



**QuantumFlo™**  
an exponential leap in technology

# Variable Speed Pumping Handbook

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**Forward:**

Over 12 years ago in 1997, some colleagues and I published the Pressure Booster Systems Designers Handbook (© Armstrong Pumps, Inc.). At the time of publication, we believed that this handbook represented the “state of the art” in current technology as applied to pressure booster systems. Many consultants agreed with us, however, like everything else, these technologies have evolved and improved. I would be remiss if I did not explain how these new technologies have changed those opinions, and what QuantumFlo, Inc. is doing as a manufacturer, to continually improve and refine the pump control process.

In 1997 it would have been unspeakable to discuss variable frequency drives in the same breath as “constant speed pressure boosters”. For one thing, they were “constant speed” and varying the speed did not seem to make any sense when applied to a system with a “fixed” static height. “These pumps didn’t need to slow down”, we would say. “The load is static; therefore, constant speed pumping is the best and only application suitable for these systems.”

Although we were correct in the assumption that the load was static and in the case of current centrifugal pump technology, the potential change in speed approaching shutoff is marginal; however, this small change in shutoff can have a significant effect on energy savings. If we consider the duty cycle of these packages, and then apply simple mathematics using the Fan & Pump Affinity Laws, we realize that even nominal speed changes create large energy cost savings! Furthermore, the application of variable speed drives today, allows us to reduce our reliance on “mechanically-controlled” pressure regulation, which has always been prone to high maintenance in the past. Additionally, we have been able to remove bladder tanks which pose not only a maintenance problem, but require larger floor areas and do not create any value in the system.

As “students of the industry”, QuantumFlo will never stop looking at ways that we can improve lives through our commitment to our representatives and customers.

*If you are receiving this publication through electronic distribution, please feel free to pass it along. If you have anything to add to this manual or you have a concern or question, please feel free to contact me via e-mail at [dave@quantumflo.com](mailto:dave@quantumflo.com) or you can call me in my office. My direct line phone number is (386) 753-9702, Ext. 302. I am also available to speak at your local ASHRAE or ASPE weekly meeting. Thank you for your contribution in making this world a better place to live, through quality engineering!*

## ***Introduction:***

The purpose of this manual is to introduce you to the proper sizing and selection of variable and constant speed pressure boosting stations. Due to the rapidly changing availability of technologically superior materials and electronic controls, much of the information in this manual was not even available until the recent past. We at QuantumFlo, Inc. hope that you gain a better understanding of the concepts of modern-day pressure boosting and urge you to notify us if you require additional copies of this manual which are freely distributed. We will attempt to make this manual widely available through the use of electronic publishing and distribution through our website. The most up to date information can be found on-line at [www.quantumflo.com](http://www.quantumflo.com).

### **1. Chapter 1 - System Types**

**Invariably, the booster system pressure control falls into two general categories; Constant Speed and Variable Speed.**

#### **A. Constant Speed**

Constant Speed Systems are packages which utilize “mechanical means” to reduce the system discharge pressure. Since the pump impeller diameter and speed is constant, the system requires hydraulically controlled mechanical forces to regulate pump pressure regardless of where it is operating along its performance curve to maintain a constant downstream pressure. This is the basis for the first systems ever manufactured and is typical of most manufacturers of constant speed systems today. The primary component for achieving this is the **pressure regulating valve (PRV)**. The PRV changes its pressure drop, adding or subtracting “friction head” as the pump moves along its demand curve, by opening and closing the valve seat through the use of a diaphragm. As the pump curve moves to the right to satisfy flow demand, the over-pumped head lessens and the PRV opens further, adding less friction head to allow more flow across the seat. (Fig. #1) Since the pump is falling in pressure as flow increases, the PRV does not have to add



as much friction head to regulate downstream pressure. Conversely, as the demand flow diminishes and the pump curve begins to rise in pressure, the valve closes, creating a higher pressure drop through the increased friction head added to the pump. In this way, the PRV is constantly reacting to maintain a constant downstream pressure as the flow demand increases and decreases. (Fig. #2) Unfortunately, in most cases, the demand on a plumbing system varies constantly, but is typically operating at a very low flow condition which means that the PRV must typically close off at or near closure to regulate these low-flow, high pressure scenarios.

### **The 80/20 Rule for Booster Systems**

*All buildings use water differently, however, a good general rule for booster systems is the 80/20 rule. Generally speaking, you can assume that 80% of the time the system demand flow is at 20% capacity or less. As a result, the PRV would be throttled nearly closed 80% of the time requiring the valve to add significant friction head which is wasted to heat.*

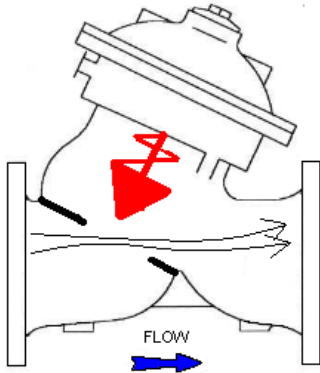
### **B. Variable Speed**

Variable speed pumping systems of today typically use a variable frequency drive to change the pump speed, thereby changing the flow and pressure that the pump is producing. These systems require a signal, such as a pressure or flow transmitter which sends an analog voltage signal to the input of the drive and the drive adjusts the speed to match the signal. If you want to maintain a 70 PSI system pressure, you would simply dial in the threshold of 70 PSI as a reference signal for the processor to compare and the drive will increase or decrease the speed of the motor and pump to follow that signal. As system demand and other factors change, the drive either ramps the pump up or down to match this signal. All of these values, including the ramping speed can be set on the drive using a PID (Proportional Integral Derivative) controller which is built into the drive logic.

*Note: We have discovered that variable frequency drive controlled pumps tend to be quieter than constant speed systems, suggesting that much of the sound generated with a constant speed pump is actually noise caused by PRV water velocity rather than fan, motor and bearing noise.*



Figure #1



Full Flow Operation - Valve is fully open imparting very little friction loss on the system. Pump is operating at or near its design point

**Show me the Money!**

It has been the question of the millennium. How can I save more money? Should I specify Constant Speed or Variable Speed? Is it more cost effective to run the pump at full speed most of the time and occasionally store water to a bladder tank or is it more cost effective to slow the pump down most of the time with only occasional shutdowns taking advantage of the reduction in power?

First you need to understand these control types and how they impact energy consumption:

**2. Chapter 2 – Constant Speed Systems – Throttling demand “Mechanically”**

The load demand of a booster system can significantly impact the related power consumption of the package. In any system, the significance of cost savings is based squarely on the actual load the building is demanding. Therefore, if we could somehow predict what the building load will be, we could size a booster system which is very cost effective. But let’s face it; there is simply no way to predict what a building load will be for any given moment in time. The load is constantly changing, requiring the constant speed system to be throttled open or closed to follow the system load profile. It is a forgone conclusion that, regardless of the type of system control, the pump system will have to be throttled to follow the building load. As the load decreases, the pump rises on its curve toward shutoff, requiring that the pressure regulating valve increase its pressure drop to account for the over-pressurization condition. (Fig. #1A)

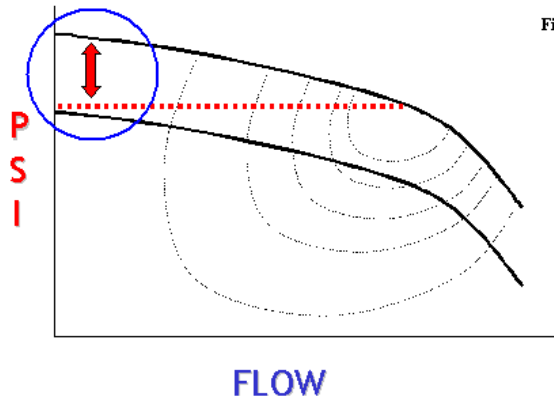


Figure 1A

The pump curve above illustrates the increase in pressure as the pump rises to shutoff. The red arrow indicates this pressure increase which is reduced by the PRV. The lower the pump demand, as in a typical booster, the greater the wasted head.

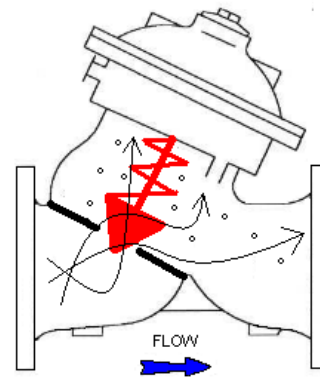
It costs money to run a constant speed pump at high amperage loads even though most booster systems operate at 20% capacity, 80% of the time<sup>1</sup>. (See 80/20 Rule) That is to say that, even though the demand in the building is low, the pump still runs at full capacity and requires a pressure regulating valve to throttle the additional pressure. This translates into an operationally-inefficient booster system which is wasting energy

nearly its entire operational time. Since the system is producing high-pressures at the pump discharge, this pressure is piped to a tank which acts as temporary storage until small system demands allow flow back to the system. Unfortunately, due to the fact that the tank must first be pre-charged with air equal to the system pressure where the tank is located, these tanks generally do not store more than a few gallons of “leak load” storage. The system will shut down, but only for short periods of time until such time as the storage volume has been exhausted. (See 2B, Bladder Tanks) At this point, the process begins again. Let’s look at the key components required for a Constant Speed pressure booster in detail.

- A. **Pressure regulators** - usually go hand-in-hand with constant speed systems; however, they have been used in conjunction with variable speed systems (Hybrid Systems). PRV’s typically consist of a valve body, diaphragm, bonnet and (in most booster system applications) a pilot valve and piping assembly. The piping assembly can also include additional features such as a built-in check feature. This is achieved by adding a check valve to the pilot assembly. The large valves mimic the actuation of the pilots and thus, achieve pressure reduction and/or flow checking. As mentioned previously, the job of the PRV is to adjust its pressure drop to match the “over-pumped” pressure of the constant speed pump at a given location on its demand curve. Generally speaking, the lower the pump flow, the greater the over-pumped pressure, requiring the PRV to close off further to add friction across the valve. The additional friction matches the over-pumped head pressure of the pump and the net result downstream of the valve is a relatively constant pressure.

***The Good News*** – Pressure regulators have typically been cost effective and, when properly sized and applied, can provide relatively trouble-free operation. The best way to assure trouble free operation is to reduce the throttle-load which in essence, pre-loads the PRV. Much like a water tempering valve, the PRV works best when it is operating in its optimally sized flow range. This assures that the valve is at least 50% “pre-loaded” requiring it to absorb less friction head, thereby reducing the amount of friction impact on the valve seat. This marginalizes the energy absorption making the valve more cost effective since the pump is not wasting as much amperage in maintaining system pressure due to throttling by the valve. QuantumFlo systems constructed with variable frequency drive controls will nearly eliminate pulsing and pressure anomalies. Furthermore, these systems are easily filed changeable from the touch screen and do not require additional “tweaking” in the field just to change the system pressure.

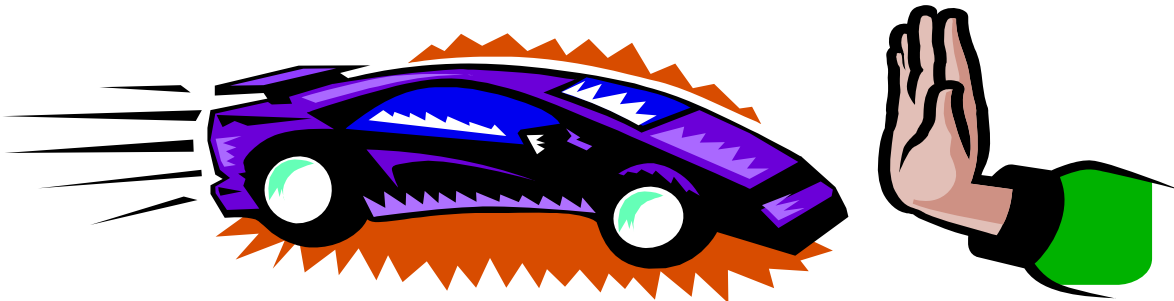
Figure #2



Typical Booster Operation: Valve is at or near fully closed creating high pressure drop across the valve seat.

***The Bad News*** – PRV’s are energy absorbers which rob you of efficient pump energy consumption if not properly sized. Most booster manufacturers use separate PRV’s mounted immediately upstream of each individual pump. Most of these valves are operating at or near the fully closed position, causing water velocity noise and sometimes even dangerous sudden pressure changes that the PRV is unable to respond to quickly enough. Since most boosters are at a small capacity, most of the time (see 80/20 rule), the over-pressurization of the pump shutoff head is wasted. Additional problems can arise as a result of poor application, such as: valve seat wire draw, excessive pulsing, loss of pressure control and diaphragm and pilot failure. The cost of the PRV has steadily risen over the years while the cost of electronic variable frequency control has declined. The cost of variable speed control today is nearly the same as constant speed pumping so it makes more sense to most engineers to use variable speed drives today. We will likely see electronic speed control become even more competitive than PRV’s in the very near future as microprocessor costs continue to drop.

***Analysis of the Constant Speed System*** - One of the reasons that Constant Speed packages with PRV’s have been so widely used is because of their relatively low initial cost. Up until recently, we simply had nothing better or more cost effective for reducing pressure. In the case of the constant speed package, the PRV is nothing more than an “energy absorber”. Unfortunately, energy “absorption” is not the most economical means of pump control. Think of the PRV in terms of the brake pads in your car. The brakes allow you to stop due to the friction imparted on the rotor or cylinder. All this energy, however generates heat, which is essentially, wasted. Would you get into your car today, slam the gas pedal all the way to the floor and regulate your speed with the brake? If you did this, what kind of gas mileage do you think you would get?



Why would you use all this pent-up energy within the engine and then waste that energy due to the friction of your brake pads? This is not very inefficient, but it would get the job done. This is the very premise, however, of the PRV. Your pump is running at full (constant) speed and the PRV is placed immediately downstream to literally “knock-down” over-pressurization as the pump approaches shutoff. PRV manufacturers recommend that when the load falls outside the capacity of a single valve, that multiple valves be installed in series. This allows the use of a small valve to serve the smaller loads until it “hands-off” the

load to a larger valve when the load exceeds the smaller one. This is not required when applying a variable speed controlled system as the capacity is changed by simply varying the speed of the pump. This is the new method and, as a result, it involves new sizing considerations and methods too. (See Chapter 5)

- B. **Bladder Tanks** - Bladder tanks (or Hydro-accumulator tanks as they are sometimes called) are usually recommended with the constant speed, pressure regulated system. They are not required for shut-down as many manufacturers will have you believe. The primary reason for the bladder tank is to increase the length of system shutdown when a shutdown is possible since the only means of energy savings in a constant speed system is to shut down the pump during periods of low or no activity.

Over the years as a sales agent, I supplied many systems in hospitals without bladder tanks due to the constant water demands of a 24 hour facility. If these systems somehow reached their shut-off pressure for longer than five minutes they would shut off. (tank or no tank!) It is highly desirable to add the tank to a constant pressure system because of the very nature of the system “inefficiency”. As stated in the previous chapter, the PRV is often wasting over-

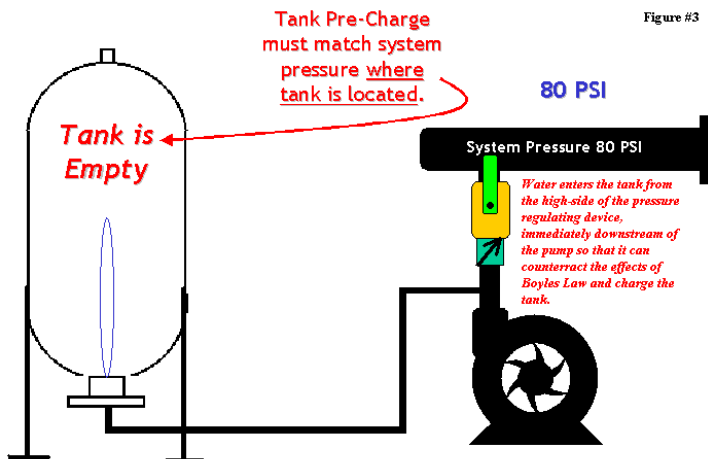


Figure #3

pressurization for most of the operational time of the system, therefore, at very low flows (such as the middle of the night) it is cost effective to turn off the component which is wasting energy, in this case, the pump motor. It can be a difficult task with some engineers to consider removing a bladder-tank which has been used in the past out of “tradition” rather than need. Variable speed systems allow the tank to be deleted due to the fact that the cost savings occur when the pump is running NOT by turning it off, but turning the speed down. This allows constant

energy savings over constant speed rather than short shutdown periods with a bladder-tank. Bladder tanks (particularly ASME rated) can be very costly. We have come to the conclusion that the cost versus benefit is marginal or even non-existent on these system types. One of the reasons for this is the fact that most tanks do not allow for efficient operation due to the requirements of the air pre-charge and the realities of Boyle’s Law<sup>2</sup>.

Boyles Law states that:

*At a constant temperature, the volume and pressure of a certain quantity of a gas are inversely proportional. In the case of a bladder tank, as the air charge is compressed the pressure increases proportionately on both sides of the bladder; water side and air side.*

*A practical application illustrating Boyles Law would be the action of a syringe. When we draw fluids into a syringe, we increase the volume inside the syringe, this correspondingly decreases the pressure on the inside where the pressure on the outside of the syringe is greater and forces fluid into the syringe. If we reverse the action and push the plunger in on the syringe we are decreasing the volume on the inside which will increase the pressure inside making the pressure greater than on the outside and fluids are forced out.<sup>3</sup>*

The tank must be pre-charged to the system pressure at the location of the tank; hence, as water is added to the tank, the tank pre-charge pressure rises as the cubic area of air volume is decreased. This additional pressure allows the water to be forced out during drawdown. As a result of Boyles Law, tank drawdown charts have been created which show that the volume of water which the tank stores prior to pump shutoff pressure is insufficient to provide adequate storage for anything other than system “leak-loads”. Unfortunately, the original premise that bladder tanks store a sufficient amount of drawdown to achieve extended system shutdown, is simply unfounded.

C. **Jockey Pumps** - Obviously, the system is using energy whenever it is running. As a result of this reality, many manufacturers have developed control sequences which attempt to make better use of this inevitable energy usage. In the past, some manufacturers would utilize a jockey pump lead and two larger lag pumps in order to “shift” the small loads to a pump which will operate most of the time. This confirms the fact that many system manufacturers have realized the shortcoming of simple Lead/Lag staging in favor of a more cost effective approach. The unfortunate reality is that since the system is operating at low flow most of the time(80%), this jockey pump is used and worn while the lag pumps have hardly been touched (20%). Additionally, this method requires complicated sensing and sequencing controls which do not respond quickly enough to justify their usage. Current PLC controls and VFD’s can simplify this by only running the amount of KW (or power) for a particular load at very specific staging intervals and rapid response times.

D. **Pressure Switch Sequencing**

Pressure switch sequencing is not very accurate and a change in suction pressure can actually change the flow at which the next pump sequences. Furthermore, pressure has nothing to do with flow and yet many manufacturers still use pressure switches to determine the location of a pump on its flow curve. This can create a dangerous situation where the pump must run beyond its demand point, potentially overloading the motor to sequence the next pump. (See Figure #4)

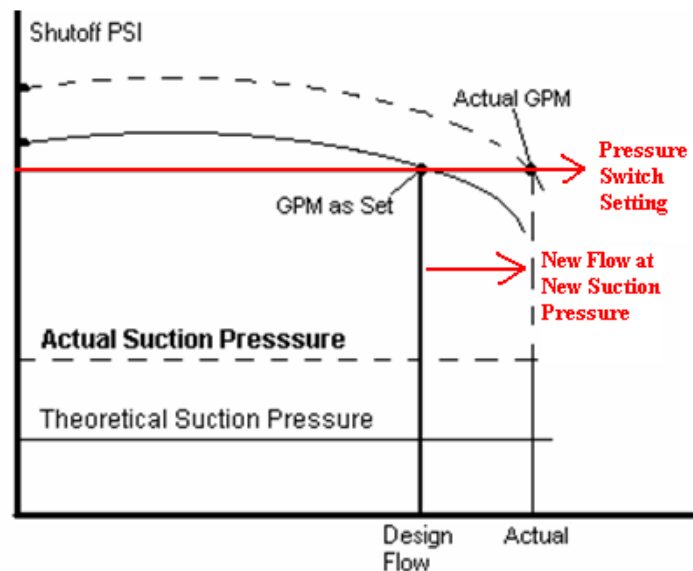
The inaccuracy of pressure-based sequencing is best exemplified in the Triplex, Lead/Lag/Lag scenario. With “pressure based” sequencing, the only sequence available is as follows:

***Pump #1 or Pump #1 & #2 or Pump #1 & #2 & #3***

This is an example of 3-Step sequencing. It is the most common sequencing option with triplex constant speed systems, and the only type of sequencing available with pressure-based sequencing actuation. It is sometimes called “additive sequencing”. If we used additive sequencing and assumed that each pump was sized for one third of the load, your operational capacity would be sequenced as; 33% then 66% then 100%. In this scenario, if the system had a demand of 35%, the system would respond with a 66% combination of pumps #1 & #2. This would not be a very efficient combination of pumps in this case.

Figure #4

One major problem with pressure-based sequencing control is that as the suction pressure changes, the location on the pump curve also changes. If the pressure switch stays at the same value the pump may actually be sequencing at a higher flow if the suction pressure increases.



As time went by, and sequencing technology advanced we discovered that we could intelligently sequence the Brake Horsepower (BHP) load in order to maximize energy efficiency. Adding additional steps of control (multi-step control) allowed the pumps to be sequenced and re-sequenced in order to match the load. This allowed the system to determine which pump or combination of pumps could be run to meet the flow load. This system is typically sized for a 20%-40%-40% load split. In triplex packages, the multi-step sequencing can be configured for up to 5-step sequencing, which is expressed as follows:

***Pump #1 or Pump #2 or Pump #1 & #2 or Pump #2 & #3 or Pump #1 & #2 & #3***

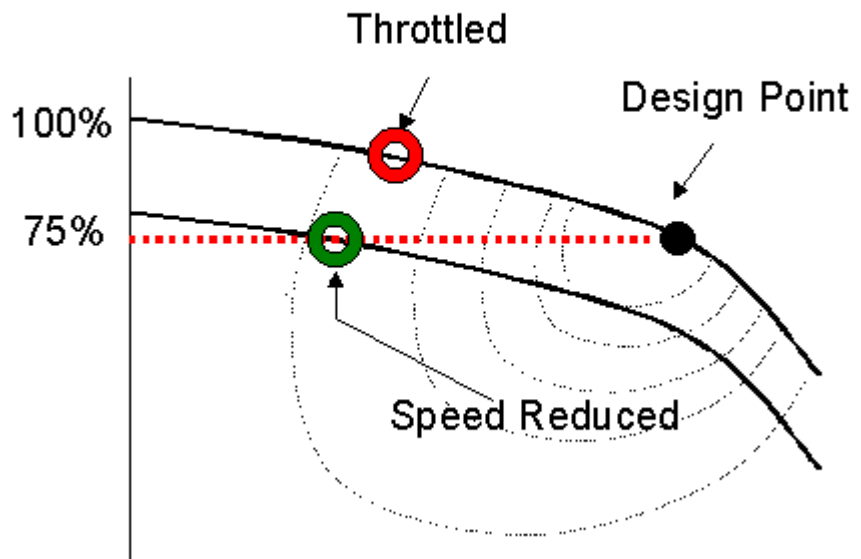
The breakdown percentage in a 5-Step Triplex sequenced system with 20-40-40% split would be 20% then 40% then 60% then 80% and 100%. Multi-step sequencing took the additive

sequencing control a step further; however, since these systems are still constant speed, we cannot take advantage of the power saving properties of the pump affinity laws which achieve cost savings through variable speed control. Furthermore, even though we understand the impact of the 80/20 rule, the additive sequence control required minimum run timers be added to avoid short-cycling of the system. During the OFF-Delay period, these constant speed pumps continue to consume full load amperage in order to satisfy the timers, whereas the variable speed system is constantly adjusting amperage draw. In other occasions, the system is operating most of the time to maintain system piping pressure and satisfy leak-loads. The pump is still drawing 100% of full load amperage during these low flow periods. If that power consumption is factored into the operational time of the pump, you would discover a significant waste in pumping energy merely to maintain the system pressure, charge a tank or satisfy the minimum pump run timers in addition to having a more complex sequencing method.

### 3. Variable Speed Pressure Control – Varying the speed to more closely match the building load profile.

- A. **Pump & Fan Affinity Laws** - As we discussed in the section on Constant Speed Pressure Control, there is no way to predict the building system load. The best way to cost effectively service this load is to match the flow rate as required without unnecessary over-pressurization. Constant speed systems cannot do this since the PRV is wasting energy through its throttling, jockey pumps and “staged” pumps are satisfying their minimum run timers at full amperage and the tank system is attempting to charge in preparation for for a very abbreviated shutdown period. Variable Speed Pressure Control modulates pump

Variable Speed control takes advantage of the quantum savings gained by **Figure #5** reducing the pump BHP by slowing the pump (bottom circle) versus simply throttling the flow with a PRV (top circle)



speed, effectively following the building load and eliminating the need to throttle the pump and waste shutoff-head pressure. This yields a quantum energy reduction as clarified by the pump and fan affinity laws as well as Ohm's Electrical Laws..

## Pump & Fan Affinity Laws

### *Law #1*

Doubling the pump speed gives you double the amount of flow.  
*(i.e. When you double the speed, the flow is doubled)*

### *Law#2*

The pump pressure changes to the square of the speed.  
*(i.e. When you double the speed, the head increases by 4 times)*

### *Law #3*

The pump power consumption changes to the cube of the speed.  
*(i.e. When you double the speed, the power consumption increases by 8 times)*

Many engineers are familiar with the pump affinity laws, particularly Law #1 & #2 listed above. The Law which best describes the potential value of pump speed reduction however, is Law #3. Motor power consumption increases or decreases to the cube of the motor speed. This means that even minor changes in pump pressure (or speed via the VFD) impact the power consumption most dramatically.

I am sure many will agree that most plumbing systems do not have a widely varying pressure load. In fact, it is the system load (or GPM) which most impacts sequential starting and stopping of pumps. There is, however, a system pressure variation which the drive can capitalize on and the three most common areas for change are as follows:

#### 1. **Pressure Variation #1:**

When calculating your plumbing system, you must account for FRICTION LOSS at the FULL system pump flow. As the demand on the system decreases, this friction loss is "recovered" since partial flow does not carry as much friction loss demand. The frequency drives reduce the speed as the system flow decreases.

Figure #6

- Assuming only a 15% reduction in speed, your power reduction is reduced by nearly HALF!

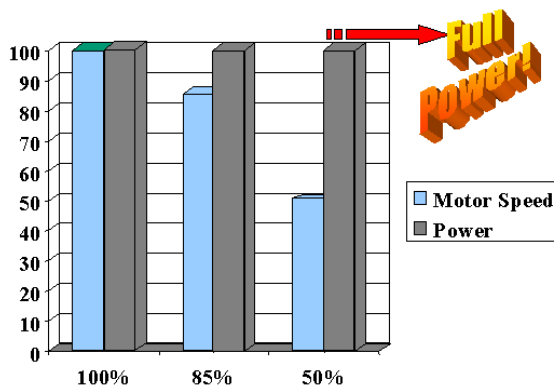
***(.85<sup>3</sup> = .614) = 61.4% of power consumed with only a 15% speed reduction!***

- This, coupled with the elimination of high maintenance PRV's and large costly Hydro-tanks which hold nearly nothing in drawdown yield greater lifetime cost savings.

2. **Pressure Variation #2:** Suction pressure should be based on the minimum worst-case condition. As a result, most suction pressure values given are not typical of the actual jobsite pressure. Furthermore, as flow decreases, the suction pressure will increase slightly. In this case the frequency drive will reduce pump speed as suction pressure increases. Always call the municipality to get an accurate suction pressure at the site.
3. **Pressure Variation #3:** As the pump rises toward shutoff and away from system design point, the pump pressure generally increases. This recovered shutoff pressure is offset by a reduction in motor speed as the pump rises to shutoff.
4. **Pressure Variation #4:** Pressure regulating valves generally have higher pressure drop requirements than standard fittings. This loss must be added to your design criteria when calculating system boost pressure. A typical variable speed system only requires system check valves downstream of the pump.

In many cases, all of the pressure variations mentioned above will add additional head pressure to your system calculations. Furthermore, many of these values vary depending on the pump system load. There are basically two schools of thought for how to deal with these pressure variations:

## Power Consumption - Constant Speed



- No appreciable power reduction.
- High shaft and bearing radial loads.
- Unnecessary high pressures at pump discharge.
- Pressure Regulator increases resistance to match load

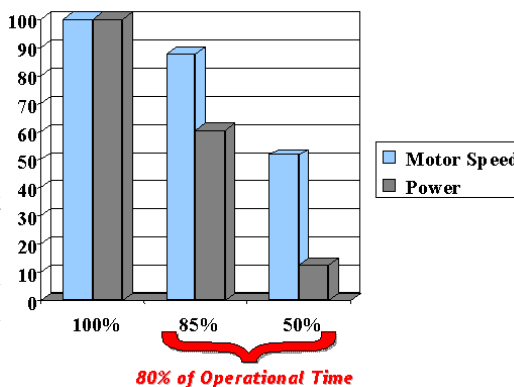
Figure #8 are basically two schools of thought for how to deal with these pressure variations:

- Chop off the over pressure condition using “Mechanical Means”.
- Reduce the speed of the driver to match both head and flow conditions by “Electronic Means”.

We’ve already discussed the ways in which pressure

regulating valves use “Mechanical means” so let’s discuss how variable frequency drives use “Active means” to contain this system pressure. Remember our Four Pressure Variations discussed above? Let’s consider the effect that these have on a typical 3500 RPM pressure booster system assuming only a 15% speed reduction and our Affinity Law #3. (Figure #6)

## Power Consumption - Variable Speed



- Power is reduced as the “cube” of the speed!
- Lower shaft and bearing radial loads.
- Constant pressure is always maintained.
- No wasted energy loss through throttling valves.

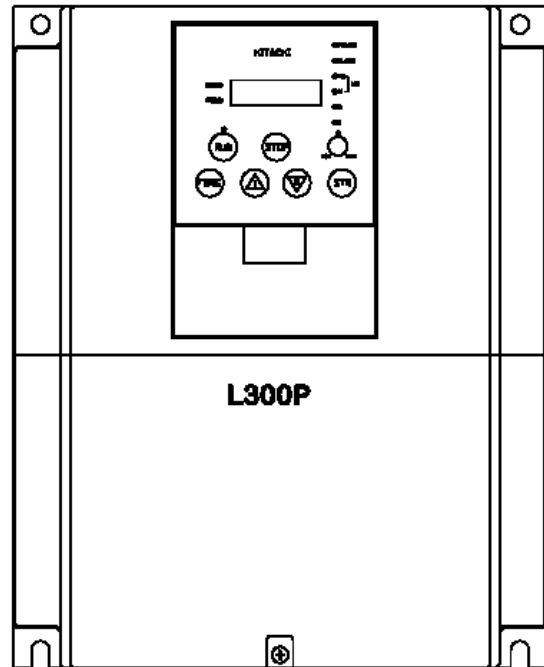
Figure #9

***15% Reduction of the normal 3600 RPM speed expressed as its reciprocal is .85  
Power savings is equal to the cube of the speed reduction, therefore we express this as:  
(.85<sup>3</sup> = .614) = 61.4% Power required at 3060 RPM  
This translates to a 38.6% power reduction with only a 15% speed reduction!***

That's a very significant power savings for a very modest speed change. More importantly, the value is exponential depending on the amount of speed which can be reduced! Consider the comparison in Fig. #8 & 9 showing constant speed versus variable speed. Apply the 80/20 Rule mentioned in Chapter #1 and you can see the how variable speed saves money when it's running, not when it's off.

### **B. Why VFD's – Why Now? Why NOT!**

When my colleagues and I wrote the first Booster Designers Handbook we dedicated very little time to the effectiveness of variable speed pump control. One of the questions asked most often is; “Why didn't you promote the use of variable speed pressure control in the past?” This is a very simple, straightforward question and it deserves a very simple straightforward answer. The answer is COST! At the time that book was written in 1997, the cost of variable speed frequency drives was relatively expensive! The potential energy savings was still valuable, however, promotion of the concept at the time would have been difficult to sell in the bid & spec community. The best part of this story is that, much like the cost of personal computers, the cost of these technologies has dropped significantly and now rivals the cost of installing pressure regulating valves, pump starters and bladder tanks. Best of all; variable speed controls are far more reliable than “hydraulically-based” pressure control through the use of PRV's and marginal shutdown storage with bladder tanks.



*The Advantages of Variable Speed Pressure control are as follows:*

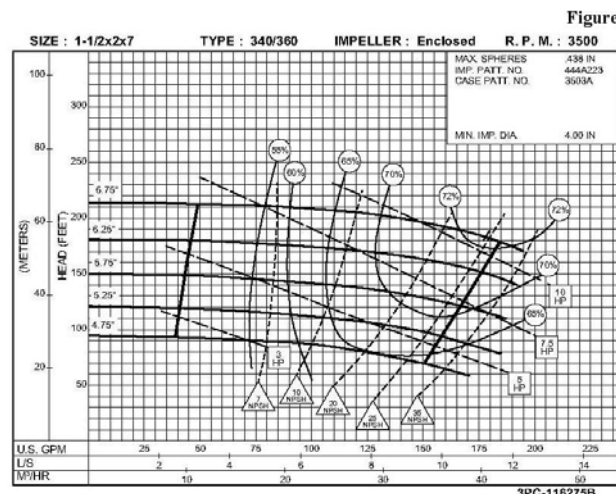
- a. Electronic reduction of pressure reduces valve noise due to velocity across the nearly closed PRV Valve Seat.
- b. Bearing radial loads are significantly reduced since pump reduces in speed as it approaches shut-off. These hydraulic loads contribute significantly to pre-mature bearing failure due to heat and stress.
- c. A variable speed system can more closely follow the building load profile, applying

- only as much energy as required for any given condition.
- High maintenance bladder tanks can be eliminated since the variable speed system maintains a constant pressure regardless of suction, system and any other pressure changes in the system.
  - PID firmware control in each drive allows a more accurate means of pressure control which is not subject to tampering in the field.
  - Elimination of pump PRV's which cannot control rapid system changes and sometimes require opening or closing speed controls or both.
  - A reduction in power equal to the cube of the speed reduction rendering energy saving devices (like bladder tanks and some jockey pumps) completely unnecessary.

### C. Pump Curves and their Affect on Power Reduction

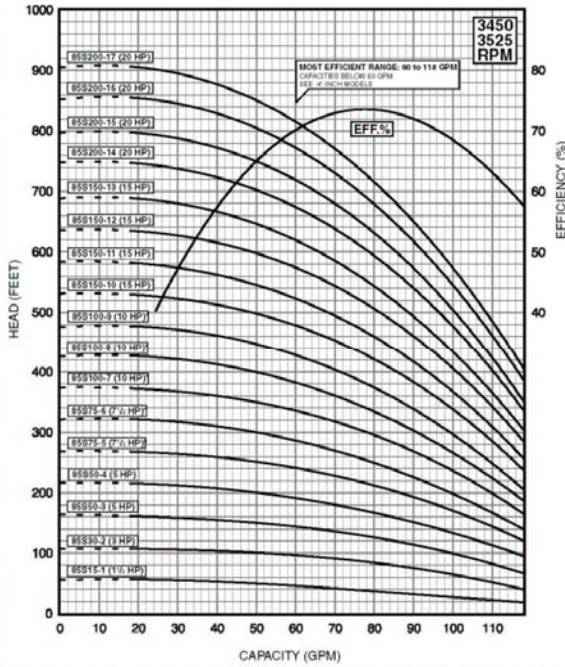
Earlier we discussed the effect of the speed change on power consumption. The shape of the pump curve can further enhance this condition. The variable speed drive does not care what it is connected to. As a result, since the affinity laws state that the power is reduced as the cube of the speed, the more the speed reduction, the greater the power savings. If a flat curve is used on a variable speed booster system, the change may be small, however the cost savings can still be significant. This savings can be multiplied exponentially on a pump curve with a very high rise to shutoff. As the pump flow is reduced the pressure rises dramatically, however, the variable speed drive is maintaining a given pressure. In the case of a steep curve, the pump can reduce its speed more while still maintaining system pressure in the system. The net result is a pump running much slower to maintain the same system pressure. High shutoff curves have been something which engineers have tried to avoid in the past due to the danger of the PRV failing, however, this is no longer an issue with variable speed control.

The best energy-efficient scenario would be to only use the amount of brake horsepower required for the building at any given time. We accomplish this through the use of variable frequency drive controllers for each pump in the system. Instead of ON/OFF operation, high radial loads and high pressure-drop valves which “chop-off” pump overpressure, this system utilizes a modulating speed operation making use of the cubic relationship between pump speed and output pressure as per the pump affinity laws. (The power changes to the cube of the reduction in speed) This means that even a small reduction such as 15%, (a minimal differential in a building such as this) relates to nearly 40% energy reduction( $.85^3 = .614$ ) by the pump! Since the high pressures no



A typical centrifugal pump curve showing the flat rise to shutoff. The “flat” nature of these curves impacts the variable speed drives bandwidth of speed control. This is a good example of how pump shapes affect power savings.

Figure #11

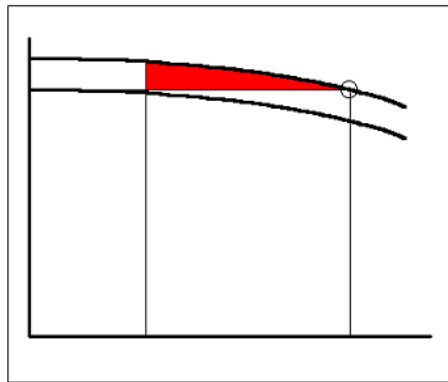


The curve shown here is an example of a multi-stage pump with high shutoff head. In the past, this curve would have been considered dangerous due to the high potential pressures, however, with variable speed control, this is more cost effective due to the bandwidth of pressure control as the pump reduces in flow. Less flow = Lower speed.

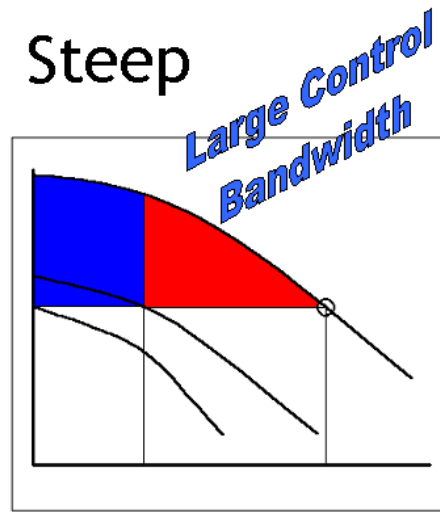
longer exist at the pump discharge, there is no additional pressure required to charge a pressure tank, nor is there any reason to use a tank anymore since the pump is saving energy most of its operational time versus constant speed. This system is actually more cost effective since the pump speed is changing 80% of the time. This reduces the power required versus the constant speed system which is ON at full speed and high amperage draw the entire time of operation. Additionally, if the system demand has been reached, the system will still shut down completely utilizing no energy.

Figure #12

## Flat vs. Steep



Typical Centrifugal



Typical Multi-Stage

Comparison of flat curves versus steep curves. As you can see, the multi-stage curve to the right generates a higher pressure as flow is reduced toward shutoff. The VFD will reduce the pump speed progressively as the pump moved along its curve to the left taking advantage of significant cost savings. In a constant speed application, the power consumed would be constant. In most cases, the variable speed system is saving energy at any given time that it's running.

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*In general, fans and blowers are moving air, but centrifugal pumps can be moving many kinds of liquids, including water, petroleum products and coolants.*

*There are two mysterious characteristics that these loads have. The first is the way they act when the speed is changed. The rules that cover these characteristics are called the "affinity laws". Where the load is not changing (simplest case), take a pump circulating chilled or hot water through a closed system. The torque required to drive the pump goes up with the square of the speed. So changing the speed from 1160 to 1760 rpm, causes the required torque to go up by 2.3 times the original value. Also, since power is based on speed times torque, this goes up with the cube of the speed almost 3.5 times the power required at the original speed.*

*The dramatic increases in the power required to drive these loads when speed increases is a little difficult to understand but it is very important. It is also important because small decreases can result in great energy savings. For example, decreasing the speed of a variable torque load by only 20% will result in a driving energy reduction of nearly 50%. This obviously has great importance when conservation is considered. It also accounts for the tremendous market that exists for variable*

*frequency drives operating variable air volume (VAV) systems used in heating, ventilating, air conditioning and variable speed pumping.*

*The second puzzling thing that occurs with variable torque loads is that the motor load actually decreases as the output or input to the blower or pump is blocked off or restricted. It defies good judgement to think that adding a restriction to the output of a blower would decrease the motor load. If you don't believe it, here's a simple test. Take a vacuum cleaner and listen to it carefully while you alternately open and close the suction. At first, you might think that the "heavier" noise is the motor straining when the suction is greatest, but if you listen more carefully you will notice that the pitch of the motor goes up when the suction is closed. What this means is that the load is being reduced on the motor and it speeds up. If you still don't believe, you can do the same test, but with an ammeter on the motor. What you will find is that the amps drop as the suction level is increased. The same is true of centrifugal pumps: closing down or restricting the output causes the pump to draw less mechanical power. Another way of looking at this is when the output of a centrifugal pump or a squirrel cage blower is closed off, the air or fluid inside the housing becomes a "liquid flywheel". It just spins around the vanes of the pump or blower. Since there is no new fluid coming in to be accelerated, the only energy needed is what it takes to make up for the friction losses within the housing of the pump or blower. It doesn't seem to make sense, but that's the way it is!*

#### D. The Shutoff Anomaly

There is another reason that variable speed pumps are cost effective and it has become the question of the century for those new to variable speed booster applications. Those who do not understand the dynamics of variable speed control will try to justify their position with the following incorrect comment:

*“If the plumbing system is static and does not increase or decrease in size, the pump needs to run at full speed to maintain the plumbing pressure load even if there is no water flowing.”*

This comment is totally false, since we have known for quite some time that the VFD will actually ramp below its minimum speed to maintain system pressure at dead-head condition. The fact is; the reduction in torque is proportional to the same reduction in power as the speed is decreased until the pump eventually becomes nothing more than a “liquid flywheel”. Since many plumbing packages operate at or near this condition a significant portion of the time, it stands to reason that the potential energy savings with variable speed is far greater than any potential savings which can be achieved during shutdown of a constant speed pump with a bladder tank. You can say that a variable speed package is saving energy nearly every moment that it is running since the flow demand is typically very low most of the time. Consider the excerpt on the previous page, from an article written by Mr. Edward Cowern with Baldor, USA for Industrial Technologies magazine.

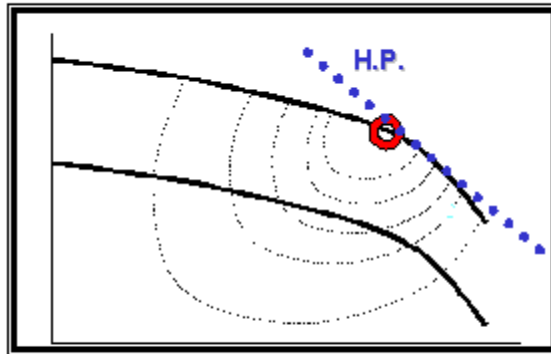
### 4. Chapter 4—Pump Sequencing for Energy Efficiency

#### A. Flow Sequencing

Flow sequencing is usually accomplished with the use of an insertion-type flow meter which measures the velocity of water through the pipe and determines the optimal selection of pumping power to accommodate the condition. This is the most accurate means of flow sequencing and we sometimes call this “direct measurement” since the flow sensor is in direct contact with the medium. This is certainly a very accurate means of determining when to sequence on an additional pump when one is running, however, there are several things to consider when using this type of flow sequencing.

- The device is in direct contact with the pumped medium and is therefore, subject to corrosion, clogging or some other “mechanical” failure.
- The device cannot be seen from the outside since it is inside the piping and there is no way to tell if it is in good working order.
- These devices require access to be serviced such as a “wet-tap” which allows frequent servicing and cleaning.
- Flow meters are requires to be installed with a minimum of (10) pipe diameters upstream and (5) pipe diameters downstream to work properly and accurately.

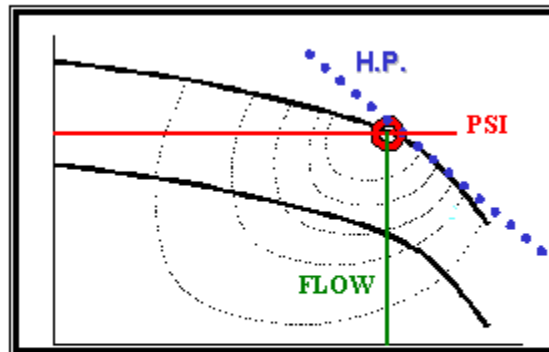
Figure #13



**Pump sequencing is so critical to maintaining energy efficiency. The ideal situation would be to sequence the next pump in series only after the previous pump has met its maximum available Brake Horsepower. Better sequencing yields less waste and more control over pump energy consumption.**

It is possible to achieve some cost effectiveness with flow-based sequencing since the flow is more accurate, however, you must still consider the fact that even a constant speed system with flow sequencing requires a means of achieving pump shutdown such as a bladder tank, etc. Additionally, these meters require that a packaged system manufacturer provide a long spool piece with the device pre-mounted in order to achieve the proper pipe diameters to work correctly. This does not apply well to factory pre-packaged systems.

Figure 13A

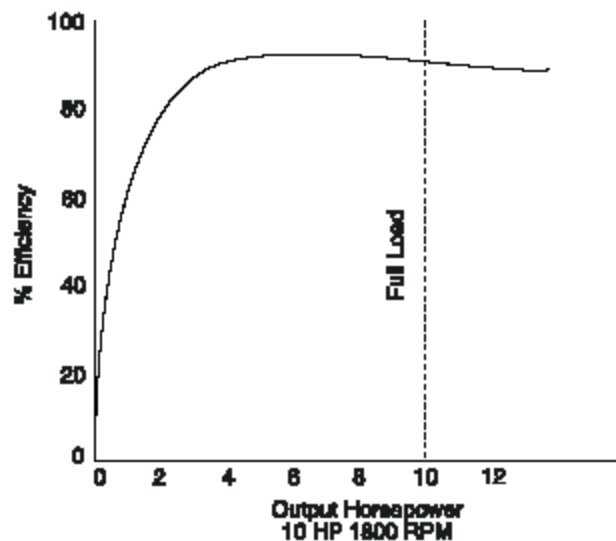


**Pressure-based sequencing relies on the intersection of the pump and suction pressure condition and the pressure switch setting which must intersect the pump curve at the desired flow. Since suction pressure affects the pump shutoff head, this value is not "repeatable" and the value can change from moment to moment.**

## B. Pressure Sequencing

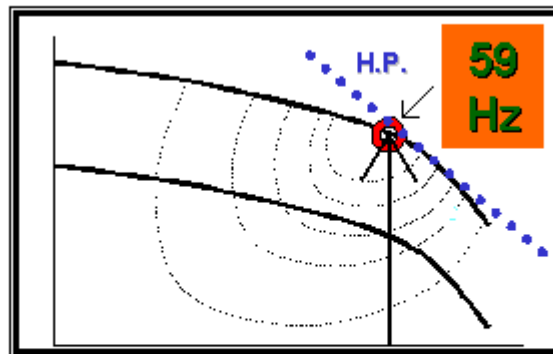
If flow sequencing is considered “direct measurement” then pressure-switch sequencing is as close as you can get to “indirect sequencing”. (Fig. 13A) As mentioned in Chapter 2, pressure switch sequencing relies on the predictable nature of a pump curve to intersect at a point which coincides with the preferred flow that you want to occur when triggering the second pump. Unfortunately, there is no way to accurately predict (at any given time) where the pressure set point will fall on the pump curve. If the suction pressure increases, the trigger value is moved farther to the right on the curve and conversely as the suction pressure drops. The danger in using this type of sequencing is that it is actually possible to overload the motor if the suction pressure increases too much. (See Chapter 2) This is obviously not good for your pump and will lead to premature motor failure. It is this “repeatability” which is not available with pressure sequencing as it can be when using a new non-fouled flow meter or sensor. Pressure switch sequencing is the most common practice of system manufacturers today because it is the most cost effective. Unfortunately, it is the client who loses by receiving a system which is not energy efficient.

Figure #14



**A motor that is overloaded will have short life.  
 In the opposite situation, a motor that is grossly oversized for the job it is asked to do, is inefficient.  
 However, on continuously loaded motors it is desirable to load motors...more ideally in the range of 75 to 80%.**

Figure #15



Frequency-based sequencing of pumps is more accurate due to the fact that the sequence point is a "repeatable" value. In the case shown above, when the motor reaches 59 Hertz the next pump in sequence is actuated. This not only occurs at the same point every time but also incorporates over-current protection.

### C. Frequency (or Speed) Sequencing

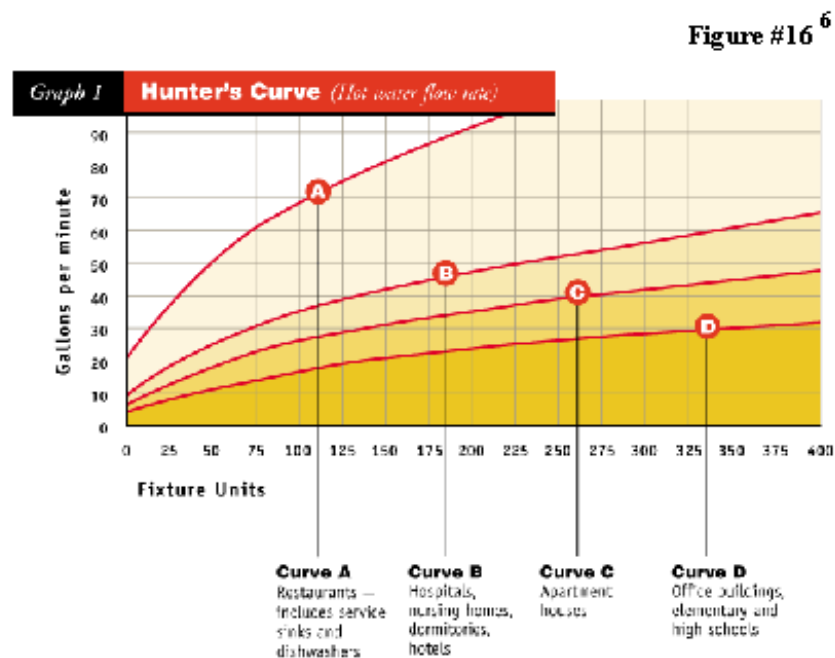
Your final choice for pump sequencing is determined by the speed of the motor. Speed-based sequencing is typically done with a Variable Speed Drive which measures pump speed by Hertz. Most motors in the U.S. are based on 60 Hz operation. This means that a 3500 RPM motor has reached full speed when it has reached 60 Hz. It is simple to avoid an over-current situation when sequencing a motor on hertz. If set at 60 Hz or below the motor will never overload before another pump is actuated to assist. A setting value of 59 Hz will bring on pump # 2 at the same time for every cycle. This is a "repeatable" value which is never influenced by pump suction pressure or any other outside influences. We can now use this reference signal to not only sequence the next pump, but also avoid allowing the motor to run under an overload or excessive amperage condition. This condition would damage the motor considerably. The best part of speed-based sequencing is that we can use all the available energy that the motor has to give. In frequent cases, a constant speed, pressure-based system would require a larger motor to avoid a potential overload condition (as shown in Figure #13A). However, many engineers are unaware of the fact that all squirrel cage induction motors are most efficient when they are 80% loaded. If we substitute a larger motor due to the fact that we do not have a repeatable sequence value, we are actually making the system even less energy efficient.<sup>5</sup>

Finally, the use of frequency or speed-based sequencing is "directly" proportionate to motor "work" and since it is done electronically, this means of sequencing is not subject to one of the most insidious of field problems; field tampering.

## 5. Chapter 5—Calculating System Redundancy

### A. The QuantumFlo Formula—The 70% Rule.

All manufacturers use (essentially) the same basic formula for calculating system flow capacity. The 1940, report entitled “Methods of Estimating Loads in Plumbing Systems,” by Dr. Roy B. Hunter (The Hunters Curve) has been used for nearly 70 years. As plumbing systems continue to evolve and become more efficient, these values were further enhanced by creating different variations of the Hunters Curve to apply to varying types of buildings. Different buildings, such as Hospitals and Institutions consume water differently than Offices and Residential Facilities, therefore the Hunters Curve is revised either up or down to account for the deviation in diversity loads. (Figure #16)




Today there are continual changes to the interpretation and usage of the Hunter's Curve. Low flow toilets are a good example of this. The Hunter's Curve was created at a time when water conservation was not even a consideration. As a result, a system sized with the single-line Hunter's Curve (not optimized by building-type as in Fig. 16) is typically oversized. Although this may provide comfort to the engineer, it is not always a good thing to over-size the pumps as mentioned in the previous topic, especially when you are using variable speed pumps!

The second problem that contributes to over-sizing loads is system redundancy. Unfortunately, many consultants are mis-informed as to the purpose and procedure for providing an additional capacity to compensate for a pump which may be out of service or under repair. Typically the capacity is arbitrary and based upon the flow of other pump capacities in the system. This adds unnecessary cost to the system since a typical duplex

system may become a triplex only for the purpose of providing redundancy. We recommend adding the redundancy capacity first, before determining the number of system pumps you intend to use. Furthermore, we have discovered a “rule of thumb” for calculating system redundancy which meets the need to have extra capacity without adversely affecting system capacity. We believe that the best way to determine both the system capacity and the number of pumps in the system is determined by the initial flow capacity and the total BHP which may be required for the entire system combined horsepower. Let’s look at the following sizing example:

Figure 17



PACKAGED BOOSTER SIZING SHEET	
Prepared By	Design-Build
Project Name	7 Story Student Housing

Sizing Criteria Provided			
Building Height	TDH As Entered	Residual PSI	Suction PSI
75 FT		40	0
Drawing From Above Grade Break Tank		Minimum Tank Level	12 FT

Pressure Boost Calculations				
Building Height	75	Multiplied By	0.433	32.48
Friction Losses	30%	Multiplied By	32.48	9.90
Losses in campus site piping or additional system-side friction losses				0.00
Residual PSI required at remote fixture (QuantumFlo Recommends 40 PSI)				40.00
Required System Pressure = A + B + C + D			<b>Sub Total</b>	<b>78.97</b>

Available Suction Pressure				
Suction Supply				5
Supply Piping Loss	Backflow Preventer	Water Meter	Package Losses	
0	0	0	5	
Minus Suction Losses of				5
Total Available Suction Pressure = A - B				0.1960001

Net Boost Required	
System Pressure Set Point (PSI)	78.97
Available Suction Pressure (PSI)	0.20
Differential Boost Pressure Required (PSI)	78.77

Figure #17 represents the first page of our EINSTEIN sizing and selection program. This page determines the total boost pressure required.

Figure #17A is page # 2 of the EINSTEIN sizing program. The recommended raw capacity and redundant capacity is shown at the bottom of the page. How is this calculated?

What happens if the system is selected as a Duplex instead of Triplex? Actually, the total required GPM would increase with fewer pumps available as backup. Let’s look at the formula we used in 17A & B with the same 70% redundancy using a 2 pump system in lieu of a 3 pump package. Remember that the redundancy is based on any one pump being isolated for service. Since the Duplex system only has a single pump spare, this spare pump must be capable of 70% system capacity with the other pump isolated for service.

Figure 17A

**Flow Calculations**

Quantity	Description	Value	Quantity	Description	Value
0	Bath Group flush tank	0	0	Laundry Tub or Svc. Sink	0
0	Bath Group flush valve	0	0	Urinal - 1" flush valve	0
324	Private Bath or Shower	648	0	Urinal - 3/4" flush valve	0
0	Public Shower/Roman Tub	0	0	Wash Machine - Res."	0
0	Private Lavatory"	0	37	Wash Machine - Public"	148
610	Public Lavatory"	1220	0	Drinking Fountain	0
0	W.C. flush tank - Private"	0	0	Hose Bibb	0
0	W.C. flush tank - Public"	0	0	Bidet	0
0	W.C. flush valve - Private"	0	0	1/2" Connection	0
324	W.C. flush valve - Public"	3240	0	3/4" Connection	0
14	Kitchen Sink - Private"	28	0	1" Connection	0
0	Kitchen Sink - Public"	0	0	1 1/2" Pool Connection	0
0	D.W. or icemaker conn	0			
<b>Grand Total Fixture Units</b>					<b>5264</b>

**Selected System**

You have selected a Triplex System	RAW SYSTEM CAPACITY BEFORE REDUNDANCY	370	GPM
	RECOMMENDED REDUNDANCY FLOW ADDER	18	GPM
	TOTAL SYSTEM FLOW AFTER REDUNDANCY	388	GPM
	BOOST REQUIRED IN TOTAL DYNAMIC HEAD	181.97	TDH

## B. Simplex, Duplex or Triplex?

The “Raw” system flow capacity is the value derived from total fixture units plotted against an optimized Hunter’s Curve. In this case, the “Apartment/Office/Condo” curve was used. The raw GPM is 250 and the program adds an additional 13 GPM to provide a 70% redundancy. This 70% redundancy was calculated as follows:  
 70% of the total system flow of 250 = 175 GPM; We have selected a Triplex system. If any (1) of the (3) pumps is isolated for service, the system must provide 175 GPM (70%) with the remaining (2) pumps. 175 divided by the (2) remaining pumps is 87.5 GPM x 3 Pumps (Triplex) = a total capacity required of 262.50 GPM which is then rounded to the next whole number (263 GPM). 263 GPM Redundancy minus 250 GPM raw = 13 Added GPM.

### DUPLEX REDUNDANCY CALCULATION

*70% of the total system flow of 250 = 175 GPM; In a Duplex system, if any (1) of the (2) pumps is isolated for service, the system must provide 175 GPM (70%) with the remaining (1) pump. 175 divided by the (1) remaining pump is 175 GPM x 2 Pumps (Duplex) = a total capacity required of 350 GPM. 350 GPM Redundancy minus 250 GPM raw = 100 Added GPM A Duplex system would actually have to be sized for 350 GPM or (2) pumps at 175 GPM.*

Now that we have established the system flow capacity and understand the concept of system redundancy, how many pumps should be used? What is the determinant factor? These are very common questions of which consultants will discover a multitude of responses. I have always felt that there are two key considerations when selecting a number of system pumps:

Let's look at Key Sizing Consideration #1. Why is it best to select to the right of BEP?

In Chapter 3 we discussed the conditions and reasons that VFD control was more cost effective than constant speed control. Most centrifugal pump curves drop in pressure as the flow is increased, therefore, the farther the pump moves to the right of BEP the greater the potential rise to shutoff for a given pump trim. By selecting a pump that has a significant rise to shutoff, you take advantage of greater speed reductions as the pump begins moving to the left. (See Section 3C)

**NOTE: Bear in mind the fact that the pump capacity (unlike HVAC practices) is the point at which the system activates the next pump in its sequence, therefore, the pump flow and pressure selection is the point where the pump will be at its maximum flow just before another pump starts to assist.**

### KEY SIZING CONSIDERATIONS

1. In Plumbing packages (particularly VFD-based Systems) it is recommended to select a pump capacity to the right of the Best Efficiency Point (BEP).
2. Select a pump or combination of pumps that use less HP for the redundant total flow capacity.

What about Key Sizing Consideration #2? This is based on the value flow value arrived at with EINSTEIN when redundancy is added. Let's use the example shown in Figure #17A.

As shown in the example, we chose to use a Triplex system. If we used the redundancy formula for a Duplex system we would have to provide (2) pumps at 175 GPM @ 160' TDH compared to (3) pumps of 87.5 GPM @ 160' TDH for a Triplex. The pump selections are as follows:

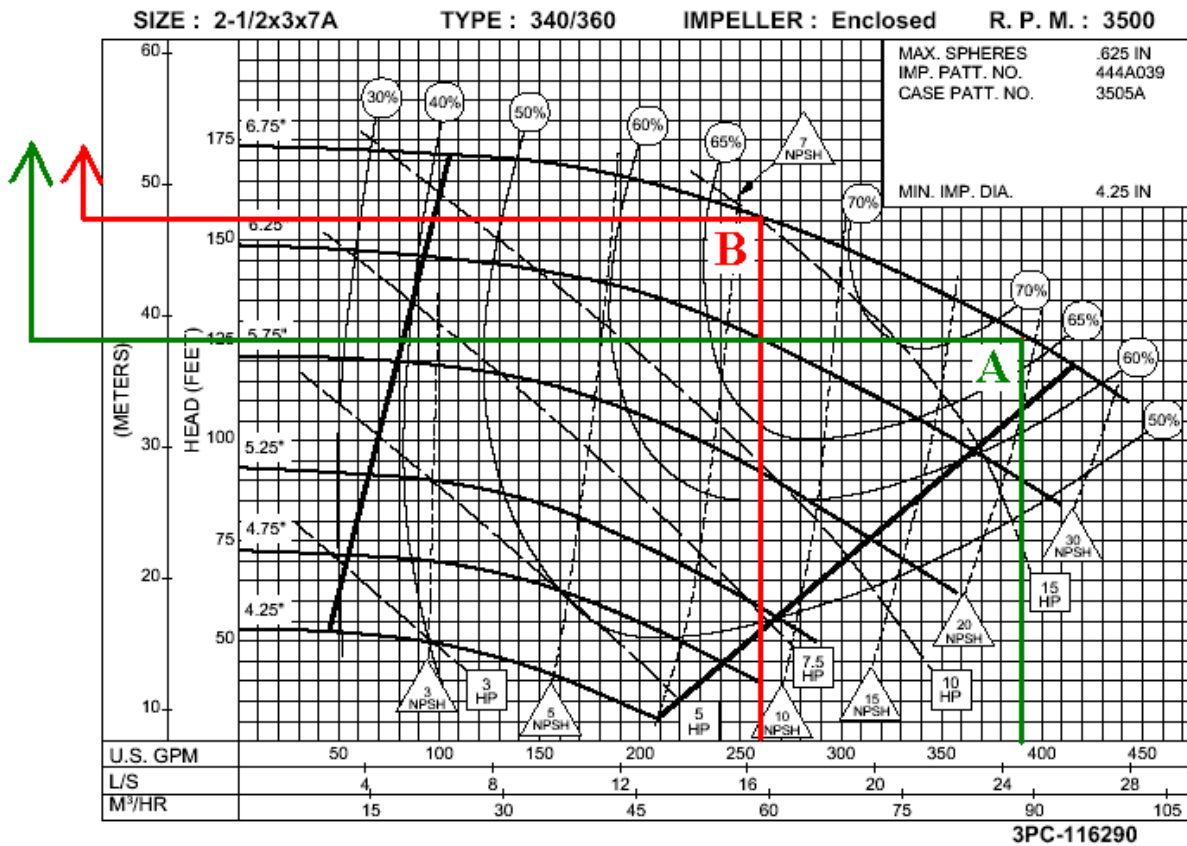
**DUPLEX (Figure #19)**

(2) Pumps: 175 GPM @ 160' TDH = 15 H.P. Ea x 2 Pumps = 30 Total H.P.

**TRIPLEX (Figure #19A)**

(3) Pumps: 88 GPM @ 160' TDH = 7.5 H.P. EA x 3 Pumps = 22.5 Total H.P.

Figure #18



Look at Figure #18 above. Notice the difference between Point A, which is right of the BEP and Point B, which is to the left of BEP. Point A has a greater rise to shutoff from the system sequence point. In all cases a plumbing booster system is operating somewhere between the point of sequence and pump shutoff condition. If the pump is selected at Point B, when the system begins its rise to shutoff, there is much less potential for pressure increase from the sequence point. The potential energy savings is equal to the speed reduction cubed.

In this case, it is more cost effective to use a Triplex. Furthermore, this system will maintain a smaller H.P. motor operation at low flow conditions requiring less electrical energy. It is important to remember, however, that the flows will differ as you consider Duplex versus Triplex since the Triplex system has a greater number of pumps to satisfy the redundancy load. Therefore, a Triplex system (in our formula) is actually smaller in total flow than a Duplex system. These are the only legitimate considerations to factor into your sizing criteria when evaluating the number of pumps required.



## 5. Chapter 5—The Future of Variable Speed

The future of variable speed technology is very bright. As we move toward an environment which encourages sustainable design and the LEED Points Program, variable speed control of motors is a central focal point of the discussion. As a member of the US Green Building Council, QuantumFlo is focused on maintaining our relevance as a leader in these new Green initiatives. We commit ourselves to the betterment of our planet through sustainable technology and design and the desire to continually “push the envelope” of future designs through technological development of our products and our involvement with our engineering network!



EINSTEIN is the QuantumFlo sizing and selection program available online at [www.quantumflo.com](http://www.quantumflo.com)

Notes:

## Sources:

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