PUMP CONTROL IN A MUNICIPAL PUMPING STATION

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Pumping applications with variable-duty requirements offer great potential for savings energy and may include improved performance and reliability, and reduced life cycle costs. An important enhancement can be achieved by driving some pumps with asynchronous motors fed by variable frequency. The paper focuses on how to create an efficient control algorithm that reduces the energy losses by variable speed of several driven pumps.

Keywords: pump control, control algorithm, asynchronous motor, variable speed

1. INTRODUCTION

Industries worldwide depend on pumping systems for their daily operation. Pumping systems account for nearly 20% of the world's energy used by electric motors and 25% – 50% of the total electrical energy consumption in certain industrial facilities [1]. Independent studies have identified the potential to save over a million tons of carbon emissions a year through the better design of pumping systems.

In particular, many pumping applications with variable-duty requirements offer great potential for savings. The savings go well beyond energy, and may include improved performance, improved reliability, and reduced life cycle costs. Most existing systems requiring flow control make use of bypass lines, throttling valves, or pump speed adjustments. The most efficient of these is pump speed control.

One of the applications with variable-duty requirements is the feeding of residential areas with cold and warm water by municipal pumping stations. The daily demand of water is variable with the day time, the season, week-end and many other parameters. We can identify two demand peaks, one in the morning and another in the evening. Statistical observations permit to draw approximates of a daily diagram of demand. Usually, pumping stations have a number of pumps that can fit into the daily diagram in a satisfactory manner with a number of steps equal to the number of pumps. The best choice is to have identical pumps or pumps with very close rated parameters. If the drive of these pumps is done with asynchronous motors fed directly from industrial grid, the head adjustment at the station exit will be made by bypass lines or throttling valves, which are wasteful energy methods.

An important enhancement can be achieved by driving some pumps with asynchronous motors fed by variable frequency. This means that a number of pumps will control the head at the exit by variable speed of pumps. In this way, a great amount of electrical energy will be saved.

The paper focuses on how to create an efficient control algorithm that reduces the energy losses by variable speed of several driven pumps. Considering a possible daily diagram of water demand, this paper will demonstrate how efficient the proposed algorithm is by looking at the amount of saved energy.

2. THE PUMP DRIVEN AT VARIABLE SPEED

The recommended pumps used in a water pumping station have a flat characteristic $H – Q$ in order to produce small head variations ΔH at strong flow variations ΔQ, from rated flow Q_N to very small flows, for instance 10% of rated flow. The head – flow $H – Q$ characteristic can be modelled by simple analytical equations. Often a second degree polynome is good enough for such modelling:

$$Hp = Ho - k_p Q^2 \tag{1}$$

The constants Ho – static head at $Q = 0$ and kp are identified by applying the equation in two well chosen duty points. Load characteristic has a typical quadratic form because of the phenomenon associated with water flow through the pipe line:

$$H_L = Hg + k_L Q^2 \tag{2}$$

The constant Hg – represents the geodesic position of the consumer and K_L is the load constant. Usually, the flow control to the consumer is done by throttling valves, an additional hydraulic resistance. Fig. 1 shows this flow control method. On the additional resistance appears a head fall Hv, which, multiplied by load flow Q_1, results in the hydraulic power loss on this resistance: $\Delta P = HvQ$.

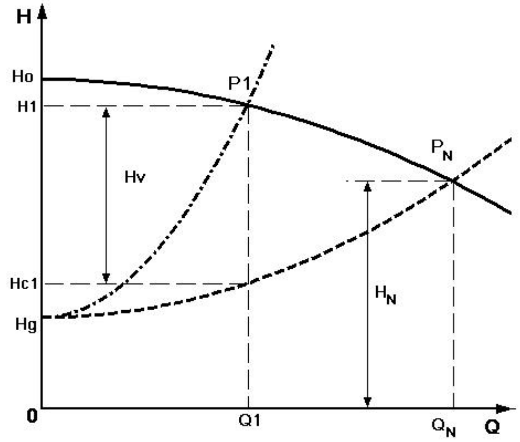

Fig. 1. Flow control by throttling valve

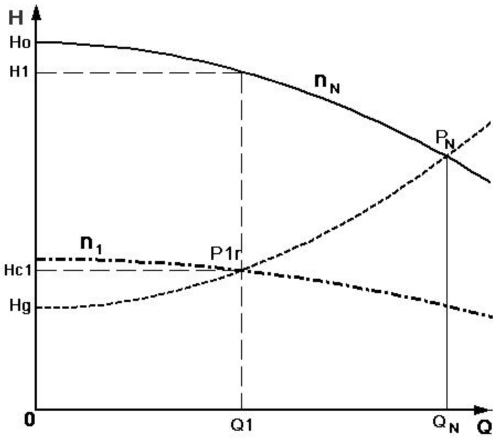

Fig. 2. Flow control by driving the pump at variable speed

The flow control by driving the pump at variable speed is presented in Fig. 2. In fact, the real duty point at the consumer has the coordinates (Q_1; Hc_1) in the $H – Q$ reference frame. Through this point, an artificial pump characteristic has to pass. Each duty point (at given flow Q) on this artificial characteristic corresponds to a constant speed that can be calculated by the affine transformation: $(n_1/n_N)^2 = H_{c1}/H_1$.

The head H_1 is the head on the pump's natural characteristic measured at rated speed given in the pump's catalogue. The new impeller speed n_1 is smaller than the rated speed and can be obtained by feeding the asynchronous motor by PWM converter. In this case, there is no additional in series hydraulic resistance, therefore no additional losses. The recovered hydraulic power can be calculated as follows:

$$\Delta P_{h-rec} = (H_p - H_L)Q = Q\left(H_0 - H_g - (k_p + k_L)Q^2\right) \tag{5}$$

The necessary mechanical power at pump impeller is greater than the hydraulic power because of the pump losses. For the calculation of the recovered electric power absorbed from the industrial grid, we have to know the pump’s efficiency characteristics in any point of the reference frame $H – Q$. There are two great families of efficiency characteristics: 1. $\eta = F(H,Q)$ with impeller’s diameter as family parameter; 2. $\eta = F(H,Q)$ with impeller’s speed as family parameter.

In our case, the useful family is the second, because once installed, the pump doesn’t change the impeller’s diameter and the two families aren’t equivalent! Fig. 3 presents the form of these kinds of characteristics.

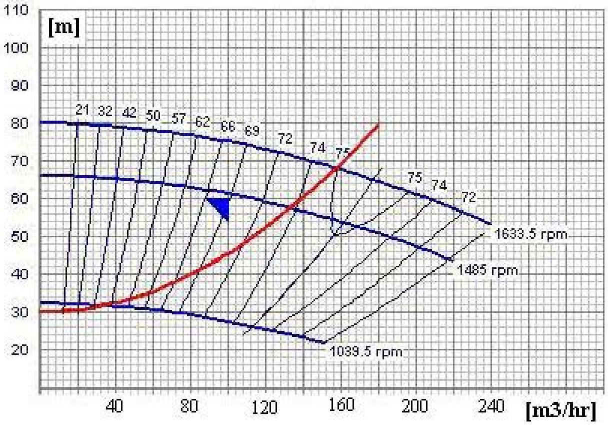

Fig. 3. $\eta = F(H,Q)$ characteristics with impeller’s speed as parameter

To obtain efficiency values on a load characteristic, we have to place the load characteristic on the figure’s frame, at the figure’s scale. After that, we can read the efficiency in some duty points and finally the load efficiency characteristic can be precisely fitted with the Matlab *polyfit* function. A polynome of fourth degree is just enough. We proceed in the same way with the pump’s efficiency on the natural characteristic with rated speed. In order to calculate easily the recovered electrical power by driving the pump at variable speed, we need analytical equations for efficiency in the frame $H – Q$. The actual mechanical power at pumps impeller and the electrical power absorbed from the industrial grid are much stronger than the pure hydraulic power necessary for the pump’s behaviour. At very small flows, the efficiency drops to values near zero! Taking the example of Fig. 3, the curve of 50% efficiency crosses the pump’s characteristic for n = 1485 rpm in the duty point (46 m^3/hr; 64.5 m). This results in a hydraulic power of 8241.67 W and a mechanical power at pump’s impeller twice as strong! Much worse is that at Q = 0, the mechanical power at pumps impeller is not zero! In fact, the pump creates the head H_0 and the water in the pump at flow zero will warm up because of friction losses. At Q = 0 m^3/hr the polynome that approximates the efficiency has to be zero, too! This means that the term of Q^0 has to be zero. The mechanical power at pumps impeller becomes:

$$\eta_p = c(1)Q^4 + c(2)Q^3 + c(3)Q^2 + c(4)Q$$

$$P_m = \frac{(H_0 - k_p Q^2)Q}{Q[c(1)Q^3 + c(2)Q^2 + c(3)Q^1 + c(4)]}; \qquad \lim_{Q \to 0} P_m = \frac{H_0}{c(4)} \tag{6}$$

and the limit at zero flow isn’t zero, because of the inner losses in the pump.

The mechanical recovered power at pump’s impeller can be calculated as follows:

$$\Delta P_{mec-rec} = Q\left(\frac{H_0}{\eta_p} - \frac{H_g}{\eta_L}\right) - Q^3\left(\frac{k_p}{\eta_p} - \frac{k_L}{\eta_L}\right) \tag{7}$$

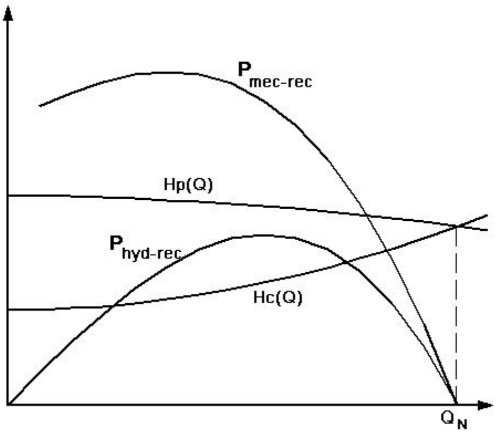

Fig. 4. Recovered hydraulical and mechanical power versus flow

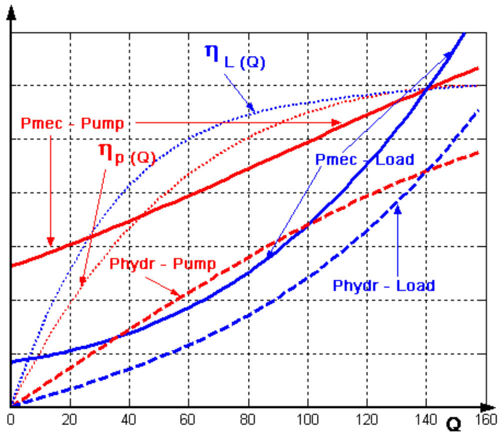

Fig. 5. Explanations to the difference between hydraulical and mechanical powers

Fig. 4 highlights the difference between the recovered hydraulical power and the recovered mechanical power at pump’s impeller and Fig. 5 shows how much the mechanical power exceeds the hydraulical power, both to pump and load.

3. FLOW CONTROL IN A MUNICIPAL PUMPING STATION

The main task of the pumping station is to maintain the imposed head H^* at the exit pipe of the station at any demanded flow. The first approach to solve the problem is the “classical” one, which is the number of pumps in duty will assure the flow, and the head will be adjusted by hydraulic resistance (of a throttling valve) in series with load, or by bypassed flow which increases artificially the pump flow in order to diminish the head at pump’s exit. Nowadays, the head control will be made by speed control of several pumps impeller by driving these pumps with asynchronous motors fed by PWM inverters. The main ideas of flow and head control are:

- A certain part of total flow Q at the moment will be covered by a pumps number Npf working at rated point, at imposed head H^* , equal to rated head of each pump working in parallel,
- The rest of the necessary flow (strong variable component of daily diagram) will be covered by $Npvs$ pumps driven at variable speed,
- All the pumps have the same characteristics and rated parameters.

We’ll describe the procedure to control the head by the speed with the help of Fig. 6, where two identical pumps are working in parallel. One should represent the “fixed speed pump” and another the “controlled speed pump”.

The natural characteristic of both in parallel is given by “1 + 2”. At a flow $Q_N + Qx < 2Q_N$ the head should increase beyond H^*. Driving the second pump with controlled speed, it will have the artificial characteristic “2reg”, that together with the “fixed pump” give the characteristic “1 + 2reg” crossing the H^* line at $Q = Q_N + Qx$.

The number of $Npvs$ pumps in a station is decided by optimization criteria. The economical criteria decide if each driving asynchronous motor is fed by an individual PWM inverter, or all variable speed motors are fed by a single inverter with the necessary installed power.

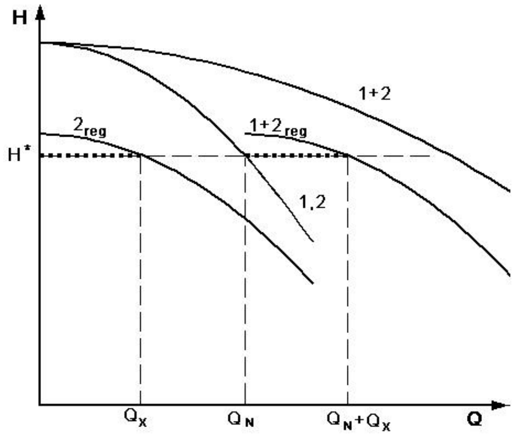

Fig. 6. Help to description of head control by impeller's speed

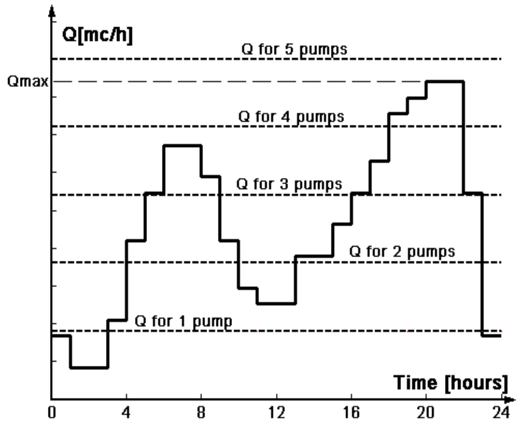

Fig. 7. Daily diagram and the rated flow for increasing number of pumps on duty

The number of pumps in a pumping station have to cover the maximum flow ever asked and a certain "reserve" for future development. The control algorithm will take into account the existence of a daily diagram of demand with two peaks. The construction of such algorithm has the following steps:

- on the "work daily diagram" the flow of number of pumps in duty is drawn,
- where the instant demanded flow is smaller than twice of rated flow of one pump, we'll use one or two pumps with adjustable speed,
- for flows that overpass this limit, a suitable number of pumps with fixed speed will be started in order to cover the more or less "steady" part of flow,
- the feeding frequency for asynchronous motors will be adjusted by a head controller whose exit signal can be related with the necessary impeller speed,
- if two or three pumps with variable speed have to work together, they will have the same flow and impeller speed,
- for very small flows, less than 10% of rated flow of a pump, it is recommended to start a single pump without speed control.

In order to outline how easy such an algorithm works, we will present the results of a LabVIEW program for a pumping station with five identical pumps, three of them drived with variable speed. The daily diagram has the form given in Fig. 7, with $Q_{max} = 840$ m^3/hr and an imposed head of $H^* = 60$ m. These parameters are given because we'll present the evolution of active and reactive power for the case "without speed control" and for the case "with speed control" of three pumps. The LabVIEW program gives the daily diagram and the number of pumps in duty as shown in Fig. 8.

The input parameters for the LabVIEW program are the daily diagram in m^3/hr and the rated flow Q_N of the pumps. The algorithm can be simply deducted from Fig. 7 and Fig. 8. To start with, we know the number of pumps with variable speed *Npvs* and the number of pumps with constant speed *Npf*. If the ratio *r* of instantaneous flow $Q(t)$ and rated flow is smaller than 2 (*Npvs* – 1), only the variable speed pumps will work. For important flows the ratio will be greater, $r > 3$, and then $r - 2$ pumps will be driven with constant speed. Fig. 9 shows the active and reactive energy evolution per day without speed control and final values of energies. This will be the reference for evaluation of energy savings. The final value of active energy in Fig. 10

is smaller than in the previous case. But the difference is very great at the reactive energy, because the PWM converters don't use reactive power from the grid. Unfortunately, the THD of the current exchanged with the industrial grid is high.

The amount of saved energy is important. For the example discussed, the saved energy per day is about 22.15% from the energy without speed control.

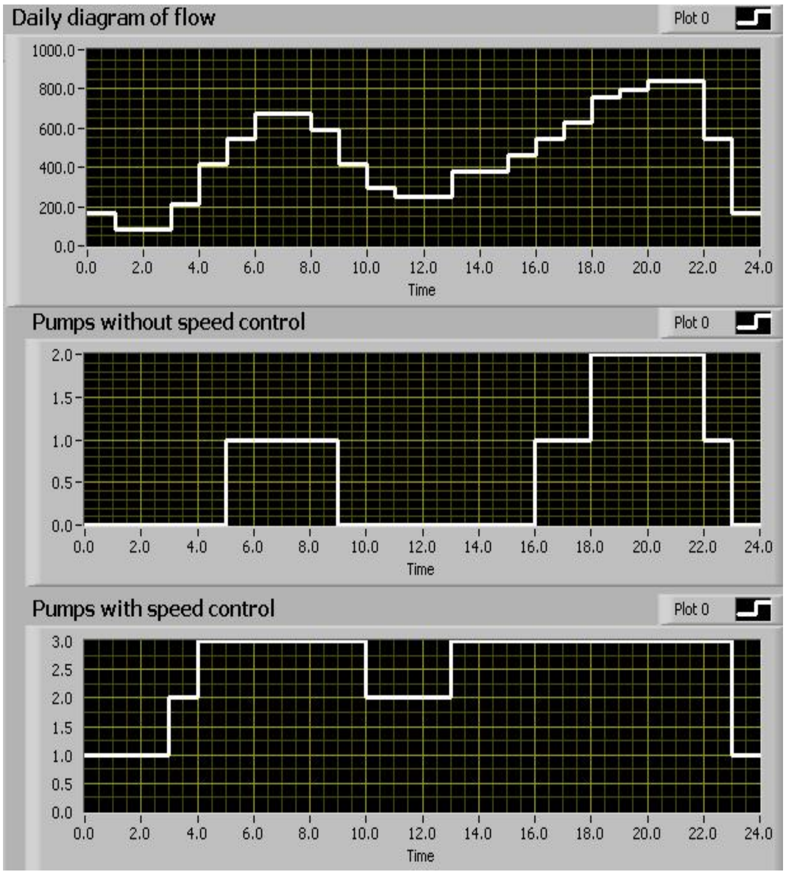

Fig. 8. Daily diagram and the number of pumps in duty

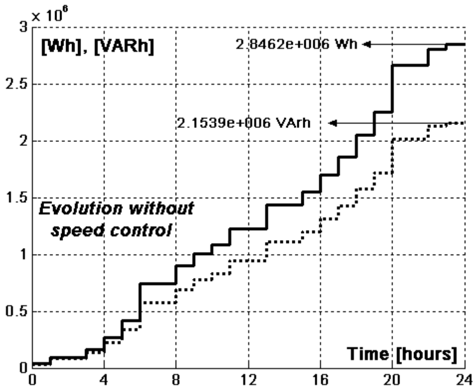

Fig. 9. Evolution of active and reactive energy without speed control

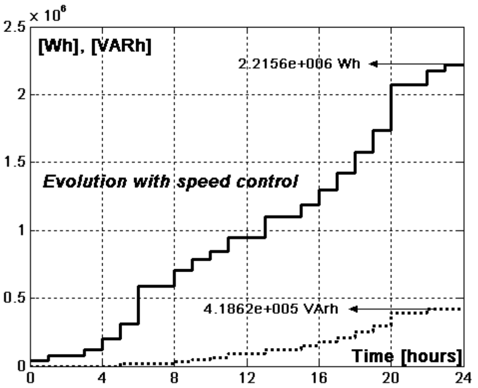

Fig. 10. Evolution of active and reactive energy with speed control

4. CONCLUSIONS

The paper presents a method to evaluate the advantages of driving a pump at variable speed in order to control the head at consumer (for any flow) without additional hydraulic losses.

For a municipal pumping station that feeds consumers with drinking water, there are some additional conditions: the head control at station's exit; the existence of an algorithm to decide how many pumps work for a given flow and how many working pumps have to have variable speed in order to optimize the energy consumption.

5. REFERENCES

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