OPTIMIZING THE ENERGY CONSUMPTION OF AN INDUSTRIAL WATER-SUPPLY PUMP SYSTEM BY ENSURING AN OPTIMAL FLOW RATE

Prof. PhD Gencho POPOV[1], PhD Boris KOSTOV[1], PhD Miglena HRISTOVA[1], Asc. Prof. Donka IVANOVA[1], Asc. Prof. Anka KRUSTEVA[1]

[1] University of Ruse "Angel Kanchev", Bulgaria: gspopov@uni-ruse.bg; bkostov@uni-ruse.bg; mcankova@uni-ruse.bg; divanova@uni-ruse.bg; akrasteva@uni-ruse.bg

Abstract: *This work presents theoretical research about the possibilities for achieving at the same time a system's optimal flow and maximum energy efficiency, by using the frequency method of flow rate regulation, with a number of pumps working in parallel. Features, concerning the change of the specific energy consumption, used for the ensuring of different total flow rates in the system, are also indicated. An interesting conclusion is the fact that the minimal energy consumption for the investigated system can be achieved by the realization of different work regimes.*

Keywords*: pump systems, energy efficiency, energy consumption*

1. Introduction

Often, even if the existing requirements for the correct designing of a pump system are being observed, the system doesn't work effectively, which leads to unnecessary energy loses. One of the main reasons for this is the failure to use the full potential of the given system, especially in terms of the utilization of its input power. As a result, finding a way for achieving the optimization of a system's work performance, in terms of energy efficiency, has become a key priority. However, for various reasons, related mainly to the changing consumer needs, it is necessary for the system's flow rate to be increased or decreased, which automatically makes the initial selection of pumps inappropriate. It is well-known that to replace the pump aggregates with new and more effective pumps will be very expensive, so it is necessary to find an alternative solution for this problem.

2. Materials and methods

A scheme of the pump system

The focus of this research is an actual pump system, used for industrial water-supplying, has been selected. Its scheme is given in figure 1. In the designing of this system it was decided that the reservoir, whose volume is 4000 m^3, should be filled with water before the start of the working day, i.e. during the time of its filling there will be no water consumption. Because of the global economic crisis, which began a few years ago, the company which runs this pump system had to reduce their production cycle, reducing daily water consumption by 25%. As a result, a decision was made to exclude one of the existing drilling pumps (P_0), so that the water supply would be provided only by two pumps (P_1) and (P_2). Analogically, in the II^{nd} stage of the pump system only two of the existing three pumps are working – in the designing of this system it was decided that there will be 2 more pumps in reserve (the third pump and the two reserve pumps are not given in figure 1).

Therefore, the initial water supply will be ensured by pumps P_1 and P_2, which are the same type, but located at different heights and distances from each other. Their nominal work parameters are: flow rate - Q=50 l/s; head - H=250 m; induction motor's power - P_M=190 kW; speed of rotation– n=2880 min^{-1}. The main purpose of these two pumps is to ensure the water transportation from I^{st} to II^{nd} stage. After that the pumps P_3 and P_4: Q=69.5 l/s; H=192; P_M=250 kW and n=1500 min^{-1}, have to ensure the water transportation to the main reservoir. In the designing of the system's II^{nd} stage, it is planned to install a buffer tank, whose volume is 100 m^3. Its main purpose is to compensate for the different flow rates of the pumps in I[st] stage and the pumps in II[nd] stage.

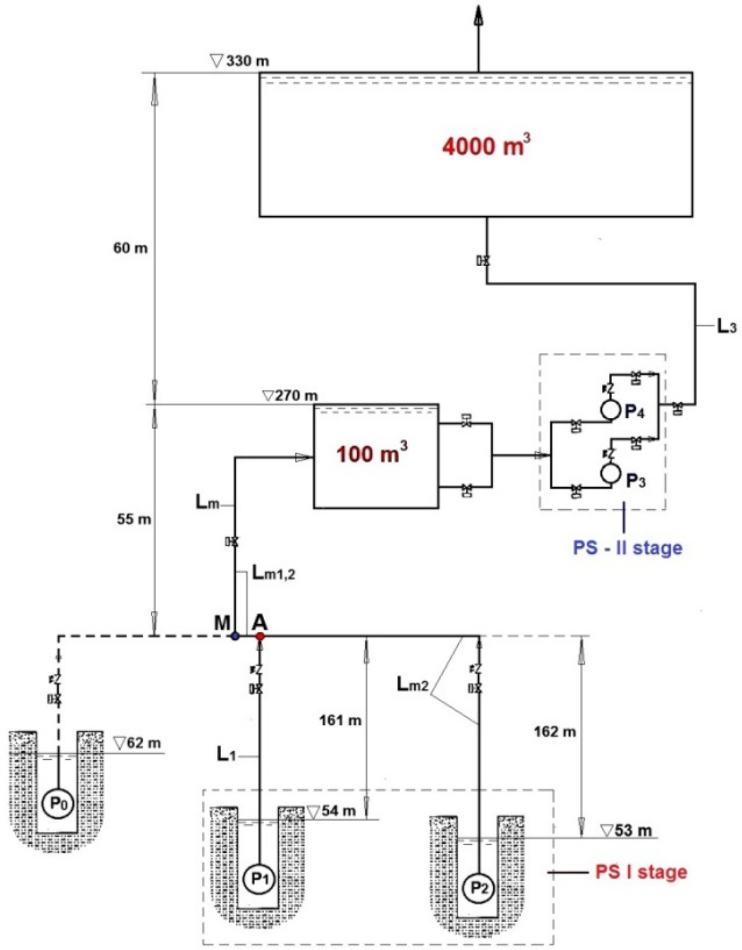

Figure 1 A scheme of the investigated pump system

In table 1 the data for the relative lengths and diameters of each of the pipes forming the pipe system are given.

TAB. 1 Lengths and diameters of the different pipes forming the pipe system

name	length, *m*	diameter, *mm*
L_1	162	273
L_{m2}	812	273
$L_{m1,2}$	2517	-
L_m	1842	325
L_3	1514	325

Structuring of the optimization model

The ensuring of an energy-effective system's work can be possible if the system is divided into two parts: **1** part – from the drilling pumps (P_1, P_2) to the buffer tank; **2** part – from the pumps P_3 and P_4 to the main reservoir). The determination of the flow rate $Q_{opt,1}$ (in **1**) will largely be predestinated and the determination of the flow rate $Q_{opt,2}$, (in **2**) ensured by the pumps P_3 and P_4 – as an additional restrictive condition the volume of the buffer tank and the time for its filling should be selected.

If the characteristics of the pumps and pipe system are known, the determination of the pump system's flow rate shouldn't be a problem. However, this doesn't mean that this will be the most appropriate value, in terms of energy efficiency, where the energy consumption to be minimal. But the optimal value of the flow rate can be achieved, when the two pumps P_1 and P_2 are working together (in parallel), because it is possible to regulate each of them by changing their speed of rotation. In this case, it will be interesting to find out what is the best proportion between the individual flow rates of each of the two pumps, remembering that the total system's flow rate is a sum of these two individual flow rates.

For finding a solution of the hydraulic part of this optimization problem it is necessary to determine (in advance) the work regime, when the pumps P_1 and P_2 work together with their nominal speed of rotations. This can be done by using some well-known hydraulic features (equations) and the method, described in [5]:

- determining the coefficient "k" of the pipe system by estimating the relative coefficient of friction "λ";

- by using equation (1) the (summary) head characteristic (referring to a common point of the pipe system – in this case, it is: p. A – fig. 1) of "n" number of pumps, located in different stations and positioned on different heights, can be found. Therefore, selecting different values of the head H_i, the relevant value of the flow rate Q_i can be estimated by (2):

$$H_i = (a_i - k_i)Q_i^2 + b_i Q_i + c_i - h_i; \tag{1}$$

$$Q_i = \frac{-b_i - \sqrt{b_i^2 - 4(a_i - k_i)(c_i - h_i - H_i)}}{2(a_i - k_i)}, \tag{2}$$

where a_i, b_i and c_i are the coefficients of the head characteristics respectively of a given pump, determined on the basis of the catalogues, provided by the manufacturer; h – the distance between the levels of the water in the exhaustive well and the beginning of the common pipe. For the selected (sample) three different values of the head (H', H'', H'''), taking places in the interval, belonging to the summary head characteristic, the relevant flow rates for each of the parallel working pumps can be estimated by (2) and summed, so to achieve the overall flow rates (Q', Q'', Q'''). This can be used for the determination of the coordinates (Q and H) of three points, belonging to the summary head characteristic. Finding a solution of (3) first and then using (4), (5) and (6) will ensure the determination of the coefficients (a_0, b_0, c_0) of the equation of the summary head characteristic:

$$\left| \begin{array}{l} H' = a_0(Q_1' + Q_2' + \ldots + Q_n')^2 + b_0(Q_1' + Q_2' + \ldots + Q_n') + c_0 \\ H'' = a_0(Q_1'' + Q_2'' + \ldots + Q'')^2 + b_0(Q_1'' + Q_2'' + \ldots + Q_n'') + c_0 \\ H''' = a_0(Q_1''' + Q_2''' + \ldots + Q_n''')^2 + b_0(Q_1''' + Q_2''' + \ldots + Q_n''') + c_0 \end{array} \right. ; \tag{3}$$

$$b_0 = \frac{(H'' - H''')(Q_{0,1}^2 - Q_{0,2}^2) - (H' - H'')(Q_{0,2}^2 - Q_{0,3}^2)}{(Q_{0,2} - Q_{0,3})(Q_{0,1}^2 - Q_{0,2}^2) - (Q_{0,1} - Q_{0,2})(Q_{0,2}^2 - Q_{0,3}^2)}; \tag{4}$$

$$a_0 = \frac{H' - H'' - b_0(Q_{0,1} - Q_{0,2})}{Q_{0,1}^2 - Q_{0,2}^2}; \tag{5}$$

$$c_0 = H''' - a_0 Q_{0,3}^2 - b_0 Q_{0,3}, \tag{6}$$

where $Q_{01}=Q_1'+Q_2'+\ldots+Q_n'$; $Q_{02}=Q_1''+Q_2''+\ldots+Q_n''$; $Q_{03}=Q_1'''+Q_2'''+\ldots+Q_n'''$ (1, 2...n – parallel working pumps in the system).

Fig. 2 presents graphically the determination of the system's work regime in case the pumps P_1 and P_2 work together on their nominal speeds of rotation:

$$\boldsymbol{Q_{nom(P1+P2)}=103{,}6} \text{ l/s},$$

where the work parameters of each of the pumps respectively are:

$$Q_{P1}=51{,}9 \text{ l/s}, H_{P1}=235{,}9 \text{ m u } \eta_{P1}=0{,}76;$$
$$Q_{P2}=51{,}7 \text{ l/s}, H_{P2}=236{,}9 \text{ m u } \eta_{P2}=0{,}76.$$

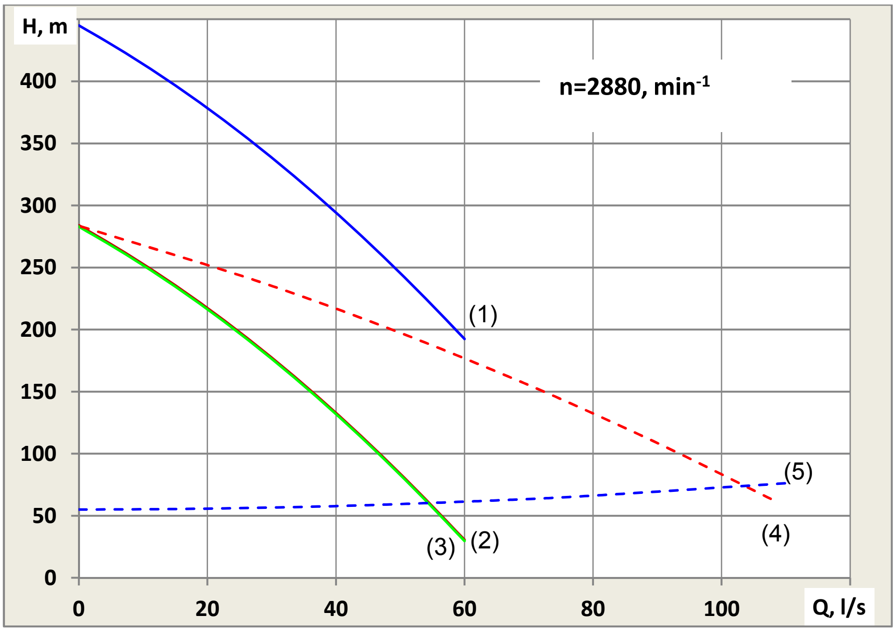

Figure 2. Graphic-analytical determination of the system's work regime, when the pumps P_1 and P_2 work in parallel: (1) pump's head characteristic, provided by the manufacturer; (2), (3) – the head characteristics (referred to p. A) of pumps P_1 and P_2; (4) – the characteristic of the common pipe

For the determination of a system's optimal flow rate, it is necessary to consider the restrictive condition of not allowing too low or high velocity ϑ to be implemented (the existence of too low a velocity leads to the accumulation of sediments in pipes, which interrupts smooth fluid transportation, while the existence of too high a velocity leads to the appearance of corrosion and also causes higher energy loses). That is why as a recommendable standard diapason of allowable velocity values the interval $\vartheta \in 0{,}75...3$, m/s, is considered, [1]. Considering that the common pipe (in **1**) consists of pipes, having different diameters and the relevant maximum and minimum allowable values, according to the restrictive velocity conditions, the relevant flow rates in these pipes, are determined. As a result, it can be indicated that the total system's flow rate Q belongs to the following interval: $Q \in (0{,}8\ldots1{,}15)Q_{nom(P1+P2)}$, m^3/s. The additional verification check indicates that the low boundary condition of the defined interval (for Q) should be changed to 0,9, for the velocities in the pipes, whose lengths are L_1 and L_{m2}, to be valid. It is clear that this final interval for Q represents 100% of the possible flow rates, as for the purpose of this research 4 values, which belongs to it, are selected - $(0{,}9;\ 1{,}05;\ 1{,}1;\ 1{,}15)Q_{nom(P1+P2)}$. For each of these values, where the total system's flow rate (Q) is being ensured by the two pumps (P_1 and P_2), working in parallel, the investigated ratios between the individual pump's flow rates (Q_{P1}/Q_{P2}) respectively are: 50-50 %; 55-45%; 45-55%; 60-40 %; 40-60 %; 65-35 %; 35-65 %.

For the achieving of the required flow rates, it is necessary for the frequency method of regulation to be used. Therefore, the summary (system's) head characteristic ((3) – fig.2), concerning the two pumps P_1 and P_2 – after they have been referred to the common point A, will intersect the characteristic of the common pipe in different points. Then if the equation ($H'_{(i)}=H_{st}+kQ_{(i)}^2$), describing the pipe system's characteristic, is known, the estimation of the head

$H'_{(i)}$ (in the common pipe – from p. A to the buffer tank) for the selected values of the flow rate Q_i is possible. The determination of the total head $H_{(P1,2)}$, for each of the parallel working pumps (P_1, P_2), can be achieved by the preliminary determination of the energy losses ($h_{v(i)}$) – for any of the selected values of Q_i, in the pipes (it is also necessary for the impact of the heights - h_1 and h_2, which have to be overcome, to be indicated):

$$H_{(P1,2)} = H'_{(P1,2)} + h_{v(P1,P2)} + h_{(1,2)},\ m. \quad (7)$$

To determine the coefficient of efficiency for each of the pumps (P_1, P_2), remembering the fact that after the change in the speed of rotation there will be a new work regime, it is necessary for the new speed of rotation to be determinate in advance. For this aim the graphic-analytical method, described in [2], where the parabola of similarity, which is a key factor (it is assumed that a given pump works at the same value of its coefficient of efficiency for all the ensured similar work regimes), can be used. The results, found after the completion of the hydraulic calculations, representing the parameters of the new work regimes, are given in table 2.

TAB. 2 Work parameters of the provided new work regimes

$Q=0,9.Q_{(P1+P2)}$	P_1+P_2		P_1+P_2		P_1+P_2		P_1+P_2		P_1+P_2	
$Q_1:Q_2$, %	50-50		55-45		45-55		60-40		40-60	
Q, m^3/s	0,047	0,047	0,051	0,042	0,042	0,051	0,056	0,037	0,037	0,056
H, m	220,35	223,01	221,26	221,87	219,52	224,27	222,26	220,85	218,78	225,65
η	0,78	0,78	0,75	0,78	0,78	0,75	0,70	0,77	0,77	0,71
n, min^{-1}	2714	2720	2810	2631	2621	2796	2911	2542	2533	2924
e_V, kWh/m^3	0,774	0,783	0,804	0,773	0,765	0,819	0,866	0,779	0,771	0,872
$Q=1,05.Q_{(P1+P2)}$	P_1+P_2		P_1+P_2		P_1+P_2		P_1+P_2		P_1+P_2	
$Q_1:Q_2$, %	50-50		55-45		45-55		60-40		40-60	
Q, m^3/s	0,054	0,054	0,060	0,049	0,049	0,060	0,065	0,044	0,044	0,065
H, m	221,92	225,18	223,16	223,63	220,79	226,90	224,52	222,23	219,79	228,78
η	0,72	0,72	0,64	0,77	0,77	0,65	0,53	0,78	0,78	0,54
n, min^{-1}	2877	2890	2997	2773	2762	3011	3121	2662	2652	3137
e_V, kWh/m^3	0,841	0,848	0,951	0,794	0,786	0,954	1,164	0,774	0,766	1,153
$Q=1,1.Q_{(P1+P2)}$	P_1+P_2		P_1+P_2		P_1+P_2		P_1+P_2		P_1+P_2	
$Q_1:Q_2$, %	50-50		55-45		45-55		60-40		40-60	
Q, m^3/s	0,057	0,057	0,063	0,051	0,051	0,063	0,068	0,046	0,046	0,068
H, m	222,50	225,98	223,86	223,43	221,26	227,86	225,36	222,75	220,16	229,93
η	0,69	0,69	0,58	0,75	0,75	0,60	0,45	0,78	0,78	0,47
n, min^{-1}	2933	2947	3061	2819	2810	3076	3194	2703	2693	3211
e_V, kWh/m^3	0,885	0,890	1,045	0,838	0,830	1,166	1,377	0,779	0,770	1,347
$Q=1,15.Q_{(P1+P2)}$	P_1+P_2		P_1+P_2		P_1+P_2		P_1+P_2		P_1+P_2	
$Q_1:Q_2$, [%]	50-50		55-45		45-55		60-40		40-60	
Q, m^3/s	0,060	0,060	0,066	0,054	0,054	0,066	0,071	0,048	0,048	0,071
H, m	223,10	226,81	224,59	224,95	221,75	228,87	226,23	223,28	220,54	231,13
η	0,64	0,65	0,52	0,73	0,73	0,53	0,36	0,77	0,77	0,38
n, min^{-1}	2991	3005	3127	2873	2860	3143	3268	2746	2735	3286
e_V, kWh/m^3	0,944	0,947	1,178	0,838	0,830	1,166	1,731	0,787	0,779	1,657

The results found in this research make it clear that because of the imposed restrictive condition, concerning the velocity of fluid's transportation through the pipes, the work regimes at Q_{P1}/Q_{P2}= (65:35 and 35:65 %), are not available.

As a key factor, determining the effectiveness of the system's work performance the specific energy consumption e_V, is selected. According to [4], it can be estimated by (8):

$$e_V = (g/3600).(H/\eta),\ \ kWh/m^3 \quad (8)$$

where H is the pump's head, $\eta=\eta_E\eta_T\eta_P$ - the total coefficient of efficiency of the pump aggregate (it is assumed that the motor's coefficient of efficiency has a constant value [3], and the transmission's coefficient of efficiency is equal to 1).

After the completion of the estimations, used to determine the specific energy consumption $e_{V(i)}$ for each of the required work regimes, it has been presented as a dimensionless parameter. For this aim $e_{V(i)}$ has been referred to the specific energy consumption e_{Vnom}, which is realized when the pumps (P_1 and P_2) work on their nominal speed of rotation ($n=2880$ min^{-1}).

3. Results and discussion

Figure 3 graphically presents the change of the relative specific energy consumption $e_{V,OTH}$ for different ratios between the flow rates (Q_{P1} and Q_{P2}) of the pumps P_1 and P_2, when the values of the system's total flow rate are preliminary asked.

Figure 4 graphically presents the relative specific energy consumptions $e_{V,OTH(i)}$, provided at same asked values of the ratio between the flow rates of the pumps P_1 and P_2, for the required values of the system's total flow rate.

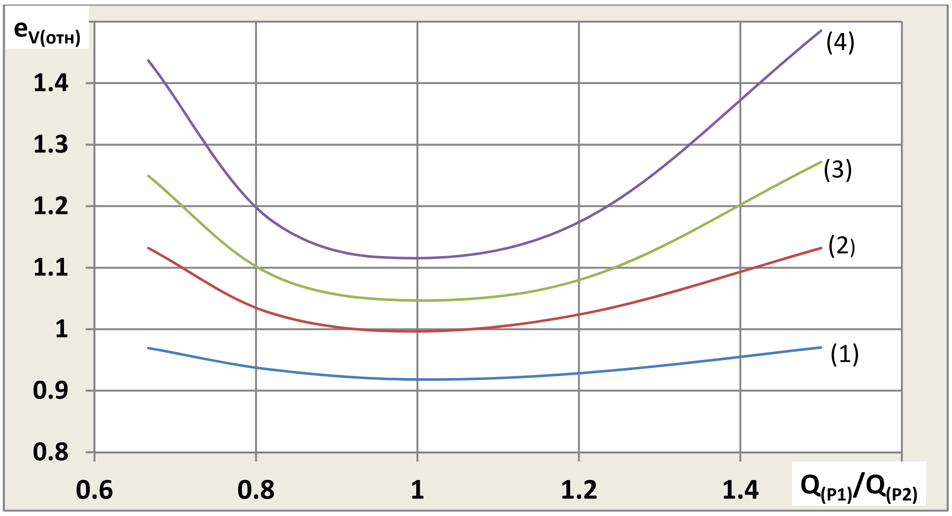

Figure 3. Change in the relative specific energy consumption e_{VOTH}, for different ratios between the flow rates Q_{P1} and Q_{P2}, ensured by the pumps P_1 and P_2: (1) - $Q=0,9Q_{nom}$; (2) – $Q=1,05Q_{nom}$; (3) – $Q=1,1Q_{nom}$; (4) – $Q=1,15Q_{nom}$

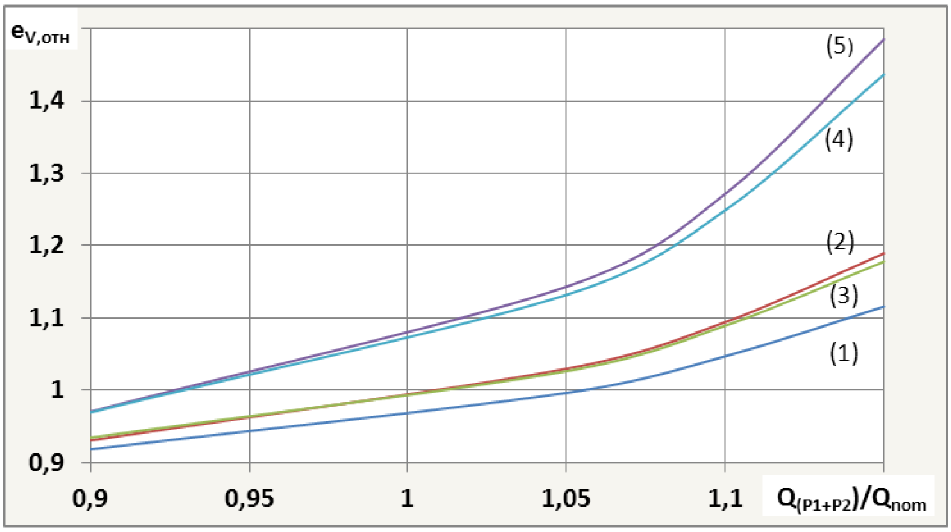

Figure 4. Relative specific energy consumption for the investigated ratios of Q_{P1}/ Q_{P2}, when the system's total flow rate changes: (1) – 50-50%; (2) – 55-45%; (3) – 45-55%; (4) – 60-40%; (5) - 40-60%

Figure 4 graphically presents the relative specific energy consumptions $e_{V,отн(i)}$, provided at same asked values of the ratio between the flow rates of the pumps P_1 and P_2, for the required values of the system's total flow rate.

Analyzing the results found in fig. 3, it can be clearly seen that with the increasing of the system's total flow rate Q the energy, used for the transportation of 1 m^3 of water, also start to increase. This happens faster, when: $Q_{P1}/Q_{P2}<0,8$ and $Q_{P1}/Q_{P2}>1,2$. The increased energy consumption, when the system's total flow rate has been increased, can be explained with the fact that the energy losses in the pipe system are also increased. Because of the significant length of the pipes these losses can be defined as one of the key factors, having an impact on the head of the pumps, respectively on their specific energy consumptions. While the above statement is expected, it is more interesting to indicate that for all the investigated work regimes, independently of the change of Q, (for a given pump system (and pumps)), the specific energy consumption will be minimal at $Q_{P1}/Q_{P2} \approx$ 50-50%. Whether this conclusion is valid for a wide range of pumps, or it is true only for this particular case, could be found when additional research, where different combinations of pumps are provided, is accomplished.

For the given pump system the optimal value of the relative specific energy consumption will be **$e_{V,отн}\approx 0,92$**, which represents the minimum of the function of equation (9), describing curve (1) – fig. 3:

$$e_{V,отн} = -0,2181(Q_{P1}+Q_{P2})^3 + 0,9968(Q_{P1}+Q_{P2})^2 - \\ -1,353(Q_{P1}+Q_{P2}) + 1,4929. \quad (9)$$

The results, graphically presented in fig. 4, confirm the tendency that $e_{V,отн}$ increases with the increasing of the system's total flow rate ($Q_{(P1+P2)}$). More interesting in this case is for the change of $e_{V,отн}$, when a given flow rate is ensured at different values of the ratio Q_{P1}/Q_{P2}, to be indicated. Over again, $Q_{P1}=Q_{P2}$ can be determinate as the most appropriate distribution between the flow rates of P_1 and P_2. Also, it can be seen that for both the "mirror" ratios (55-45↔45-55; 60-40↔40-60), when the system's total flow rate has a constant value, the values of $e_{V,отн}$ are very similar, almost covering each other at $Q_{(P1+P2)}=(0,9...1).Q_{nom}$. However, at $Q_{(P1+P2)}>Q_{nom}$ the difference between the values of $e_{V,отн}$ starts to increase.

A better imagination for the change of $e_{V,отн}$, when the frequency method of flow rate regulation is used, can be ensured if the graphs, presented in figures 3 and 4, are being combined in a common three-dimensional graph (fig. 5). In this graph the $e_{V,отн}$ is presented as a function of two parameters - Q_{P1}/Q_{P2} and $Q_{(P1+P2)}/Q_{nom}$.

For this aim with the help of Matlab a mathematical model is established. Using this model make it possible for a given number (17) of curves, describing the presented spatial plane, to be found. The model representing the relation between $e_{V,отн}$ and Q_{P1}/Q_{P2}; $Q_{(P1+P2)}/Q_{nom}$, is given by (10):

$$e_{v,отн} = 1/(b_1 + b_2(Q_{P1}/Q_{P2} + b_3(Q_{P1}/Q_{P2})^2) + b_4 + \\ b_5 Q_{P1}/Q_{P2} + b_6(Q_{P1}/Q_{P2})^2 + b_7(Q_{P1}/Q_{P2})(Q_{(P1+P2)}/Q_{nom}) + \\ + b_8(Q_{(P1+P2)}/Q_{nom}) + b_9, \quad (10)$$

where the values of the coefficient b are given in Table 3.

TAB. 3 Values of the coefficient, taking place in model: $e_{V,отн}=f(Q_{P1}/Q_{P2}, Q_{(P1+P2)}/Q_{nom})$

b_1	b_2	b_3	b_4	b_5	b_6	b_7	b_8	b_9
0.6700	0.9002	3.0907	0.2134	-1.6840	0.7584	0.3838	0.8563	0.2134

The coefficient of correlation between the estimated and predicted (according to the established model) values of the relative specific energy consumption is R=0.9297. Using the Student's t-criteria the significance of the coefficient of correlation is found.

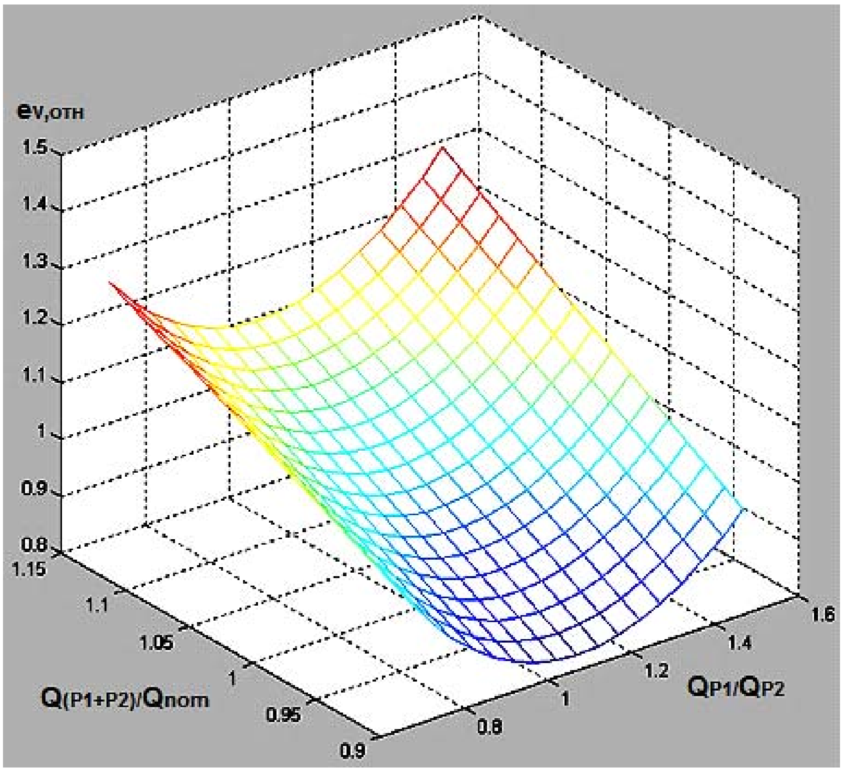

Figure 5. Change of the relative specific energy consumption $e_{V,OTH}$, presented as a function between Q_{P1}/Q_{P2} and $Q_{(P1+P2)}/Q_{nom}$

In fig. 5 it can be clearly seen the existence of different work regimes, belonging to the defined working area, where the energy consumption has the same value $e_{V,OTH}$=const. The curves, found by (9), consisting of that kind of work regimes, are given in fig. 6.

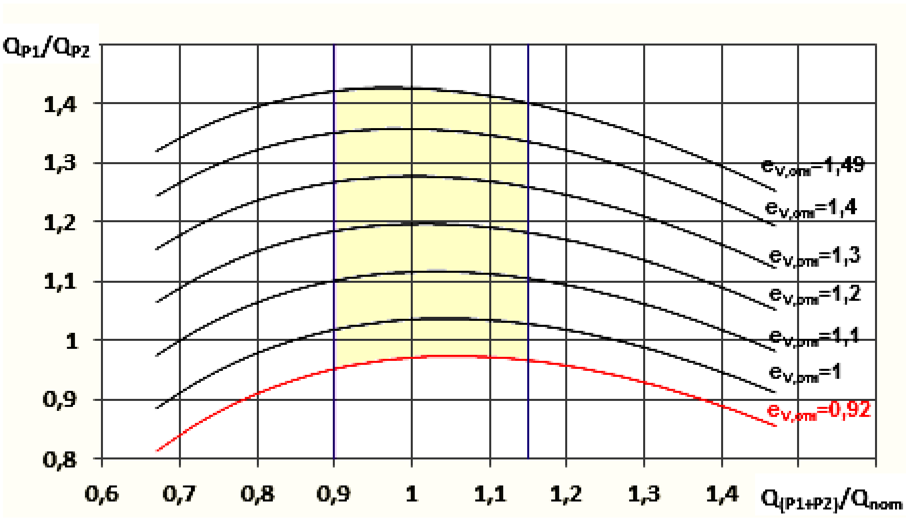

Figure 6. Curves of constant relative specific energy consumption

Analyzing the results found in fig. 6, we can see the interesting fact that the minimal (relative) specific energy consumption (the red line) – for a given system, can be ensured by providing different system's total flow rates, in case that some strictly regulated ratios between the flow rates of P_1 and P_2 are met.

The determination of the optimal flow rate for the second part of the pump system can be done in a similar manner, however it is necessary for the capacity of the buffer tank to be indicated.

4. Conclusions

According to this research, the following are the most important conclusions:

- the established method for the determination of the optimal system’s flow rate, in terms of energy efficiency, can be applied for any given pump system;
- after some trends about the change in the specific energy consumption, provided for the ensuring of different system’s flow rates, which also are distributed differently between the two supplying pumps, are indicated, the next step is to investigate whether these trends are valid only in this particular case or can be used in general;
- it is proved that the system’s minimal (relative) specific energy consumption can be provided by the ensuring of different work regimes.

REFERENCES

[1] Designing norms of water-supply systems, ABC Technics Ltd., 2003
[2] Popov G., Basic data and investigation of the energy efficiency of pump systems, University of Ruse, 2008
[3] Popov, G., Klimentov Kl., Kostov B., “Methods to estimate the energy consumption in regulating the flow rate of pump systems”. DEMI’2011, Banja Luka (Bosnia), 2011, pp. 495-500
[4] Burt C., Piao X., Gaudi F., Busch B., Electric Motor Efficiency under Variable Frequencies and Loads, California, USA, 2006
[5] Popov, G., A. Krasteva, B. Kostov, D. Ivanova, Kl. Klimentov, Optimization of the energy consumption of a pump system used for industrial water supply. Annals of faculty engineering Hunedoara – international journal of engineering, Tome XII ISSN: 1584-2665, Romania, 2014