PUMP SYSTEM IN THE ENERGY MANAGER TRAINING CENTER AT THE FACULTY OF MECHANICAL ENGINEERING UNIVERSITY OF BELGRADE

Đorđe ČANTRAK*, Miloš BANJAC, Novica JANKOVIĆ, Dejan ILIĆ
University of Belgrade, Faculty of Mechanical Engineering, Belgrade, Serbia

INTRODUCTION

The Organization for the training of energy managers and authorized energy advisors or Training Organization (TO) was established on 20 November 2015 on the basis of a decision adopted by the Minister of Mining and Energy Republic of Serbia, which gave authorization to the University of Belgrade, Faculty of Mechanical Engineering (MFBU) to perform tasks relating to the training of energy managers (EM) and authorized energy advisors (EA). The legal basis for the adoption of this Decision has been laid down by the Law on Efficient Use of Energy. The establishment of the TO is one of the main results of two projects: “Introduction of Energy Management System in Energy Consumption Sectors in the Republic of Serbia” and “Project for Assistance of Enhancement of Energy Management System in Energy Consumption Sectors in the Republic of Serbia”, which the Ministry of Mining and Energy (MOME) conducted jointly with Japan International Cooperation Agency (JICA). In the scope of the second project JICA donated to MOME laboratory equipment and measuring instruments necessary for the practical training of EM and authorized EA, install equipment and make operational the Laboratory for practical training. The equipment was officially put to use on 11 October 2016.
Three training plants established in the Laboratory are pump, air compressor and boiler units. In this paper is presented pump unit.
Pumps consume a great share of energy in today's world. So, adequate matching of pump performance and system requirements is necessary. Various methods for pump regulation could be applied [1]:

- throttling,
- switching pumps on/off (operating in parallel or series),
- by-pass control,

*Email: djcantrak@mas.bg.ac.rs

- speed control,
- impeller vane adjustments,
- pre-rotation control (changing the angle of the guided vanes),and
- modifying the impeller geometry (reduction of impeller diameter (impeller trimming) and sharpening of impeller blade trailing edges).

Life cycle cost (LCC) analysis, which takes into account the purchase cost, installation, pump operation which include also energy cost, maintain and recycle, i.e. disposal of all pump components, reveals that energy cost share in many typical pumping systems is approximately 50% of the total cost [2].

"Electrical energy consumed by pumps, fans and compressors represents a significant proportion of the electricity used around the world. It is estimated that in industrial processes and building utilities, 72 % of electricity is consumed by motors, of which 63 % is used to drive fluid flow in pumps, fans and compressors." [3]. Valves are still the most commonly used device for controlling the flow rate. "However substantial energy savings can be obtained by using variable speed drives to control the flow rate or pressure in pumps..." [3]. The greatest savings could be achieved with radial, i. e. centrifugal pumps [3]. This is demonstrated at the pump system in the Energy manager training center at the Faculty of Mechanical Engineering University of Belgrade. Some possibilities of this test rig are presented here.

TEST RIG

Pump system is presented in Fig. 1. City water is supplied, without previous preparation, to the water circulation tank equipped with the water meter glass and thermometer (1). Pressures on the pump suction side (i. e. inlet pressure) and the pump outlet are measured by use of the Bourdon tube pressure gauges, and parallel with appropriate pressure transducers (2 and 3, respectively). Pressure transducers are connected to the electric board (14). Pump unit (4), manufactured by Grundfos, is type NB32-160.1/169 A-F2-A-BAQE, with specified flow rate$Q$ = 21.7 m^3/h, head H = 28.9 m, speed of rotation$n$ = 2900 min^{-1} and efficiency η = 60%. Electric motor has 3 kW and cosφ = 0.82 to 0.87 and belongs to the category IE3. It can be regulated by setting the motor frequency by use of inverter placed in the electric board (14).It is provided in the technical specification [3] that actual centrifugal pump impeller diameter is 0.169 m. It is made of cast iron. Pump inlet and outlet diameters are respectively 50 and 32 mm. Shaft is made of stainless steel and has diameter 24 mm.

Discharge valves are 5, 7 - valve V3 (Fig. 2(a)) and 12 - valve V2 (Fig. 2(a)). Valve at position 5 is almost always fully open, while regulation is performed by use of valves V2 and V3. Flow rate is measured by use of electromagneticflowmeterKrohne type OPTIFLUX 2050 (6) andvariable area Krohne type H250/RR/M40. The second one, with indicator, is manufactured by Krohne and has limitation of maximum flow rate 10 m^3/h. Pipe 1/2" inlet ball valve (10, BV1 at Fig. 2(a)) and pipe 3/4" inlet ball valve (9, i. e. BV2 at Fig. 2(a)) are open/closed in order to demonstrate pressure drops on these two pipes of the same length. Pressure drops are measured only on the Bourdon tube pressure gauges (11 and 13, respectively).

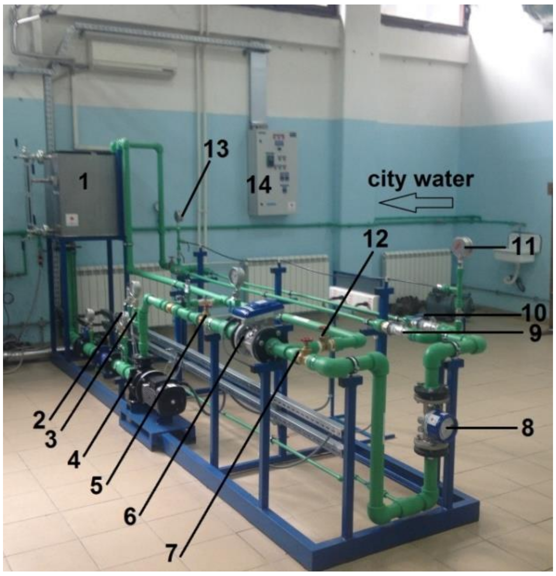


Fig. 1.Main elements of the pump system

The main purposes of this facility is to demonstrate the following:

1. study of the centrifugal pump - basis,
2. determination of the pump performance curve,
3. to demonstrate the influence of the pipe diameter on the system hydraulic losses, i. e. pump duty point,
4. to measure power consumption decrease by using inverter instead of the valve regulation and
5. to demonstrate the usage of the energy audit equipment.

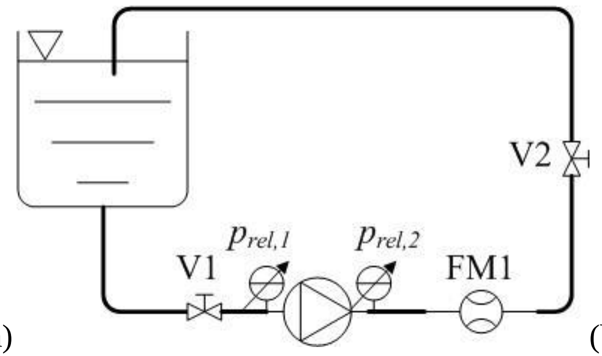
A lot of new ideas and upgrade could be involved such as measurement and calculation of the pump inverter effectiveness and etc.

MEASUREMENTS

Here will be presented three experiments which are performed at the installation.

Pump performance curve determination

First experiment is performed to demonstrate pump performance curve determination. Scheme of the circulation test rig is presented in Fig. 2(a). In this case valve V3, i. e. pos. 7 at Fig. 1 is closed.

(a) 

(b) 

Figure 2.a) Circulation test rig for pump performance curve determination and b) pump with measurement sections

Pump is positioned under the level of the water in the inlet tank. City water is used and its level in the tank is checked by use of the analogue level meter. Valve on the pump inlet V1 (Fig. 2(a)) is always open. Pressures on the pump inlet and pressure sides (Fig. 2) are measured by two devices - Bourdon tube pressure gauges and pressure transducers (in Fig. 1 positions 2 and 3, respectively, Fig. 2(b)).Measurement positions are more than two inner diameters from the inlet and pressure flanges. Flow is measured by use of the electromagneticflowmeter (Fig. 2(a), FM1). Pump duty point is changed with valve V2 (Fig. 2(a)). Valve V2 is closed for the start and afterwards slowly open, so the duty point could be "translated" in the region of higher power slowly.
The whole measurements are performed on the maximum rotation number.
Pump head H [m], or by ISO 9906 [4] total head, is determined as follows if the compressibility of the pumped liquid is neglected:

$$H = \frac{p_2 - p_1}{\rho g} + \frac{U_2^2 - U_1^2}{2g} + (z_2 - z_1) = \frac{p_{G,2} - p_{G,1}}{\rho g} + (z_{G,2} - z_{G,1}) + \frac{U_2^2 - U_1^2}{2g} + \Delta z, \quad (1)$$

where$p$[Pa] is the absolute pressure in the specified cross-sections on the pump inlet and outlet,U[m/s] is mean axial velocity, z[m] is the height of the centre of the cross-section above the reference plane, ρ[kg/m^3] is water density and g[m/s^2] is acceleration due to gravity. It is sufficient to useg = 9.81 m/s^2 for lower grades of the standard ISO 9906. Absolute pressure is determined on the basis of the readings (p_G) and height corrections (z_G). Inlet sections is denoted with "1", while outlet with "2".
Energy per unit mass Y [J/kg] could be derived as follows [4, 5]:

$$Y = H \cdot g. \quad (2)$$

Mean velocity is determined in inlet and outlet sections as follows:

$$U_i = \frac{Q}{A_i}, \quad (3)$$

where$Q$ is volume flow rate and$A_i$is pipe cross section area on the pump inlet (i =1) and outlet (i = 2).
"Pump power output is mechanical power transferred to the liquid during its passage through the pump." [3] It is also called hydraulic power P_h[kW] and calculated as follows [5-8]:

$$P_h = \rho Q Y. \quad (4)$$

Pump efficiency η is determined as follows:

$$\eta = P_h / P, \quad (5)$$

where$P$[kW] is the pump power input, what is read on the control table (Fig. 1, pos. 14).
Pump curve is steadily rising and it is tested for following flow rates: 0, 6, 12 and 18 m^3/h. These flow rates are adjusted approximately and read at the control table (Fig.1, pos. 14). Velocity head (U^2/(2g)), what represents "kinetic energy per unit mass of the liquid in

movement, divided by g" [4] is calculated afterwards. For the highest volume flow rate (Q = 18 m^3/h) mean axial velocity on the pump pressure side achieve more than 5 m/s. So, now hydraulic power could be determined for all four flow rates, i. e. positions of the valve V2.

Pump power input P is also measured on the control table (Fig. 1, pos. 14) and its increase with flow rate increment is also obvious. The pump efficiency is determined on the basis of eq. (5).

Pipe inlet to the water circulation tank could be in atmosphere, i. e. pipe outlet could be above water level. In this case piping system characteristic would have an additional term - a static component which is independent of the flow rate involves H_{geo}, This represents position of the pipe outlet to the water level in the tank:

$$Y_A = \mathrm{g}H_{geo} + mQ^2, \text{ or } H_A = H_{geo} + \frac{m}{\mathrm{g}}Q^2 \tag{6}$$

where$m$ [(J/kg)/(m^3/s)^2] is the pipe coefficient which involves all hydraulic losses described with Darcy and Weisbach formulas:

$$m_j = \frac{8}{d_j^4 \pi^2}\left(\sum_{i=1}^{N} \zeta_i + \lambda_j \frac{L_j}{d_j}\right), \tag{7}$$

where$j$ denotes the$j$-th pipe section, N number of local hydraulic losses ζ, $\lambda$the dimensionless pipe friction loss coefficient, which in the case of the turbulent flow depends on Reynolds number and relative pipe roughness, L pipe length and d is inner pipe diameter. Pump duty point is generated when the pump head curve intersects pipe hydraulic curve. This is just another way to write the Bernoulli equation [5-9].In this experiment static head is zero (H_{geo} = 0), so the system head is entirely due to dynamic head, i. e. hydraulic losses.

This pump testing could be done also on some other rotation speeds, not only on the maximum one. Measuring uncertainties calculation is specified in standard ISO 9906 [4].

Pump performance curve, as well as regulation with valve on the pump pressure side are demonstrated in this experiment.

Influence of the pipe diameter on the system hydraulic losses

In this experiment will be demonstrated variation of the system pipe coefficient m while using two various pipelines with inner diameters 1/2" and 3/4".

These pipe sections have the same length 2.582 m. Pump operates on the highest speed of rotation.

Scheme of the circulation test rig for this experiment is presented in Fig. 3. In this case valve V2, i. e. pos. 12 at Fig. 1 is closed, while valve V3, i. e. pos. 7 at Fig. 1 is open.

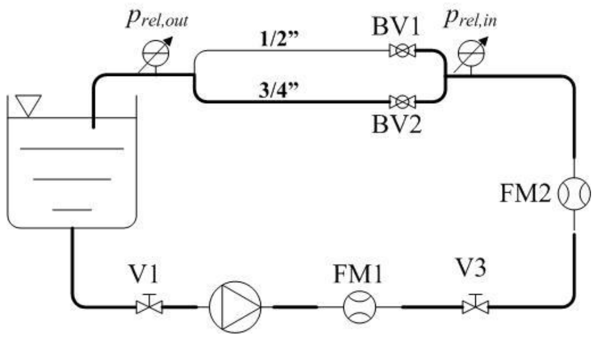
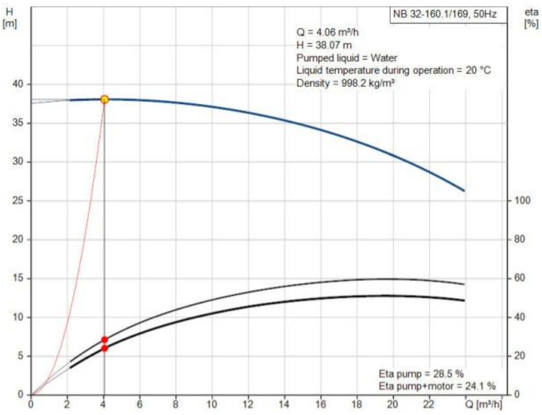


Figure 3. Circulation test rig for demonstration of the pipe diameter on the pressure losses

In this experiment is demonstrated also usage of the analogous flowmeter (FM2) which operates on the basis of the pressure drop.
In the first case valve BV2 (Fig. 3) is closed so the pipe with smaller diameter (1/2") is in function. In this case is achieved 4.06 m^3/h (Fig. 4(a)).

(a)


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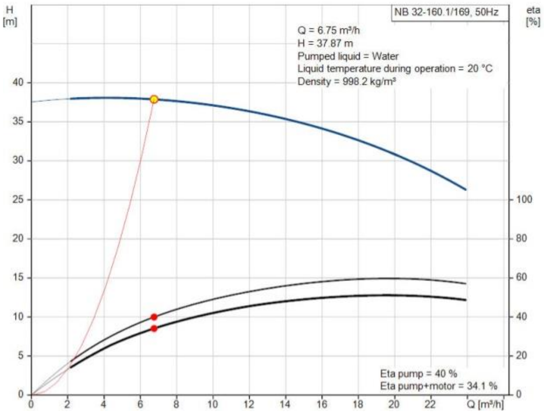

Figure 4. Pump duty point for pipe with: (a) smaller and (b) bigger diameter in function [8]

In the second case valve BV1 (Fig. 3) is closed, so pipe with bigger diameter (3/4") is in function. In this case is achieved 6.75 m^3/h (Fig. 4(b)).
Pump doesn't have steep characteristic, so the head is almost the same, while flow rate is significantly changed what implies increase in the input power of approximately 15%.
Pressure drop on these pipe sections is measured by use of the Bourdon tube pressure gauges (Fig. 3). It is shown that in this case pressure drop is for 26.7% higher for pipe with smaller diameter.
Increase in temperature could be observed in both cases, because pump doesn't operate in the best efficiency point (Fig. 4). Experiment could be performed also on some other speeds of rotation.
In this case is demonstrated how pipe diameter influences, with the power of five (eq. (7)), on hydraulic losses. In fact, this is the same effect like one achieved by using valve.

Power consumption decrease by using inverter instead of the valve regulation

This experiment is performed on the test rig presented in Fig. 2. Pump electric motor inverter is turned on the control table 14 (Fig. 1).
First, pump rotation speed is on the maximum and flow rate is adjusted by use of the valve V2 (Fig. 2(a)) to the value Q'. Pump duty point DP is obtained (Fig. 5). System hydraulic characteristic is now H_{A2} (Fig. 5). Input power is measured and it is P.
Afterwards, the valve V2 is open and system characteristic is now H_{A1} (Fig. 5), while the same flow rate is adjusted by decreasing the pump rotation speed to n' and obtaining new pump duty point DP'. Input power is again measured and it is now P'. So, the input power is decreased. Saved power, as well as hydraulic losses due to regulation by valve V2 are marked in Fig. 5.

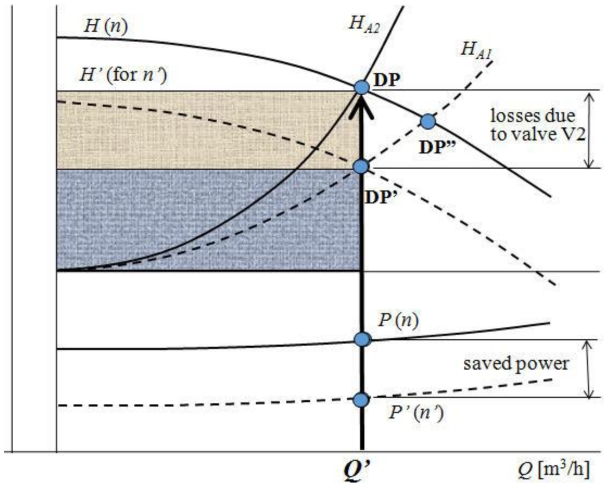

Figure 5.Saved power by using inverter instead of the valve regulation

Ratio of the surfaces of rectangulars defined with pump duty points DP and DP' is:

$$\frac{A_{DP}}{A_{DP'}}=\frac{P_h}{P_h^{'}}=\frac{H}{H'}. \tag{8}$$

Saved input power is also marked in Fig. 5, as well as losses due to local losses on the valve V2 for specified flow rate Q':

$$\varsigma_{V2}\big|_{Q'}=\left(H_{A2}-H_{A1}\right)\big|_{Q'}. \tag{9}$$

In this case of the circulation test rig, curve H_{A1} is identical to the parabola describing affinity law. This means that points DP' and DP" are similar (Fig. 5), so following very known [3, 6, 7] relations could be applied:

$$\frac{Q'}{Q''}=\frac{n'}{n''},\frac{H'}{H''}=\left(\frac{n'}{n''}\right)^2,\frac{P'}{P''}=\left(\frac{n'}{n''}\right)^3 \text{ and } \eta'=\eta''. \tag{10}$$

It is worth of notice that effect of increasing the pipe coefficient (m) could be also achieved by decreasing the pipe inner diameter, what is shown in the previous experiment. The same discussion presented in Fig. 5 could be applied in this case.

CONCLUSIONS

Pump unit in the Energy Manager Training Center at the Faculty of Mechanical Engineering University of Belgrade is established with the aim to educate students and engineers in the field of energy efficiency in pump systems. Lectures in pumps and pump systems precede training in situ, on the installed pump system. Various pump constructions,Cordierdiagram, Euler'sturbomachine equation, energy parameters, hydraulic pipeline equation, cavitation, measurements for energy savings, and etc. are discussed. Here are presented some of the experiments conducted at this installation.

"ISO 9906:2012 specifies hydraulic performance tests for customers' acceptance of rotodynamic pumps (centrifugal, mixed flow and axial pumps). It is intended to be used for pump acceptance testing at pump test facilities, such as manufacturers' pump test facilities or laboratories. ISO 9906:2012 can be applied to pumps of any size and to any pumped liquids which behave as clean, cold water..." [4].

In chapter 5.3.2 Standard test arrangements are specified the best measuring conditions in the measuring sections which are not verified in this test rig [4]. In fact, this is not always possible in practice. In fact at the presented pump test rig, recommendation of avoiding local hydraulic losses four diameters from the measuring section is not fulfilled. Anyhow, this test rig is very useful in demonstration how the pump testing could be performed in situ and what could be done in industry for checking pump performance curves. By manufacturer technical specifications [3] pump is tested in accordance with standard ISO 9906:2012, grade 3B with broad tolerance, so experimental data obtained in this test rig should be comparable with the ones in catalogue [3]. Of course, there would be some differences due to the reasons reported above.
In the pump test rig is also demonstrated loss in pipe of various inner diameters. It is shown how pipe diameter with a power five, influences hydraulic losses in the system. Decreasing pipe diameter generates the same effect like closing the valve. Whole experiment could be repeated also for some other pump speeds of rotation.
The third experiment demonstrated power consumption decrease by using inverter instead of the valve regulation. In fact, the same volume flow rate is achieved by closing the valve and with reducing the pump speed of rotation *n*. Significantenergy saving is achieved in the second case. Savings depend on the duty point and here are demonstrated for one flow rate.

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