WHITE PAPER

Modern Pump Selection:

Maximizing Benefits of Variable Speed Control

By JMP Equipment Company

WHITE PAPER

The key to variable speed pump performance has as much (if not more) to do with pump selection as it does with the application variable speed controls. This white paper will go over selection criteria to help engineers maximize efficiency and avoid common operational problems due to improper pump sizing.

Selection of pumps based on peak load conditions and the Best Efficiency Point (BEP) will undercut the savings potential for these modern systems. Throughout this paper we will refer to "Efficiency Islands" which enable engineers to optimize pump selection for the load profile of the system. We'll talk about real world conditions versus theory, variable versus constant head and how to apply integrated part load values.

Efficiency Islands

Most of us realize that centrifugal pumps in closed hydronic systems typically only operate at peak load for a short period of the time. So it doesn't make sense to select a pump based on its performance at peak load; it makes much more sense to select a pump based on a wider range of conditions.

Such a selection will generally yield far more savings than a pump selection based on the singular "sweet spot" of Best Efficiency Point (BEP). Instead of focusing on that point where the system curve intersects with the BEP, we need to look at a broader operating range, the boundaries of which form what is called an "Efficiency Island."

Efficiency Islands are bound vertically by the minimum and maximum impeller diameter performance and horizontally by the iso-efficiency lines, those concentric elliptical curves that "wrap around" the BEP. Iso-efficiency lines are based on what the Hydraulic Institute has deemed as the Preferred Operating Range for a given pump, which is defined as the flow range from 70 to 120 percent of the pump's BEP.

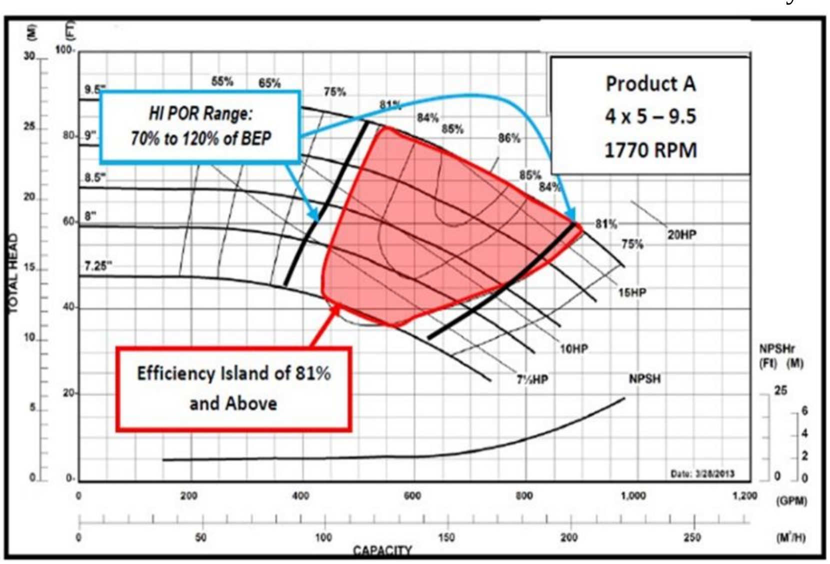

Figure 1

The part shaded in red in Figure 1

represents the Efficiency Island for a particular pump. This can be used in conjunction with the system curve to select a pump that will operate efficiently over most of the system's load profile. A pump will always operate at the point where it intersects the system curve. Of course, in most systems, the system curve is constantly moving as valves close and open in response to demand. Therefore when selecting a pump we must consider the entire operating range of the system, which is defined by the control area (Figure 2).

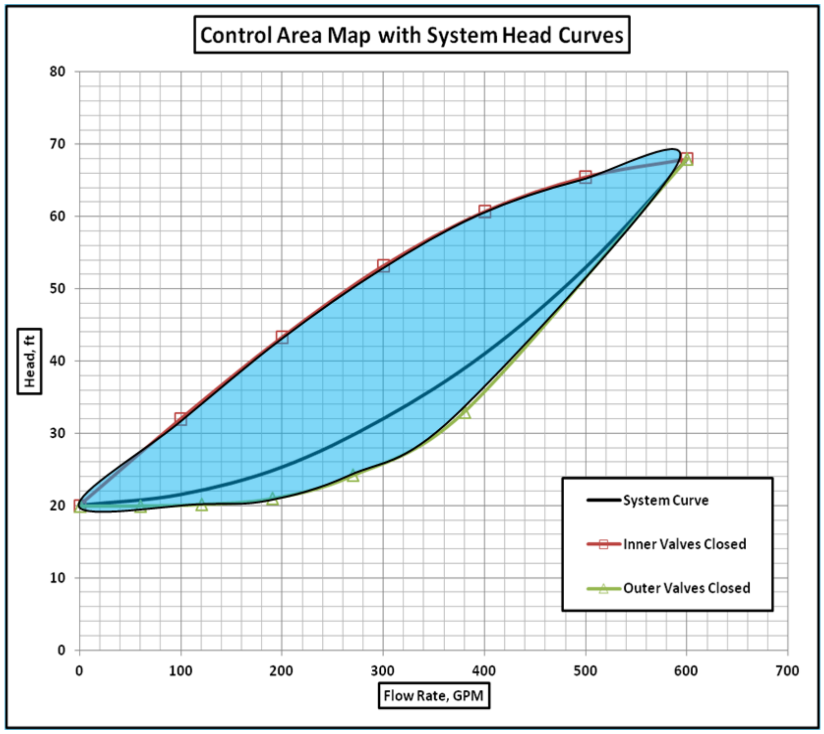

Figure 2

If we transpose the upper and lower system curves on a given pump curve, as we have in Figure 3, it is apparent that the best overall operating efficiency will occur with pumps that have the deepest and widest efficiency islands. This essentially expands the cross section area between the control curve and the Efficiency Island. If high efficiencies are confined within a narrow flow range then there will be a rapid drop in pump efficiency as the impeller diameter is reduced. This is especially important to remember when selecting pumps with Variable Frequency Drives (VFDs) because VFDs enable pumps to operate as though they have various impeller trims.

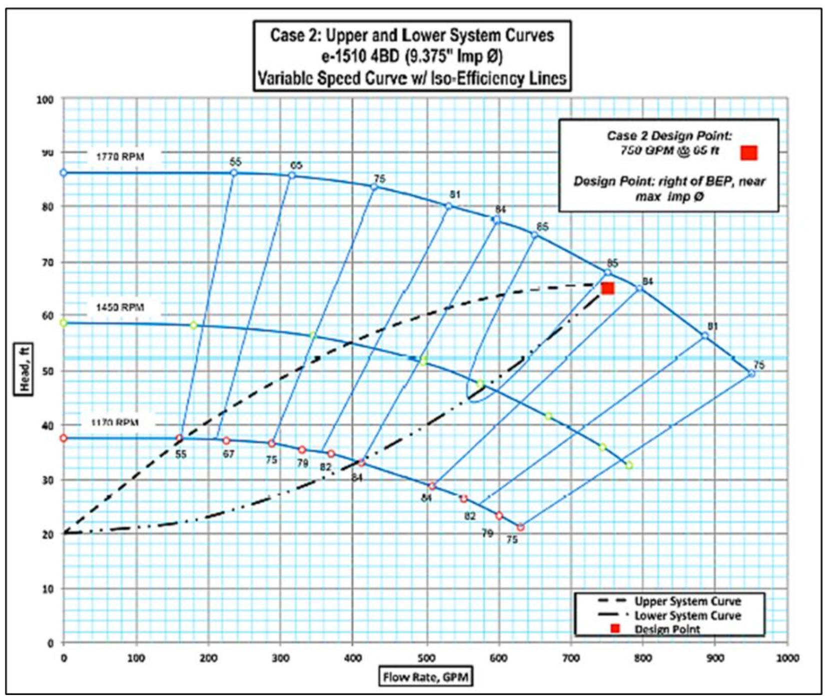

Figure 3

Notice also in Figure 3 that the design point is located just outside of the pump's BEP. That's perfectly acceptable since this pump moves back through the higher Efficiency Islands as it unloads, maintaining an efficiency of 82% even at 50% of the design flow of 750 GPM.

Variable Speed Pumping

Theory Vs. Real World Operation

Variable speed pumps decrease in rotation as demand drops, but that doesn't mean that zero demand results in zero pump speed. In closed hydronic systems with 2-way valves there is almost always some impeller rotation – even when variable speed drives are employed.

That's not to say variable speed pump drives don't save a lot of money over their operating life. But what variable speed can do in theory may lead to inflated expectations of what variable speed can do in a real life hydronic system.

The efficiency projections of variable speed pump operation are based on the pump affinity laws. These laws state that:

1. Pump GPM capacity varies DIRECTLY as the speed (RPM) or impeller diameter ratio change.
2. Total pump head varies directly as the SQUARE of the speed (RPM) or impeller ratio change.
3. BHP varies directly as the CUBE of the speed (RPM) or impeller diameter ratio change.

Plot these mathematical facts on a graph that relates directly to pump flow as we have in Figure 4 and the numbers are pretty compelling.

The pump curves at various speeds are shown in black. The green line represents the system head curve and the purple line represents the brake horsepower as it corresponds to the various speeds. A 50% reduction in speed will result in a head loss reduction from 100% to 25% and a brake horsepower reduction to 12.5%. In other words, per the Affinity Laws, if we cut the speed of the pump in half, the head drops by the square to 25% and the brake horsepower drops by the cube (100 x 0.125 = 12.5% BHP).

The numbers don't lie, so why is it that we never actually see such reductions in real world applications of variable speed pumps? It is because the theory assumes 100% variable flow, which simply does not occur in typical closed HVAC systems where there is always some amount of constant fixed head loss.

When designers perform system head loss calculations as prescribed by ASHRAE 90.1, they must keep in mind that only a portion of this head will vary with load. This is because in closed systems with 2-way valves at each circuit, a constant fixed pressure must always be maintained at the coil even at zero flow. Otherwise, if there is a sudden requirement for 100% flow, causing the 2-way valve to open, there would not be enough differential pressure at the coil to establish full flow through the heat exchanger.

Figure 5 shows the piping of a constant primary/variable secondary chilled water

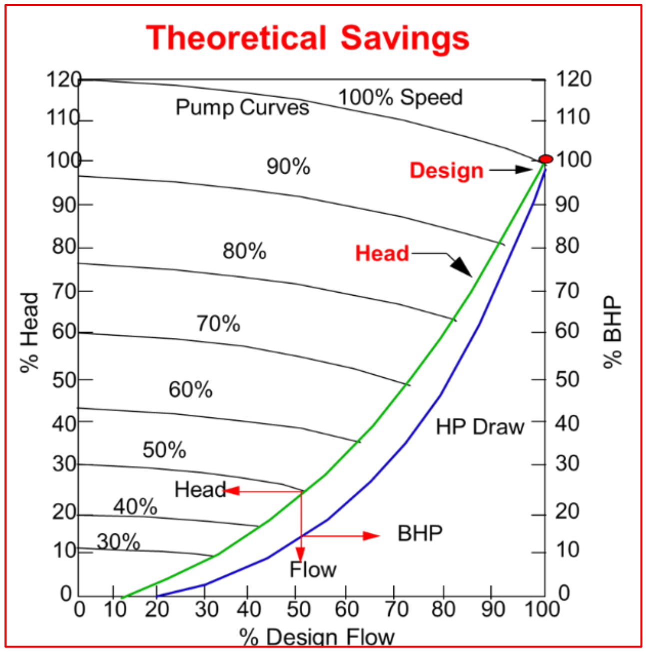

Figure 4

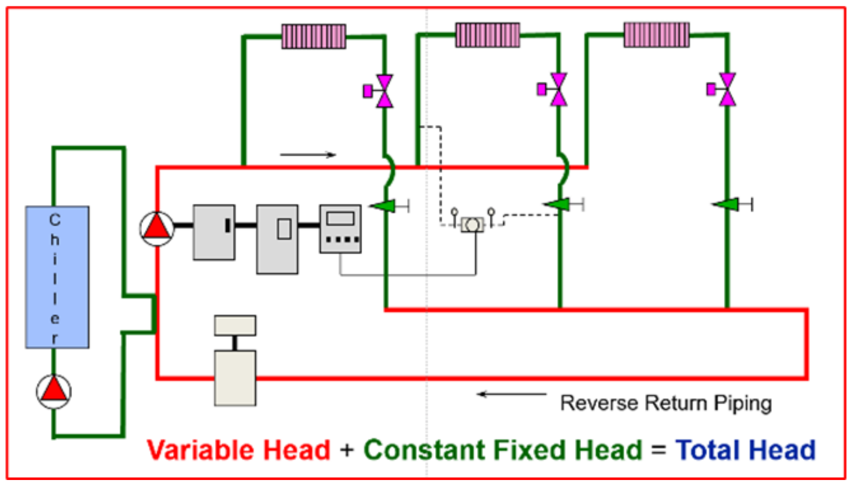

Figure 5

system. The variable head is shown in red and the constant fixed head is shown in green. The pump head loss calculations are shown in Table 1.

In this particular system, the pump has a variable head of 52 feet. This value defines the range to which the pump speed can be varied. The potential for variable speed bottoms out precisely at the minimum head required at the coil, which in this case is 28 ft. of head (the sum total of head loss through the coil piping & circuit setter, the two control valve and the coils. The actual system (control) curve doesn't begin at 0 GPM and 0 head. It begins at 28 feet and 0

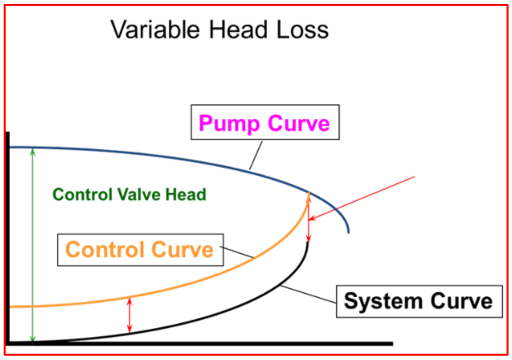

Figure 6

flow and might look something like the control curve shown in orange in Figure 6.

This is the reality of hydronic system design. The payback of variable speed systems will always be limited by the coil piping, control devices, and the coil itself.

Impact of Constant Fixed Head On Pump BEP

Even at no flow, the impeller must be spinning enough to produce some amount of pressure at the 2-way valve in order to quickly establish full flow at the critical circuit if demand suddenly occurs. It is important to understand how this impacts the pump's best efficiency point, or BEP.

Figure 7 shows a variable speed pump curve. The purple curve represents the system curve, which also happens to be at the BEP. The blue vertical example line indicates the operational points for this pump at 50% flow, which in this is 400 GPM or one half of 800 GPM. If this pump were applied to a system with no constant fixed head and all variable head, it would deliver 85% efficiency at every speed.

But what happens if we have a constant fixed head, as most variable speed HVAC systems do? In other words, what happens to the pump efficiency when you increase the minimum control head or differential pressure set point?

For example, let's say there is a minimum differential pressure of 40 feet. Figure 8 shows the same pump curve with a

Pump Head Calculation

Variable Head + Constant Fixed Head = Total Head

52 ft + 28 ft = 80 ft

Component		Head Loss
Pipe - 1756 ft of 10" Pipe at a Flow Rate of 2200 GPM		40.4 Ft.
Elbows - 15 - 10" Flanged Reg Ells Total Equivalent Feet (210) (2.3 ft per 100 ft)	52 ft	4.8 Ft.
Valves - 10 - 10" Gate Valves Total Equivalent Feet (28)		.6 Ft.
Check Valves - 3DS-8" Triple Duty At 1100 GPM		1.2 Ft.
Coil Piping & Circuit Setter	Coil & Valve 28 ft	3.0 Ft.
Control -- Two Way Control Valve On Cooling CV=292 @ 733 GPM		15 Ft.
Coils - Cooling Coil Pressure Drop Pressure Drop Is (Catalog Tables)		10 Ft.
Rolairtrol - RL-10 With 2200 GPM Flow	+	5 Ft.
Total System Head Loss @ 2200 GPM		**80 Ft.**

Table 1

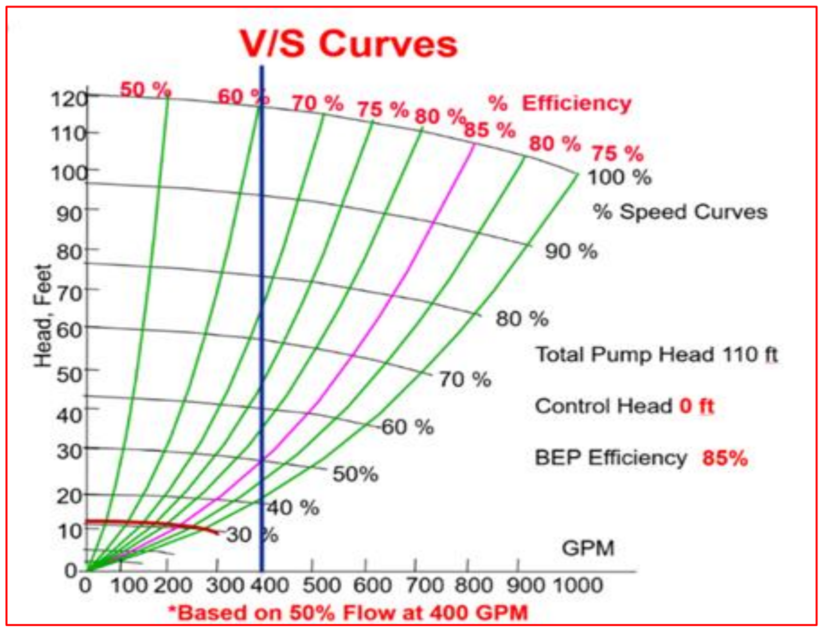

Figure 7

control curve of 40 feet of constant fixed head. The purple line is the new control curve.

Notice that at a full flow of 800 GPM we have 85% efficiency, but as we reduce flow to 50% or 400 GPM, the BEP efficiency drops. In this system with this particular pump, we have a BEP of 85% only at full flow. If we reduce flow to 400 GPM, then our BEP drops to 72%.

The intersection of the dark blue line and the purple line shows that the pump will have to operate at 70% speed in order to deliver 50% of flow. This in effect shifts the efficiency curve (shown in orange) to the left, somewhere between 70 and 75%. The orange line indicates the operating efficiency of the pump (approximately 72%) under these load conditions. If we were to continue to increase the constant fixed head, efficiency would continue to diminish.

Pitfalls of Over Heading Pumps

Variable speed drives are an excellent way to enhance the efficiency of a well-designed system. They are not and should never be thought of as an insurance policy against an imperfect pump selection. It is okay to be slightly conservative in your pump selection if you are not 100% sure how the pumps are going to be piped, but common sense must be applied to the pump head calculation. Also, don't forget ASHRAE 90.1's requirement that a detailed pump head calculation be performed and documented.

An over headed pump operating at variable speed can cavitate at full flow conditions. Granted, full flow only occurs about 1% of the time, but pump cavitation creates serious and costly problems that should be avoided. The variable speed pump curve in Figure 9 illustrates the consequences.

In this system we have a pump capable of generating 110 feet of head at a design flow of 800 GPM. At 100% speed we're fine, but what if we don't need 100% speed to achieve full flow? The variable speed drives will begin to slow the pump down. At 80% speed the pump falls off its curve and begins to cavitate, creating explosive noise and vibration. The pump impeller is taking a huge beating.

The only way to resolve the situation (without buying a new pump) it is to throttle the pump at the discharge valve – essentially adding resistance to the system. This, of course, negates the advantages of variable speed drives and violates ASHRAE 90.1 (and the building

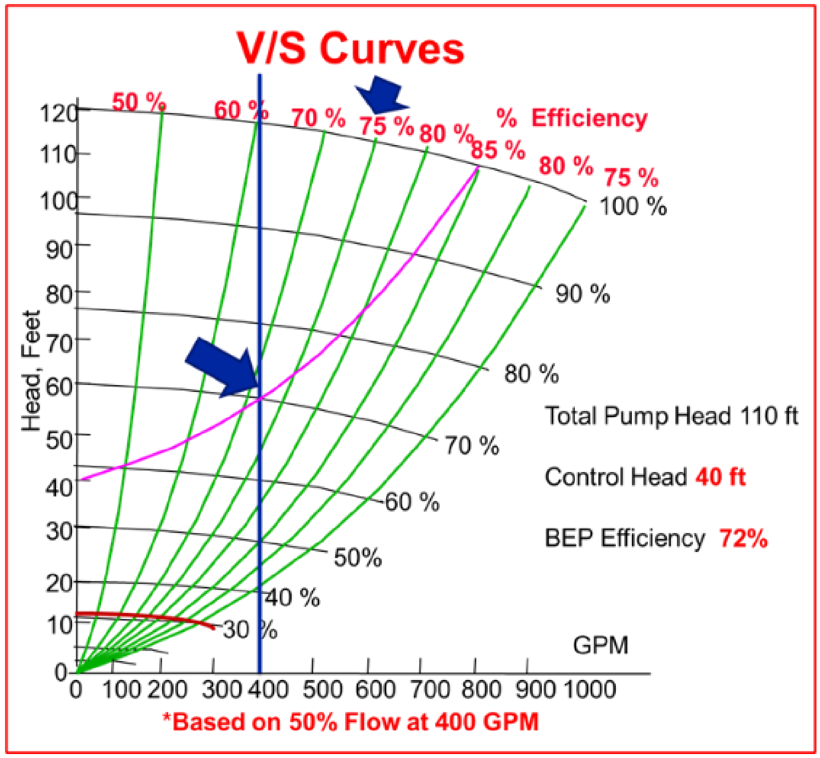

Figure 8

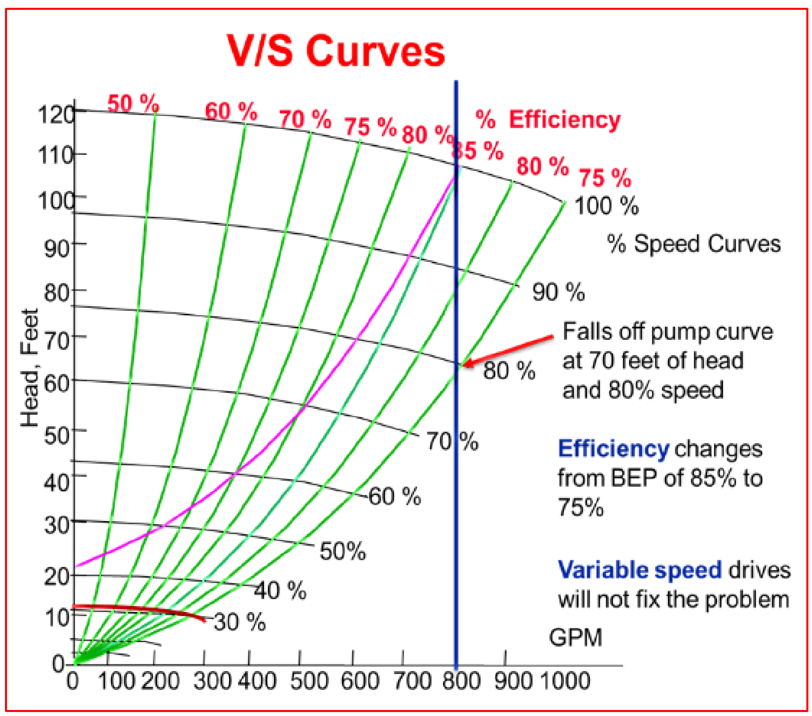

Figure 9

code) which says you must minimize pump throttling losses.

Clearly, it's better to avoid this situation in the first place by improving your pump selection. Start with a more accurate calculation of your total system resistances and design flow. Don't select

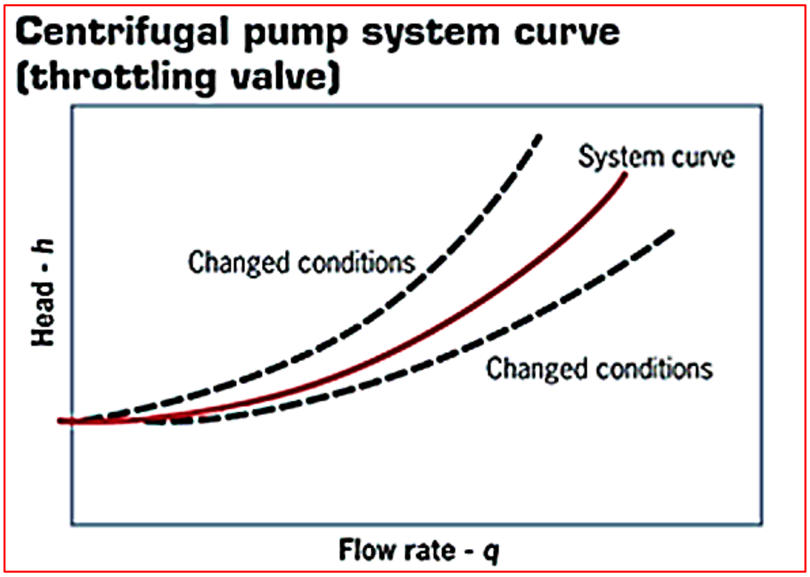

Figure 10

a pump for 110 feet of head when 70 feet of head will do. Furthermore, don't select a pump based on BEP at full flow conditions, which only occurs about 1% of the time. It's better to select the pump based on Efficiency Islands as discussed earlier in this paper. This means selecting a pump that will operate to the left of its BEP most of the time.

Smarter Pump Selection

How the System Curve Moves

Effective and efficient pump selection demands a thorough understanding of the system curve and how it reflects what happens in the actual system. This is especially true in today's variable speed systems.

The system curve pivots up or down with each and every flow change that occurs in a live system (Figure 10). As 2-way valves open to increase flow, the system curve pivots down. As 2-way valves close to reduce flow, increasing system resistance, the system curve pivots up. Consequently, the operating point of a variable speed pump (where the system curve intersects with the pump curve) will also move either to the right and down (valves opening), or to the left and up (valves closing). Every point in between represents the control range of the pump in a given system.

While the pump curve never changes, the speed at which the pump operates will. Therefore, we must consider not just the 100% speed curve, but the entire operating range of the pump that is represented by all the parallel pump curve lines on a single pump curve.

If we put a dot at every corresponding operational point that occurs in a system over its entire operating range, there would be a concentration of dots in the areas where the system operates most frequently. Throw a lasso around all those dots and you've captured the control range for your pump (Figure 11).

Notice the lower blue line with the triangles. This curve represents the operation of a multi-zone system when all of the outer valves (those furthest from the pump) are closed. The blue line with the squares reflects what is happening in the same system when the inner valves (those closest to the pump) are closed.

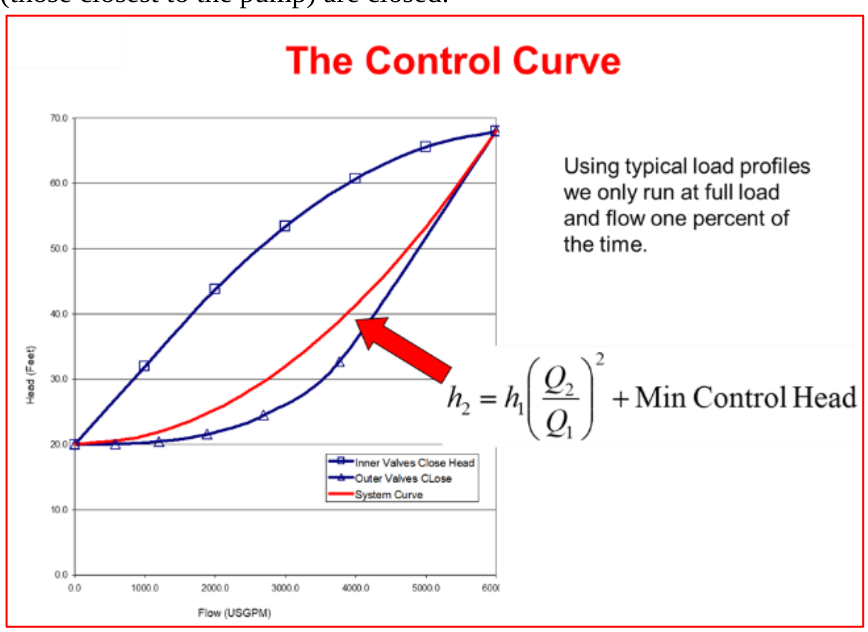

Figure 11

Why the distinction between the two? Because in a direct return system under part load conditions, it takes more pump head to drive flow to zones that are furthest from the pump than it does to drive flow through zones closest to the pump.

We want to pick a pump in which the areas of highest efficiency are mostly contained within the operating range of our system. Assuming we have done a detailed system head loss calculation and we know that the system will be installed as designed, we can pick a pump accordingly, maximizing its operating efficiency in our system.

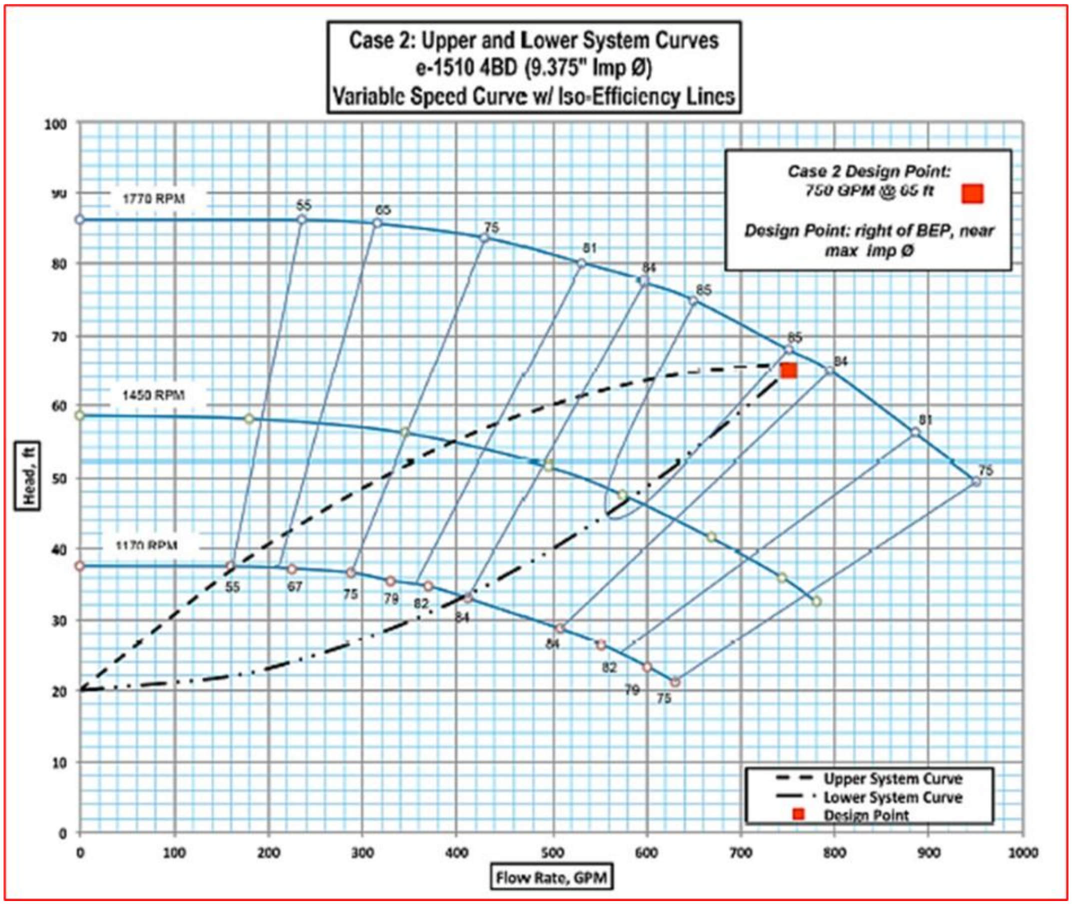

Figure 12

This method of selection places our design point just to the right of BEP, or the best efficiency point for our pump (Figure 12). That's fine because in a typical variable volume system the design conditions only occur about 1% of the time. Notice that most of the operating range of the system keeps the pump operating at or above 75% efficiency. However, ASHRAE recommends that we select a pump so that the design point of our system falls just to the *left* of BEP, which typically isn't quite as efficient. Why is that?

If we are bold in our quest for maximum efficiency and pick the pump so that our design point is just to the right of BEP, then we run the risk of unstable operation.

Ultimately it is up to the engineer to decide how confident he or she is in the accuracy of the system head loss calculations. In either case we want as much of the system operating range to fall within the efficiency island of the pump as possible.

Worst case scenario, our pump falls off its curve and cavitates. For that reason, ASHRAE recommends building in a little bit of a safety factor by picking a pump to operate slightly to the left of BEP. Specifically ASHRAE states:

"Where possible, pumps should be chosen to operate to the left of the BEP because the pressure in the actual system may be less than design due to overstated data for pipe friction and for other equipment. Otherwise, the pump operates at a higher flow and possibly in the turbulent region."[1]

Applying Integrated Part Load Values

The key to efficient pump selection for modern day systems is establishing a load profile. We want to select a pump that is most efficient at the heads and flows that

[1] 2012 ASHRAE Systems and Equipment Handbook, p 44.11

dominate our load profile – where we will be operating most of the time.

In a perfect world the engineer will have calculated an exact load profile for the system he or she is designing. If this isn't possible, the Integrated Part Load Value (IPLV) load profile prescribed by AHI Standard 550/590 (Table 2) may be used. Although the IPLV was developed for chillers, it can be used for chilled water pumps and boiler pumps as a means to evaluate part load pump efficiency.

It is reasonable to assume that pump flow rates will match the percent of chiller load. Obviously, we want to focus on 50 to 100% chiller load because that's where we will be operating approximately 88% of the time. We just have to make sure that the pump is capable of delivering the minimum amount or "constant head" the system requires in addition to peak load.

Let's assume we've picked our pump based on the IPLV load profile. We've calculated our pump efficiencies according to the IPLV recommendations and we've calculated our annual operating cost as shown in Table 3.

Notice that this calculation is based on 30 feet of constant head. Again, it is absolutely critical that we recognize that there will always be some constant head requirement. No pumping system with 2-way valves is ever 100% variable! Failure to acknowledge this in our selection will result in a grossly under-estimated system operating cost.

To help illustrate this point, we've calculated the annual operating cost for the same system as above, based on zero constant head and 100% of variable flow (Table 4). Notice the operating cost is much lower than it was when we had 30 feet of constant head: $10,057.43 vs. $14,502.21. That's a $4,444.78 mistake that no one wants to explain to an owner.

Using the IPLV as a guide for selecting pumps for today's variable speed systems is perfectly acceptable as long as you include a reasonable amount of constant

ARI 550/590 Standard

"IPLV" Pump Efficiency - Load Profile

(Integrated Part Load Value)

Based on 40% constant fixed head

IPLV Formula Weighting Factors & Water Pump Flow Rates

Chiller Load	Weighting	Pump Flow Rate	Pump kw	Run Point	Pump Efficiency	Operating Hours
100%	1%	100%		A		
75%	42%	75%		B		
50%	45%	50%		C		
25%	12%	25%		D		

$$\text{Pump IPLV} = \frac{1}{\frac{1\%}{A} + \frac{42\%}{B} + \frac{45\%}{C} + \frac{12\%}{D}}$$

expressed in blended efficiency

Note: Assume pump flow rates match % load

Table 2

Load	Hours	Flow GPM	Head Feet	RPM	Pump Eff.	BHP	Drive/ Motor Eff.	kWHR	Cost/day	Wire/Water Eff
25%	2.88	500.0	34.1	964	68.06	6.32	88.48	15.34	$1.53	60.2%
50%	10.80	1,000.0	46.3	1161	85.70	13.63	88.14	124.55	$12.46	75.5%
75%	10.08	1,500.0	66.6	1439	88.06	28.64	87.43	246.23	$24.62	77.0%
100%	0.24	2,000.0	95.0	1759	88.52	54.23	86.62	11.20	$1.12	76.7%

Variable Speed Operating Cost	
Total Kilowatt Hours = 145,022.1	Cost per kwhr = $0.10
Total Hours per Year = 8,760	Annual Operating Cost = $14,502.21

"30 Feet Constant Head"

Table 3

Load	Hours	Flow GPM	Head Feet	RPM	Pump Eff.	BHP	Drive/ Motor Eff.	kWHR	Cost/day	Wire/Water Eff
25%	2.88	500.0	6.9	466	88.44	0.98	83.25	2.54	$0.25	73.6%
50%	10.80	1,000.0	24.6	891	88.56	7.00	88.51	63.74	$6.37	78.4%
75%	10.08	1,500.0	54.0	1325	88.53	23.11	87.72	198.04	$19.80	77.7%
100%	0.24	2,000.0	95.3	1761	88.52	54.35	86.61	11.23	$1.12	76.7%

Variable Speed Operating Cost	
Total Kilowatt Hours = 100,574.3	Cost per kwhr = $0.10
Total Hours per Year = 8,760	Annual Operating Cost = $10,057.43

"100% Variable Head"

Table 4

head in your calculation. If you are unsure of the exact amount of constant head, we recommend plugging in 40 percent to be safe.

Summary

Selecting pumps based solely on BEP and peak load will not yield the performance or efficiency that is expected from today's variable speed systems. The best way to optimize variable speed pump performance involves accurate head loss calculations, utilization of efficiency islands and incorporating integrated part load values into your selection.

Establishing a control curve that reflects the constant fixed head of the system also helps to deliver predictable results, since no closed hydronic system is ever at zero head – even when it is a zero flow. We must consider the entire control range of the system and choose the pump for best overall efficiency within this range. It may involve a bit more time but will offer the best payback and trouble-free pump operation for the life of the system.