How to Avoid Overestimating Variable Speed Drive Savings

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ABSTRACT

This paper addresses eight factors that can cause incorrect and often excessive savings estimates for pump and fan variable speed drive applications. To avoid overestimating savings:

1. Identify system elements that affect head pressure independently of flow rate.
2. Identify system elements that change head pressure in proportion to less than the square of flow rate.
3. Account for dynamic system elements, especially when in systems with minimum static pressure controls.
4. Consider changes in fan efficiency.
5. Account for decreases in motor efficiency at part load, particularly for smaller motors below about 35 percent load.
6. Recognize that existing part load controls may be more efficient than expected.
7. Account for drive losses.
8. Measure full flow power, rather than assuming it is the same as motor nameplate or design power.

For many pump and fan systems, none of the eight factors will apply, or their effects will be negligible. However, analysts should consider their applicability when estimating savings for a particular system. This paper provides tools for accounting for the factors.

INTRODUCTION

The author's firm conducts energy audits, follow-up metering studies and verifications of estimated savings for municipal, state, and federal, agencies and utility companies, and investor-owned utility company incentive programs. We have found that the predicted savings for variable speed drive (VSD) applications are frequently overestimated. This appears to occur more often for VSDs than for other energy efficiency technologies.

This paper addresses eight factors that can cause excessive savings estimates, identifies warning signs that such conditions exist in a system, and provides techniques for adjusting savings estimates.

BACKGROUND—STANDARD PART LOAD CURVES

Data on the part load efficiency of pump and fan controls are available from a variety of sources. Example curves for most common control mechanisms are shown in Figure 1.

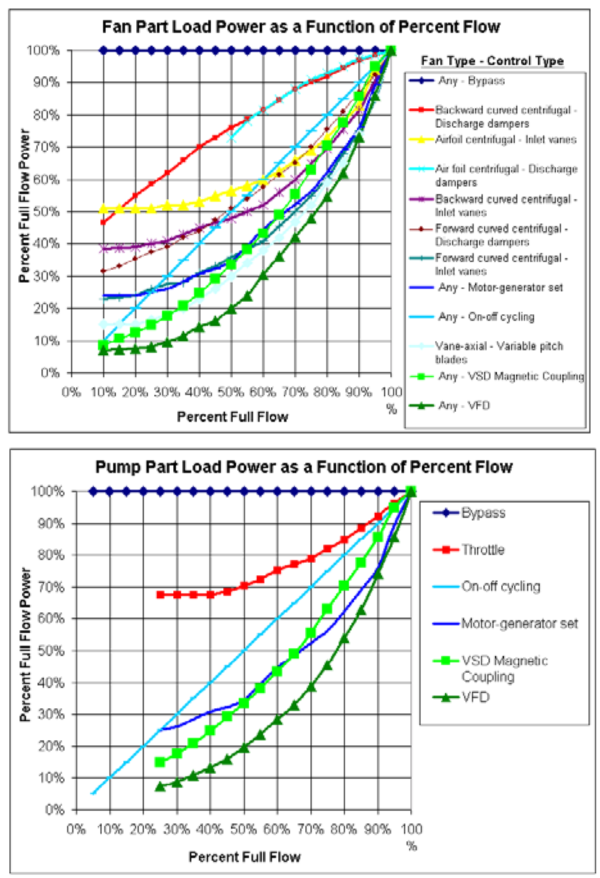

Figure 1. Fan and Pump Part Load Curves[1]

[1] The curves are based on standardized conditions. References for each fan and pump curve shown are

It should be noted that "full flow power" is not numerically the same for all types of controls. Full flow power with a variable frequency drive (VFD)-controlled pump typically will be 3 to 4 percent higher than full flow power for the same pump and system without a VFD due to drive power requirements, for example. It will be 5 to 10 percent higher for magnetically coupled VSDs. This is discussed in more detail later. Full flow power for motor-generator sets is much higher than other for other control system types—up to 35 percent higher. Also the straight line depiction of the relationship for "On-off cycling" control is an approximation depicting average power as a function of average flow rate, not instantaneous, and does not include startup power requirements. The deviation from the line shown may be considerable if a large number of stops and starts are needed.

These curves are convenient and generally recognized and are appropriate for use in many applications that are near the standardized conditions. This paper addresses factors that should be incorporated to modify the curves for certain applications, and when to not use them at all.

BACKGROUND—PIPE AND DUCT SYSTEM LOSSES

This background is relevant for discussion regarding the first three factors noted in the abstract.

For many pump and fan systems, the "affinity laws" govern power required by the fan or pump. The affinity laws hold that power is proportional to the cube of velocity, and to the cube of fan/pump rotational speed, and to the cube of flow rate in a constant diameter system. This relationship drives the savings for many variable speed drive applications.

$$P \; \alpha \; rpm^3 \; \alpha \; Q^3 \; \alpha \; \dot{m}^3 \qquad \text{Equation (1)}$$

Where,

- P = power (kW)
- α = "is proportional to"
- rpm = rotational speed of fan/pump (rpm)
- Q = volumetric flow rate (cfm or gpm)
- $\dot{m}$ = mass flow rate (lb_m/min)

The affinity laws only apply to duct and pipe systems characterized entirely by "velocity losses" and "friction or minor losses." Velocity losses usually are negligible in fluid systems and constant diameter duct air systems. This means all system losses must be friction losses for the affinity laws to apply. Examples of system elements that cause friction losses are:

- Duct and pipe straight runs
- Duct and pipe entrances/exits
- Bends/elbows/tees/wye branch fittings
- Expansion and contraction transition fittings
- Turning vanes
- Balancing valves and vanes

These are the most common elements that create friction and head loss in systems. But duct and pipe systems can have other types of losses that contribute to pressure drop in the system and increased pressure differential across the pump or fan. If these losses are not accounted for, users will find that VSDs save less energy than was projected.

1. IDENTIFY SYSTEM ELEMENTS THAT AFFECT HEAD PRESSURE INDEPENDENTLY OF FLOW RATE

If the pump or fan system includes elements that affect pressure drop independently of flow rate, VSD savings as a percentage of full flow power will be lower than a system without such elements. This effect is not reflected in the standard part load curves.

Any pump that raises the fluid from a lower to a higher elevation in an open system must impart energy into the fluid to do so. This "potential head" or elevation head is the most common example of a system element that increases head independent of flow rate. Systems with potential head include:

- Condenser water pumps for open cooling towers[2]
- Refinery pumps that transfer oil into an elevated open tank[3]

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included in the Bibliography Cross-Reference section at the end of this paper.

[2] For detailed examples of the effect of ignoring elevation head, see Martino (16). Two examples provided by Martino show overestimation of savings by 49 and 79 percent.

[3] For a detailed example of the effect of ignoring elevation head on a large industrial system, see Carlson (6). One example results in overestimation of savings of 364 bhp and 18 percent. This source also includes example calculations for a multiple large crude oil pump system.

Potential head is negligible in most air systems because of the low density of air. Also, there is no potential head in closed fluid systems because the net elevation change is zero.

Any pump or fan system that maintains a constant differential pressure also has a pressure component that is independent of flow rate. Systems with a constant differential pressure component include:

- Variable speed primary-secondary HVAC pump systems[4]
- Water pressure booster pumps[5]
- Positive or negative pressurization fans

Mathematical expression of the factor

When flow rates and duration are unknown auditors sometimes measure power with short term metering equipment and then use part load curve profiles to estimate flow rates from the power data. In pumping applications where constant potential head contributes to the power requirement, the potential-associated power must be subtracted from all measurements before flow is estimated from the power-flow curves. While electric power due to potential head often cannot be measured explicitly before the retrofit, it can be estimated as a percentage of total head or calculated using the equation:

$$P = \dot{m}\, \Delta z\, g / (44{,}236\, g_c\, \eta_p\, \eta_m) \qquad \text{Equation (2)}$$

Where,

P = power (kW)
$\dot{m}$ = gpm * 8.33 $lb_{m\ water}$/gallon (lb_m/min)
Δz = change in elevation or constant pressure differential (ft)
g/g_c = accel. of gravity/gravitational constant = 1.0 lb_f / lb_m
44,236 = conversion factor[6] (ft-lb_f / min / kW)
η_p = pump efficiency
η_m = motor efficiency

[4] For detail see Hegberg (12), especially Fig. 12.

[5] VSDs usually save money because they reduce system pressure at the same time as they reduce flows. For a discussion on how VSDs can save money on booster pumps—an application that is seemingly a constant pressure application—see ITT (13). Even so, savings is less than it would be for a similar variable flow application without pressure boosting.

[6] 44,236=550 ft-lb_f/sec/hp*60 sec/min / 0.746 kW/hp

Exclude such power from the measured total power before using it to estimate non-elevation full flow power and then percent full flow non-elevation power and then percent full flow, using the standard curves in Fig. 1. The analyst must know both power and flow at one operating point, usually design conditions, to use this approach.

Example of Savings Overestimation

A 20-hp water pump with throttle flow control is being considered for a VFD. Power at full flow is 10 kW. Elevation head of 20 feet is observed and maximum flow is known to be 200 gpm. Using Eq. 2 and assuming 65% pump efficiency and 85% motor efficiency, the power to overcome elevation head at maximum flow is 1.4 kW. The auditor measures power over a two week period and finds that the pump runs at 10 kW 10% of the time, 9 kW 20% of the time, 8 kW 30% of the time, and 7 kW 40% of the time.

Table 1 summarizes estimated flow and projected VSD power using the throttle curve from Figure 1, with and without consideration of the elevation head. Ignoring the effect of elevation head results in estimating 25% more savings that the estimate that takes elevation head into account.

The above approach can require multiple iterations to estimate flow because percent flow is an input and an output, and it is not as precise as using the actual pump performance curve and system curve. However, it is quick, can be coded into a spreadsheet, makes use of standard relationships, does not require curve data that may not be available, and the results account for elevation head.

2. IDENTIFY SYSTEM ELEMENTS THAT CHANGE HEAD PRESSURE IN PROPORTION TO LESS THAN THE SQUARE OF FLOW RATE

Lost head due to friction typically is a function of the square of velocity, as illustrated in the Darcy formula:

$$l_f = f\,\frac{L}{D}\,\frac{V^2}{2g} \qquad \text{Equation (3)}$$

Where,

l_f = Friction head loss (ft)
f = Moody friction factor
L = length of pipe or duct (ft)
D = diameter of pipe or duct (ft)
V = average velocity in conduit (ft/sec)
g = acceleration of gravity (ft/sec^2)

Without Considering Elevation Head

Measured Power (kW)			Percent of Full Flow Power	%Max Flow *(From Fig. 1)*	VSD Power (kW) *(From Fig. 1)*	Savings (kW)
10			100%	100%	10.4	(0.4)
9			90%	87%	6.8	2.2
8			80%	69%	4.3	3.7
7			70%	50%	2.2	4.8
Weighted average hourly power savings from VSD:					3.4 kW	

With Consideration of Elevation Head

Measured Power (kW)	Approx. Pwr For Elevation *(From Eq. 2 & %Flow above)* (kW)	Power For Other than Elevation (kW)	Percent of Full Flow Non-Elevation Power	%Max Flow *(From Fig. 1)*	VSD Power (kW) *(From Fig. 1)*	Savings (kW)
10	1.4	8.6	100%	100%	10.4	(0.4)
9	1.2	7.8	91%	89%	7.4	1.6
8	0.9	7.1	83%	75%	5.1	2.9
7	0.7	6.3	73%	57%	3	4.0
Weighted average hourly power savings from VSD:					2.8 kW	

Table 1: Example of Overestimation of VSD Savings Due to Ignoring Potential Head

For other than straight runs, friction loss is typically calculated in equivalent lengths. In all cases the equation reduces to a head loss that is proportional to the square of velocity. The power equation is the same as Eq. (2) except that l_f is substituted for Δz. Power is proportional to the cube of velocity. This is the basis of the affinity laws.

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Mathematical Expression of Losses

Friction and other such losses are not always proportional to the square of velocity. Examples of system elements to look for that change head pressure in proportion to less than the square of velocity include:

Element	Head Loss Is Proportional To
HEPA filters	V
Bag filters	$V^{1.2}$
Throwaway filters	$V^{1.65}$
Air handler coils[7]	V
Disc type water meters[8]	V

[7] See for example ASHRAE HVAC Systems handbook (2), Fig. 12, page 21.13.

Figure 2 compares the filter head loss-velocity relationship with that for standard V^2 friction losses.

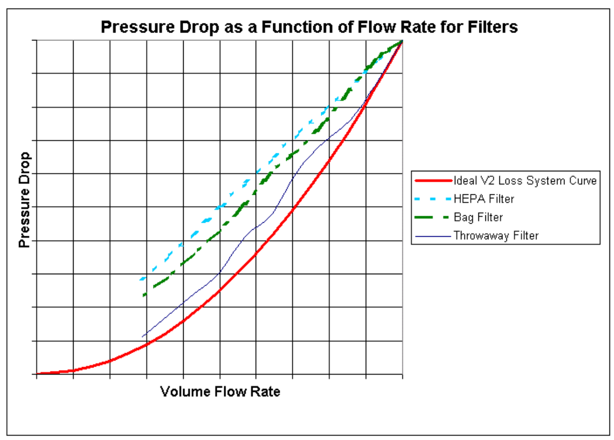

Figure 2. Filter Head as a Function of Velocity[9]

Example of Savings Overestimation

Figure 3 illustrates the error potential from treating losses that are proportional to flow rate as though they are proportional to the square of velocity in estimating VSD power.

[8] See for example ASHRAE Fundamentals handbook (1), Fig. 8, page 35.9.

[9] For reference data on all three filter types, see Carrier (7), page 22.

In all cases ignoring this factor will cause overestimation of VSD savings.

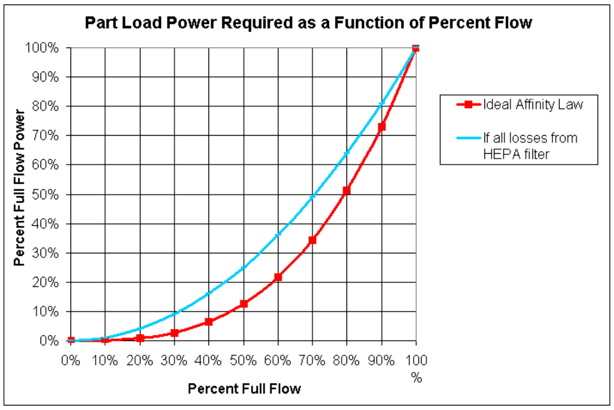

Figure 3. Ideal Affinity Law Part Load Power Compared to Part Load Power if All Pressure Losses are Proportional to Flow Rate

3. ACCOUNT FOR DYNAMIC SYSTEM ELEMENTS, ESPECIALLY WHEN IN SYSTEMS WITH MINIMUM STATIC PRESSURE CONTROLS

Variable air volume (VAV) HVAC systems modulate air flow with dampers in VAV boxes that are well downstream of the air handlers. The dampers continue to modulate even after a VSD retrofit. The dampers effectively continuously change the system characteristics. Head loss in such a dynamic system not only is not proportional to the square of velocity, the theoretical maximum flow for the fan is different for each VAV damper position. The entire system pressure curve changes with each change in damper position.

Viewed another way, after a retrofit it is as if the fan has VSD control but still has discharge dampers attached as well. Discharge dampers are among the least efficient control techniques, as Fig. 1 illustrates. All of the discharge damper curves show high power at low flow rates, and if extended to 0% flow, would still have substantial power. The dampers increase head pressure at low flow rates regardless of fan control type, and this increases power requirements.

Any industrial fan or pump system that has flow modulation at the point of use will function similarly.

The analytical challenge of estimating savings in such circumstances is compounded by the presence of minimum static pressure settings. VAV boxes, balancing valves, and many other devices specify minimum pressure settings to ensure their proportional controllers or other hardware can function properly. VAV boxes typically specify a minimum static pressure of 0.3 to 0.8 inw. in order for the dampers to control flow properly. Operators in turn often set the minimum pressure to 1 inw. or higher because the pressure sensors can be well upstream of the VAV boxes. Separately, as part of balancing exercises technicians and engineers sometimes set the minimum pressure excessively high as a cure-all to solve downstream problems that would be better remedied at the root cause. Raising the minimum pressure is an energy intentive solution because the fan then runs at nearly full capacity all the time with little or no part flow power savings.

Once flow requirements are low enough that the minimum pressure setting comes in to play, the system becomes a constant pressure system and energy use patterns grossly deviate from the affinity law relationships.

In short, analysts must account for additional head loss at part load in systems with:

- Downstream modulating dampers or throttles (e.g. VAV boxes)
- High minimum pressure settings relative to the design full flow head pressure

<u>Mathematical Expression of Losses</u>

If such a system is under consideration for a retrofit, the issue of accounting for system dynamics and minimum pressure settings is difficult to model.

Analysts have found that a reasonable approximation of such a system is to treat the minimum static pressure control point as being the equivalent of elevation head in pump systems. The formula is shown below:

$$p = p_{min} + (p_{design} - p_{min}) * \%Q^2 \qquad \text{Eq. (4)}$$

Where,

p = operating pressure (ft)
p_{min} = minimum pressure (ft)
p_{design} = design pressure (ft)
$\%Q$ = percent of design flow rate

To use the formula to calculate power, substitute p for Δz in Eq. (2). The first part of the equation, p_{min}, has the same relation to power as Δz. The second part has the same relation between flow and power as the affinity law, except that the pressure used is the difference between the design and minimum pressure instead of just the design pressure.

4. CONSIDER CHANGES IN FAN EFFICIENCY

The sources for the generic power curves provided in Fig. 1 either make no assumptions about the change in fan and pump efficiency as flow rate varies or do not state what the assumptions are. Figure 4 illustrates the typical efficiency patterns taken from performance curves for fans.

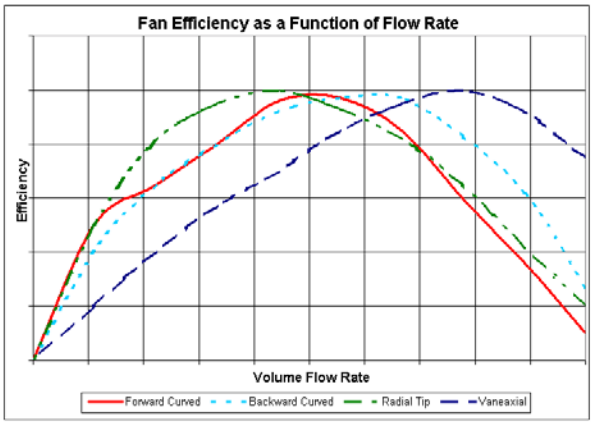

Figure 4. Fan Efficiency Curves[10]

Where "full flow" lands on the efficiency curves obviously affect full flow power. It also can have a significant effect on part flow power curves for a particular application. Throttling and reducing speed can either increase or decrease the fan/pump efficiency.

In the field, the best way to spot systems in which this could be a material issue is to look out for systems modified through the addition or removal of equipment that changes the system head (filters, ducts, heat exchangers, etc.) or flow (new sheaves, etc.). In such instances the pump or fan selected to meet the original full flow head and flow rate combination might be mismatched with the current system it serves. The mismatch could move the intersection of the system and pump curves to the point where efficiency is sub-optimal and changes rapidly with minor variation in system flow or head. In such circumstances, the only accurate approach to estimating system energy use is to use the pump and fan curves.

The only way to know for certain where the fan or pump is operating is to make careful field measurements of pressure, flow and power and have the fan performance curves available. Unfortunately in many retrofit applications this is not possible, and in small horsepower applications the analytical effort simply cannot be cost-justified even if the data could be made available.

Mathematical Expression of Losses

As a practical matter, the author only has been able to account for this factor in routine analysis through use of the fan and pump performance curves.

5. ACCOUNT FOR DECREASES IN MOTOR EFFICIENCY AT PART LOAD, PARTICULARLY FOR SMALLER MOTORS BELOW ABOUT 35 PERCENT LOAD

Motor efficiency varies with load. Many publications have addressed the causes of losses and they are not repeated here.[11] Efficiency is relatively constant above 50 percent load, typically with a slight peak near 70 percent. Efficiency decreases below 50 percent load as load-independent losses become a larger portion of the total input power. Figure 5 illustrates motor efficiency patterns.

VSD conversions often move the input power from above 50 percent load to below 50 percent load. If the VSD controls a smaller motor, 10 horsepower or less, the corresponding decrease in motor efficiency can have a material effect on savings.

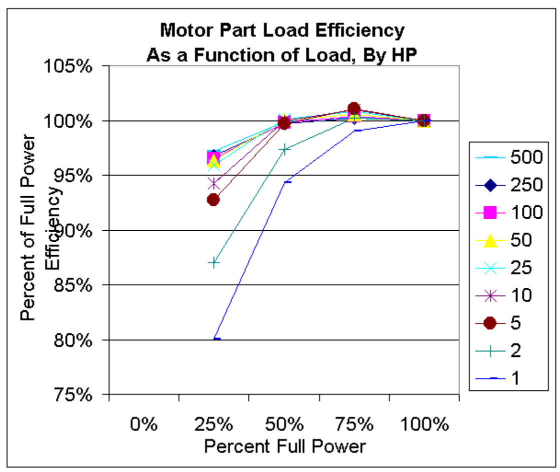

Figure 5. Motor Part Load Efficiency Curves—Efficiency As a Function of Load By HP[12]

[10] From McQuiston (17), p. 426-427.

[11] See Andreas (3) p. 35-42, for example.

[12] Plotted values are the average efficiencies of all TEFC and ODP NEMA Design B motors in the MotorMaster+ database for which 100%, 75%, 50%, and 25% efficiency data were available. Data downloaded February 1, 2003.

While the curves above are nonlinear, making percent full load input power the dependent variable instead results in a more linear relationship that can be expressed simply and can be easily applied. This expression also more accurately correlates with the real world basis of losses, some of which are proportional to operating load while others are proportional to maximum load.

Mathematical Expression of Losses

$$\text{Motor \% full load input power} = M * \text{\%full load bhp required to fan} + B \qquad \text{Eq. (5)}$$

M and B can be found in Table 2.[13] The table also includes typical motor full load efficiency. In the table, N is the MotorMaster+ population.

HP	N	Average Full Load Efficiency	M slope	B intercept
0.5	9	73.7	0.862	0.128
0.75	13	77.0	0.906	0.082
1	391	80.1	0.916	0.077
1.5	459	82.3	0.935	0.059
2	469	83.6	0.949	0.044
3	535	86.0	0.958	0.035
5	511	87.2	0.973	0.020
7	20	88.9	0.982	0.007
7.5	433	88.5	0.972	0.022
10	445	89.4	0.979	0.015
15	384	90.3	0.982	0.013
20	364	90.9	0.986	0.008
25	382	91.8	0.985	0.010
30	362	92.1	0.987	0.008
40	302	92.7	0.985	0.011
50	294	93.1	0.987	0.010
60	239	93.6	0.986	0.011
75	225	94.0	0.988	0.010
100	217	94.5	0.988	0.010
125	177	94.7	0.988	0.009
150	170	95.1	0.990	0.008
200	145	95.3	0.990	0.008
250	61	95.1	0.989	0.009
300	43	95.2	0.989	0.009
350	23	94.9	0.986	0.012
400	20	95.3	0.993	0.005
450	5	94.8	0.981	0.016
500	9	95.3	0.990	0.007

Table 2. Motor Part Load Efficiency Coefficients

[13] M and B are based on regressions of the same data used in Figure 5.

Example of Savings Overestimation:
A cleanroom VSD conversion reduces makeup air fan shaft power requirements from 5 to 2.5 hp on (20) 10-hp, 90-percent full load efficiency motors. Without accounting for the change in motor efficiency, savings is calculated to be 20.7 kW. Accounting for a six percent decrease in efficiency reduces the savings by 6.4 percent to 19.4 kW.

Ignoring this factor almost always results in overestimating savings because VSD operation leads to lower motor efficiency. The error typically is less than five percent but up to ten percent or more occurs for small motors at 25 percent or less output power.

6. RECOGNIZE THAT EXISTING PART LOAD CONTROLS MAY BE MORE EFFICIENT THAN EXPECTED

End users often are not aware of just how efficient their existing fan's flow control system may be. VFD vendors have little incentive to highlight it. A close look at Figure 1 reveals that vane axial fans and forward curved fans with inlet vanes (the latter combination being relatively common in smaller air handlers) use no more than about 10 percent more power across the flow range. Given the costs of VFDs, such small savings can make it hard to cost-justify a retrofit.

Example of Savings Overestimation
Airfoil and forward curved centrifugal fans with inlet vanes are sometimes used in similar applications. For a fan running at 30 percent flow, savings potential for an airfoil centrifugal fan is fairly high, about 40 percent of full flow power. In contrast, because forward curved fans with inlet vanes have such good part load efficiency, savings is only about 15 percent of full flow power. This 166 percent difference in savings likely would be the difference between a viable and non-cost effective project.

7. ACCOUNT FOR DRIVE LOSSES

The electronics in a variable frequency drive—or the magnets and mechanical components in a magnetically coupled drive—are not 100 percent efficient. Drive enclosure cooling is a major design consideration that must be addressed in certain applications. Drive losses follow a pattern similar to the part load efficiency curves to that discussed previously for motors. Certain losses are load dependent while others are load independent.

Figure 6 illustrates the loss pattern for variable frequency drives as well as a model curve approximation of the same. The 'modeled' curve is based on the relatively simple assumptions that total losses at full load are four percent, and that half of the losses are operating load-dependent. The benefit of the modeled curve is that it is easy for the analyst to incorporate a simple explicit formula relationship into the calculations.

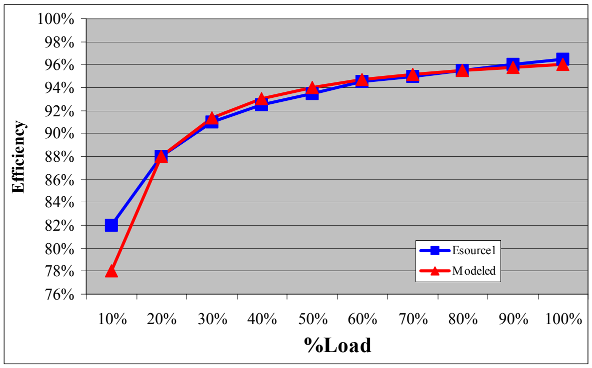

Figure 6. VFD Losses as a Function of Load[14]

As the curve illustrates, drive energy requirements are no less than about three percent of motor input power and can become a more significant percentage of total system energy use if load is less than 50 percent.

Magnetically coupled drives have a different pattern of losses. Losses for such drives range from five to 25 percent of motor nameplate power depending on load and application. Losses peak in the 30% to 70% load range.[15]

Mathematical Expression of Losses

VFD drive loss as a percent of full load power = 0.02 * %full load pwr rating (%FP) + 0.02 Eq. (6)

Magnetically coupled drive loss as a percent of full load power $= 0.05 + 0.54 * \%FP - 0.51 * \%FP^2$ Eq. (7)

Example of Savings Overestimation

Opening a pump system balancing valve and replacing throttle control with a 50 hp VFD results in a reduction in power from 35 to 15 hp. Without accounting for motor losses, savings is 14.9 kW.

Accounting for the 2.6% of full load drive losses at 30% flow reduces the savings by 6.5 percent and 1.0 kW to 13.9 kW.

Omitting drive losses results in overestimating savings by at least two percent and up to ten percent or more at low flow rates.

8. MEASURE FULL FLOW POWER, RATHER THAN ASSUMING IT IS THE SAME AS MOTOR NAMEPLATE OR DESIGN POWER

Few motors and drives operate at 100 percent of rated power at 100% flow. Both design safety factor engineering and practical considerations result in oversized motors. "First pass" and "rule of thumb" type savings estimates based on motor nameplate power as equal to full flow power will routinely overestimate savings by 30 to 40 percent. Four reasonable reasons why full flow power is not the same as nameplate power are:

1. The fan or air handler component designer matched the "next largest" nominal motor size to their fans when shipped from the factory. Even when maximum design power is just below the next higher nominal power, 14.5 bhp for example, a designer may reasonably specify a 20-hp motor be paired with a fan or pump instead of 15 hp to ensure that unanticipated head pressure can be overcome.

2. The fan or pump system designer must account for the wide variety of system losses described earlier, plus many others. It is only prudent to add a safety factor to the calculated losses. That safety factor, if not necessary, means additional oversizing.

3. The fan system designer must also allow for filter dirt accumulation that is not always present. This head pressure allowance can be a quarter of the total system design pressure. Under most circumstances dirt accumulation is less than the maximum.

4. Urgent replacement of burnt out motors by facilities staff can result in upsizing. Emergency replacement of burnt out motor with one of the next largest nominal hp size but same frame size to get the plant back operating as quickly as possible is not uncommon.

Since modern motors are designed to run at peak efficiency near 70 percent of full load power and have relatively constant efficiency down to about 50

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[14] Esource (11), Fig 11.6, p.11.3. The curve is for fans but should be applicable for pumps as well. Esource describes their basis as "composite curves developed from diverse resources including drive testing and theoretical modeling."

[15] The mathematical expression is an original derivation of laboratory test data in Chvala (8).

percent power, these decisions typically result in a negligible power penalty. Misestimating the full flow power can grossly distort savings calculations, however.

Mathematical expression of the factor

The best method to account for this factor by far is to open the dampers, valves, or vanes to the "100 percent open" position and measure actual power at full flow with a real power meter. Then, divide by the motor efficiency and 0.746 to get full flow bhp.

If a real power meter is not available, data are available to estimate power factor and in turn real power based on measured volts, amperes, and phase.

Example of Savings Overestimation

Overestimation is proportional to the overestimation in full flow power. If actual full flow power is 30 percent less than that used for savings calculations, then savings will be overestimated by 43 percent (1 / 0.7).

SUMMARY

This paper provides curves for fans that analysts can use to estimate part load energy use for 13 different fan part flow control systems and six types of pump part flow control. The curves apply to "standard" conditions. Non-standard conditions affect energy use and savings potential. In almost every case, deviation from standard conditions decreases the savings that variable speed drives can realize compared to other means of control.

When auditors are considering a fan or pump system for variable speed drive retrofit, look out for these features, among others, that can reduce savings potential:

- Open pump systems with elevation change
- Fan systems with HEPA filters or an unusually high number of filters and coils
- Pump or fan systems with downstream modulation at point of use, especially when combined with minimum pressure settings
- Pressure booster pumps
- Pumps and fans below 20 hp with flow rates well below the maximum
- Systems modified through the addition or removal of equipment that results in full flow operation that is well off of the originally design flow-head combination.

The paper provides analytical techniques to account for the effect of most of these features on energy use.

REFERENCES - PART LOAD CURVE GRAPHS CROSS-REFERENCE INFORMATION

The part load curve data in Fig. 1 came from the sources noted in the table below. In many cases curves were available from multiple sources. When this occurred, one curve was selected or curve data was combined as described.

Fans

Any – Bypass	LBL p. IV.209 (and self evident), (14)
Backward curved centrifugal – Discharge dampers	ASW Application Note (4).
Airfoil centrifugal – Inlet vanes	CEC, p. 4-48 (5).
Air foil centrifugal – Discharge dampers	CEC (5).
Backward curved centrifugal - Inlet vanes	ASW Application Note (4).
Forward curved centrifugal – Discharge dampers	CEC (5).
Forward curved centrifugal - Inlet vanes	CEC (5).
Any – Motor-generator set	EPRI (9).
Any - On-off cycling	LBL p. IV.209 (and self-evident), (14).
Vane-axial – Variable pitch blades	OSU (18).
Any – VFD	EPRI (9) combined with Esource (11).
Any – VSD Magnetic Coupling	Chvala Table B-1 averaged and normalized to 100% (8).

Pumps

Bypass	LBL p. IV.209 (and self-evident), (14).
Throttle	OSU (18).
On-off cycling	LBL p. IV.209 (and self-evident), (14).
Motor-generator set	None found. Used same curve as fan.
VFD	OSU (18).
VSD Magnetic Coupling	Chvala (8).

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