District heating substation elements modeling for the development of the real-time model

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In the heating system of residential and commercial buildings, heating substation has an important role because it is used as a separator between primary and secondary supply sides. In this paper, we focused on the development of the simulation model of all heating substation elements, which influence the change of relevant parameters: water temperature, flow and pressure. The primary objective of the paper is to analyze and develop mathematical models of the heat exchanger, control valve, three-way valve and frequency-regulated centrifugal pump that are configurable and generic as much as possible, so they can be used for the development of the model that could operate in real time. A real-time model could be used as a suitable replacement for an actual physical system in the process of testing and improvement of the control system performance. Different elements setups of district heating substation are considered, and the model-based simulation is presented for one of them.

Key words*: district heating substation, model, control valve, heat exchanger*

1. Introduction

District heating substations are mainly used when the parameters of primary heat source are not allowed in the building heating system [1]. As a separator between primary and secondary water flow it is common to use a heat exchanger, allowing adjustments in the water temperature that is supplied to the building. This connection method is known as indirect connection method [2]. Aside from the heat exchanger, in district heating substation, there can be found: circulation pump and different types of valves (e.g. control valve, three-way valve, safety valve etc.).

There has been a number of researches on modeling and improvements of district heating substation. Many authors examined enhancement of the control methods, but proper modeling of the district heating substation is lacking on details due to many simplifications. Gustafsson et al. in [3] considered a novel control approach for radiator heating system with indirect connection based on the maximization of ΔT, but the model of the district heating substation [4] considered heat exchanger based on differential equations. In [5] Brand et al. studied the same mathematical model of the heat exchanger emphasizing its drawbacks. Wang et al. [6] considered mathematical model of plate heat exchanger and feedback control on a valve on primary side to ensure operational stability of the district heating substation.

In the recent past, traditional district heating systems are being replaced due to their uneconomical aspects. In low-temperature district heating systems – LTDH, supply temperature is reduced as much as possible [7] allowing more efficient use of energy resources. LTDH system was considered in [5], and a study in [8] analyses the challenges in the transition to LTDH. In [9] Tol et al. considered low-energy district heating networks with booster pumps and different substation types.

If we look at the district heating substation modeling problem from the perspective of automatic control, it is, first, necessary to identify all variables that could be useful for control purposes. The main task of district heating substation is to deliver the water of the appropriate temperature and flow to the building. However, both parameters are depending on the water pressure in the pipes. So, we identified water temperature, flow and pressure as relevant parameters for the modeling.

Concerning the temperature, to a large extent it depends on the type of heat exchanger (i.e. construction type and flow arrangement). In district heating substations, it is usual to install concentric tubes heat exchanger or plate heat exchanger [1]. Flow arrangements in these heat exchangers could be: 1. parallel flow - both fluid flows are oriented in the same direction; 2. counterflow – fluid flows are oriented in opposite directions, and 3. crossflow – fluid flows are mutually perpendicular [10]. Accordingly, heat exchangers have two inputs that are commonly referred to as primary and secondary sides. The primary side of heat exchanger is connected to the primary heat source, which is district heating system in most cases. For that matter, the water temperature one the primary side is mainly predetermined by the city regulations or weather conditions [1]. The primary side could, also, be connected to the heat pump, boiler or any other system that can produce hot water. The secondary side of the heat exchanger is connected to the building heating system, and, in general, all relevant parameters on the secondary side are changing more often.

The three most common methods for mathematical modeling of the heat exchangers are basic energy balance formulation [11], Log Mean Temperature Difference – LMTD and ε-NTU (i.e. effectiveness – Number of Transfer Units) methods. First order model in [12] based on energy balance introduced time delay to attenuate the transient behavior of a plate heat exchanger. LMTD method can be used for the design and performance purposes. Although it is analytically quite simple, values of all input and output temperatures must be known in order to use this method for more detailed analysis [10]. If, for example, the values of input temperatures are known, LMTD method requires the implementation of the iterative procedure and the output temperatures must be determined via the energy balance equations. Numerical model of a district heating substation presented in [13] considered variations of the flow rates, pressure losses in the district heating system and the overall heat transfer coefficient of the plate heat exchanger based on LMTD method. The same method was used by Oevelen et al. in [14] to evaluate the impact of optimized control curves on the district heating network return temperature. In [15] analytical models of concentric tubes heat exchanger based both on LMTD and ε-NTU method were compared with numerical model. In [16] ε-NTU method was used for analysis of economic aspects of heat exchangers based on linear equation systems.

As an additional element for temperature control, three-way valve can be mounted on the secondary side. Three-way valves are also known as mixing valves because they are used to mix two water flows of different temperatures. In that way, three-way valves are used if the water of different temperature set points needs to be delivered to various end users in the building (e.g. underfloor radiant heating 45/30 °C, fan-coil unit 55/35 °C, radiator heating 85/60 °C) [2]. On the inlets of the three-way valve are brought: 1. the water that is heated in the heat exchanger and 2. the return water from the building that was “consumed” by the building heating system. Depending on the temperature set point for the particular user, mixing valve will open to an appropriate value. According to the author’s best knowledge, in the attainable literature, three-way valve model was not considered as a part of a district heating substation.

In order to regulate the water flow in the district heating substation, the circulation pump is set up to maintain a constant flow rate depending on the pressure drop in the system, Fig. 1a. Circulation pumps are frequency-regulated centrifugal pumps. For a specific district heating substation, a circulation pump is chosen based on the system’s demand flow and the effort (i.e. head) that the pump needs to overcome in order to deliver water to each user.

Bajsič and Bobič in [17] introduced the control valve model as second order differential equation. In [6] a nonlinear equal-percentage control valve was considered. Depending on the position where they are mounted, control valves could be used for [18]:

1. Flow control – control valve is mounted on the secondary side and it is used as bypass for the circulation pump that is set to a constant speed, Fig. 1b.
2. Temperature control – control valve is mounted on the primary side. Regulation of the flow on the primary side is affecting water temperature on the secondary side, Fig. 1c.

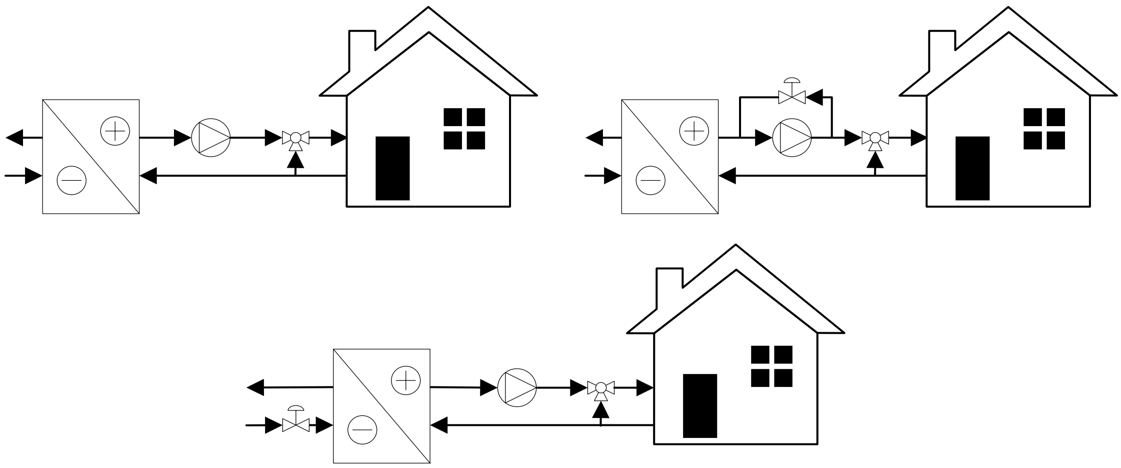

Figure 1. Different setups of district heating substation elements

This paper presents detailed analysis and precise modelling of all heating substation elements needed for development of the real-time model. The given models are comprehensive, generic and configurable, enabling transparent fine-tuning of the real-time model. In that way, real-time model of district heating substation can be used as a suitable replacement of the real physical system for testing and improvement of the control system performance, but also for diagnostics and fault detection providing more reliable control of the district heating substation parameters.

2. Mathematical formulation of district heating substation elements

The numerical simulation of the district heating substation elements presented in this paper is implemented in Simulink®. The elements that are considered are heat exchanger, frequency-regulated centrifugal pump, control valve and three-way valve.

2.1. Heat exchanger model

The heat exchanger is a device in which a heat exchange process occurs between two fluids of different temperature. In order to predict the thermal performance of the heat exchanger, it is necessary to create a relationship between total heat transfer and parameters such as input and output water temperatures, overall heat transfer coefficient U and total heat transfer area A. Based on the laws of thermodynamics, the energy balance can be expressed by the means of enthalpy, but, however, if a constant specific heat of a fluid can be assumed, the heat transfer rate Q is given as following [10]:

$$\dot{Q} = \dot{m}_h c_{p,h}\left(T_{h,i} - T_{h,o}\right) \tag{1}$$

$$\dot{Q} = \dot{m}_c c_{p,c}\left(T_{c,o} - T_{c,i}\right) \tag{2}$$

where i and o in the index denote input and output, retrospectively, h and c in the index denote hot and cold, also retrospectively, $\dot{m}$ is mass flow, c_p is specific heat (for water 4.1813 $Jg^{-1}K^{-1}$) and T is temperature. It can be noticed that eq. (1) and (2) are independent of the type of construction and flow arrangements of the heat exchanger. Therefore, the connection is made between the heat transfer rate and temperature difference $\Delta T = T_h - T_c$. Concerning that ΔT has different values in different parts of the heat exchanger, the mean value of temperature difference ΔT_m is used, so $Q = UA\Delta T_m$.

The algebraic formulation of ε-NTU method is computationally less demanding and simple of implementation. In ε-NTU method which is in detail given in [10], the effectiveness of the heat exchanger is defined by the maximum possible heat transfer rate Q_{max}:

$$\varepsilon = \frac{Q}{Q_{max}} \tag{3}$$

$$Q_{max} = C_{min}\left(T_{h,i} - T_{c,i}\right) \tag{4}$$

where $C_{min} = min\{C_c, C_h\}$; $C_c = \dot{m}_c c_{p,c}$ and $C_h = \dot{m}_h c_{p,h}$. Eq. (3) is more often used to express actual heat transfer rate:

$$Q = \varepsilon Q_{max} = \varepsilon C_{min}\left(T_{h,i} - T_{c,i}\right)$$

and effectiveness of the heat exchanger is given as $\varepsilon = f(NTU, C_r)$, where NTU is number of transfer units and C_r is heat capacity ratio:

$$NTU = \frac{UA}{C_{min}} \tag{5}$$

$$C_r = \frac{C_{min}}{C_{max}} \tag{6}$$

where $C_{max} = max\{C_c, C_h\}$. The effectiveness as such, depends on both the type of construction and flow arrangement of the heat exchanger. As an example, effectiveness for concentric tubes heat exchanger with parallel flow arrangement is:

$$\varepsilon = \frac{1-\exp(-NTU(1+C_r))}{1+C_r} \tag{7}$$

and for concentric tubes heat exchanger with counterflow arrangement is:

$$\varepsilon = \frac{1-\exp(-NTU(1-C_r))}{1-C_r\exp(-NTU(1-C_r))} \tag{8}$$

The output temperatures for hot and cold fluid can be determined from eq. (1) and (2) as:

$$T_{h,o} = T_{h,i} - \frac{Q}{C_h}$$

$$T_{c,o} = T_{T_{c,i}} + \frac{Q}{C_c}$$

2.2. Control valve

Mathematical modeling of control valves is complicated because of the design complexity and various hydraulic phenomena. In addition, the mathematical model of one control valve will differ if it is applied to different hydraulic systems. In other words, the control valve, depending on the hydraulic system, will give different flow rate Q for the same relative position of the valve h. For that reason, the valves are represented by its inherent and installed characteristics.

Inherent characteristic of the valve defines a change in the relative flow coefficient K_v/K_{vs} with respect to relative valve position h/h_{100}. The flow coefficient K_v is defined as the flow of the water through the valve expressed in m^3/h, at temperature 5-30 °C, for a pressure drop of 1 bar and the relative valve position in the range 0-1. The nominal flow coefficient K_{vs} is defined in the similar way, only it relates to the relative valve position $h = h_{100}$ and it is usually given by the manufacturer. Depending on the type of the valve, inherent characteristic can be [19]:

1. Linear – linear valve:

$$\frac{K_v}{K_{vs}} = \frac{K_{v0}}{K_{vs}} + \left(1 - \frac{K_{v0}}{K_{vs}}\right)\frac{h}{h_{100}}$$

2. Exponential – equal-percentage valve:

$$\frac{K_v}{K_{vs}} = \frac{K_{v0}}{K_{vs}} \exp\left(\frac{h}{h_{100}} \ln\left(\frac{K_{vs}}{K_{v0}}\right)\right)$$

3. Square – fast-opening valve:

$$\frac{K_v}{K_{vs}} = 1 + \frac{K_{v0}}{K_{vs}} - \left(1 - \frac{h}{h_{100}}\right)^2$$

where K_{v0} is the value of value of K_v when $h = 0$. Inherent characteristic of the valve is valid only when the pressure drop on the valve is constant, and that will happen only when the relationship between the flow and valve position depends solely on the valve geometry. In order to keep constant pressure drop on the valve, there should be no pressure drop in the pipes, which applies only if the length of pipes is minimal. To determine the installed characteristic of the valve, the following expression is used:

$$Q/Q_{100} = \frac{1}{\sqrt{1+\psi(1/(K_v/K_{vs})^2-1)}} \tag{9}$$

where $\psi = \Delta p_{V100}/\Delta \mathrm{p}_{hs}$ is the ratio between the maximum pressure drop on the control valve and the maximum dynamic pressure drop in the system.

When selecting the control valve, it is common to choose between linear and equal-percentage valve. The equal-percentage valves are used for the processes of heat exchange or for the transfer of a fluid through long pipes using centrifugal pump (i.e. due to this, the installed characteristic of the valve will be approximately linear). Linear valves are useful when the overall static characteristic of the process is approximately constant. Fast-opening valves are commonly used for on-off operations [20].

2.2.1 Three-way valve

In the heating or cooling systems, three-way valves are frequently used. Three-way valve, whose cross section is shown in Fig. 2, has two inlets A and B and one outlet AB. On the inlet A comes the water at temperature T_A under pressure p_A, and on the inlet B comes the water at temperature T_B under pressure p_B.

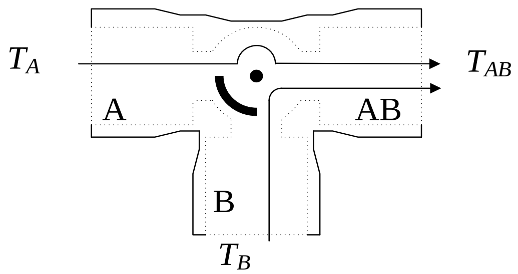

Figure 2. Cross section of three-way valve

The total pressure drop on three-way valve is, in general, constant. Because of this, three-way valves are modeled as the control valves with linear or approximately linear installed characteristic. If the water pressure at inlets A and B are different, water flows at the inlets are determined as $q_A = \beta q_{100}$ and $q_B = q_{100} - q_A$, and β is:

$$\beta = \frac{q}{\sqrt{\alpha+(1-\alpha)\cdot q^2}} \tag{10}$$

where $\alpha = p_A/p_B$. The temperature of the water on the outlet of the three-way valve can be determined from Richmann's calorimetry mixing formula:

$$T = \frac{Q_A T_A + Q_B T_B}{Q_{100}}$$

2.3. Centrifugal pump

The capacity of the centrifugal pump is commonly given by the pump characteristic that describes the dependency of the pump head H on the volumetric flow rate Q [18]. The pump head H can be determined if the total differential pressure across the pump Δp_{tot} is known:

$$H = \frac{\Delta p_{tot}}{\rho g} \tag{11}$$

where ρ is fluid density and g is the gravitational acceleration. Considering the Bernoulli principle, total differential pressure is given as the sum of static, dynamic and geodetic differential pressures:

$$\Delta p_{tot} = \Delta p_s + \Delta p_d + \Delta p_g \tag{12}$$

Static differential pressure Δp_s is usually measured directly, but it is given as the difference of static pressure at the inlet and outlet of the pump. Dynamic differential pressure Δp_d is given as:

$$\Delta p_d = \frac{1}{2}\rho v_o^2 - \frac{1}{2}\rho v_i^2 = \frac{1}{2}\rho\left(\frac{4Q}{\pi}\right)^2\left(\frac{1}{d_o^2} - \frac{1}{d_i^2}\right) \tag{13}$$

where v is flow velocity, and d is pipe diameter. Geodetic differential pressure can be determined as $\Delta p_g = \Delta z \rho g$, where Δz is the difference on the vertical positions of the inlet and outlet of the pump.

Once the appropriate pump has been chosen, control methods must be considered. Some of the most common flow control methods [18] are:

1. Throttle regulation – supplementary hydraulic resistance is introduced in the pipeline after pump (i.e. valve). This type of control is suitable for the pumps with a high head compared to the flow.

2. Bypass regulation – a control valve is placed parallel to the pump, allowing part of the flow to go back to the pump inlet. This type of control is suitable for the pumps with a low head compared to the flow.

3. Speed control – given as affinity laws:

$$\frac{Q_1}{Q_0} = \frac{n_1}{n_0} \qquad \frac{H_1}{H_0} = \left(\frac{n_1}{n_0}\right)^2 \qquad \frac{P_1}{P_0} = \left(\frac{n_1}{n_0}\right)^3 \tag{14}$$

where the rotating speed n of the motor is a function of frequency f, slip s and the number of poles p [21]:

$$n = 120\ f(1-s)/p \tag{16}$$

3. Model-based simulation of the district heating substation

The model of district heating substation is implemented in Simulink® and it is presented on Fig. 3. The flow control valve is connected to the primary side of the heat exchanger and on the secondary side there are frequency-regulated centrifugal pump and three-way valve, as shown on Figure 1c.

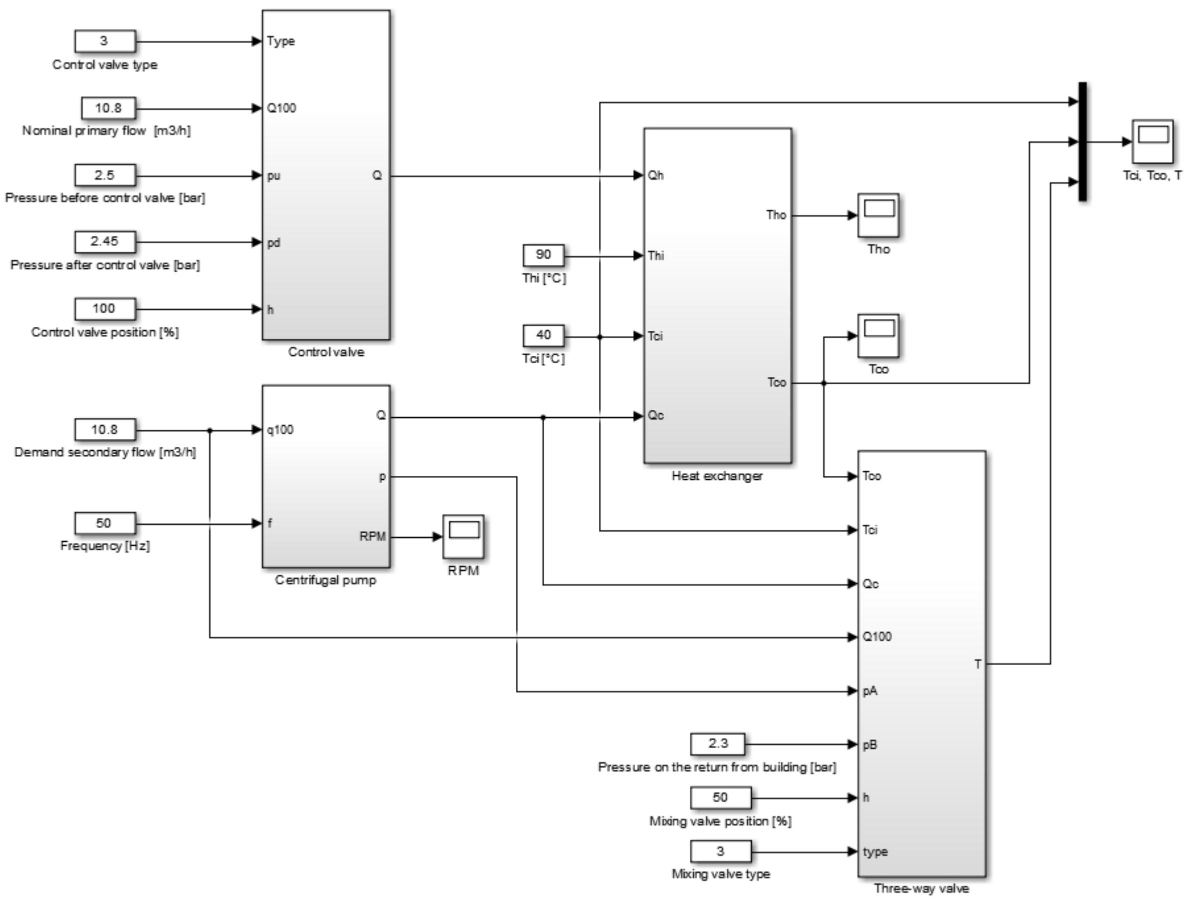

Figure 3. Simulink® model of district heating substation

Presented simulation setup considers counterflow concentric tubes heat exchanger with two configurable parameters: overall heat transfer coefficient U and heat transfer area A. The overall heat transfer coefficient for water-to-water heat exchangers is given in the range 850-1700 W/(m^{2}K) [10] and the heat transfer area of the concentric tubes heat exchanger can be obtained from $A = r^2\,\pi/4$, where r is the diameter of the inner tube. The overall heat transfer coefficient is one of the most uncertain parameters since it is subjective to the fluid impurities, rust and other factors due to which the thermal resistance of the heat exchanger changes.

For the simulation purposes, nominal volumetric flow rates on both primary and secondary side of the district heating substation are set to the same value of 10.8 m^3/h. Pressure drops on control valve and three-way valve are selected so that relative valve position of 50% gives 50% of the maximal flow rate. K_{vs} value for the valves can be determined form as $K_{vs} = q_{100}(\rho/(1000\Delta p_V))^{1/2}$[20] [4], [20]. As an example, for the nominal primary flow rate of 10.8 m^3/h, and pressure drop on the valve of 0.05 bar, K_{vs} is 48.23. Considering this value and the equal-percentage valve, control flow valve characteristic is presented on Fig. 4.

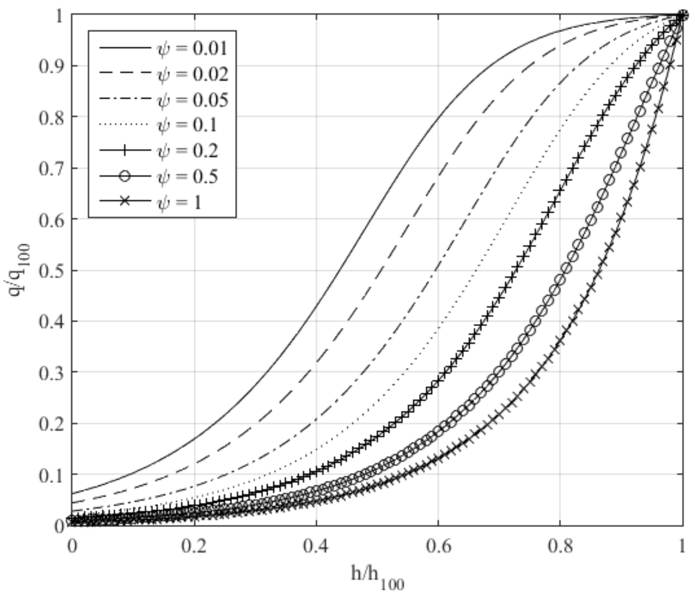

Figure 4. Installed characteristic for equal-percentage valve

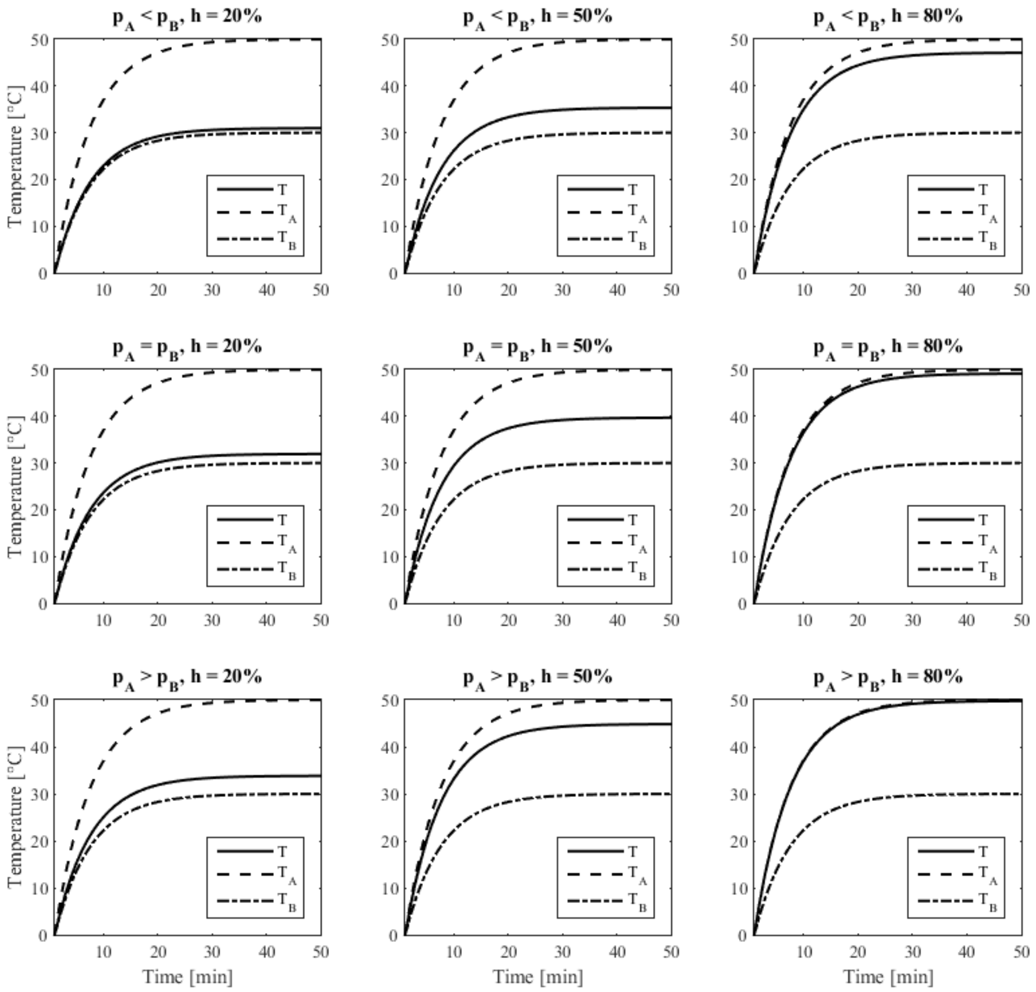

Figure 5. Three-way valve model

The parameters relevant for control are flows on the primary and secondary side of the district heating substation, and the relative position of the three-way valve. In the model-based simulation, all other variables were set to constant values, such as $T_{hi} = 90$ °C, $T_{ci} = 30$ °C, $\Delta p_{tot} = 2.5$ bar etc. The model for three-way valve is implemented following the eq. (9) and (10) in Section 2.2 and 2.2.1. The resulting water temperature T is subordinate to the: water temperatures ($T_A = T_{co}$ and $T_B = T_{ci}$) and pressures on both inlets (i.e. p_A is the pressure after centrifugal pump, and p_B is the pressure in the return pipeline from the building), together with the relative valve position h. As it can be seen on Fig. 5, the temperature T will have the mean value of T_{co} and T_{ci} if the relative valve position is 50% and pressures on both inlets are equal.

3.1. Simulation examples and discussion

For the control valve that is installed on the primary side of the district heating substation, different values of the relative valve position were considered: 20%, 100% and 50%. As the volumetric flow rate on the primary side of the heat exchanger is changed, the amount of heat transferred on the secondary side changes and resulting output temperature T_{co} varies, Fig. 6. Three-way valve position is set to 50% with equal pressures on its inlets.

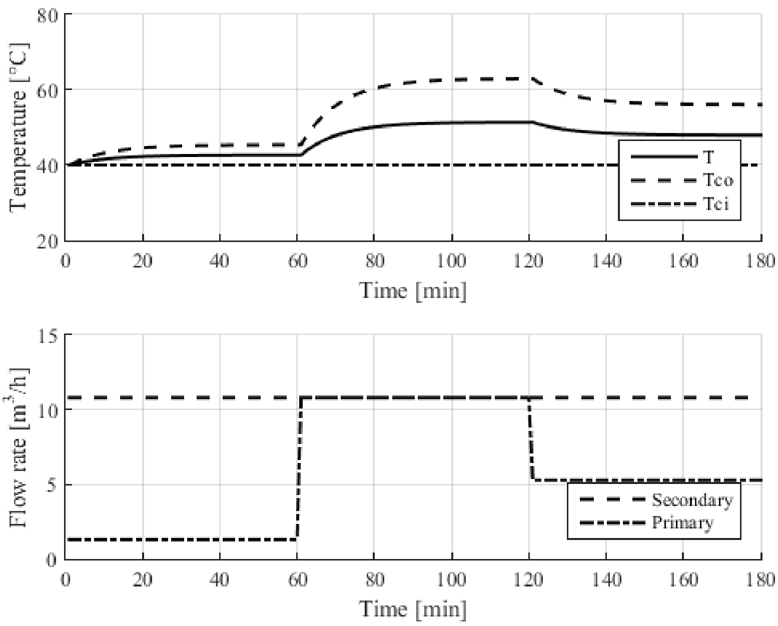

Figure 6. Effect on the model output due to the changes in the primary flow

Similarly, in the second example of model-based simulation, Fig. 7, for the frequency-regulated centrifugal pump, the operating frequency is 40, 50 and 45 Hz. As the volumetric flow rate on the secondary side reaches higher values, the output temperature T_{co} is lower. Three-way valve position is set to 50% with equal pressures on its inlets.

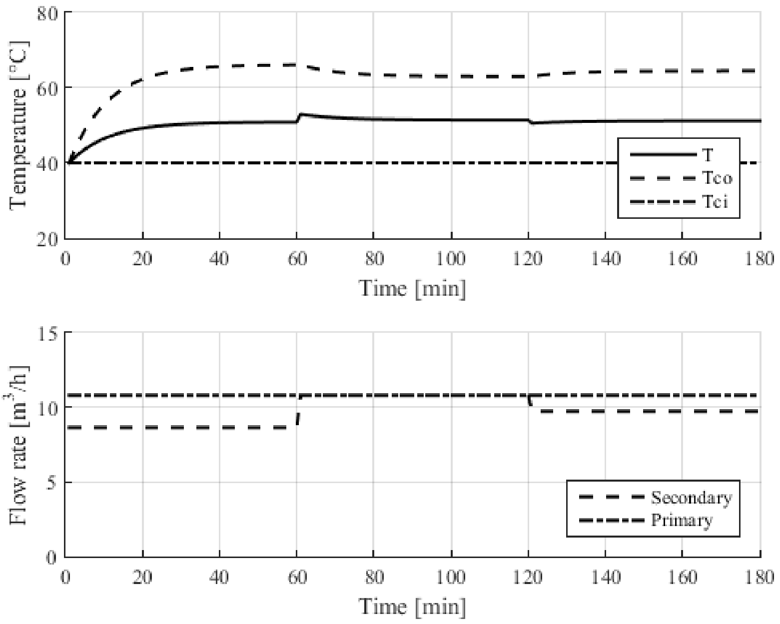

Figure 7. Effect on the model output due to changes in the secondary flow

The third simulation example shows maximal volumetric flows on both sides of the heat exchanger, but the relative position of the three-way valve changes from 20% to 100% and 50%, Fig. 8.

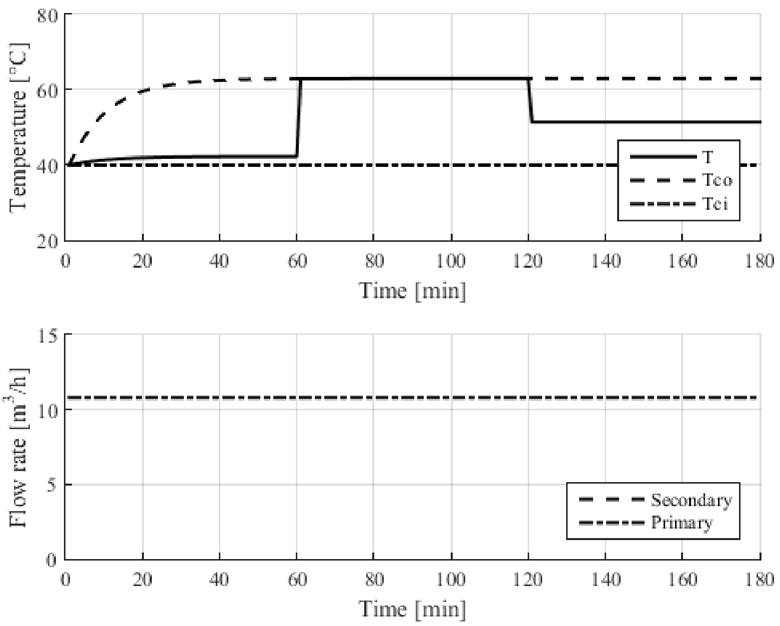

Figure 8. Effect on the model output due to relative position of the three-way valve

4. Conclusion

Detailed mathematical models of district heating substation elements are presented in this paper. The mathematical models of heat exchanger, control valve, three-way valve and frequency-regulated pump that were analyzed provide a general view of the system which can be easily configured and validated for a real physical system, if the data from a real system is available.

The interconnection of such detailed modeled elements of the district heating substation that is corresponding to the real physical system has been performed through the model-based simulation in Simulink® tool. The effect of all relevant control parameters on the water temperature that is delivered to the building heating system (i.e. model output) was observed.

The analyzed models are configurable, and after validation with the data from an actual physical system, they will suitable for the development of the real-time model of district heating substation. In the future work, a real-time model will be used for testing, improvement and verification of control and optimization methods giving the possibility of more efficient performance of the control system.

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