Pump ED 101

Pump Selection for VFD Operation

Part 1 – Constant Pressure

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Introduction

All centrifugal pumps can be controlled by a VFD. That said, some are better suited for variable speed operation than others. In order to find the best fit, we have to take a very close look at the application's requirements. In doing so, we will find that a generic pump model that meets the head and flow requirements is seldom the best choice. A specific model, with hydraulic characteristics tailored to the application, may provide far better results. Another finding, that results from dissection of the application, is that not all applications benefit equally from VFD control. Some may be better suited to other modes of operation.

Although there are many pumping applications that can take advantage of changes in pump speed, constant pressure is one that resides at the very top of the list. There are two reasons that a speed change can be an important factor in this application. First, a change in pump speed can allow the pump to meet the pressure and flow requirements of the application without outside intervention (a PRV for example). Second, and even more important in many cases, a change in pump speed can effect a rather large reduction in the power required to operate the pump when demand is less than full flow. Additionally, VFD operation can provide several other significant benefits. With higher horsepower pumps the improved power factor that results from drive control allows more efficient use of electrical power. And, "soft" starts and stops decrease electrical and mechanical stress while reducing the potential for water hammer.

We will take a close look at a PRV and VFD controlled constant pressure application and provide some basic rules for pump selection. Additionally we will show examples of pumps that will work well and those that are less desirable. And, we will also discuss examples where VFD operation may not be justified. Although sizing can be somewhat subjective and based upon one's experience, I will try to be as objective as possible. If you are new to constant pressure applications download Constant Pressure Booster Systems for an overview.

Constant Pressure

A unique capability of the centrifugal pump is it's ability to operate normally under conditions that throttle its discharge. Since pressure is completely predictable across it's entire operating range damage, due to decreasing flow, will not occur if the system is designed to accommodate the increased pressure and temperature. Some designs can even be throttled to shut off without cause for concern (download the "Puzzler" for a comprehensive discussion of radial and axial forces). Conversely, a positive displacement pump, due to its inherent operating characteristics, requires a bypass if it is to allow throttled flow. If one were not installed, even a relatively small reduction in flow could cause the system to overload and destroy the driver or, for that matter, literally explode if the components could not handle the increased pressure.

Because of this innate ability, centrifugal pumps dominate booster applications where constant pressure must be maintained over a broad range of flows. The addition of a simple pressure reducing valve (PRV) on the discharge of the pump will provide some predetermined constant pressure across the entire flow range of the pump. Figure 1 is an example of a PRV controlled constant pressure booster. The blue, curved line is the head / capacity curve of the pump and the red, horizontal line is the system curve (constant pressure).

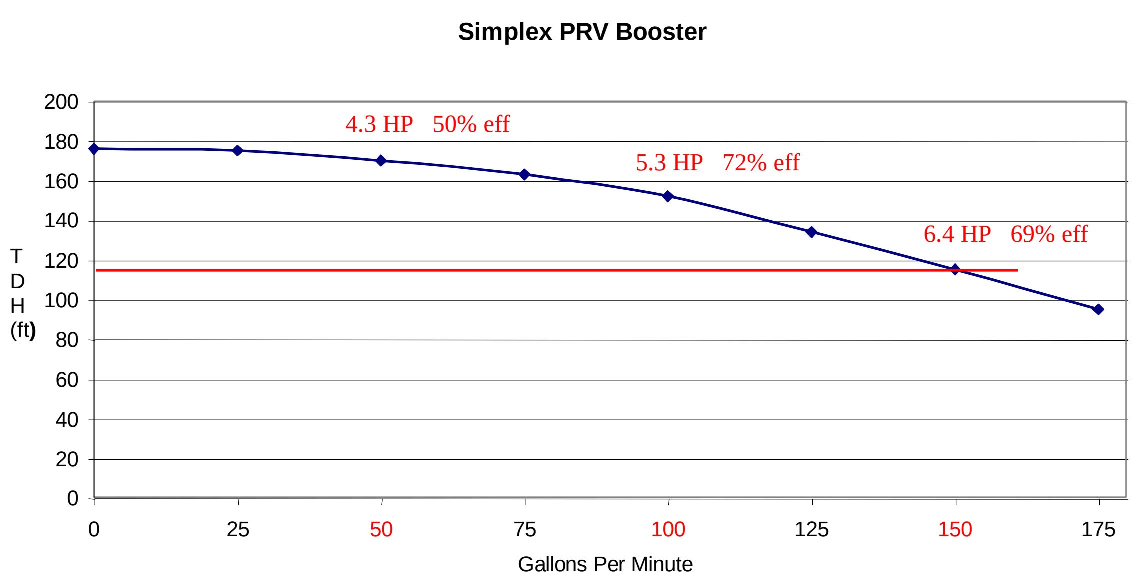

Figure 1

In this example, the PRV will provide a constant discharge pressure of 115′ from shut off to 150 gpm. At flows lower than 150 gpm, the pressure at the valve's inlet is that of the head/capacity curve. At 150 gpm and above, the valve is fully open and the pressures on each side are equal. You will also notice another unique trait of the centrifugal pump. As the pump's discharge is throttled and its flow decreases, so does the power consumed. Unfortunately, this reduction in power is not in direct proportion because the hydraulic efficiency of the pump also decreases as flow moves to either side of the pump's best efficiency point (BEP). When pump speed changes, however, the corresponding change in the power required is very different from that caused by throttling. Brake HP can be calculated at any point on the head / capacity curve by using the following equation:

BHP = (GPM x Head) / (3960 x Pump Efficiency).

The table in Figure 2 illustrates the effect of a change in frequency, and thus motor and pump speed, on the performance of a centrifugal pump. The top row shows decreasing frequency from 60 to 40 hz. The three rows below show the corresponding flow, head, and horsepower (as a percentage and rounded to the nearest whole number) predicted by the affinity laws. These values are valid for all points on the 60 hz head / capacity curve and their corresponding points. We will discuss these corresponding points in more detail a little later. If you were to "do the math" you would find that flow (%Q) changes directly with frequency, head (%H) changes as the square of a change in frequency, and power (%HP) changes as the cube. For example, at 50 hz (83% of 60 hz speed) flow is reduced to 83%, head is reduced to 69% and power is but 58% of it's 60 hz requirement. Our brains can deal fairly easily with a change in flow because it is a linear function. Reduce the speed by ½ and the flow is reduced by ½ . Double speed and flow doubles. Head and power, on the other hand, usually require a graphical representation for us to comprehend the effect, especially when multiple points are involved (i.e. the entire pump curve).

Hz	60	59	58	57	56	55	54	53	52	51	50	49	48	47	46	45	44	43	42	41	40
%Q	100	98	97	95	93	92	90	88	87	85	83	82	80	78	77	75	73	72	70	68	67
%H	100	97	93	90	87	84	81	78	75	72	69	67	64	61	59	56	54	51	49	47	44
%HP	100	95	90	86	81	77	73	69	65	61	58	55	51	48	45	42	39	37	34	32	30

Figure 2

Figure 3 illustrates this last point. Here we see the same pump under VFD control. Again, the red line is the system curve and the blue, violet, and green curves are the pump's head / capacity curves at 60, 55, and 50 hz. The single and fractional hz curves that are generated by the VFD are not shown in this example although they will always be there during VFD operation. The black angled lines (frequency

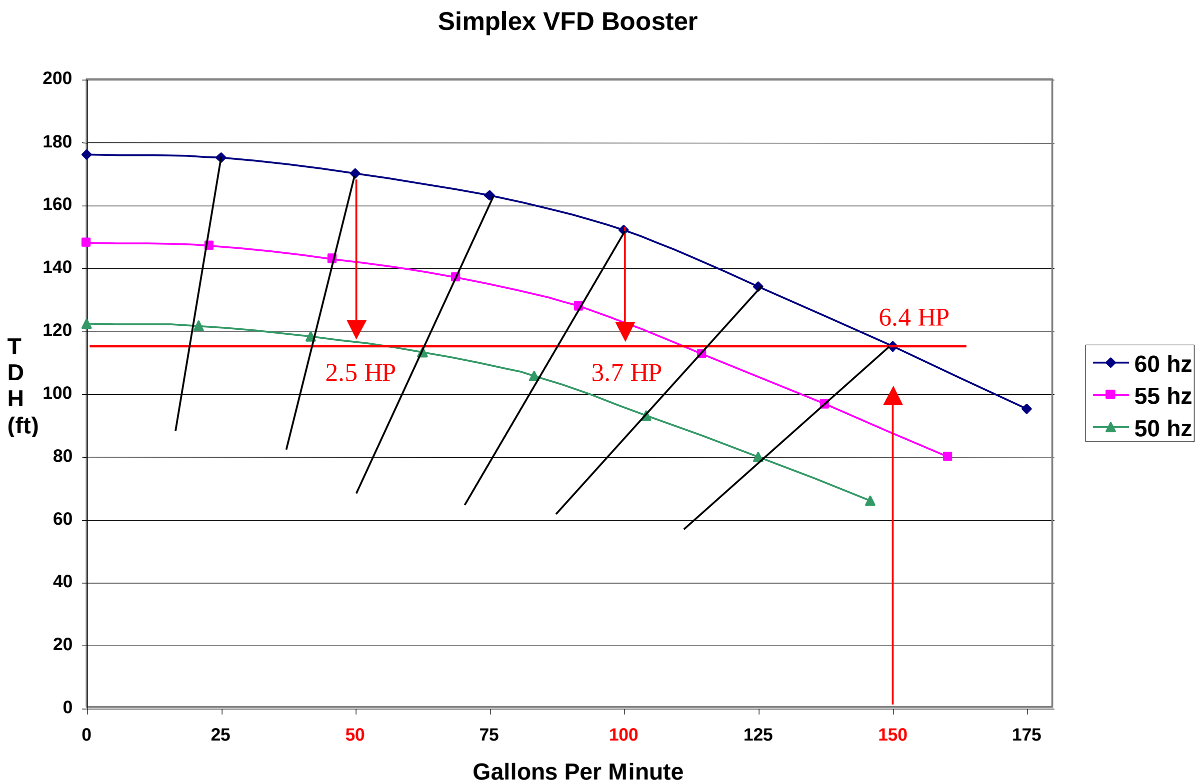

Figure 3

isomers) show the lower frequency, head / capacity points that correspond to that of the 60 hz curve. Even though they move to the left as frequency is reduced, their efficiency is equal or very similar to that of the corresponding point on the 60 hz curve.

The system curve not only shows the pressure but also intersects each operating frequency from shut off to 150 gpm. As you can see, the frequency range available to the VFD to maintain a constant pressure of 115' ranges from approximately 49 hz at shut off to 60 hz at 150 gpm. The BHP required at 150 gpm is the same as the PRV controlled system but as frequency, flow, and head decrease, the corresponding reductions in BHP are considerably larger than those of the PRV booster. Reductions in flow, head, and power follow the affinity laws and a major factor behind the third law (BHP required) is that the efficiency of each point on the 60 hz curve moves to the left with flow as speed decreases. Is this pump a good choice for VFD operation in this application?

Well, at 150 gpm, the corresponding points on the 55 and 50 hz curves (and all of those in between) are unusable because their pressures are below that of the system curve (115'). As flow decreases, however, lower frequencies become available. For example, for flows of 100 to 150 gpm a frequency range of 54 to 60 hz can be utilized and for flows of 50 to 150 gpm 50 to 60 hertz can be employed. This may not sound like a wide range but the frequency resolution of most modern VFD's is 0.01 hz. This means that the drive can produce fractional frequencies and, an application like the one above could utilize up to 1000 individual frequencies between 50 and 60 hz and thus provide accurate control of flow and pressure over an otherwise narrow frequency range.

Now back to our question – is this pump a good choice for VFD operation in this application? The answer is – maybe. Even though the frequency control range is narrow (10 hz), the resolution offered by many drives will provide accurate control from shut off to 150 gpm. Also we see a power reduction (over that of PRV control) that ranges from 33% at 115 gpm to 42% at 50 gpm. Accurate control and adequate power savings are important attributes of good pump selection. If the system were to operate in a range of 50 to 115 gpm the majority of the time, this pump could be considered a good choice. If the normal operating range were lower or higher (say 10 to 100 gpm or 75 to 150 gpm) there are probably better choices available. Why? Because 49 hz is the minimum frequency on the low end and pump efficiency begins to drop once the pump exceeds 100 gpm. In each of these cases power reduction would not be maximized and better pump choices are probably available.

Suppose this pump were supplying an application that requires a constant pressure of 115' and continuous flows of 125 to 150 gpm? In other words flow never exceeds 150 gpm nor does it drop below 125 gpm. Although this pump will meet these conditions, a much better choice would be one with a BEP between 125 and 150 gpm. Also, you would be hard pressed to justify a VFD over a PRV for this application because of the limited flow range and power savings.

The following are a several rules that should be followed when selecting a pump for VFD operation in a constant pressure application.

1 In a constant pressure application the goal is to select a pump that will meet the head and flow requirements and achieve the maximum power savings while operating in a reasonable control frequency range. An unnecessarily large control range can cause "hunting" and reduce control accuracy where as a very narrow range can limit control and reduce power savings.

2 Sketch a simple system curve for the application. It should show the desired pressure, minimum & maximum flows, and the average flow range expected during peak demand periods. If it is an existing application, install a non-invasive, recording flow meter and monitor flow for several days or weeks. If it is a new application use the design engineer's estimates and, if possible, verify them by monitoring a similar existing installation. Some constant pressure applications are more consistent and therefore more predictable than others. A process application, for example, will probably have similar flows during the same times each day. A municipal or high-rise booster, on the other hand, will experience daily and weekend variations. The more you know about the application the better your selection will be.

3 Select the highest efficiency pump that has its BEP as close as possible to peak average flow and still meets maximum flow at or slightly above the desired pressure. As a rule of thumb, maximum flow should not exceed BEP flow by more than 20%. This % varies by pump and tends to be greater with higher flow pumps. The objective is to keep hydraulic efficiency as high as possible to the right of BEP. The pump curve must also meet the conditions outlined in 4 below.

4 Typically, you should avoid flat pump curves and those that do not exhibit a continuous rise to shut off. Although they are often preferred for PRV controlled systems, flat curves tend to offer an extremely narrow frequency range which results in a lower power reduction under VFD control. Similarly, curves that fall towards shut off reduce the control frequency range and thus the power savings in that region. In our constant pressure example the head rise from 150 gpm to shut off is 60' or 52%. This allowed an 11 hz frequency range from full flow to shutoff and resulted in a 39% power reduction at mid curve. In my opinion, a minimum operating range of 13 hz or 60% head rise to shutoff is a reasonable goal.

5 Uses Hertz or a similar program to obtain a graphical comparison of potential pumps based upon the required pressure. You can adjust the

system curve, add frequency isomers, and enter efficiency and HP labels to give you a clearer view of a pump's overall performance.

Figure 4 is an example of a pump that meets or exceeds all of the selection criteria outlined above. The system design calls for flows of 100 to 750 gpm at a constant pressure of 90′. Peak demand averages 450 to 625 gpm. Head rise from maximum to minimum flow is 70′ and provides a frequency control range of 15 hz. Power reduction averages 30% at peak flow and increases to 55% at minimum flow.

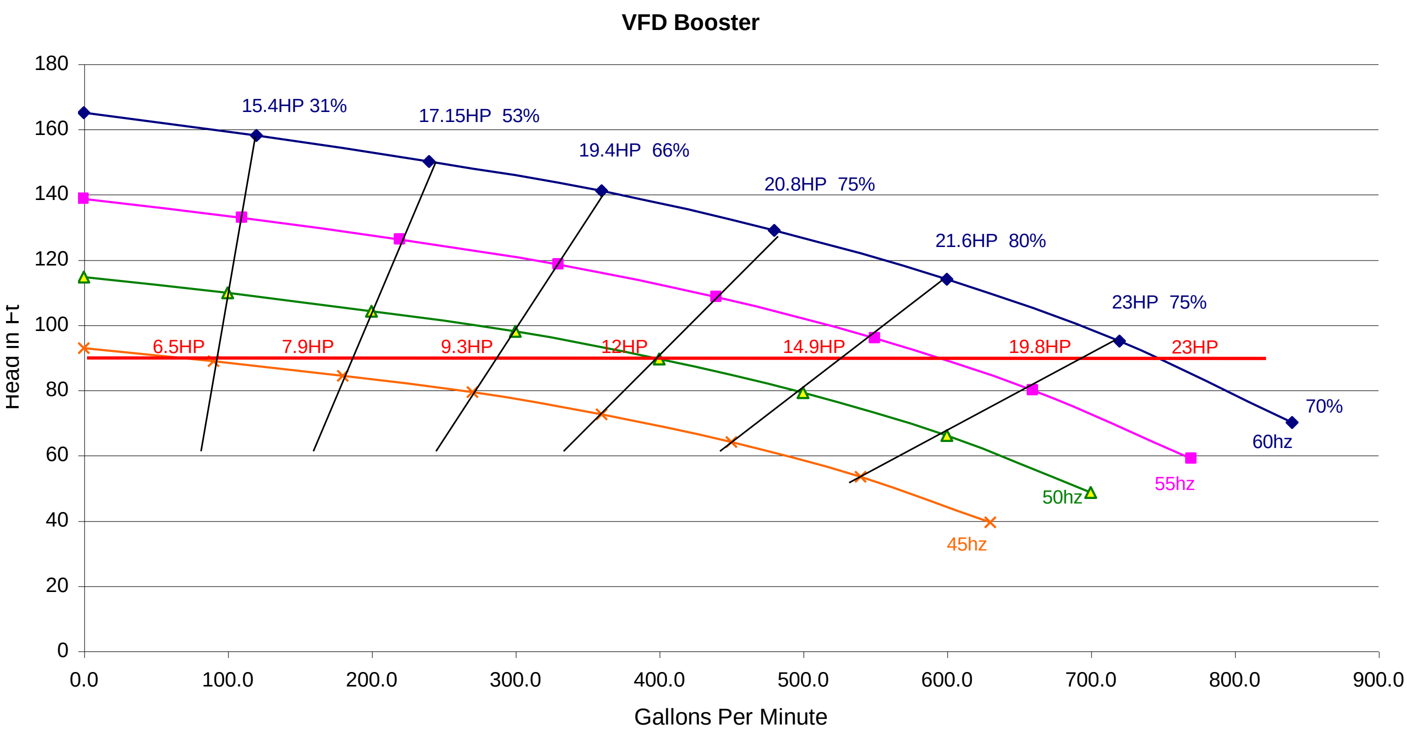

Figure 4

Figure 5 is an example of a pump performance curve that is too flat for effective operation via a VFD. The system curve shows a constant pressure of 120′ and a flow range of 0 to 95 gpm. The head rise from maximum flow to shut off is about 21′ or 17% which corresponds to a control range of 56 – 60 hz. Note that from shut off to 65 gpm frequency varies by less than 2 hz. Power savings at 80 gpm (BEP) is less than 10% and at 20 gpm it is about 19%. I stated earlier that flat curves are often preferred in PRV controlled boosters because of the increased power savings they can offer as flow is reduced but, just the opposite occurs under VFD control.

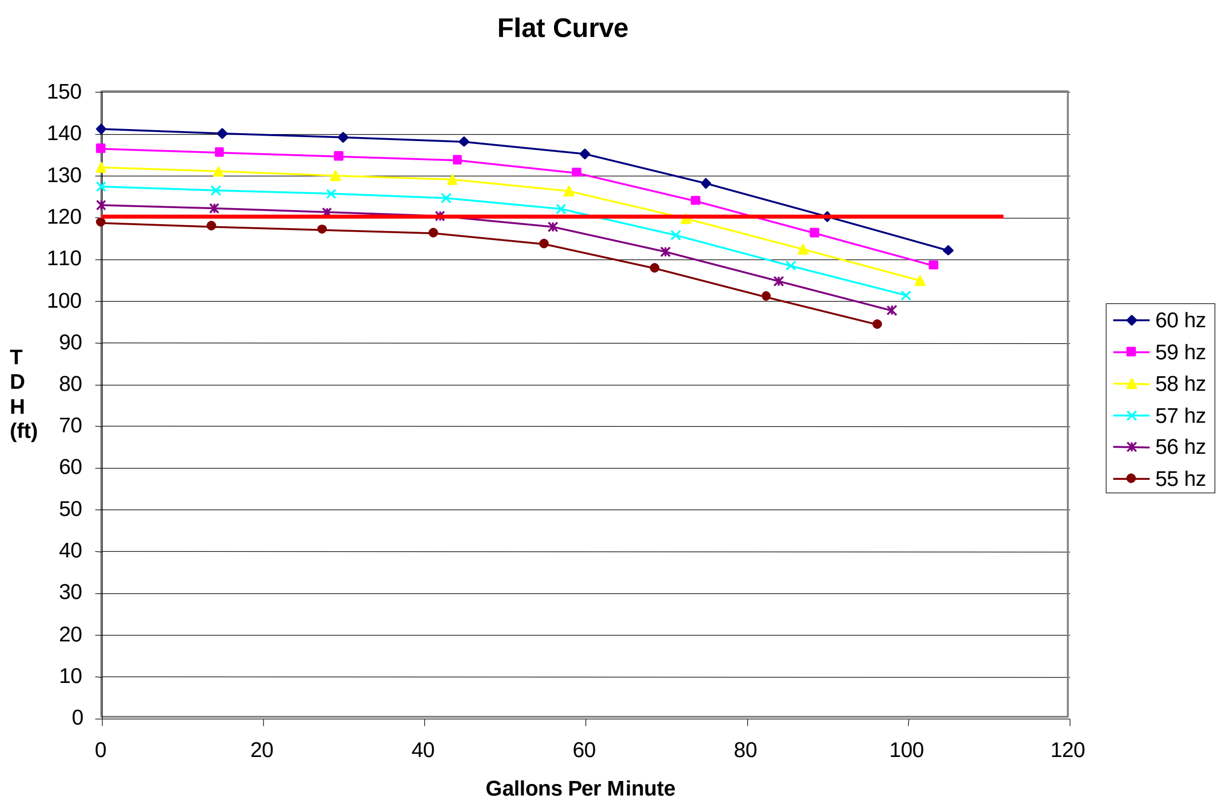

Figure 5

There are, however, applications where flat curves and VFD's are a good combination. We will discuss these in Part 2 of this series.

Down Stream Pressure Compensation

All of the applications we have seen so far provide constant pressure at the discharge of the pump (or the location of the pressure transducer). Unless the piping system is designed for minimum friction loss at full flow, down stream pressure will drop as flow increases due to an increase in friction in the system. In many applications this is not a concern because, in well designed systems, the drop is usually small. There are, however, incidences where it may be important to avoid any pressure drop down stream of the pump.

Figure 6 is an example of such an application. The system curve (in red) shows a constant pressure of 55′ for flows up to approximately 300 gpm. Above 300 gpm (the piping design point) it rises slowly and reaches 71′ at the maximum system flow of 600 gpm. This rise is due to an increase in system friction as flow increases.

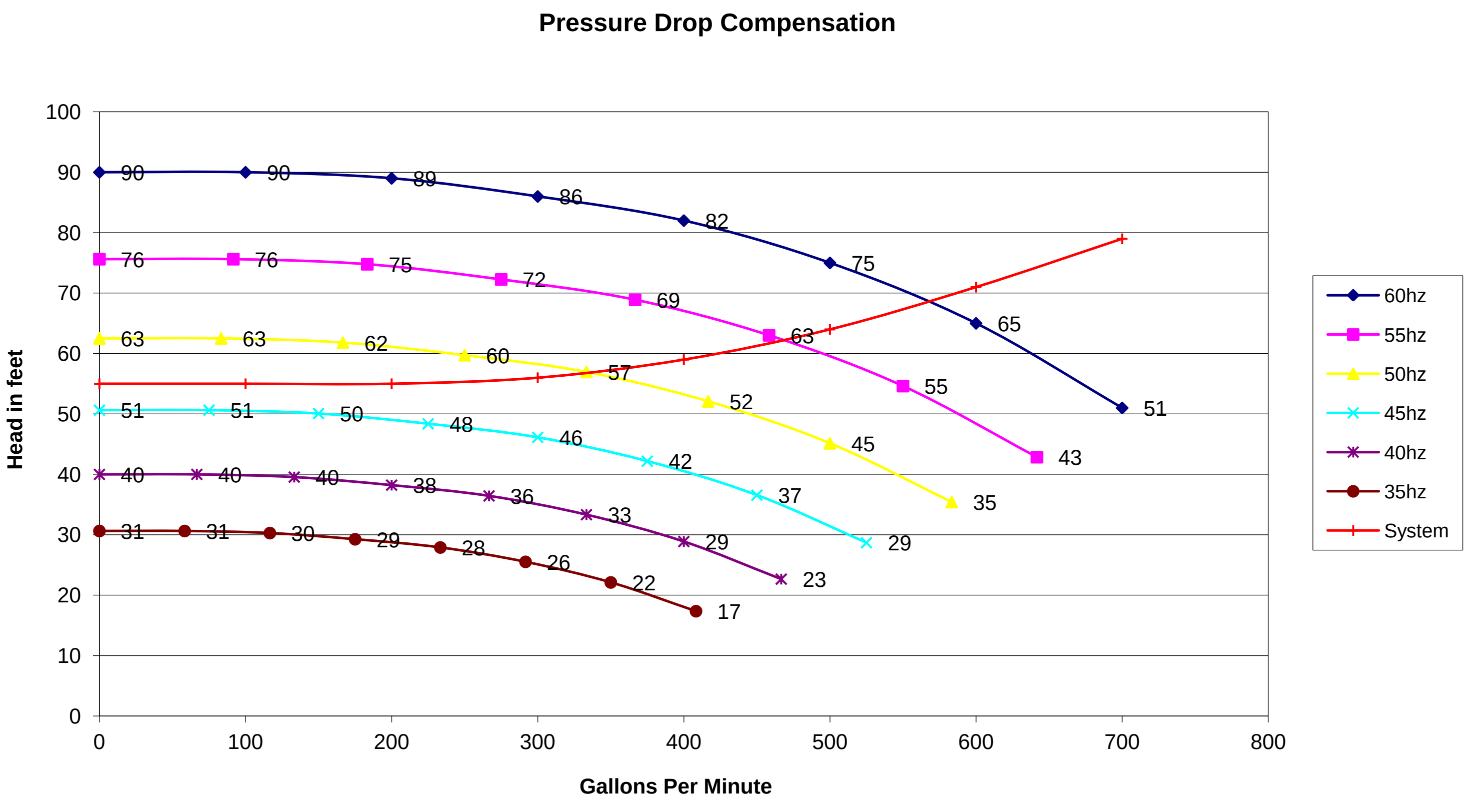

Figure 6

Constant pressure applications of this type are more difficult to control, however, a VFD can be programmed to compensate. The addition of a flow meter in the circuit can trigger the VFD to ignore the transducer as flow passes the piping design point. Above this point pressure is maintained by operating at some predetermined frequency points based upon flow meter input.

Joe Evans July 2004