DOI 10.1007/s10749-023-01516-w
Power Technology and Engineering *Vol. 56, No. 3, September, 2022*

IMPROVEMENT OF THE PROCEDURES FOR ASSESSING THE ENERGY EFFICIENCY OF PUMPING SYSTEMS

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Translated from *Gidrotekhnicheskoe Stroitel'svo*, No. 3, March 2022, pp. 26 – 38. DOI: 10.34831/EP.2022.52.79.005

The authors put forth a procedure for evaluating the energy efficiency of pumping equipment powered by electricity and operating under variable load, as well as a procedure created according to the guidelines of the European Pump Manufacturers Association (ENPA) based on the energy efficiency index (EEI). The article presents the results of analysis of energy efficiency values calculated based on different methods and the inherent disadvantages of the mathematical model used to determine the EEI index.

Keywords: energy efficiency; pumping systems; energy saving; mathematical model; pump-drive-network system; efficiency gradient; energy efficiency indices; EEI; PEI.

Processes in systems of water supply, water disposal, heat supply, oil and gas supply, etc. are in most cases maintained by bladed equipment such as pumps, compressors, fans, and smoke exhausters. The fraction of the electric power consumed by such equipment in the majority of economies is more than 20% of the total power consumption.

The main mechanism consuming power in pumping systems is the pump unit. Therefore, experts in the field of design and operation of pumping systems has recently focused on increasing the energy efficiency of commercial pumping equipment.

The EU and the USA have long resolved energy efficiency issues legislatively. In 2005, the European Commission and U.S. Department of Energy enacted guidelines and regulations on the energy efficiency of bladed equipment of various types and sizes. The most significant of them are regulation No. 547/2012 [1] of the European Commission, EN16480 [2], and the PEI Standard [3] of the U.S. Department of Energy.

However, experience suggests that installing a highly efficient pump does not guarantee its effective operation in a piping system. According to five leading pump manufacturers in the USA, more than 60% of the sold pumps have been operated out of their operating range, which is, in 95% of cases, due to consumers' providing incorrect data for pumps required.

Thus, high efficiency of pumping systems depends on two factors: the efficiency of pumps and the correct selection of their characteristics to fit pumping systems in which they will operate.

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Because of the complexity of the processes in hydrodynamic pump-piping-consumer systems, the current technical literature and regulations do not provide theoretically grounded and experimentally validated procedures for assessing the energy efficiency of pumping systems equipped with a variable-speed drive and operating under a variable load. Due to the limited regulatory and legal framework, the inspection of pumping systems is in most cases spontaneous, chaotic. In this connection, the European standards on assessment of the energy efficiency of pumps and pumping systems enacted in 2016 [2, 4] and the amended and revised version, as well as GOST 33970–2016 and GOST 33969–2016 Russia State Standards are supposed to play an important role in providing legal, technical, and methodical support.

It is of considerable theoretical and applied interest to consider standards on the energy efficiency of pumps and pumping systems [2 – 7].

The standard on assessing the energy efficiency of pumping systems offers three levels of tests. Of principal interest is the third level recommended for pumping systems operating under a variable load. The tests of the third level necessarily involve recording a process histogram. The standard imposes no requirements on its quality. However, a number of publications, including European regulations on environmental design of pump water-supply systems and determination of the energy efficiency index (EEI) [8, 9] recommend using a standardized histogram for a statistical sample of four intervals.

The number of statistical intervals of any nonstationary random process is known to represent the behavior of this process, not to be very small to adequately describe the real physical process, and not to be excessively large to react to possible fluctuations and excesses.

Experience of drawing histograms for real water-supply systems suggests that decreasing the number of intervals by enlarging them leads to considerable gradients of the parameters of interest (head, flow rate, efficiency) within each statistical interval. This makes the water supply model inadequate to the real physical process and leads to insufficiently substantiated conclusions and recommendations.

The standards on the energy efficiency of pumping systems [4 – 7] do not give any theoretically grounded and experimentally validated procedure for assessing the energy efficiency of pumping systems equipped with a variable-speed drive (VSD) and operating under a variable load. The standard only recommends documenting the used method, data source, and the formulas and methods used to draw conclusions.

It is also worth mentioning two important characteristics of the system: head-flow curve of the piping system and load profile. The standard imposes too soft requirements onto the input data for drawing these characteristics. For example, the standard allows using only two points of the head-flow curve (one corresponding to zero flow rate; in complex cases, only one point, i.e., operating point, can be used) to draw the characteristic of a piping system. To draw the load profile, the standard allows consulting, whenever possible, with the personnel of the station on approximate running hours per year, season, week, and day for different flow rates.

The two major characteristics of a pumping system mentioned above are fundamental for an objective assessment of the energy efficiency of the system. They radically affect the end results, conclusions, and decisions. This circumstance requires the standard to impose stricter requirements on the reliability and quality of experimental and other data.

Along with the general recommendations on increasing the energy efficiency of pumping systems by decreasing the friction pressure loss in pipelines and local hydraulic resistance, the standard should provide recommendations on improving complex piping systems through horizontal and vertical zoning.

Since the pressure in the discharge header of a pumping station depends on the required pressure at the critical control point of the network, it is necessary to experimentally establish the correlation between the pressure at the critical control point and the header of the pumping station. First, this allows comparing this pressure with the minimum admissible pressure at the critical control point defined by the appropriate regulations. If the pressure appears excessive (higher than minimum), then it can be reduced to the necessary level in the discharge header of the pumping station. Second, if the pressure in the discharge header is determined by the necessity of maintaining its value at the critical control point, which is a separate building (structure) or a small group of buildings, it would be well to install in them booster pumps with characteristics determined experimentally. This will allow transferring the critical control point to a lower piezometric level. As a result, conditions can be created for decreasing the pressure in the discharge header of the pumping station feeding a whole region or a city and, hence, considerably increasing the energy efficiency of the pumping system.

Moreover, the standard does not give recommendations on reducing the energy consumption by pumping systems when covering quite wide ranges of flow rate by connecting pumps in parallel. Pumps operating in parallel usually have identical characteristics. As our studies show, this solution is energetically justified only when the dynamic component of the required head is insignificant, i.e., when the hydraulic exponent $(N_{st}/N_w) > 0.8$ [10]. If this condition is not satisfied, it is energetically reasonable to connect pumps with different characteristics (head and flow rate) in parallel. Experience suggests that using different types of pumps with VSD allows avoiding jumps of flow rate and power upon putting different number of pumps into parallel operation. Although the control of the pumps is more complicated in this case, the maximum energy efficiency of the system is achieved by maintaining high efficiency over the whole load range, which is impossible with pumps of the same type.

Summing up the consideration of the European standards on the minimum efficiency index (MEI) of a pump and on the energy efficiency of pumping systems, it should be emphasized that, despite the wide use of variable-speed drives, these standards do not change in any way the conventional procedure for selecting the characteristics of pumping equipment.

The absence of a reliable theoretically grounded procedure complicates the objective assessment of the energy efficiency of pumping systems and, therefore, reduces the investment attractiveness of energy-saving projects. Therefore, for feasibility study of projects on modification of pumping systems, we have developed a new procedure for assessing, with an accuracy sufficient for practical purposes, the energy efficiency of designed pumping systems and for selecting energy-efficient pumping equipment and optimal control of pumps.

The main performance characteristics of a vane pump in a nonstationary process change significantly with time. Since in most processes supported by vane pumps, the load (flow rate) changes slowly, the process can be partitioned into statistical time intervals within which the flow rate, head, and efficiency can be considered constant with an adequate accuracy. This allows determining the energy consumed by a pump over any period, say, a year:

$$S_w = \sum_{j=1}^{m} \frac{\rho g Q_j H_j}{1000 \eta_{jp}; \eta_{jm}; \eta_{jd}} P_j T, \tag{1}$$

where ρ is the density of water, kg/m^3; g is the acceleration of gravity, m/sec^2; Q_j is the flow rate of the pump in the jth interval, m^3/sec; H is the pump head in the jth interval, m; η_{jp}, η_{jm}, η_{jd} are the efficiencies of the pump, electric motor, and drive in the jth interval; P_j is the probability of flow rate Q_j in a year; T is the time of operation of the pump during a year, h; m is the number of terms in the statistical series.

For a deeper insight into our procedure for assessing the efficiency of pumping systems, it is advisable to compare it with the conventional procedure for selecting pumping equipment that has widely been used in the last decades.

The input data in the conventional procedure are the coordinates of the operating point, i.e., the required head and flow rate of the pump. The EN16480 standard indicates that the characteristics of a pump at the operating point are determined by the customer, while the characteristics of pumping equipment are selected by its manufacturer. Moreover, data on static head, type, viscosity, and temperature of fluid have been acquired in recent years.

A pump is selected using special software and enumerating the characteristics of pump units whose flow rate and head in the optimal operating mode coincides or somewhat exceeds the characteristics of the pump at the operating point defined by the customer. Therefore, the main criterion in the conventional procedure used to select pumps is the condition $Q_{opt} = Q_{max} = Q_{opp}$, i.e., the characteristics (Q_{opt} and H_{opt}) of the selected pump are bound to those of the system using the only operating point compatible with the system.

This approach to the selection of pumps is justified for vane pumps operating at a constant speed (n_t = const) and a constant or slightly changing flow rate. However, the conventional procedure was extrapolated, without any justification, to pumps equipped with VSD and operating under a variable load.

The flow rate of a pump at the operating point is not usually equal to the maximum flow rate of the system. The flow rate at the operating point exceeds the maximum flow rate of the system mainly because of the limitation of the range of pumps manufactured; therefore, at the design stage, pumps are usually selected so that their head and flow rate are somewhat higher than those required. Moreover, in designing pumping systems, the major task is to ensure their reliability rather than to achieve high efficiency.

The regulations on design of pumping systems do not provide for instrumental determination of their energy efficiency during commissioning. This does not stimulate designers to improve the quality of projects, which in many cases leads to overdesign and selection of oversize pumps.

However, documents on selection of pumps from catalogs of some leading manufacturers (Flygt, Grundfos, KSB, etc.) recommend choosing the optimal flow rate of the chosen pump a little to the left of the operating point on its head-flow curve in the case of variable-load operation. These recommendations based on long-term observations of the energy efficiency of pumps are general and have not been brought to engineering calculations.

Our studies [11, 12] show that the necessary match of the characteristics of pumps and the system can be achieved if the procedure for selection of pumps equipped with VSD and operating under a variable load accounts for the influence of the following factors on energy efficiency: range of flow rates of the system and distribution of their probabilities within the range, position of the optimal flow rate of the pump relative to the statistical distribution of flow rates, and position of the maximum flow rate of the system relative to the flow rate of the chosen pump at the operating point.

Let us consider the procedure for selection of pumping equipment and determination of its energy efficiency for different control methods using, as an example, the mathematical modeling of the pump unit-pipeline network system.

The range of flow rates of the system and the distribution of their probabilities are presented in Table 1. The distributions of probabilities in the Table are based on real data of inspection of the water pumping station WPS-2 (Balashikha, Moscow oblast').

The static head of the piping system H_{st} = 20 m; the hydraulic exponent of the system $\alpha = 20/43.7 = 0.45$; the hydraulic resistance coefficient $\beta = 6.596 \times 10^{-5}$ [h^2/m^5]. The pump characteristics at the operating point: Q_{opp} = 648 m^3/h; H_{opp} = 47.7 m (at Q = 600 m^3/h).

An analysis of the energy functional that defines the power consumption by the pump unit (1) shows that the flow

TABLE 1. Statistical Distribution of Pump Flow Rates

Number j of statistical interval	Flow rate Q, m^3/h	Probability P_j	Number j of statistical interval	Flow rate Q, m^3/h	Probability P_j
1	140	0.0075	13	380	0.085
2	160	0.0075	14	400	0.06
3	180	0.01	15	420	0.05
4	200	0.01	16	440	0.05
5	220	0.03	17	460	0.035
6	240	0.055	18	480	0.035
7	260	0.055	19	500	0.06
8	280	0.055	20	520	0.06
9	300	0.035	21	540	0.06
10	320	0.035	22	560	0.03
11	340	0.06	23	580	0.02
12	360	0.06	24	600	0.01

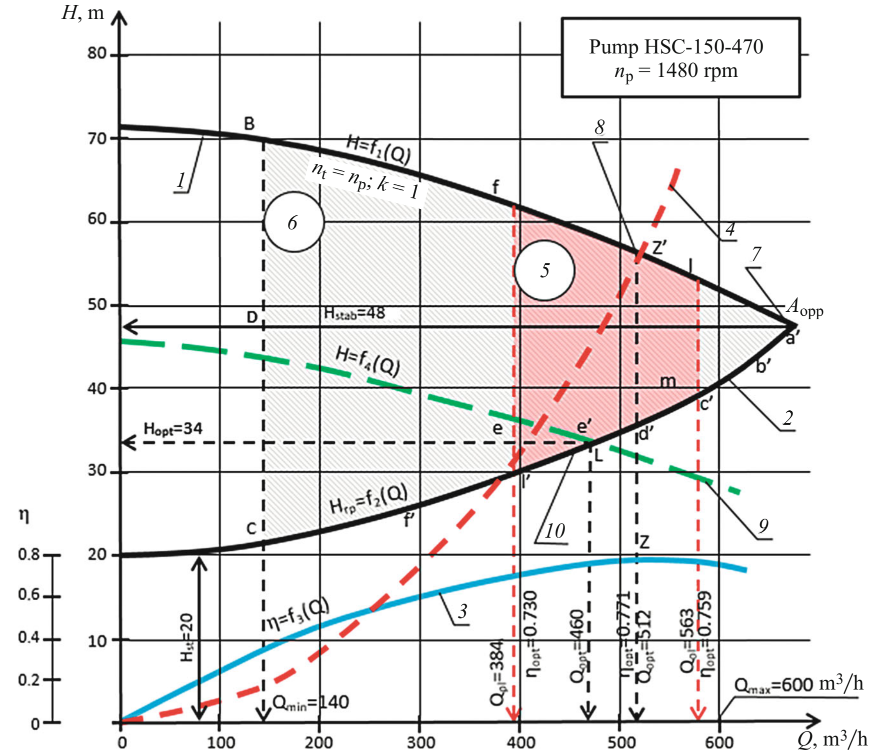

Fig. 1. Behavior of the operating point of the pump and the area of possible operating modes for different procedures of selecting characteristics of pumps and methods of controlling them using a variable-speed drive: *1*, head-flow curve of the pump (at $n_t = n_p$); *2*, head-flow curve of the piping system; *3*, efficiency curve of the pump (at $n_t = n_p$); *4*, curve (OZ') of similar modes of maximum efficiency; *5*, area (eflm) of effective operating modes of the pump according to the Standard EN 16480; *6*, area ($A_{opp}BC$) of possible operating modes of the pump; *7*, operating point A_{opp} of the pump at rated speed of the impeller (Q_{opp} = 648 m³/h; N_{opp} = 47.7 m); *8*, point (Z') corresponding to the optimal characteristics of the base pump; *9*, head-flow curve of the virtual pump; *10*, point (L) corresponding to the optimal characteristics of the virtual pump.

rate Q_j in water supply systems is specified by the customer and is an uncontrollable parameter.

To exclude the effect of the efficiencies of the electric motor and the drive on power consumption, their values were set equal to 1. Since the specific weight of the fluid can be assumed constant (γ = const), the energy efficiency of the pump is fully determined by the ratio H_j/η_j.

Therefore, the energy functional can be minimized if the redundant heads are eliminated in the whole range of loads, which represents the method of control through the minimization of redundant heads (proportional control), the efficiency being maximum in the whole load range.

While the former condition can be satisfied by decreasing the speed of the impeller, it is impossible to maintain the maximum efficiency over the whole range of loads because a decrease in the impeller speed automatically leads to a decrease in the efficiency of the pump.

When the pump operates at a constant (rated) speed np, its efficiency can uniquely be determined from the flow rate Q, which is an uncontrollable parameter and specified by the customer. If a variable-speed drive is used, then the efficiency depends on the trajectory of the operating point on the $H-Q$ plane and is, generally, a function of two parameters: $\eta = f(Q, H)$. For a specific flow rate in a certain statistical interval, the efficiency only depends on the head set by the control system, and its value is determined by the deviation of the current speed from the rated value, i.e., from the ratio of change in speed Kt ($Kt = n_t/n_p$). The power consumed by the pump is affected by not only the deviation ($\Delta\eta$) of the actual efficiency from its maximum value in the jth interval of flow rates, but also the time of operation with this deviation. Therefore, the minimum statistical expectation of the deviation of the actual efficiency from its maximum was considered a criterion of maximum efficiency under a variable load:

$$M(\Delta\eta_j) = \sum_{i=1}^{m} \Delta\mu_j P_j = \sum_{i=1}^{n} (\eta_{\max} - \eta_j) P_j \to \min. \quad (2)$$

To analyze the piping system, we use an HSC-150-470 pump (n_p = 1480 rpm) with optimal flow rate Q_{opt} = = 512 m³/h that is located in the range of flow rates defined by the system (Fig. 1).

It can be seen from Fig. 1 that the head-flow curve (curve *1*) of the pump crosses that (curve *2*) of the piping system at the operating point A_{opp} (point *7*) with coordinates Q_{opp} = = 648 m³/h and N_{opp} = 47.7 m. The figure shows the efficiency curve $\eta = f_3(Q)$ (curve *3*) with peak at the point Z for the rated speed ($n_t = n_p$) and the optimal flow rate Q_{opt} =

= 512 m^3/h. Also, Fig. 1 shows the range of possible flow rates of the pump from $Q_{min} = 140$ m^3/h to $Q_{max} = 600$ m^3/h and the curve of similar modes (curve *4*) corresponding to the maximum efficiency with peak at the point Z'.

It can be seen from Fig. 1 that the VSD allows considerably expanding (compared with throttling) the area of possible operating modes of the pump which is a curvilinear triangle of $A_{opp}BC$ (shaded area *6*) bounded by the head-flow curve $H = f_1(Q)$ of the pump from above, by the head-flow curve $H_{pl} = f_2(Q)$ of the piping system (curve $A_{opp}C$) from below, and by the vertical interval BC parallel to the ordinate axis from left. The trajectory of the operating point within the area $A_{opp}BC$ is defined by the chosen control method.

The operating point moves along the head-flow curve (curve $A_{opp}B$) of the pump in the case of throttling of the piping system and along the straight line $A_{opp}D$ (at $H_{stab} = 48$ m) or another line in the case of pressure stabilization, depending on the chosen stabilization pressure.

In the case of minimization of redundant heads, the operating point moves along the curve $A_{opp}C$ of the piping system (Fig. 1).

The areas of effective operating modes of the pump according to the EN 16480 standard are imposed on the areas of possible operating modes (eflm area *5*). Figure 1 shows that the area of effective operating modes of the pump is only a part of the area of possible operating modes and is in the flow-rate range from $Q_{pl} = 384$ m^3/h (underload) to Q_{ol} = = 563 m^3/h (overload), the optimal flow rate being Q_{opt} = = 512 m^3/h ($\eta_{opt} = 0.771$).

The mathematical model of the pumping system and the LAB-MZ software we have developed based on this model allow us to accomplish the following tasks:

— assessment of the energy efficiency of the installed pumping equipment (base case) or pumps planned to be installed and chosen control method;

— determination of the theoretical minimum of energy consumption (minimum of the energy functional (1)), using the minimum objective energy-consumption function as the base (reference) value of the maximum energy efficiency of the pump operating under a variable load. The objective function will become minimum if the head at the critical control point of the network (or the correlated head in the header of the pumping station) is minimum permissible and the deviations of efficiency from the maximum are zero over the whole range of loads;

— calculation of the available energy saving potential as the difference of power consumptions for the chosen control method and the theoretical minimum of possible power consumptions;

— determination of the energy efficiency of the pumping system that can be achieved by changing the control method rather than replacing the pump (for example, stabilization of pressure or minimization of redundant heads instead of throttling, proportional control);

— selection of the characteristics of the virtual pump that fit best the characteristics of the system;

— assessment of the efficiency of various procedures of selecting pumping equipment and associated control methods or other energy saving measures according to the criterion of maximum use of the energy saving potential;

— analysis of the performance of the pump in the whole range of loads; identification of the portions of the range of flow rates where the operation is inefficient; selection of an additional pump to cover this portion to ensure high performance of the pumping system as a whole.

Since we have introduced the new concept of virtual pump, we have to discuss it in detail.

As indicated above, the energy functional (1) becomes minimum when the redundant and excess heads are zero over the whole range of loads, which is possible if the operating point moves along the $A_{opp}C$ curve of the piping system (Fig. 1) and the mathematical expectation of the deviation of the actual efficiency from the maximum is minimum according to criterion (2). To this end, the peak of the efficiency curve (point Z, Fig. 1) becomes “floating,” i.e., moving horizontally, depending on the flow rate, maintaining its value ($\eta_t = \eta_{opt}$). The point Z' simultaneously moves along the head-flow curve (curve *1*, Fig. 1) and, therefore, the curve of similar modes of maximum efficiency (curve *4*, Fig. 1) takes different positions relative to the head-flow curve of the piping system and the range of statistical distribution of flow rates, tending to meet the optimization criterion (2). The head of the virtual pump is determined from the optimal flow rate of the virtual pump using the curve $H_{pl1} = f_2(Q_t)$ of the piping system.

Figure 1 shows the point L at which the flow rate and head correspond to the optimal parameters of the virtual pump Qvirtopt and Hvirtopt (curve *10*). The speed ns of a vane pump is known to have a very strong effect on its head-flow curve (steepness, presence of dropping left branch, etc.). The speed of the virtual pump is determined from the calculated flow rate and head in the optimal mode. The calculated values of Q_{opt}, H_{opt}, and n_s of the virtual pump and the statistical database on Russian and foreign vane pumps (in the range $15 \leq n_s \leq 75$) are used to draw the head-flow curve of the virtual pump. Then a real pump with characteristics closest to those of the virtual pump is selected from catalogs of pumping equipment.

The head-flow curve (curve *9*) of the virtual pump is shown in Fig. 1. It can be seen that if a pump with parameters close to those of the virtual pump is used, the operating point moves from the point L with the coordinates Q_{opt}^{virt} = = 460 m^3/h and $H_{opt}^{virt} = 34$ m to the point A_{opp} (Fig. 1, point *7*). Flow rates higher than the optimal are achieved by increasing the speed of the impeller (within 15 – 20%) relative to the rated value and decreasing the flow rate below Q_{opt} by decreasing the speed.

Studies of the energy efficiency of pumps with VSD operating under a variable load show that the efficiency de-

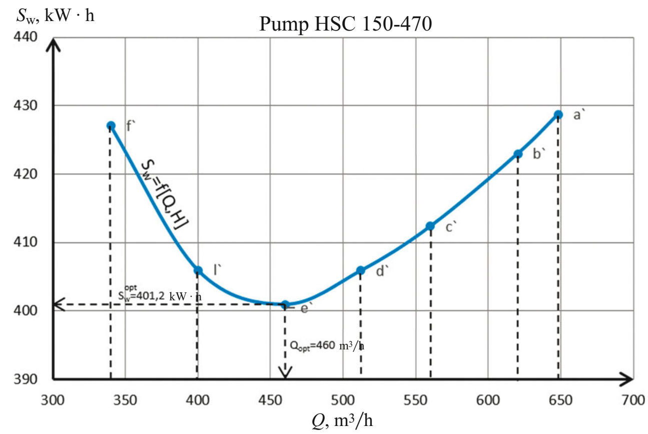

Fig. 2. Dependence of the energy functional on the flow rate of the pump to determine the optimal parameters of the virtual pump under a variable load.

pends on the coordinates Q and H in the area of possible operating modes and on the deviation of the current speed of the impeller from the rated value. The position of the operating point L within the range of flow rates leads to the fact that, as the flow rate decreases in the range from the A_{opp} to L, the efficiency of the pump first increases and then decreases in the range from L to C. This circumstance allows ensuring more effective functioning of the pump in the whole range of flow rates.

Moreover, the position of the operating point within the operating range of flow rates open_s access to those high values of efficiency which are inaccessible for the conventional approach. The issues of determining the optimal parameters of a virtual pump and drawing its curves are detailed in [11, 12].

For the mathematical model of a pumping system with HSC-150-470 pump, the parameters of the virtual pump are $Q_{opt}^{virt} = 460$ m³/h and $H_{opt}^{virt} = 34$ m. Thus, the flow rate of the chosen base pump in the optimal mode $Q_{opt} = 512$ m³/h, which is a little higher (by 11%) than the required flow rate of the virtual pump.

It is interesting to know how much the energy indicators of the base pump can worsen compared with those of the virtual pump because of the deviation of its flow rate in the optimal mode ($Q_{opt} \geq Q_{opt}^{virt}$). The LAB-MZ software permits calculating the power consumption by a virtual pump after manually entering the parameters of the virtual pump into the program. In the case of minimization of redundant heads, the operating point moves along the curve $A_{opp}C$ of the piping system (Fig. 1). We select points $a'b'c'd'e'l'f'$ on this curve, calculate their coordinates (flow rate Q and head H), and the corresponding values of the energy functional $S_w = f(Q_{opt})$ defined by (1). The calculated results are presented in Table 2.

TABLE 2. Dependence of the energy functional on flow rate and head for determining the parameter of the virtual pump

Point in Fig. 1	Pump parameters		Energy functional, S_w, ths. kW · h/year
	flow rate Q, m³/h	head H, m	
a'	648	47.7	428.8
b'	630	45.3	423.0
c'	560	40.7	412.4
d'	512	37.0	405.9
e'	460	34.0	401.2
l'	400	30.5	403.1
f'	340	27	427.1

The functional $S_w = f(Q)$ is visualized in Fig. 2 using the data of Table 2. It can be seen that the curve is extremal. At the point a' (corresponding to the operating point of the pump), the functional has the maximum value ($S_w = 428{,}800$ kW · h/year), which then smoothly decreases to the minimum value ($S_w = 401{,}200$ kW · h/year at $Q = 460$ m³/h) with decrease in the optimal flow rate of the pump. The minimum value of the functional found by the semigraphical method is consistent with that calculated by the optimizing module of the LAB-MZ software. Further, the energy functional increases considerably with decrease in the flow rate (Fig. 2).

It is of interest to compare the functionals calculated for the optimal flow rate of the virtual pump ($S_w = 401{,}200$ kW · h/year) and for the optimal flow rate of the base pump ($S_w = 405{,}900$ kW · h/year at $Q = 512$ m³/h). The comparison shows that the change in the flow rate from 460 to 512 m³/h slightly increases the power consumption.

These results suggest that the efficiency curve of the base HSC-150-470 pump was initially selected so as to be as close as possible to the parameters of the pumping system, similarly to the efficiency curve of the virtual pump.

In this connection, it is of interest to calculate and compare the energy efficiencies of the base pump for different control methods with the efficiency of this pump in the case where the optimal flow rate is transferred from the point L ($Q_{opt} = 512$ m³/h) to the operating point A_{opp} ($Q_{opp} = 648$ m³/h) as in the conventional procedure for selecting pumping equipment, i.e., $Q_{opt} = Q_{opp}$. The results of mathematical modeling of such a transfer are compared in Table 3. Table 3 summarizes energy indicators of the HSC-150-470 pump for two cases (Nos. 1 and 2) of the position of its optimal flow rate relative to the range of statistical distribution of flow rates.

In case No. 1, the optimal flow rate of the pump meets the optimization criterion (2), which minimizes the mathematical expectation of deviation_s of the current efficiency from its maximum value. By meeting the optimization criterion, we can establish the maximum correspondence between the characteristics of the pump and the network under a variable load.

In case No. 2, the characteristics of the pump are selected by the conventional procedure in which the optimal flow rate equals the flow rate at the operating point: $Q_{opt} = Q_{opp}$ (Fig. 1).

From Table 3 it can be seen that in case No. 2, the energy consumed by a pump selected by the conventional procedure increases for all control methods, namely: by 10.2% in the case of throttling of the piping system, by 7.5% in the case of stabilization of the pressure in the discharge header, and by 5.3% in the case of minimization of redundant heads.

The theoretical minima of the possible energy consumption in the two cases (minimum of the energy functional) remain almost equal: 387,800 and 388,200 kW · h/year. From Table 3 it can be seen that the efficiency of the pump is maximized by minimizing the redundant heads with preliminary optimization of the parameters of the pump by the procedure proposed. This leads to the maximum correspondence between the characteristics of the pump and the system, which is confirmed by 96.6% recovery of the energy saving potential, unlike all the other methods.

Since the European standard on the assessment of the energy efficiency of pumping systems does not give recommendation_s on and references to any procedure, it is of interest to consider the energy efficiency index (EEI) based on the concept introduced by European Pump Manufacturers Association (ENPA).

The energy efficiency criterion has been developed for the pump-motor-drive-network system. For different types of pumping systems, it is proposed to calculate the energy efficiency index taking into account the load profile and the control method:

$$EEI = P_{1,\text{avg}} \big/ P_{1,\text{ref}} \,, \tag{3}$$

where $P_{1,\text{avg}}$ is the power consumed by the pump for a certain load profile and a certain flow rate during certain periods.

The power consumption (3) is determined from the expression

$$P_{1,\text{avg}} = 0.06P_{1;100\%} + 0{,}15P_{1;75\%} + \\ + 0{,}35P_{1;50\%} + 0{,}44P_{1;25\%}, \tag{4}$$

where 0.06, 0.15, 0.35, 0.44 are the relative times of operation within statistical intervals (t_j/t), t is the total time of operation of the pump, t_j is the time of operation in the jth inter-

TABLE 3. Comparison of Performance Indicators of a Pump with Optimal Flow Rate Located in the Range of Loads and Shifted to the Operating Point

Parameters of compared pumps and control method	Unit of measurement	Pump HSC-150-470, n = 1480 rpm, D_2 = 426 mm. Compared cases: No. 1 — Position of Q_{opt} in the range of flow rates of statistical distribution	No. 2 — Shift of Q_{opt} to operating point beyond the range of flow rates	Comparison of energy efficiency indices for cases Nos. 1 and 2 (per year)
Parameters of pump in optimal mode:				
flow rate Q_{opt}	m³/h	512	648	–”–
head H_{opt}	m	56.1	47.7	–”–
efficiency η_{opt}	—	0.77	0.77	–”–
Parameters at operating point:				
flow rate Q_{opp}	m³/h	648	648	–”–
head H_{opp}	m	47.7	47.7	–”–
efficiency η_{opp}	—	0.77	0.77	–”–
Efficiency of pump in case No. 2 for Q = 512 m³/h	—	0.73	—	–”–
Electric power consumed by pump with chosen control method:	ths. kW · h/year	—	—	-
throttling of discharge pipeline	(%)	799.1	889.9	+90.8 (10.2%)
stabilization of pressure in discharge header		614.4	664.8	+49.9 (7.5%)
minimization of redundant heads in header of PS		405.9	428.8	+22.9 (5.3%)
minimization of redundant heads with preliminary optimization of pump characteristics (optimization)		401.6	401.6	0
Theoretical minimum of possible power consumption (minimum of energy functional)	ths. kW · h/year (%)	387.8	388.2	+0.4 (0.1%)
Energy saving potential	ths. kW · h/year	417.3	501.7	—
Energy saving for chosen control method:	ths. kW · h/year			
pressure stabilization		184.7	225.1	+40.4 (17.%)
minimization of redundant heads		393.2	461.1	+67.9 (14.7%)
optimization of pump characteristics		397.4	488.3	+90.8 (18.6%)
Efficiency of energy saving potential for chosen control method:				
pressure stabilization	%	44.9	44.9	0
minimization of redundant heads	%	95.6	91.9	–3.7 (3.9%)
optimization of pump characteristics	%	96.6	97.3	+0.6 (%)

val; $P_{1,\text{ref}}$ is the base (reference) power consumption defined in regulations and determined from the data of analysis of highly efficient electric motors taking into account the specific speed and flow rate of the pump:

$$P_{1,\text{ref}} = \frac{\rho g H_{100\%} Q_{100\%}}{\eta_{\text{pump}} \eta_{\text{mot}}}, \quad (5)$$

where ρ and g are the density of the fluid, kg/m^3, and the acceleration of gravity, m/sec^2; $H_{100\%}$ and $Q_{100\%}$ are the head and flow rate of the pump (selected by the conventional procedure: $H_{100\%} = H_{\text{opp}}$ and $Q_{100\%} = Q_{\text{opp}}$); P_{pump} and P_{mot} are the efficiency of the pump and the electric motor under 100% load.

It is of interest to compare the power consumption and the EEI using our procedure (for a greater number of statistical intervals, $j = 24$) and the ENPA concept (for enlarged intervals, $j = 4$). For comparison, the model of a pumping system with HSC-150-470 pump mentioned earlier was used. Since, as was shown earlier (Table 3), the position of the optimal flow rate Q_{opt} relative to the range of flow rates of the pump has a significant effect on its energy consumption, two cases were considered. The optimal flow rated was located within the range of flow rates ($Q_{\text{opt}} = 512$ m^3/h and $H_{\text{opt}} = 56.1$ m; $Q_{\text{opp}} = 648$ m^3/h and $H_{\text{opp}} = 47.7$ m) in case No. 1 (Table 4) and at the operating point ($Q_{\text{opt}} = Q_{\text{opp}} = 648$ m^3/h; $H_{\text{opp}} = H_{\text{opt}} = 47.7$ m) in case No. 2. The calculated energy consumption and associated EEI are summarized in Table 4.

From Table 4 it can be seen that decreasing the number of statistical intervals from 24 to 4 by enlarging them leads to an increase in the calculated power consumption and the EEI for both cases and all control methods. For example, if the optimal flow rate is in the range of flow rates (case No. 1), the power consumption increases by 2% in the case of throttling and by 14.7% in the case of the minimization of redundant heads, and the EEI increases by 2.1 and 12.6%, respectively. When the optimal flow rate at the operating point is beyond the range of flow rates of the system (case No. 2), the power consumption increases by 0% (in the case of throttling) and by 12.6% (in the case of minimization of redundant heads). The EEI increases if the statistical intervals are enlarged from 0 to 12.5%.

The main causes of the increase in the power consumption and EEI calculated as recommended by guidelines (EU) No. 547/2012 compared to those calculated by our procedure are the following:

TABLE 4. Power Consumed by the Pump and the EEI Depending on the Number of Statistical Intervals in the Mathematical Model for Different Control Methods and Cases of Optimal Parameters

No.	Control method, parameters of pump at operating point and in optimal mode	Energy efficiency of HSC-150-470 pump for different number of intervals of statistical distribution					
		Power consumption per year, ths. kW · h			Energy efficiency index EEI		
		for greater number of intervals ($j = 24$)	for enlarged number of intervals ($j = 4$)	difference of power consumptions	for greater number of intervals ($j = 24$)	for enlarged number of intervals ($j = 4$)	Comparison of EEIs
1	2	3	4	5	6	7	8
	Case No. 1 optimal flow rate located within the range of flow rates)						
	Pump parameters	$Q_{\text{opp}} = 648$ m^3/h, $H_{\text{opp}} = 47.7$ m, $Q_{\text{opt}} = 512$ m^3/h, $H = 56.1$ m					
	Control method:						
1	Throttling of piping system	799.1	815.7	+16.6 (2.0%)	0.900	0.919	+0.019 (2.1%)
2	Stabilization of pressure in discharge head	614.4	649.4	+35.5 (5.5%)	0.692	0.730	+0.038 (5.2%)
3	Minimization of redundant heads in discharge header	405.9	476.1	+70.2 (14.7%)	0.474	0.536	+0.062 (11.6%)
4	Minimization of redundant heads with preliminary optimization of pump parameters	401.6	459.2	+57.6 (12.5%)	0.452	0.517	+0.065 (12.6%)
5	Theoretical minimum of possible power consumption (minimum of energy functional)	378.8	438.1	+59.3(13.5%)	0.437	0.494	+0.057 (11.5%)
	Case No. 2 (optimal flow rate located outside the range of flow rates, at operating point)						
	Pump parameters	$Q_{\text{opp}} = Q_{\text{opt}} = 648$ m^3/h, $H_{\text{opp}} = H_{\text{opt}} = 47.7$ m					
	Control method						
1	Throttling of piping system	889.9	893.0	+3.1 (0.0%)	0.93	0.933	+0.003 (0.0%)
2	Stabilization of pressure in discharge head	664.8	689.4	+24.2 (3.51%)	0.695	0.721	+0.026 (3.6%)
3	Minimization of redundant heads in discharge header	428.8	476.6	+47.8 (10.6%)	0.448	0.498	+0.06 (12.5%)
4	Minimization of redundant heads with preliminary optimization of pump parameters	401.6	459.7	+58.1 (12.6%)	0.420	0.480	+0.06 (12.5%)
5	Theoretical minimum of possible power consumption (minimum of energy functional)	388.2	438.5	+50.3 (11.5%)	0.405	0.458	+0.053 (11.6%)

— despite the wide range of possible flow rates (from 0.25 to 1.0 Q_{max}), the number of intervals of statistical distribution of flow rates recommended by guidelines No. 547/2012 is excessively limited. The standardization of the load profile by representing it as a four-bin histogram in the procedure considerably simplifies the description of the process, yet cannot adequately describe the real physical processes in pumping systems;

— instrument-aided inspection and mathematical modeling of a number of pumping systems demonstrate considerable gradients of efficiency and head within each statistical interval of the histogram. For example, the efficiency gradient is 12 – 20% in the first quarter of the flow rate range, 7 – 12% in the second quarter, 2 – 7% in the third quarter, and 1 – 2% in the fourth quarter. Great gradients of efficiency and head lead to great gradients of the EEI calculated at the ends of almost every enlarged interval. This circumstance does not allow us to determine, with an accuracy sufficient for practical purposes (1 – 2%), the efficiency and head at any point of the range of flow rates. Therefore, the EEI calculated for the whole range of flow rates is an integral parameter, which cannot be used to study changes in such important parameters as efficiency and head within the range of flow rates;

— to calculate the base (reference) power consumption $P_{1,ref}$, use is made of head, flow rate, and efficiency corresponding to 100% load of pump (5). When selecting the characteristics of a pump by the conventional procedure, it is necessary to satisfy the following conditions: $Q_{opt} = Q_{opp}$ and $H_{opt} = H_{opp}$. Usually, these conditions are not met when the pump operates under a variable load. As shown above, the shift of the optimal flow rate and efficiency relative to the range of flow rates of the system leads to a significant change in the power consumption according to the load profile, the base (reference) power, and, hence, the EEI.

CONCLUSIONS

1. The standard on the assessment of the energy efficiency of pumping systems details the organizational part, from the creation of an expert group to conclusions, recommendations, and requirements to report preparation. The standard still offers the user to use the conventional procedure for selecting pumping equipment from the extreme (peak) load and extrapolates it, without any justification, to pumping systems equipped with a variable-frequency drive and operating under a variable load. Moreover, the standard does not recommend any theoretically grounded and experimentally tested procedure for objective assessment of the energy efficiency of pumping systems.

2. The most significant European document on the assessment of the energy efficiency of pumping systems is the procedure for assessing the energy efficiency of various pumping systems using the so-called energy efficiency index (EEI) developed by the European Pump Manufacturers Association. For analysis of the procedure of calculating the EEI, mathematical models and special software we had developed were used. The analysis was conducted by comparing the calculated power consumption and EEI for two compared methods. The results obtained allow us to reveal the following shortcomings of the procedure for determining the EEI.

The experience of inspecting operating pumping systems suggests that each of them is individual, with inherent range of flow rates and distribution of probabilities within it and an individual head-flow curve of the network and installed pumping equipment. Therefore, the profile standardization consisting of four enlarged statistical intervals used in the procedure of determining the EEI greatly simplifies the modeled process and, hence, cannot describe adequately the real physical processes in pumping systems. Since the resulting value of the EEI for pumping systems is formed from the head and efficiency calculated for each statistical interval, great gradients of these quantities within each interval reduce the accuracy of the EEI for the whole pumping system. Being an integral parameter determining the energy efficiency of a pumping system as a whole, the EEI does not allow us to study changes in head and efficiency within the system, which are important parameters forming the value of power consumption, and to determine their deviation from the design values for all possible operating modes of the pump. However, it is these parameters and their deviations that allow us to infer the causes of additional energy loss, to evaluate the energy saving potential, and to develop measures to recover it.

For the majority of pumping systems operating under a variable load, the position of the optimal flow rate does not coincide with the flow rate at the operating point, which is recommended by the conventional procedure for selection of pumps, i.e., the main pump selection condition $Q_{opt} = Q_{opp} = Q_{100\%}$ is not satisfied. Our studies show that the deviation of the position of the optimal flow rate of the pump from the flow rate of the system at the operating point leads to a change in both consumed power and base (reference) power and, hence, a change in the EEI calculated from them. However, the procedure for determination of the EEI neglects this factor. Moreover, this procedure cannot be applied to pumping systems consisting of two and more pumps connected in parallel.

3. Based on theoretical studies, mathematical simulation, and instrument-aided inspection of pumping systems, we have developed and experimentally tested the following procedure for assessing their energy efficiency:

— it was established that the base (reference) maximum energy efficiency needed to determine the energy saving potential of pumps for water supply systems is the theoretical minimum of the objective power-consumption function (minimum of the energy functional) that is achieved when the head at the critical control point of the water supply system or at the outlet of the pump is minimum permissible over the whole range of flow rates and the deviations of efficiency from its maximum are equal to zero. In this case, the energy saving potential is defined as the difference between the power consumption for the chosen control method and the minimum of the energy functional (minimum possible energy consumption);

— optimization (minimization) of energy consumption by pumping equipment operating under a variable load allows us to theoretically determine the most effective parameters of the pump, i.e., its flow rate, head, and speed in optimal mode. The characteristics of the most effective equipment are determined by matching the mathematical model of a virtual pump and the head-flow characteristics of the system and the statistical load distribution, rather than by the conventional procedure (enumeration of characteristics of commercial pumps). Then the calculated characteristics of the virtual pump are used to select real equipment with characteristics as close as possible to those of the virtual pump;

— in the recommended procedure for selecting the characteristics of a vane pump, its head and flow rate in the optimal mode are located between the minimum and maximum loads so that the energy loss due to the deviation of the actual efficiency from its maximum is minimum. With such choice of characteristics, the pump most of the time operates in the range of most probable flow rates with maximum or nearly maximum efficiency. In this case, the maximum (peak) load is served by briefly increasing the impeller speed relative the rating. Off-peak loads are compensated by reducing the load;

— the recommended procedure for selecting the optimal characteristics of a vane pump does not depend on design solutions and, thus, is almost free to select its characteristics. This makes it possible to objectively determine the qualities of a modern vane pump intended for operation under a variable load. The procedure allows us to see the near-term outlook for the development of fluid machinery when the market will demand energy-efficient pumps capable of:

— ensuring reliable and steady operation with controllable drive and speed changing from $0.6n_p$ to $1.3n_p$;

— effective operation in a range of flow rates exceeding the optimal flow rate by a factor of 1.3 to 1.5 (in the absence of motor capacity constraints and cavitation);

— having a high-quality hill (control) diagram with minimum deviations of efficiency from its maximum when the current speed deviates from the rating over a wide control range;

— objective assessment of any energy saving measures (replacement of pumping equipment or its control method, etc.) by introducing the usage of the energy saving potential as a criterion. A comparative analysis of the energy efficiencies of different control methods for vane pumps equipped with a controllable drive shows that the minimization of redundant heads with preliminary optimization of the characteristics of the pump is the most effective method. The recommended method allows energy saving from 35 to 60% compared with throttling of pipelines and the most complete (95 – 97%) use of the energy saving potential, which is not possible with any other methods.

The above procedure was experimentally tested and is successfully used at water- and heat-supply facilities of the Moscow and Vladimir regions.

REFERENCES

1. *Study of Commission Regulation (EU) No. 547/2012*.
2. *European Standard EN-16480. Minimum Required Efficiency of Rotodynamic Water Pumps* (2016).
3. *Energy Conservation Standards for Pumps*, USA (2015).
4. *European Standard Pump System Energy Assessment ISO/ACME 14414:2015*.
5. *European Standard Pump System Energy Assessment ISO/ACME 14414:2019* (amended and revised).
6. *State Standard GOST 33970–2016. Energy Efficiency. Centrifugal Water Pumps. Determination of Efficiency and Energy Efficiency Index* [in Russian].
7. *State Standard GOST 33916–2016. Energy Efficiency. Assessment of the Energy Efficiency of Pumping Systems* [in Russian].
8. B. Stoffel, *Presented at the EUROPUMP roundtable*, Brussels, October 6 (2011).
9. *Review Study Pumps Final Report European Commission*, December (2018).
10. V. G. Nikolaev, "Energy saving methods of controlling the operating modes of a group of parallel vane pumps," *Pribory Sist. Upravl. Kontr. Diagn.*, No. 6 (2008).
11. V. G. Nikolaev, "Analysis of the energy efficiency of various methods of control of pumping units with controllable drive," *Vodosnab. Sanit. Tekhn.*, No. 11, Part 2 (2006).
12. V. G. Nikolaev, "Influence of the method of control of a vane pump operating under a variable load on the determination of its optimal parameters," *Pribory Sist. Upravl. Kontr. Diagn.*, No. 5 (2008).