Frequency pressure regulation in water supply systems
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ABSTRACT

In water supply systems, pressure management in most cases is proven to be the most cost-effective activity related to water loss control. As an advanced method of pressure control, it is possible to use variable frequency drives for centrifugal pump control. Pressure regulation can be performed with constant pressure or with proportional pressure control. The application of proportional pressure control is particularly applicable in water supply systems as the operating pump performance is constantly adapting the pressure to the actual demand. Along with lower leakage losses, it also results in lower energy consumption and the elimination of non-stationary phenomena, thereby extending the pump lifetime. Therefore, the paper presents a theoretical discussion of the proportional pressure control. Possible savings are shown on the numerical example of water supply system of the city of Velika Gorica.

Key words | proportional pressure control, water loss, water system

INTRODUCTION

The application of frequency regulation as a pump control method stands out in water supply pumping systems characterized with high variability in demand through a relatively short time. Implementing this method to a system with negligible demand variation results in small savings. By applying frequency regulation control, pump head and flow can be directly affected. Additionally, high energy savings and water loss reduction can be achieved by changing the impeller speed as a result of changing the frequency of electricity.

The paper first analyzes the theoretical aspects of pump operation and control along with the associated energy and water savings. Then the possible savings are evaluated using a numerical model of Velika Gorica city’s water supply system. Two scenarios of frequency control are considered. The first considers the possibility of introducing frequency control to meet the existing conditions together with the water tank in the center of the city. The second scenario examines the effects of frequency control in the system with the exclusion of the water tank.

Energy savings are presented by analyzing the characteristics of the water supply system and the operating conditions of the pump. The influence of changing impeller speed on water loss reduction is presented through pump affinity laws and FAVAD (Fixed and Variable Area Discharges) method.

PUMP AND SYSTEM CURVES

The pump characteristics are normally described graphically by the manufacturer as pump performance curves.

Figure 1 shows a set of typical performance curves for a centrifugal pump with a constant motor speed, \( N \).

The basic independent variable is taken to be flow, \( Q \). The dependent variables, or ‘output’, are taken to be head (pressure), \( H \), power consumption, \( P \), efficiency, \( \eta \), and net positive suction head required, \( NPSHR \). The \( QH \)-curve shows the head which the pump is able to perform at a given flow. The head falls gradually with the increasing flow.

As shown in Figure 1, the power required to drive the pump typically rises monotonically with the flow rate.

The efficiency is always zero at no flow and at \( Q_{max} \), and it reaches a maximum, perhaps 80–90 \( \% \), at about...
0.6 \( Q_{\text{max}} \). This is the design flow rate, \( Q^* \), or best efficiency point (BEP) at \( \eta = \eta_{\text{max}} \). It is desirable that the efficiency curve is flat near \( \eta_{\text{max}} \) so that a wide range of efficient operation is achieved.

The NPSHR-value of a pump is the minimum absolute pressure that has to be present at the suction side of the pump to avoid cavitation. The NPSHR-value depends on the flow: when the flow increases, the NPSHR-value increases as well.

A proper discussion of pumping considers not just the pump, but the entire pumping system: pipes, valves, fittings and equipment in the system.

Typical characteristics of a pump system are static head and friction head. Static head is the height the water must be lifted from the source to the destination. Friction head is power loss caused by the flow of the water through the pumping system.

Figure 2 shows a typical open system with positive static head. A pump has to transport water from a source to a required destination, e.g. filling a high level reservoir. In such systems the pump has to both deal with the static head and overcome friction losses in the pipes and components. In the case of fittings, friction is stated as an equivalent length of pipe of the same size.

Adding the two heads together creates the system curve, Figure 3, which describes what kind of flow will occur given a specific head. System curve also defines the system resistance at various flow rates.

When a pump is installed in a system, the effect can be illustrated graphically by superimposing pump and system curves. The point where the pump curve and the system curve cross determines the operating point or duty point of the system, Figure 3. The system will have only one operating point, so if variable flow is required, something needs to be added.

The power consumption, \( P \) [kW], is defined as:

\[
P = \frac{9.81QH}{\eta}
\]

where \( Q \) [m\(^3\)/s] is flow rate, \( H \) [m] is head and \( \eta \) [1] is efficiency.

![Figure 1](https://iwaponline.com/ws/article-pdf/13/4/896/414985/896.pdf) | Typical centrifugal pump performance curves at constant impeller-rotation speed. The units are arbitrary (White 2003).

![Figure 2](https://iwaponline.com/ws/article-pdf/13/4/896/414985/896.pdf) | Open system with positive static head (Pumps and Pumping System 2006).

![Figure 3](https://iwaponline.com/ws/article-pdf/13/4/896/414985/896.pdf) | System characteristic curve together with the pump performance curve for the open system in Figure 2 (Pumps and Pumping System 2006). The units are arbitrary.
Graphically, power is represented by the area under the rectangle defined by the operating point on a pump performance chart, Figure 3.

When selecting a pump for a given application, it is important to choose the one which has the duty point in the high-efficiency area of the pump. Otherwise, the power consumption of the pump is unnecessarily high.

However, sometimes it is not possible to select a pump that fits the optimum duty point because the requirements of the system change or the system curve changed over time.

**PUMP CONTROL METHODS**

Where a single pump has been installed for a range of duties, it will have been sized to meet the greatest output demand. Therefore, it will usually be oversized and will be operating inefficiently for other duties.

Many water supply systems require a variation of flow demand or pressure and either the system curve or the pump curve must be changed to get a different operating point. In this case, it can be necessary to adjust the pump performance so that it meets the changed requirements.

The most common methods of changing pump performance are (Oleson & Bech 2004):

- Throttle control – the pump runs continuously and a valve in the pump discharge line is fully open or partially closed to adjust the flow to the required value.
- Bypass control – the pump runs continuously and a valve in the bypass line attached to the outlet can be used to adjust the pump performance.
- Impeller trimming – the pump performance will change by modifying the impeller diameters in the pump, meaning.
- Speed control – the pump performance parameters (flow rate, head and power) will change with the varying impeller speeds.

Choosing a method of adjusting the pump performance is based on an evaluation of the initial investment together with the life cycle costs of the pump. Compared with the other methods, which can be carried out during operation, modifying the impeller diameter has to be done in advance, before the pump is installed.

Traditional control methods – throttling valves or bypass lines – waste energy and frequently increase operating and maintenance costs, because valves reduce the flow, but not the energy consumed by the pumps.

The variable speed control method is without no doubt the most efficient way of adjusting pump performance exposed to variable flow rates and pressures (Oleson & Bech 2004). This is because the power consumption reduces as the pump’s speed reduces. Compared with the other methods, the speed control method also makes it possible to extend the performance range of the pump above the nominal speed level of the pump.

The most commonly used device for pump speed variation (reduction) is variable speed drive (VSD) (Pumping Systems Field Monitoring and Application PSAT 2002; European Association for Pump Manufacturers, Hydraulic Institute (US), US Department of Energy 2005; Application guide No. 2 2006). With the introduction of pumps with integrated VSD, it became possible to operate the pump at different impeller speeds, which made it possible to realize other relations between flow and head and between flow and power.

By far the most popular type of VSD is the variable frequency drive (VFD). VFDs are a type of adjustable speed drives used to control the motor (rotational or impeller) speed of an alternating current electric motor by adjusting the frequency and voltage applied to the motor.

During the last decade, VFDs have become increasingly popular for the pump characteristics control.

The use of VFD control offers several advantages (Kaiser et al. 2008; Payne & Harrington 2009). The most significant benefit is the potential to reduce electrical energy consumption and demand from motor-driven processes.

VFD also has the potential to reduce system maintenance and related costs. Control with a VFD affords the capability to ‘soft start’ a motor, which means the motor can be brought up to its running speed slowly rather than abruptly starting and stopping. Soft starting motor results in less mechanical stress on the equipment and, over time, less maintenance is required. Similarly, running the motor at lower speeds extends the lifetime of other equipment components, including shafts and bearings.
EFFECT OF SPEED VARIATION

Varying the motor speed therefore has a direct effect on the pump’s performance. The equations relating rotational (centrifugal) pump performance parameters of flow, \( Q \), to motor speed, \( N \), and head, \( H \), and power, \( P \), to motor speed are known as the Affinity Laws (UNEP Division of Technology, Industry and Economics 2006; Pumps and Pumping System 2006; Prachyl 2010), Figure 4.

- Flow is proportional to the motor speed; \( Q \propto N \).
- Head is proportional to the square of the motor speed; \( H \propto N^2 \).
- Power is proportional to the cube of the motor speed; \( P \propto N^3 \).

Efficiency, \( \eta \), is essentially independent of motor speed, Figure 5. In practice, this is not quite correct.

Affinity equations are shown in Figure 5, where the subscript \( n \) stands for the value under the rated (nominal) condition and the subscript \( x \) stands for the value under the condition with \( N_x \) speed.

As it can be seen from the equations, doubling the motor speed will increase the power consumption by eight times. Conversely, a small reduction in the motor speed will result in a very large reduction of power consumption.

This forms the basis for energy conservation in centrifugal pumps with varying flow requirements. For practical predictions the pressure change effect on the leakage rate is given by FAVAD method (Thornton 2003):

\[
\frac{L_x}{L_n} = \left(\frac{H_n}{H_x}\right)^{n_1}
\]

where \( L_x \) [l/s] is the intensity of the leakage after the pressure change, \( L_n \) [l/s] is the intensity of the leakage before the pressure change, \( H_x \) [m] is the changed pressure value, \( H_n \) [m] is the initial pressure value, \( n_1 \) [1] is the pressure ratio exponent dependent on the type of pipe material.

Combining the speed and pressure ratio affinity law with FAVAD method results in:

\[
\frac{L_x}{L_n} = \left(\frac{N_x}{N_n}\right)^{2n_1}
\]

\[
L_x = L_n \left(\frac{N_x}{N_n}\right)^{2n_1}
\]

where Equation (4) is intensity of the leakage after the speed change.
A large number of pump applications do not require full pump performance 24 hours a day. Therefore, it is an advantage to be able to adjust the pump’s performance. As mentioned earlier, the best possible way of adapting the performance of a centrifugal pump is by means of speed control of the pump.

Figure 6 shows an example of the performance curves of a speed-controlled pump in an open system. The first set of curves shows the \( QH \)-curves and the second one shows the corresponding power consumption curves.

As is indicated in the diagram, it is possible to point out a specific duty point and find out at which speed and at which power consumption level the duty point can be reached. Points of equal efficiency on the curves for different speeds are joined to make the iso-efficiency lines, Figure 6.

In an open system, which is a pump system with a static head, the system curve does not follow the curve of constant efficiency. Instead, it intersects it. This means that the pump efficiency changes along with the speed of the pump. For a pump system without a static head (closed system) where the head required by a system is mainly friction loss, the pump efficiency in general tends to be more constant over a range of speeds, but only to a certain point.

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**SPEED-CONTROLLED PUMP SOLUTIONS**

The usual regulation solutions of the pressure in the water supply systems with speed-controlled pumps are (Bidstrup 2002; Intelligent Variable Speed Pumps 2011):

1. **Constant pressure control** – water supply systems with constant pressure.
2. **Proportional pressure control** – water supply systems with pressure proportional to the flow.

**Constant pressure control**

With constant pressure regulation set pressure, \( H_{\text{set}} \), occurs at the beginning of the pressure pipe, right at the outlet of the pump. The set pressure must provide sufficient pressure for the critical (authoritative) consumer when maximum demand occurs. Therefore, the pressure in the pumping system, except for the maximum demand, will be higher than required. This excess of pressure will be more conspicuous with higher flow resistance. Constant pressure control is recommended if the flow resistance in the water supply system is low.

However, the demand for water varies, so the system characteristic varies as well, according to the required flow. Due to comfort and energy savings, a constant pressure is recommended.

Figure 7 shows the performance curves of a speed-controlled pump delivering constant pressure to the system. The supply pressure is constant and independent of the flow in the whole range of 0–\( Q_{\text{max}} \). A1–A4 are different duty points.

The speed of the pump is reduced from \( N_1 \) to \( N_4 \) in order to ensure that the required head is \( H_1 = H_2 = H_3 = H_4 = H_{\text{set}} \).

The shaded area in Figure 7 represents the decrease in head compared with an uncontrolled pump.
Proportional pressure control

If the resistance in the water supply systems is not negligible, proportional pressure control is recommended. This pressure control also maintains the constant pressure in the system, not at the beginning but at the end of pressure pipe, where critical consumer is defined.

Figure 8 shows the performance curves of a speed-controlled pump delivering proportional pressure to the system. The pressure increases when the flow increases and vice versa. A1–A5 are different duty points.

Different slopes of the proportional system control curves can be selected to fit the pump to overcome the resistance in the system in which it is installed.

The system control curve between two operating points $H_{\text{max}}$ and $H_{\text{min}}$ is usually defined with a straight line, Figure 8(a). $H_{\text{max}}$ is the design duty head of the pump and $H_{\text{min}}$ is the head at minimum flow which is set. However, the system control curve can be modified as a quadratic curve between $H_{\text{max}}$ and $H_{\text{min}}$, Figure 8(b).

The speed of the pump is reduced from $N_1$ to $N_5$ in order to ensure that the required head is $H_1 > H_2 > H_3 > H_4 > H_5$.

The shaded area in Figure 8 represents the decrease in head compared with an uncontrolled pump.

ENERGY COST SAVINGS

Considerable savings in energy can be achieved through efficient pumping.

The first step in calculating the potential energy savings is determining the load requirements for the system.

Many water supply pumping systems require a variation of flow demand or pressure. A helpful way to show the flow demand is to use a duration diagram (European Association for Pump Manufacturers, Hydraulic Institute (US), US Department of Energy 2005), Figure 9.

A duration diagram shows how many hours during a year a given flow rate is needed. Each point of the solid curve, which is interpreted differently, tells how many hours during a year the flow rate exceeds the value on the y-axis.

The water supply system must be able to deliver the peak flow, but, from an economic point of view, it is also important to know at which flow rates the system will operate most of the time.

The second step is defining the energy cost savings by comparing energy costs of traditional control methods (e.g. throttling valves or bypass lines) and the energy costs of variable speed control method. This calculation should be done at each specific flow rate from the duration diagram.

Energy saving for a pumping system with a throttle valve and speed control is shown in Figure 10. Partially closing the valve adds another resistance in the system, raising the system losses and the system curve. The flow rate will now be determined by the point B where the new system curve crosses the pump curve, Figure 10.
Suppose that we have estimated a new operating condition as $Q_2$ and reduced head, $H_r$. In fact, what we actually want is to operate at point C. To achieve this we need to reduce pump speed.

In the first case, using throttling valve, the amount of energy the system consumes is proportional to the flow rate, $Q_2$, and the head, $H_2$.

In the second case, using speed-controlled pump, the amount of energy the system consumes is proportional to the reduced head, $H_r$, and flow rate, $Q_2$.

The difference:

$$Q_2H_2 - Q_2H_r = Q_2(H_2 - H_r) = Q_2\Delta H$$  \hspace{1cm} (5)

represents the energy cost saving of the pumping system. In other words, this difference represents wasted power.

To determine the total cost savings of operating the pump during a year, the running cost savings at each operating condition during a year must be calculated and summed.

Consequently, there is an opportunity to achieve energy cost savings by using the variable speed control method which reduces the power to drive the pump during periods of reduced demand.

**NUMERICAL EXAMPLE**

The effects of frequency regulation control are theoretically examined using a calibrated numerical model of Velika Gorica city’s water supply system. City of Velika Gorica is located in central Croatia, south of the capital city of Croatia, Zagreb. The total length of the water supply network is about 500.0 [km] supplying the total population of 42,500 and several industrial facilities throughout three municipalities. The supply system is a combination of a direct supply through a pumping station, CS2, with the intake capacity of 200.0 [l/s] and an elevated storage tank located at the single highest residential building in the center of the city. The volume of the water tank is 1,000.0 [m$^3$] with a static lift between 49.0 [m] and 53.0 [m] depending on the water level in the tank. Two additional pumping stations are interpolated in the system to ensure the required head for distant parts of the system.

The installed pump (power input of 190.0 [kW] and operating point $Q/H = 200.0$ [l/s]/70.0 [m]) largely exceeds maximum hourly demand that does not have significant seasonal variation. This results in a very short water tank filling time, causing the pump to turn on and off too often while generating unwanted non-stationary phenomena and bursting old pipes. Therefore, the pump is controlled by a throttling valve in order to reduce the flow and extend the operating period, Figure 11. The existing throttle valve regulation is characterized with an average leakage of 34.0 [l/s], the pump electricity cost is in the amount of 285.2 [EUR/d] and the pressure in the system ranges from 4.0 to 4.5 [bar]. Pump operating time is 71.3 [%] and the average total efficiency ($\eta_{pump} \cdot \eta_{motor}$) is 62.3 [%].
The numerical model was designed using the computer program EPANET, version 2.0.12. Measured data included 80 discharge gauge points and 65 pressure gauge points in total. The leakage flow was expressed as an emitter flow in terms of pressure (applying emitter coefficients) so that it was possible to monitor leakage changes in real time in relation to the pressure changes within the system (Walski et al. 2006; Clayton & van Zyl 2007). Two scenarios were analyzed: one regarding a supply system with a storage tank (open system) and another regarding supply system without storage (closed system) with the introduction of variable speed control. The first scenario analyzes the uncontrolled pump and the effects of impeller trimming.

Uncontrolled pump operation analysis of the existing pump is characterized with the leakage of 38.2 [l/s] because of the higher pressure in the system ranging from 4.0 to 5.0 [bar] due to lack of local head loss caused by the throttle valve. With the increased average total efficiency of 67.3 [%] and less operating time (47.3 [%]) the operating cost of the pump would be 193.8 [EUR/d].

Trimming the impeller of the existing pump to meet the existing demand would result in an average total efficiency of 64.8 [%], the operating time of 68.5 [%] and the operating cost of 191.9 [EUR/d]. As it is necessary to maintain the pressure from 4.0 to 4.5 [bar] which allows the filling of the water tank, the average leakage would be 33.8 [l/s], thus practically the same as in the existing conditions.

With the exclusion of the water tank, static head is only 10.0 [m] (flat system area). However, the required fire-fighting pressure is 2.5 [bar]. Considering the above, a minimum pressure value of 3.0 [bar] would secure the need for regular supply of the critical consumer and the fire-fighting conditions. This results in system curves according to Figure 12 (Walski et al. 2010). The analysis of variable speed pumping includes pressure regulation with constant pressure control and proportional pressure control, Figure 12. System demand including leakage ranges from 35.0 to 130.0 [l/s].

As the existing pump is too old, all negative side effects caused by a high pulsing frequency may occur. Thus a new pump with an adequate protection is considered. For this analysis, two pumps (2 x 65.0 [kW]) in parallel connection with two VFD drives are considered as the most expensive solution. Pump and motor efficiency were obtained for the exact pump and motor. VFD drive efficiency is defined as a function of percentage of full operating speed according to the rated VFD horsepower (Rooks & Wallace 2003; Burt et al. 2006).

Introducing proportional pressure control with a pressure ranging from 3.0 to 3.5 [bar], whereby the higher pressure is realized with increased demand, would reduce the existing amount of water losses for 9.1 [l/s] in average, Figure 13. Water loss due to leakage would now be 24.9 [l/s]. With the average total efficiency ($\eta_{\text{pump}} \cdot \eta_{\text{motor}} \cdot \eta_{\text{VFD drive}}$) of 55.6 [%] and 100.0 [%] of time in operation, the pump electricity cost would be 157.3 [EUR/d]. With a constant pressure regulation of 3.5 [bar] and 100.0 [%] of time in operation the average total efficiency would be 56.8 [%] with the operating cost of 162.2 [EUR/d] and the average water loss of 26.2 [l/s], Figure 14.

![Figure 12](https://iwaponline.com/ws/article-pdf/13/4/896/414985/896.pdf)

**Figure 12** | Pump and system curves. (a) With a storage tank; (b) without storage.
To obtain the least costly solution life cycle cost analysis is carried out, Table 1.

Life cycle cost analysis shows that proportional pressure control results in the least costly solution. Data obtained by the utility shows that with the existing throttle control, distribution network repair costs due to leakage are approximately 24,700.0 [EUR/year]. Reduced water losses would reduce repair costs to about 18,050.0 [EUR/year], Table 2. Reduced pressure would cause a decrease in the number of new bursts for about 35.0 [%] (Lambert & Thornton 2011) and would also reduce the background leakage.

**CONCLUSION**

Commonly used methods for pump control in water supply systems include throttle control, bypass control, impeller trimming and pump speed control. However, some

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<tr>
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* 3.0 [%] real interest rate
traditional flow control methods sacrifice energy efficiency and result in unnecessary costs.

VSDs can provide an economically sound and operationally effective solution for flow or pressure control and reduced power consumption, as well as water loss.

The usual pressure regulation solutions in the water supply systems with speed controlled pumps are constant pressure control and proportional pressure control. Proportional pressure control is applied more often, because this method is recommended in water supply systems with relatively great head losses.

By comparing the energy requirements and costs when a throttling device is used for flow control on a centrifugal pump with the power used when a VFD is used to control the same flow, the potential savings become evident.

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