

Energy Efficiency of Machinery Hydro-Pneumatic and Climate Impact

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Abstract- In practice, due to various over-the loss of pregnancy considered by designers as a security measure, the pump provides a flow chosen much higher than calculated. The main greenhouse gases (GHGs) producer is represented by the fossil burning systems. The paper presents the effect on the GHGs emission combined with the economic effects of a new pumping system method of automatic mixing of hot and cold water circulated with variable speed driving pumps and adjustable vanes. Considering the potential economic and environmental effects combined with the social one, The heating systems represent a sensitive sector of the energy consumers' basket, strengthen by the unpredictable primary energy price fluctuation, the recent financial crisis and the population expectations. This paper tries optimizing operation of machinery, hydro-pneumatic.

Key-Words: - Modeling and optimization, Greenhouse gases, Heating systems, Economic effects.

1 Introduction

In the actual international and European frame, it is important to accelerate the development and deployment of cost-effective low carbon and energy efficiency technologies. The European Union considers an absolute priority the GHGs emission mitigation, in regulation and in technical initiatives frame for low carbon technologies improvement and generalization [1].

One specific European initiative is the commitment to reduce with 20% GHGs emission by the year 2020. One of the most important solution proposed by the European Commission are the Energy efficient technologies, which tend to have high upfront costs. From energy changing point of view, the attention is focused to hydraulic generator and electrical motor characteristics, referring to the energy efficiency and general behavior.

The Eastern European Countries have a special situation, considering the heating systems exceptional diversity of dimensions, caloric power necessities, fuel, technical solutions, age, ownership and technical disposal, as a result of the chaotic and rapid rehabilitation and modernization process, after 1991. If the large energy producing units were

more or less modernized on different types of funds, especially by Joint Implementation projects after the Kyoto Protocol ratification by many potential donor and host countries, the secondary heating installations are not in the light of investors and authorities. In the conclusion there are analyzed the ecologic and economic energy reduction effects. In the last years, the heating solutions register a dynamic evolution in two directions: utilization of the renewable energy sources; modernization of the producing and utilization equipments. It remains an important part of poor people clients of the old central heating systems, but some of them try to avoid even a normal exploitation. This situation produces unbalances and disturbs the central systems operation, reducing drastically their efficiency. A good opportunity for both individual consumers and central systems is a consistent improvement of their efficiency, by rehabilitating the existing solutions and adopting new technical efficient solution.

2 The Heating Systems

Conventional, mixing or injection-type, systems heating set-ups characteristics are:

- The circulator pump and the control valve fitted in operate independently, without coordination between the two components.

- There is no system knowledge about the hydraulic conditions in the heating circuit, so the hydraulic energy supplied may be destroyed by control elements.

- Discharge head of the variable-speed circulator pump remains constant, irrespective of the actual load conditions, i.e. irrespective of external temperature.

Systems produce a constant volume flow rate through the consumer installations. As a result, part-load conditions, which account for more than 95% of the operating period, result in the circulator pump handling predominantly cold return water most of the time.

In the systems with water recirculation, the volume flow in the supply line is coupled with the volume flow of the cold return water pumped through the mixing line, as seen in Fig. 1.

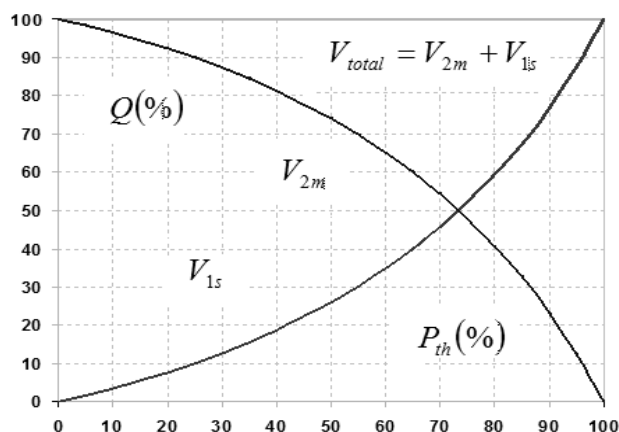


Fig. 1: Static characteristics of pump and installation.

$$Q_{total} = Q_{1s} + Q_{2m} \quad (1)$$

where:

Q_{total} - radiator total flow,

Q_{1s} - supply flow from the kettle,

Q_{2m} - mixing flow transferred from by-pass.

Thermal output of the heating circuit can be adjusted according to the temperature differential. Then, predominantly cold return water is pumped through the heating circuit under part-load conditions.

3 Methods of Adjustment

It is assumed that the initial network characteristic curve calculated and determines the operation of a recirculation pump is provided in the operating point A, point near optimal efficiency (Fig. 2a).

Real network generates fewer miscarriages and actual operating point will be point B, with flow $Q_B > Q_A$ (Fig. 2b) [11], [13].

Opportunities to return to the desired flow Q_A are:

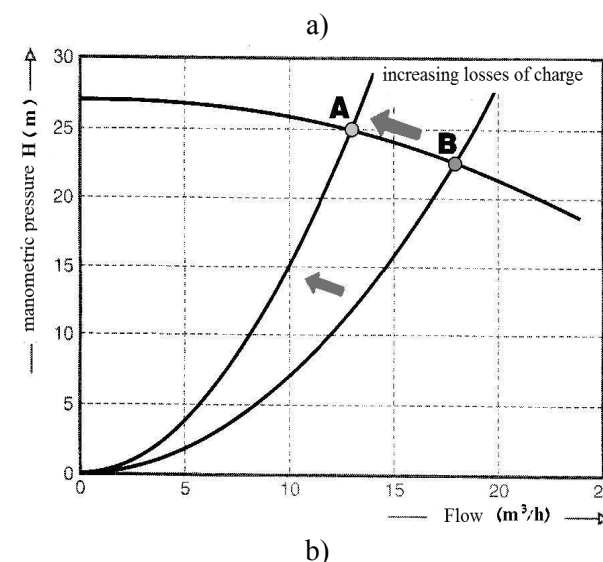
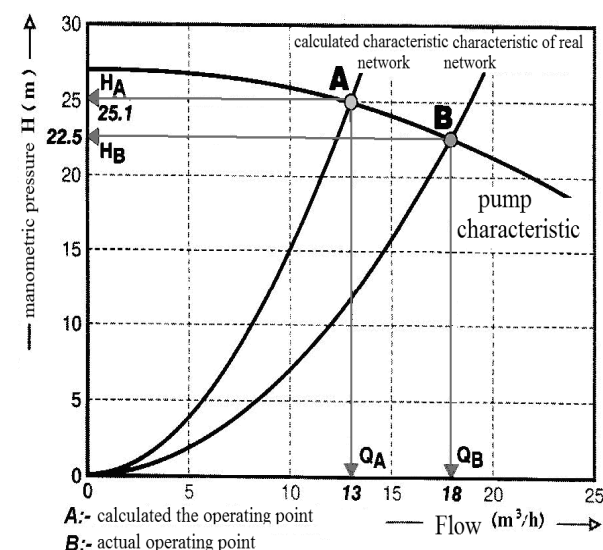


Fig. 2: Explanatory flow adjustment.

1. *Obstruction by a valve.* By closing its progressively increasing load losses, the characteristic redresându network by focusing near the operating point B to point A (this is achieved by thermostatic valves fitted to radiators). Also, to achieve additional burden may be placed between 2 clamps on a diaphragm pump repression [3], [8].

The advantage of using this method are:

- is very simple and provides a gradual adjustment (made of bath), without major investment;
- to achieve a slight reduction in energy consumption of the pump.

The limitations of the method are given by the fact that this solution adjustment determined, in comparison with other solutions, a large waste of energy.

1. *Decrease rotor diameter machine.* Changing the rotor diameter is established at a lower pump characteristic feature of the network at the point C, point of operation with the same rate as the point A (Fig. 3).

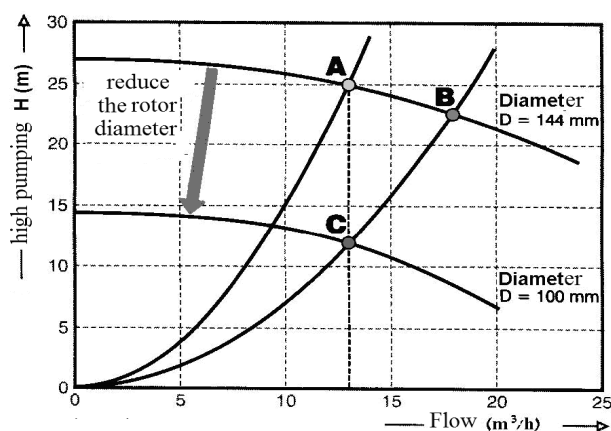


Fig. 3: Explanation of flow adjustment first hydro machine rotor diameter change.

Advantages:

- simple and inexpensive solution (rotor being provided by the manufacturer), applicable pump power between between 5 and 20 kW;
- saving more than obturării method.

Disadvantages:

- immobilization pump for removal and changing of the rotor;
- beach limited adjustment (wheels can not be reduced only to a certain limit) - in case of excessive contraction, the rotor will be replaced with a new one - small loss of hydraulic efficiency.

3. *Change speed* [4], [6]. This method, like the previous one, is based on reducing the speed of the pump characteristic curve in a way that it passes through point C. Thus, a rotational speed of 2117 rpm results in a characteristic of the pump virtually identical to that obtained using a rotor diameter of 110 (Fig. 4). And so the same operating point C.

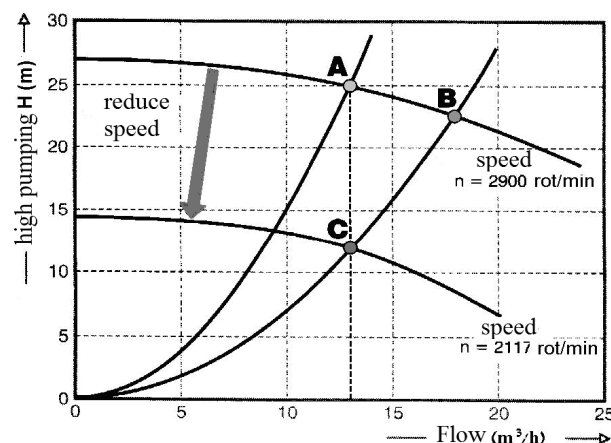


Fig. 4: Explanation of flow adjustment by changing the rotational speed of the pump.

Changing speed can be simple and a good functioning but requires expensive investment in relation to characteristics of the equipment.

Pump:

- some have more easily switch speeds (2 or 3 speed pumps with selector can be mounted in different positions);

- other (modern) are provided from the factory with variable speed - but others (in generally the oldest manufacturing) have only one speed. In this case, involving the purchase of a speed variation frequency converter, quite expensive. This is the most easily adaptable to the conditions of various flow regimes [1].

Benefits:

- to achieve a significant reduction in energy consumption from the point of view, the same order as the rotor diameter is reduced;
- adapting the exact desired flow depending on the situation (or pulley for variable speed fans).

Disadvantages:

- the solution is costly if necessary purchase of a variable speed;
- for switchable speeds, imperfect adaptability to supradabit.

1. *Obstruction and speed variation.* While other solutions are possible intermediate stop on the proposals manufacturers, ie pump speed integrated regulators. In this pump its rotational angular velocity decreases to the point of the network determined which allows maintaining a constant pressure. This is a solution used for circuits or radiators with thermostatic valves for circuits with two-way valves (Fig. 5).

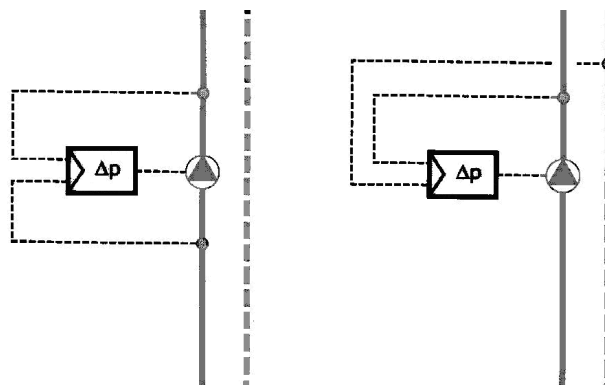


Fig. 5: Explanation for adjusting flow through clogged and flow changes.

In Fig. 6 to represent adequately the situation, moving the operating point of B in D is needed in differential pressure probe. In its absence, the increased pressure would be reduced when the flow rate regime would remain constant. If shown in Fig. 6a, the pressure remains constant but if hunt is closed, the flow of the network decreases, resulting in a decrease in pressure on the joint sections.

Therefore, some producers, believes that the decrease in flow rate with change can be achieved by reducing pressure linear (Fig. 6b).

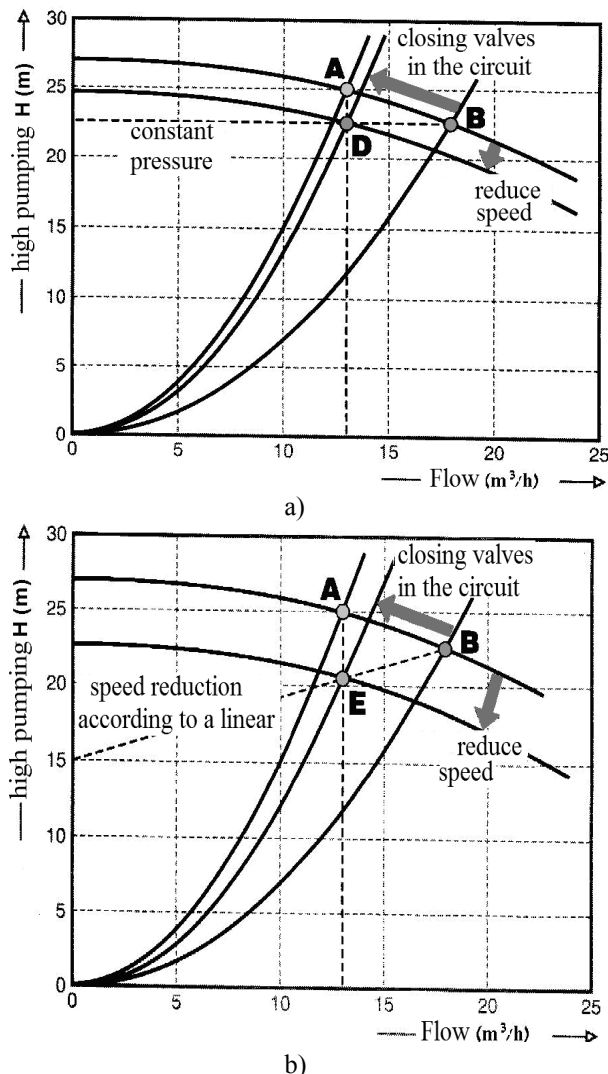


Fig. 6: Explanation of the pump rotational speed changes depending on pressure.

This solution (shown in the figure by shifting the operating point B to E), in terms of energy use is more economical than the previous one, agrees in most cases the circuits provided by thermostatic valves [2], [10], [12].

3.1 Influence of the Method of Adjustment Used on Energy Eonsumption

If you believe that yields are constant, hydraulic power absorbed is proportional to the product of pressure and flow Q and H is the rectangle area defined by the axes H , Q and operating points (eg A, B, C, ...).

It is noted that (Fig. 7):

- reducing the flow through seal (on point of operation of B in A) provides a slight decrease of the surface of the rectangle (not too big a leap), and hence power;

- to maintain constant pressure (operating point moving from B to D) is without a decrease in energy consumption;
- altering the characteristics of the pump or fan, by reducing the rotor diameter or decrease speed to maintaining the characteristic curve of the circuit, this providing a significant loss of hydraulic power (rectangle O, H_C , C, Q_A , compared with O, H_B , B, Q_B , or O, H_A , A, Q_A). A decrease in flow by 20% for a flow equal to 80% of flow rate, results in a decrease of 50% power ($= (80\%)^3$ under the laws of hydro-pneumatic machines).

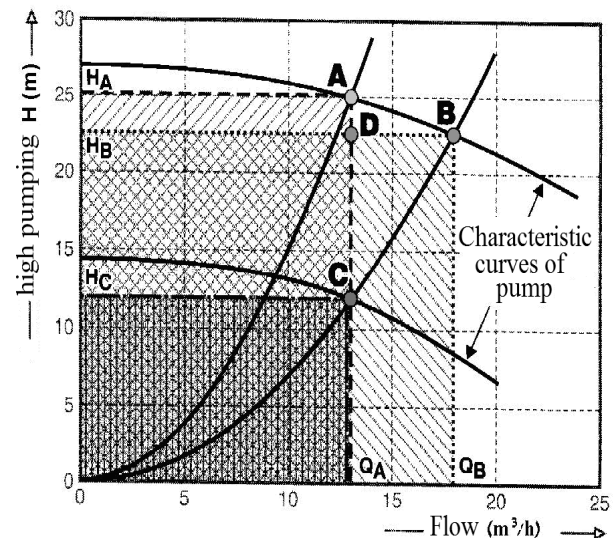


Fig. 7: Evaluation of power characteristic curves of centrifugal hydro-pneumatic machines.

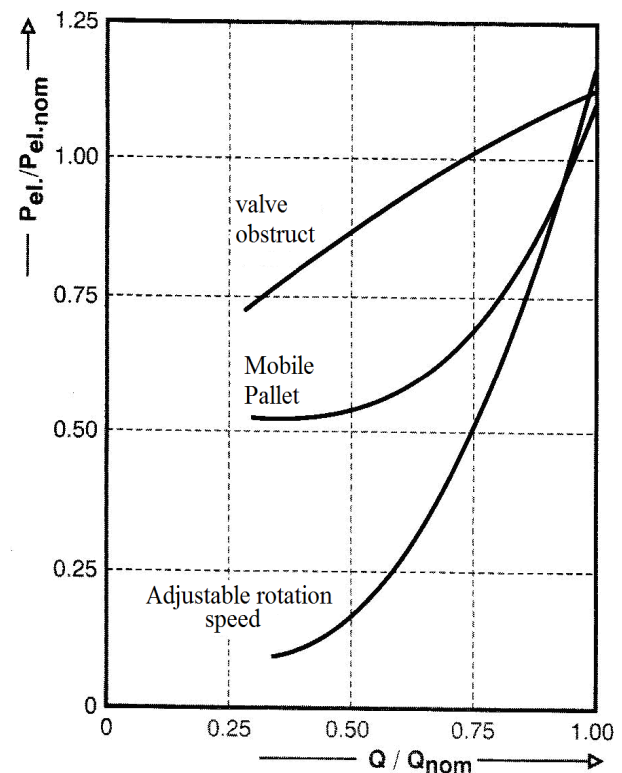


Fig. 8: Electric power absorbed when used methods of regulating hydro-pneumatic machines.

Figure 8 summarizes the reduction of consumption using different methods of adjusting the flow for a pump. Adjust flow by adjusting the angular velocity of rotation is considered the most elegant method, both in terms of energy consumption and vibration at low speeds [1], [5].

It is also the most expensive method, but we thank you and a decrease in speed by means of fixed (by changing drive pulleys).

In Fig. 9 is shown the variation of noise in decibels when adjustment methods described above.

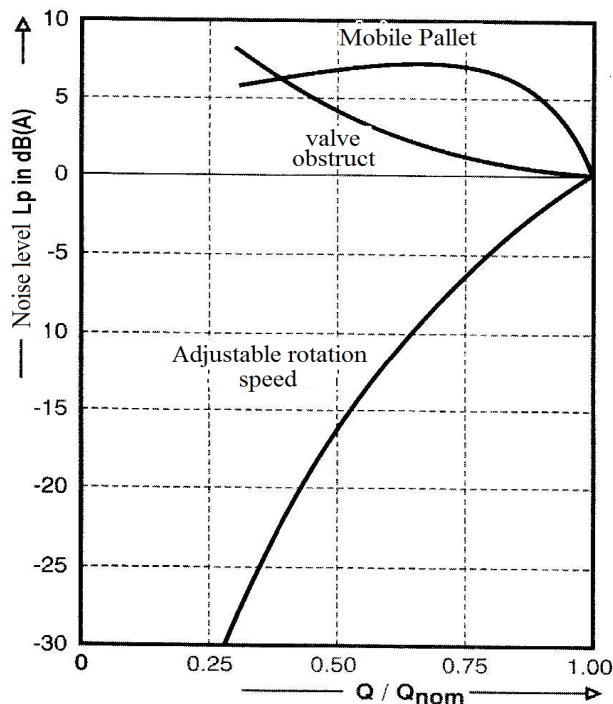


Fig. 9: The noise level for different methods of hydro-pneumatic adjustment centrifugal machine.

4 Innovative Pump Management System

The system is a control one (high controller), installed downstream of the heating system's control unit, as seeing in Fig. 10 [1].

It coordinates the operation of circulator pump and control valve. Depending on the control signal, the two control valves adjust the resulting volume flow rate pumped through the consumer installations. At the same time, the appropriate discharge head set point is transmitted to the variable-speed drive of the circulation pump.

It transforms the conventional, constant-flow mixing or injection-type system into a variable-flow system, as it is presented in Fig. 11, and adjusts the discharge head of the circulator pump to the reduced volume flow rates via the system control curve.

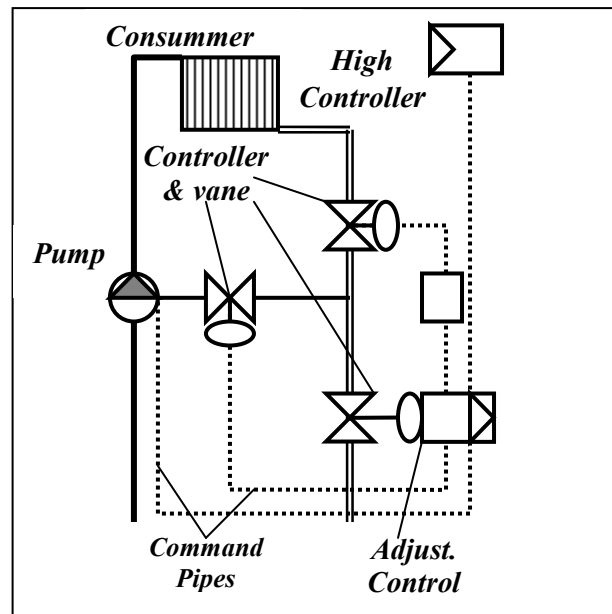


Fig. 10: The heating system.

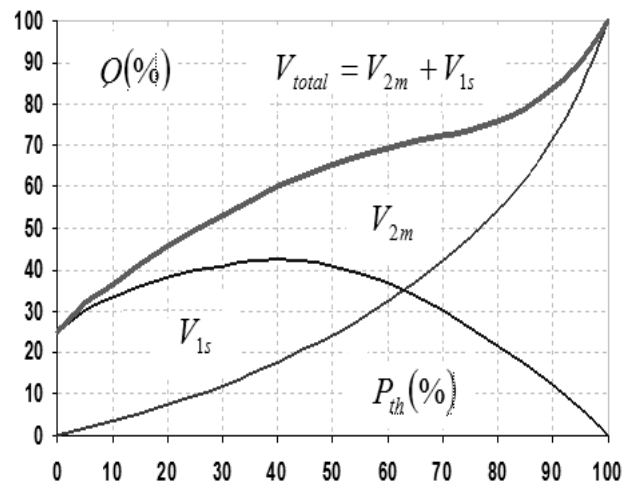


Fig. 11: Static characteristics of the pump.

Thermal output of the heating circuit, P_{th} , is calculated for the design point as the product of volume flow rate Q and temperature. Then:

$$P_{th} = 1.163 \cdot Q \cdot \Delta T \quad (2)$$

The innovative control concept consists in regulating thermal output by variation of the temperature differential and the volume flow rate, considerably reducing the water volume to be pumped through the heating circuit.

The radiator diagram illustrates the physical fact that, at identical heat output, the volume flow rates provided to the consumer installations can be reduced if the supply temperature is increased at the same time, using the higher-level controller. This results in hydraulic savings regarding both volume flow rate ΔQ and discharge head ΔH , then electrical power savings are:

$$P_{el} = \int K \cdot Q(t) \cdot H(t) \cdot dt \quad (3)$$

where:

P_{el} - electrical necessary power,

K - constant describing the efficiency of the pump,

$Q(t)$ - flow-time characteristic,

$H(t)$ - head-time characteristic.

Another positive effect is that static hydraulic balancing at the main feed manifold is now automatically performed via the circulator pump, which reduces commissioning costs for the heating circuit.

The new device of the system is the Higher-level control unit operation. The pressure drop across the valve may amount to several tenths of bar. Supply temperature control is the task of a higher-level controller. Input signals for this controller are, among others, the external temperature measured and the supply temperature measured in the heating circuit. Helping with the heating curve stored in the higher-level control unit, the set point for the supply temperature of the heating circuit is generated on the basis of the external temperature measured. Based on this set point and the supply temperature measured, the higher-level controller generates the control error signal, which is the input of a control algorithm.

This control algorithm generates a signal, which, in conventional systems, is then transmitted to the control valve. This output control signal from the higher-level controller is the input for control system, i.e. it is transmitted to the valves and pump control unit. The control unit translates the signal into two separate control signals for the two control valves and into a discharge head set point.

5 Mathematical Model of Hydro-pneumatic Machinery in Relative Coordinates

Dimensionless coefficients for pressure, flow and power output and relative coordinate drive motor speed of the car, defined by reports [7]

$$\frac{\Delta p}{p_0}, \quad \frac{d}{d_0}, \quad \frac{P_t}{P_{t_0}}, \quad v = \frac{\omega}{\omega_N} \quad (4)$$

where: Δp is total pressure developed hydraulic machine, defined as the difference between total pressure (static and dynamic) to output and total pressure at the entrance, p_0 is the maximum pressure developed by the machine at zero flow, d is the flow developed by car, d_0 is maximum flow

at zero pressure developed car, P_t is the effective power flow corresponding to pressure p and d ; P_{t_0} is effective power that would be obtained for a pressure p_0 and a flow d_0 ; ω is the absolute value and ω_N the nominal motor speed drive.

Mathematical model used to determine optimum operating point of the pump is obtained from the application of the laws of two hydro-pneumatic machinery namely:

- Act I, which shows that the flow developed by car d , is proportional to angular velocity ω drive it, assuming a constant values of fluid density ($d \approx \omega$);

- Act II, which shows that the total pressure Δp developed car, is proportional to the square drive angular velocity ($\Delta p \approx \omega^2$).

Hydro pneumatic machine characteristic equation

$$\Delta p = \left[v - \left(\frac{d}{d_0} \right)^2 \right] p_0. \quad (5)$$

Power output, developed the machine, fluid involvement will be needed once the product of flow and pressure

$$d \Delta p = d \left[v - \left(\frac{d}{d_0} \right)^2 \right] p_0. \quad (6)$$

6 Determination of Optimal Operation of Hydro-Pneumatic Machines

Phenomena in the hydro-pneumatic pumps or centrifugal fans used to work in a well-defined operating point on the curve characteristic of the machine.

The criterion of optimization. To assess the functioning machine is a criterion adopted energy, represented by achieving a maximum power output transmitted to the fluid in unit time (useful energy), expressed by function:

$$J(d) = d \Delta p. \quad (7)$$

Formulation of optimization problem. Optimization problem is to determine optimum operating point of the car (d^* , p^*) to ensure maximum energy required for fluid transport, evaluated by title:

$$J(d) = d \left[v - \left(\frac{d}{d_0} \right)^2 \right] p_0 = \max. \quad (8)$$

Necessary condition for optimum. Is obtained from the equation resulting from the cancellation of partial derivatives with respect to useful energy flow:

$$\frac{\partial(d\Delta p)}{\partial d} = \left(v - 3 \frac{d^2}{d_0^2} \right) p_0 = 0. \quad (9)$$

Optimal amount of flow and pressure. Solving the previous equation, for $v=1$ (nominal regime of operation), to obtain optimum flow value $d^* = \frac{d_0}{\sqrt{3}}$, which replaced in equation (4) will give optimum value of pressure $p^* = \frac{2}{3} p_0$, which means a maximum power useful, equal to $d^* \Delta p^* = \frac{2}{3\sqrt{3}} p_0 d_0$. Optimal operating point thus determined (A - Fig. 12), can provide an indication as training hydropneumatic cars electric motor requires the existence of friction.

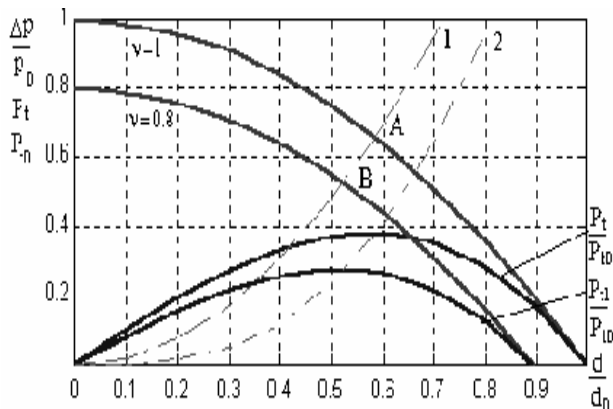


Fig. 12: Dimensionless characteristics of hydro-pneumatic machines.

If asynchronous motor involvement there will be a very big difference between theoretical values of flow and pressure resulting from optimization and resulting in reality from effective involvement of the machine. Locus of maximum power points, under the law 3 hydro-pneumatic machine, the cubic parabola denoted by 2. For flows d_0 and d developed car, the pressure losses from the plant will be given by producible $R_0 d_0^2$ and $R d^2$. Turning to the relative coordinates to obtain dimensionless loss of pressure curve (denoted by 1)

$$\frac{R}{R_0} \left(\frac{d}{d_0} \right)^2 = r \left(\frac{d}{d_0} \right)^2, \quad (10)$$

Analysis of natural extremes. Since the maximum pressure p_0 and the maximum flow d_0 have positive values and the second derivative of energy (power) useful, calculated optimal flow value is negative:

$$\left. \frac{\partial^2(d\Delta p)}{\partial d^2} \right|_{d=\frac{d_0}{\sqrt{3}}} = -\frac{6}{\sqrt{3}} \frac{p_0}{d_0} < 0, \quad (11)$$

means that the result achieved absolute maximum criterion function J . Considering that the optimum

operating point of $A \left(\frac{\Delta p^*}{p_0}, \frac{d^*}{d_0} \right)$ pump,

corresponding to a certain amount of heat generated by the boiler on site, where the quantity of heat from the chamber decreases, the car will move a quantity less fluid, the corresponding new thermal load, reducing the angular speed $v=1$ to $v=0.8$, corresponding to the new optimal operating point $B \left(\frac{\Delta p_1^*}{p_0}, \frac{d_1^*}{d_0} \right)$. This leads to a decrease in pressure

and power to the $\left(\frac{P_t}{P_0} \right)$ useful $\left(\frac{P_{t1}}{P_0} \right)$. Proportional

to the effective power, reduced power required involvement of the car, thereby decreasing energy consumption [9].

7 Energy Savings Practical Demo by Measurements

7.1 System Characteristics

For energy saving demonstration, there is considered twin-heating installations, using two different automatic control systems, with the main characteristics presented in table no.1: a conventional 3-ways system and a new pump management system, both with a thermal input of 2 MW, equipped with two stage burner. The main feed manifold is connected directly to the boilers and feeds four heating circuits. Both distribution systems have readjustment thermo-static valves, considering the rooms' temperature disturbances, and differential pressure ($\Delta p = ct$).

Tab. 1

Parameters	Conventional system	New system
Thermal output (kW)	$P_{th}=330.00$	$P_{th}=330.00$
Temp. differential (K)	$\Delta T=20.00$	$\Delta T=20.00$
Flow rate max. (m ³ /h)	$Q_n=13.00$	$Q_{sn}=9.75$
Discharge max. (m)	$H_n=8.00$	$H_{sn}=4.50$
Pump main diameters	65-100	50-100
Circuit diameter	Dn 65	Dn 50
Parallel shift of Heating curve (K)	no	+3.5

The pipes characteristic is calculated as the quotient of the pump head and the square of the flow rate.

$$H = k \cdot Q^2 \quad (12)$$

where

$$k - \text{characteristic coefficient} \left(k = \frac{8}{13^2} = 0.0473 \right).$$

The measurements were taken on 550 days in the period of the years 2007-2009. The measured values were recorder two minutes' intervals of time, resulting 710 values per 24 hours. The measured characteristics were: external temperature, flow rate at main feed manifold, differential pressure and electrical power input.

The measurements show an average (medium) external temperature of 100C in the monitored period. During the operation period (winter time), the external temperature is about -120C...-150C. Regulation stipulates that the building must be heated until external temperature reach 160C. The differential temperature between the design point and switch-off temperature of the heating system is 280C. The temperature difference between the design point and the average temperature is 22 °C. The ratio between the values is 22/28=0.79. This means that the monitoring period both heating circuits on average only require about 21% of the thermal output they were selected for. This theory is confirmed by a comparison of the flow rate measured, the differential pump pressure and the pump power consumption of both systems.

The required volume flow rate for the design point is:

$$Q = \frac{P_{th}}{1.163 \cdot \Delta T} = 13. m^3 / h \quad (13)$$

This value is confirmed by the measurements. The conventional system always supplies the nominal flow rate. The new system, by contrast, adjusts the resulting flow rate in the main feed circuit depending on the opening degree of the two control valves and thus depending on the control signal

issued by heating controller. As expected the flow rate supplied to the heating circuit by new system during the utilization period is considerably smaller than the flow rate the conventional circuit (Tab. 2).

Tab. 2

Controlling system	Flow rate	Ratio (Q_{sn}/Q_n)
3-way configuration	$Q_n=11.3$ (m ³ /h)	
Pump management	$Q_{sn}=3.3$ (m ³ /h)	0.3

The measurements of the differential pressure of the pumps shows that for the new system, the pump is operated on average at only about 0.66 of the nominal discharge head at design point (Tab. 3).

Tab.3

Controlling system	Differential pressure	Ratio (H_{sn}/H_n)
3-way configuration	$H_n=4.5$ (m)	
Pump management	$H_{sn}=3.0$ (m)	0.66

The pump input power, and thus its power consumption, is proportional to the product of the discharge head and volume flow rate. The pump draws much less power from the electricity grid. The new system was shown to save approx. 70% in electrical energy (see table no.4) [12].

Tab. 4

Controlling system	Input power	Ratio (P_{sn}/P_n)
3-way configuration	$P_n=560$ (W)	
Pump management	$P_{sn}=185$ (W)	0.33

7.2 Economic Evaluation of the Energy Savings

Considering the mentioned measurements and an electrical energy average price of 0.11 euro, the average saving per year is calculated as in Tab. 5.

Tab. 5

Cost factor	Unit	Value	Difference
Energy price	Eur/kWh	0.11	
Operation	h	6.900	
Efficiency	%	35	
Power conv.	kWh	4.500	
Power new	kWh	1.100	-3.900
Power cost conventional	Euro	545	
Power cost new system	Euro	130	- 420

Total energy savings economic effect. Considering savings and savings interests (3%), the

effects of system modernization for a period of 20 years is presented in Tab. 6. In the first year, the reduced investment costs for the circuit is -1,059 Euro, the extra price for the new equipments is 402 Euro and the reduced commissioning costs for heating circuit is - 45 Euro.

Tab. 6

Period (y)	Reduced operating costs (Euro)	Savings (Euro)	Interests on savings (Euro)	Savings & interests (Euro)
1	- 415	-1.118	- 33	- 1.501
2	- 450	-1.606	- 49	- 1.655
3	- 450	-2,110	- 61	- 2.173
4	- 490	-2.666	- 80	- 2.744
5	- 490	-3.235	- 95	- 3.329
6	- 530	-3.867	- 115	- 3.978
7	- 530	-4.525	- 130	- 4.649
8	- 570	-5.215	- 155	- 5.367
9	- 570	-5.940	- 177	- 6.126
10	- 609	-6.731	-201	- 6.942
11	- 609	-7.543	- 225	- 7.759
12	- 650	-8.401	- 251	- 8.671
13	- 650	-9.303	- 278	- 9.590
14	- 680	-10.267	- 307	- 10.581
15	- 680	-11.262	- 335	- 11.599
16	- 720	-12.316	- 370	- 12.689
17	- 720	-13.405	- 401	- 13.810
18	- 760	-14.563	- 436	- 15.002
19	- 760	-15.771	- 475	- 16.245
20	- 760	-16.991	- 515	- 17.501

GHGs emission mitigation effect. The total CO₂ mitigation corresponding to the energy reduction ΔE is [2]

$$\Delta C = 0.36 \times \frac{44}{12} \times \Delta E \times 0.8 \text{ kg CO}_2 / \text{year} \quad (14)$$

Then, for the estimated energy saving of 3,800kWh/year the GHGs mitigation is about 4,012.8 kg CO₂/year.

8 Conclusions

The economy generated by the proper adjustment of pump is always profitable if: item is oversized power is high, annual operating time is great variation in loads during operation in different seasons or different times of day or night is great, the cost of electricity is high and investment is minimal.

When the pump factory has more speed, the economics justify changing speed pump (electrical coupling modification or use of a selector), so the cost is virtually nil.

Also, in case of emergency replacement (for technical reasons), increased cost of an integrated variable speed pumps is rapidly amortized (pay-back time 2-3 years). Instead, use an external variable speed power is not justified for amri and / or a highly variable load profile.

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