Basis of Energy Efficiency Economical and Ecological Approach Method for Pumping Equipments and Systems

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Abstract: The paper presents the basis of an original economical and ecological approach of the pumping systems energy efficiency, offering a holistic picture of the pumping efficiency, with emphasis in economical and GHGs emissions mitigation effects. The main contributions consists in the original energy efficiency evaluation method of pumping equipments, a particularization of the pumping systems optimal operation characteristic determination for variable requested flow conditions, and a practical application for parallel operation variable speed driving adjustment, the actual most applied system. In the application, there are emphasis economic and environmental effects, with a special focus on the GHGs emissions. The paper results application consists in automatic driving operation and in new pumping optimization operation solution evaluation, including pumps and motors energy classification, considering the actual international trends and European Commission commitments in energy and climate changes fields.

Key-Words: energy efficiency, GHGs emission mitigation, operation modeling, variable speed driving

1. European frame of energy efficiency innovation

Pumping systems and equipments energy efficiency improvement is vital in reaching the energy and climate change European policy objectives: to reduce greenhouse gas emissions by 20% and ensure 20% of renewable energy sources in the EU energy mix; to reduce EU primary energy use by 20% by 2020. Considering that energy consumption in this type of installations represents more that 20% of the total energy consumption of a modern economy, pumping system efficiency improvement could have a consistent contribution to the effort to accelerate the development and deployment of cost-effective low carbon and energy efficiency technologies. Energy industry has five clear components (resources, production, transport, distribution and consumption), each of them with specific problems and solutions. In the longer term, the solutions for decarbonizes the economy and improve energy efficiency, new generations of technologies have to be developed through breakthrough in research. Since the oil price shocks in the 70s and 80s, Europe has enjoyed inexpensive and plentiful energy supplies. The easy availability of resources, no carbon constraints and the commercial imperatives of market forces have not only left the union dependent on fossil fuels, but have also tempered the interest for innovation and investment in new energy technologies. This has been described as the greatest and widest-ranging market failure ever seen.

Intrinsic weaknesses in energy innovation

The energy innovation process, from initial conception to market penetration, also suffers from unique structural weaknesses. It is characterized by long lead times, often decades, to mass market due to the scale of the investments needed and the technological and regulatory inertia inherent in existing energy systems. Innovation faces entrenched 'locked-in' carbon based infrastructure investments, dominant actors, imposed price caps, changing regulatory frameworks and network connection challenges. The market take-up of new energy technologies is additionally hampered by the commodity nature of energy. New technologies are generally more expensive than those they replace while not providing a better energy service. The immediate benefits tend to accrue to society rather than the buyers. Some technologies face social acceptance issues and often require additional up-front integration costs to fit into the existing energy system. Legal and administrative barriers complete this innovation adverse framework. In short, there is neither a natural market appetite nor a short-term business benefit for such technologies. This market gap between supply and demand is often referred to as the 'valley of death' for low carbon energy technologies. Public intervention to support energy innovation is thus both necessary and justified.

Between the key EU technological challenges for the next 10 years to meet the 2020 targets are:
– *Bring to mass market more efficient energy conversion and end-use devices and systems, in buildings, transport and industry, such as poly-generation and fuel cells;*
– *Maintain competitiveness in fission technologies, together with long-term waste management solutions.*
And between the Key EU technology challenges for the next 10 years to meet the 2050 vision is: *achieve breakthroughs in enabling research for energy efficiency: e.g. materials, nano-science, information and communication technologies, bio-science and computation.*
Between the European regulations with an important impact to energy efficiency, are:
- New and unitary Legislation or Amending the Directive 92/75/EEC-for the global energy and environmental performance-then-the better and simplification
- Eco-design: Directive 2005/32/EC of the European Parliament and of the Council of 6 July 2005, establishing a framework for the setting of eco-design requirements for energy-using products and amending Council Directive 92/42/EEC and Directives 96/57/EC and 2000/55/EC of the European
Parliament and of the Council
- Energy-Star: Regulation (EC) No. 2422/2001 of the European Parliament and of the Council of 6 November 2001 on a Community energy efficiency labeling program for office equipment as amended.
- Energy Labeling: Council Directive 92/75/EEC of 22 September 1992 on the indication by labeling and standard product information of the consumption of energy and other resources by household appliances.
- Eco-label: Regulation (EC) No 1980/2000 of the European Parliament and of the Council of 17 July 2000.
According to the European plans for eco-design implementation measures, the pumps, funs and electric motors should be in the phase of drafting measures and impact assessment by the end of this year.

2. Operation pumps adequate representation solutions

2.1.Pumps operation point representation

Operation point could be obtain graphically as the intersection between the pump head-flow characteristic curve $H_p(Q)$, equation (1), with the pipe head-flow resulting characteristic curve $H_r(Q)$, equation (2), as it is presented in fig. 1 [1], or analytically solving the equations system representing the polynomial approximation of the mentioned characteristics curves, as follows [2]:

$$H(Q) = a_1 \cdot Q^2 + a_2 \cdot Q + a_3 \tag{1}$$

$$H_r(Q) = b_1 \cdot Q^2 + b_2 \tag{2}$$

where:
$a_1,...,a_x$ – polynomial coefficients for the pump head-flow characteristic;
$b_0,...,b_x$ - polynomial coefficients for the network head-flow characteristic.

It is to mention that the pump characteristic represent the resulting head-flow characteristic of one pump or of a whole pumping station, and the pipe characteristic represent the resulting head-flow characteristic of one pipe or a whole network [13].

Other energetic parameters, efficiency and hydraulic power, are also represented in fig 1, and could be approximated by the polynomial equations (3) and (4).

$$\eta(Q) = c_1 \cdot Q^2 + c_2 \cdot Q + c_3 \tag{3}$$

$$P(Q) = d_1 \cdot Q^2 + d_2 \cdot Q + d_3 \tag{4}$$

where:
$c_1,..,c_x$ - polynomial coefficients for the pump efficiency-flow characteristic;
$d_1,..,d_x$ - polynomial coefficients for the pump power-flow characteristic.

It is to mention the remarks:
(a) The pipe polynomial form of the head-flow characteristic can be represented like by the equation (5), considering that part of the dynamic losses depends only of armatures geometrical characteristics, materials, dimensions and surfaces qualities, but another part of these losses depends on the adjustment operation armatures positions. There are accepted the same notation as for equation (2).

$$H_r(Q) = (b_1 + b_0)Q^2 + b_2 \tag{5}$$

(b) In several other conditions, the pumps and system characteristics could have different aspects, but for the definition purpose it is considered the present representation.

(c) In the picture, the working point is considered the most efficient operation point of the system. It could be other operation conditions, characterized by other parameters, and the pump efficiency could be lower then the maximum value.

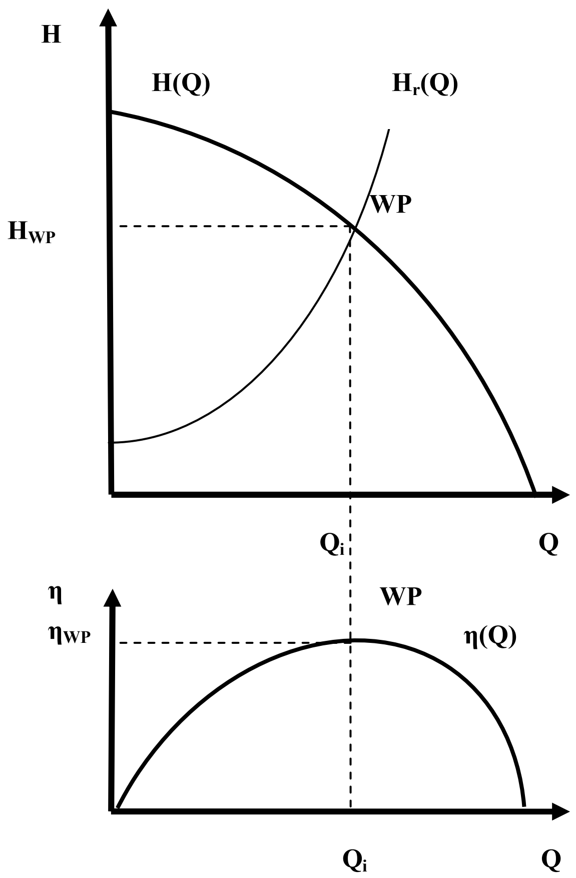

Fig. 1.Operation point representation

For generalization purposes, in the present paper, all the pump parameters are represented in non-dimensional coordinates, defined as follows (6):

$$q(\%) = \frac{Q_x}{Q_n}, h(\%) = \frac{H_x}{H_n},$$
$$\eta'(\%) = \frac{\eta_x}{\eta_n}, p(\%) = \frac{P_x}{P_n} \qquad (6)$$

where:

Q, H, η, P - operation parameters (flow rate, pumping head, efficiency and hydraulic power);

x - notation for nominal operation parameters; notation for momentum operation parameters;

q, h, η', p -non-dimensional operation parameters (flow rate, pumping head, efficiency and hydraulic power).

2.2.Working point operation adjustment using variable speed driving solution

If for the maximum efficiency of the pumping equipments at the nominal working regime is hardly to obtain significant improvements, taking into consideration the actual performances of the devices, there is an important potential to be valuated in pumping systems operation optimal adjustment.
There are different ways to operate adjustments of pumps working point, depending to the period of regime changing, the pumps power and dimensions, pumps and installation type, adjustment sharpness, etc.

For definitive operation point changing, it is preferable to action on the pump itself, by using different types of impellers in the same casing, the same impeller in different casings, or providing a permanent changing to a standard pump type.

For operating point changing for a long period of time, it could be used different types of diaphragms at the pump outlet or in appropriate points of the network.

For short term changes, it is preferable to use special adjustable devices of the installation (vanes, by passes), of the driving engines (variable speed motors), or pump itself (adjustable impellers for axial flow pumps, or same diagonal pumps).

Generally, the challenge is between adjustable vanes and variable speed motors for shot term changes in the majority types of installations.

For variable speed driving motor and using the same notations, the pump polynomial form of parameters characteristics become: head-flow characteristic - (7), Efficiency-flow characteristic – (8), power-flow characteristic – (9).

$$H_p(Q,N) = a_1 \cdot Q^2 + a_2 \cdot Q \cdot N + a_3 \cdot N^2 + a_4 \cdot Q + \\ + a_5 \cdot N + a_6 \qquad (7)$$

$$\eta(Q,N) = c_1 \cdot Q^2 + c_2 \cdot Q \cdot N + c_3 \cdot N^2 + c_4 \cdot Q + \\ + c_5 \cdot N + c_6 \qquad (8)$$

$$P(Q,N) = d_1 \cdot Q^2 + d_2 \cdot Q \cdot N + d_3 \cdot N^2 + d_4 \cdot Q + \\ + d_5 \cdot N + d_6 \tag{9}$$

where:
N - rotation speed at the operating point.

In non-dimensional coordinates and with using the non-dimensional rotation speed n, according to (10), the equations (1), (2), (3), (4), (5), (7), (8), (9) become the equations (11)- (17).

$$n = \frac{N_x}{N_x} \tag{10}$$

where:
N_x - rotation speed at the momentum operating point;
N_n - rotation speed at the optimum operating point.

$$h(q) = a_1^{'} \cdot q^2 + a_2^{'} \cdot q + a_3^{'} \tag{11}$$

$$h_r(q) = \left(b_1^{'} + b_0^{'}\right)q^2 + b_2^{'} \tag{12}$$

$$\eta^{'}(Q) = c_1^{'} \cdot q^2 + c_2^{'} \cdot q + c_3^{'} \tag{13}$$

$$p(q) = d_1^{'} \cdot q^2 + d_2^{'} \cdot q + d_3^{'} \tag{14}$$

$$h(q,n) = a_1^{'} \cdot q^2 + a_2^{'} \cdot q \cdot n + a_3^{'} \cdot n^2 + a_4^{'} \cdot q + \\ + a_5^{'} \cdot n + a_6^{'} \tag{15}$$

$$\eta^{'}(q,n) = c_1^{'} \cdot q^2 + c_2^{'} \cdot q \cdot n + c_3^{'} \cdot n^2 + c_4^{'} \cdot q + \\ + c_5^{'} \cdot N + c_6^{'} \tag{16}$$

$$p(q,n) = d_1^{'} \cdot q^2 + d_2^{'} \cdot q \cdot n + d_3^{'} \cdot n^2 + d_4^{'} \cdot q + \\ + d_5^{'} \cdot n + d_6^{'} \tag{17}$$

The equations coefficients for non-dimensional parameters take the indices $\left(m_i^{'}\right)$.

In practice, for variable speed driving motor, it is preferable to generate each working point using affinity lows [1], applied the ratio between current speed and nominal speed for different working point of the initial head-flow and power-flow characteristics and calculating the efficiency for each point.

$$\frac{H_x}{H} = \left(\frac{N_x}{N_N}\right)^2, \ \frac{Q_x}{Q} = \left(\frac{N_x}{N_N}\right), \ \frac{P_x}{P} = \left(\frac{N_x}{N_N}\right)^3 \tag{18}$$

$$\frac{h_x}{h} = \left(\frac{n_x}{100}\right)^2, \ \frac{q_x}{q} = \left(\frac{n_x}{100}\right), \ \frac{p_x}{p} = \left(\frac{n_x}{100}\right)^3 \tag{19}$$

The equations (18) and (19) represent the dimensional and non-dimensional affinity low expressions for turbo pumps.
For non-dimensional parameters equations the optimum operation point parameters values are considered 100%.

3. Controlled operation parabola definition and computation for special purposes

3.1.Controlled-operation characteristic definition

The controlled-operation characteristic is defined [5] as a theoretical curve along which the operating point variation is limited, in order to provide the most efficient operation with the restriction to ensure that from the minimum to the nominal flow rate, even at variable seep driving, there is always sufficient pump head available to cover the piping pressure losses and the useful pressure at the consumer installation.
The practical controlled-operation characteristic application is in any automatic pumping system operation design, but mainly for parallel operation pumps stations, where working point adjustment is realized modifying the different individual pumps working points, delivering on the same pipe. For one or more operating pumps, the most effective working point should be the crossing point of the pump(s) head-flow characteristic with the similar pipe characteristic, but some momentum specific operation condition could produce damages, then it is important to ensure the operation safety, working only above the controlled operation curve.
The controlled-operation characteristic is a system characteristic and its practical definition depends on the specific installation and pumps characteristics.
The difference between head-flow characteristic and the controlled-operation characteristic rise from the adjusted with the necessary head for the operation controllers, variable speed devices, and some more controlled vanes.

3.2.Controlled-operation characteristic particularization

Generally, this characteristic is considered a second degree curve, which could be defined by two points.
One of them is the optimum operation pump(s) point, common with the head-flow pipe characteristic [11]. For the second point, there are specific installation condition to be considered, and also specific operation restriction generated by the power characteristic, minimum admitted efficiency, pump characteristic aspect, or parameters variation consideration.

Examples
For a heating system station with two active parallel pumps, the controlled-operation characteristic is presented in fig.2, with the notation $h_{OC}(q)$. One operation point is placed at the intersection of the pumps system and the pipe head-flow characteristics, $B_N(100\%.100\%)$. Then, to attend the points of the controlled-operation characteristic, the pipe characteristic is corrected with adjustment devices, in order to attend the safety operation condition. The maximum flow point of the controlled-operation characteristic is the intersection of the pumps (or pump) hear-flow characteristic with the pipe characteristic. The minim operating point, $h_{OC}(q_{OC}=0, h_{OC}=h_W)$, is obtained practically, for each case, imposing a reserve of power for the electric motor over-load.

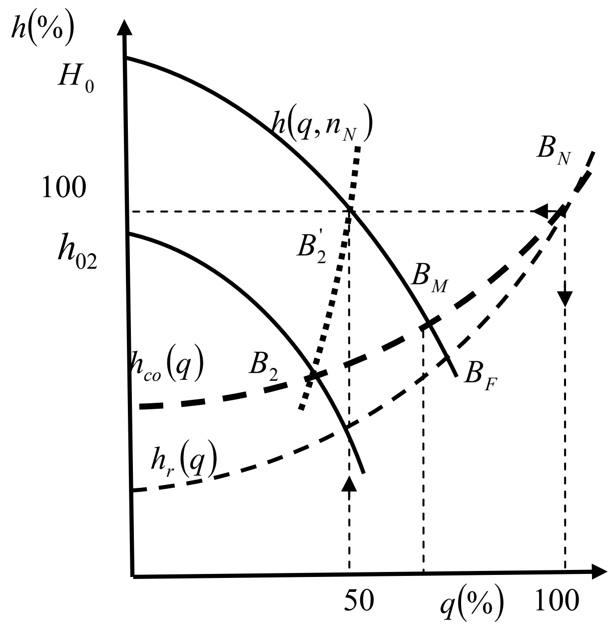

Fig.2. Controlled operation parabola

A usual specific condition is to ensure a minimum admitted efficiency at the minimum flow rate, realized by rotational speed reduction. In fig. 2, this point is B_2 .It indicate the crossing of the pump head-flow characteristic at a minimum rotational speed, with the affinity parabola through the optimal operation point of one pump $B_2^{'}(50\%.100\%)$ at nominal rotational speed being in the same time on the system controlled-operation characteristic.

Considering controlled-operation characteristic is a parabola through these two points, the intersection with head axis has the coordinates, $q=0$, and the head as follows [11]:

$$H_W = H_N - \left(\frac{H_N - H_{B1M}}{Q_N^2 - Q_{B1M}^2}\right)\cdot Q_N^2 \tag{20}$$

or, for non-dimensional parameters:

$$h_W = h_N - \left(\frac{h_N - h_{B1M}}{q_N^2 - q_{B1M}^2}\right)\cdot q_N^2 \tag{21}$$

Another limitation condition is to establish the necessary head at null flow, the value H_w itself. It can depend upon the following influencing factors: operating behavior of the consumer installation; similar load behavior over time or time-independent load behavior; system dimensions.

Following this limitations, the determination of one pump maximum accepted operation flow rate, Q_{1M} , is characteristic of the maximum accepted pump operation point $B_{1M}(Q_{1M}, H_{1M})$, which be positioned on the operation controlled characteristic. There are some additional steps for determining B_{1M} position.
Maximum flow operating, $B_{1M}(q_{1M}, h_{1M})$, is placed on the pump head characteristic of nominal speed, $n=100\%$, and depends on the mentioned considerations and power characteristic.

If the station has not a stand-by pump, in the event of one pump failure, it is necessary that the remain pump characteristic ensure the pipe characteristic required, since otherwise the remaining pump will be overloaded. Then, the maximum possible operation flow is Q_F . It is characteristic of the maximum accepted pump operation point $B_F(Q_F, H_F)$, placed at the intersection of the pump head-flow characteristic at nominal rotation speed and head-flow characteristic of the piping system. This is an abnormal function situation, but it can be attended at damage situation.

4. Pumping Station with Parallel Operating Pumps

The division of the flow into several variable speed pumps is used in all applications where demand fluctuates substantially and where the following requirements must be met the minimization of power consumption and compliance with minimum flow rate [3]. The fine adjustment is achieved by infinitely variable speed adjustment of one or more centrifugal pumps. In this process, the pumps operating are limited by several specific characteristics: power reserve, some unstable part of the head-flow characteristic, unaccepted low efficiency. Then, the pump could not attend the installation characteristic curve points.

The pumps parallel operation head-flow characteristic is obtain summarizing the flow rates offered by the two, or more, functioning pumps at the same head requested by the pipes network.

Considering the same mentioned operation hypothesis for a station with two pumps, the function condition is, as in fig. 3, the flow equation is

$$Q_p = Q_1 + Q_2 \qquad (22)$$

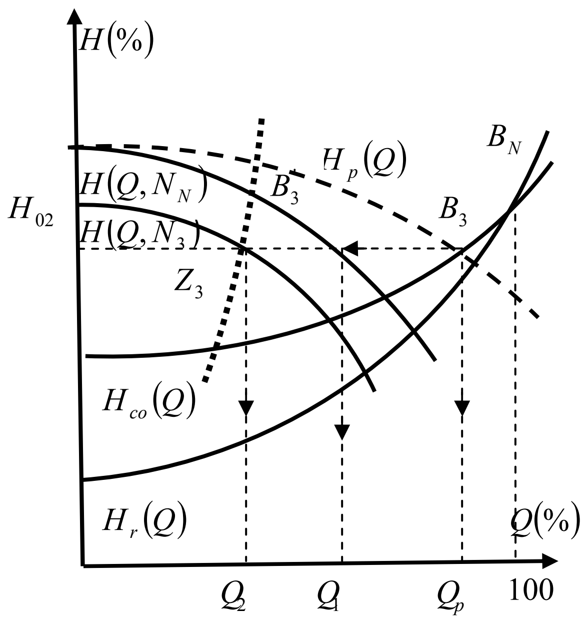

Fig.3. Parallel operation characteristic

where:

Q_p – current station operation flow;

$Q_{1,2}$ – current pumps 1,2 operation flows.

The other notations are similar to the notations from fig.2 [6].

The largest part of is offered by the pump working at the nominal speed, N_N considered Q_1, and the difference, by the pump working at the smaller speed, n_2, considered Q_2.

$$Q_2 = Q_p - Q_1 \qquad (23)$$

The optimum system operation characteristic is realized for the pumps working at the nominal rotation speed, N_N, with the optimum nominal operation point B_N, where $Q_p = Q_N$. This is one of the point which are defining the controlled-operation characteristic of the system.

The current system operation point is marked on the controlled-operation characteristic $B_3(Q_p, H_p)$, with Q_p requested by the user, and H_p is the corresponded head on the controlled-operation characteristic.

The first pump working point is $Z_3^{'}(Q_1, H_p)$, from the pump characteristic $H(Q, N_N)$. Consequently, the second pump working point will be, on the pumps characteristic $H(Q, N_3)$, with $N_3 < N_N$. In order to estimate the second pump necessary speed, will be consider the intersection of the Affinity parabola through Z_3 and the pump characteristic (1).

$$H(Q) = \left(\frac{H_p}{Q_2^2}\right) \cdot Q^2 \qquad (24)$$

Results $B_3^{'}(Q_{B3}^{'}, H_{B3}^{'})$ and, from the affinity law

$$N_3 = \left(\frac{H_p}{H_{B3}^{'}}\right)^{\frac{1}{2}} \cdot N_N \qquad (25)$$

In non-dimensional coordinates, all the equations have the same structure, but the notations are changed according to the definitions (8).

The mentioned operation points reach the coordinates as follows:

$$B_3^{'}(q_{B3}^{'}, h_{B3}^{'}),\ Z_3^{'}(q_{Z3}^{'}, h_{Z3}^{'}),\ B_3(q_3, h_3)$$

The necessary rotation speed on variable speed divining pumps is:

$$n_3 = \left(\frac{h_{B3}}{h'_{B3}}\right)^{\frac{1}{2}} \cdot 100. \qquad (26)$$

The algorithm application is the automatic adjustment operation points of the pumps of the parallel ensemble, following the system requested flow.

5. Requested power estimation for two pumps parallel operation adjustment

5.1.Parallel operation variable driving speed requested power estimation

It is considered the ensemble with one fix speed driving pump and one variable speed driving pump. Knowing the flow rate for both pumps, it can be calculated the requested power from the power-flow characteristic (6) for nominal speed pump and, in addition, applying to the affinity laws (18) and (19) for the variable speed pump, as follows [5].

The fix speed driving pump power consumption P_{1N} for the requested flow rate Q_1 and nominal rotation seed N_N is computed as follows:

$$P_{1F}(Q_1, N_N) = d_1 \cdot Q_1^2 + d_2 \cdot Q_1 + d_3 \qquad (25)$$

The variable speed driving pump power consumption P_{2V} for the requested flow rate Q'_{B3} and the momentum rotation speed N_3 is computed as follows:

$$\begin{aligned} P_{2V} &= P'_2 \cdot \left(\frac{N_3}{N_N}\right)^3 = \\ &= \left(d_2 \cdot \left(Q'_{B3}\right)^2 + d_1 \cdot Q'_{B3} + d_0\right) \cdot \left(\frac{N_3}{N_N}\right)^3 \end{aligned} \qquad (26)$$

Then, the total power consumption of the two pumps assembly is:

$$P = P_{1F} + P_{2V} \qquad (27)$$

Considering the non-dimensional parameters, the total requested power becomes

$$\begin{aligned} p_t &= d'_1 \cdot q_1^2 + d'_2 \cdot q_1 + d'_3 + \\ &+ \left(d'_1 \left(q'_{B3}\right)^2 + d'_2 \cdot q'_{B3} + d'_3\right) \cdot \left(\frac{n_3}{100}\right) \end{aligned} \qquad (28)$$

For real power consumption is should be considered the equipments efficiency.

5.2.Parallel operation fix driving speed and vane adjustment power requested

For the ensemble with both pumps have fix speed driving motors, the necessary flow Q_2 could be obtained by changing the pipe characteristic with a vane. This is the most unfavorable procedure for operation adjustment.

One of the pumps is functioning at nominal operation point (maximum efficiency) and the other pump is working at momentum requested flow rate, but the operation point is achieved by flow throttling with a vane.

The requested power P_1^V is computed with equation (25) and the second pump operating point is situated on the initial head characteristic, for nominal speed. Then

$$P_2^V = d_2 \cdot \left(Q'_{B3}\right)^2 + d_1 \cdot Q'_{B3} + d \qquad (29)$$

The total requested power is

$$\begin{aligned} P^V(Q_1, N_N) &= d_1 \cdot \left(Q_1^2 + Q'^2_{B3}\right) + \\ &+ d_2 \cdot \left(Q_1 + Q'_{B3}\right) + 2 \cdot d_3 \end{aligned} \qquad (30)$$

and

$$\begin{aligned} p^V(q_1, 100) &= d'_1 \cdot \left(q_1^2 + q'^2_{B3}\right) + \\ &+ d'_2 \cdot \left(q_1 + q'_{B3}\right) + 2 \cdot d'_3 \end{aligned} \qquad (31)$$

the equation (31) is calculating the non-dimensional requested power.

6. Energy efficiency evaluation method for pumping systems and equipments

The method consists in comparing current operation flow adjustment energy consumption with the most disadvantageous adjustment solution that is the vane utilization. There are compared two situations in pumping station design: one variable speed pump, working in parallel with one or more fixed speed pumps; all the operating pumps have fix rotation driving motors and the adjustment of the flow rate is obtain by throttling the flow with a vane. In the same way could by analyzed the situation of more then one variable speed pumps parallel operating.

Generally the pumps are similar, and this is the example analyzed, but they could be different also.

The savings can be estimated considering the necessary and the consumed power for all the pumps working in the same time together.

For the method demonstration there are considered two identical pumps working in parallel. One of the pumps is remaining at the nominal operation point and the nominal rotation speed. The flow rate adjustment is realized by throttling the flow for the other by maintaining the nominal rotation speed and throttling the flow, or by varying the rotation speed and maintaining the vane at the complete opening and minimum throttle of the flow. In fig.4, it is presented the variable requested flow pump providing by the two described procedures, for the adjustment operation pump.

The figure notations are as follows:

$H(Q,N_N)$ - pump head-flow characteristic at nominal operation point;

$H(Q,N_n)$- pump head-flow characteristic at momentum operation point;

$H_r(Q,x_0)$- piping system head-flow characteristic at the complete opening and minimum throttling of the vane;

$H_r(Q,x_i)$ - piping system head-flow characteristic at a momentum opening and throttling of the vane;

$WP_1(Q_i,H_{WP1})$- initial operation point of one pump;

$WP_2(Q_r,H_{WP2})$- pump operation point at the requested flow rate Q_r, obtain by throttling the flow with a vane;

$WP_3(Q_r,H_{WP3})$- pump operation point at the requested flow rate Q_r, obtain by varying the rotational speed;

η_{WP1} - pump efficiency at the initial operation point;

η_{WP2} - pump efficiency at the requested operation flow rate, obtain by throttling the flow;

η_{WP3} - pump efficiency at the requested operation flow rate, obtain varying the rotational speed.

The other flow rate is considered constant.

For the whole system, the fix flow operation pumps power is summarized with the variable flow pump, whatever should be the adjustment procedure[12].

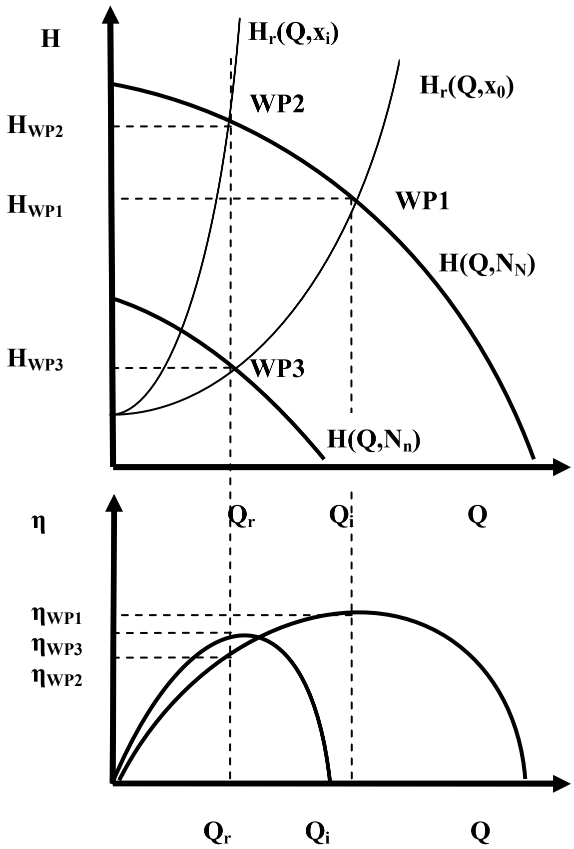

Fig.4. Vane and variable speed operation adjustment

The electrical consumed power, P_{eli}, provided by the electric network, takes into consideration both the utile power consumption and the momentum electric motor operation efficiency, η_{EMi}, where *i* is the indice of the specific operational point[13].

Vane adjustment efficiency

For the requested flow rate, Q_r, the utile power is the met in the crossing point of the vertical line through Q_r and the piping system head-flow characteristic at the complete opening and minimum throttling of the vane, $H_r(Q,x_0)$, notated P_{WP3}. The real consumed power from the pump is P_{WP2}, in the working point *WP₂*, the crossing point of the vertical line through Q_r and the pump head-flow characteristic at nominal operation point, $H(Q,N_N)$.

$$P_{WP2} = \rho \cdot g \cdot Q_r \cdot H_{WP2} \tag{31}$$

$$P_{WP3} = \rho \cdot g \cdot Q_r \cdot H_{WP3} \tag{32}$$

The consumed power in the mentioned working point is computed considering the specific efficiencies.

$$P_{C2}^{Vane} = \frac{P_{WP2}}{\eta_{WP2}} = \frac{\rho \cdot \mathrm{g} \cdot \mathrm{Q_r} \cdot H_{WP2}}{\eta_{WP2}} \tag{33}$$

$$P_{C3}^{Vane} = \frac{P_{WP3}}{\eta_{WP2}} = \frac{\rho \cdot g \cdot Q_r \cdot H_{WP3}}{\eta_{WP2}} \tag{34}$$

The vane adjustment efficiency (η_v) is

$$\eta_v = \frac{P_{C3}^{Vane}}{P_{C2}^{Vane}} = \frac{\rho \cdot g \cdot Q_r \cdot H_{WP3} \cdot \eta_{WP2}}{\eta_{WP2} \cdot \rho \cdot g \cdot Q_r \cdot H_{WP2}} = \frac{H_{WP3}}{H_{WP2}} \tag{35}$$

where the efficiency could be expressed as

$$\eta_h^{Vane} = \frac{H_{WP3}}{H_{WP2}} \tag{36}$$

is defined as hydraulic efficiency of the vane adjustment procedure. Then the vane adjustment efficiency procedure is defined as a hydraulic efficiency.

Variable speed engine adjustment efficiency
Both the utile and consumed power are computed for the point WP_3 and the electric consumed power is computed as follows

$$P_{C3}^{Speed} = \frac{\rho \cdot g \cdot Q_r \cdot H_{WP3}}{\eta_{WP3}} \tag{37}$$

It is to mention the electric motor efficiency which vary with the requested power, according to the specific working points.

Variable speed adjustment efficiency compared to the vane adjustment
They are compared the consumed power adjusting the flow with a vane$\left(P_{C2}^{Vane}\right)$, and the consumed power adjusting the flow with a variable speed device$\left(P_{C3}^{Speed}\right)$, by defining a specific power saving, (ΔP_{SV}), and a specific efficiency (η_{SV}) [4].

$$\Delta P_{SV} = P_{C2}^{Vane} - P_{C3}^{Speed} = \rho \cdot g \cdot Q_r \cdot \left(\frac{H_{WP2}}{\eta_{WP2}} - \frac{H_{WP3}}{\eta_{WP3}} \right) \tag{38}$$

$$\eta_{SV} = \frac{P_{C3}^{Speed}}{P_{C2}^{Vane}} \frac{\rho \cdot g \cdot Q_r \cdot H_{WP3} \cdot \eta_{WP2}}{\rho \cdot g \cdot Q_r \cdot H_{WP2} \cdot \eta_{WP3}} = \eta_h^{Vane} \cdot \frac{\eta_{WP3}}{\eta_{WP2}} \tag{39}$$

Energy savings are computed considering the operation duration of he pumps (t)

$$\Delta E = t \cdot \Delta P_{SV} = t \cdot \left(P_{C2}^{Vane} - P_{C3}^{Speed}\right) = t \cdot \rho \cdot g \cdot Q_r \cdot \left(\frac{H_{WP2}}{\eta_{WP2}} - \frac{H_{WP3}}{\eta_{WP3}} \right) \tag{40}$$

The total savings should take into consideration the total electrical system [9].

$$\Delta E_{total} = t \cdot \Delta P_{SV}^{Total} = t \cdot \left(\frac{P_{C2}^{Vane}}{\eta_{EM2}} - \frac{P_{C3}^{Speed}}{\eta_{EM3}} \right) = t \cdot \rho \cdot g \cdot Q_r \cdot \left(\frac{H_{WP2}}{\eta_{EM2} \cdot \eta_{WP2}} - \frac{H_{WP3}}{\eta_{EM3} \cdot \eta_{WP3}} \right) \tag{41}$$

The electricity price it is not consider in the efficiency computation because the general results could be disturbed by the price variation. In practice, the final result is presented considering the momentum financial effects also.

7. Specific influences on the automatic operation modelling

Influence of the System Design
The operating point of a centrifugal pump or group of pumps is always the point of intersection between the system characteristic curve and the pump characteristic curve. All control methods thus change either the pump or the installation curve.
The installation characteristic curve denotes the pressure requirement of the installation depending the flow rate. It always contains dynamic components that increase quadratiqually with the flow rate due to the flow

resistances – for example in circulatory installations (heating, cooling etc). However, it may also incorporate additional static components, such as differences in geodesic head or pressure differences caused by other factors – for example in transport systems (pressure boosting). In circulatory installations characteristic curves has no static components and thus begins at the origin *(H=0)*. In practice, to prevent consumer installations being undersupplied, the necessary pressure graph lies above the system characteristic curve. Its precise path is dependent upon the system in question. The controlled operation curve, along which the operating point should move, must consequently lie on or above the necessary pressure line [3].

Influences as a result of the loading of the system over time

The flow rate Q of the centrifugal pump system can, in the most extreme case, fluctuate between a maximum value and zero. If we order the required flow rate over a year according to size we obtain the ordered annual load duration curve. Its precise path is dependent upon the system in question and can differ from one year to the next one.
The longer the operating period and the smaller the area between the curves, the greater is the potential for the possible savings.
For example, the pump is designated for 100% flow rate. This output is seldom required in the year. Most of the time, a lower flow rate is required. To save pump driving power, the control system automatically matches the pump speed to the momentary system demand.
Fig. 5 presents two load duration alternatives in order to calculate the energy saving taking into consideration the real operation duration of the system.

Influence of the pump or group of pumps

The pump can influence the extent of possible savings realized by pump control in different ways: by the path of its characteristic curve, by the different motors size required and by the design of the pump.
The pumps solution is important for the different types of internal losses, but also for the characteristic curves steepness.
The graph of the pump input power depends upon the gradient of the head and the graph of the efficiency.
In general, the steeper of pump characteristic curve, the falter the power characteristic curves.
The motor size of a pump unit has influence, since experience tells that the ratio of investment to motor size falls as power increases.
In multi-pump systems the economy calculations performed according to the same way as described bellow [4].

The pump power consumption from the electric grid

Generally, the pumps behaviour is evaluated considering the input power (shaft power) of the pump. However, the more specific and correct computation have to consider the electric power absorbed from the electric grid.
The power consumption $\left(P_{eg}\right)$ in fixed speed operation is increased in relation to the pump shaft power $\left(P_{wg}\right)$ by the motor losses.
The power consumption in variable speed operation is determined by the shaft power plus the losses of the frequency inverter plus the motor losses (which may increase slightly depending upon the frequency inverter type).
The additional losses as a result of variable speed operation are negligible, since a power saving is achieved as soon as the flow rate falls bellow appreciatively at 95% compared to fixed speed operation.

For practical applications, it is not necessary to determine the power consumption in details. It is fully adequate to base the calculation upon the pump input power (shaft power). This is because, as it is presented in fig.5, the absolute electrical power losses in variable speed and fixed speed operation are almost identical.

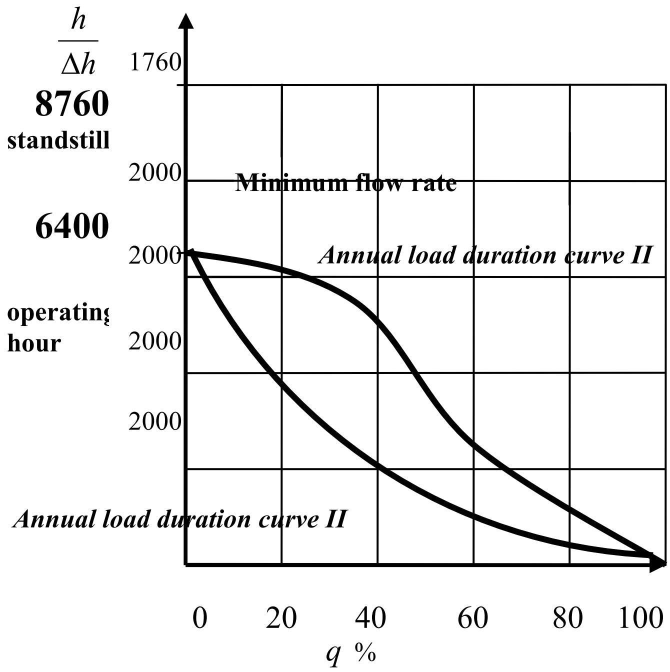

Fig.5. The annual load duration in two situations

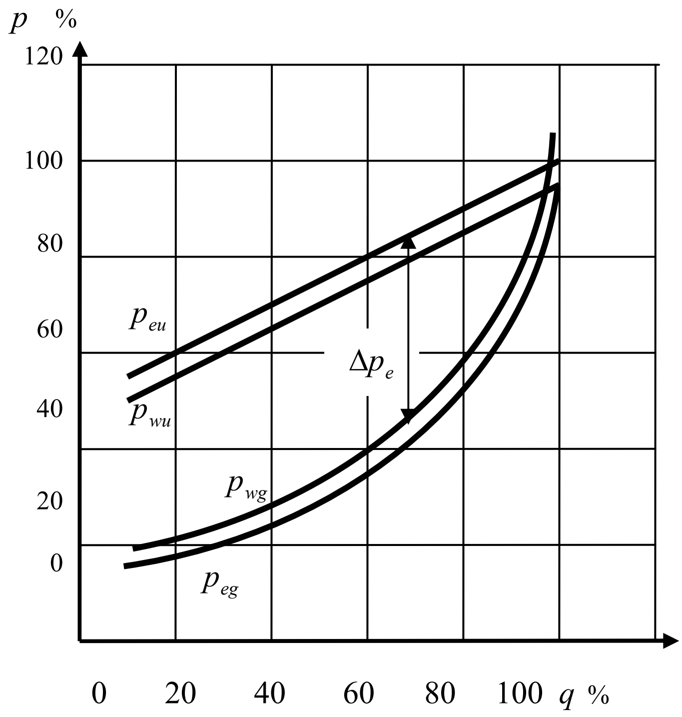

Fig.6. Example of saved electric power

The automatic operation methods of the pumping systems are using analytical representation of the power head-flow, power-flow and efficiency-flow representation, the load-flow diagrams in dimensional coordinates, which are similar to the non-dimensional ones previously presented [4].

8. Green house Gases Mitigation result by using variable speed driving devices

The most important ecological effect of the energy saving is the greenhouse gases emission reduction, that consists one of the European actual challenging, following the recent regulations and commitments [7].
Considering the conventional fuel burning CO_2 emission, it can be computed the gaze emission reduction using the energy saving equation (41).

The caloric effect of one kilogram of conventional fuel is divided between the carbon and hydrogen components, considering: x – Carbon part quantity, (1-x) – Hydrogen part quantity [1].
According to the international norms, one kWh of electrical energy is produced by burning 350-360 g of conventional fuel.
Also, it is known that burning one kilogram of conventional fuel is producing around 7000 kcal.

Then, considering the Carbon and Hydrogen caloric capacity and molecular mass

$$Q = 8{,}100\frac{kcal}{kgC}\cdot x + 28{,}000\frac{kcal}{kgH_2}\cdot(1-x) = 7{,}000\,\tfrac{kcal}{kg} \tag{42}$$

where, for one kilogram of conventional fuel

$$x = 0{,}8\frac{kgC}{kg} \tag{43}$$

Burning one kilogram of conventional fuel it is produced $\frac{44}{12}xkgCO_2$, and CO2 mitigation corresponding to the energy savings is

$$\Delta C(CO_2 / year) = 0.36\times\frac{44}{12}\times\Delta E_{total}\times 0.8 \tag{44}$$

The procedure is applicable for any energy saving procedure, including variable speed driving of pumps and other hydraulic machines.

9. Conclusion

The proposed energy saving evaluation method have practical application in such as:
- Greenhouse gases mitigation evaluation of the efficient pumping solutions;
- Being based on the energetic efficiency maximization, the procedure represents the optimal condition for automatic operation of all types of pumping systems.

Energy savings using variable speed engine adjustment can be estimated, considering the period of operation time during one year, the pumps number and power and efficiency. At national level, there are not reliable studies of the present situation in pumping installation efficiency. Even at the European level, the developed countries are just starting such types of inventories. But, a limited energy saving of about 10%, could lead to a general energy saving at national level of about 2%, which means a huge effect at European level.
Considering the relation (38) of the total energy savings, it can be represented in non-dimensional coordinates $(\Delta p, q)$. In fig. 7, it is presented a demo diagram for a parallel operation pumping system composed by two identical pumps, with operation adjustment as describe in § 6. The savings are null for the optimal operation at the initial flow rate and is growing when the requested flow rate is decreasing [8].

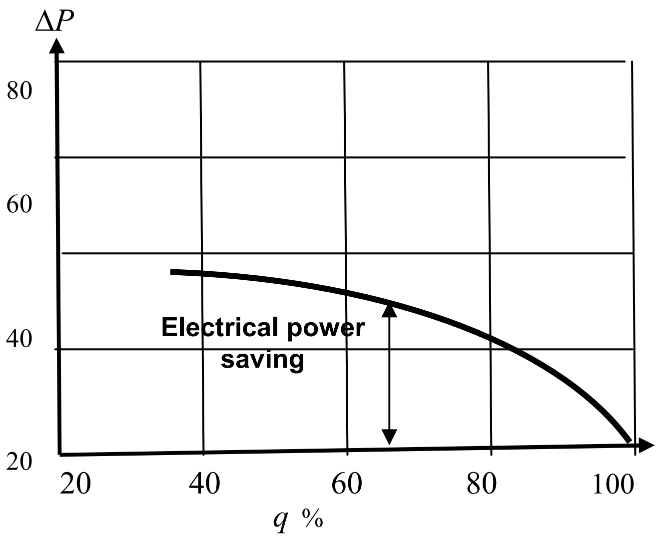

Fig.7. Explanatory plotting electrical power saving diagram

The most important ecologic effect of the energy efficiency application is the greenhouse gasses mitigation, reducing the electrical energy consumption and, by consequence production [10]. Burning fossil fuels produces CO2 emissions of roughly 0.53 kg per 1 kWh of electrical energy produced. Thanks to the drastic reduction in power consumption, the new system therefore is a positive contribution to environmental protection.

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