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Optimization of Pump Unit Operation Modes With Respect to Minimum of Specific Energy Consumption

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ABSTRACT: This article considers the optimization of the pump unit operation modes based on analytical description of main (nameplate) operation characteristics of the centrifugal pump considering the change of its rotation frequency and on the selection the specific energy consumption on the rising of volume unit of water as a criteria of operation efficiency evaluation of the pump unit. When optimizing the operation modes of the pump unit the nonlinear programming problem has been solved where the target function (specific energy consumption) is minimized and the limitations are represented as inequations. Based on the generalized Lagrange multiplier rule and known Kuhn-Tucker theorem, the system of nonlinear equations is obtained, having solved thereof allows to determine the optimal values of the parameters under control.

KEY WORDS: specific power consumption, pumping assembly, pumping unit, pumping station, geometric, kinematic and dynamic similarity, pump rate, head, power, efficiency factor, rotation frequency control, approximation coefficient.

I. INTRODUCTION

In meliorative pumping stations (PS) the basis of hydro mechanical equipment is centrifugal pumps which ensure water supply to irrigation areas.

The base parameters of centrifugal pumps, particularly Q, head H, power P, efficiency factor (EF) η, rotation frequency n are in strictly defined interrelation. The head value, pump shaft power and efficiency factor depending on the delivery of the pump are represented as the system of points in coordinates $Q-H$, $Q-P_M$ and $Q-\eta$, the characteristics obtained herewith are named delivery and head, delivery and power, delivery and EF characteristics of the pump . The specified characteristics determined experimentally at $n = const$ according to the results of factory rig tests of this model of centrifugal pump represent nameplate or cataloged characteristics.

The analytical dependency between delivery Q and head H is determined based on the flow continuity equation analysis, Euler’s main equation of vane-type centrifugal hydraulic machines and K. Pfleiderer’s energy balance. At the same time, E.A Preger has suggested to consider the delivery and head characteristic of the pump as a parabola equation as a result of mathematical processing of pumps characteristics obtained experimentally. Therefore, the analytical dependency of actual centrifugal pump characteristic is generally described by the trinominal of second degree:

$$H = A_H * n^2 + B_H * n * Q + C_H * Q^2, \tag{1}$$

where A_H, B_H and C_H - constant coefficients of delivery and head characteristic of the pump depending both on the type of its configuration and the design parameters.
The power delivered to the pump shaft is determined as:

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$$P_M = A_P * n^2 * Q - B_P * n * Q^2 + C_P * n^3, \qquad (2)$$

where Ap, Bp and Cp - constant coefficients of delivery and head characteristic of the pump depending both on the type of its configuration and design parameters.

Let us determine the EF of the pump as the ratio of useful power $P_{пол}$ to delivered power from the pump shaft side $Pм$:

$$\eta = \frac{P_{пол}}{Pм} = \frac{\gamma * Q * H}{102 * Pм}, \qquad (3)$$

where γ - density of water pumped

or

$$\eta = \frac{\gamma * Q}{102} * \frac{A_H * n^2 * Q + B_H * n * Q^2 + C_H * Q^3}{A_P * n^2 * Q - B_P * n * Q^2 + C_P * n^3}. \qquad (4)$$

Relating to centrifugal pumps their constant parabola coefficients (delivery and head, and delivery and power characteristics) are determined from nameplate or cataloged characteristics using approximation by least square method or experimentally as a result of statistical processing of operation parameters NA [2].

II. RELATED WORK

A significant contribution to the development of the theory and practice of modern electromechanics – automation of industrial plants, peculiarities of functioning of Electromechanical and hydro-mechanical equipment of pumping stations as a control and regulation system, as well as energy saving by means of a regulated electric drive were devoted to the work of Onishchenko G. B., Yunkova M. G., Ilyinsky N. F., Braslavsky I. Ya., Moskalenko V. V., Klyucheva V. I., Basharina A.V., Leznov B. S., Rychagova V. V., florinsky M. M., Minaeva A.V., Karelin V. Ya., Walker C., Jahns T. M., Walters D. G. and others.

III. METHODOLOGY AND DISCUSSION

We use the similarity theory of hydraulic pumps that uses geometric, kinematic and dynamic similarity as the basis of hydraulic flow events modeling in order the analytical form characteristics $Q-H$ and $Q-Pм$ allow to consider rotation frequency variation during its control as well as the value of outside diameter of the pump impeller in case of its machining on outside diameter.

The geometric similarity that suggest a proportional change of the pump impeller is conveyed through the ratio of external diameter of impeller of the pump under consideration to the diameter of operational initial centrifugal pump:

$$i_d = D_Ф / D_{KAT}, \qquad (5)$$

where $D_Ф$ – actual external diameter of impeller of considered centrifugal pump;

D_{KAT} – cataloged (nameplate) external diameter of impeller of initial centrifugal pump.

Kinematic similarity suggests that the ratio of flow rates both inlet and outlet of considered and initial centrifugal pumps are constant.

Finally, dynamic similarity of centrifugal pumps may be represented by the rotation frequency ratio of their impellers:

$$i_n = n / n_H, \qquad (6)$$

where n – rotation frequency of considered centrifugal pump;

n_H – rated (according to technical passport) rotation frequency of initial centrifugal pump.

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Therefore, the set of regularities can be obtained in accordance with the similarity theory of hydraulic pumps:

$$Q / Q_{И} = i_d^3 * i_n, \tag{7}$$

$$H / H_{И} = i_d^2 * i_n^2, \tag{8}$$

$$P_M / P_{MИ} = i_d^5 * i_n^3, \tag{9}$$

where Q, H, P_M – delivery, head and mechanical power accordingly of considered centrifugal pump;

$Q_{И}$, $H_{И}$, $P_{MИ}$ – accordingly, delivery, head and mechanical power of initial centrifugal pump.

At $i_d = 1$ the dynamic similarity is only retained that corresponds to the change of rotation frequency of centrifugal pump.

At $i_n = 1$ the geometric similarity of impeller of centrifugal pump is only preserved.

The method of impeller machining of centrifugal pump without change of blade front side shape is frequently used in practice of meliorative irrigation when it is necessary to change delivery or head of centrifugal pump. At that the geometric similarity is broken. In this case the ratios (7) - (9) are become as follows:

$$Q / Q_{И} = i_d^2 * i_n, \tag{10}$$

$$H / H_{И} = i_d^2 * i_n^2, \tag{11}$$

$$P_M / P_{MИ} = i_d^4 * i_n^3, \tag{12}$$

The following expressions ensure more accurate result because of changed gap of centrifugal pumps with specific speed $n_S < 150$ more widely used in meliorative PS:

$$Q / Q_{И} = i_d * i_n, \tag{13}$$

$$H / H_{И} = i_d^2 * i_n^2, \tag{14}$$

$$P_M / P_{MИ} = i_d^3 * i_n^3, \tag{15}$$

Then, in accordance with the expression (1) we'll obtain:

$$H = i_d^2 * i_n^2 (A_H + B_H * Q_{И} + C_H * Q_{И}^2) = A_H * i_d^2 * i_n^2 + B_H * i_d * i_n * Q + C_H * Q^2 \tag{16}$$

Considering expression (2) we have:

$$P_M = i_d^3 * i_n^3 (A_P * Q_{И} - B_P * Q_{И}^2 + C_P) = A_P * i_d^2 * i_n^2 * Q - B_P * i_d * i_n * Q^2 + C_P * i_d^3 * i_n^3 \tag{17}$$

In accordance with the expression (1.7), using obtained expressions (1.19) and (1.20) which describe cataloged (nameplate) $Q - H$ and $Q - P_M$ characteristics of centrifugal pump, the EF dependency from its delivery value $\eta = \varphi.(Q)$ as analytical can be obtained.

$$\eta = \frac{\gamma * Q}{102} * \frac{A_H * i_d^2 * i_n^2 + B_H * i_d * i_n * Q + C_H * Q^2}{A_P * i_d^2 * i_n^2 * Q - B_P * i_d * i_n * Q^2 + C_P * i_d^3 * i_n^3}. \tag{18}$$

So, the ratios (16), (17) and (18) with approximation coefficients calculated according to the least square method for the type of centrifugal pump considered allow to describe analytically its operation characteristics at specified rotation frequency and actual diameter of its impeller may be used for mathematical model creation of the pump unit, where the limitations imposed on the operation parameters of PU are determined as [3]:

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$$Q \geq Q_{ГР};$$
$$H_{\min} \leq H_T \leq H_{\max};$$
$$n_{\min} \leq n \leq n_{\max}, \quad (19)$$

where $H_T = H_{CT} + R_{TP} * Q^2$,

$Q_{ГР}$ - required water supply in accordance with the water consumption schedule for irrigating land areas;

$H_T, H_{\min}, H_{\max}$ -allowable current, minimal and maximal created head value according to operation conditions of PU;

$n_{\min}$ - minimal rotation frequency of PA in accordance with the specified operation mode of PU;

$$n_{\min} = n_H * \sqrt{H_{\min} / H_{\max}};$$

$n_{\max}$ - maximum allowable according to manufacturing conditions of the pump, rotation frequency of PA;

n_H - rated rotation frequency of PA;

R_{TP} - resistance of pipe pressure network that includes, the resistances of each suction, connection, inlet pipelines of PA and common pressure network of PU.

The main technical and economic operation indicators of PU as well as the all PS is the EF of pumping unit and specific power consumption consumed to supply a unit of water volume to hydraulic pressure network [3].

As known [4] the most reliable and practically realizable method to determine a specific power consumption rate is a ratio of consumed power to effective work being done for the same time by a pumping assembly. The expressions corresponding to this method are as follows:

$$\Delta Э = \frac{P_{ПОЛ}}{W * \eta_H * \eta_{ДВ} * \eta_{TP}} \text{ kW} * \text{h/m}^3; \quad (20)$$

$$P_{ПОЛ} = \frac{\gamma * Q * H}{102} \text{ kW}; \quad (21)$$

$$W = Q * 3600 \text{ m}^3\text{/h}, \quad (22)$$

Where $\Delta Э$ - specific power consumption;

$P_{ПОЛ}$ - effective power of the pump;

W - volume of pumped liquid;

$\eta_H, \eta_{ДВ}$ - EF of the pump and motor;

η_{TP} - pressure pipeline EF;

γ - density of the liquid being pumped;

Q - pump delivery m^3/s;

H - head developed by the pump.

or $$\Delta Э = \frac{1000 * H}{102 * 3600 * \eta_H * \eta_{ДВ} * \eta_{TP}} = \frac{0{,}002724 * H}{\eta_H * \eta_{ДВ} * \eta_{TP}} \text{ kW} * \text{h/m}^3. \quad (23)$$

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Specific power consumption rate is a power consumption, $kW*h$ to supply 1000 m^3 of the liquid being pumped to a height of $H_{Г}$ m for stable operation conditions of a pump and its motor to pressure network. In this connection, particular power consumption rate of PA of water lifting pumping PS is determined as:

$$\Delta Э^{И}_{HA.i} = \frac{2{,}724 * H_{Г.i}}{\eta_{HA.i}}; \ kW*h/1thous.*m^3 \quad (24)$$

where $\eta_{HA.i} = \eta_i * \eta_{ДВ.i} * \eta_{TP.i}$ (25)

$H_{Г.i}$ - geometric lifting height of pressure pipeline i – th PA;

η_{HAi} - EF of i – th pumping assembly as a part of PU;

η_i - pump EF of i – th PA;

$\eta_{ДВ.i}$ - EF of drive motor of i – th PA;

$\eta_{TP.i}$ - pressure pipeline EF of i - th PU.

IV. RESEARCH RESULTS

It is known that at constantly preserved characteristic of pipeline pressure network (H_{CT} = *const*, R_{TP} = *const*), specified geometry of centrifugal pumps impellers, (i_d = *const*) the position of performance point of PU is determined by rotation frequency of PA n and their total delivery $Q_T \approx N * Q_{HA}$.

Therefore, for specified operation conditions of PU, the optimizing of its operation modes according to minimum criteria of specific electric energy consumption can be represented as the target function dependent on rotation frequency of PA and irrigation water consumption. It is evident that the minimum of specific electric energy consumption of PU will be reached at its consumption minimum by each of together operating PA while ensuring agreed and proportional control of rotating frequency of PA.

Specific power consumption of PA can be represented as:

$$\Delta Э = \frac{\left[m_1 * \left(m_2 * n^6 + m_3 * Q * n^5 + m_4 * Q^2 * n^4 + m_5 * Q^3 * n^3 + m_6 * n^3 + m_7 * Q * n^2 + m_8 * Q^4 * n^2 + m_9 * Q^2 * n + m_{10}\right)\right]}{\left[Q * \left(à_1 * n^2 + b_1 * Q * n + c_1 * Q^2\right)\right]}, \quad (26)$$

where $m_1 = 2724 * Kt * P_{ДВ.Н} / (0{,}001 * \gamma)$;

$$m_2 = \{(1 - \eta_{ДВ.Н}) / [\eta_{ДВ.Н} * (1 + A_П) * (K_{ИЗН} * Kt * P_{ДВ.Н})^2]\} * c_2^2 = m_{11} * c_2^2;$$

$m_3 = 2 * m_{11} * a_2 * c_2$; $\quad m_4 = m_{11} * (a_2^2 - 2 * b_2 * c_2)$;

$m_5 = -2 * m_{11} * a_2 * b_2$; $\quad m_6 = c_2 / (K_{ИЗН} * Kt * P_{ДВ.Н})$;

$m_7 = a_2 / (K_{ИЗН} * Kt * P_{ДВ.Н})$; $\quad m_8 = m_{11} * b_2^2$;

$m_9 = -b_2 / (K_{ИЗН} * Kt * P_{ДВ.Н})$; $\quad m_{10} = A_П * (1 - \eta_{ДВ.Н}) / [\eta_{ДВ.Н} * (1 + A_П)]$;

$a_1 = A_H * i_d^2 / n_H^2$; $\quad b_1 = B_H / (n_H * i_d)$; $\quad c_1 = C_H / i_d^4$;

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$$a_2 = A_p * i_d^2 / n_H^2; \qquad b_2 = B_p /(n_H * i_d); \qquad c_2 = C_p * i_d^5 / n_H^3;$$

$$K_{ИЗН} = e^{-0{,}00000833*T_{пр}}; \qquad Kt = 1{,}24 - 0{,}000196 * t_0^2; \qquad i_d = D_ф / D_{KAT},$$

$K_{ИЗН}$ - wearing coefficient of pump working elements;
Kt - coefficient considering changes of rated power of motor of PA depending on environment temperature;
D_{KAT} - cataloged (nameplate) diameter of impeller of the pump.

We designate the delivery of Q the pumping assembly x_1 and rotation frequency n as x_2.

Then the optimization task of operation modes of PU can be represented as the following nonlinear programming task:

$$\Delta Э (x_j) \Rightarrow \min; \qquad (27)$$

$$g_{i.j}(x_j) \le 0, \quad i = 1,\dots,m; \qquad (28)$$

$$g_{l.j}(x_j) = 0, \quad l = m+1,\dots,p; \qquad (29)$$

$$x_j > 0, \quad j = 1,\dots,k. \qquad (30)$$

We minimize the target function $\Delta Э(x)$ within allowable set specified by the limitations (28) – (30) based on generalized Lagrange multiplier rule and known Kuhn-Tucker theorem. Then the minimum point of specified target function may be determined as the solution of equation systems with additional variables $\lambda_{\cdot i}$, x_{k+i}, $i = 1,\dots,m$; μ_j, $j = 1,\dots,k$; $\lambda_{\cdot l}$, $l = m+1,\dots,p$:

$$\left.\begin{aligned} &\frac{\partial \Delta Э(x_j)}{\partial x_j} + \sum_{i=1}^{m} \lambda_{\cdot i} * \frac{\partial g_{i.j}(x_j)}{\partial x_j} + \sum_{l=m+1}^{p} \lambda_l * \frac{\partial g_{l.j}(x_j)}{\partial x_j} - \mu_j = 0 \\ &g_{i.j}(x_j) + x_{k+1}^2 = 0 \\ &g_{l.j}(x_j) = 0 \\ &\lambda_i * x_{k+1} = 0 \\ &\mu_j * x_j = 0 \end{aligned}\right\} \qquad (31)$$

At that $x_j \ge 0$, $\mu_j \ge 0$, $\lambda_i \ge 0$, $\lambda_l > 0$,

where $g_{i.j}(x_j)$ - limitations represented in (19) as inequations;
$g_{l.j}(x_j)$ - limitations represented in (19) by equations;
x_j - dependent variables x_1 and x_2 $(k = 2)$.

As the control of rotation frequency of PA structurally integrated for concurrent working of PU and consequently its delivery is carried out between the stages then for specified range of the head change $H_{min} < H < H_{max}$ ensured at stage (per assembly) control of PU performance with automation devices, we write the limitations (19) as:

$$Q_{ГР} - Q_T = Q_{ГР} - N * Q = Q_{ГР} - N * x_1 < 0 \qquad (32)$$

$$n - n_{max} = x_2 - n_{max} < 0 \qquad (33)$$

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$$n_{\min} - n = n_{\min} - x_2 < 0 \quad (34)$$

$$a_3 * n^2 + b_3 * Q * N * n + c_3 * N^2 * Q^2 - H_{CT} - R_{TP} * N^2 * Q^2 =$$

$$= a_3 * x_2^2 + b_3 * N * x_1 * x_2 + N^2 * x_1^2 * (c_3 - R_{TP}) - H_{CT} = 0, \quad (35)$$

where $a_3 = A_{SH} * i_d^2 / n_H$; $b_3 = B_{SH} /(n_H * i_d)$; $c_3 = C_{SH} / i_d^4$, (36)

A_{SH}, B_{SH}, C_{SH} - approximation coefficients of total delivery and head characteristic of N of concurrently working PA. As a result, we will obtain the system of nonlinear equations:

$$\left.\begin{aligned}
&\partial\Delta Э(x_1,x_2)/\partial x_1 - \lambda_1 * N + \lambda_4 * [b_3 * N * x_2 + 2 * N^2 * x_1 * (c_3 - R_{TP})] - \mu_1 = 0 \\
&\partial\Delta Э(x_1,x_2)/\partial x_2 + \lambda_2 - \lambda_3 + \lambda_4 * (2 * a_3 * x_2 + b_3 * N * x_1) - \mu_2 = 0 \\
&Q_{ГР} - N * x_1 + x_3^2 = 0 \\
&x_2 - n_{\max} + x_4^2 = 0 \\
&n_{\min} - x_2 + x_5^2 = 0 \\
&a_3 * x_2^2 + b_3 * N * x_1 * x_2 + N^2 * x_1^2 * (c_3 - R_{TP}) - H_{CT} = 0 \\
&\lambda_1 * x_3 = 0;\ \lambda_2 * x_4 = 0;\ \lambda_3 * x_5 = 0;\ \mu_1 * x_1 = 0;\ \mu_2 * x_2 = 0;
\end{aligned}\right\} \quad (37)$$

$$x_1, x_2 > 0;\quad \mu_1, \mu_2 > 0;\quad \lambda_1, \lambda_2, \lambda_3, \lambda_4 > 0.$$

This equations system can be solved using Newton-Raphson method that is the most common and widely used method to solve the system of nonlinear equations described as known by recurrent formula:

$$x_{K+1} = x_K - \alpha_K * [\varphi(x_K) / \varphi'(x_K)], \quad (38)$$

where $\varphi(x_K)$ - scalar function of some argument x_K;

α_K - length of iteration step.

In solving (36) by Newton-Raphson method it is necessary to specify the relative error ε, number of the system's equations, maximum iteration number as well as initial approximation for each x_i, $i = 1,\dots,m$.

For x_1 and x_2 as initial approximations the actual delivery Q of PA and its current rotation frequency n are taken:

$$x_{10} = Q = Q_T / N; \qquad x_{20} = n. \quad (39)$$

Then:

$$x_{30} = \sqrt{N * Q - Q_{ГР}};\quad x_{40} = \sqrt{n_{\max} - n};\quad x_{50} = \sqrt{n - n_{\min}}. \quad (40)$$

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V. CONCLUSIONS

So, based on the expressions that specify mathematical model of PU as well as the system of nonlinear equations (37) and initial conditions (39), (40) it is possible to determine the optimal values of rotation frequency of PA and delivery of PU which ensure the minimum of specific power consumption while the parameters $R_{TP} = const$, $H_{TP} = const$, $i_d = const$ *constantly preserved*

It is determined that the optimization of operation parameters of pumping unit as a function of rotation frequency of the pumping assembly at control of its drive motor rotation frequency below the rated value ensures the reduction of specific power consumption up to 15-20% depending on its load as compared with the operation modes of the pumping unit equipped with non-adjustable electric drive.

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