power.

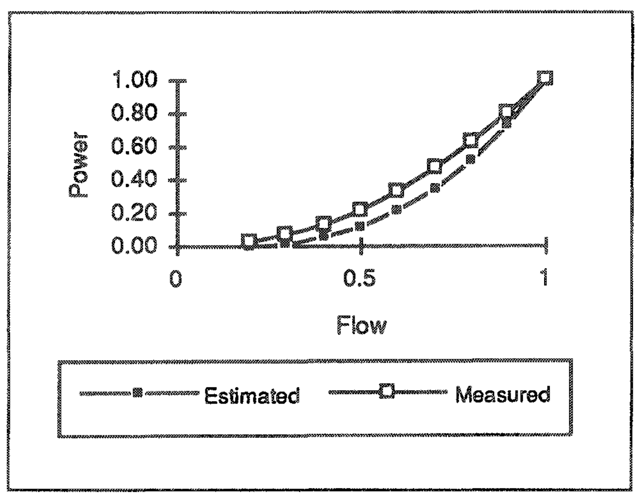

Figure 15. Electrical Power as Estimated by the Simplified Model and as Measured by Englander and Norford (1992), for VSD Control of a Building Ventilation Return Fan with no Static Pressure Set Point.

The two dimensionless power relationships are useful in practice if they can be scaled up to physical values based on site information. First, power at maximum flow can be

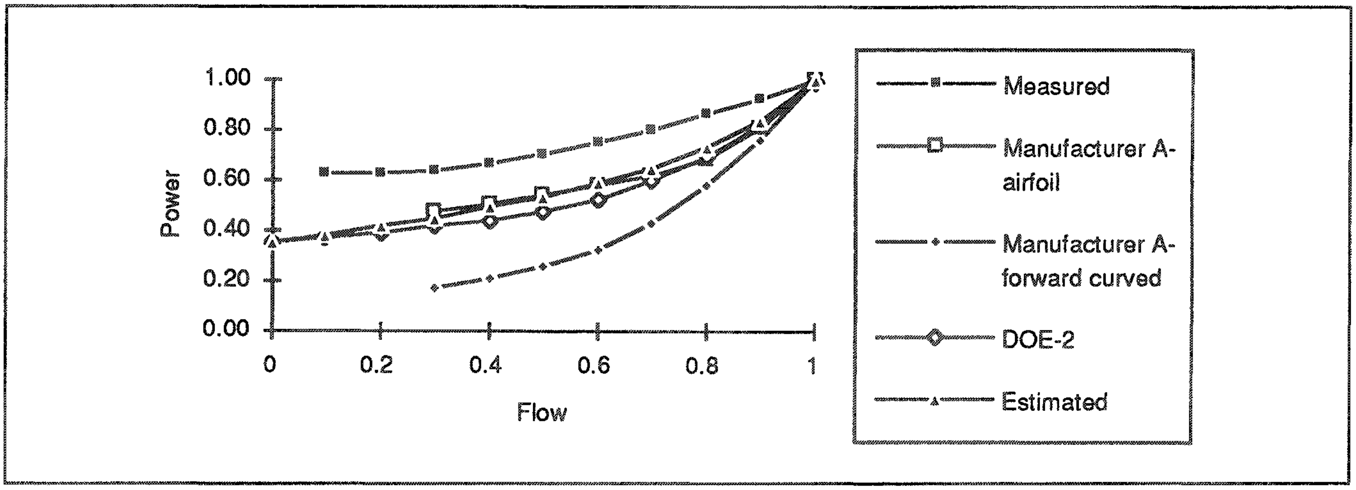

Figure 16. Comparison of Predicted Powers for Inlet Vanes, with no Static Pressure Offset. Values estimated from the model agree well with manufacturer's data for backward inclined fan blades with airfoil blade shape, but disagree markedly with forward curved fan blades. Measured data are much higher, suggesting a need for more monitoring.

taken from motor nameplate ratings and an estimate of motor and VSD efficiencies, if the motor is not significantly oversized. Even with a properly sized motor, a VSD may run at less than 60 Hz at maximum flow, indicating that the fan or pump itself is oversized. If so, motor nameplate power can be reduced by the following factors:

$(\omega_{max}/60)^3$ variable-flow variable-pressure systems
$(\omega_{max}/60)^2$ variable-flow constant-pressure systems

Second, static pressure as a fraction of design pressure can be obtained directly or, for pumps in particular, can be expressed as the fraction of VSD speed under no flow conditions.

Building VAV Systems

As discussed above, a typical VAV system involves both a variable flow/pressure offset system[1] (the supply fan) and a system with variable flow and pressure proportional to the square of flow (the return fan).[2] For this reason, the analysis must be performed separately for each system. Annual energy savings can be estimated by taking the difference between estimated energy consumption for constant speed operation and for VSD operation. Energy consumption can be estimated by combining fan power curves with flow distribution histograms specific to a particular installation. Since neither power curves nor flow histograms are typically available for a given site, they must be approximated based on a few parameters that are known or can be closely estimated. The context in which the analysis is being done (e.g., cost-effectiveness screening for utility planning, pre-retrofit site-specific payback analysis, post-retrofit performance evaluation) will normally determine the depth of the analysis and, in turn, the types of inputs required. Depending on the context, the inputs might range from engineering assumptions to information collected in phone interviews with building operators to instrumented measurements made on-site.

Once power curves and flow (or dimensionless part-load) histograms are developed, the normalized annual energy use for a given fan can be estimated as:

$$E = \sum_{i=1}^{n} h_i \cdot P_e(f)\big|_{f=f_i} \tag{14}$$

where

- n = number of bins in flow rate or part load distribution
- h_i = number of annual run-time hours in bin i
- P_e = motor input power as a function of flow or part-load fraction, evaluated at f_i
- f = flow rate or part-load fraction
- f_i = mean flow rate or part-load fraction of bin i

The utility demand requirement can be estimated as

$$D = \frac{\sum_{i=1}^{n} h_i' \cdot P_e(f)\big|_{f=f_i}}{\sum_{i=1}^{n} h_i'} \tag{15}$$

where h'_i is the number of operating hours in bin i occurring during the period of interest (e.g., summer or winter), weighted by probabilities of utility peak load occurrence. In an analysis of VAV/VSD retrofits performed by the utility, historical peak load data were used to determine probabilities of peak occurrence for each time/temperature bin.

Studies based on measured data have shown that supply fan power can be modeled as a two-parameter function of outside air temperature (Lorenzetti and Norford, 1992); it follows that supply air flow can be modeled as a similar function. It is likely that below some outside temperature $T_o = T_{ref}$, the supply air flow rate f for a given building depends on internal gains and is relatively constant,[3] and above T_{ref}, an additional portion that depends on outside temperature:

$$(T_o < T_{ref}) \quad f = f_{\min}$$

$$(T_o \geq T_{ref})$$

$$f = f_{\min} + \frac{1 - f_{\min}}{T_{des} - T_{ref}}(T_o - T_{ref}) \qquad (16)$$

$$(T_o > T_{des}) \quad f = 1.0$$

The balance temperature T_{ref} might be known by the building operator, or can be estimated based on the type of construction. The minimum flow rate $f_{min,}$, if unknown by the building operator, can be estimated from several instantaneous measurements or from the maximum flow rate (if known) and an assumed turn-down ratio. The outdoor temperature at which the fans are designed to provide full flow, T_{des}, if not available from building specifications, can be taken as the ASHRAE 99% design drybulb temperature for the location. If measurements of flow and temperature over a range of loads are available, flow as a function of T_o can be determined by regression. Since hourly measurements of temperature are available for most locations in the U.S., normal year histograms of flow can be constructed easily, given flow as a function of temperature and the operating schedule of the fans.

Supply Fan

Rather than substitute dimensioned forms of the variables into Equation (13), it is perhaps easier to first determine $p_o = p_s/p_{max}$, where p_s is the static pressure set point and p_{max} is the pressure at the design point or under full flow conditions, and then work directly with the remaining dimensionless equation. The dimensionless power can be computed for a range of f (0 to 1); these and the dimensionless values of flow can then be scaled up to actual values. In this way, fan design parameters can be changed without having to recompute coefficients of the equation each time.

In the same manner, Equation (4) is used to compute supply fan power under VSD operation; alternately, Equation (8) may be used if speed rather than flow is known. Combining the power data for each condition with a flow histogram will yield annual energy consumption, from which savings can be computed.

Note that to calculate electricity use and demand with Equations (14) and (15), power estimates must be for motor input, rather than fan shaft power. Because VAV fans typically operate above 30 or 40% of full flow, and motor and VSD efficiency curves are relatively flat in the fan's operating range, it is reasonable to assume some constant average efficiency for converting fan shaft power to electrical power. The following equation may be used:

$$p_e = \frac{0.746H}{\eta_{mt}} \qquad (17)$$

where H is the fan shaft power in horsepower (or kW, in which case the conversion factor of 0.746 is unnecessary), and η_{mt} is a combined average motor, VSD, and transmission efficiency. Of course, if maximum power is based directly on field measurements of motor electrical input, no conversion is necessary.

Return Fan

With pressure offset set to 0, Equation (13) for fan power with inlet vanes reduces to

$$H = \frac{0.646f}{2 - f^2} + 0.354 \qquad (18)$$

Equation (9) is used for fan power with discharge dampers, for return as well as supply fans. As expected, Equation (4) for VSD operation reduces to the simple cubic

$$H = f^3 \qquad (19)$$

Energy and demand savings are computed in the same manner as for the supply fan. Since power drops rapidly with decreasing flow, however, the constant motor efficiency assumption is no longer a good one; it is recommended that lower efficiency at low flow rates be accounted for in some way.

Conclusion and Recommendations

We have shown that centrifugal VSD applications, when considered on the basis of energy use characteristics, can be broken down into five distinct categories, with most common applications falling into three of them. Savings estimation software offered by EPRI and VSD manufacturers, as well as popular building energy simulation software, largely fails to account for system differences, and produces poor savings estimates as a result.

Of the software reviewed, only that of the fan manufacturer adequately accounted for static pressure offset, a factor crucial to accurate energy estimation. Notably, the fan manufacturer's power estimates for inlet vane control agreed with our theoretical approximation, while VSD manufacturers' estimates of inlet vane power are higher, leading to greater estimates of VSD energy savings. EPRI's, the fan manufacturer's, and one VSD manufacturer's software account for motor and VSD efficiencies that change with load. As it turns out, this is significant only for some applications in which motor load drops below about 50%; the impact of the estimated shape of load distribution is far greater.

We have presented simplified general expressions for pump or fan power as a function of flow and pressure offset, for both throttled and VSD control, including the special case of inlet vane control for fans. The equations agree well with measured data for fans, and with previously published data for pumps. A method for estimating energy savings in building VAV systems with VSDs, given limited system design or performance information, is presented.

Flow distribution information is crucial, given its strong influence on savings predictions. Site-specific information is needed. The three flow distributions for building ventilation systems provided by the fan manufacturer, although useful as examples, produce markedly different results and leave the user with concerns that a particular site might fall between two of the three categories. Our recommended savings estimation procedure for building ventilation systems rests on an assumed relation of airflow to outside temperature that needs to be confirmed with instrumented monitoring performed over several seasons.

Flow data may be routinely collected for large commercial and industrial applications of VSDs and should be sought out by VSD savings evaluators. For example, one of the utility's customers installed VSDs on four large wastewater pumps. After the retrofit, motor speed varied between 70 and 100 percent, with the lower bound corresponding to no-flow conditions. This narrow speed range produces a large range in power, an uncertainty that can be reduced by taking advantage of wastewater flow data collected by the treatment plant.

Although the estimates of power presented here are in good agreement with published results that adequately account for the differences in system type we've highlighted, we recommend further instrumented monitoring to confirm the accuracy of the equations for specific applications.

Acknowledgments

The authors acknowledge the support of the New England Power Service Company and the assistance of J. Greg Byrd.

Endnotes

1. Although with advanced control systems it is possible to vary the static pressure set point in VAV systems, the set point in typical systems is fixed.

2. Some VAV systems do not have return fans; for the purpose of this discussion, we will use the general case.

3. This minimum supply air flow rate can be thought of as the sum of two components: one representing the fresh air requirement of all the zones served by the fan, and the other representing cooling to offset internal gains in interior zones.

References

Armintor, J. K. and Connors, D. P. 1987. "Pumping Applications in the Petroleum and Chemical Industries." *IEEE Transactions on Industry Applications*, Vol. 1A-23.

Englander, S. L. and Norford, L.K. 1992. "Saving Fan Energy in VAV Systems--Part 1: Analysis of a Variable-Speed-Drive Retrofit. *ASHRAE Transactions*, Vol. 98, Pt. 1.

Electric Power Research Institute, 1989. *Adjustable Speed Drives Analysis (ASCON-I) Energy Savings and Return on Investment User's Manual.* Palo Alto, CA.

Electric Power Research Institute, 1990. *Adjustable Speed Drive Retrofit Economic Analysis Computer Program ASCON II Version 1.0 User's Manual.* Draft report. Palo Alto, CA.

Electric Power Research Institute, 1991. *Adjustable Speed Drives Directory.* Palo Alto, CA.

Larson, Eric D. and Nilsson, Lars J. 1990. "A System-Oriented Assessment of Electricity Use and Efficiency in Pumping and Air-Handling." Report 253, Center for Energy and Environmental Studies, Princeton University, Princeton, NJ.

U.S. Department of Energy, Assistant Secretary for Conservation and Solar Energy. 1980. "Classification and Evaluation of Electric Motors and Pumps." Washington, D.C.

Background Reading

Trane Company. 1983. "Adjustable Frequency Fan Speed Control." Applications Engineering Manual FND-AM-11 (783). La Crosse, WI.

Trane Company. 1982. "Fans and Their Application in Air Conditioning." Applications Engineering Seminar ED-FAN 4/71. La Crosse, WI.