Parameter control of the adjustable pipeline heat supply system

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Abstract. Controlled pipeline heat supply systems are equipped with shut-off valves that causes a decrease in the stabilization level of the heat-transfer agent transportation process and reducing the pressure in the network during transition processes. Booster pumping stations are installed to stabilize the pressure in the heat supply system. The heat-transfer agent transportation process becomes turbulent that significantly reduces the hydraulic system stability when such systems are working. The process of transporting the coolant. Regulator that contains proportional, integral, and differential control components is used to control the dynamic parameters. Studies have revealed the reduction level of dynamic characteristics depending on the regulator components. The use of PID control allows to change the oscillatory process into acyclic process.

Introduction
Modern heat supply systems are the facilities with distributed parameters and have a significant pipelines length. Pressure drop at the end sections of the chain, pressure boost in the return pipelines, and insufficient pressure drop occur while the heat-transfer agent transportation through long pipelines. Therefore, heat networks are equipped with automated pumping stations and shut-off valves. This leads to nonsteady processes in their work and technological parameters changes. Therefore, the pressure changes in the system during starting operation mode and transition process. The process takes on an oscillatory form. Cavitation can also occur on the suction lines in the network. All this reduces the hydraulic system stability. Famous scientists studied the parameter control issues of an adjustable pipeline heat supply system.

In the works [1-3], the issues of the load flow in the heating pipeline system with independent circulating pump connection and their mathematical simulation in heat points are considered. However, the author does not consider the system condition on the stability theory criteria. In the works [4-5], the issues of heat networks parameter optimization in the Western Siberia conditions were considered, and the frequency-response analysis of the dynamic component of the heat transfer agent behavior in the main pipeline was given. The results of the research allow to increase the hydraulic control stabilization of the heat supply system. However, the control of dynamics in transition processes by the proportional-integral-derivative controller requires additional studies in terms of the ratios of proportional, integral and differential components.

In the works [6-7] heat calculation of the hot water supply transmission networks systems, analysis techniques of thermal and hydraulic conditions and computerization of district heat supply systems are
given. The issues of a synthesis of relevant parameters are not considered in these works. These issues are partially considered in the studies of the authors [8-10].

Overview of district heating supply and cooling systems for a sustainable future is provided, issues of sustainable design of renewable energy supply chains integrated with district heating supply systems are addressed, assessment of approaches to simulate a temperature wave in the heating pipe mains area is evaluated, connection method between urban heat supply systems is provided and also a comparative analysis of the residential heat supply system of Japan and the district heating system based on local sources using the SPECO method is undertaken, modeling and analysis of the heating network state are provided in works of foreign authors [11–12].

The purpose of these studies is to improve the energy efficiency of the pipeline heat supply system in a transient process.

Transition processes as well as the system operation in the starting operation mode significantly influence on the process parameters nature of the pipeline heat supply system. The main characteristic of modern control valves (thermic controllers, heating agent flow regulators, balancing valves, etc.) is accepted a relieving capacity depending on the volumetric water discharge at a pressure drop of 105 Pa. So a ratio change of the relief valve leads to a parameters change such as pressure drop and coolant flow through the valve. All this significantly reduces the stability of the pipeline heat supply system and can lead to the misalignment [7].

Dependences of the pressure change in the pump starting operation mode are shown in figure 1.

![Figure 1. Dependences of the pressure change in the pump starting operation mode](image)

Analysis of the pressure change dependence in the pump starting operation mode shown in figure 1. The start of the main line pump occurs on a closed valve. The starting time of the pump ranges from 1 to 10 sec. The heat transfer agent pressure at start on a closed valve at the outlet is 20 m. After 10 sec., it is characterized by opening the valve at the pipeline section adjacent to the pump. At the same time, there is a pressure decrease at the outlet. The pressure on the pipeline sections increases when switching to the operating mode after 50 sec. At the same time, the consumption of a heat-transfer agent increases. This process takes 20 seconds. Heat transfer agent sharp increase leads to the hydraulic resistance change of the pipeline and is accompanied by a pulsating character. The pressure amplitude of oscillations is greater in the area adjacent to the pump unit. Such a process is accompanied by the technological and energy parameter change of the adjustable pipeline heat supply system.

To control the processes in the pipeline sections, the PID Controller is used, see: figure 2. The signal from the temperature sensor is fed to the regulator, then to the controlled facility (circulation pump and valve). Perturbation influence effects on the system. As a result, a control signal, which action is aimed at the dynamic characteristics control of the heat supply system is produced at the regulator output.
Figure 2. Block diagram of the tuning parameters synthesis

The transfer function of the adjustable pipeline system in accordance with the block diagram is written in the form [13]

\[ W_{OP}(p) = \frac{Ke^{-\tau B,O^*p}}{T_{B,O}^*p + 1} \]  (1)

where \( k \) – facility gain constant;
\( \tau \) – delay time for turning on / off the device;
\( T_{B,O}^* \) – time constant;
\( p \) – operator.

The transfer function of the PID controller is

\[ C(s) = K_p + \frac{K_i}{s} + K_d s = \frac{K_d s^2 + K_p s + K_i}{s} \]  (2)

where \( s \) – Laplace converter;
\( K_p, K_i, K_d \) – are the coefficients of proportionality, integration and differentiation.

The facilities control of the heat supply system are the pump and the valve. The pump provides the fluid supply into the pressure pipe [14], which is an elementary pipeline section. In this regard, the physical processes in the pump and the adjacent pipeline can be described by RLC - circuit with transfer functions of the form

\[ W_{N_i}(p) = \frac{Q(p)}{H_{N_i}(p) - \Delta H_{N_i}(p)} = \frac{1}{T_z \cdot p + 1}; \]  \( i \)  (3)

\[ W_{N_e}(p) = \frac{\Delta H_{N_e}(p)}{Q(p)} = R_{N}(p); \]  \( i \)  (4)

\[ W_{N_r}(p) = \frac{\Delta H(p)}{Q(p) - Q_{r,}(p)} = \frac{1}{T_e \cdot p}; \]  \( i \)  (5)

\[ W_{N_z}(p) = \frac{Q_{z,}(p)}{\Delta H(p)} = R_{Cz}(p). \]  \( i \)  (6)

Where \( T_z = T_N + T_u \) – time of the heat transfer agent passage through the pump and the adjacent pipeline, \( c \); \( T_N = 4(d_2 - d_1)/(\Omega z_p \ln(d_2 / d_1)) \) – time constant of the pump, \( c \); \( d_2, d_1 \) – inlet and outlet diameters of the circular grating of the centrifugal pump, \( m \); \( \Omega \) – relative velocity of the fluid in the interblade space; \( z_p \) – blades number of the circular grid; \( T_u \) – pipeline inertial time constant, \( c \); \( T_e \) –
pipeline capacitive time constant, \( c; H_N(p) = H_0(p)\nu^2(p) - \Delta H_N(p) = H_0(p) - R_{\text{RVN}}Q^2(p) - \)
pressure developed by the pump, \( m; \nu(p) = \frac{\omega_1(p)}{\omega_2(p)} \) - speed ratio of the pump impeller; \( \omega_1(p), \omega_2(p) \) - current and nominal frequencies of the pump impeller rotary velocity, \( c^{-1}; H_0(p) \) - shutoff head, \( m; \Delta H_N(p) \) - pressure loss in the pump, \( m; Q(p) \) - pumping power, \( m^3/c; \text{RVN} \) - internal pump resistance, \( c^2/m^3; \Delta H_2(p) \) - pressure losses in the hydraulic system (in this case \( \Delta H_2(p) = \Delta H(p) \), \( m; \Delta H(p) \) - pressure losses in the adjacent pipeline section, \( m; R_\chi(p) = R_\chi(p) + R_{\text{RVN}}(p) \) - hydrodynamic resistance of the adjacent pipeline section, \( c^2/m^3; R_{\text{RVN}}(p) \) - intrinsic hydrodynamic resistance of the pipeline section, \( c^2/m^3; R_{\text{RVN}}(p) = \frac{\zeta_{\text{RVN}}(p)}{2gs^2} = K_{\text{RVN}} \times V_{\text{RVN}}(p) \) - hydrodynamic resistance corresponding to the presence of cavitation in the fluid flow, \( c^2/m^3; V_{\text{RVN}}(p) \) - the cavity pocket volume, \( m^3 \).

The valve transfer function is written as

\[
W_{\text{OP}}(p) = \frac{K \cdot e^{-\tau_{\text{B,O}}p}}{T_{\text{B,O}} \cdot p + 1}, \tag{7}
\]

where \( k \) - facility gain constant; \( \tau \) - delay time for turning on / off the device; \( T_{\text{B,O}} \) - time constant; \( p \) - operator.

As a result of the synthesis of the two systems, we obtain an integral pressure control in general terms. The transfer function of the pipeline section will be written as

\[
W_{\text{OP}} = \frac{k \cdot e^{i\omega_0p}}{T_{\text{B,O}} \cdot p + 1} + \frac{k_{\text{RVN}}}{T_{\text{n}} \cdot p + 1} + \frac{k_{\text{f}}}{T_{\text{f}} \cdot p + 1} + \frac{k_{\text{f}} \cdot T_{\text{f}} \cdot p + k_{\text{f}} \cdot e^{i\omega_0p} \cdot T_{\text{f}} \cdot p + k \cdot e^{i\omega_0p} \cdot T_{\text{f}} \cdot p + k \cdot e^{i\omega_0p} \cdot p \cdot T_{\text{f}} \cdot p + k \cdot e^{i\omega_0p} \cdot p \cdot T_{\text{f}} \cdot p + k \cdot e^{i\omega_0p} \cdot p \cdot T_{\text{f}} \cdot p + k \cdot e^{i\omega_0p} \cdot p \cdot T_{\text{f}} \cdot p + k \cdot e^{i\omega_0p} \cdot p \cdot T_{\text{f}} \cdot p + k \cdot e^{i\omega_0p} \cdot p \cdot T_{\text{f}} \cdot p + k \cdot e^{i\omega_0p} \cdot p \cdot T_{\text{f}} \cdot p + k \cdot e^{i\omega_0p} \cdot p \cdot T_{\text{f}} \cdot p + k \cdot e^{i\omega_0p} \cdot p \cdot T_{\text{f}} \cdot p + k \cdot e^{i\omega_0p} \cdot p \cdot T_{\text{f}} \cdot p + k \cdot e^{i\omega_0p} \cdot p \cdot T_{\text{f}} \cdot p + k \cdot e^{i\omega_0p} \cdot p \cdot T_{\text{f}} \cdot p + k \cdot e^{i\omega_0p} \cdot p \cdot T_{\text{f}} \cdot p + k \cdot e^{i\omega_0p} \cdot p \cdot T_{\text{f}} \cdot p + k \cdot e^{i\omega_0p} \cdot p \cdot T_{\text{f}} \cdot p + k \cdot e^{i\omega_0p} \cdot p \cdot T_{\text{f}} \cdot p + k \cdot e^{i\omega_0p} \cdot p \cdot T_{\text{f}} \cdot p + k \cdot e^{i\omega_0p} \cdot p \cdot T_{\text{f}} \cdot p + k \cdot e^{i\omega_0p} \cdot p \cdot T_{\text{f}} \cdot p + k \cdot e^{i\omega_0p} \cdot p \cdot T_{\text{f}} \cdot p + k \cdot e^{i\omega_0p} \cdot p \cdot T_{\text{f}} \cdot p + k \cdot e^{i\omega_0p} \cdot p \cdot T_{\text{f}} \cdot p + k \cdot e^{i\omega_0p} \cdot p \cdot T_{\text{f}} \cdot p + k \cdot e^{i\omega_0p} \cdot p \cdot T_{\text{f}} \cdot p + k \cdot e^{i\omega_0p} \cdot p \cdot T_{\text{f}} \cdot p + k \cdot e^{i\omega_0p} \cdot p \cdot T_{\text{f}} \cdot p + k \cdot e^{i\omega_0p} \cdot p \cdot T_{\text{f}} \cdot p + k \cdot e^{i\omega_0p} \cdot p \cdot T_{\text{f}} \cdot p + k \cdot e^{i\omega_0p} \cdot p \cdot T_{\text{f}} \cdot p + k \cdot e^{i\omega_0p} \cdot p \cdot T_{\text{f}} \cdot p + k \cdot e^{i\omega_0p} \cdot p \cdot T_{\text{f}} \cdot p + k \cdot e^{i\omega_0p} \cdot p \cdot T_{\text{f}} \cdot p + k \cdot e^{i\omega_0p} \cdot p \cdot T_{\text{f}} \cdot p + k \cdot e^{i\omega_0p} \cdot p \cdot T_{\text{f}} \cdot p + k \cdot e^{i\omega_0p} \cdot p \cdot T_{\text{f}} \cdot p + k \cdot e^{i\omega_0p} \cdot p \cdot T_{\text{f}} \cdot p + "class"=""style"=""font-size"=""24px""">
**Figure 3.** Transient curves of the heat supply system during regulation

1 - proportional component of the regulator; 2 - integral component of the regulator; 3 - differential component of the regulator

Transient curve of the heat supply system during the system regulation by a proportional regulator shows that the process is of oscillating nature with an oscillation amplitude of 0.5 mm and a frequency of 20 Hz. Figure 3 shows a transient curve of the heat supply system during the system regulation by an integral regulator. The nature of curve has changed, the oscillation amplitude has decreased to 0.4 mm and a frequency of 10 Hz. Regulation of the heat supply system by a proportional-integral-derivative controller ensures the obtaining of a transient curve of an aperiodic mode and approaches the mode embedded in the device controller. Due to the integral and differential PID component presence – a controller at a certain ratio of $k_p$, $k_i$ and $k_d$, the transients dynamics is smoothed. System transient characteristic has a static response. In the linear version it is controlled and with the best quality of the transition process, the overcontrol value is less than 5%. A closed system remains stable with a new equilibrium position.

**Summary**

1. The block diagram has been developed and the body of mathematics has been proposed to describe the hydrodynamic processes of an adjustable pipeline system that allows to investigate transients, set different operating modes for pumping units, taking into account changes in flow conditions and heat transfer agent properties.

2. Time constants in the mathematical model take into account the design and technological parameters of the pumping facility. In this case, the inertial time constant significantly affects the transition process time and in a greater or lesser degree extent determines the hydraulic system stability to the development of oscillatory processes in the network.

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