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Control of a Plate Heat Exchanger Using the Terminal Sliding Mode Technique

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ABSTRACT: Heat exchangers are devices used to transfer heat from one fluid to another. In this paper we investigate the application of three different terminal sliding mode (TSM) controllers to a plate heat exchanger. The main objective of the designed controllers is to adjust the outlet cold water temperature to a desired set point temperature. First, a fast TSM controller is designed. To reduce chattering in the control signal, a second-order fast TSM controller is designed. In addition, a dynamic fast TSM controller is designed to further reduce the chattering. The performances of the closed loop system are simulated using MATLAB. The simulation results clearly show that the proposed three control schemes work very well. Moreover, the proposed control schemes are implemented using an experimental setup. The implementation results indicate that the proposed controllers work very well.

1. INTRODUCTION

The exchange of heat between two fluids that are at different temperatures and which are separated by a solid wall occurs in many engineering applications.[1] The device used to implement this exchange is called a "heat exchanger". Heat exchangers are typically classified according to their flow arrangement (countercurrent or cocurrent) and to their type of construction (plate, tubular, or shell). In this paper, we will consider the temperature control problem of a countercurrent flow plate heat exchanger. We will propose three different terminal sliding mode controllers. The main objective of these controllers is to adjust the outlet cold water temperature to a desired (warm) temperature.

Several researchers investigated the problem of controlling liquid temperature in a heat exchanger plant. In ref 2 a model predictive control (MPC) applied to a heat exchanger network is proposed; a nonlinear model of a shell-and-tube heat exchanger is used. The control structure of the heat exchanger includes two levels. The upper level controller is devoted to determining an optimal steady state such that a minimum service cost is achieved; the lower level controller takes care of temperature targets and drives the system toward the optimal steady state while focusing on the dynamic performance of the system. A comparison between MPC and conventional proportional-integral-derivative (PID) control laws applied to an industrial heat exchanger is given in ref 3. A min−max MPC scheme is applied to a heat exchanger in ref 4. Nevertheless, the design methodology in ref 4 generally gives rise to control algorithms that suffer a large computational burden due to the numerical min−max problem which has to be solved at every sampling time. The use of hinging hyperplanes reduced this disadvantage, but it complicates the controller design procedure. Other model predictive controllers applied to heat exchangers can be found in refs 5−7. Controlling the internal fluid temperature at the outlet of a parallel-flow heat exchanger by manipulating the inlet external fluid temperature is proposed in ref 8. A robust controller for temperature control in a heat exchanger with uncertainty estimation is proposed in ref 9. An approach which guarantees the boundedness of the input flow rate without entailing a complex control algorithm is proposed in ref 10. The outlet temperature is regulated through a controller which does not need feedback of the whole state vector and does not depend on the exact value of the process parameters. Furthermore, conventional P, PI, and PID, and a multiloop controller tuned using game theory were proposed in refs 11 and 12, respectively. However, these controllers lack formal stability proofs. In all of these methods, direct manipulation of the flow rate of the hot water or cold water is commonly used. If both flow rates are set by the process requirements, then heat exchanger bypassing is widely used.[13]

Recently, artificial intelligent control techniques such as fuzzy logic and neural networks have been extensively used in the modeling and control of heat exchanger systems.[14−22] In ref 14, an optimal PI fuzzy controller is used to control the heat exchanger system. The design procedure consists of optimizing the scaling factors of the linear fuzzy controller. This is done by solving an unconstrained optimization problem obtained from the simplification of a constrained optimization where the objective function is the integral of the measured error, and the constraints are the relationships between fuzzy and conventional PID gains. An adaptive fuzzy sliding mode controller for the heat exchanger is proposed in ref 15. A predictive control law based on a nonlinear fuzzy model is developed in ref 16. On the other hand, some controllers based on neural network models can be found in refs 18−22.

The paper is organized as follows. The dynamic model of the plate heat exchanger is presented in section 2. A fast terminal sliding mode (TSM) control scheme, a second-order fast TSM, and a dynamic fast TSM controllers are developed in section 3. The simulation results are given in section 4. The experimental results are presented in section 5 to show the effectiveness and

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the applicability of the proposed control schemes. Finally, the conclusion is given in section 6.

Sometimes, the arguments of a function will be omitted in the analysis when no confusion may arise.

2. DYNAMIC MODEL OF THE PLATE HEAT EXCHANGER

The schematic diagram of the plate heat exchanger is shown in Figure 1. It consists of a stack of parallel thin plates that are stacked between heavy end plates. The cold water and hot water pass alternately between adjoining plates in the stack; heat is exchanged through these plates. The objective of the control problem is to adjust the outlet temperature of the cold water, $T_{co}(t)$, to maintain this temperature as close as possible to the desired set point temperature.

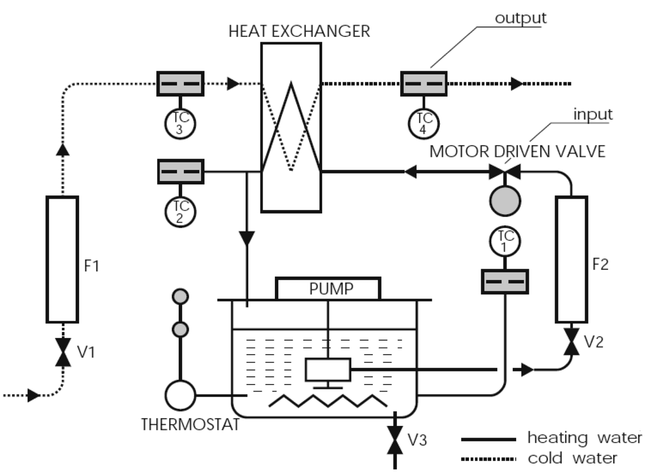

Figure 1. Schematic diagram of the plate heat exchanger.[23]

The parameters of the system are as follows. The rate U_c is the fixed volumetric flow rate of the cold water; V_c and V_h are the volumes of the cold water and the hot water, respectively. The temperatures T_{ci} and T_{hi} are the fixed inlet temperatures of the cold water and the hot water, respectively. The parameter U is the constant heat transfer coefficient; the area A is the summation of the areas of all the plates. The coefficients $C_{p,c}$ and $C_{p,h}$ are the specific heat coefficients for the cold water and the hot water, respectively. The coefficients ρ_c and ρ_h are the water densities of the cold water and the hot water, respectively.

The state variables of the system are the outlet cold and hot temperatures $T_{co}(t)$ and $T_{ho}(t)$, respectively. The input to the system is $u(t)$, which is the volumetric flow rate of the hot water.

The plate heat exchanger dynamic model can be written as follows:[24]

$$\dot{T}_{co}(t) = -k_1(T_{co}(t) - T_{ho}(t)) + \frac{U_c}{V_c}(T_{ci} - T_{co}(t)) \tag{1}$$

$$\dot{T}_{ho}(t) = -k_2(T_{ho}(t) - T_{co}(t)) + \frac{1}{V_h}(T_{hi} - T_{ho}(t))u(t) \tag{2}$$

where $k_1 = UA/(C_{p,c}\rho_c V_c)$ and $k_2 = UA/(C_{p,h}\rho_h V_h)$.

In the following, we will transform the above nonlinear model into a model that facilitates the use of different control design techniques. First, define the state variables $z_1(t)$ and $z_2(t)$ such that

$$z_1(t) = T_{co}(t)$$

$$z_2(t) = T_{ho}(t) - T_{co}(t)$$

Note that z_1 and z_2 are positive variables.

The output of the system is taken to be the outlet temperature of the cold water. Hence, the above dynamic model can be written as

$$\begin{aligned} \dot{z}_1(t) &= k_1 z_2(t) + \frac{U_c}{V_c}(T_{ci} - z_1(t)) \\ \dot{z}_2(t) &= -k_2 z_2(t) + \frac{1}{V_h}(T_{hi} - z_1(t) - z_2(t))u(t) \\ y(t) &= z_1(t) \end{aligned} \tag{3}$$

The objective of the control scheme is to regulate the output $y(t) = z_1(t) = T_{co}(t)$ to a desired constant temperature T_{cr}. Using the first equation of the above system, it is easy to show that if $y(t)$ is regulated to the desired value T_{cr}, then $z_2(t) = T_{ho}(t) - T_{co}(t)$ will be regulated to the value $(U_c/(k_1V_c))(T_{cr} - T_{ci})$.

Clearly, the dynamic model of the plate heat exchanger is nonlinear. Therefore, we will introduce a transformation so the dynamic model of the system can be written in a form that facilitates the control design.

Let

$$\mathbf{x}(t) = \begin{bmatrix} x_1(t) \\ x_2(t) \end{bmatrix} \quad \text{and} \quad \mathbf{z}(t) = \begin{bmatrix} z_1(t) \\ z_2(t) \end{bmatrix}$$

Define the transformation $\mathbf{x}(t) = \mathbb{T}(\mathbf{z}(t))$ such that

$$\begin{aligned} x_1(t) &= z_1(t) - T_{cr} \\ x_2(t) &= k_1 z_2(t) + \frac{U_c}{V_c}(T_{ci} - z_1(t)) \end{aligned} \tag{4}$$

Using the transformation defined by eq 4 leads to a system model which is in the normal form. It should be noted that the inverse transformation $\mathbf{z}(t) = \mathbb{T}^{-1}(\mathbf{x}(t))$ is such that

$$\begin{aligned} z_1(t) &= x_1(t) + T_{cr} \\ z_2(t) &= \frac{1}{k_1}\left[x_2(t) + \frac{U_c}{V_c}(x_1(t) + T_{cr} - T_{ci})\right] \end{aligned} \tag{5}$$

Hence, the dynamic model of the plate heat exchanger system can be written in the following compact form:

$$\begin{aligned} \dot{x}_1(t) &= x_2(t) \\ \dot{x}_2(t) &= f(\mathbf{x}) + g(\mathbf{x})\, u(t) \\ y(t) &= x_1(t) + T_{cr} \end{aligned} \tag{6}$$

where

$$f(\mathbf{x}) = f_0 - f_1 x_1(t) - f_2 x_2(t)$$

$$g(\mathbf{x}) = g_0 - g_1 x_1(t) - g_2 x_2(t)$$

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and the constants f_0, f_1, f_2, g_0, g_1, and g_2 are such that

$$f_0 = \frac{k_2 U_c}{V_c}(T_{ci} - T_{cr})$$

$$f_1 = \frac{k_2 U_c}{V_c}$$

$$f_2 = k_2 + \frac{U_c}{V_c}$$

$$g_0 = \frac{k_1}{V_h}\left[T_{hi} - \left(1 + \frac{U_c}{k_1 V_c}\right)T_{cr} + \frac{U_c}{k_1 V_c}T_{ci}\right]$$

$$g_1 = \frac{k_1}{V_h}\left[1 + \frac{U_c}{k_1 V_c}\right]$$

$$g_2 = \frac{1}{V_h}$$

The dynamic model in eq 6 will be used to design control schemes for the plate heat exchanger.

3. DESIGN OF TERMINAL SLIDING MODE CONTROLLERS

3.1. Motivation for Using Nonlinear Controllers for the Plate Heat Exchanger. To motivate the use of nonlinear controllers, we will first attempt to control the system using a simple linear state feedback controller of the form

$$u = -K_1 x_1 - K_2 x_2 + r_0$$

where K_1 and K_2 are constant design parameters and $r_0 = -f_0/g_0$ is the reference input. It follows that the closed loop system can be written as

$$\dot{x}_1 = x_2$$

$$\dot{x}_2 = -(f_1 + g_0 K_1 + g_1 r_0)x_1 - (f_2 + g_0 K_2 + g_2 r_0)x_2 + g_1 K_1 {x_1}^2 + (g_2 K_1 + g_1 K_2)x_1 x_2 + g_2 K_2 {x_2}^2$$

Thus, we end up with a nonlinear closed loop system. It is not clear whether the closed loop system is stable or not.

On the other hand, we will show next that the plate heart exchanger system can be linearized through the use of a nonlinear feedback linearization controller. Recall eq 6:

$$\dot{x}_1(t) = x_2(t)$$

$$\dot{x}_2(t) = f_0 - f_1 x_1(t) - f_2 x_2(t) + g(\mathbf{x})\, u(t) \tag{7}$$

We want to drive the states (x_1, x_2) to (0,0) asymptotically. Let α_1 and α_2 be constant design parameters; also let the controller u be such that

$$u = \frac{1}{g(\mathbf{x})}(-\alpha_1 x_1 - \alpha_2 x_2 - f_0) \tag{8}$$

Obviously, to use the controller in eq 8 we need to ensure that $g(\mathbf{x}) \neq 0$ for all x_1 and x_2. Applying the controller given by eq 8 to the system given in eq 7, we obtain the following linear closed loop system:

$$\dot{x}_1(t) = x_2(t)$$

$$\dot{x}_2(t) = -(f_1 + \alpha_1)x_1(t) - (f_2 + \alpha_2)x_2(t)$$

The closed loop system can be written as

$$\dot{\mathbf{x}} = \mathbf{A}\mathbf{x}$$

where

$$\mathbf{A} = \begin{bmatrix} 0 & 1 \\ -(f_1 + \alpha_1) & -(f_2 + \alpha_2) \end{bmatrix}$$

The characteristic polynomial of the closed loop system matrix **A** is such that

$$\det(\lambda \mathbf{I} - \mathbf{A}) = \lambda^2 + (f_2 + \alpha_2)\lambda + (f_1 + \alpha_1)$$

Therefore, as long as $f_1 + \alpha_1 > 0$ and $f_2 + \alpha_2 > 0$, the closed loop system is asymptotically stable. Different choices of α_1 and α_2 will influence the performance of the closed loop system.

Hence, the above linear and nonlinear control designs indicate that in some cases proving the stability of the closed loop system is simpler when a nonlinear controller is used.

In this paper we propose to use a sliding mode control technique to control the plate heat exchanger system. The choice of this technique is due to the attractive properties (fast response and robustness) of this control technique.

3.2. Overview of Sliding Mode Control. The sliding mode control (SMC) technique is a well-known control method which is used to control different types of systems. The works of DeCarlo et al.,[25] Hung et al.,[26] Edwards and Spurgeon,[27] and Utkin et al.[28] present very good reviews of this technique; also they present several applications of SMC. The dynamics of a system which is controlled using an SMC controller goes through two phases (or two modes). The first phase represents the dynamics of the controlled system before reaching the sliding surface ($\sigma(\mathbf{x}) = 0$); this mode is called the "reaching mode". The second phase represents the dynamics of the controlled system after reaching the sliding surface ($\sigma(\mathbf{x}) = 0$); this mode is called the "sliding mode". On the sliding surface (the sliding mode), the dynamics of the controlled system are of a lower order than the dynamics of the original model of the system. Therefore, the design of an SMC controller involves the design of a sliding surface and the design of a sliding controller. The designer needs to ensure that the sliding surface is reached and that the trajectories slide to the desired equilibrium point.

Terminal sliding mode (TSM) control is a class of sliding mode control which was first introduced by Venkataraman and Gulati.[29] The difference between the standard sliding mode control technique and the TSM technique is that the latter guarantees the convergence of the trajectories of the controlled system in finite time. Venkataraman and Gulati proposed a sliding surface of the form

$$\sigma_{vg}(\mathbf{x}) = x_2 + \beta {x_1}^{q_1/p_1}$$

They showed that the sliding surface $\sigma_{vg}(\mathbf{x})$ is reachable in finite time. Later on, other researchers[30] proposed the use of a sliding surface of the form

$$\sigma_{vg}(\mathbf{x}) = x_2 + \alpha x_1 + \beta {x_1}^{q_1/p_1}$$

The addition of the term αx_1, with $\alpha > 0$, to the sliding surface $\sigma(\mathbf{x})$ guarantees faster finite-time stability. Hence, the term "fast TSM" is used.

Moreover, during the past few years, higher-order sliding mode control schemes have received significant interest and have become well-established with good potential for practical applications. These techniques have been applied in many control fields which include robot control,[31−34] motor control,[35−38] flight control,[39] renewable energy control,[40,41] and process control.[42−44]

Sections 3.3, 3.4, and 3.5 present the use of terminal sliding mode controllers for the plate heat exchanger system.

3.3. Design of a Fast TSM Controller. In this subsection, a fast terminal sliding mode control scheme is designed. Let α, β, and W be positive scalars, and let p_1 and q_1 be positive odd integers such that $p_1 > q_1$. Also, define the signum function such that

$$\mathrm{sgn}(\sigma) = \begin{cases} +1 & \text{if } \sigma > 0 \\ 0 & \text{if } \sigma = 0 \\ -1 & \text{if } \sigma < 0 \end{cases}$$

Also, let the sliding surface σ be such that

$$\sigma = x_2 + \alpha x_1 + \beta x_1^{q_1/p_1} \tag{9}$$

Proposition 1. The fast terminal sliding mode control law

$$u = \frac{1}{g(\mathbf{x})}\left\{-f_0 + f_1 x_1 + f_2 x_2 - \alpha x_2 - \beta\frac{q_1}{p_1}x_1^{(q_1-p_1)/p_1}x_2 - W\,\mathrm{sgn}(\sigma)\right\} \tag{10}$$

when applied to the plate heat exchanger system given by eq 6 regulates the states x_1 and x_2 to their desired values in finite time.

It can be shown that the application of the above controller to the system given by eq 6 leads to

$$\dot{\sigma} = -W\,\mathrm{sgn}(\sigma) \tag{11}$$

Equation 11 guarantees that the sliding surface $\sigma = 0$ is reached in finite time.

Using the first equation in eq 6 and $\sigma = 0$, it can be concluded that the states x_1 and x_2 converge to 0 in finite time. Consequently, using the transformation given by eq 4, it can be concluded that z_1 and z_2 converge to their desired values in finite time. Therefore, the controller given by eq 10 guarantees the convergence of the temperature of the cold water $z_1(t) = T_{co}(t)$ to its desired value T_{cr} in finite time.

The control scheme proposed in this subsection suffers from the well-known chattering problem. The chattering is due to the assumption that the control can be switched from one value to another at any moment of time and with almost zero time delay. One can reduce the chattering by using a boundary layer.[45] However, we decided to use a higher-order fast terminal sliding mode controller to reduce the chattering. In addition, a dynamic fast terminal sliding mode controller is also proposed to deal with the chattering problem.

3.4. Design of a Second-Order Fast TSM Controller. In this subsection, a second-order fast terminal sliding mode control scheme is designed. We will use the same sliding surface as the one given by eq 9. Let Υ be a positive scalar.

Proposition 2. The second-order fast terminal sliding mode control law

$$u = \frac{1}{g(\mathbf{x})}\left\{-f_0 + f_1 x_1 + f_2 x_2 - \alpha x_2 - \beta\frac{q_1}{p_1}x_1^{(q_1-p_1)/p_1}x_2 - \Upsilon\frac{\sigma}{|\sigma|^{0.5}}\right\} \tag{12}$$

when applied to the plate heat exchanger system given by eq 6 regulates the states x_1 and x_2 to their desired values in finite time.

It can be shown that the application of the controller given by eq 12 to the system given by eq 6 yields

$$\dot{\sigma} = -\Upsilon\frac{\sigma}{|\sigma|^{0.5}} \tag{13}$$

Equation 13 guarantees that the sliding surface $\sigma = 0$ is reached in finite time.

Using the first equation in eq 6 and $\sigma = 0$, it can be concluded that the system states x_1 and x_2 converge to 0 in finite time. Using the transformation given by eq 4, it can be concluded that z_1 and z_2 converge to their desired values in finite time. Therefore, the controller scheme given by eq 12 guarantees the convergence of the temperature of the cold water $z_1(t) = T_{co}(t)$ to its desired value T_{cr} in finite time.

Remark 1. The controller given by eq 12 is classified as a second-order sliding mode controller since the condition $\sigma = \dot{\sigma} = 0$ is satisfied on the sliding surface.[46] Also, note that the dynamics given by eq 13 can be written as

$$\dot{\sigma} = -\Upsilon\,|\sigma|^{0.5}\,\mathrm{sgn}(\sigma)$$

3.5. Design of a Dynamic Fast TSM Controller. In this subsection, a dynamic fast terminal sliding mode control scheme is designed. We will use the same sliding surface as the one given by eq 9. Let Γ be a positive scalar, and let p_2 and q_2 be positive odd integers such that $p_2 > q_2$.

Proposition 3. The dynamic fast terminal sliding mode control law

$$u = \frac{1}{g(\mathbf{x})}\left\{-f_0 + f_1 x_1 + f_2 x_2 - \alpha x_2 - \beta\frac{q_1}{p_1}x_1^{(q_1-p_1)/p_1}x_2 - \Gamma\sigma^{q_2/p_2}\right\} \tag{14}$$

when applied to the plate heat exchanger system given by eq 6 regulates the states x_1 and x_2 to their desired values in finite time.

Proof. Taking the time derivative of σ in eq 9 and using eq 6, one obtains

$$\begin{aligned}\dot{\sigma} &= \dot{x}_2 + \alpha\dot{x}_1 + \beta\frac{q_1}{p_1}x_1^{(q_1-p_1)/p_1}\dot{x}_1 \\ &= f(\mathbf{x}) + g(\mathbf{x})\,u(t) + \alpha x_2 + \beta\frac{q_1}{p_1}x_1^{(q_1-p_1)/p_1}x_2\end{aligned} \tag{15}$$

Using the controller given by eq 14 into the above equation, its follows that

$$\dot{\sigma} = -\Gamma\sigma^{q_2/p_2} \tag{16}$$

The trajectories associated with the unforced discontinuous dynamics in eq 16 exhibit a finite time reachability to 0 from any given initial condition provided that the constant Γ is positive. The system dynamics on the sliding surface $\sigma = 0$ are determined by the following nonlinear differential equation:

$$\dot{x}_1 = -\alpha x_1 - \beta x_1^{q_1/p_1} \tag{17}$$

Hence, the system states x_1 and x_2 converge to 0 in finite time. Using the transformation given in eq 4, it can be concluded that z_1 and z_2 converge to their desired values in finite time. Therefore, the controller scheme in eq 14 guarantees the convergence of the temperature of the cold water $z_1(t) = T_{co}(t)$ to its desired value T_{cr} in finite time.

Remark 2. The controller given by eq 14 is a second-order sliding mode controller since on the sliding surface $\sigma = \dot{\sigma} = 0$.[46] Higher-order sliding mode controllers are sometimes known as dynamic sliding mode controllers.[47] Therefore, we labeled the proposed controller in eq 14 as "dynamic fast" since we have already used the "second-order fast" label for the controller given by eq 12.

4. SIMULATION RESULTS

The control schemes designed in section 3 are simulated using the MATLAB/SIMULINK software.[48] The parameters of the system are taken to be the parameters of the plate heat exchanger apparatus which is used to implement the proposed control schemes. The values of the plate heat exchanger parameters are listed in Table 1.

Table 1. Values of the Parameters of the Plate Heat Exchanger

parameter	value	parameter	value
U	300 W/m^2·°C	$C_{p,h}$	4180 J/kg·°C
A	0.0672 m^2	$C_{p,c}$	4180 J/kg·°C
V_h	5.37×10^{-4} m^3	ρ_c	1000 kg/m^3
V_c	5.37×10^{-4} m^3	ρ_h	1000 kg/m^3

The constant heat transfer coefficient, U, was experimentally determined using steady-state measurements. To simulate realistic cases, the simulations are carried out using the following input constraint:

$$0\ \frac{\text{cm}^3}{\text{min}} \le u \le 280\ \frac{\text{cm}^3}{\text{min}} \tag{18}$$

In all the simulations, the parameters of the controllers are taken such that $\alpha = 0.5$, $\beta = 0.1$, $p_1 = p_2 = 9$, $q_1 = q_2 = 7$, and $W = \Upsilon = \Gamma = 0.1$. The volumetric flow rate of the cold water, U_c, is fixed, and it is equal to 150 cm^3/min. The fixed inlet temperatures of the cold water and the hot water are $T_{ci} = 20$ °C and $T_{hi} = 80$ °C, respectively. The desired temperature for the cold water is $T_{cr} = 40$ °C.

4.1. Results of the Fast TSM Controllers. The plate heat exchanger system described by eq 3 is simulated using the proposed control schemes given by eqs 10, 12, and 14. Figure 2 presents the simulation results when the fast TSM controller given by eq 10 is applied. Figure 2a depicts the cold water temperature and the hot water temperature versus time. Clearly, the output of the system (the cold water temperature) converges to its desired value in about 300 s. The hot water temperature converges to a constant value in about 330 s. Figure 2b shows the controller versus time. The controller displays a lot of chattering.

Figure 3 presents the simulation results when the second-order fast TSM controller given by eq 12 is used. Figure 3a depicts the cold and hot water temperatures versus time. The output of the system converges to its desired value in about 260 s. The hot water temperature converges to a steady-state value in about 300 s. Figure 3b shows the controller versus time.

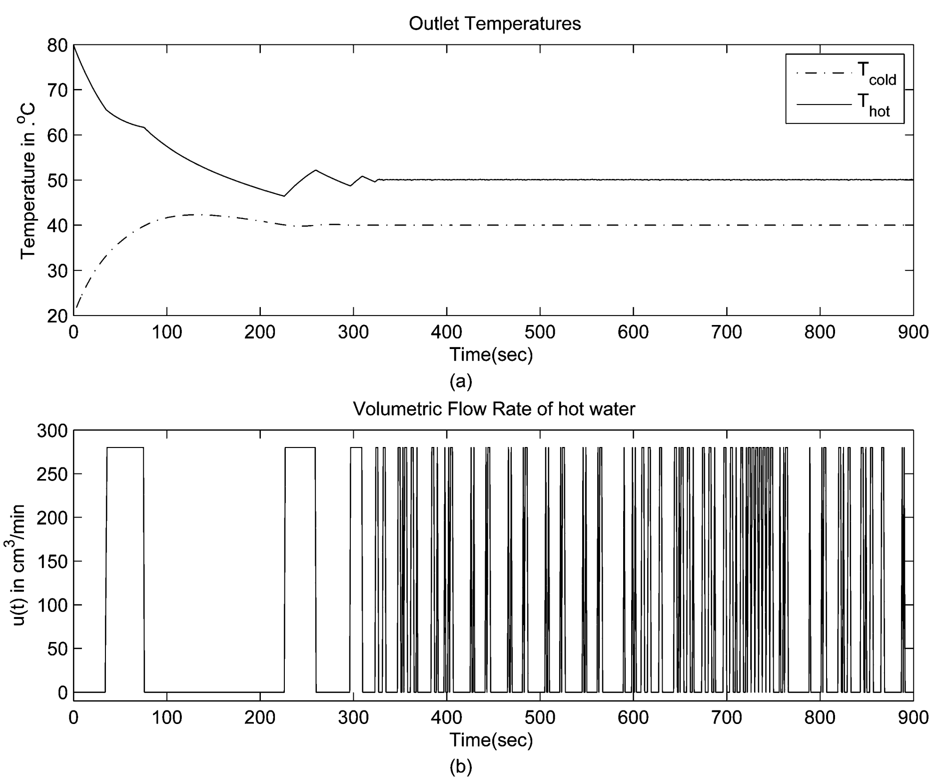

Figure 2. Simulation results when the fast TSM controller is used.

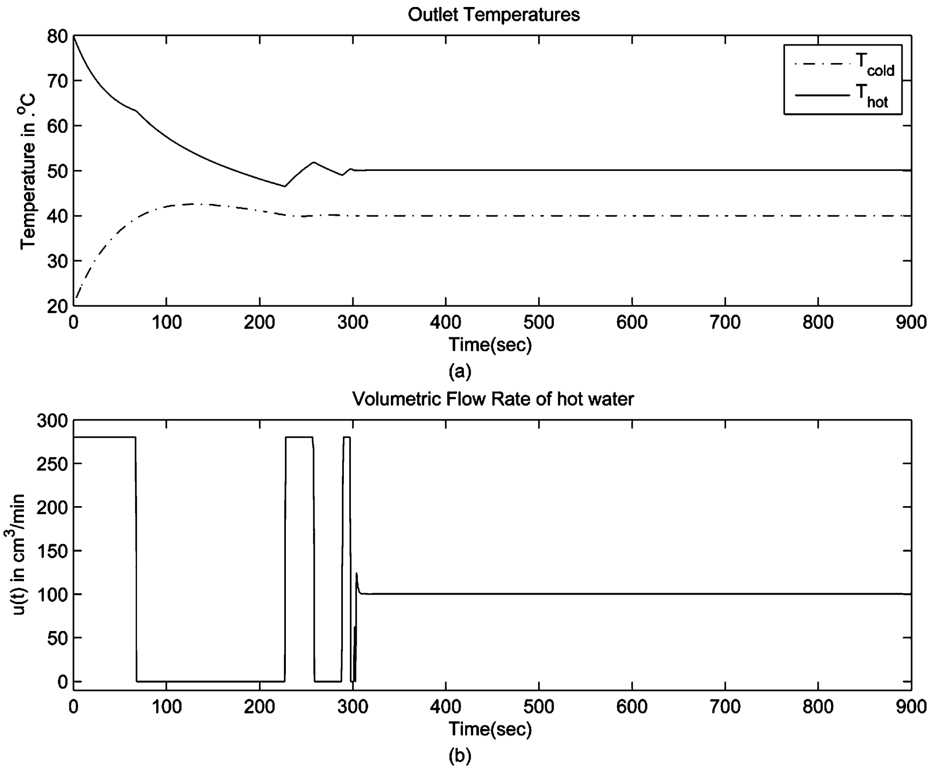

Figure 3. Simulation results when the second-order fast TSM controller is used.

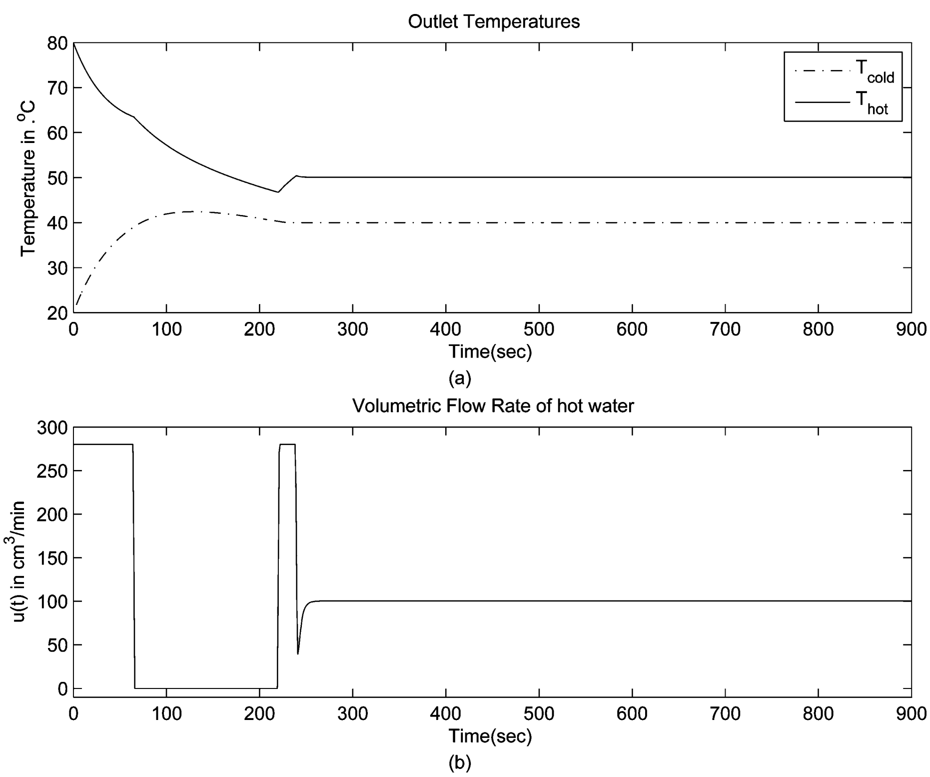

Figure 4. Simulation results when the dynamic fast TSM controller is used.

Clearly, the chattering is greatly reduced compared to the results of the first controller.

The simulation results when the dynamic fast TSM controller given by eq 14 is used are presented in Figure 4. Figure 4a depicts the cold and hot water temperatures versus time. The output of the system converges to its desired value T_{cr} = 40 °C in about 230 s. The hot water temperature converges to a constant steady-state value in about the same time. Figure 4b displays the controller versus time. This controller shows very little chattering.

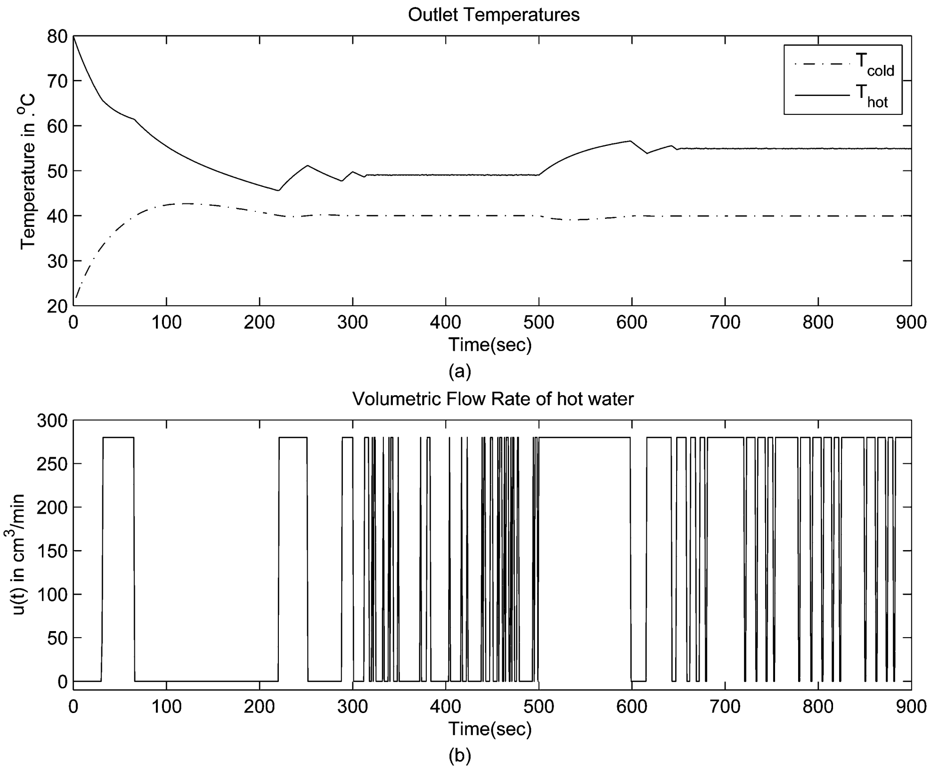

Figure 5. Simulation results using the fast TSM controller when U is increased by 15% and in the presence of some disturbances.

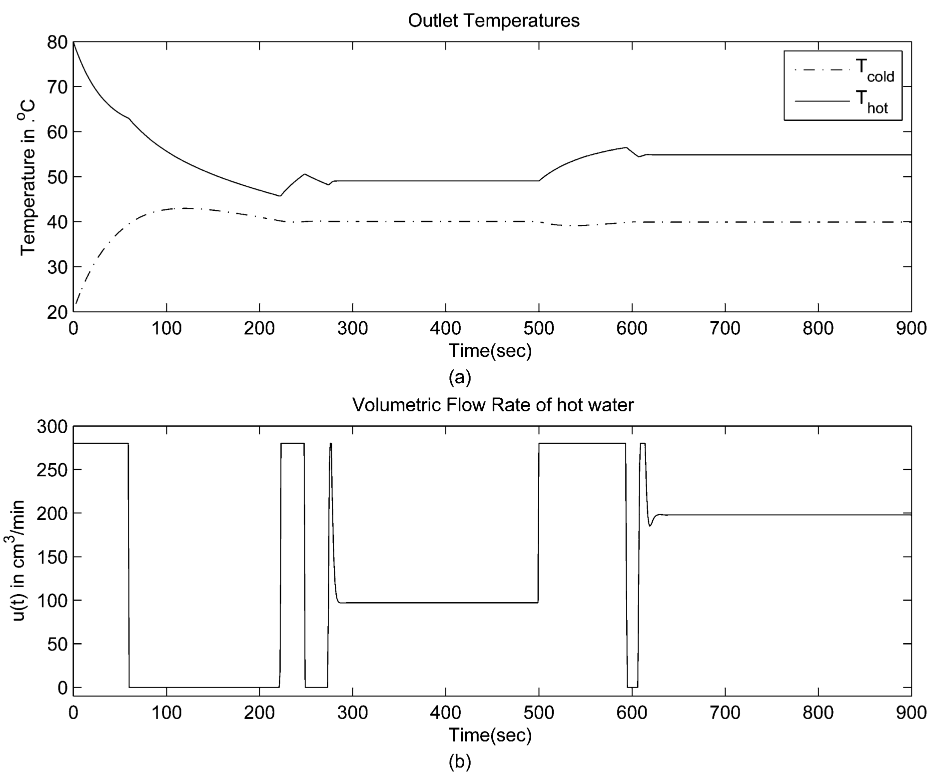

Figure 6. Simulation results using the second-order fast TSM controller when U is increased by 15% and in the presence of some disturbances.

Clearly, the simulation results indicate that the proposed control schemes works well. However, the fast TSM controller suffers from chattering and the chattering is greatly reduced when using the second-order fast TSM controller or the dynamic fast TSM controller. In addition, it is clear that the dynamic fast TSM controller gave the best performance among the proposed three controllers since it has the fastest settling time of the output response.

4.2. Robustness and Disturbance Rejection. The performance of the system is simulated when some parameters

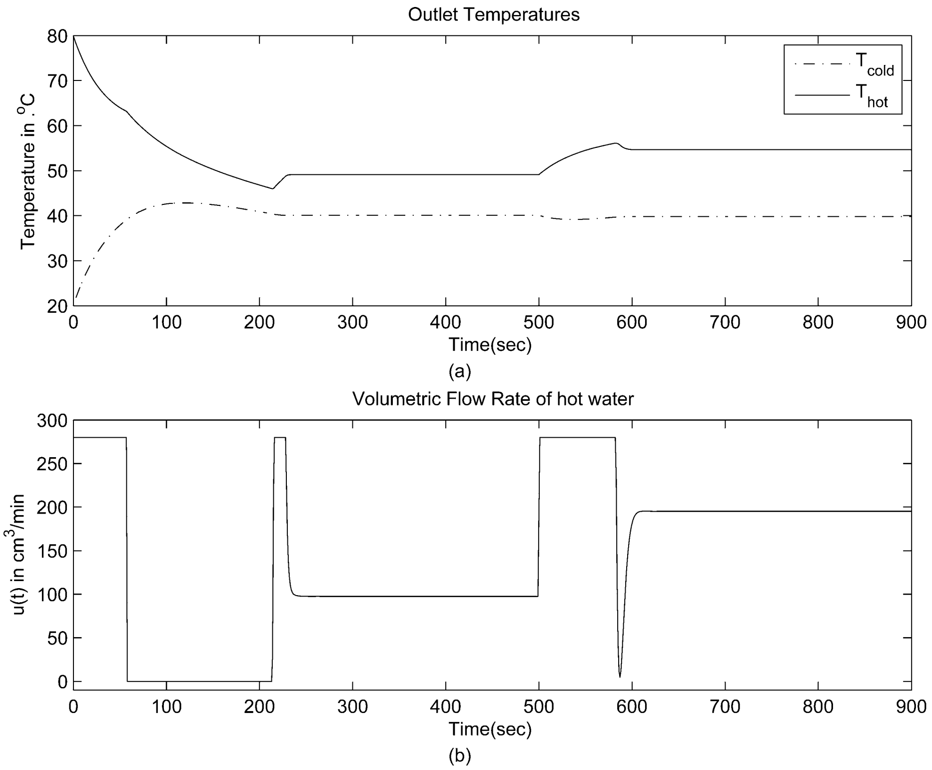

Figure 7. Simulation results using the dynamic fast TSM controller when U is increased by 15% and in the presence of some disturbances.

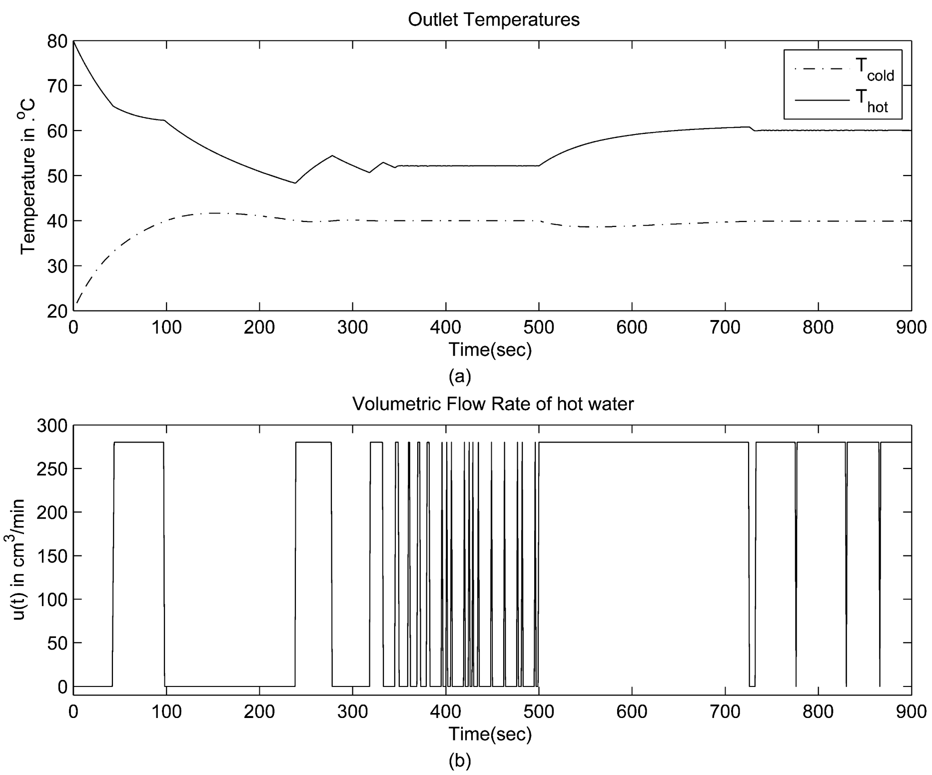

Figure 8. Simulation results using the fast TSM controller when U is decreased by 15% and in the presence of some disturbances.

of the system are assumed not to be known exactly and when some disturbances are acting on the system. In particular, an external disturbance representing an increase in the cold water inflow rate of magnitude 100 cm^3/s is applied to the system at $t = 500$ s. In addition, the constant heat transfer coefficient, U, which is experimentally determined, is increased by 15% of its nominal value. The performance of the system when applying the fast TSM controller is depicted in Figure 5, while the

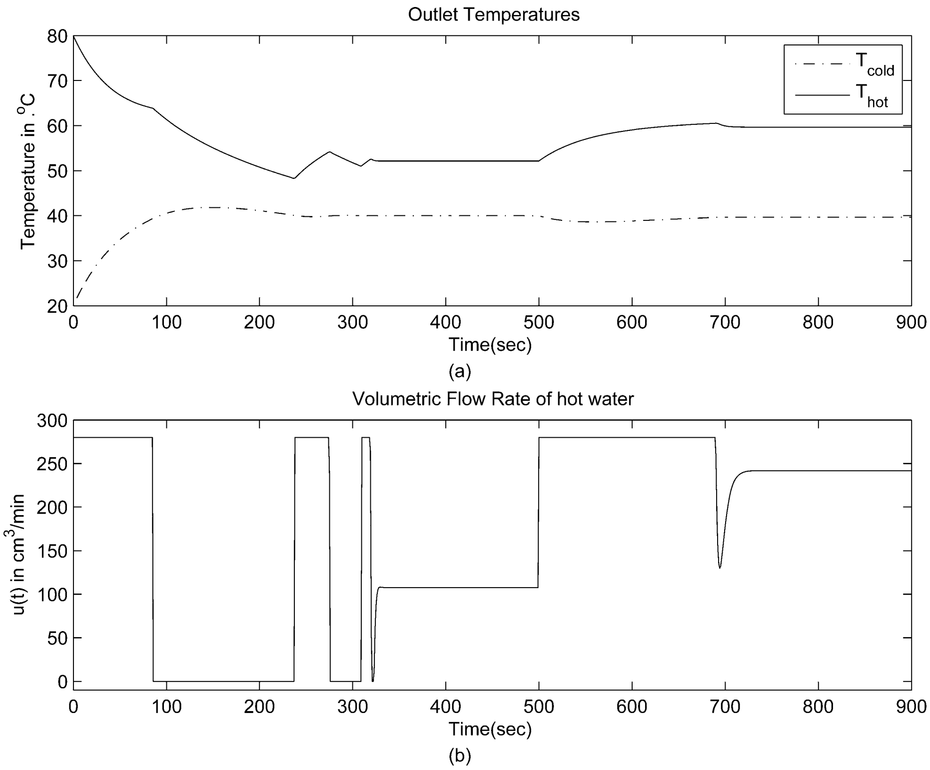

Figure 9. Simulation results using the second-order fast TSM controller when U is decreased by 15% and in the presence of some disturbances.

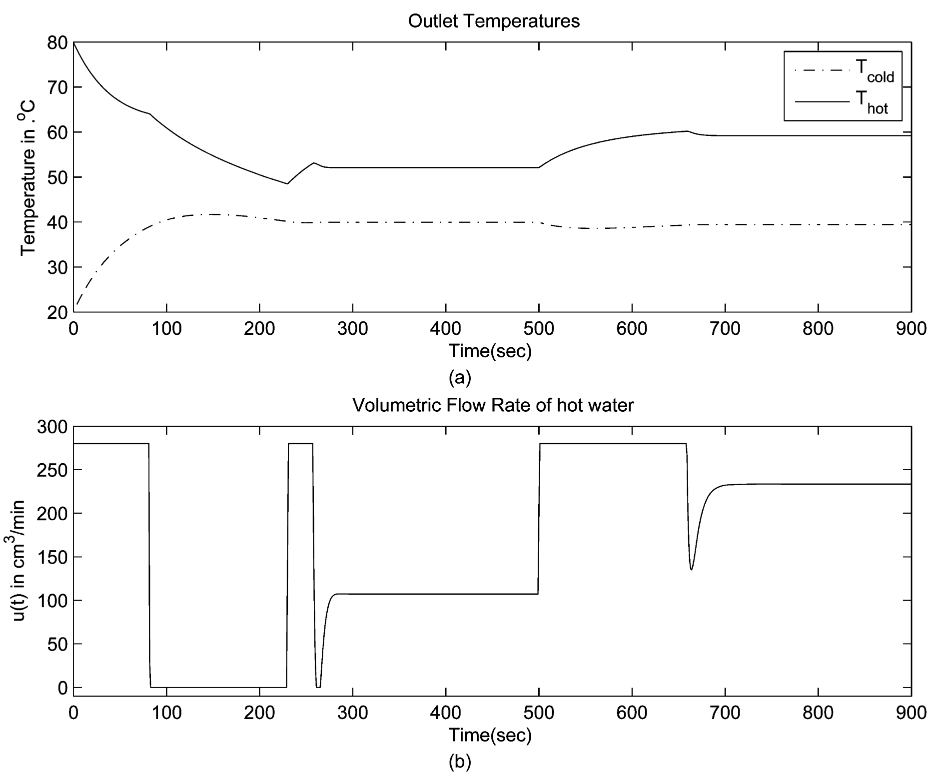

Figure 10. Simulation results using the dynamic fast TSM controller when U is decreased by 15% and in the presence of some disturbances.

performance of the second-order fast TSM controller is depicted in Figure 6, and finally the performance of the dynamic fast TSM is depicted in Figure 7. It can be seen from these responses that the outlet cold water temperature converges to its desired value in all three cases. In addition, these figures show how the control signals adjusted themselves so that the output of the system converges to its desired value.

Moreover, simulations were carried out when the constant heat transfer coefficient, U, is decreased by 15% of its nominal value and in the presence of the external disturbance

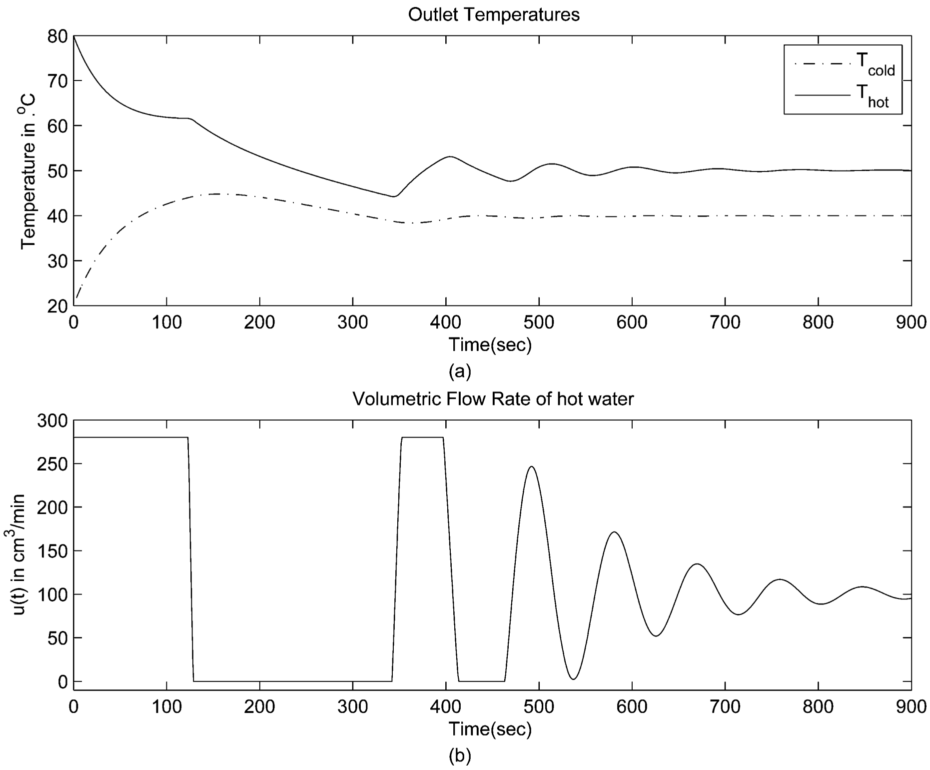

Figure 11. Simulation results when the PI controller is used.

Table 2. ISE Measures for the Proposed Controllers and the PI Controller

	fast TSM	second-order fast TSM	dynamic fast TSM	PI controller
ISE × 10^3	7.0999	7.0950	7.0410	9.3218

on the inflow rate of the cold water. The results for the fast TSM, the second-order fast TSM, and the dynamic fast TSM controllers are given in Figure 8, Figure 9, and Figure 10, respectively. Again, it is clear from these figures that the outlet cold water temperature converges to its desired value for the three different controllers.

Therefore, the simulation results indicate that the proposed control schemes are robust to changes in some of the parameters of the system and to some disturbances acting on the system. It should be mentioned that this is an expected result as it is well established that sliding mode controllers are robust.

4.3. Comparison with PI Controller.

For comparison purposes, we use a standard PI controller to regulate the outlet cold water temperature to its desired value T_{cr} = 40 °C. The transfer function of the PI controller is

$$D(s) = \frac{K_p s + K_i}{s} \tag{19}$$

where K_p is the proportional gain and K_i is the integral gain. After numerous trail-and-error simulation tests, the PI controller gains were determined to be K_p = 600 and K_i = 360. With these values, regulation was achieved while avoiding a large number of oscillations in the control signal. Figure 11 shows the simulation results when the standard PI controller is used. It can be seen that the outlet cold water temperature converges to its desired value T_{cr} in about 550 s. The hot water temperature converges to a constant value in about 800 s. Note that the settling time for the

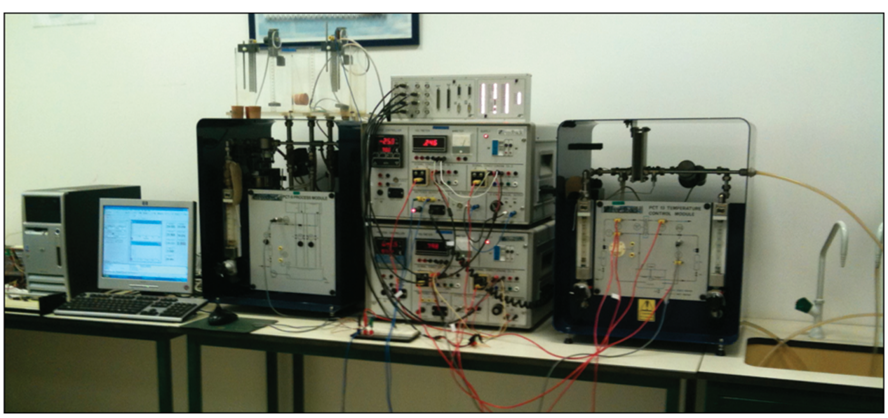

Figure 12. Experimental setup.

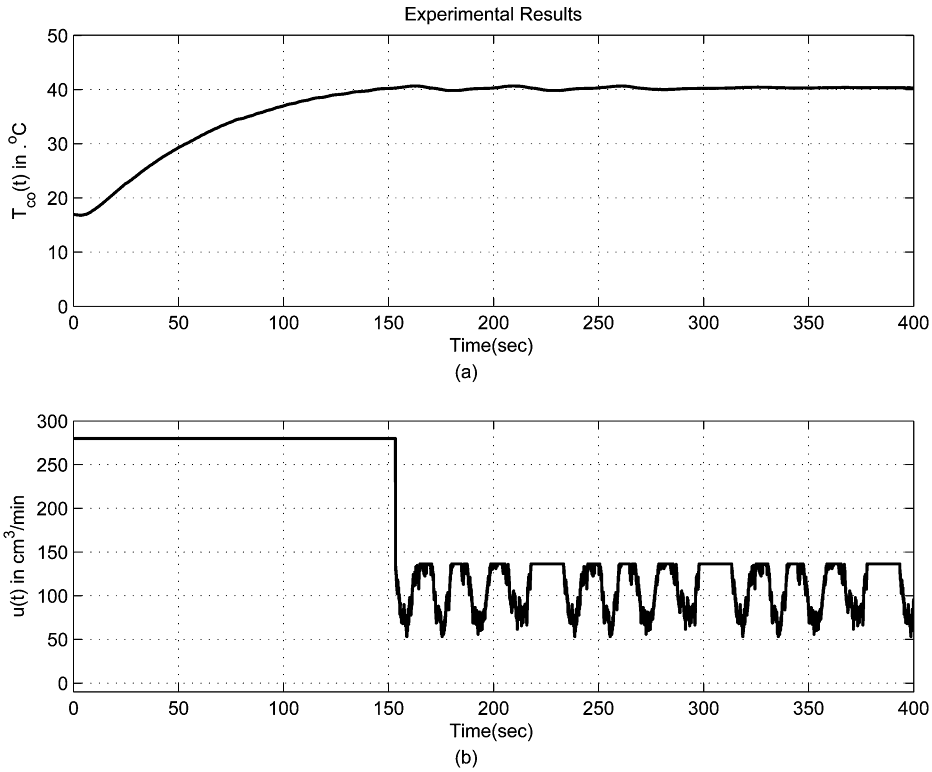

Figure 13. Implementation results when using the fast TSM controller.

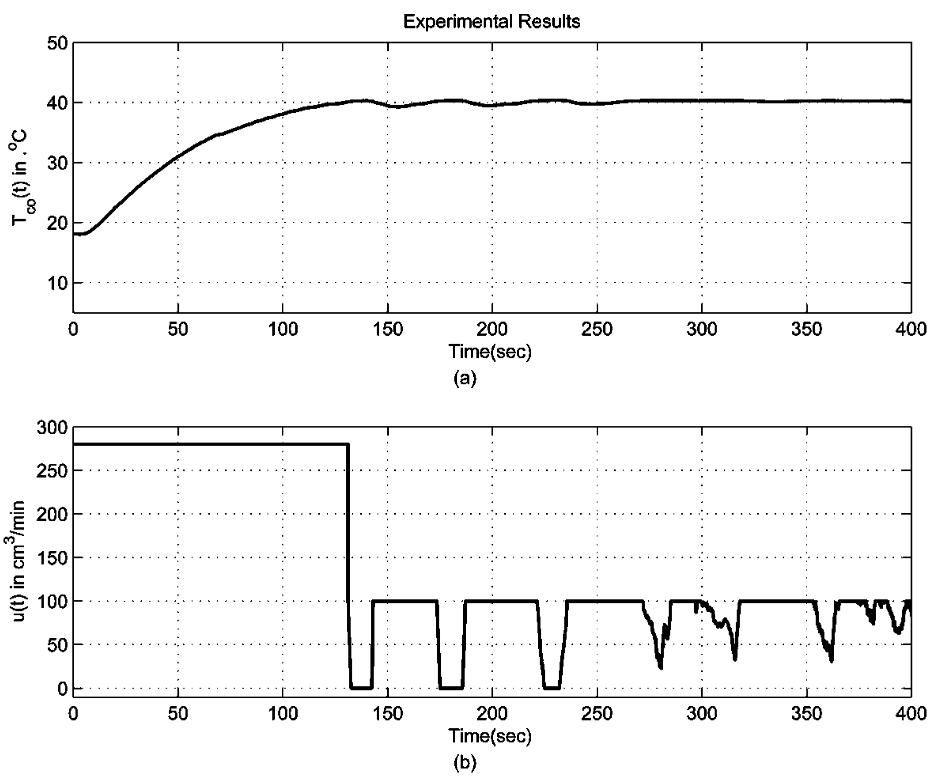

Figure 14. Implementation results when using the second-order fast TSM controller.

temperatures is about twice the settling time obtained when the proposed fast TSM controllers are used. In addition, the integral square error (ISE) measure is used for further comparison. The results are summarized in Table 2.

It can be seen from Table 2 that the dynamic fast TSM controller gave the smallest ISE value among the proposed fast TSM controller; the PI controller gave the largest ISE value.

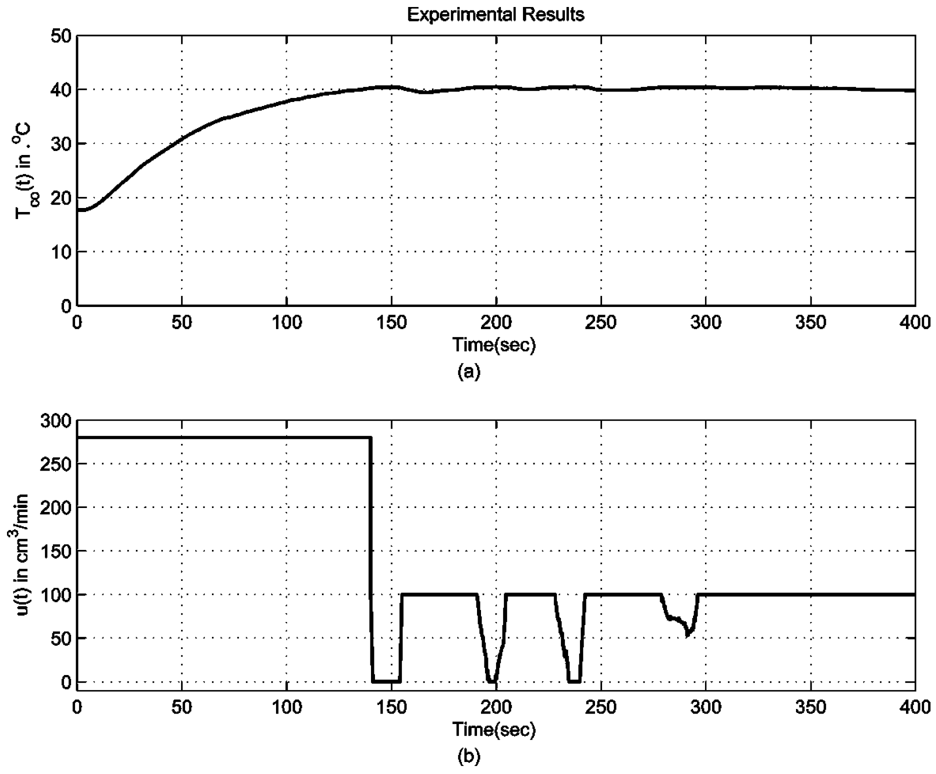

Figure 15. Implementation results when using the dynamic fast TSM controller.

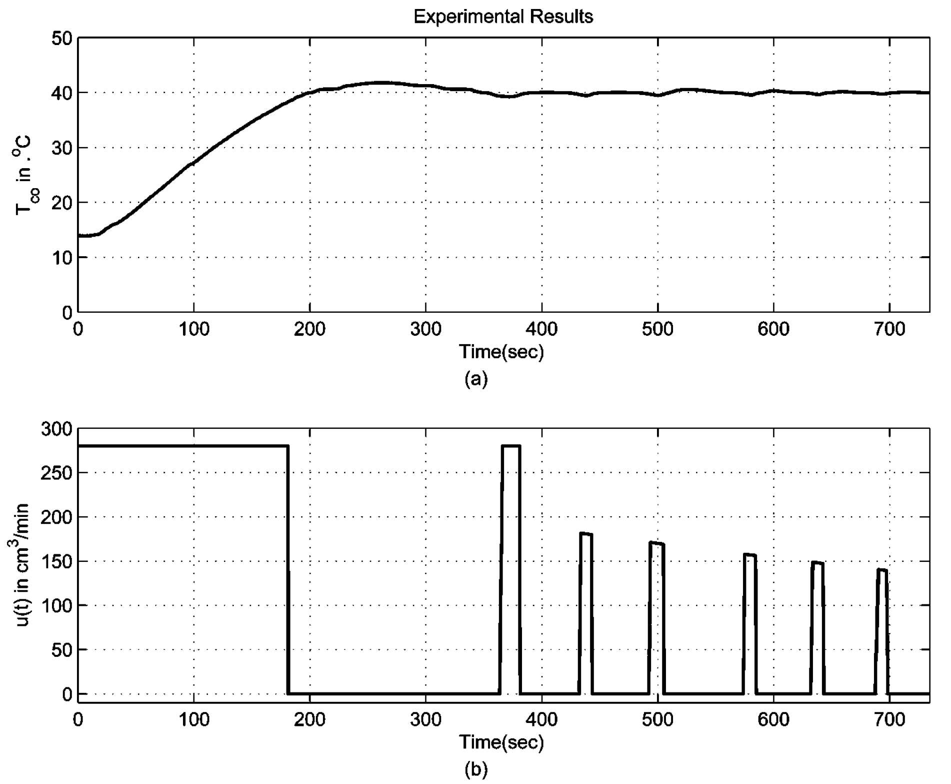

Figure 16. Implementation results when using the PI controller.

Since the simulation results show that the proposed controllers work very well, we decided to implement the controllers using an experimental setup.

5. EXPERIMENTAL RESULTS

The controllers designed in section 3 are implemented on the plate heat exchanger apparatus using the MATLAB/SIMULINK

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software. A dSPACE 1104 card is used for implementation purposes. A photograph of the experimental setup is shown in Figure 12.

The experimental setup consists of a plate heat exchanger and a reservoir with heated water. The plate heat exchanger continuously circulates (in a countercurrent flow) the hot water from an electrically heated reservoir with the cold water. A motor-driven valve is used to control the flow of the heated water. Two thermocouples are used to measure the outlet temperatures of the cold water and the hot water. The input signal to the plate heat exchanger is the voltage $v_{in}(t)$ (in volts); this voltage is applied to the electrical pump which supplies the hot water to the plate heat exchanger. The flow of the hot water through the pump is proportional to the voltage $v_{in}(t)$ and it is given as $u(t) = Kv_{in}(t)$, where K is experimentally determined to be 280. The sampling rate used is 10 Hz. In performing the experiments, we used the set point for the hot water temperature as $T_r = 40$ °C; this is the same set point which was used in the simulations.

5.1. Implementation Results of the Fast TSM Controllers. The parameters of the controllers are taken to be the same values as the ones used for simulation purposes. Figure 13 shows the implementation results when the fast TSM controller given by eq 10 is applied. It can be seen from Figure 13a that the outlet cold water temperature converges to its desired value in about 250 s. The control input $u(t)$ versus time is shown in Figure 13b; chattering is evident in this graph.

Chattering occurs due to the nonlinearity of the sign function that performs the switching operation and generates a discontinuous control signal. This phenomenon manifests itself as undesired oscillations on the system trajectory with finite frequency and amplitude. This phenomenon leads to a reduction in the control accuracy, high wear of the moving mechanical parts of the actuator, and high heat loss in power circuits. It may cause the system to be instable. This phenomenon can be avoided by implementing the proposed second and third fast TSM controllers.

Figure 14 shows the implementation results when the second-order TSM controller given by eq 12 is applied. It can be seen from Figure 14a that the outlet cold water temperature converges to its desired value in about 250 s. The control input $u(t)$ versus time is shown in Figure 14b.

The implementation results clearly show that the chattering is reduced in comparison to the implementation results of the fast TSM controller.

Figure 15 shows the implementation results when the dynamic fast TSM controller given by eq 14 is applied. It can be seen from Figure 15a that the outlet cold water temperature converges to its desired value in about 250 s. The control input $u(t)$ versus time is shown in Figure 15b. Again, note that chattering is reduced compared to the implementation of fast TSM controller.

5.2. Implementation Results of the PI Controller. The parameters of the PI controller are taken to be the same values as the ones used for simulation purposes. Figure 16 shows the implementation results when the standard PI controller is used. It can be seen that the outlet cold water temperature converges to its desired value T_{cr} in about 700 s. The settling time for the PI controller is about triple the settling time of the three proposed control schemes.

Therefore, the implementation results indicate that the three fast TSM controllers work very well. However, it is clear that the dynamic fast TSM controller gives the best results among the three proposed controllers because the resulting control signal is reasonably smooth. Moreover, the three fast TSM controllers gave better responses than a standard PI controller. It should be mentioned that the robustness feature of the sliding mode controllers makes them very attractive compared to the PI controller.

6. CONCLUSION

The control of a plate heat exchanger is addressed in this paper. Three fast TSM control schemes are proposed for the system. These controllers regulate the states of the system to their desired values in finite time. The performances of the controlled system are studied under variations in one of the system parameters and in the presence of an external disturbance. The simulation results indicate that the proposed control schemes work very well and are robust to change in some of the parameters of the system as well as to disturbances acting on the system. In addition, the proposed controllers are implemented using an experimental setup. The implementation results indicate that the proposed controllers work well. A comparison with the standard PI controller shows that the proposed schemes gave better results. Future work will address the multi-input plate heat exchanger system where the temperature of the inlet heated water to the heat exchanger becomes a state variable in the system model. Hence, the model of the system will be higher than two and a new input variable, which is the heater control signal, is introduced in the model.

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Notes

The authors declare no competing financial interest.

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