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Original Research Article

Methodology for Highly Efficient Pump System Management

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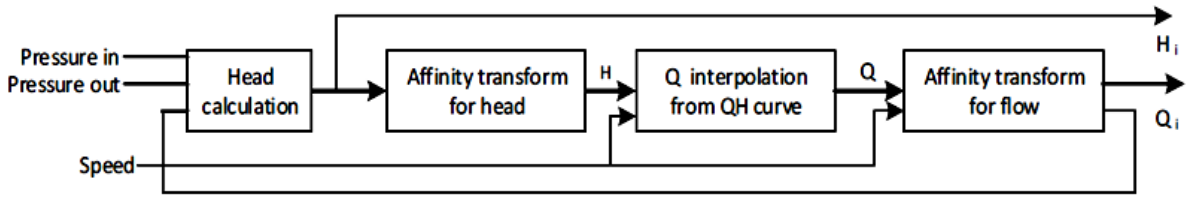
ABSTRACT

Maintaining pump operation in the best efficiency zone (BEZ) is one of the most important tasks of the control system. The main management tasks include detecting work outside the BEZ, preventing actions that may move the work point to undesirable areas, and regulating pumping parameters to keep work within the recommended limits. Thanks to the process control architecture of pump systems that use PLCs and programmable frequency converters, new algorithms can be implemented on these devices. This study provides suggestions for using sensor data, internal frequency converter parameters, or pump performance data from performance graphs for new control algorithms. For this purpose, the methodology of predictive control and performance monitoring were described. Pump operation outside BEZ and related phenomena were considered. The proposed algorithm for high-efficiency pump control in certain speed ranges were elaborated. Simulation of test runs of a pumping system relied on computer simulation tools modeling and software for VSD selection. Experimental tests were also introduced on a laboratory installation that simulates a pumping system.

GRAPHICAL ABSTRACT



Stages of the evaluation method based on the HQ graph

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Introduction

In terms of efficiency, the most appropriate way to start a centrifugal pump is to keep it running as close to the BEP as possible. Thus, the region near the BEP is the preferred or recommended working area for pumping. Performance in this area ensures good reliability, efficiency, energy consumption and long pump life [3]. Harmful phenomena such as hydraulic driving forces on the impeller and cavitation are less likely to occur in this working area, which has a positive effect on pump reliability (Figure 1). When the pump operates far from the fuel cell, outside the recommended range, its efficiency is reduced, and the probability of occurrence of phenomena such as cavitation and vibration is reduced, which reduces the reliability and service life of the pump. Moreover, there is an increased risk of events that damage the process in which the pump is used. These phenomena are an increase in temperature, which may be critical for some types of fluids being treated, and flow recirculation. These data refer to pumps manufactured in accordance with the standard ISO 5199 for use in chemical and technological applications.

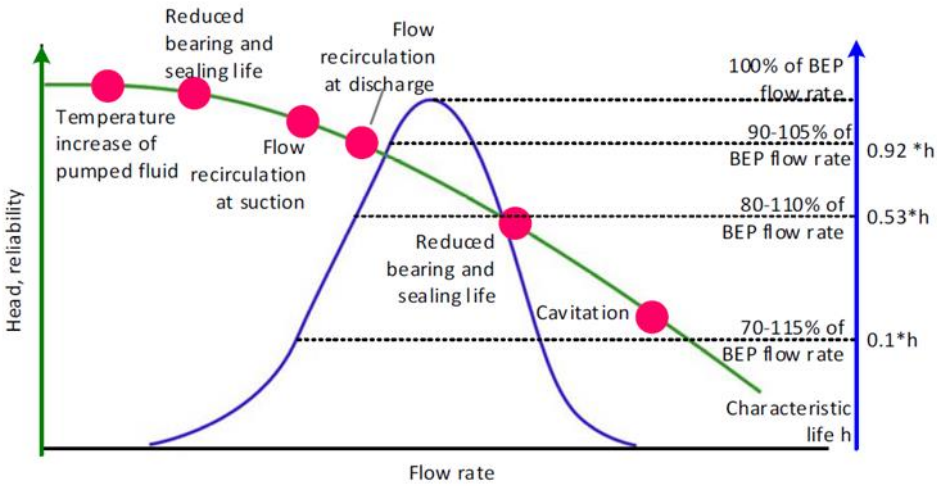

Figure 1: Pump reliability in terms of head and flow. The right axis shows the expected reduction in pump life. Disruptions caused by working outside the recommended area are shown on the graph

The definition of acceptable and preferred operating areas were compared with the flow characteristics depending on the pump head, so that it was divided into two parts. The service life of the pump is not significantly reduced in the preferred working area (PWA) due to vibration and hydraulic forces. It was indicated that PWA could be determined within 70-120% of the consumption for BEP for centrifugal pumps with radial flow. However, according to the reduction in pump life estimated (Figure 1), in this case the estimated pump life will be reduced to 0.1 of the service life. To reduce the service life, the pump should be kept within a narrower range. The permissible working area (PWA) was determined over a wider range of flow values, within which the risk of pump life is lower. Vibration and noise are also present in this region, and their values are higher than in PWA. Bearing life is also reduced in this region. Hence, the assumed the pump's service life is at risk of being reduced when operating in PWA. However, the operation of the pump in this region does not cause immediate destruction of any of its parts.

It is indicated that the limits cannot be estimated theoretically due to the dissimilarity of individual applications. These limitations are often derived from experience and should not be considered a strict definition. To differentiate between acceptable performances areas, the following criteria can be taken into account:

- Pump type and application
- The vibration profile of the pump and system
- HQ curve stability

- Power consumption of the motor
- Physical parameters of the pumped liquid and its temperature.
- Risk of cavitation
- Risk of recirculation, including hydraulic actuation forces, noise
- Energy costs
- Preheating of the pumped liquid is possible
- The power rating of the pump

The limits of permissible continuous operation can be defined for those conditions under which the pump can operate for several thousand hours without damage or excessive wear. For example, the limits of the continuous allowable operating range can be defined so that the efficiency is maintained between 80 and 85% of the maximum. This restriction is justified in terms of energy consumption. In addition, it tends to eliminate the situation where the pump is running with excessive recirculation at partial load (which can occur below Q <50% of full Q) or split flow rate (occurring at high flow rates above BEP).

In practice, for reasons discussed earlier, many pumps are too large due to the fields imposed on the head or flow during the design phase. As a result, these pumps often operate with reasonable recirculation at partial load. Given the reliability of the design, this usually does not lead to noticeable damage at low or medium circumferential speeds, if the pumps are operating in the range of permissible continuous operation.

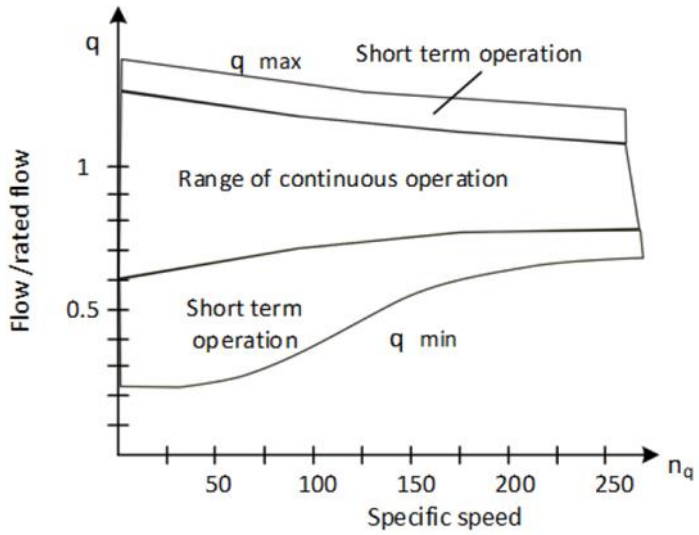

Figure 2: Recommended ranges for continuous and short-term operation relative to the flow

Figure 2 shows the permissible range of continuous operation of the pump depending on the current speed, which is obtained by applying the efficiency criteria discussed earlier. This recommendation is mainly relevant for large pumps with capacities from 500 to 1000 kW. The higher the power and head, the greater the need to work mostly close to the BEP. Again, the recommended range can be changed by a detailed analysis of the specific pump and pumping system, taking into account the mentioned criteria. Small pumps, especially those designed to operate at low speeds, even operate continuously at low loads (below 50% of full capacity).

The risk of cavitation at high flow rates is the main reason for limiting the maximum flow rate. However, the general rules for this case can hardly be determined, since the occurrence of cavitation depends on the net positive suction head, the design of the suction impeller, the circumferential velocity at the inlet of the working wheel, the physical properties of the liquid, and other parameters.

It was it would be correct to recognize that the upper limit of the diagram in Figure 2 mainly applies to systems with a low risk of cavitation. Due to the relatively high reliability of low-speed pumps, the DRO of these devices is often limited:

- Stable area of the HQ curve
- Maximum power consumption of the motor

• The effect of cavitation is observed at partial load and overload

Short -term operation can be defined as an abnormal condition that usually leads to premature wear of pumps, while the cumulative duration per year usually does not exceed 100 hours.

The two approaches mentioned give a general idea of determining the working areas in which the pump should operate. As mentioned earlier, this is highly dependent on the current application conditions. Thus, the display of acceptable and recommended areas is very specific for different applications. In addition, the mentioned approaches are mainly concerned with mechanical reliability and energy consumption. The use of VCD can make it more feasible to drive the pump to more remote locations from the BEP with less risk of physical damage to the pump.

In accordance with these approaches, the limits of the working regions, acceptable or recommended, should be known. Storing the pump in these areas will reduce the risk of physical failure or premature equipment wear. This requires methods for determining the current location of the operating point. In addition, a useful indication of the overall mechanical reliability of the pump in the short and possibly long term can be obtained. While setting limits cannot eliminate the possibility of unexpected failure, it can help prevent pump operation in regions with low energy efficiency or a high risk of physical failure. Thus, knowledge of the current operating status, efficiency and location of the pump operating point can provide a powerful monitoring and protection tool for controlling a pumping system. In the current approach, tools for identifying these parameters and the risk situation are key factors. Frequency converters integrated into control systems provide methods for obtaining the required current parameters without the need to introduce new measuring equipment. As an example, the parameters needed to evaluate the current efficiency can be obtained directly from the frequency converter.

As the first approximation, it is advisable to have a matrix representation of the flow rate and the total head (or pressure) at the nominal speed [Hrs, Qrs] in the form of a reference table stored in the memory of the control unit. Affine transformations allow you to transfer the nominal values from the table to the current values corresponding to the state of the process. The data for filling in the reference table should be taken from the main characteristic graphs of the pump. The same graphs can be used to obtain the required performance for a multi-pump system. The corresponding transformations should be applied to the characteristic curves depending on the topology of the pump connections (Table 1).

Table 1: Search the rated characteristics pump

Levels the system	I	II	III	IV
Total pressure, m	28	26	25	23
Flow rate, m^3/ h	4.32	5.45	6.21	7.12

Using the characteristics of a multi-pump system, the same tables can be created for each number Z. However, the limitation of the table and the need to approximate the data between the table cells, which leads to a decrease in the resolution of the estimates, is a disadvantage of the search approach.

Here we propose a different methodology, where the efficiency criterion is used to calculate the required number of pumps. Each operating point this can be achieved by using a different number of pumps, from 1 to Z. Pumping efficiency is the ratio between the hydraulic power developed by the pump and the electrical power consumed by the pump:

$$\eta = \frac{P_{WP}}{P_{in}} = g\rho HQ\eta_{Pump}\eta_{motor}\eta_{FC} \quad (1)$$

The power consumption can be obtained from the drive documentation, where it is presented in the

form of graphs and tables. It can also be obtained from the drive manufacturer's software for selecting and emulating the drive. Many manufacturers provide such software for determining the size of drives at the system design stage.

The power and torque characteristics are represented by an inverted parabolic curve. Its current shape and location also depend on the speed. Therefore, this characteristic can be described by a second-order polynomial, a function of torque and speed. Typically, the loss P has a non-linear relationship with the speed and torque of the engine:

$$P_{const}(T,n) = P + \Delta P_0(n) - C_{\Delta 1}(n)T - C_{\Delta 2}(n)T^2 \quad (2)$$

where C - is the coefficient of friction.

The efficiency distribution shown in Figure 3 was obtained using this method. Here, the nominal operating characteristics at 3000 rpm were replaced with performance characteristics similar to those obtained at increased speeds of 3750 rpm. Curved areas indicate areas of constant efficiency. As shown in Fig. 3, a system consisting of five parallel centrifugal pumps, where the highest efficiency was observed -33-35%. They are located in the upper-right part of the work area just below and above the solid lines of performance characteristics recorded at rated speeds and dashed lines of system characteristics recommended by the manufacturer. When the pipeline narrows and the speed decreases, the efficiency decreases. Therefore, in order to achieve a relatively high operating efficiency, it is advisable to choose the minimum number of pumps operating at the highest possible speed. Two groups of areas attract the most attention in Figure 3.

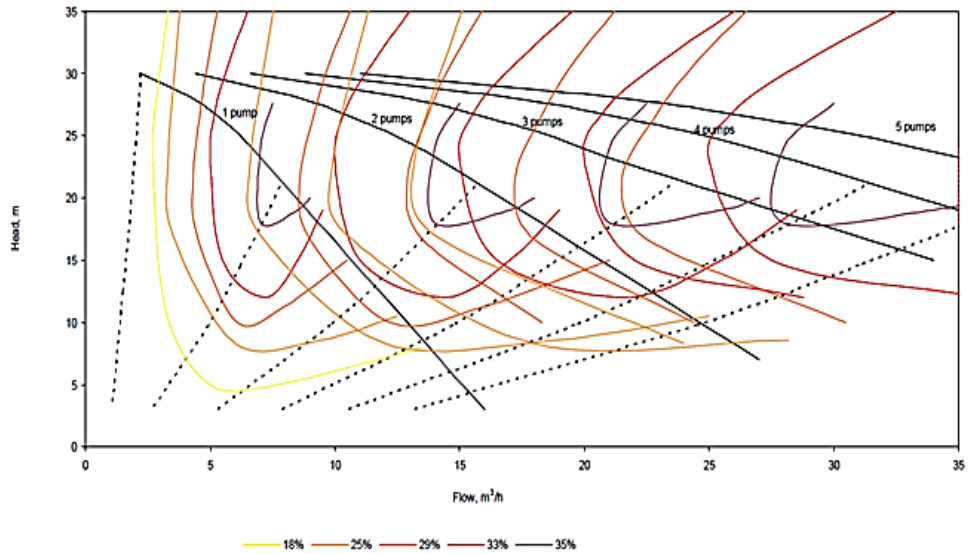

Figure 3: Distribution of efficiency by performance and system characteristics of five parallel-connected Ebara CDX 120/12 pumps.

When low-speed operation is required, when it is not possible to select a smaller number of pumps operating at high speed, the areas to the right of the dotted lines are of interest. In these areas, the efficiency is reduced, and the risk of premature mechanical wear and cavitation is increased, and the service life is shortened.

Fortunately, Figure 3 shows several areas where efficiency can be achieved thanks to additional working pumps higher than with fewer pumps running. For example, to keep the total head below 15m at a flow rate above 7m^3, it is preferable to run two pumps instead of one. In the same way, to keep the total head low when the flow rate is higher than 17 m^3/h, it is better to use three pumps instead of two, whereas at flow rates above 27 m^3, four pumps instead of three look more efficient.

Areas above the rated speed are also worth considering. VSDs provide optimal performance in

the so-called "constant power region", operating at speeds above the rated level with limited engine torque. Consequently, pumping in this area with precise torque control is likely to yield many advantages if fewer pumps are used.

In practice, monitoring and controlling the operation of a centrifugal pump is hardly possible without information about the current location of the pump's operating point. The duty point of the pump can be determined by measuring the available flow or head. In this case, the location of the operating point is indicated by the head generated by the pump and its flow rate. Typically, external meters provide the data needed to control the pump. Generally, in many pumping applications, only the pressure at the pump outlet is measured. Therefore, alternative estimation methods are needed to determine the location of the operating point.

The position of the operating point of a centrifugal pump can be obtained directly by measuring the head H and the flow rate Q generated by the pump. The flow rate can be obtained in various ways. The most commonly used methods are differential pressure and flow rate. The pump head is equal to the total pressure drop across the pump. Therefore, it can be estimated by measuring the difference in static pressure and taking into account the influence of fluid characteristics, flow losses and head velocity, which is due to the difference in fluid velocity in the pump. Thanks to these direct measurements, the duty point can be displayed on the pump performance graph. These measurements provide information regarding the hydraulic or output component of the pump efficiency. The remainder, or input efficiency, is the input power. This can be determined by measuring the mechanical power on the pump shaft, which requires measurement of torque and speed. In the case where the pump motor is driven by a frequency converter, the input power component can be derived from the current parameters of the converter.

Another approach to determining effectiveness is based on evaluation methods. A good example here is the model-based flow calculation method. The model uses centrifugal pump modes based on the HQ pump characteristic provided by the manufacturer and on the Bernoulli principle. The model is configured with values such as fluid density, geometrical parameters of the pipeline and the speed of rotation of the pump. The HQ characteristic of the pump is fed to the model in the form of a reference table, which shows the key points of the curve. The model approach strongly depends on input parameters such as the key points of the HQ curve. The fact that the pump performance may change over time is a disadvantage of this method. On the other hand, this approach makes it possible to monitor the operating state of the pump system using the internal measurements of the frequency converter.

Material and methods

In this approach, the location of the pump's operating point is estimated from the pump's HQ curve. The pump head can be defined as the total pressure drop at the pump. It can be measured by two separate pressure sensors located at the pump inlet and outlet. The speed of rotation of the pump is taken from the corresponding parameters of the frequency converter. The P curve is shifted on the graphical panel of the pump characteristics depending on the value of the current speed. Therefore, to use it when estimating the flow rate, it must be taken according to the current pump rotation speed. Affine transformations provide such an acceptance. Figure 4 describes the process of evaluating the pump flow and head.

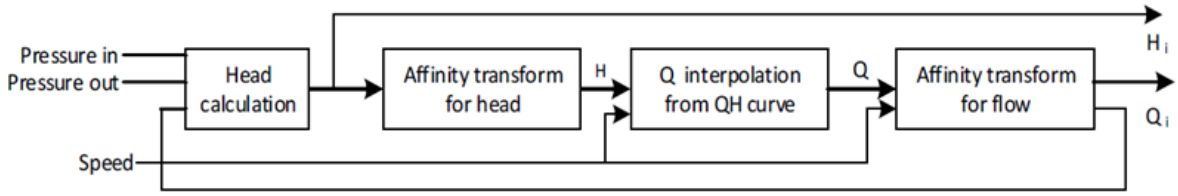

Figure 4: Stages of the evaluation method based on the HQ graph

Typically, the calculations are performed inside a PLC or a programmable frequency converter control unit. The head calculation is based on the Bernoulli equation for incompressible fluids, which can be expressed as:

$$H = \frac{P_d - P_s}{\rho . g} + \frac{8}{\pi^2 . g} . \left(\frac{1 + k_{f,d}}{{d_d}^4} - \frac{1 - k_{f,s}}{{d_s}^4} \right) . Q^2 + (Z_d - Z_s) \quad (3)$$

where, p is the static fluid pressure, k_f is the coefficient of friction loss between the measurement point and the corresponding pump section, and z is the vertical distance between the inlet and outlet pressure sensors. The indexes d and S indicate the discharge and suction side of the pump, respectively. There may be valves and other components of the pipeline system causing flow losses between the pump. The coefficient of friction loss k_f allows you to take these factors into account.

Once the pump head has been calculated, the corresponding flow rate can be estimated from the pump characteristic curve HQ using the interpolation method. The published HQ curve characteristic refers to the rated speed case. Therefore, if the current pump speed differs from the nominal speed, an affinity transformation must be applied to estimate the flow rate relative to the current speed.

The flow rate should be applied to the head calculation as soon as it is estimated for the first time. The working point estimates should be performed at an appropriate time interval that provides instantaneous detection of the changing flow rate. The time interval mainly depends on the dynamic speed and state of the process. The minimum recommended time interval can range from tens of milliseconds to several seconds. In practice, modern control units provide flexible synchronization and launch of tasks, and traditionally process control applications are implemented within tens of milliseconds.

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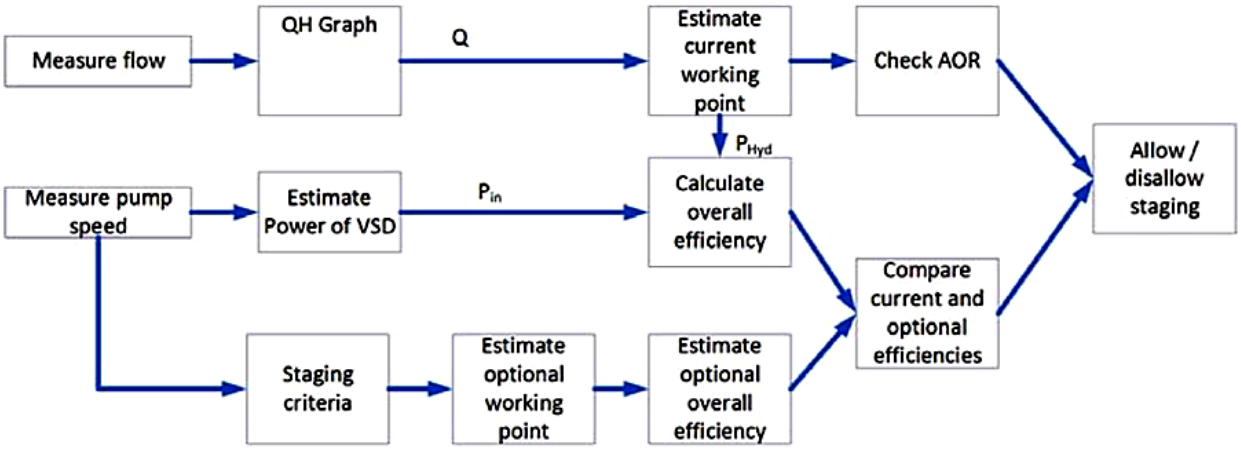


Figure 5: Software architecture for implementing an efficiency-based control method on a PLC platform

It can be stated that each step moves individual pump operating points to the left, moving them away from the fuel pump. This principle is fully justified when the pump is operating at or near the maximum capacity of the AOR limit before setting. However, if the pump capacity limit is still not reached, it may be advantageous to operate at high speed rather than putting the system into operation under AOR conditions.

The assessments and solutions described above can be implemented on the platform of commonly used process control equipment. The following diagram shows the program architecture for these evaluations and preparing data for decision-making

Result and Dissection

The proposed algorithm allows you to evaluate the efficiency of the pump in real time together with its drive. In addition, the additional efficiency for the following cases is constantly re-evaluated and compared with the current efficiency:

• Efficiency for the case when the pump system is switched on (then the additional pump is started); and,

• Efficiency for the case when the pumping system is switched off (one of several pumps is stopped).

These estimates predict the state of the system in the event of a change in the number of pumps in operation and provide information for comparing the current state with some optional (predicted) state. Figures 6 and 7 show the current and predicted operating points for both cases.

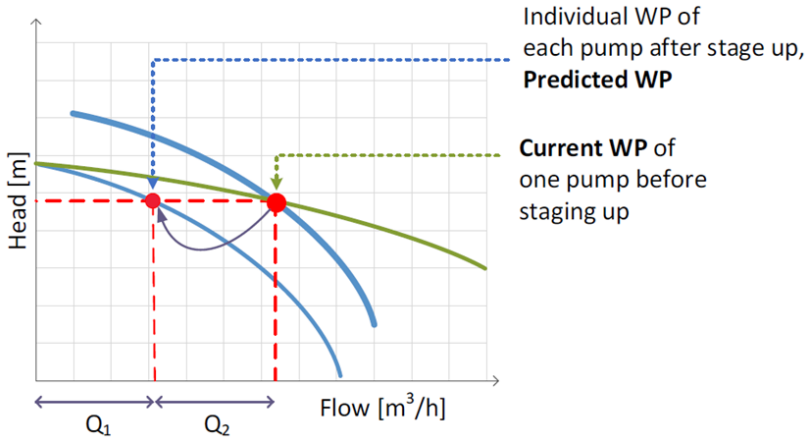


Figure 6: Graphical representation of the predicted state for preparation

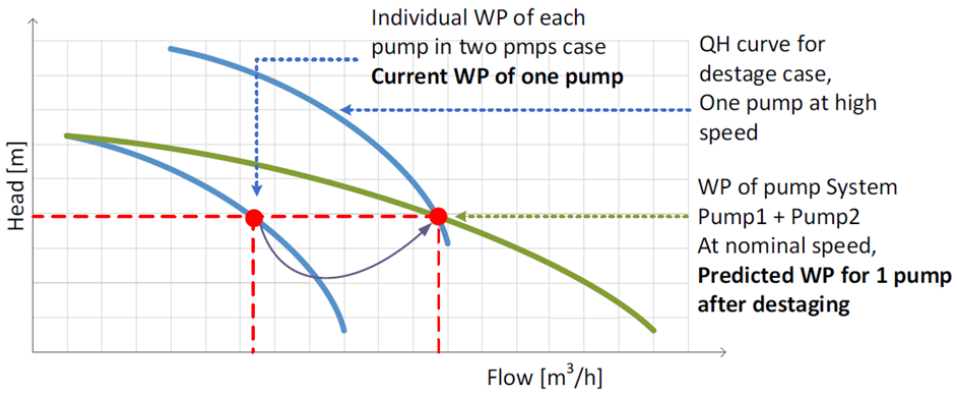


Figure 7: Graphical representation of the predicted state for the event of a shutdown

Both projected work points are reviewed during each program cycle. Therefore, if the preparation criteria are met, the predicted work point is ready to be used for decision-making. The steps of the work points, the overall performance assessment and forecast, and the decision-making are described below.

Setting or disabling is required when the corresponding condition is met. This condition is an increase in speed or an increase in the difference between the process variable and its reference. The first step of the algorithm is to identify the active setting criterion. In the current study, the criterion check was implemented by the standard ACQ 810 frequency converter firmware, based on comparing the current speed with the initial speed limit and the stop speed. The delay in setting is taken into account. If the speed exceeds the specified limits and remains outside the limits for a time exceeding the power-on delay, the start-up stage will be activated.

• To enable a stage, the current efficiency must be compared with the projected performance after the stage. The efficiency after the stage includes the pump efficiency values together with the VCD in the case of setting and shutting down. Therefore, at this stage, it is necessary to evaluate three types of efficiency:

• Current efficiency;

• Efficiency for setting the case; and,

• Effectiveness in the case of destabilization.

The efficiency calculation is divided into two stages: The evaluation of the pump operating point (which is expressed in hydraulic power) and the calculation of the VSD efficiency.

The predictive method based on the model is used to evaluate the current and subsequent stages of the work. To estimate the current working point, you need to get the value of the current flow. This value can be obtained from an external flow meter or calculated using non-sensor estimation methods.

The current expense value is applied go to the HQ curve to get the current pump head value. A numerical representation of the HQ curve in the form of a correspondence table containing the coordinates of key points is used to calculate the pump head. The calculation itself is performed using the interpolation method. Naturally, the accuracy of the calculation depends on the number of key points in the matching table and the rate of curvature of the HQ curve.

To estimate the working point for the preparation case, the HQ curve is used, based on the corresponding (after preparation) number of working pumps. An example of the HQ curve for two working pumps is shown in Figure 7. The HQ curve after the stage for the nominal velocity is estimated using the principle of parallel distribution of the pump flow applied to the key points of the nominal velocity shape of the active HQ curve. The resulting curve is then taken to preserve the desired flow using an affine transformation. The speed value of each pump after the cascade is calculated based on the location of the individual predicted operating point and the HQ curve of the pump.

The same principle is used to determine the operating point in the event of destabilization. The principle of parallel pump flow distribution is applied taking into account the reduction in the number of working pumps. The resulting predicted operating point and the speed value after cleaning are used in further calculations.

The obtained coordinates of the individual operating points provide data for calculating the output power of the pump. The input power is calculated taking into account the power loss in the frequency converter and the motor. The power on the shaft is obtained from the current parameters of the frequency converter (as in the case of the ACQ810).

Power loss data are derived from VSD model. This study used the model provided by the ABB DriveSize tool. The power loss data should be obtained from graphs and integrated into the program in the form of a multidimensional reference table. The current VSD power loss was extracted from the model by applying an interpolation method for the current speed. Predicted speeds were used for post-set cases. The input power was estimated by summing the shaft power from the frequency converter and the power loss from the VSD model. Post-stage current and efficiency were calculated using the hydraulic power values and the received input data.

At this stage, you can compare the current and optional performance. The comparison result provides optional criteria for setting up or shutting down a pumping system. In this comparison, it is possible to increase or decrease the number of pumps in operation, depending on the efficiency for a staged or destabilized version. However, this method of determining staging criteria is beyond the scope of this study.

Once the predicted performance for a post-stage case is known, arming can be enabled or disabled by running the VSD micro-software. If a stage is disabled, the AOR must be considered. The pump is capable of operating in the AOR for a specified period without compromising equipment components.

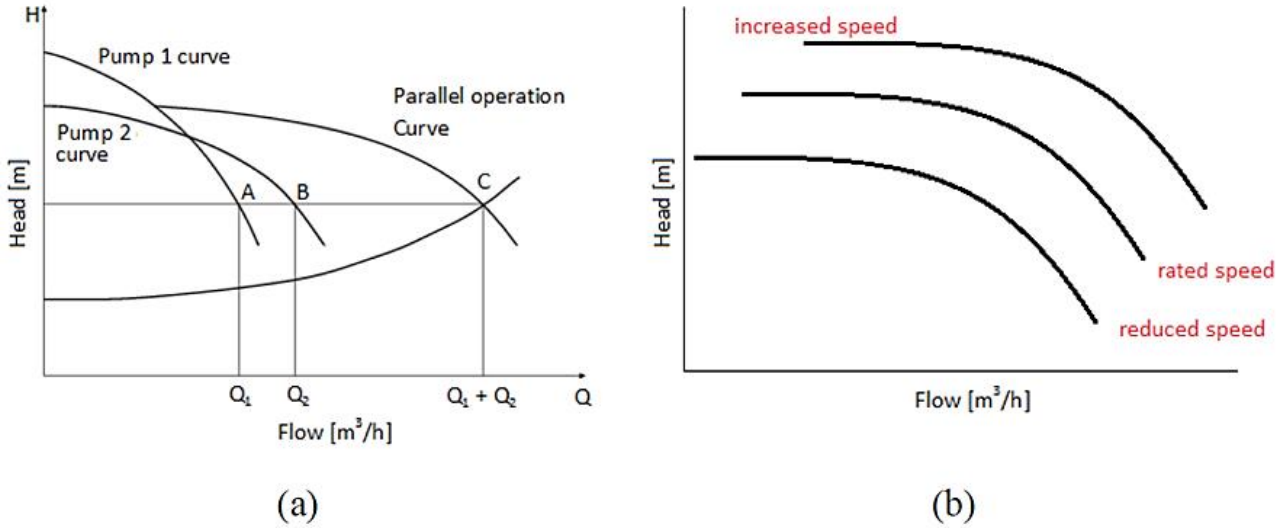
As a rule, efficiency drops when staged. This is due to the shape of the efficiency map. In particular, from the Ebara 120/12 efficiency map working with the ACQ810-04-02A7-4, it can be seen that efficiency gains are only possible when the operating point is moved to the right and up along the HQ performance graph panel. This is possible when the pump speed is increased. In the case of a decrease in speed that occurs when starting a new pump, the duty point (and therefore efficiency) shifts to the left. Taking into account the direction of displacement of the operating point and the drop in the efficiency of the frequency converter at

low speed, it can be argued that the start of a new pump leads to losses when the system is safely operating in the ABM.

To prevent the pump control program from making decisions that could harm the process, reduce the efficiency of the entire pumping system, and move the duty point to undesirable work areas; the potential effects of capacity adjustments must be considered. In order to provide the ability to predict the consequences of changes in the number of operating pumps and its effect on the overall efficiency of the pumping system, this section implements a predictive control method using a polynomial approach.

The proposed method provides a forecast of the efficiency of the pump system in conjunction with a frequency converter. The approach is based on a mathematical model of the pump system, which takes into account the power consumed by the pump itself, as well as the power loss of the frequency converter. This prediction algorithm is implemented on a typical industrial PLC platform, which provides high flexibility in configuration and monitoring. The pump models and VSD are stored in the PLC memory as reference tables. These reference tables contain key VSD and pump specifications published by the manufacturer the pump.

The characteristic RI curves contain information about the performance characteristics of a single pump under rated operating conditions. The operating characteristics of a pump system consisting of Z identical pumps can be obtained from the graph of the HQ curve for a single pump, taking into account that the total capacity of all parallel connected pumps is the total capacity of each of them (Q1 + Q2 +...). The resulting graph is shown in Figure 8 (a).



(a) (b)

Figure 8: Parallel operation of two pumps (operating points A and B). The combined curve and the resulting location of the working point C with the total flow rate Q1 + Q2 (a) and the displacement, respectively, of the velocity dynamics

This adaptation is reflected in the shift of the HQ curve to the right and left in the performance graph panel, depending on the pump rotation speed. The combined HQ curve for multiple pumps slides across the graphic panel in a similar way. The velocity-related dynamics of the combined HQ curve is shown in Figure 8 (b).

As mentioned above, the VSD model was derived from the ABB Drive Size tool. The input data for this tool is the engine parameters, the type of load and its dynamics, information about the parameters of the extreme operation of the engine and the parameters of the power source. The output data in the form of VSD power loss as a function of the pump speed is shown in Figure 9.

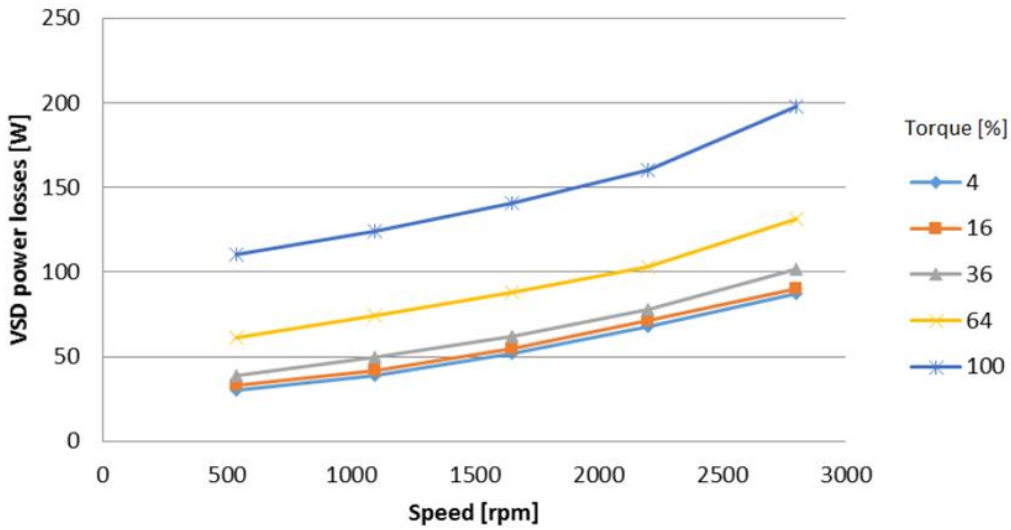


Figure 9: Characteristics of the frequency converter, representing the relationship between the pump speed and the drive losses. The graph shows the power loss of a two-pole asynchronous motor

A graph showing the power loss of the drive as a function of the generated speed allows you to derive the power consumed by the VSD and calculate the total power consumption of the pumping system, including the motor and frequency converter. This applies to both single-pump and multi-pump systems. The input data is stored in the control unit as a reference table containing the key points.

The presented head control algorithm, shown in Fig. 10, is used as follows. First, the required total head H * is determined together with the pump characteristic parameters. At each iteration, the current readings of the flow rate Qi and the total head Hi are obtained. The current common head Hi is then compared to the reference head H *. To reduce their difference, the appropriate flow rate Q * is calculated using the affinity laws (1.5-1.6) as follows:

$$Q^* = Q_i \sqrt{\frac{H^*}{H_i}} \tag{4}$$

Substituting H *, Q * , and Hi, Qi into the system characteristic results in the head Hs of the system and the head Cs being the solution of two equations with two unknown variables:

$$C_s = \frac{H^* - H_i}{Q^{*2} - Q_i^2} \tag{5}$$

$$H_s = H^* - C_s Q^{*2} \tag{6}$$

$$Q_{rs} = -\frac{a}{2} + \sqrt{\frac{a^2}{4} - b} \tag{7}$$

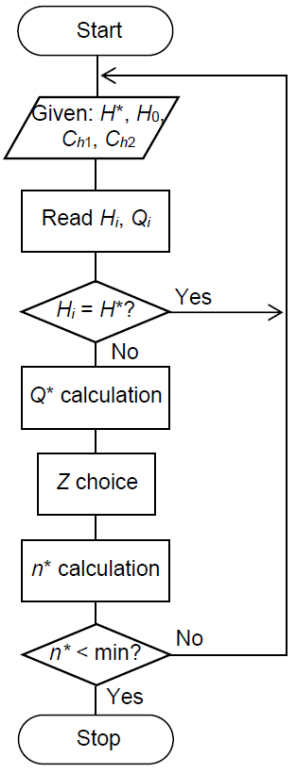


Figure 10: Control algorithm

Conclusion

After evaluating the effectiveness of the system, two areas drew attention: the low-speed area and the high-speed area. It has been shown that in the first case, thanks to the additional working pump,

higher efficiency can be achieved than in the case of a smaller number of pumps. In the latter case, pumping with precise torque control will provide many advantages if you use fewer pumps than if you use more pumps.

As a result, an efficiency map was created, intended both for the management of the pumping system and for analysis at the design stage. Once the control unit is stored in memory, it becomes the core of a new control approach suitable for selecting the optimal number of operating pumps and calculating the required speed setting for these pumps.

The developed scenario of high-performance predictive control provides step by step instructions for the movement of the working point inside and outside the BER.

The proposed new methodology allows evaluating the pump efficiency in real time, taking into account the losses in the drive. It is shown how an algorithm for predicting the location of the operating point can be implemented on a typical industrial PLC platform.

The developed pump model allows combining VSD models with a centrifugal pump in a common simulation environment with ABB DriveSize. This allows you to experiment with different VSD models.

The developed experimental setup provides the possibility of starting the pumping process near real conditions. Real equipment simulates the same physical phenomena that can be observed in a real pumping system.

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Authors' contributions

All authors contributed toward data analysis, drafting and revising the paper and agreed to be responsible for all the aspects of this work.

Conflict of Interest

We have no conflicts of interest to disclose.

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