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Operation of pumps in a district heating system supplying a distant major industrial user

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Abstract

Water pump control methods are presented for a district heating system supplying an industrial heat user. The analysis sought to determine the optimum operating conditions for the pump system. The study focused on control parameters of the pumps installed at the heating substation delivering the heated water to the end-user. The chosen method consisted of throttling control of the pump using a control valve in the discharge header. The values of characteristic parameters and control characteristics of the pumps were used to evaluate the performance of the pump system.

Keywords: district heating system, pump, water flow rate, pump control.

1. Introduction

Sustainable energy policy trends have presented the Polish energy environment with a task to ensure efficiency in the management of electricity and heat production resources. One major challenge relates to the obligation to make a significant reduction in CO_2 emissions from 2013 onwards. Leaving aside political issues, all appropriate measures have to be taken to systematically reduce emissions. Increasing the energy efficiency of all partial processes in electricity and heat production is one possible and efficient way to do this. Through reduced energy intensity and improved failure rate of system components, including pumps operating in district heating systems, operating costs could be far lower. To achieve reliable and economic performance of equipment, all guidelines relating to operation, including operation modes, defect prevention and recovery have to be observed. Failure to comply with these requirements results in serious equipment operation problems that cause substantial losses in productivity and performance.

The optimal method of pump operation control consists of providing sufficient pump system capacity to meet the needs of the system and to economize on power consumption while taking into account potential constraints and reliability requirements.

This paper analyzes the methods of regulating the pumps transporting district heating system water in the heating circuit supplying an industrial end-user. This study seeks to determine optimal conditions for pump system operation. The study focuses on the performance parameters of pumps installed at the district heating station providing heated water to the Osiek Sulfur Mine (KSO). The method used consists of throttling control of the pump using a control valve in the discharge header. The values of characteristic parameters and control characteristics of the pumps were used to evaluate the performance of the pump system.

2. Control of pump performance parameters

Liquid transport in heating systems consumes huge amounts of electrical energy. Significant energy savings in a pump system can be achieved through efficiently regulating the pump operating conditions. For the hot water system analyzed in this study, the following basic pump performance control methods [1–4] are discussed.

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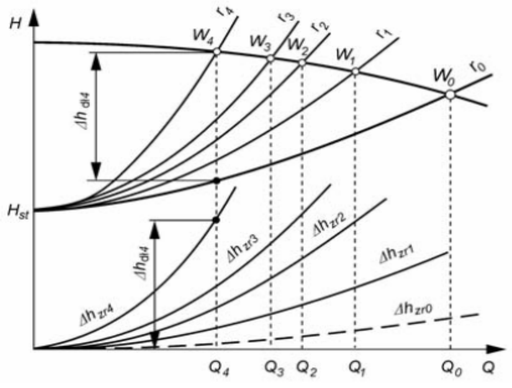


Figure 1: Change in pipeline characteristics $r = Hp(Q)$ and operating point W of the pump as a result of the change in the control valve opening level [1]

Regulation through throttling control. Throttling is performed by adjusting the control valve placed in the discharge line of the pump. When the valve is throttled, the water flow is restricted and the pump head changes. Regulation of the heating system pump adjusts operation parameters: water volumetric flow rate Q, pump head Hp and the installation head Huk to varied in time t demands of the end user. In contrast to the pump head, the water volumetric flow rate is the parameter regulated most often; alternatively, both parameters are jointly adjusted. Pressure drop $\Delta p_{z_r} = \Delta h_{dl}$ in the control valve depends on the valve opening level, expressed as 1.

$$\varDelta h_{z_r} = \frac{\varDelta p_{z_r}}{\rho g} = \frac{\xi c^2}{2g} \tag{1}$$

where: Δhz_r—head loss on the valve; constant $c = Q/A$; area $A = \pi d2/4$, d—nominal diameter of the valve; ρ—water density; g—gravitational acceleration; ζ—loss coefficient.

Closing the valve z_r increases angle α, and by altering the characteristics of the pipeline, changes the position of the pump operating point W0→W1→W2→... (Fig. 1).

Thus, for the water volumetric flow rate Q, the lost power P_z will be

$$P_z = \frac{\rho g Q \varDelta hz_r}{\eta} \tag{2}$$

where: Δhzr—head loss due to partial closing of the valve, ρ—water density, $\underline{g}$—gravitational acceleration, η–pump efficiency.

The energy effect of throttle control is thus identical to that obtained when a new pump with lower head and efficiency is installed.

Regulation through changes in rotational speed. This method is most effective in those pump systems where head loss dominates over the static pumping head (e.g. circulating pumps).

Its widespread use is a result of adopting frequency inverters to variably adjust the rotational speed of the pump asynchronous motors. Three basic pump operating parameters change with the pump speed change: water volumetric flow rate, pumping head, and power on the shaft. Performance characteristics of the pump are described by the following relationships

$$\frac{Q_1}{Q} = \frac{n_1}{n} \tag{3}$$

$$\frac{H_{pl}}{H_p} = \left(\frac{n_1}{n}\right)^2 = \left(\frac{Q_1}{Q}\right)^2 \tag{4}$$

$$\frac{P_1}{P} = \left(\frac{n_1}{n}\right)^3 \frac{n}{n_1} \tag{5}$$

where: Q_1 is the water volumetric flow rate after the speed change, n and n_1 denote the rotary motion of the pump impeller before and after the speed change, H_{p1} is the pumping head after the change in impeller rotational speed, P and P_1 denote power on the pump shaft before and after the impeller speed change.

Although this type of control needs relatively high financial investment, it delivers dramatic energy savings compared with throttling steering.

Regulation through changes in rotational speed. In practice, nominal pump operating parameters often exceed the values of both the water volumetric flow rate and flow resistance needed by the pump system. To minimize throttling losses, the pump operating parameters can be reduced by trimming the outer diameter of the impeller (or impellers in multistage pumps). The liquid flow conditions change with the change in impeller tangential velocity and (to a lesser degree) the radius angle of the vane at the outlet, the width, length and contact ratio of the vanes. Impeller outer diameters can be reduced by approximately 20% with no significant effect on pump efficiency. The impeller diameter d' after cutting is determined from the following dependence (assuming constant pump efficiency before and after impeller design changes)

$$d' = d\sqrt{\frac{H_p'}{H_p}} = \sqrt{\frac{Q'}{Q}} \tag{6}$$

where: d is the impeller diameter before cutting, H_p' is the pumping head after impeller diameter reduction, Q'

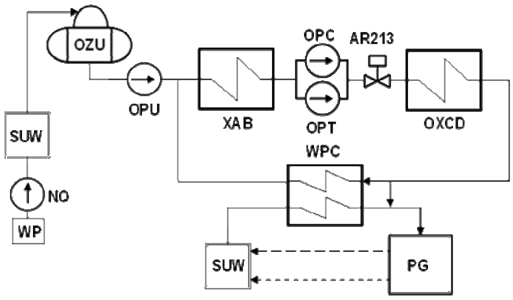


Figure 2: A simplified flow chart of the water closed circuit covering the power plant and the sulfur mine: WP—post-cooling water, NQ—after-cooling water pump, SUW—water treatment station, OZU—water storage tank, OPU—water storage pump, XAB—primary heat exchanger, OPC—hot water pump, OPT—district heating water pump, AR213—control valve, OXCD—peak demand heat exchangers, WPC—recuperative heat exchanger, PG—minefield, SUW—water treatment station

is the water volumetric flow rate after the diameter reduction.

2.1. Pump control in the analyzed district heating circuit

The district heating unit at the Połaniec Power Plant heats and transports water to a sulfur mine in Osiek, 12 kilometers away, where hot water is used to extract sulfur through the Frasch process. Sulfur is melted underground by superheated water and the liquid mixture of sulfur and water is forced to the surface through specially designed production boreholes. The water removed from the depression boreholes is sent to the water treatment station, where it is reheated in heat exchangers. The extraction of sulfur is conducted in a continuous process [5]. The heating system has been designed as an open type, operating at non return water transport with the mass flow rate of 91.67 kg/s (330 Mg/h), which corresponds to the heat flux of 70 MW_t; or as a partially open type, in which 60% water flow returns to the power plant. The nominal heat flux is also 70 MW_t. The water supplied to the heating system and that used to replenish water loss in the mine is taken from the system outlet of the water that cools the steam turbine-set condensers (WP), called post-cooling water. It is then sent to the water treatment station (SUW), where it is softened and mechanically cleaned and then sent to storage tanks (OZG). The water is heated in heat exchangers (XAB, OXCD) at four stages of heating and reaches a nominal temperature of 190°C. The return water from the Osiek Sulfur Mine (KSO) with a temperature of 60...90°C and pressure of approximately 0.7...0.9 MPa is re-circulated to power unit basic heaters, condensate heaters and finally to peak-load heaters. Then the water is transported through the water main pipelines to the re-

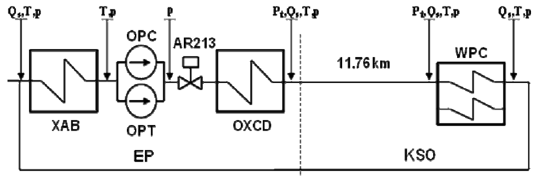


Figure 3: Flow chart of the district heating water closed loop with the measurement point indicated

cuperative heat exchangers (water-water) in KSO. A simplified schematic diagram of the closed water circulation loop in the power plant and the sulfur mine is shown in Fig. 2. Power plant pumps (NQ) compensate the heating water losses in the district heating system.

The designed heating system uses multistage pumps connected to the transport pipes in pressure mode. The pump nominal mass flow rate is 180.6 kg/s at the head of 1.76 MPa. The pumps are driven by 24.75 1/s, 500 kW ring—type motors. Under normal KSO working conditions, one pump is sufficient to cover the needs of the end user. However, in view of hot water supply reliability, scheduled pump—motor maintenance work and the variable demand for heat in the sulfur mine, there is a need to provide stand—by and regulation of the water mass flow rate in the system. The pump operating parameters were adjusted to the variable requirements of the pump system by:

a) thyristor cascade used in the OPT pump systems,
b) ontrol valve fixed in the discharge header of OPT, with the thyristor cascade off,
c) transition to OPC pumps in the period of lower hot water demand,
d) varying the number of pumps switched in parallel connection.

As the thyristor cascades were withdrawn from service, currently the water outlet pressure is regulated through an AR213 pneumatic pressure regulator mounted on the OPT and OPC pumping pipes. System water pumps OPT are multistage pumps for transporting system water within a partially closed loop. Under normal conditions of water circulation, the mass flow rate of water flowing out of the OPT pump is approx. 125.0...152.8 kg/s (450...550 Mg/h). OPC pumps have the same functions as OPT pumps. One pump provides KSO with water mass flow rate of approx. 97.2 kg/s (350 Mg/h). Two OPC pumps in parallel connection are acceptable in an emergency and when the required mass flow rate must exceed 97.2 kg/s (350 Mg/h).

Table 1 shows nominal operating parameters of OPT and OPC pumps. The measurement points for the key parameters in the water loop are shown in Fig. 3.

Table 1: Nominal values of OPT and OPC pump parameters

Name	OPT	OPC
	Pump	
Type	rotodynamic	centrifugal
Model	250W P–3	20W30x46V
Water volumetric flow rate, m^3/s (m^3/h)	0.1806 (650)	0.0917 (330)
Rotational speed, 1/s (rpm)	24.7 (1480)	24.7 (1480)
Head, MPa (m)	1.77 (180)	1.71 (174)
	Motor	
Type	induction, three–phase,	asynchronous
Model	SCh cm 124 SE	SZco 194s
Power, kW	500	320
Voltage, V	6000	6000
Current, A	58.1	38
Power factor	0.89	0.87
Rotational speed, 1/s (rpm)	24.7 (1485)	24.8 (1490)

3. Characteristic quantities of the heating system and control characteristics of system water pumps

The measuring control apparatus used in this study has valid certification in compliance with the requirements of the Quality Management System (SZJ). A schematic diagram of the water system, its main equipment and measurement points is shown in Fig. 5.

The following parameters were measured to evaluate the performance of the heating system:

a) heating water mass flow rate,
b) pressure/head:
  - discharge head and suction lift for OPC and OPT pumps,
  - in the outlet of the control valve, at the district heating unit outlet towards KSO,
  - on the return pipeline from KSO and to the XAB heaters,
  - on the water-water exchangers in KSO (data received from KSO).
c) heating water temperature,
d) electric power, voltage and current of the pump prime movers.

Current working conditions of the heating unit were taken into account in the tests, which included [6]:

- measurement of hydraulic resistances in the heating water system of the district heating unit,

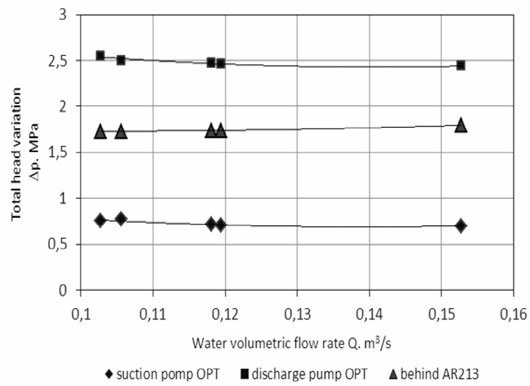


Figure 4: Hydraulic resistance characteristics in the heating water system

- measurement of OPT and OPC pumps efficiency,
- determination and assessment of the energy results of OPT pump throttling,
- determination and assessment of the energy results of trimming the OPT2 pump impeller,
- comparative evaluation of both regulation methods.

The test results are discussed and illustrated in figures later in this paper. Total hydraulic resistance in the heating water system is equal to the OPT pump head minus the pressure drop in the control valve. Fig. 4 shows volumetric flow rate range of 0.100. . . 0.153 m3/s (360...550 m3/h), the recorded hydraulic resistance values were 0.96. . . 1.08 MPa (98. . . 110 m H_2O). The water pressure drop in the control

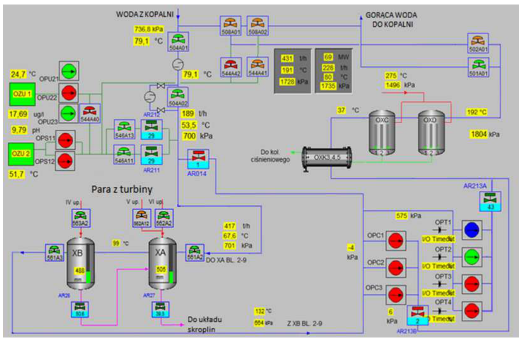


Figure 5: Diagram of system water circuit, main equipment and measurement points

valve was 0.63. . . 0.80 MPa (64.5. . . 82.0 m H_2O). In the heating system analyzed, water pressure rises at the inlets of OPT and OPC pumps to protect them from the effects of cavitation through operation of the makeup water pumps OPS and OPU (Fig. 5).

They also contribute to the reduction in pressure drop in the system of basic heat exchangers XAB. Owing to the obtained pressure reserve, the heating system can operate at lower pressure on the discharge side of the OPT pump, with no rise in water temperature at the outlet to KSO. Lowering the discharge pressure by approx. 0.35 MPa allows the pump to operate within a parameter range where cavitation does not occur. The pump efficiency values shown in Fig. 6 are in the following ranges:

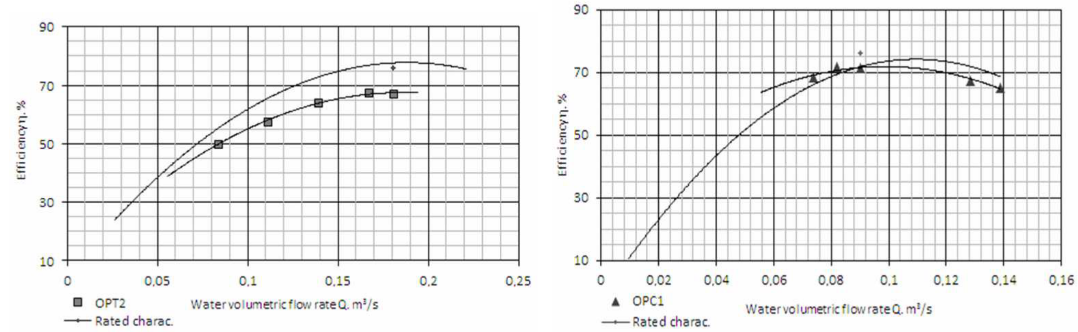
- approx. (50. . . 67)% for OPT2, at the head of approx. 1.95. . . 2.08 MPa (199...212 m H_2O), for volumetric flow rates
approx. 0.086. . . 0.181 m3/s (310. . . 650 m3/h),
- (65. . . 71.6)% for OPC1, at the head of
approx. 1.17. . . 1.65 MPa (119...168 m H_2O), for volumetric flow rates approx. 0.074. . . 0.128 m3/s (265.2. . . 462.3 m3/h).

The measured efficiency of OPT2 is about 12% lower than the rated efficiency given by the manufacturer. Changes in OPC1 pump efficiency correspond with the manufacturer's specification. Considering the efficiency values calculated from efficiency measurements, the condition of the pumps qualifies as good. With one OPT pump operating, the water mass flow rate for KSO was changed in the range 83.88. . . 150.00 kg/s (300...540 Mg/h) at the thermal power of 52...90 MWt. The heating water pressure at the heating station outlet (after heaters OXCD) was maintained at the level of about 1.8 MPa using the control valve. With the OPC pump, the water mass flow rate to KSO was varied within the range 83.88. . . 180.06 kg/s (300...648.2 Mg/h) at thermal power of 52...70 MWt. Water pressure was maintained within 1.5...1.6 MPa using the control valve. The OPT2 pump was reconditioned and the diameter of its impeller was reduced. The regulation test for this OPT2 pump was made for a reduced impeller diameter equal to 90% of its rated value.

Pump heads were 1.57. . . 1.65 MPa (160.7. . . 168.7 m H_2O), respectively. For the reference volumetric flow of 0.125 m3/s (450 m3/h), the power on the shaft after the impeller diameter reduction is lower than the primary power of about 90. . . 100 kW. The operating point of the OPT2 pump changed, but the pump head obtained was sufficient to ensure proper operation of the hot water system for the pipeline hydraulic resistance of 0.96. . . 1.08 MPa (98.0. . . 110.0 m H_2O). The modified pump is provided with a suitable hydraulic standby in case operating conditions worsened. This is important since excessive reduction in the impeller diameter lowers the pump head irreversibly. In the evaluation of system performance it is assumed that a properly selected pump should operate at fully opened discharge valve, with no throttling. The OPT pumps, with no rotational speed regulation or impeller diameter modification, operate at a significant pressure drop on the control valve, which at the water volumetric flow rate average value of 0.125 m3/s (450 m3/h), is 0.80 MPa (82 m H_2O) with total hydraulic resistance of the system (without the control valve) of 1.21 MPa (123 m H_2O). Then the pump head achieves the excessive value of up to 2.02 MPa (206 m H_2O). Depending on the end user's demand, the following pumps can operate in the heating system:

- OPC at the hot water volumetric flow rate of up to 0.097 m3/s (350 m3/h),
- OPT2 at the hot water volumetric flow rate from 0.083 m3/s (300 m3/h) to 0.139 m3/s (500 m3/h),
- other OPT pumps at the hot water volumetric flow rates from 0.111 m3/s (400 m3/h) to 0.181 m3/s (650 m3/h).

The alternative pump operation described here in the heating system helps optimize energy consumption and lower operating costs. Regulation of OPT2 pump parameters, for the reference volumetric flow rate value of 0.125 m3/s (450 m3/h), compared with the throttling control reduces the power input by about 100 kW—after the impeller diameter reduction of 10% with throttling continued. When only the OPC pump is used at the volumetric flow rate of 0.125 m3/s (450 m3/h), the calculated drop in electrical energy consumption is approx. 200 kW. When the OPC pump operates at the volumetric flow reference value of



(a) OPT2

(b) OPC1

Figure 6: Hot water pump efficiency

0.083 m3/s (300 m3/h), the electric power input can also be reduced. For that purpose, the diameter of the impeller should be reduced by 10%. This will prevent the OPC pump from replacing the OPT pump at the heating water volumetric flow rate of approximately 0.111 m3/s (400 m3/h), which is undesirable. Since the heating system installation supplying heat to an industrial user operates continuously at a steady water flow rate, the results presented in this paper, based on the pump efficiency measured values, can be used to evaluate the performance of the pumps and the entire system.

4. Conclusions and comments

Based on the results of the analysis conducted, the following general conclusions are appropriate. One of the major operating problems of pump systems is the optimization of energy consumption in existing systems. The optimization can involve a more efficient adaptation and adjustment of the pump power generation capability and performance parameters to the service conditions of the system. This can be achieved by providing appropriate modifications during repairs. Proper operation and diagnosis allows plant operators to upgrade the pumps through water flow adjustment and elimination of energy loss inside the pumps. Identification of the operating condition of the pumps is an essential prerequisite for improved energy efficiency. It is the basis for decisions related to the best selection of pumps in terms of the set service conditions, correction of the pump parameters or changes in the pump design. The accuracy of a pump or a pump system performance assessment is directly related to the number of monitored and controlled parameters, the accuracy of the measurement and determination of optimal characteristics of the system and its components. The adjustment of performance characteristics of the pumps, which involved changes in their design (reduced impeller diameters) to match operating conditions, led to reduced levels of power required to supply the pump engines—without the extra costs involved in using expensive frequency converters. More detailed conclusions derived from the analyses of the pump and the heating system performance are presented below. The measured pump efficiencies are in the range (50. . . 67)% for the OPT2 pump—before the impeller reduction and (65. . . 71.6)% for the OPC1 pump. The underperforming pump OPT2 qualified for an overhaul. The OPT pumps work is normally performed at a significant drop in pressure value on the control valve, which at an average water flow of 0.125 m3/s (450 m3/h) is more than 0.80 MPa (82 m H_2O) or 1.21 MPa (123 m H_2O) at the total hydraulic resistance of the system (without the control valve). The adjustment of the OPT2 pump parameters after the repairs and the 10% reduction in impeller diameter helped reduce the power necessary to supply the pump engine relative to the throttle control used to date. The hydraulic resistance of the heating water system during OPT2 pump operation after impeller diameter reduction was 1.08 MPa (110 m H_2O). The water pressure drop on the control valve was 0.80 MPa (82.0 m H_2O) at the water flow of 0.153 m3/s (550 m3/h). The power on the pump shaft for the reference value of the pump flow of 0.125 m3/s (450 m3/h) is 90 to 100 kW lower than the initial/original power, bringing tangible benefits in terms of reduced power and electrical energy demand. With the use of the OPC pump, at water flow of 0.125 m3/s (450 m3/h), the established reduction in electrical energy consumption is about 200 kW. Since the pump is operating outside of the reference water flow range, the flexibility of the heating system is limited, rendering this option impracticable. The optimal pump for this system should operate with the discharge control valve fully open and without throttle control, which will further reduce its electricity consumption.

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