Pressure Regulation in Centrifugal Pumping System with PLC

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***Abstract* — A method of pressure regulation in a pumping system is described. A pressure control system with PLC and additional components including digital/analog input/output module DA501 and profibus communication module CM572_DP was developed. To analyze the system, a series of tests were conducted. Experiments proved that the designed system is quite flexible, easy to tune, and reliable. Advantages of the system lie in its relatively high accuracy of pressure maintenance at both the static and the dynamic mode. The proposed pressure regulation system can be used either in industry or in household.**

I. INTRODUCTION

Pumping systems based on centrifugal pumps play an important role in water supply, waste water treatment, oil production and other areas of industry [1]. Centrifugal pumps are the most common type of pumps in different industrial applications. They constitute about 80 to 90 % of all water treatment equipment [2]. Being a sub-class of the dynamic asymmetrical absorbing turbo machinery, they are used to transport liquids by means of converting the rotational kinetic energy of the fluid to the hydrodynamic energy.

Pumping systems are very popular in the variable loading processes, therefore they often require pressure maintenance to enhance the pumping efficiency and quality. To this aim, the variable speed drives (VSDs) are commonly used. These drives provide a wide range of functionalities, including speed adjustment, technology management as well as power quality improvement. In some cases it is reasonable to combine the VSDs with some external equipment to provide more accurate adjustment and to increase control possibilities.

The main purpose of pressure maintenance is to sustain a desired pressure level in a pumping installation or circulation system. In fact, in many applications the liquid being pumped is often hot and occasionally at high temperature, it contains also some amount of vapour. One of the main advantages of pressure maintenance in heating systems is that it prevents cavitations at the high points of pipelines and pumps.

Pressure maintenance also compensates unintentional leakages and makes up variations in the volume due to the temperature changes. Some common cases where it is necessary to have pressure maintenance are the district heating networks and hot water circulation systems in buildings [3].

The aim of this paper is to enhance the design of the control systems that help to maintain constant pressure in the pipes in spite of different phenomena that affect it in the pipeline. To develop and explore the system, an original pump test bench was used.

In this study a control system was built on the basis of AC500 PLC series of ABB, which combines some advantages of classical control methods and at the same time offers new opportunities to enhance the productivity of centrifugal pumps.

The paper is organized as follows. First, the most common traditional methods for pressure maintenance of centrifugal pumps are described. Next, the topology of the developed system is presented with the experimental outcomes. Finally, the test results are discussed and the conclusions drawn.

II. MATHEMATICAL MODEL OF THE CENTRIFUGAL PUMP

A topology of the traditional centrifugal pumping system is shown in Fig. 1. The system includes the pump fed by the variable speed drive (VSD). The supply grid, fluid pipe, a sensor producing the pressure feedback, and a discharge valve are the compulsory parts of the system. The VSD incorporates an electric motor directly connected to the pump. Other important units of the VSD are the power electronic converter and the programmable logical controller (PLC). Usually, the fluid enters the pump along the rotating axis and after acceleration by the impeller, flows outward into a diffuser or volute chamber from which it exits.

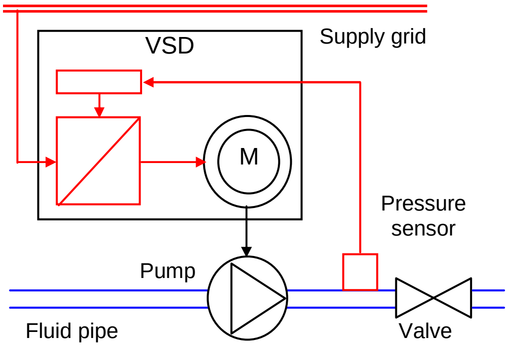

Fig. 1. Functional diagram of the pumping system.

Every pump is described by the fluid flow rate Q (m^3/s) and the fluid energy head H (m) at the definite rotational speed of the motor shaft n (rpm). The non-linear relations between the fluid flow rate and the head known as the performance characteristics, or the QH curves are usually provided by the pump manufacturers at the rated rotational speed. Some examples of the performance curves for Ebara CDX pumps from ABB are presented in [4]. Similar data can be found for example in [5].

According to the QH and QP curves, in all types of the centrifugal pumps the head decreases as the flow rate increases whereas the power grows along with the flow rate.

An important property of hydraulic systems is expressed with the affinity laws [6] that describe the relationships between variables involved in the pump performance, such as the flow rate and the head at the changing speed:

$$\frac{Q_1}{Q_2}=\frac{n_1}{n_2}, \qquad (1)$$

$$\frac{H_1}{H_2}=\left(\frac{n_1}{n_2}\right)^2, \qquad (2)$$

where index 1 denotes the initial states and index 2 – the final states of the process variables.

Every set of points found with the help of the affinity laws defines a locus of constant speed useful to predict the pump characteristics based on the known characteristics measured at another speed for the same pumps. Using the above affinity transformations, the family of the performance curves can be designed. Such a family calculated with (1), (2) for Ebara CDX 120/12 [7] is shown in Fig. 2.

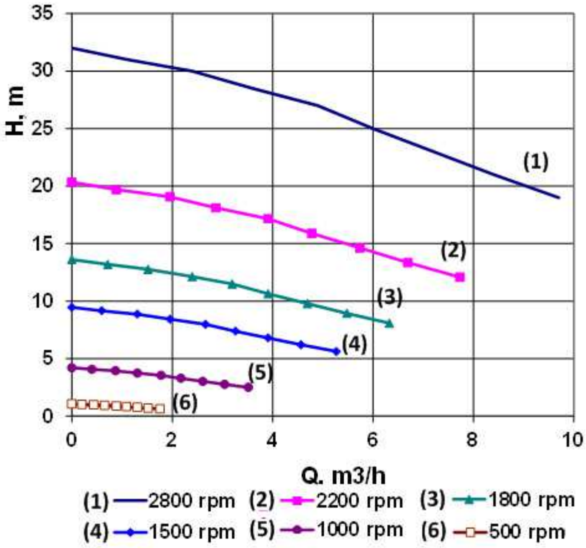

Fig. 2. A family of QH-characteristics for Ebara CDX 120/12.

Here, the curves H2800 to H500 describe the flow-head relations at 2800, 2200, 1800, 1500, 1000, and 500 rpm respectively.

III. EXPERIMENTAL DEFINITION OF THE SYSTEM CURVES

Applying the pump QH ratios to the real pumping system affects the pressure in the pipeline, which can be found from the Bernoulli's equation [8] depending on the system topology as follows:

$$p=g\rho\left(H-\frac{\upsilon^2}{2g}-z\right), \qquad (3)$$

where

p – fluid pressure, N/m^2

$g\rho H$ – initial pressure at the intake, N/m^2

$g\rho z$ – static pressure, N/m^2

$\rho\frac{\upsilon^2}{2}$ – dynamic pressure, N/m^2

$\upsilon=\frac{Q}{A}$ – fluid velocity in a pipeline, m/s

A – cross-sectional area of the pipeline, m^2

ρ – fluid density, kg/m^3

g – acceleration due to gravity, m/s^2

z – elevation of the point above the reference plane, m

Using (3), the above family of performance characteristics for Ebara CDX 120/12 was recalculated to the family of QP curves representing the flow rate versus pressure relation shown in Fig. 3.

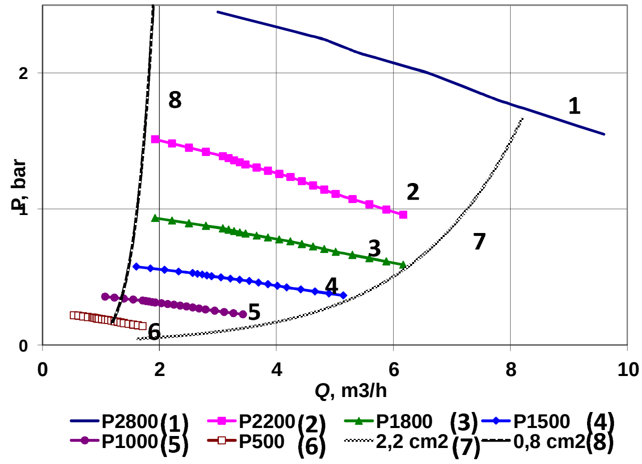

Fig. 3. A family of QP-characteristics and system curves for Ebara CDX 120/12.

Applying (3), any particular pumping application at every condition can be described with its individual system curve, which represents the resistance of the pipeline resulting from a specific consumer design [9]:

$$H_s=H_0+k_H Q_s^2, \qquad (4)$$

where H_0 is the static head that the pumping system lifts, Q_s is the system flow rate, and k_H is the head loss coefficient.

Generic shapes and slopes of the system curves cannot be provided with the manufacturer's pump datasheets because they alternate along with the consumer's environment changing. Variations in the household daily consumption are one of the factors affecting the shape of the system curve, which tends left when the resistance is high and tends right when the resistance falls. Particularly, if the pump discharge valve is closed, the provided head is at its maximum and the flow rate is near the zero point.

For the VSD-fed pumping applications, every system curve is superimposed upon the family of performance curves. The intersection of a pump performance curve with the system curve indicates the working point of pumping known as a pump operating point [9]. By plotting the points of the intersection of the system curve with the pump performance curves, a set of operating points can be found, one for every speed along the system curve. From these points, different pump operating regions can be defined, such as the best efficiency region, maximum productivity region [4], etc.

To find the system curves, an experimental ABB manufactured pumping station was used in this study. The station is composed of pumps EBARA 120/12 fed by the power electronic converters ABB ACQ810, the discharge valves for the consumer's environment emulation, the pressure sensor and the manometer, digital switches, potentiometers, relay circuitry, wiring, and data cables for RS485 connection. All the pumps are of nominal power 0.9 kW, current 3 A, voltage 400 V, and speed 2800 rpm. This station includes all the required computing hardware and software to perform the pump speed adjustment and process monitoring. The toolkit Drive Studio provides the distance station control and data acquisition.

The main variables for the experimental study are the desired and the actual speed as well as the actual speed and the pressure. The Drive Studio adjusts the speed reference for the pump, assigning the speed required for every measurement level. To explore different system curves, the operator changes the discharge valve position, thus providing the regulation of the pipe cross-sectional area. As the VSD executes the commands, the Drive Studio collects continuously updated data from experimentation. Depending on the operational state of the pump and the process system characteristics, the minimum required time interval for such estimations can vary from hundreds of milliseconds to several seconds.

Based on the above family of the performance curves, the system characteristics of the pumping station were read. To this aim, using the pressure measurements at different discharge valve positions, 2.2 cm^2 pipeline and 0.8 cm^2 pipeline, the flow rates and pressure were acquired. The family of the system curves obtained is shown in Fig. 3 above the pump performance curves. Here, the vertical traces of 2.2 cm^2 and 0.8 cm^2 bound the area of the system family studied at different discharge valve positions.

IV. SYSTEM ORGANIZATION

To play the PLC role, an AC500 PLC series was chosen with a powerful processor and a standard input/output DA501 module. For communication with other devices, this PLC uses such communication possibilities like Ethernet, Profibus DP, Modbus TCP, Modbus serial. AC500 connection with ACQ810 and the pumping station are shown in Figs. 4 and 5.

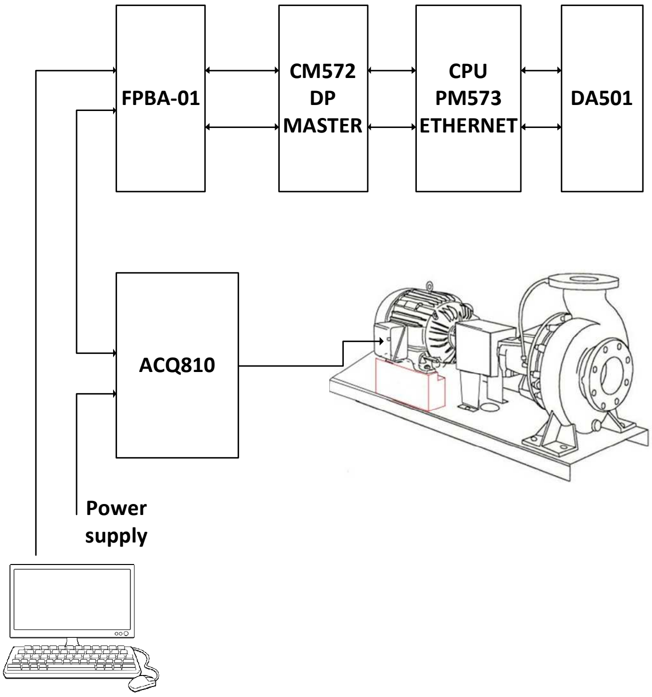

Fig. 4. Topology of the designed control system.

Here,

CM572 DP – communication module DP Master
PM573 – PLC interface module
DA501 – digital-to-analog input/output module
FPBA-01 – Profybus DP adapter module
ACQ810 – power electronic converter

For operation design, the ABB Automation Builder was used. This integrated software designed for machine builders and system integrators combines all the tools needed for configuring, programming, debugging and maintenance of the automation projects using an intuitive interface. Automation Builder platform allows common data storage and usage of the common functionality by the integrated software tools. Also, for AC500 PLC the specially designed control panel was assembled with 16 on/off switches, two resistive potentiometers and a voltage measuring device.

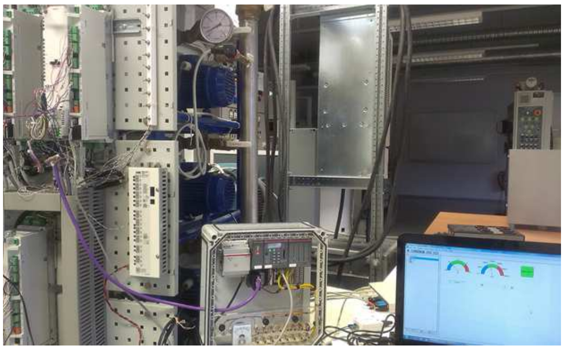

Fig. 5. Experimental pumping station.

The functional diagram of the ACQ810 VSD with AC500 PLC is shown in Fig. 6.

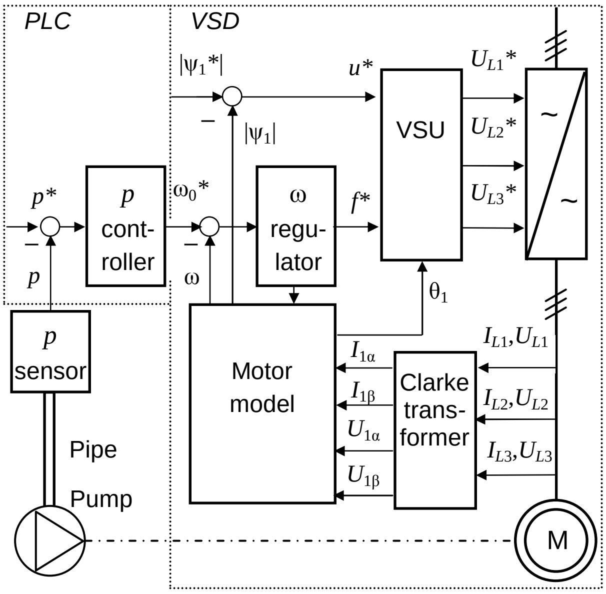

Fig. 6. Functional diagram of VSD with PLC.

Here, an induction motor M connected to the pump is supplied with a power electronic converter. The Clarke transformer converts the motor currents I_{L1}, I_{L2}, I_{L3} and voltages U_{L1}, U_{L2}, U_{L3} obtained by sensors from the natural coordinate frame L_{123} to the direct current signals $I_{1\alpha}$, $I_{1\beta}$, $U_{1\alpha}$, $U_{1\beta}$. The built-in model of the induction motor calculates the stator flux $|\psi_1|$ and the motor speed ω.

The output magnitude $|\psi_1|$ is compared with the required flux $|\psi_1^*|$ and their difference comes to the voltage switching unit (VSU) as the voltage set-point u^*. If the flux linkage ψ_1 is less than the reference flux linkage ψ_1^*, the VSU steps up the required supply voltages U_{L1}^*, U_{L2}^*, U_{L3}^*, otherwise it steps them down. The speed regulator generates the slip frequency proportional to the electromagnetic torque error, which also comes to the VSU as the frequency set-point *f**.

The input of this system is the proportional-integral-differential (PID) controller of the pressure where the difference of the pressure set-point and actual measured values is estimated. Its output serves as the reference for a speed loop. The speed loop is the innermost loop and must therefore respond quickly, as this will determine the speed response of the pump. The limiter of the pressure controller restricts the set-point pressure to a predetermined maximum value.

The system is connected via the network cable to FPBA-01 DP adapter module mounted directly on the ACQ810. The designed connection transmits commands to the converter and receives data related to the working mode of the pumping system. In this topology the control system works as a master and the converter as a slave.

The software was designed in the Controller Development System within the 3S-Smart Software Solutions (CoDeSys) V2.3 environment. This hardware-independent IEC 61131-3 Windows development system is intended for the PLC programming and creating PLC applications [10], [11], [12]. CoDeSys combines such programming tools as instruction language, structural text language, ladder diagram language, function block diagrams, and sequential function charts.

In the CoDeSys environment, a user interface has been created (Fig. 7). It includes two arrow indicators for the motor rpm speed, pressure, and an emergency situation indicator activated when the pressure exceeds the predefined limit. Using the pressure reference bar, the desired pressure level is assigned. The Start/Stop toggle button turns the pumping unit on/off.

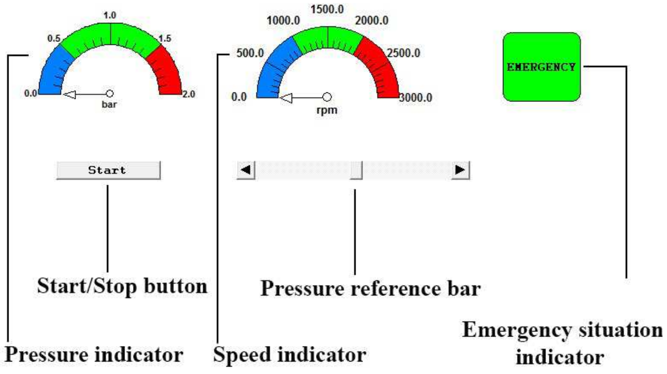

Fig. 7. User interface.

V. TUNING AND EXPERIMENTATION

The tuning process of the closed-loop drive ensures evaluation of the system response on the step inputs and disturbances as well as on the non-step inputs [13]. As a rule, permissible and approvable drive outputs meet such requirements as

- to approach the desired pressure and speed
- to keep the steady-state flux, torque, current, and voltage within the rated areas
- to provide the transient overpressure and overspeed within the rated restrictions or 150 to 200 % of the rated values for 0.2 to 10 s

The program for the pressure controller was developed using the Structure Text language and Continuous Function Chart language from the CodeSys. The controller with a transfer function of a Laplasian *s*

$$W_{r\omega}(s) = k + k_{int} + k_{dif} = k\left(1 + \frac{1}{\tau_{\text{int}} s} + \tau_{dif} s\right) \tag{5}$$

consists of the proportional (*k*), integral (k_{int}) and differential (k_{dif}) coefficients.

The transient response characteristics of a pump can be initially characterized by the first order model **Error! Reference source not found.** described with the following function:

$$p(t) = \left(p_{set}\left[1 - \exp(-\frac{t-L}{T}\right] + p_0, t \geq L \right), \tag{6}$$

$$p(t) = \left(p_0, t < L\right)$$

where

p(*t*) – output pressure without PID control
T – process time constant
L – process time delay

To find the PID parameters, use the classical Ziegler-Nichols method for determining the proportional gain, integral time constant τ_{int}, and differential time constant τ_{dif}. (Fig. 8).

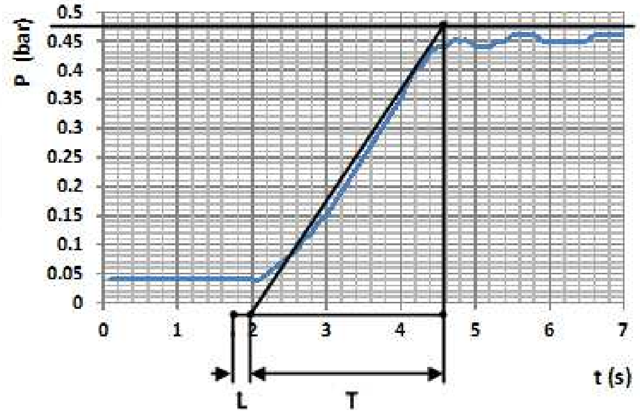

Fig. 8. Transient response and estimation of the gain and time constants.

Obtained *T* and *L* values were used for PID tuning **Error! Reference source not found.**. According to (6) and Fig. 8, the following controller parameters were obtained:

$$k = 1.2 \times \frac{T}{L} \approx 14, \tag{7}$$

$$\tau_{\text{int}} = 2 \times L \approx 0.4s, \tag{8}$$

$$\tau_{dif} = 0.5 \times L \approx 0.1s, \tag{9}$$

The fine tuning of the cascading control system is accomplished step by step. First, a proportional regulator is assigned with a *k* gain and small inputs are applied to the system without its capturing. In this way, the transients and steady processes are investigated along with the smooth rising of the *k* gain. As the gain increases, the response enhances but a too high value will make the system liable to vibrate. Next, to eliminate stationary deviations, an integral control component with τ_{int} is introduced, which slightly increases and/or decreases to improve the response. Since the system is liable to vibrate, the integral time constant change is completed. If the loop response remains slow, the differential component τ_{dif} may be added with stepping up/down gently and the overshoot and vibration are observed. Since their level becomes dangerous, the change of the differential time constant is completed. Thereby, the pressure loop is tuned.

Since the linear system is properly tuned, the regulator limiters are assigned and the testing input signals are increased to examine the drive response of the constrained system.

For the system testing in the ACQ810 pumping station, a valve was installed in the pipeline. During the experiments the valve angle changed discretely, as shown in Fig. 8.

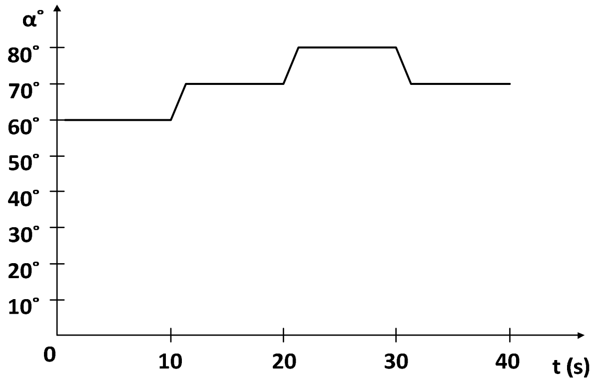

Fig. 8. Valve angle regulation diagram.

An example of the properly tuned system is shown in Fig. 9.

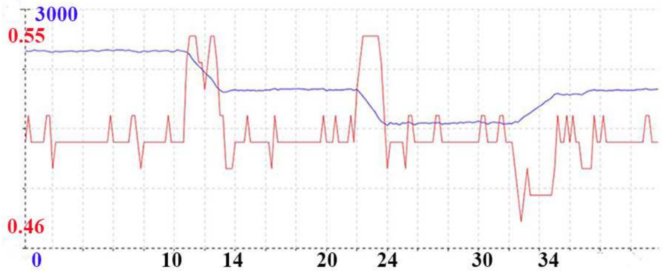

Fig. 9. Reference, pressure and speed transients.

The experiments showed that once the system curve rushes left (valve closes in Fig. 8), the working point in Fig. 3 moves up along the pump performance characteristic, thus increasing the pressure. Once the sensor feels the pressure growth, the pressure controller in Fig. 6 decreases the reference speed, motivating the VSU to select the reduced voltage space vector of the power converter. As a result, the motor speed drops stabilizing the pressure on the desired set-point level, as shown in Fig. 9. And vice versa, if the system curve rushes right (valve opens in Fig. 8), the working point moves down decreasing the pressure. Once the sensor feels the pressure lowering, the pressure controller increases the reference speed, motivating the VSU to choose the enlarged voltage space vector of the power converter. As a result, the motor speed grows, stabilizing the pressure on the desired set-point level.

VI. CONCLUSION

Though many various automatic control systems can be used for pressure maintenance in pumping applications, there is still a need in more flexible, accurate and reliable control methods. The proposed system has some advantages over other industrial equipment designed for pressure regulation. Our tests proved that the system is capable of maintaining a desired level of pressure in a wide range. It supports the required pressure level and helps to avoid serious danger in the pipeline during the operation if the pressure varies unpredictably or exceeds the limits.

ACKNOWLEDGMENT

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REFERENCES

[1] H. L. Stewart, *Pumps*, Indianapolis: Sams, 1977, 465 p.

[2] G. Hovstadius, V. Tutterow and S. Bolles, “Getting it right. Applying a system approach to variable speed pumping,” *4th International Conference on Energy Efficiency in Motor Driven Systems EEMODS 2005*, Heidelberg, Germany, 2005, pp. 304 – 314.

[3] B. Nesbitt, *Handbook of Pumps and Pumping*, Elsevier, 2006, 424 p.

[4] *Centrifugal Pumps CDX*, Available at: http://ebara-pumps-online.com/CDX.pdf.

[5] *Danfoss Products and Solutions*, Available at: http://www.danfoss.com.

[6] P. Girdhar and O. Moniz, *Practical Centrifugal Pumps: Design, Operation and Maintenance*, Imprint Oxford [etc.]: Newnes, 2005, 250 p.

[7] L. Gevorkov, I. Bakman, V. Vodovozov, “Hardware-in-the-loop simulation of motor drives for pumping applications,” *Electric Power Quality and Supply Reliability Conference PQ 2014*, Rakvere, Estonia, 2014, pp. 1 – 6.

[8] J. E. Finnemore and J. B. Franzini, *Fluid Mechanics with Engineering Applications*, NY: McGraw Hill, Inc., 2002, 500 p.

[9] I. I. Ionel, *Pumps and Pumping with Particular Reference to Variable-Duty Pumps*, Amsterdam: Elsevier, 1986, 715 p.

[10] *CODESYS in Embedded Automation Complete IEC 61131-3 IDE for industrial embedded applications*, Available at: http://www.codesys.com/.

[11] T. Avery, *CoDeSys*, Cel Publishing, 2012, 96 p.

[12] B. Vogel-Heuser, *Automation & Embedded Systems*, Oldenbourg Industrieverlag, 2008, 78 p.

[13] V. Vodovozov, *Electrical Drive: Performance, Design and Control*, Lambert Academic Publishing, Saarbrücken, 2014, 320 p.

[14] R. Shell, *Handbook of Industrial Automation*, CRC Press, 2000, 904 p.

[15] K. Ogata, *Modern Control Engineering*, Prentice Hall, 2010, 894 p