# RESEARCH REGARDING THE OPERATION OF WATER PUMPING HYDRAULIC SYSTEMS

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**Abstract**

*Important control and limit energy consumption is related through the component pumps hydraulic systems can help save energy. These issues need to be addressed, both qualitatively and quantitatively.*

*The testing of rotating volumetric hydraulic pumps and the simulation of hydraulic networks on the laboratory stand presented in the paper allows simulation of pump load, measurement pressure and flow.*

*The quantification of the effects of flow regulation by regulating pump speed is a current concern in the effort to reduce industrial energy consumption.*

*The quantification of the effects of the flow regulation by regulating pump speed is a current concern in the context of efforts to reduce industrial energy consumption. It presents the analysis of a pump driven by a frequency converter engine, for which a reduction in annual energy losses of 11% was achieved by the speed variation process, compared to the valve adjustment process [2].*

**Key words:** *volume pumps, laboratory stand, debit regulation, speed adjustment, energy quantification*

## 1. Introduction

Electrical energy consumed by pumps, fans and compressors is an important part of the electricity consumed worldwide. It is estimated that industrial processes and building utilities, 72% of electricity is consumed by engines, 63% of this energy being used for circulating fluids.

To adjust the flow, the traditional method consists in varying the useful section of the fluid through which the fluid flows: valves, taps and valves are the most commonly used devices. This method is inefficient from the energy point of view, as well as an additional load on pump discharge [1].

A significant improvement in the power of the pump can be achieved by engaging the hydraulic machine with a variable speed, which leads to a much closer utilization to the requirements of the powered network. Compared to solutions where flow control is done through traditional methods, the flow rate will always have the necessary value according to networking requirements.

In the field of pumping, the most important gains are obtained through centrifugal pumps.

## 2. General references on water support systems

Water pumping systems are used in a multitude of industrial activities and fields, with the particularities of structure and operation being determined by the specific conditions of use.

Pumps from underground water abstraction plants are essential components of the technological system, thus participating in an important share of total electricity consumption during the operation of the plant. Their number, position and operating characteristics will be determined according to the exhaust system adopted and the operating mode.

Aggregates from pumping stations ensure the transport of some volumes of water from the suction basin to the network to which consumers are connected.

In order to analyze the operation of the pumping aggregates, it is necessary to know in detail both the characteristic of the aggregate and the network.

## 3. Experimental research to establish the performance of the systems with the adjustment of the debit by changing the refulation task

For the experimental lifting of characteristic curves, the pump whose characteristics have to be determined is mounted in a test facility. The diagram of a stand variant for the industrial energy test is shown in fig. 1.

The stand is in open circuit, one of the tanks being in contact with air at atmospheric pressure. The pump 1 aspires from the open tank 2 via the pipe 3 provided with a valve 4. The water enters the pipe section 5 connected to the pump inlet connection. At the outlet of the pump is connected a pipe 6 provided with a regulating valve 7. A discharge measuring system 8 is mounted on the discharge line.

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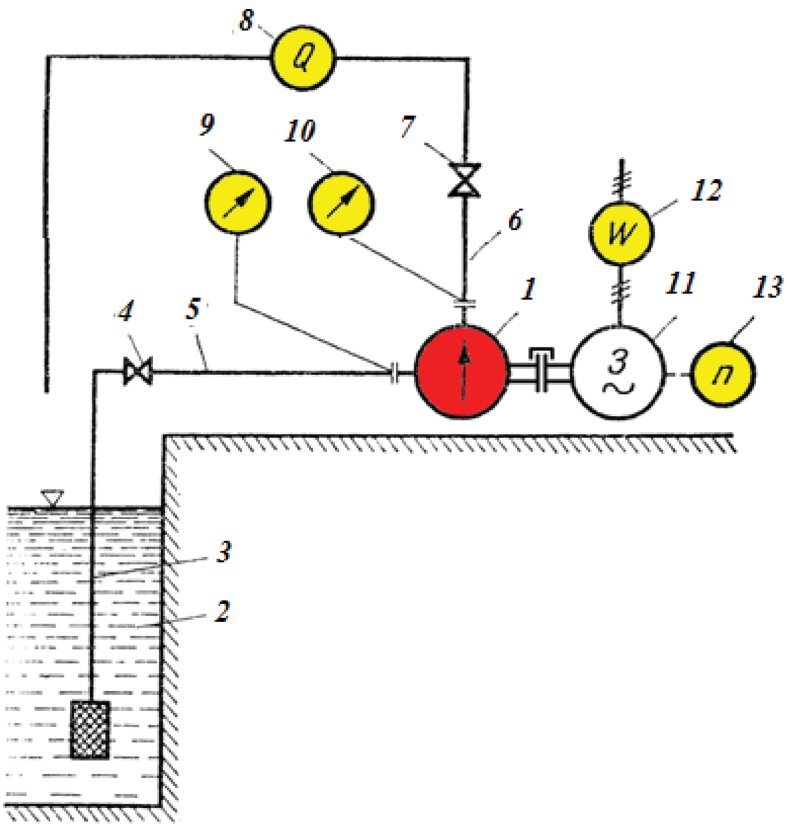



*Fig. 1.* **Stand scheme for industrial energy charging**

At the inlet and outlet of the pump are mounted a manometer-vacuum meter 9 and a pressure gauge 10, respectively, for determining the inlet and outlet pressure from the pump. The pump is driven by a electric motor 11 (the connection between the power absorbed by the grid and the power developed on the shaft is known), the power absorbed by it from the grid being measured by means of a wattmeter 12 and the speed by means of a tachometer 13.

The purpose of the industrial energy test is to determine the dependencies *H(Q)* or *Y(Q)*, *P(Q)* and *η(Q)* having parameters *n* and rotor diameter *D* for centrifugal pumps and geometric cylinder $V_g$ for volumetric pumps. After the pump is mounted in the test bench, the valve 4 is fully opened and the pump is added, then the electric motor for the pump is started. A position of the valve 7 is fixed and determinations of the instruments determining the flow rate.

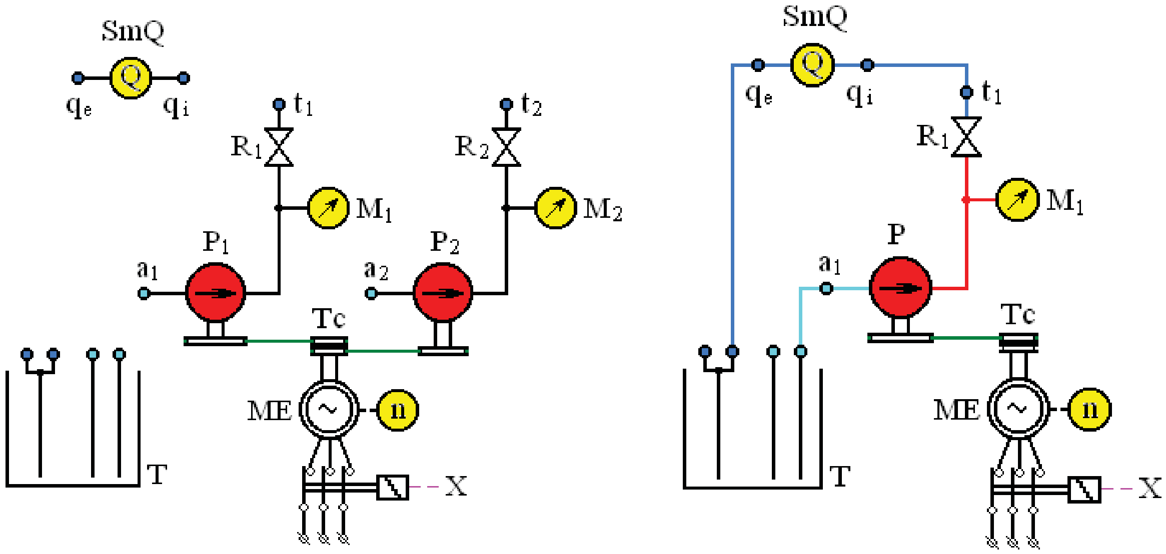
The power absorbed by the grid, the speed and the pressures at the inlet and outlet of the pump are determined. Based on these, the pumping height *H* or *Y* and the power absorbed by the pump (power to the motor shaft) *P* can be calculated, depending on the characteristic curve of the motor for centrifugal pump.

Repeat these operations for other positions of the control valve 7. In the case of variations in the speed from the constant speed at which the characteristic curves are desired, the *Q*, *H* or *Y* and *P* quantities are recalculated at the desired speed using similarity relations.

The set of measurements for volumetric pumps is similar and allows the calculation of the hydraulic power (useful) and, if known, the pump power of the mechanical power absorbed by the drive motor: when the value of the speed at the mortar shaft is known, the torque Required pump according to discharge pressure.

Based on the experimental results, the dependencies *H(Q)* or *Y(Q)* and *P(Q)* for the centrifugal pumps *Q(p)* and *P(p)* for the volumetric pumps are plotted: Measures inherent. Following the plotting of the above mentioned dependences, we can calculate and draw the flow according to the flow *η(Q)*, and the pressure *η*(p). Finally, a diagram is obtained that fully characterizes the pump's energy performance. The operating point corresponding to the maximum efficiency is the rated operating point.

For the purpose of checking these considerations, a volumetric sliding plenum pump designed to drive the fuels was tested. The test was carried out on a laboratory stand (Fig. 2) designed for testing rotary volumetric hydraulic pumps and simulation of the hydraulic networks in which they are active.




**a)** **b)**

*Fig. 2.* **Lab stand for hydraulic pump testing**

The stand for testing the pumps and hydraulic networks shown in Fig. 2, a) as a structure and the hydraulic diagram of principle and in Fig. 2, b) as a hydraulic scheme for determining the characteristic curves of a pump *P*, has the following components: *ME* - electric engine; $P_1$, $P_2$ - pump; $R_1$, $R_2$ - load generating taps; *T* - tank; $t_1$, $t_2$ - connection sockets for test networks; *X* - electric engine control system; *N* - speedometer / tachometer; $M_1$, $M_2$ - manometers; *SmQ* - flow measurement system.

The stand can be used as a source of hydraulic pressure for testing different network configuration and nominal diameter of max.3/4"; inputs can be connected to the network connections $t_1$ and/or $t_2$, and output to the input $q_i$ of the measuring system flow *SmQ*. Transmission of the energy from the electromotor to pump is made via trapezoidal belt transmission $T_c$ and allows to driving individual/simultaneous pumps. A hydraulic rotary volumetric pumps for debit determination should be based on a few basic requirements which it must carry out:

-able to be simulated charge pump in load;

-to be able to measure the pressure and the pressure difference in domain 0-1 $MP_a$;

-to be able to measure the debit in domain 0-100 L/min, with precision of at least ± 0.01 L/min.

Corresponding to hydraulic scheme of principle as shown in figure 1 can be seen that the two pumps $P_1$ and $P_2$ are driver simultaneously or individually, according to the necessities tests, by an electromotor three-phase *ME*. Aspiration plugs of the two pumps $a_1$ and $a_2$ can be connected to the aspiration pipes existing in the open tank *T*. The repress orifices of the pumps are connected to pressure lines which contains one tap $R_i$ and one manometer $M_i$, at the end of these lines are arranged pressure sockets $t_1$ and $t_2$.

The stand is designed so as to permit measuring the following parameters:

- the lifting pressures on the two columns with manometers $M_1$ and $M_2$;

- the lifting debits-through debit measurement system *SmQ*;

- the electric motor speed to drive of the pumps-through tachometer n;

- the working fluid: adapted to the type of pump being tested;

- the M1 manometer was selected and was used according to the maximum pressure and pressure variation range (0 ... 10 bar), precision class 2.5.

The test results for the above mentioned pump correspond to two different drive speeds at the hydraulic machine shaft: these were obtained by producing a trapezoidal belt transmission, with pulleys of different diameters mounted on the pump shaft. Table 1 shows the values of the pressure *p*, the *Q* flow rate and the pump shaft speed for two distinct situations characterized by the ratio of the pulley diameters of the drive / pump motor shafts, when the idling speed of the drive motor is $n_{M0}$ = 920 rpm.

The test of the pump implied a variation of the load on the discharge pipe made by means of the valve located on it. Several sets of tests have been performed which have led to the overlapping of the four values listed in the table several times. A view was made of how the pump discharge rate *Q* is influenced by the value of the load *p* on the discharge port.

*Table 1.* **Parameters of the test pump**

| Diameter of pulleys | No. | Hydraulic parameters | | Mechanical parameters |
|---|---|---|---|---|
| | | p, [bar] | Q, [L/min] | n, [rot/min] |
| $\phi_1 / \phi_2$ 76/137 | 1. | 0,7 | 30,83 | 400 |
| | 2. | 1 | 27,5 | 395 |
| | 3. | 1,5 | 18,33 | 385 |
| | 4. | 2 | 14,16 | 380 |
| $\phi_1 / \phi_3$ 76/94 | 1. | 1,7 | 43,33 | 580 |
| | 2. | 1,8 | 32,92 | 540 |
| | 3. | 1,9 | 23,66 | 430 |
| | 4. | 2 | 11,66 | 360 |

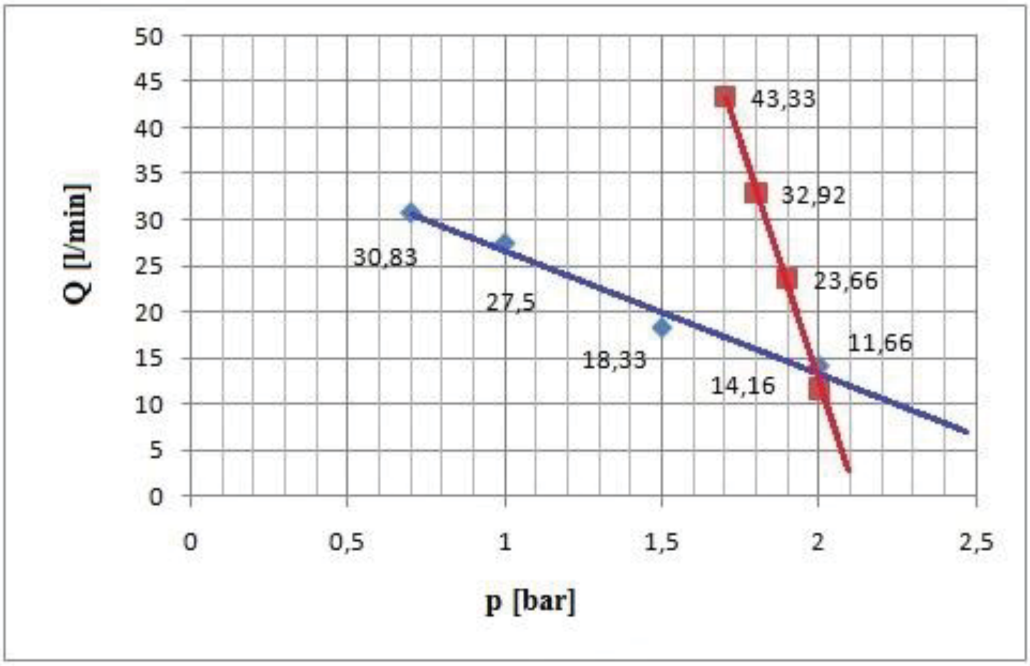



**Fig. 3.**

The diagrams plotted in figure 3, taking into account the values in Table 1, confirm the quasi-linear dependencies *Q(p)* for the volumetric pumps. The blue curve with a lower slope corresponds to a lower pump speed and a flow adjustment range of 30.83 to 14.16 L/min corresponding to a pressure range of 0.7 to 2 bar. When mounting a smaller diameter pulley on the pump shaft, the pump drive speed increased by 50% compared to the previous case, resulting in a flow rate from 43.33 to 11.66 L/min for a range of the load from 1.7 to 2 bar. It can be seen from the red diagram that the debit being greater the *Q(p)* curve moves to the right and becomes much more abrupt, the influence of the pump discharge load being much more pronounced, the adjustment of the discharge flow being much harder to achieve.

We believe that the flow adjustment results for the second pump drive situation would have been better if the pipeline route had had a larger diameter on the machine.

It follows that when adjusting the flow rate of the volumetric pumps by variation of the discharge load, the drive speed and discharge flow rate must be aligned with the diameter of the system being fed so that the slope of the diagram *Q(p)* is not greatly increased and the desired flow rate adjusted easy to achieve.

The pump cylinder, the main machine parameter required to determine the average flow rate when the machine drive speed is known, is a parameter that can be determined either on the basis of the pump geometrical dimensions or by tests under certain operating conditions: operation at Idling because the drive and discharge rates allow the starting point of the pump curve *Q(p)* to be set.

## 4. Energy environmental enhancement of the efficiency of the adjustment of the power of pumping turning

Energy consumption differs depending on debit control methods adopted in exploitation.

Quantitative adjusting can be achieved:

-with pumps with different characteristics (debit and discharge head);

-variable speed with pumps having the possibility of modifying the debit and discharge head of needs, to minimize electrical power related for pump.

Variable speed drive motor of the pump can be done by using the frequency converters or DC motors, of which the asynchronous motor with short-circuited rotor, associated with a static frequency converter with thyristors or power transistors (200 kW respectively 25 kW) forms an electrical drive equipment with adjustable speed in large limits [1].

Any way speed control involves energy losses. For calculating energy losses when using debit adjustment valve, should be considered power and efficiency of the system motor-frequency converter, lower, must be correlated with the substantial drop in engine power drive.

The analyzed pump (CERNA 200a - $H$ = 32 m, $Q$ = 350 $m^3$ / h - used for impregnation of impurities) is connected to a standard 55 kW engine. The yield of such a motor is 90% and is maintained constant for power supply ($P$) values between 50% and 100% of its nominal value ($Pn$).

Frequency converter output is 96% at rated load without additional engine losses [3].

Taking into account the power supply calculated in table 2, the power losses are obtained presented in table 3. As a result the method speed variation, compared to the valve control losses from 30660 kWh to 27243 kWh, respectively 11% [4].

*Table 2.* **Power supply and power consumption of a pump in the case of the valve adjustment, respectively with variable speed**

| Flow G [$m^3$/h] | During operation | Valve adjustement | | Variable speed | |
|---|---|---|---|---|---|
| | [hours] | Power P, [kW] | Energy E, [kWh] | Power P, [kW] | Energy E, [kWh] |
| 350 | 438 | 42,5 | 18615 | 42,5 | 18614 |
| 300 | 1314 | 38,5 | 50589 | 29,0 | 38106 |
| 250 | 1752 | 35,0 | 61320 | 18,5 | 32412 |
| 200 | 1752 | 31,5 | 55188 | 10,0 | 17520 |
| 150 | 1752 | 28,0 | 49056 | 6,5 | 11388 |
| 100 | 1752 | 23,0 | 40296 | 3,5 | 6132 |
| TOTAL | 8760 | - | 275064 | - | 124173 |

*Table 3.* **Power losses in the case of adjustment the valve, respectively, with variable speed**

| Flow G [$m^3$/h] | Duration [hours] | Valve adjustment | | | | Variable speed | | | |
|---|---|---|---|---|---|---|---|---|---|
| | | Power P [kW] | $\eta_M$ (%) | Power loss [kW] | Energy loss [kWh] | Power P [kW] | $\eta_M$ [%] | Power loss [kW] | Energy loss [kWh] |
| 350 | 38 | 42,5 | 90 | 4,7 | 2059 | 42,5 | 86 | 6,9 | 3022 |
| 300 | 314 | 38,5 | 90 | 4,3 | 5650 | 29,0 | 85 | 5,1 | 6701 |
| 250 | 1752 | 35,0 | 90 | 3,9 | 6833 | 18,5 | 84 | 3,5 | 6132 |
| 200 | 1752 | 31,5 | 90 | 3,5 | 6132 | 10,0 | 79 | 2,7 | 4730 |
| 150 | 752 | 28,0 | 90 | 3,1 | 5431 | 6,5 | 74 | 2,3 | 4030 |
| 100 | 752 | 23,0 | 90 | 2,6 | 4555 | 3,5 | 0 | 1,5 | 2628 |
| TOTAL | 760 | - | - | - | 30660 | - | - | - | 27243 |

Based on experimental results presented in tables 4 and 5 were constructed graphs from figures 4, 5, 6, 7 and 8.

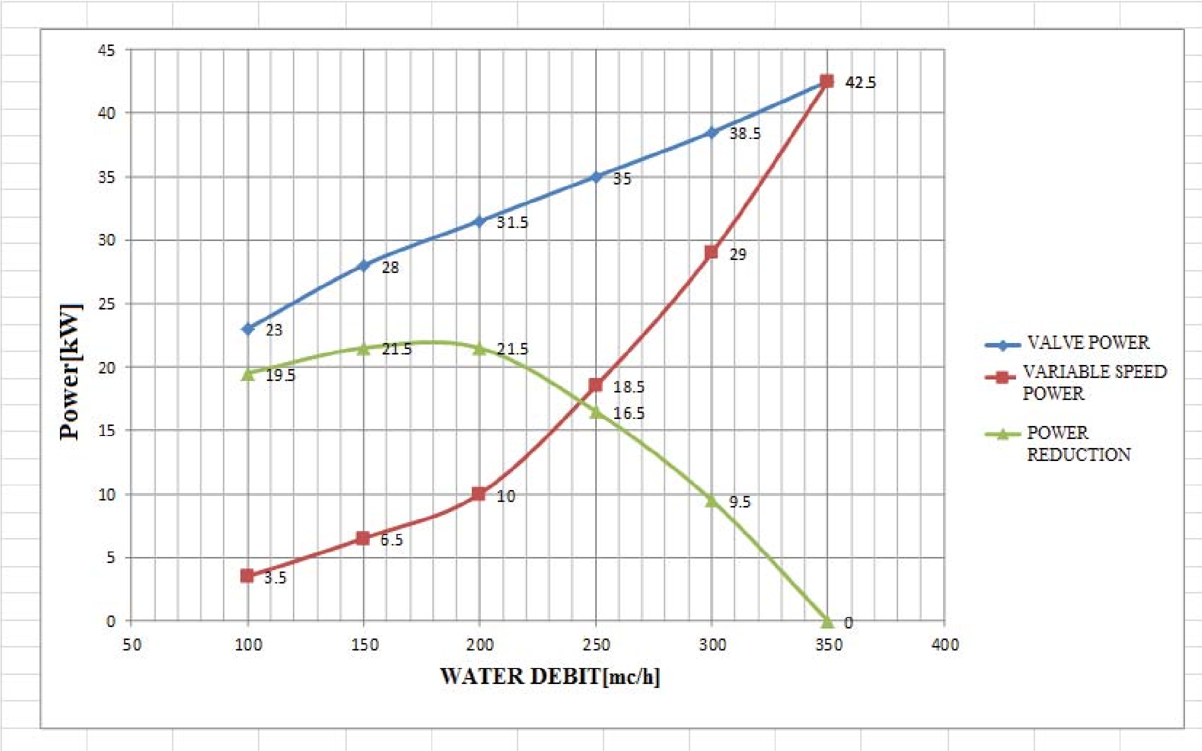



**Fig. 4.** *Power variation depending on the how to adjust and the size water debit circulated*

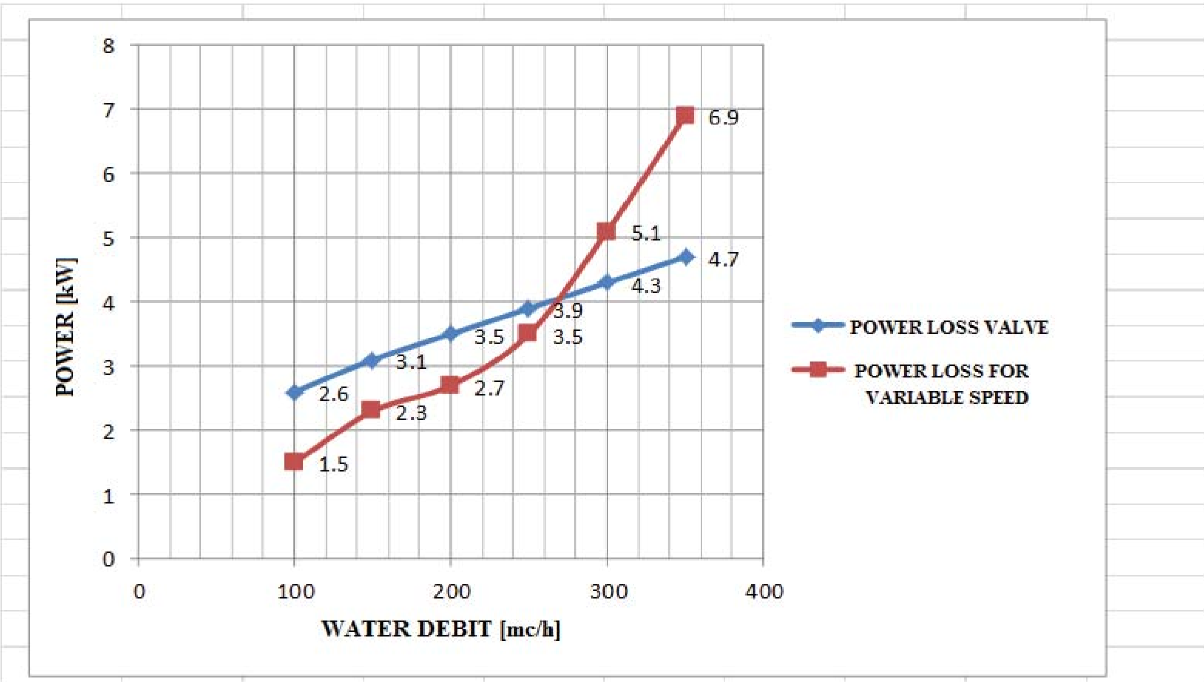



**Fig. 5.** *Power loss variation depending on the how to adjust and the size water debit circulated*

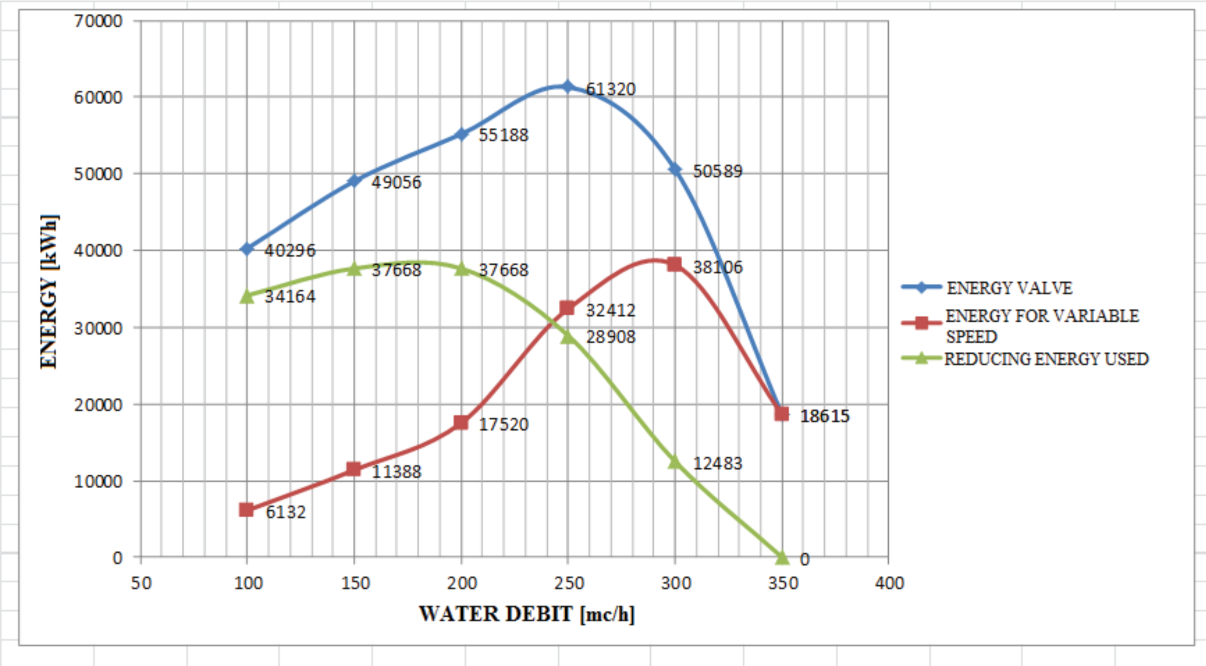



**Fig. 6.** *Variation the energy used depending on the how to adjust and the size water debit circulated*

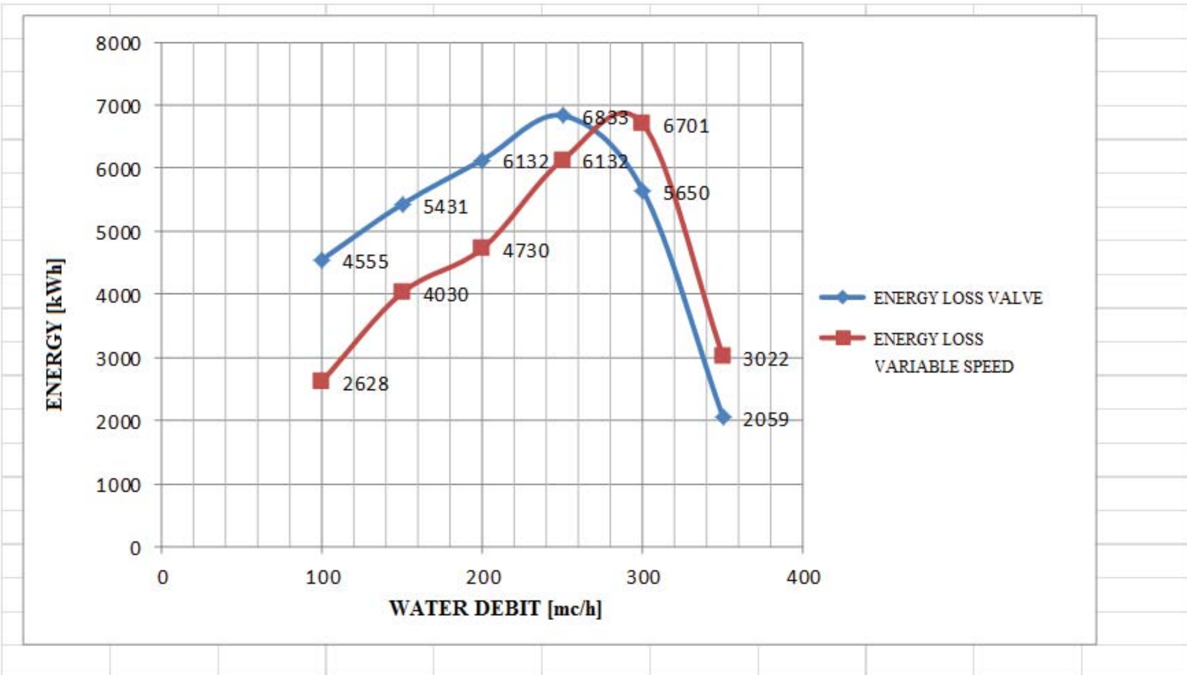



**Fig. 7.** *Energy loss variation depending on the how to adjust and size water debit circulated*

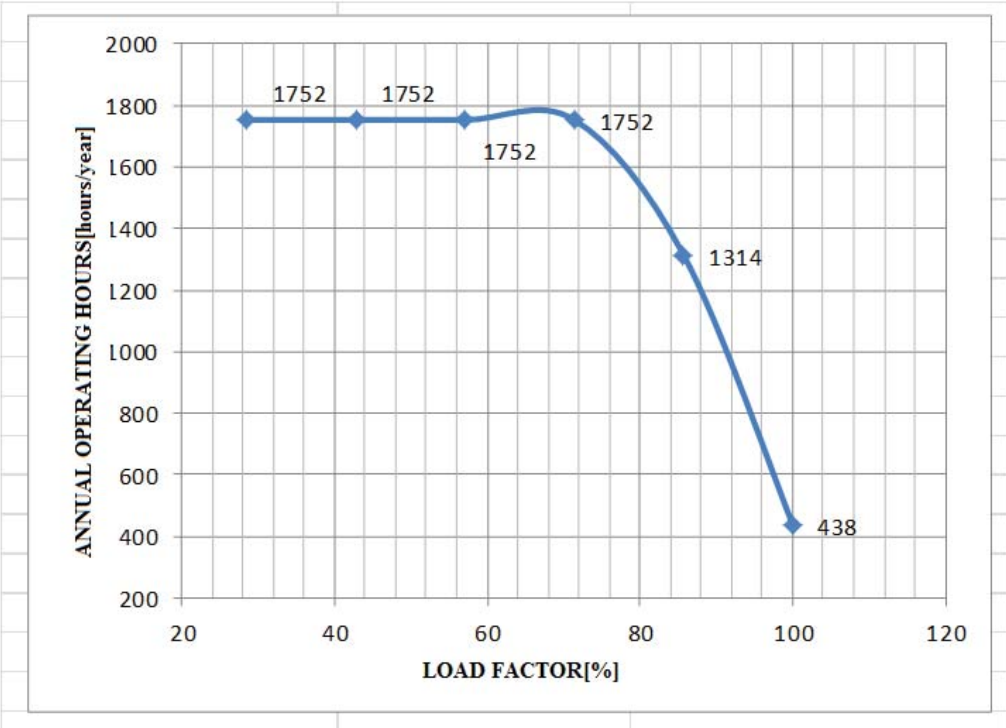



**Fig. 8.** *Annual variation of the load factor*

It is clear from figures 4 and 5 that the reduction in power and energy used, depending on the adjustment mode, is 0% at maximum load, increasing as follows: 25% at load factor of 86%, 47% at load factor of 71%: 68% at load factor of 57%, 77% at load factor of 43%, 85% at load factor of 29%. Figures 5 and 7 point out that at power values of up to 4 kW and energy up to 6750 kWh, power and energy losses no longer justify the variation of the flow through the variation of the speed.

Graphics from figures 4-8, make evident that the debit adjustment by regulating rotational speed of the pump is efficient in cases characterized by the loading factor with values around 80% when it reaches the operating time of 1700 hours.

## 4. Conclusions

- Increase energy efficiency and reduce energy consumption of pumps can be obtained by adjusting the debit with regulating the rotational speed;
- By eliminating the maximum debit adjustment valves, when the pump is over dimensioned, operation at low speed allows to avoid energy losses (caused by a debit limiting valve);
- Reducing the risk of cavitation: these phenomena related to rapid variations in pump rotational speed are avoided due to progressive start slow, controlled by variation speed device;
- The lifetime of the rotors is connected their peripheral speed: speed reduction will improve reliability;
- Speed variation to operate the maximum efficiency, increasing the service life of the bearings and joints.

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