New Techniques and Technologies in Mining – Bondarenko, Kovalevs'ka & Dychkovs'kyy (eds)
 ISBN 978-0-415-59864-4

Efficiency of water supply regulation principles

G. Pivnyak, A. Beshta & A. Balakhontsev
National Mining University, Dnipropetrovs'k, Ukraine

ABSTRACT: The paper deals with energy efficiency of water supply regulation in different applications: mains water supply, heating and cooling systems etc. Each case is characterized by a set of requirements to supply and head pressure level. Two major approaches for supply regulation are compared: throttling and pump speed regulation. It is shown that variable speed electric drive application is not always expedient and in some cases can bring no significant cost benefit. Parallel operation of several pumping units is also examined. Recommendations for pump installations' parameters selection, such as regulation range, are given.

INTRODUCTION

Energy and resource saving programmes are considered now to be key factors for word's industry due to constant growth of prices for fuels and energy resources. Specific energy intensity of gross domestic product in Ukraine is significantly higher than in countries of Central Europe thus making these problems more urgent. However, the awareness of energy management specialists of modern energy saving techniques and principles in Ukraine is still low.

Electric drive is a major consumer of electric power in industry and municipal engineering with its share up to 65%. Electric installations with continuous duty cycle and varying productivity posses the largest potential for energy saving. The prevailing mechanisms in this class are installations with so called parabolic speed-torque characteristic (centrifugal pump, screw propeller etc). Such installations are as a rule equipped with induction motors with squirrel cages. They are widely used in industry and agriculture and municipal enterprises. The electric drive of these installations is mainly uncontrolled and therefore to regulate productivity throttling and bypassing principles are applied. Unfortunately lowering productivity is not equal to lowering electric energy consumption. So this control principle meets technological demands but does not account energy efficiency of water transportation.

Thereby, to justify the expediency of variable speed drive application it is necessary to estimate the cost benefits of this solution.

1 EXISTING APPROACHES ANALYSIS

Selection of proper production control principle is to be made according to technology requirements and specific performances of equipment installed and its economical efficiency. Incorrect evaluation of economical performances of certain control principle leads to introduction of improper technical solution and thus low production efficiency.

The application of variable speed drive itself cannot provide significant power consumption decrease. Energy saving requires thorough evaluation of technological and technical factors concerning production regulation.

The production level of water supply and utilization facilities tends to vary in wide range during operation. The main controlled parameter at the pumping station is the pressure at the discharge line or at the control point (step-up pumping stations of municipal water supply system). In some cases it is water level or its flow (supply) rate.

It should be taken into account that when using uncontrolled water pumps the excessive pressure can occur under low production rate. Excessive pressure in pipeline causes electric power loss. To minimize such losses it is necessary to achieve maximal efficiency by mutual adjustment of pump's mechanical parameters and the entire system.

Energy losses can also be decreased by proper pipeline processing measures – interior surface processing, elimination, or at least, minimization of elbows and narrowings in the line (Goppe 2008). However, these measures most often cannot be applied for existing water supply facilities. It should be taken into account while designing new systems. Meanwhile, application of variable speed drives which become more and more available (Pivnyak & Volkov 2006) is possible for newly developed as good for existing systems' modernization.

The simplified water supply scheme can be presented as it is shown at Figure 1.

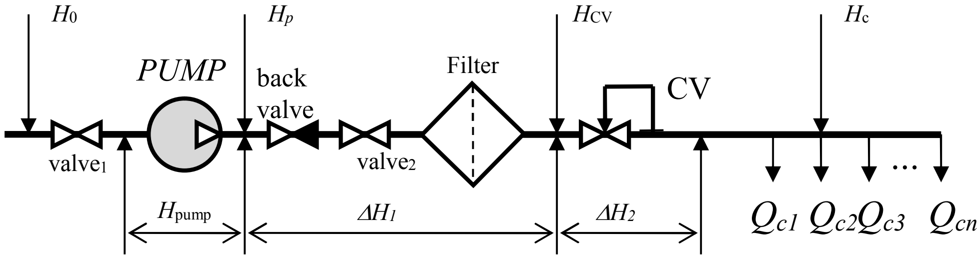

Figure 1. Simplified water supply scheme

The output of pumping unit is supply (flow) rate Q_{pump} and head pressure H_{pump} (Geyner, Dulin & Zarya 1991). Because of extra pressure H_{pump} the head pressure in the mainline grows from H_0 up to H_p. Because pressure drop on the sealing and stop valves and the filter on ΔH_1 the mains pressure downs to H_{CV}. The control valve CV determines pressure drop depending on the mains parameters H_C and Q_C control principle.

The head pressure H_C and flow consumption Q_C can be controlled according to the following principles (Geyner, Dulin & Zarya 1991):

1) $H_C = const$, $Q_C = var$;
2) $H_C = var$, $Q_C = var$;
3) $H_C = var$, $Q_C = const$.

Let us examine each case from the point of energy efficiency.

2 REGULATION UNDER CONSTANT PRESSURE ($H_C = const$, $Q_C = var$)

Let's consider the first principle, when it is necessary to maintain constant pressure in the hydraulic network under varying water consumption (supply) $Q_C = Q_{C1} + Q_2 + ... + Q_{Cn}$. This case is common for main pipelines which must provide necessary supply level for each consumer.

The control valve CV maintains constant head pressure H_C at required constant level by varying pressure drop ΔH_2 value. The output power of the pump unit is defined as

$$P_{pump} = k \cdot H_{pump} \cdot Q_c, \quad (1)$$

where k is the certain proportional gain.

This power is spent to provide necessary consumption and to cover losses $k \cdot (\Delta H_1 + \Delta H_2) \cdot Q_C$ and also to maintain constant pressure level $H_C - H_1$. Thus

$$P_{pump} = k \cdot (H_C - H_0 + \Delta H_1 + \Delta H_2) \cdot Q_c. \quad (2)$$

The electric power rate (consumed from the electric mains) can be determined via efficiency of the pump η:

$$P_{pump,el} = k \cdot (H_C - H_0 + \Delta H_1 + \Delta H_2) \cdot Q_c / \eta. \quad (3)$$

The Q-H curve of the pump is described as follows

$$H = A \cdot \omega^2 + B \cdot \omega \cdot Q + C \cdot Q^2 \quad (4)$$

where A, B, C – coefficients; ω – angular speed of the pump's wheel.

When pump speed is constant its curve can be written as

$$H = A + B \cdot Q + C \cdot Q^2, \quad (5)$$

where $A_1 = A \cdot \omega^2$, $B_1 = B \cdot \omega$, $C_1 = C$.

The hydraulic characteristic of the line is described by

$$H = H_0 + R \cdot Q^2 \quad (6)$$

where H_0 – static pressure (back pressure, or uplift pressure), R – line hydraulic pressure.

The operating mode of the pump unit is defined by the intersection point of pump (Figure 2, curve 1) and hydraulic line (Figure 2, curve 2) characteristics. The intersection point ''1'' is the ideal joint operation calculation point for the pump and the line. At this point the rated supply Q_1 is provided under required pressure $H_1 = H_{pump}$ and maximal efficiency.

When water consumption decreases down to Q_2 level, the operating point moves to the position '2'. It is caused by the rise hydraulic resistance because of the closing of consumers' valves. The head pressure rises up to H_2 value, causing the control valve CV to increases the pressure drop ΔH_2 in order to provide the required line pressure H_C. Meanwhile, the pumping unit keeps operating with pressure H_2. This is an obvious lack of throttling control.

The pressure can also be decreased by pump wheel speed regulation. In this case, the curve of the pump will move to '1' position, and the operating point will move to the '3' position (Figure 2). However, under these conditions, the required supply Q_2 is not provided. The supply will be lower and consumers will open their valves, so decreasing line's hydraulic resistance. After several iterations, a new operating point '4' will be set.

So, the decrease of the supply in the line to ΔQ value by throttling causes the increase of pressure on ΔH and, accordingly, variation of the pump power to

$$\Delta P_d = k \cdot \left(H_1 Q_1 - H_2 Q_2 \right). \tag{7}$$

And same decrease of water supply by means of speed regulation leads to the change of power on

$$\Delta P_\omega = k \cdot \left(H_1 Q_1 - H_1 Q_2 \right). \tag{8}$$

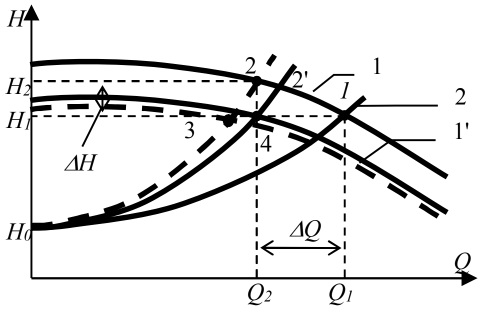

Figure 2

Thus we can asses regulation efficiency comparing power values calculated by (7) and (8) relative to base value $P_1 = k \cdot H_1 Q_1$. So

$$e_d^* = \frac{\Delta P_d}{P_1} = \frac{H_1 Q_1 - (H_1 + \Delta H)(Q_1 - \Delta Q)}{H_1 Q_1} = \\ = \Delta Q^* - \Delta H^* \left(1 - \Delta Q^* \right), \tag{9}$$

$$e_\omega^* = \frac{\Delta P_\omega}{P_1} = \frac{H_1 Q_1 - H_1 (Q_1 - \Delta Q)}{H_1 Q_1} = \Delta Q^*, \tag{10}$$

where $\Delta Q^* = \Delta Q / Q_1$, $\Delta H^* = \Delta H / H_1$.

Formulas (9) and (10) prove that speed regulation is always more efficient than throttling, because
Table 1. Pumps characteristics.
under normal operation it is always $\Delta H^* > 0$, $\Delta Q^* < 1$.

Thus

$$\Delta e^* = e_\omega^* - e_d^* = \Delta H^* \left(1 - \Delta Q^* \right) > 0 . \tag{11}$$

Lets transform (11) to

$$\Delta e^* = \left(\frac{\Delta H^*}{\Delta Q^*} \right) \left(1 - \Delta Q^* \right) \Delta Q^* . \tag{12}$$

The equation (12) shows that the value of Δe^* which determine relative efficiency of speed regulation principle in compare with throttling depends on ΔQ^* regulation range and $\Delta H^* / \Delta Q^*$ ratio at the new operating point. Let us use (5) to determine $\Delta H^* / \Delta Q^*$. Equation (5) in per units is

$$H^* = A_1^* + B_1^* \cdot Q^* + C_1^* \cdot Q^{*2} \tag{13}$$

where $H^* = H / H_1$, $Q^* = Q / Q_1$ $A_1^* = A_1 / H_1$, $B_1^* = (B_1 Q_1) / H_1$, $C_1^* = \left(C_1 Q_1^2 \right) / H_1$.

Taking increments of (13) we obtain

$$\left(\frac{\Delta H^*}{\Delta Q^*} \right) = C_1^* \Delta Q^* - \left(B_1^* + 2 C_1^* \right). \tag{14}$$

And substituting (14) in (12) we estimate efficiency

$$\Delta e^* = \left[C_1^* \Delta Q^* - \left(B_1^* + 2 C_1^* \right) \right] \cdot \left(1 - \Delta Q^* \right) \cdot \Delta Q^* . \tag{15}$$

Formula (15) shows that speed regulation efficiency relative to throttling Δe^* depends on regulation range relative to the given point ΔQ^* and pump's Q-H curve B_1^* and C_1^*

The formula estimates useful power saving under speed regulation. Losses in the pump are not taken into account. However, it is known that pumps efficiency significantly depends on supply, taking maximum value at the rated operating point. That is why in order to estimate energy saving it is necessary to introduce $\eta\left(\Delta Q^* \right)$ in (15). Figure 3 shows corrected dependency (15) for several pumps (pumps data are given at the Table 1 (Popov 1990)).

#	Pump type	Q-H curve equation $H = A_1 + B_1 \cdot Q + C_1 \cdot Q^2$	Efficiency equation
1.	CSP* 38-44-220, 2950 rpm, 38 m^3 / hr, 44 m	$26.8+0.168Q-0.00787Q^2$	$0.04Q- -0.00073Q^2+0.000003Q^3$
2.	CSP 180-76-880, 2950 rpm, 180 m^3 / hr, 76 m	$75+0.139Q-0.00098Q^2$	$0.0068Q-0.000017Q^2+ +0.017\cdot 10^{-6}Q^3$
3.	CSP 180-500-900, 2950 rpm, 180 m^3 / hr, 500 m	$105.5+0.096Q-0.0007Q^2$	$0.0085Q-0.0000276Q^2 +0.014\cdot 10^{-6}Q^3$
4.	CSP 850-240-960, 1450 rpm, 850 m^3 / hr, 24 0m	$126.2+0.035Q-0.000049Q^2$	$0.237\cdot 10^{-2}-0.024\cdot 10^{-4}\cdot Q^2+0.00062\cdot 10^{-6}Q^3$

***** CSP stands for “centrifugal sectional pump”, correspondent soviet abbreviation is CNS.

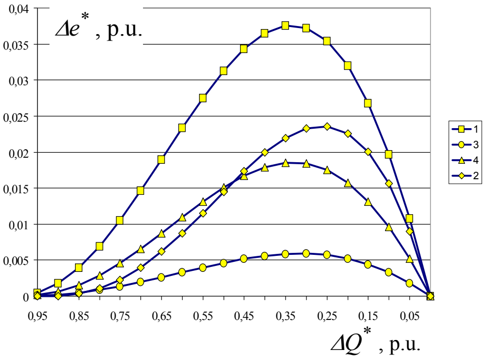

Figure 3.

The analysis of this section shows the following.

1. When head pressure is to be maintained, power (energy) benefits from introduction of speed regulation principle relative to throttling significantly depends on pump's Q-H curve stiffness. The higher stiffness the less efficiency is (look formula (12)). Figure 3 shows that benefit from supply regulation by speed does not exceed 3.7% in the entire regulation range.
2. Maximal efficiency under speed regulation falls within 10..45% supply range relative to operating point. Thus deep regulation (supply less then 45% of the rated value) so and "shallow" (regulation range within 10%) is not expedient. Measures for variable speed drive introduction will not provide cost benefits.
3. When operating at Q-H curve shifts to lower supply zone (to the left), efficiency of the pump will be even less due to higher stiffness of the curve, despite on the fact that efficiency of the motor shifts the same direction under speed regulation.

3 REGULATION UNDER VARIABLE PRESSURE AND SUPPLY

Let us consider supply regulation $H_C = var$, $Q_C = var$ i.e. when there is no requirement for keeping pressure constant. It is peculiar to cases when water supply is stipulated by technology, like refrigerating systems, irrigation and so on.

This case the decrease of supply down to Q_2 value by means of speed regulation lowers pump's Q-H curve and shifts operating point from "1" to "2'" position (Figure 4).

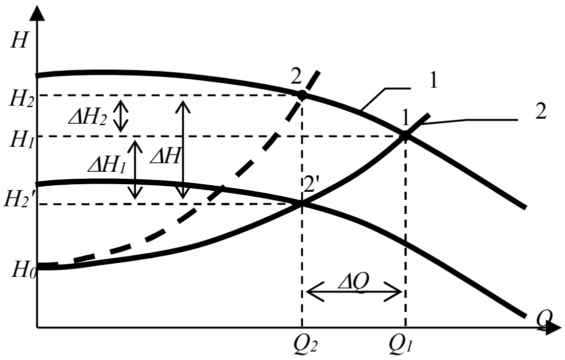

Figure 4.

Still, throttling shifts the operating point to "2" position.

Energy benefits of speed regulation principle relative to throttling is defined by (11), where the difference is calculated by the following expression

$$\Delta H^* = \Delta H^*_{21} + \Delta H^*_{12}, \qquad (16)$$

where $\Delta H^*_{21} = \Delta H_{21} / H_1$ – relative pressure increase in case of throttling; $\Delta H^*_{12} = \Delta H_{12} / H_1$ – relative pressure decrease as a result of pump speed lowering.

It is obvious that ΔH^*_{21} and corresponding efficiency is defined by (14) and (15). The value of ΔH^*_{21} can be estimated by water line's equation in per units

$$H^* = H^*_0 + R^*_1 \cdot Q^*, \qquad (17)$$

where $H^*_0 = H_0 / H_1$, $R^*_1 = R_1 Q_1^2 / H_1$.

The expression in incremental form relative to "1" point, considering that $H^*_1 = 1$, $Q^*_1 = 1$:

$$\Delta H^*_{12} = R^*_1 \Delta Q^* \left(2 - \Delta Q^*\right), \qquad (18)$$

where $R^*_1 = 1 - H^*_0$.

Substituting (14) and (18) in (16) and find

$$\Delta H^* = \left\{\left[C^*_1 \Delta Q^* - \left(B^*_1 + 2C^*_1\right)\right] + \left[R^*_1\left(2 - \Delta Q^*\right)\right]\right\}\Delta Q^*$$

. (19)

From (16) and (19) we determine

$$\Delta e^* = \left[\left(R^*_1 - C^*_1\right)\left(2 - \Delta Q^*\right) - B^*_1\right] \cdot \left(1 - \Delta Q^*\right) \cdot \Delta Q^* \quad .(20)$$

In order to examine (20) let us obtain $d\left(\Delta e^*\right) / d\left(\Delta Q^*\right)$ and find its roots:

$$\frac{d\left(\Delta e^*\right)}{d\left(\Delta Q^*\right)} = D^* - 2\left(3R^* - B^*_1\right)\Delta Q^* + 3R^* \Delta Q^{*2} = 0 \; ,(21)$$

$$\Delta Q_m^* = \frac{\left(R^* + D^*\right) - \sqrt{R^{*2} - R^* D^* + D^{*2}}}{3R^*}, \quad (22)$$

where $R^* = R_1^* - C_1^*$, $D^* = 2R^* - B_1^*$,

$R_1^* = 1 - H_0^*$.

Thus under supply level $1 - \Delta Q_m^*$ the speed regulation benefits will be maximal relative to throttling

$$\Delta e_{max}^* = \left[R^*\left(2 - \Delta Q_m^*\right) - B_1^*\right] \cdot \left(1 - \Delta Q_m^*\right) \cdot \Delta Q_m^* \quad . (23)$$

Equations (22) and (23) shows that Δe_{max}^* depends on relative static pressure H_0^* and coefficients of pump's Q-H curve B_1^* and C_1^* for pump's rated speed.

Let us analyze the dependence of ΔQ_m^* and Δe_{max}^* on the mentioned parameters. It is assumed that they do depend on $R^* = R_1^* - C_1^*$ (not on R_1^*). This assumption affects only factor R_1^* range. Thus $R^* = 1 - H_0^*$, where H_0^* variation range is shifted on C_1^* value. The B_1^* value represents stiffness of Q-H curve. The higher B_1^* is, the higher stiffness is and the less efficiency of speed regulation we obtain.

In the range for different B_1^*=0,1; 0,4; 0,9 trends of maximal efficiency were obtained (Figure 5).

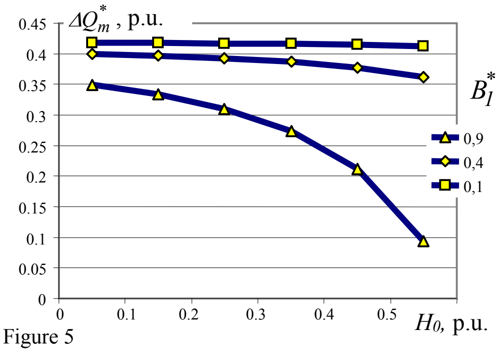

Figure 5

Figure 5 shows that maximal efficiency is achieved under maintenance of the pump with no uplift pressure. And the increase of the Q-H curve stiffness dramatically lowers the speed regulation benefits relative to throttling. The figure also proves that maximal efficiency lies within ΔQ_m^*=0,35…0,42 range. Higher ΔQ_m^* corresponds lower values of B_1^* и H_0^*. For B_1^* = 0, H_0^*=0 we have ΔQ_m^*= 0,423. Substituting this in (23) we obtain $\Delta e_{max}^* = 0{,}385$.

So, for the water supply regulation under variable pressure, the following conclusions can be made.

1. Theoretical maximal efficiency (energy benefit) of water supply by means of pump's speed regulation relative to throttling is 38.5% of the power consumed at the pump's operating point. This efficiency is obtained for 42.3% regulation depth (range) relative to rated supply.
2. The uplift pressure increase significantly decreases speed regulation efficiency (Figure 6).

4 REGULATION UNDER CONSTANT SUPPLY

In the third case for some technologies it is necessary to maintain constant supply $Q_c = const$, which is possible only by varying pressure $H_c = var$. The pumping unit can operate with constant power and supply stability can be provided with bypass system. Another, more efficient principle, is maintaining constant power by speed regulation (Figure 6). Supply stabilization, for example, in water heating system, requires the pump speed lowering.

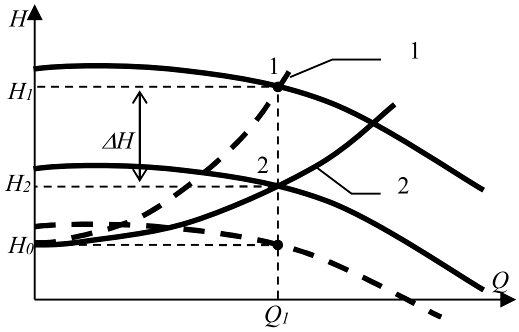

Figure 6

When consumers open their throttles the mains Q-H curve shifts from "1" to the "2" position.

So, the efficiency of speed regulation is defined by

$$\Delta e^* = e_\omega^* - e_d^* = \Delta H^*, \quad (24)$$

where $e_d^* = \dfrac{\Delta P_d}{H_1 Q_1} = 0$,

$$e_\omega^* = \frac{\Delta P_\omega}{H_1 Q_1} = \frac{H_1 Q_1 - Q_1\left(H_1 - \Delta H\right)}{H_1 Q_1} = \Delta H^*.$$

Formula (24) and Figure 7 shows that maximal benefit of pump speed regulation relative to throttling is limited by uplift pressure

$$\Delta e_{max}^* = 1 - H_0^*. \quad (25)$$

Thus when the pump is operating to the water line with no uplift pressure, the benefit of speed regulation is limited only by its stability in low supply range.

5 PARALLEL PUMPS CONNECTION

In some cases it is expedient to connect several pumping units for parallel operation in order to provide higher supply under given required pressure. Conclusions given above can be transferred for this case. For instance, supply regulation can be provided by simultaneous regulation of all connected units. However, it is not always expedient because this way implies installation of variable speed drives for all installation, which is expensive.

Let us analyze parallel operation of two pumps. Total supply, obviously, is defined as a sum of individual supplies

$$Q^* = Q_I^* + Q_{II}^* . \tag{26}$$

where $Q^* = Q/Q_I$, $Q_I^* = Q_I/Q_I$, $Q_{II}^* = Q_{II}/Q_I$, Q_I – total supply.

Let the relative regulation range is $\Delta Q^* = \Delta Q/Q_I$. Assuming that only one variable speed drive is installed (pump II), while the other is fixed speed drive. Then

$$1 - \Delta Q^* = Q_{I,y}^* + \left(Q_{II,y}^* - \Delta Q^*\right), \tag{27}$$

where $Q_{I,y}^*$, $Q_{II,y}^*$ – rated supplies of the pumps.

According to the previous statements, under constant hydraulic pressure it is reasonable to select maximal supply regulation range within λ of its rated supply, i.e. $\Delta Q_{max}^* = \lambda Q_{II,y}^*$. Then under required regulation range ΔQ^* it is necessary to install a pump with rated supply

$$Q_{II,y}^* = \frac{\Delta Q^*}{\lambda} . \tag{28}$$

The supply of the first pump is defined from

$$Q_{I,y}^* = 1 - \frac{\Delta Q^*}{\lambda} . \tag{29}$$

For example, if supply regulation range in the system with constant pressure does not exceed 10% then energy saving (benefit) of speed regulation is only 1.5% relative to throttling (Figure 3). Obviously, it is not wise to install variable speed drive in this case.

Two pumps with one of them equipped with variable speed drive can provide only 3.7% energy saving. And even this small value can only be obtained under wide regulation range – 40..50%. (Figure 7).

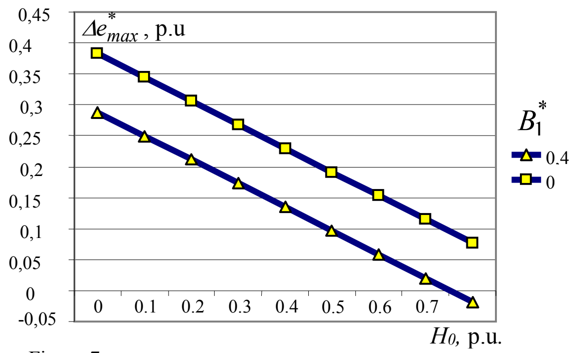

Figure 7

These results can be transferred to the case when the pressure is not to be maintained. In this case maximal regulation depth for variable speed pump is to lie within 35…42% range relative to its rated supply (Figure 5). The increase of uplift pressure at the unregulated pump output is also to be taken into account.

For the case $H_C = var$, $Q_C = const$ there is no necessity to apply pump stations with several pumps. One powerful variable speed drive should be installed.

CONCLUSIONS

1. Theoretical maximal energy benefit from pump's speed regulation application relative to throttling principle is 4% under constant pressure and about 40% when pressure can be varied.
2. When two pumps operates in parallel, supply regulation range of one of then is to be 5…50% of its rated supply.
3. Energy saving due to variable speed drive application is defined by equivalent supply regulation range.
4. Installation of variable speed drive in water mains, where constant pressure must be maintained, is not a reasonable solution. Energy saving (benefits) of in this case does not exceed several per cents relative to throttling.
5. The operation of pumping stations must be organized in a way that each pump would operate at its maximal efficiency under given pressure regulation range. Application of several unregulated drive and one equipped with variable speed drive can be expedient in this case. The regulated pump is to be chosen from required regulation range.
6. The most beneficial is application of variable speed drive in case of necessity of constant supply.

REFERENCES

Goppe, G.G. 2008. *Decreasing energy losses in header water and gas transportation pipes*. Energy saving and water processing.Vol.1(58): 68-73.

Pivnyak, G.G. & Volkov, O.V. 2006. *Modern variable speed induction drives with PWM: monograph.* Dnepropetrovsk: National Mining University: 470.

Geyner V.G., Dulin V.S., Zarya A.N. *Hydraulics and electric drive*. – Moscow: Nedra, 1991. – 321 p.

Popov V.M. *Water pumping units: reference book.* - Moscow: Nedra, 1990. – 250 p