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On-line optimization control method based on extreme value analysis for parallel variable-frequency hydraulic pumps in central air-conditioning systems

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ABSTRACT

This paper proposes a class of theoretical models based on parallel hydraulic pump characteristics and on-line control optimization methods for variable-flow hydraulic pumps in central air-conditioning systems, which includes the optimized allocation of parallel variable-frequency hydraulic pump speed ratios and the optimized number of operating units. It is suitable for use in fast load variations and for on-line control of variable-flow central air-conditioning systems that use a variable differential pressure setpoint control strategy. This study first establishes a model of parallel variable-frequency hydraulic pump characteristics and uses the extreme value analysis method to investigate the optimal combination of speed ratios of multiple parallel variable-frequency hydraulic pumps. We then explore the optimal number of operating units of parallel variable-frequency hydraulic pumps that utilize the same hydraulic operating point. Using these two methods and monitoring parameters such as the total flow rate and total head of parallel hydraulic ptumps, this study proposes an optimization control method that can perform on-line decision-making for the optimal number of operating hydraulic pumps based on automatically calculated operational zones. The proposed method was applied to a plant with a primary pump variable-flow air-conditioning system that uses a variable differential pressure setpoint control strategy. The actual application shows that this method obtained electricity savings of 15.8% and 4% compared with single-pump operation scheme and dual-pump operation scheme, respectively, for the hydraulic pumps. The method is universal and simple for on-line implementation, which is suitable for control optimization of chilled-water pumps in a variable-flow air-conditioning system.

1. Introduction

In an HVAC system, 60–70% of the power consumption is due to the water pump and fan-based energy transmission system [1], in which the water pump, in particular, has significant potential as a target for energy savings. The optimal control of water pumps in HVAC systems aims to obtain benefits from variable-flow energy efficiency technology. To meet the system user's personalized system regulation, which leads to a wide range of variable-flow features, multiple pumps with variable frequencies running in parallel are often required; frequent changes in hydraulic conditions have presented certain difficulties in developing parallel variable-frequency pumps for energy-efficient technology.

In a parallel variable-frequency pump control system, the control feedback for the pump speed is situated at the lower platform, which often refers to the differential pressure or flow amount. The control logic determining the number of water pumps used is situated at the upper platform of the speed control feedback. The optimization objective of the logic control of the water pump number is to ensure that, at a particular hydraulic operating point, all of the pumps can function with high efficiency so that the total parallel electric pump energy consumption can be reduced.

Kaya et al. [2] have used a more holistic approach to explore ways to improve the efficiency of water pumps and presented various methods of economic analysis. Chen Tao et al.[3] used a parallel variable-frequency water pump with the constant differential pressure setpoint as an example, with a parallel pump efficiency analysis figure as a reference to analyze the number of pump control units corresponding to typical working conditions.

For optimal control of the pump speed, Stan [4] has discussed the safety issues of parallel water pump operation and noted that on-line pump performance monitoring or periodic testing for pumps is necessary for developing a reasonable parallel water pump

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strategy. Leszek [5,6] has used the fixed differential pressure-control strategy as an example and took the correlation between pump efficiency and speed changes as a reference to obtain the speed changing value under optimal conditions. Hu Sike et al. [7] have noted that a parallel variable-frequency pump does not operate in the adjustment range of the synchronous parallel variable-speed control mode operating conditions; its energy efficiency and safety are not as good as those of the synchronous variable-speed control. Xu Guangying [8] has compared the performances of two sets of parallel simulated water pumps, one at constant speed and the other at variable speed, and two sets under variable-speed control.

For control optimization of the number of pump units, Lu Wei et al. [9] took the intersection of power lines and other flow lines corresponding to the pump frequency as a reference to determine the optimal number of operating pumps. Fu Yong Zheng [10] analyzed the flow increment and related issues in the unit number control of parallel pump operation and provided a reference for the design selection of a parallel variable-frequency pump system. Stan [11] analyzed the advantages and disadvantages of running a single-pump and dual-pump program based on the operating costs, capital investment, reliability, characteristics of the hydraulic work, and safety. Ma Zhenjun et al. [12] proposed an optimal pump sequence control strategy including both pump number and speed regulation based on pump power consumptions and maintenance costs. Ahonen et al. [13] developed two model-based methods for pump operation estimation and analyzed factors affecting the accuracy of the estimation methods, this study provided a good reference for analysis and control of pump operation.

Some scholars have applied intelligent algorithms, such as genetic algorithms [14], multi-objective evolutionary algorithms [15], adaptive and derivative algorithms [16] to the control optimization of pump performance.

The above studies indicate that the control optimization of parallel variable-frequency pumps has become a hot research issue for control optimization of HVAC systems. In the above studies, there are two areas of concern: on the one hand, the HVAC system load features frequently change the hydraulic transmission system conditions, which have a greater influence on the real-time operation of the pump optimization control; on the other hand, universality of the optimal control of an ideal pump is essential, where this requires the method to be established on the foundation of a theoretical derivation. To this end, this study used extreme value analysis, based on a parallel variable-frequency water pump, to propose an on-line control optimization method as the working condition reference and used a real variable-flow conditioning system with variable differential pressure setpoint control to investigate the on-line control performance of the method.

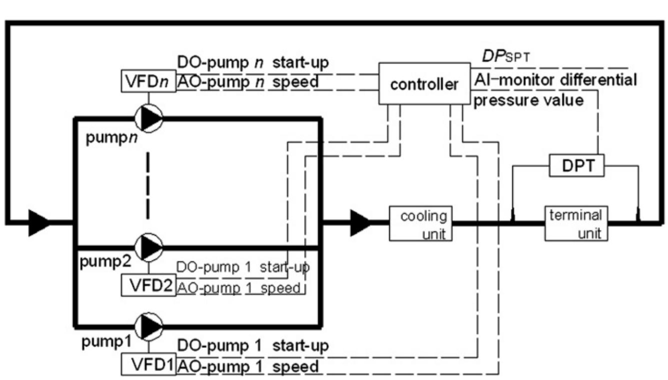

Fig. 1. Schematic diagram of a parallel variable-frequency pump control system.

2. The proposed method

A parallel variable-frequency pump control system with differential pressure control (Fig. 1) is illustrated as an example. For the system shown in Fig. 1, different control methods can achieve the purpose of pressure control. The optimization methods can mainly be described as follows.

(1) For single-pump variable-frequency control, a controller is based on the difference between monitored differential pressure and its setpoint while a corresponding control algorithm is used to calculate the frequency signal of the variable-frequency drive to adjust the pump working conditions. The main entry point of the control optimization is the optimal adjustment of the control algorithm.
(2) For variable-frequency control of parallel pumps, the number of pump units related to the switch control logic optimization must be determined. In other words, to achieve the required hydraulic conditions for the current water system (differential pressure setpoint and water flow required for the system), the selection of the pump operation mode must be optimized so that the power consumption of the parallel pumps is minimized.

The present study addresses two aspects. Assuming that n water pumps in Fig. 1 are the same model, the following working conditions during their operation may exist.

(1) In the parallel operation of multiple pumps, each pump configuration corresponding to different speeds can meet the hydraulic conditions of the current system requirements.
(2) A single-pump or multiple-pump parallel operation can meet the hydraulic conditions of the current system requirements.

In response to these two conditions, the power consumptions of pump operations with different configurations differ such that there is an optimization selection in the control configuration. The first issue is that, in the parallel operation of multiple pumps, the number of revolutions for each pump should match the others to be the most energy efficient; the second issue is that the hydraulic conditions in the system, whether using a single-pump operation mode or a multiple-pump parallel operation mode, must be more energy-efficient. The present study investigated these two problems using speed optimization and the optimal number of units.

3. Parallel variable-frequency pump model

For a pump running at a set speed, the models of the pump head, the efficiency and power can be calculated by Eqs. (1), (2) and (3), respectively.

$$H_{PR} = a_1 G_{PR}^2 + a_2 G_{PR} + a_3. \tag{1}$$

In the above equation, H_{PR} is the head of the pump running at the rated speed (m); G_{PR} is the flow (m^3/s) of the pump running at the rated speed; and a_1, a_2 and a_3 are constants of the performance, obtained from the sample or from data fitting.

$$\eta_{PR} = b_1 G_{PR}^2 + b_2 G_{PR} + b_3. \tag{2}$$

In Eq. (2), η_{PR} is the efficiency (%) of pump running at rated speed; and b_1, b_2 and b_3 are the constants for the performance, obtained from the sample or from data fitting.

$$N_{PR} = \frac{10 H_{PR} G_{PR}}{\eta_{PR} \eta_M \eta_{VFD}}. \tag{3}$$

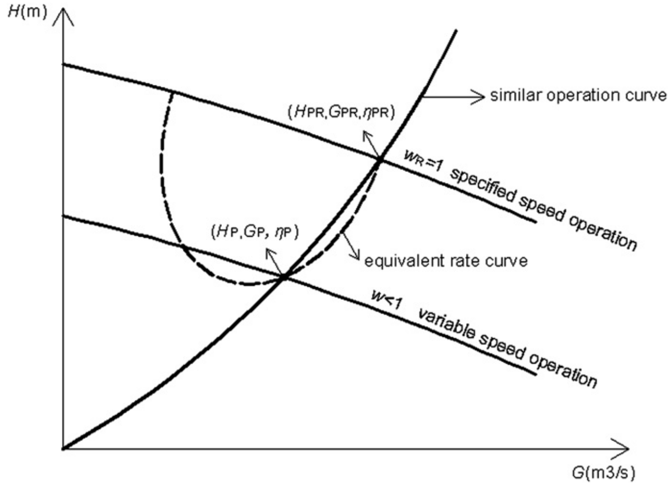

Fig. 2. Pump performance curves for the rated-speed condition and the variable-speed condition.

In the above equation, N_{PR} is the power (kW) of pump running at rated speed; η_M is the motor efficiency (%) and is correlated to pump speed; and η_{VFD} is the variable-frequency drive efficiency (%) and is correlated to pump speed.

We take the ratio of the variable speed and the rated speed of the pump to be w, where w_m is the variable-speed ratio of m parallel pump units. When the pump is in variable-speed operation ($w < 1$), a set of similar operation curves (Fig. 2) can be used to establish the relationship of the head-flow curves in the variable-speed operation and the rated speed operation, as shown in Eq. (4).

$$\begin{cases} G_{PR} = \dfrac{G_P}{w} \\ H_{PR} = \dfrac{H_P}{w^2} \\ N_{PR} = \dfrac{N_P}{w^3} \\ \eta_{PR} = \eta_P \end{cases}. \tag{4}$$

In Eq. (4), G_P is the pump flow of the variable-speed condition (m^3/s); H_P is the pump head of the variable-speed condition (m); N_P is the pump power of the variable-speed condition (kW); and η_P is the pump efficiency of the variable-speed condition (%). When η_{PR} reaches its maximum value, we set the flow to that point named the best efficiency point (BEP) as G_{BEP}. G_{BEP} exists in the variable range of the flow in each pump, i.e., $b_2 > 0$.

By substituting Eq. (4) into Eqs. (1)–(3), we obtain the pump models in the variable-frequency conditions (Eqs. (5)–(7)):

$$H_P = a_1 G_P^2 + a_2 w G_P + a_3 w^2, \tag{5}$$

$$\eta_P = b_1 w^{-2} G_P^2 + b_2 w^{-1} G_P + b_3, \tag{6}$$

$$N_P = \frac{10 H_P G_P}{\eta_P \eta_M \eta_{VFD}}. \tag{7}$$

4. The optimal allocation of parallel variable-frequency pump speed ratios

Suppose that there are m sets of identical pumps in parallel variable-frequency operation. To meet the flow needs and the differential pressure control requirements of the end-user, the working conditions, with the head as H_{RS} and flow as G_{RS}, should be determined by the parallel pump. Let us suppose that the speed ratio, flow, head, pump efficiency, motor efficiency, and variable-frequency drive efficiency of the ith ($1 < i < m$) pump unit are w_i, G_{Pi}, H_{Pi}, η_{Pi}, η_{Mi}, and η_{VFDi}, respectively; the total power of m pumps can then be expressed as $N_{PT} = \sum_{i=1}^{m} 10 H_{Pi} G_{Pi} / \eta_{Pi} \eta_{Mi} \eta_{VFDi}$, N_{PT} is the target parameter for the optimal speed configuration of the parallel variable-frequency pump. Based on the characteristics of parallel pumps, N_{PT} can be written as $N_{PT} = 10 H_{RS} \sum_{i=1}^{m} G_{Pi} / \eta_{Pi} \eta_{Mi} \eta_{VFDi}$.

The problem now consists of determining how to configure the variable-speed ratios of all of the pump units and how to consume the minimum N_{PT} while meeting the same hydraulic operation requirements.

For any pump i of m water pumps, if $G_{Pi}/\eta_{Pi}\eta_{Mi}\eta_{VFDi}$ acquires the minimum value, $\sum_{i=1}^{m} G_{Pi}/\eta_{Pi}\eta_{Mi}\eta_{VFDi}$ also acquires the minimum value, and N_{PT} is minimized. Therefore, the optimization objective can be transformed into $G_{Pi}/\eta_{Pi}\eta_{Mi}\eta_{VFDi}$. Bernier et al. [17] showed that η_M and η_{VFD} depend only on the conversion ratio. When $0.4 < w < 1$, η_M and η_{VFD} are less affected by w: the η_M variable range is [92%, 94%], and the η_{VFD} variable range is [83%, 95%]. When $w < 0.4$, w has a larger impact on η_M and η_{VFD}. Therefore, if the lower limit of the set speed ratio is 0.4, the difference between η_M and η_{VFD} is not significant. On the other hand, because the characteristic curves of the pumps used in an HVAC are relatively flat, the difference between the parallel multiple-pump and the single-pump speed ratios is small; thus, compared to η_{Pi}, η_M and η_{VFD} for the pump energy consumption can be neglected. Therefore, the optimization target is simplified as G_{Pi}/η_{Pi}. For any two pumps Pi and Pj, combined with the η_{Pi} expression results in

$$\begin{cases} \dfrac{G_{Pi}}{\eta_{Pi}} = \dfrac{1}{b_1 G_{Pi} w_{Pi}^{-2} + b_2 w_{Pi}^{-1} + b_3 G_{Pi}^{-1}} \\ \dfrac{G_{Pj}}{\eta_{Pj}} = \dfrac{1}{b_1 G_{Pj} w_{Pj}^{-2} + b_2 w_{Pi}^{-1} + b_3 G_{Pj}^{-1}} \end{cases}. \tag{8}$$

In Eq. (8), the minimum value obtained by G_{Pi}/η_{Pi} and G_{Pj}/η_{Pj} in the same condition implies that the denominators on the right side are maximized. With Pi as an example such that $b_1 G_{Pi} W_i + b_2 W_i + b_3 G_{Pi}^{-1}$ (where $W_i = w_{Pi}^{-1}$) obtains the maximum value, the extreme value conditions can be expressed as $-(b_2/2b_1 G_{Pi})$; i.e., $w_{Pi} = -(2b_1 G_{Pi}/b_2)$. We can combine this with $G_{BEP} = -(b_2/2b_1)$ to give $w_{Pi} = G_{Pi}/G_{BEP}$.

When $w_{Pi} = G_{Pi}/G_{BEP}$, G_{Pi}/η_{Pi} reaches the minimum value, and the pump is in the most energy-efficient working condition. Referring to the w_P variable range:

(1) When $G_{Pi} \le G_{BEP}$, $w_{Pi} \le 1$, which is in line with the actual variable-speed ratio, the minimum value w_{Pi} corresponding to G_{Pi}/η_{Pi} is $-(2b_1 G_{Pi}/b_2)$. Similarly, when $G_{Pj} \le G_{BEP}$, the minimum value w_{Pj} corresponding to G_{Pj}/η_{Pj} is $-(2b_1 G_{Pj}/b_2)$. According to the parallel pump characteristics:

$$a_1 G_{Pi}^2 + a_2 w_{Pi} G_{Pi} + a_3 w_{Pi}^2 = a_1 G_{Pj}^2 + a_2 w_{Pj} G_{Pj} + a_3 w_{Pj}^2 \tag{9}$$

If we substitute $w_{Pi} = -(2b_1 G_{Pi}/b_2)$ and $w_{Pj} = -(2b_1 G_{Pj}/b_2)$ into Eq. (9), we find:

$$\left(a_1 - \frac{2a_2 b_1}{b_2} + \frac{4a_3 b_1^2}{b_2^2}\right)(G_{Pi} + G_{Pj})(G_{Pi} - G_{Pj}) = 0. \tag{10}$$

Suppose that $C_1 = a_1 - (2a_2 b_1/b_2) + 4a_3 b_1^2/b_2^2$; the size relationships of a_1, a_2, a_3, b_1, and b_2 show that C_1 is non-zero. Therefore, the established condition of Eq. (10) is $G_{Pi} = G_{Pj}$, which means that, when $w_{Pi} = w_{Pj}$, the value of N_{PT} is a minimum.

(2) When $G_{Pi} > G_{BEP}$, $w_{Pi} > 1$, which exceeds the actual speed ratio range, the minimum value w_{Pi} corresponding to G_{Pi}/η_{Pi} is 1. Similarly, when $G_{Pj} > G_{BEP}$, the minimum value of the corresponding G_{Pj}/η_{Pj} is 1. By substituting $w_{Pi} = 1$ and $w_{Pj} = 1$ into Eq. (9), we obtain:

$$(G_{Pi} - G_{Pj})\left[a_1(G_{Pi} + G_{Pj}) + a_2\right] = 0. \quad (11)$$

Suppose that $C_2 = a_1(G_{Pi} + G_{Pj}) + a_2$; the size relationships of a_1 and a_2 show that C_2 is non-zero. Therefore, the established condition of Eq. (11) is $G_{Pi} = G_{Pj}$, which means that the value of N_{PT} is minimized when $w_{Pi} = w_{Pj}$.

(3) When $G_{Pi} \le G_{BEP}$, $w_{Pi} \le 1$, which is in line with the actual speed ratio range; the minimum value w_{Pi} of the corresponding G_{Pi}/η_{Pi} is $-(2b_1G_{Pi}/b_2)$. When $G_{Pj} > G_{BEP}$, the minimum value w_{Pj} of the corresponding G_{Pj}/η_{Pj} is 1. By substituting $w_{Pi} = -(2b_1G_{Pi}/b_2)$, $w_{Pj} = 1$ and $G_{BEP} = -(b_2/2b_1)$ into Eq. (9), we find:

$$a_1(G_{Pi} + G_{Pj})(G_{Pi} - G_{Pj}) + a_2\left(\frac{G_{Pi}^2}{G_{BEP}} - G_{Pj}\right) + a_3\left(\frac{G_{Pi}^2}{G_{BEP}^2} - 1\right) = 0. \quad (12)$$

From the size relationships of a_1, a_2 and a_3, the established condition of Eq. (12) is obtained as $G_{Pi} = G_{Pj} = G_{BEP}$; because $G_{Pj} > G_{BEP}$ in this working condition, G_{Pi}/η_{Pi} and G_{Pj}/η_{Pj} cannot reach the minimum values at the same time.

(4) When $G_{Pi} > G_{BEP}$, $w_{Pi} > 1$, which exceeds the actual speed ratio range; the minimum value w_{Pi} corresponding to G_{Pi}/η_{Pi} is 1. When $G_{Pj} \le G_{BEP}$, the minimum value w_{Pj} corresponding to G_{Pj}/η_{Pj} is $-(2b_1G_{Pj}/b_2)$.

By substituting $w_{Pi} = 1$, $w_{Pj} = -(2b_1G_{Pj}/b_2)$ and $G_{BEP} = -(b_2/2b_1)$ into Eq. (9), we find:

$$a_1(G_{Pi} + G_{Pj})(G_{Pi} - G_{Pj}) + a_2\left(\frac{G_{Pj}^2}{G_{BEP}} - G_{Pi}\right) + a_3\left(\frac{G_{Pj}^2}{G_{BEP}^2} - 1\right) = 0. \quad (13)$$

From the size relationships of a_1, a_2 and a_3, the established condition of Eq. (12) is obtained as $G_{Pi} = G_{Pj} = G_{BEP}$; because $G_{Pi} > G_{BEP}$ in this working condition, G_{Pi}/η_{Pi} and G_{Pj}/η_{Pj} cannot reach the minimum values at the same time.

To summarize the above information, for any two pumps of m identical parallel variable-frequency pumps that correspond to the same hydraulic conditions, the conditions with the minimum total power efficiency for the parallel pumps are that the speed ratios of the two pumps are the same, which means that the speed ratios of m parallel pumps are the same.

5. The optimal unit number for a parallel variable-frequency water pump

For m pump units, as an example, take $m = 2$, which is similar to the condition of $m > 2$. For two pumps running in parallel, we assume that the configuration of the two pumps has reached the requirement of the system (i.e., operating head H_{RS}, flow G_{RS}).

Option 1, i.e., a single pump operates at variable speed w_1, the pump head is H_{RS}, the pump flow is G_{RS}, and the pump power efficiency is $N_{P1} = 10H_{RS}G_{RS}/\eta_1\eta_{M1}\eta_{VFD1}$.

Option 2, i.e., two pumps operate in parallel at variable speed w_m, and the total pump power is N_{Pm}. Because the variable speeds of all of the pumps are the same, the pump efficiencies are the same, which gives$N_{Pm} = \sum_{i=1}^{m} 10H_{RS}G_{Pi}/\eta_m\eta_M\eta_{VFD} = 10H_{RS}G_{RS}/\eta_m\eta_M\eta_{VFD}$.

According to parallel pump operating characteristics, we have:

$$a_1G_{RS}^2 + a_2w_1G_{RS} + a_3w_1^2 = a_1\left(\frac{G_{RS}}{m}\right)^2 + a_2w_m\left(\frac{G_{RS}}{m}\right) + a_3w_m^2. \quad (14)$$

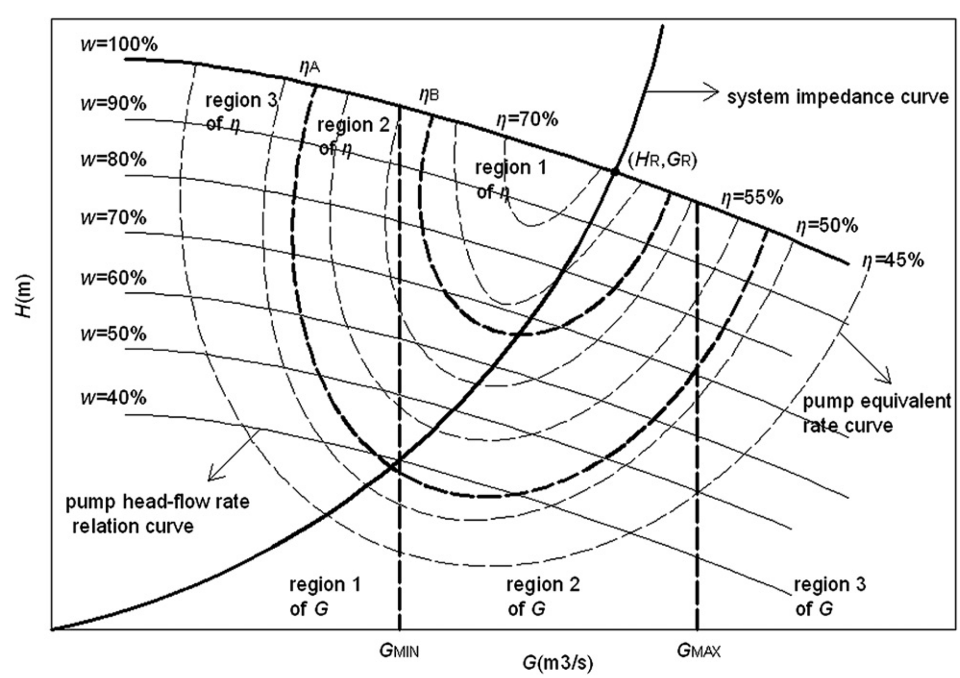

Fig. 3. The reference diagram for the optimization control of a parallel variable-frequency pump.

Table 1
The regional, working condition and optimal configuration of a parallel variable-frequency pump.

Flow region	Working condition	Efficiency region	Working condition	Optimal configuration
Region 1 of G	$G_{RS}<G_{MIN}$	Region 1 of η	$\eta_1>\eta_B$	Plan 1
		Region 2 of η	$\eta_1\in[\eta_A, \eta_B]$	Plan 1
		Region 3 of η	$\eta_1<\eta_A$	Plan 2
Region 2 of G	$G_{RS}\in[G_{MIN}, G_{MAX}]$	Region 1 of η, region 2 of η	$\eta_1\geqq \mathrm{MIN}(\eta_A, \eta_B)$	Plan 1
		Region 3 of η	$\eta_1<\mathrm{MIN}(\eta_A, \eta_B)$	Plan 2
Region 3 of G	$G_{RS}>G_{MAX}$	Region 1 of η	$\eta_1>\eta_A$	Plan 1
		Region 2 of η	$\eta_1\in[\eta_B, \eta_A]$	Plan 1
		Region 3 of η	$\eta_1<\eta_B$	Plan 2

Because $m>1$, we can refer to Eq. (14), which gives $w_m<w_1$. From the head algorithm model, we know that $H_{RS}<a_3w_m$, which means that the range of w_m is $\sqrt{H_{RS}/a_3}<w_m<w_1$. If we set the lower limit of the variable-speed ratio as 0.4, when the range of the variable speed is small, η_M and η_{VFD} can be considered to be the same. When the revolution ratio is higher than 0.4 and the variable range is small, the effects of η_M and η_{VFD} on N_P can be ignored, which means that the difference between w_1 and w_m is small. The two schemes of η_M and η_{VFD} can be considered as almost the same; the sizes of N_{P1} and N_{Pm} mainly depend on the sizes of η_1 and η_m. Assuming that $\Delta\eta=\eta_1-\eta_m$ and combining the expressions of η_1 and η_m, we find:

$$\Delta\eta=-b_1G_{RS}^2M^2-b_2G_{RS}M+b_1\left(\frac{G_{RS}}{w_1}\right)^2+b_2\frac{G_{RS}}{w_1}. \quad (15)$$

In the above equation, $M=1/mw_m$; thus, Eq. (15) gives $1/mw_1<M<1/m\sqrt{a_3/H_{RS}}$. When $M=-(b_2/2b_1G_{RS})$, Δ obtains the minimum value $\Delta\eta_{MIN}=b_1w_1^{-2}G_{RS}^2+b_2w_1^{-1}G_{RS}+b_2^2/4b_1$. When we combine this value with the expression of η_1, we obtain $\Delta\eta_{MIN}=\eta_1-b_3+b_2^2/4b_1$.

(1) When $\eta_1>b_3-b_2^2/4b_1$, $\Delta\eta_{MIN}>0$, which means that $\eta_1>\eta_m$. Thus, Option 1 is more energy-efficient. However, according to the expression of η_1, $b_3-(b_2^2/4b_1)$ is the maximum value corresponding to η_1. Therefore, $\eta_1>b_3-(b_2^2/4b_1)$ cannot be established.

(2) When $\eta_1\le b_3-(b_2^2/4b_1)$, $\Delta\eta_{MIN}\leqq 0$, and the size relationships of η_1 and η_m cannot be determined. If the M value corresponding to $\Delta\eta_{MIN}$ is M_O, then the range of M $[M_{MIN}, M_{MAX}]$ is:

$$\begin{cases} M_O=-\dfrac{b_2}{2b_1G_{RS}} \\ M_{MIN}=\dfrac{1}{mw_1} \\ M_{MAX}=\dfrac{1}{m}\sqrt{\dfrac{a_3}{H_{RS}}} \end{cases}. \quad (16)$$

(1) $M_O<M_{MIN}$, which means that $G_{RS}>-(b_2mw_1/2b_1)$; M is in the range of extreme points on the right side of the monotonically increasing region. By combining this with the expression of η_1, the minimum and maximum values of $\Delta\eta$ can be expressed respectively as:

$$\Delta\eta_{MIN}=\left(1-m^{-1}\right)\left(\eta_1-b_3+b_1m^{-1}w_1^{-2}G_{RS}^2\right), \quad (17)$$

$$\Delta\eta_{MAX}=\eta_1-b_3-b_1G_{RS}^2\left(m\sqrt{\frac{H_{RS}}{a_3}}\right)^{-2}-b_2G_{RS}\left(m\sqrt{\frac{H_{RS}}{a_3}}\right)^{-1}. \quad (18)$$

When $\Delta_{MIN}>0$, we have $\eta_1>b_3-b_1w_1^{-2}G_{RS}^2m^{-1}$, which gives $\eta_1>\eta_m$. Thus, Option 1 is more energy-efficient. When $\Delta_{MAX}<0$, we obtain $\eta_1<b_3+b_1G_{RS}^2(m\sqrt{H_{RS}/a_3})^{-2}+b_2G_{RS}(m\sqrt{H_{RS}/a_3})^{-1}$, which gives $\eta_1<\eta_m$. Thus, Option 2 is more energy-efficient.

(2) When $M_O\in[M_{MIN}, M_{MAX}]$, which means that $G_{RS}\in[-(b_2m/2b_1)\sqrt{H_{RS}/a_3}, -(b_2w_1m/2b_1)]$, if we compare Δ_{MIN} and Δ_{MAX}, the maximum values of the two are set as $\Delta\eta_M$. When $\Delta_M<0$, which gives $\eta_1<\eta_m$, Option 2 is more energy-efficient.
(3) When $M_O>M_{MAX}$, which means that $G_{RS}<-(b_2m/2b_1)\sqrt{H_{RS}/a_3}$, M is in the range of extreme points on the right side of the monotonically decreasing region. At this point, the maximum and minimum values of Δ are Δ_{MIN} and Δ_{MAX}, respectively. When $\Delta_{MAX}>0$, which means that

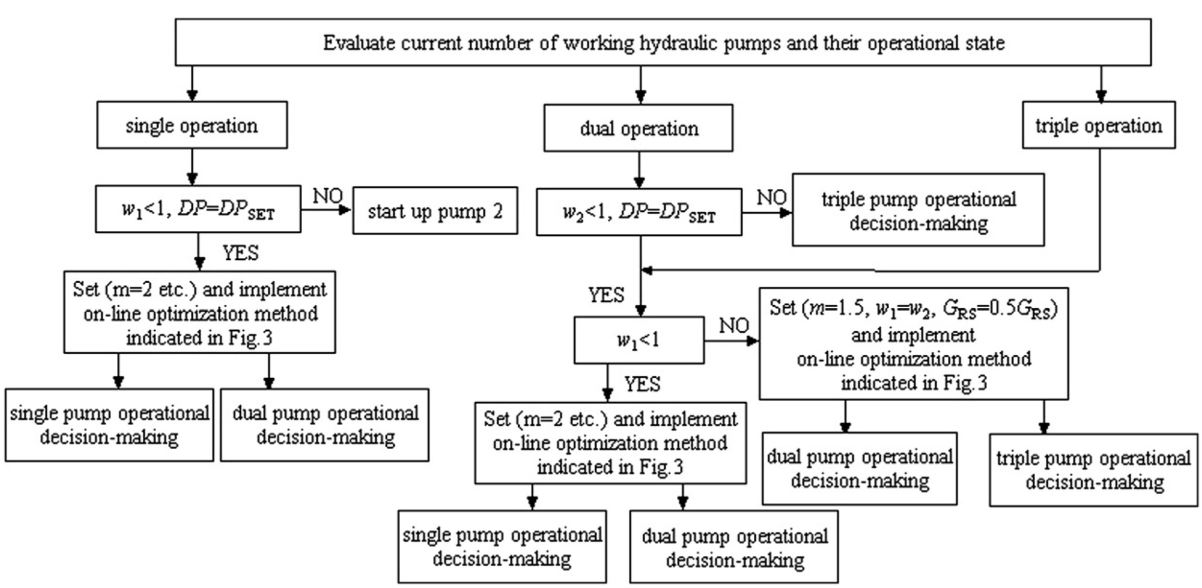

Fig. 4. A flow chart for the implementation of the optimal control method of parallel variable-frequency pumps.

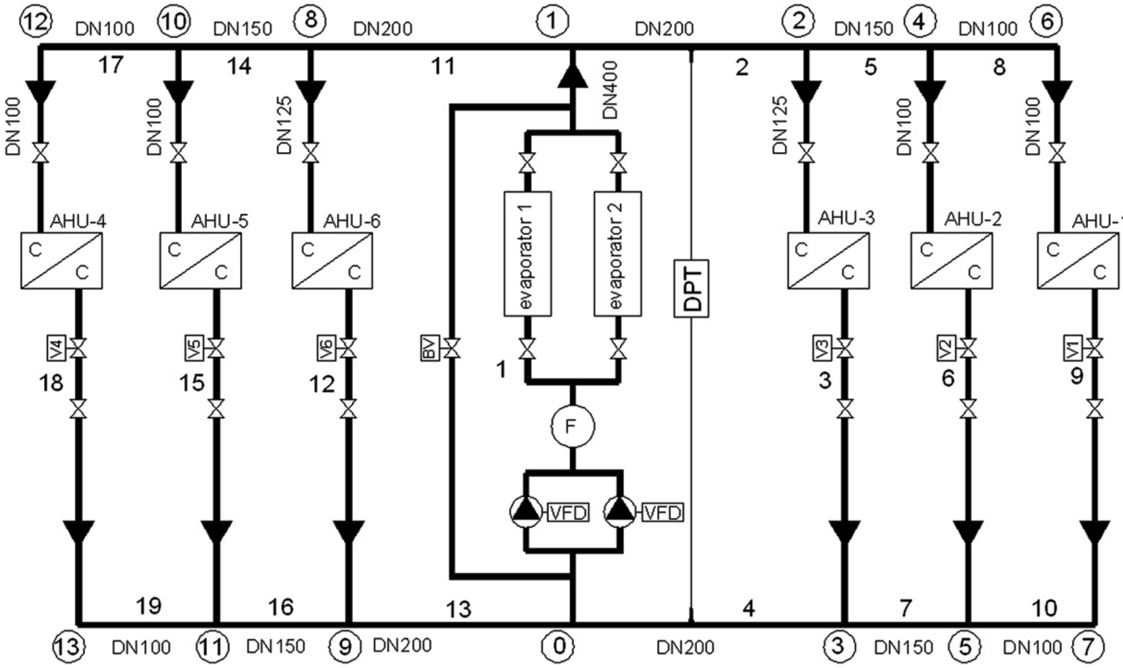

Fig. 5. A representation of a chilled-water-air-conditioning plant in a Suzhou factory.

$\eta_1 > b_3 + b_1 G_{RS}^2 (m\sqrt{H_{RS}/a_3})^{-2} + b_2 G_{RS}(m\sqrt{H_{RS}/a_3})^{-1}$, $\eta_1 > \eta_m$; thus, Option 1 is more energy-efficient. When $\Delta_{MIN} < 0$, which means that $\eta_1 < b_3 - b_1 w_1^{-2} G_{RS}^2 m^{-1}$, $\eta_1 < \eta_m$, Option 2 is more energy-efficient.

6. Optimization control method and application of a parallel variable-frequency water pump

6.1. Optimization control method of a parallel variable-frequency pump

Via extreme value analysis, the above derivation provided solutions to the two issues proposed in this study. Based on the above analysis, we can refer to the parameters in this operating condition, such as the system flow G_{RS} and the pump efficiency η_1 monitoring range, to determine the optimal number of pumps to operate. The working condition of the region can be used to express the derivation of Section 5 in the pump performance curve shown in Fig. 3. In Fig. 3, $G_{MIN} = -(b_2 m/2b_1)\sqrt{H_{RS}/a_3}$, $G_{MAX} = -(b_2 m w_1/2b_1)$,$\eta_A = b_3 - b_1 w_1^{-2} G_{RS}^2 m^{-1}$ and $\eta_B = b_3 + b_1 G_{RS}^2 (m\sqrt{H_{RS}/a_3})^{-2} + b_2 G_{RS}(m\sqrt{H_{RS}/a_3})^{-1}$. G_{MIN} and G_{MAX} divide the system flow into three regions: the region 1 of G, the region 2 of G, and the region 3 of G. η_A and η_B also divide the pump efficiency into three regions: the region 1 of η, the region 2 of η, and the region 3 of η. The region boundaries are indicated as bold dotted lines in the graph. The optimal configurations of the working conditions in each region are shown in Table 1. Because G_{MIN}, G_{MAX}, η_A, η_B, and the conditions of the hydraulic system are closely related, the regional conditions vary with the changes in working conditions in the graph. As shown in Fig. 3, the current pump operating point (H_{RS}, G_{RS}) is inside of the region 2 of G and the region 1 of η, we can use Table 1 to obtain the current working condition configuration and find that the single-pump operating configuration is more energy-efficient.

Table 1 and Fig. 3 present the optimal control method proposed by this study for a parallel variable-frequency pump. The method is based on a theoretical analysis of two issues and is generalizable. Another feature of this method is that it is very suitable for on-line application. Its conditions for implementation are as follows.

(1) Find the most accurate model of the pump's characteristics, which determines a_1, a_2, a_3, b_1, b_2, and b_3.
(2) Continuously monitor the pump head and the system flow, i.e., H_{RS} and G_{RS}.

To calculate w_1 in the η_A equation, if a single pump operates in the current condition, w_1 can be directly obtained through the pump frequency feedback signal; if two pumps operate in the current condition, w_1 can be obtained with the help of the head and flow relationship. On the one hand, H_{RS} and G_{RS} can be easily monitored; on the other hand, by means of a programmable logic

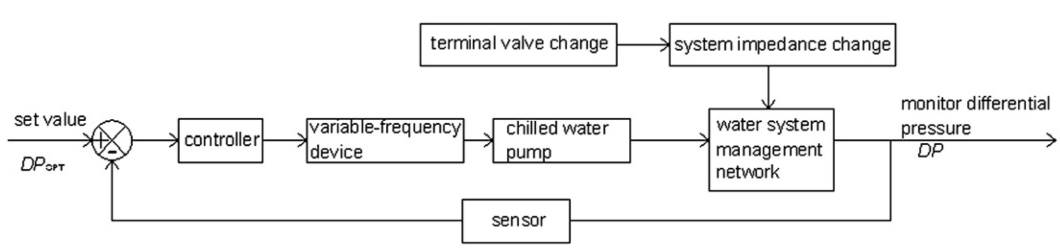

Fig. 6. Schematic diagram of a differential pressure-control loop.

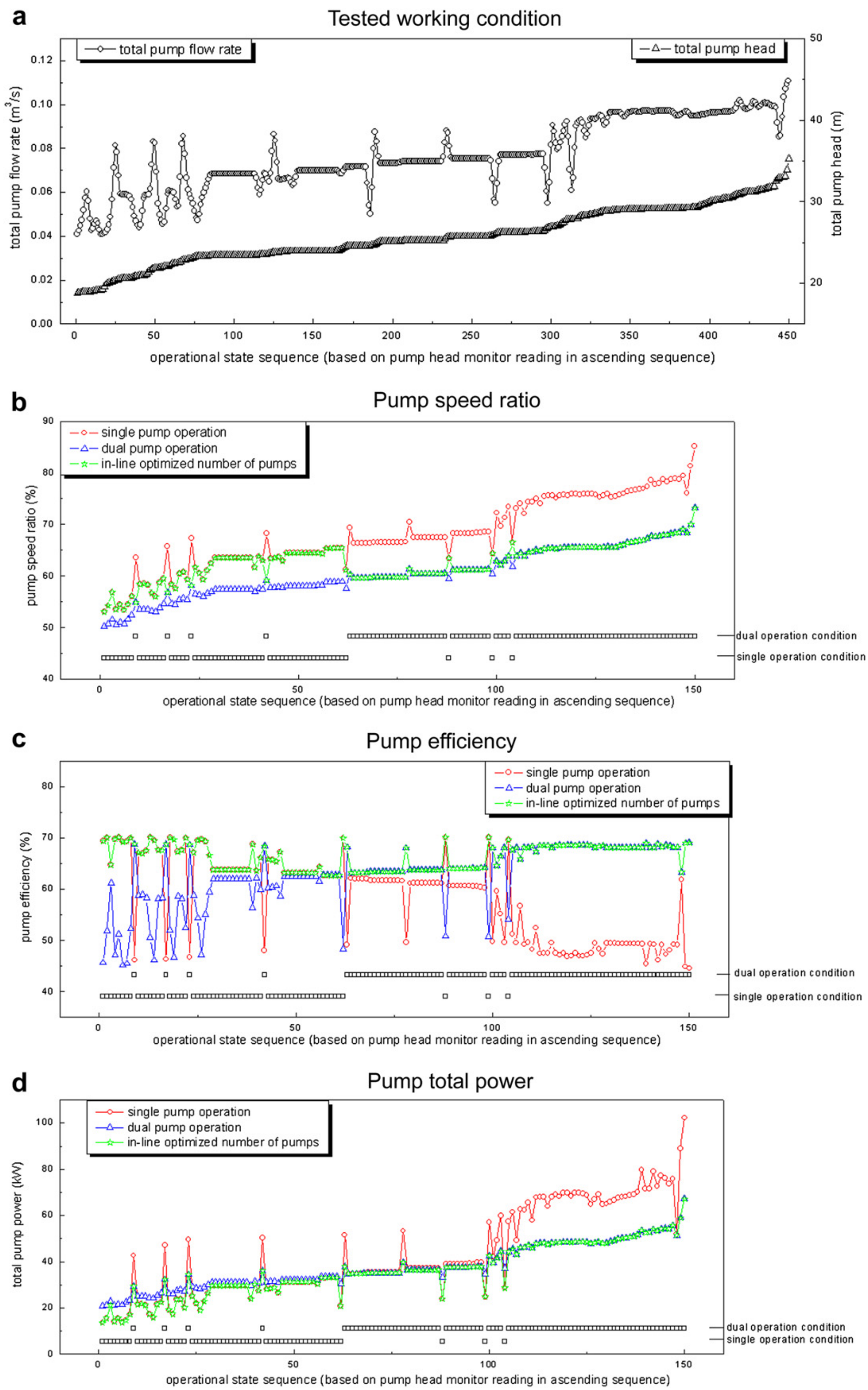

Fig. 7. Application results for the on-line number optimization method for parallel variable-frequency pump units. (a) Tested working condition; (b) pump speed ratio; (c) pump efficiency; and (d) pump total power

controller or control algorithm programming software, the analysis of the operating condition and control logic can be facilitated and used to decide the best parallel pump control plan.

For more than two sets (three sets are used as an example) in a parallel variable-frequency pump control system, the on-line allocation optimization method can be performed in accordance with the logic shown in Fig. 4 because three or more sets of the parallel pump system are similar.

6.2. Application

Here, we apply the optimal control method proposed in the present study to a variable-flow air-conditioning chilled-water system pump control in an industrial building in Suzhou, China. The schematic diagram for the chilled-water system in the factory air-conditioning is shown in Fig. 5. In Fig. 5, AHU1–AHU6 represent six air-conditioning units with variable air volume, V1–V6 represent the continuous control valves of each corresponding air-conditioning unit, VFD1 and VFD2 represent the variable-frequency drives of two chilled-water pumps, and DPT and F represent the pressure differential sensor and flow sensor, respectively.

The chilled-water-air-conditioning system uses a pump with a variable-flow control mode. The system has a total of three control loops. Each control feedback uses variable set point control mode.

(1) *Supply air control loop*: the system has a total of six air-conditioning units with variable air volume, and the air flow control is achieved by adjusting the supply fan speed. The controller is based on the supply of air static pressure and its set point deviation, according to the corresponding control algorithm output and the supply fan speed signal. The set point of the supply air static pressure changes based on real-time adjustments at room temperature.
(2) *Supply air temperature control loop*: the controller compares the air temperature and its set point, in accordance with the corresponding control, to calculate the control signals of continuous regulated valve and regulates the chilled-water flow in the cooling coil to adapt to changes in working conditions due to changes in customer load (reflected in changes in air volume) and to maintain the supply air temperature at its set point. With changes in the load of users, the control valve for each branch can be adjusted at any time, thereby affecting the impedance of the water system.
(3) *Pressure-control loop*: the pressure-control feedback acts to overcome the effect of system resistance changes in the reference pressure. By comparing the monitored pressure values and its set point, the controller based on the control algorithm outputs the corresponding control signal to the variable-frequency drive. The variable-frequency drive performs the signal control and the conversion of frequency, and it regulates the speed of the chilled-water pump to adapt to system resistance changes due to end-valve regulation and to maintain the pressure at its set point, as shown in Fig. 6. The algorithm for the adjustment of the pressure set point is described in [18].

Table 2
Average pump performance parameters.

Scheme	Parameter		
	Variable-speed ratio (%)	Efficiency (%)	Power (kW)
Single-pump operating scheme	67.9	58.8	43.0
Dual-pump operating scheme	60.6	62.9	37.7
On-line optimization scheme	62.7	66.4	36.2

This paper focuses on the application of a parallel variable-frequency pump on-line control method in a pressure-control loop. The corresponding values of a_1, a_2 and a_3 in Eq. (1) are -1308.19, -50.9 and 78.61, respectively, and those of b_1, b_2 and b_3 in Eq. (2) are -124.6,21.2 and -0.2, respectively.

The on-line control method proposed in this paper can be implemented through programmable controllers. It also runs programs with a single-pump or a dual-pump operation plan and compares the applications of three categories, in which the dual-pump operation plan adopts a synchronous frequency-control plan.

The user loads for six consecutive days, i.e., August 21, 2010 to August 26, 2010, were selected as an example to illustrate the effectiveness of the presented control method. Single-pump operation corresponded to August 21 and 22, the on-line control method proposed in this study corresponded to August 23 and 24, and the application of dual-pump simultaneous variable-frequency operation corresponded to August 25 and 26.

The pump daily working duration was 15 h, with 12-min sampling cycles, and there were 150 sets of working conditions for each group control scheme monitored. The working parameters include the chilled-water pump head and the chilled-water pipe flow. In the working condition of each group, the pump head also meets the current system requirements of flow and pressure control. Fig. 7(a) shows a comparative test for the control group, with a total of 450 sets of monitoring conditions for the chilled-water pump head and flow of chilled water.

Fig. 7(b)–(d) shows three methods corresponding to the ratio of pump speed, pump efficiency, and total pump power.

The application results show that:

(1) The on-line optimization method can ensure the highest efficiency of the water pumps and the lowest total power consumption for each test condition. It has substantially increased energy efficiency, as shown in Table 2.
(2) Generally, the optimal number of units depends on the total flow of the current system. Fig. 7(b)–(d) gives a comparison. Because of the larger flow of working condition nos. 9, 17, 23 and 42, to ensure that each pump is in its best condition, and to obtain the ideal work efficiency, the on-line optimization method adjusted the number of pump sets to two. Working condition nos. 88, 99 and 104 had similar conditions.
(3) The on-line optimization method reduced the energy consumption by 15.8% and 4% in dual-pump and single-pump operation, respectively, and the average pump operating efficiency was the highest among the three plans.

7. Conclusion

(1) Under the same hydraulic conditions, for the same type of parallel variable-frequency pump operating with the same speed ratio, the best overall energy efficiency is sought.
(2) This study proposed an optimal control method for parallel variable-frequency pumps with variable-frequency pump characteristics in the model through continuous monitoring of the parallel variable-frequency pump system total flow, with the pump head, the total flow, and the efficiency of a single pump divided into three regions. Through judgment of the current working conditions and zoning, the optimal number of operating pumps can be determined to meet the system load requirements and pressure settings.
(3) The method proposed in the present study was deduced theoretically and is universal and easy to implement on-line. It

provides a reference method for the optimal control of variable-frequency pumps with HVAC systems.

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References

[1] Yi Jiang. Building energy consumption in China and effective energy-saving method. HVAC 2005;35(5):30–40 (in Chinese).
[2] Kaya Durmus, Alptekin Yagmur E, Suleyman Yigit K, et al. Energy efficiency in pumps. Energy Convers Manage 2008;49(22–23):1662–73.
[3] Tao Chen, Qiao Wang Hai, Chun Zhang Deng. The analysis of air conditioning variable flow water system and variable-frequency pump. Xiangtan Teacher Univers (Sci Ed) 2008;30(1):37–9 (in Chinese).
[4] Shiels Stan. Centrifugal pump academy: the risk of parallel operation. World Pumps 1997;(1):34–7.
[5] Szychta Leszek. System for optimising pump station control. World Pumps 2004;2:45–8.
[6] Szychta Leszek. System for optimising pump station control – part II. World Pumps 2004;(7):32–4.
[7] Ke Hu Si, Tai Lu, Ru Bo Jing. The analysis of energy consumption of open systems parallel pump non-synchronous operating speed. Fluid Machin 2004; 32(3):28–31 (in Chinese).
[8] Ying Xu Guang. Simulation of parallel variable-frequency pump performance. HVAC 2007;37(3):49–53 (in Chinese).
[9] Wei Lu, Liang Chen Hong. The control of multiple parallel variable-frequency pumps. Electr Power Autom Equip 2005;25(4):89–91 (in Chinese).
[10] Zheng Fu Yong. The incremental flow of parallel pumps operation and related problems analysis. HVAC 2004;34(11):83–5 (in Chinese).
[11] Shiels Stan. When two pumps are cheapter than one. World Pumps 1997;(9):58–61.
[12] Ma ZJ, Wang SW. Energy efficient control of variable speed pumps in complex building central air-conditioning systems. Energy Build 2009;41: 197–205.
[13] Ahonen T, Tamminen J, Ahola J, et al. Estimation of pump operation state with model-based methods. Energy Convers Manage 2010;51:1319–25.
[14] Yi Wang, Liang Cao Shu. The optimization of genetic algorithms running in parallel pump system. Fluid Machin 2003;31(10):22–5 (in Chinese).
[15] Barán B, von Lücken C, Sotelo A. Multi-objective pump scheduling optimisation using evolutionary strategies. Adv Eng Softw 2005;36(1):39–47.
[16] Wang SW, Burnett J. Online adaptive control for optimizing variable-speed pumps of indirect water-cooled chilling systems. Appl Therm Eng 2001;21: 1083–103.
[17] Bernier MA, Bourret B. Pumping energy and variable frequency drives. ASHRAE J 1999;41(12):37–40.
[18] Zhao TY, Zhang JL, Chen YP, Lu Z, Ma LD, Sun DX. A novel differential pressure optimal reset strategy and its application in VWF system. In: Fifth international workshop on energy and environment of residential buildings and the third international conference on built environment and public health, Hunan University, Guilin; 2009.