

Inverse heat transfer problem in digital temperature control in plate fin and tube heat exchangers

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Abstract The aim of the paper is a steady-state inverse heat transfer problem for plate-fin and tube heat exchangers. The objective of the process control is to adjust the number of fan revolutions per minute so that the water temperature at the heat exchanger outlet is equal to a preset value. Two control techniques were developed. The first is based on the presented mathematical model of the heat exchanger while the second is a digital proportional–integral–derivative (PID) control. The first procedure is very stable. The digital PID controller becomes unstable if the water volumetric flow rate changes significantly. The developed techniques were implemented in digital control system of the water exit temperature in a plate fin and tube heat exchanger. The measured exit temperature of the water was very close to the set value of the temperature if the first method was used. The experiments showed that the PID controller works also well but becomes frequently unstable.

Keywords: Heat exchanger; Temperature control; Inverse heat transfer problem; Model based control; Digital PID control

Nomenclature

A – area, m²
 A_{in}, A_{oval} – inside and outside cross section area of the oval tube, m²

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A_{min}	–	minimum free flow frontal area on the air side, m^2
c	–	specific heat, $J/(kg\ K)$
d_h	–	hydraulic diameter of air flow passages, m
d_{min}, d_{max}	–	minimum and maximum outer diameter of the oval tube, respectively, m
d_r	–	hydraulic diameter on the liquid side, $4 A_{in}/P_{in}$, m
h	–	convection heat transfer coefficient, $W/(m^2K)$
k	–	thermal conductivity, $W/(m\ K)$
(k)	–	iteration number,
L_{ch}	–	length of the heat exchanger, m
\dot{m}	–	mass flow rate, kg/s
N_g, N_l	–	number of transfer units for the air and water side, respectively
Nu_w	–	water-side Nusselt number, $h_w d_z/k_w$
Nu_a	–	air-side Nusselt number, $h_a d_h/k_a$
p_1	–	pitch of tubes in plane perpendicular to flow (height of fin), m
p_2	–	pitch of tubes in direction of flow (width of fin), m
P_{in}, P_o	–	inside and outside perimeter of the oval tube, respectively, m
Pr	–	Prandtl number, $\mu c_p/k$
Re_a	–	air-side Reynolds number, $w_{max} d_h/\nu_a$
Re_w	–	liquid-side Reynolds number, $w_w d_r/\nu_w$
s	–	fin pitch, m
T	–	temperature, $^{\circ}C$ or K
T'_{am}, T''_{am}	–	mean inlet and outlet temperature of the air, $^{\circ}C$
T'_w, T''_w	–	water inlet and outlet temperature, respectively, $^{\circ}C$
$T''_{w,1}, T''_{w,2}$	–	water outlet temperature from the first and second tube row in the upper pass, respectively, $^{\circ}C$
T_{wm}	–	outlet temperature of the water after the first pass, $^{\circ}C$
$T''_{w, set}$	–	target (preset) value for the water temperature at the outlet of the heat exchanger, $^{\circ}C$
$T''_{w, meas}$	–	measured water temperature at the outlet of the heat exchanger, $^{\circ}C$
U	–	overall heat transfer coefficient which is referred to the outer surface of the bare tube, $W/(m^2K)$
\dot{V}'_w	–	water volume flow rate at the inlet of the heat exchanger, l/h or m^3/s
w_0	–	average frontal flow velocity (air velocity before the heat exchanger), m/s
w_{max}	–	mean axial velocity in the minimum free flow area, m/s
x, y, z	–	Cartesian coordinates
x^+	–	dimensionless coordinate, $x^+ = x/L_{ch}$

Greek symbols

δ_f	–	fin thickness, m
μ	–	dynamic viscosity, $Pa\ s$
ν	–	kinematic viscosity, m^2/s
ρ	–	fluid density, kg/m^3

Subscripts

a	–	air
g	–	first (upper) pass of the heat exchanger
g	–	gas
in	–	inner
l	–	liquid
p	–	at constant pressure
w	–	water

Superscripts

$+$	–	dimensionless
$-$	–	mean
$'$	–	inlet
$''$	–	outlet
I	–	first tube row
II	–	second tube row

1 Introduction

The subject of the paper is a digital control of water temperature at the outlet from the plate fin and tube heat exchangers. Plate fin and tube heat exchangers are used as economizers in steam power boilers, air-conditioning coils, convectors for home heating, waste-heat recovery systems for gas turbines, cooling towers, air-fin coolers, car radiators and heater cores, which are used to heat the air within the car passenger compartment. Compact heat exchangers with plate fins can be manufactured by electrical or laser welding of the fins to the tube or can also be extruded directly from the tube wall metal [1].

The objective of the process control is to adjust the number of fan revolutions per minute so that the water temperature at the heat exchanger outlet is equal to a preset value. Two control techniques were developed. The first is based on the presented mathematical model of the heat exchanger while the second is a digital proportional–integral–derivative (PID) control. The first procedure is very stable. Since the outlet water temperature is a nonlinear function of the fan revolution number, a nonlinear algebraic equation was solved in on line mode using the secant or interval searching method. The steady-state outlet water temperature was calculated at every search or iteration step using a numerical model of the heat exchanger. A numerical model of the plate-fin and tube heat exchanger with two tube rows and two passes was developed. A steady-state inverse heat transfer problem for plate-fin and tube heat exchangers was solved. Test calculations show that

the accuracy of the developed model is very satisfactory. The advantage of the numerical heat exchanger model is its flexibility in building mathematical models of heat exchanger with complex flow arrangements. The temperature dependent physical properties of fluids can easily be accounted for. The digital PID controller becomes unstable if the water volumetric flow rate changes significantly.

The developed techniques were implemented in digital control of the water exit temperature in a plate fin and tube heat exchanger. The measured exit temperature of the water was very close to the set value of the temperature if the first method was used. The experiments showed that the PID controller works also well but becomes frequently unstable.

2 Model based control

The aim of the model based control is to adjust the number of fan revolutions per minute n so that the measured water temperature $T''_{w, meas}$ at the heat exchanger outlet is equal to a preset value (set point) $T''_{w, set}$:

$$T''_{w, meas}(n) - T''_{w, set} = 0. \quad (1)$$

Since the outlet water temperature T''_w is a nonlinear function of the fan revolution number, a nonlinear algebraic equation was solved using the secant method. The steady-state outlet water temperature was calculated at every search or iteration step using a numerical mathematical model of the heat exchanger. A numerical model of the plate-fin and tube heat exchanger with two tube rows and two passes, allowing for different heat transfer coefficients on each tube row, was developed. The radiator is a double-pass heat exchanger, consisting of two tube rows. The hot water flows in parallel through both tube rows. The tube outlets in the upper pass converge into one manifold where the water from the first and second tube row are mixed.

A basis for the numerical modeling of the heat exchanger is two heat balance equations for a single tube (Fig. 1) in a differential form

$$\frac{dT_l}{dx^+} = -N_l [T_l(x^+) - T_{mg}(x^+)], \quad (2)$$

$$\frac{\partial T_g(x^+, y^+)}{\partial y^+} = N_g [T_l(x^+) - T_g(x^+, y^+)] \quad , \quad (3)$$

where the mean gas temperature $T_{mg}(x^+)$ over the row thickness p_2 is given by

$$T_{mg}(x^+) = \int_0^1 T_g(x^+, y^+) dy^+ . \quad (4)$$

Equations (2) and (3) are subject to the following boundary conditions

$$T_l|_{x^+=0} = T'_w , \quad (5)$$

$$T_g|_{y^+=0} = T'_{am} . \quad (6)$$

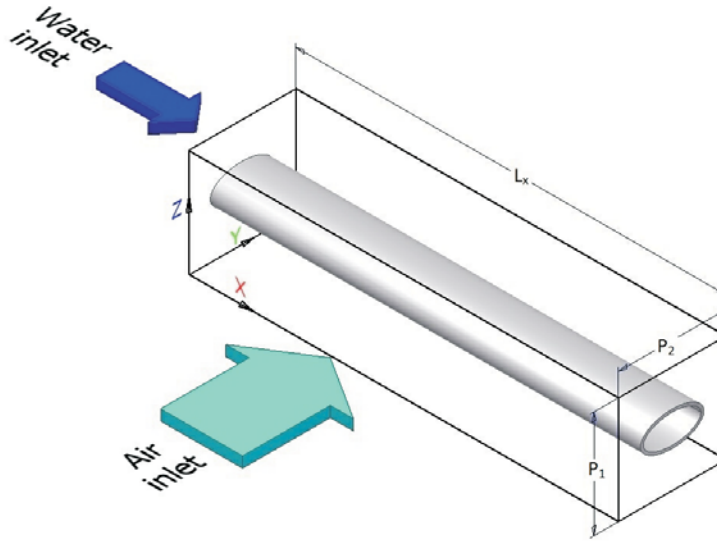


Figure 1. Single tube in cross-flow heat exchanger.

The discrete mathematical model of the heat exchanger was developed using the finite volume method. A division of fin and tube heat exchanger with oval tubes is shown in Fig. 2. Figure 3a presents control volumes for the energy balance equation on the gas-side and Fig. 3b for the energy balance equation on the liquid-side. When there are N finite volumes on the width of the heat exchanger, then $\Delta x = L_{ch}/N = L_x/N$. Assuming that the water temperature is constant in direction of air flow over the pitch p_2 and finite volume length Δx , the gas temperature $T''_{g,i}$ at the outlet of the finite volume (Fig. 3a) is calculated from the following expression:

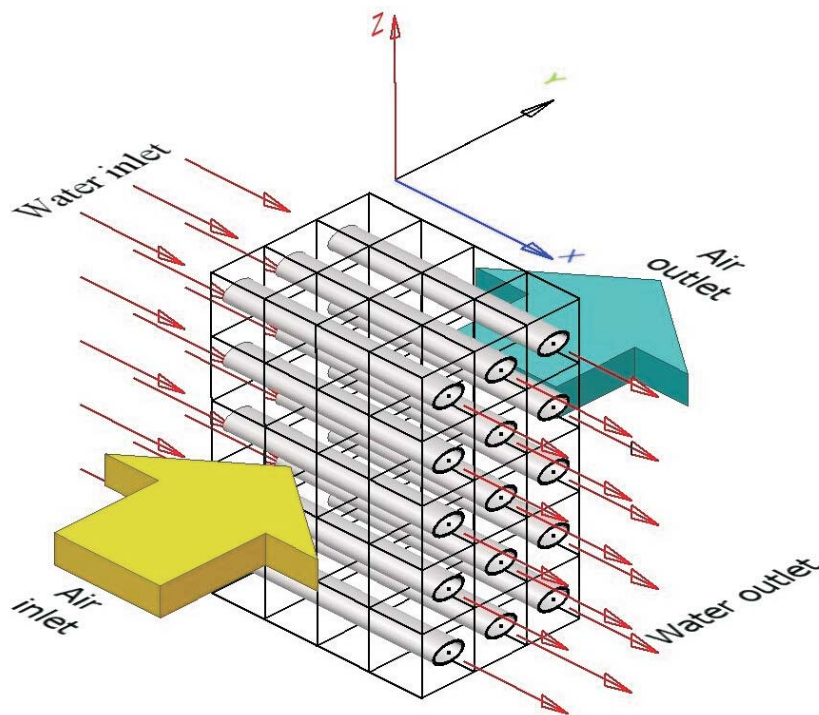


Figure 2. Division of a cross flow tube heat exchanger into control volumes.

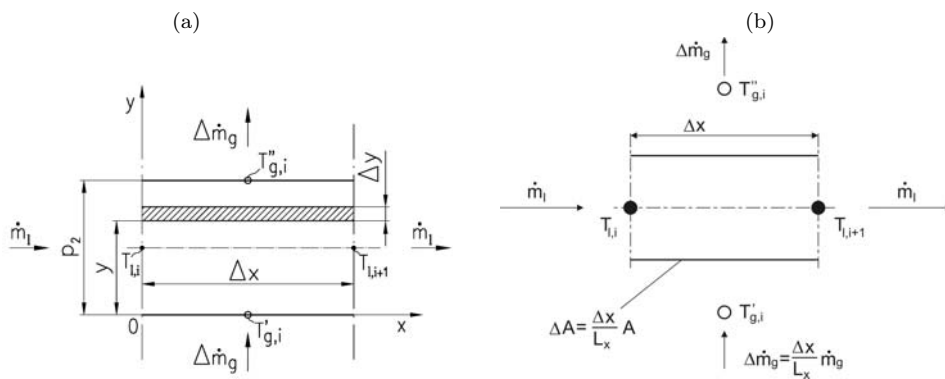


Figure 3. Control volumes used for determining gas temperature (a) and liquid temperature (b).

$$T''_{g,i} = \bar{T}_{l,i} - (\bar{T}_{l,i} - T'_{g,i}) \exp(-\Delta N_{g,i}), \quad (7)$$

where:

$$\Delta A_{oval} \frac{A_{oval}}{N}, \quad \Delta \dot{m}_g = \frac{\dot{m}_g}{N}, \quad \Delta N_{g,i} = \frac{U \Delta A_{oval}}{\Delta \dot{m}_g \bar{c}_{pg,i}} = \frac{U A_{oval}}{\dot{m}_g \bar{c}_{pg,i}}, \quad (8)$$

$$\bar{T}_{l,i} = \frac{T_{l,i} + T_{l,i+1}}{2}.$$

The liquid (water) temperature at the outlet of the finite volume (Fig. 3b) may be expressed as [4]:

$$T_{l,i+1} = \frac{1}{1 + \frac{\Delta N_{l,i+\frac{1}{2}}}{2}} \left[\left(1 - \frac{\Delta N_{l,i+\frac{1}{2}}}{2} \right) T_{l,i} + \Delta N_{l,i+\frac{1}{2}} \bar{T}_{g,i} \right], \quad i = 1, \dots, N, \quad (9)$$

where

$$\Delta N_{l,i+\frac{1}{2}} = \frac{U \Delta A_{oval}}{\dot{m}_l \bar{c}_{l,i}} = \frac{2U \Delta A_{oval}}{\dot{m}_l [c_l(T_{l,i}) + c_l(T_{l,i+1})]}, \quad (10)$$

$$\bar{T}_{g,i} = \bar{T}_{l,i} - \frac{1}{\Delta N_{g,i}} (\bar{T}_{l,i} - T'_{g,i}) [1 - \exp(-\Delta N_{g,i})].$$

The mean specific heat capacity for gas $\bar{c}_{pg,i}$ and liquid $\bar{c}_{l,i}$ are

$$\bar{c}_{pg,i} = \frac{c_{pg,i} \Big|_0^{T_g|_y} \cdot T_g|_y - c_{pg,i} \Big|_0^{T_g|_{y+\Delta y}} \cdot T_g|_{y+\Delta y}}{T_g|_y - T_g|_{y+\Delta y}}, \quad (11)$$

$$\bar{c}_{l,i} = \frac{c_c(T_{l,i}) + c_c(T_{l,i+1})}{2}.$$

The layout of the radiator is shown in Fig. 4. The whole heat exchanger was divided into finite volumes (Fig. 2). The set of nonlinear algebraic equations (7) and (9) was solved for each tube row using the Gauss-Seidel method. At first, the air and water temperatures at the first and second tube row of the first pass were calculated.

Upon mixing of water with the temperature $T''_{w,1} = T_{w,1}(L_{ch})$ from the first tube row and water with the temperature $T''_{w,2} = T_{w,2}(L_{ch})$, the feed water temperature of the second lower pass is T''_{wm} (Fig. 4). The temperature $T''_{w,3}$, $T''_{w,4}$, and T''_w can be calculated in the similar way [4].

In the second lower pass the total mass flow rate splits into two flow rates $\dot{m}_w/2$. At the outlet from the first tube row in the bottom pass the water temperature is $T''_{w,3}$, and from the second row is $T''_{w,4}$. Upon mixing the water from the first and second row, the final water temperature is T''_w . This T''_w temperature is thus the radiator exit temperature. The air stream \dot{m}_a flows crosswise through both tube rows (Fig. 4).

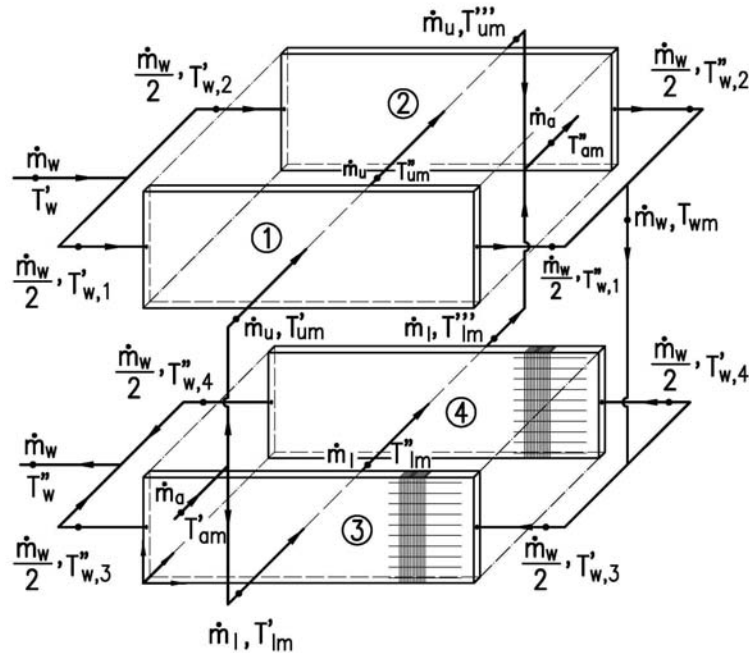


Figure 4. Two-pass plate-fin and tube heat exchanger with two in-line tube rows; 1 – first tube row in upper (first) pass, 2 – second tube row in upper pass; 3 – first tube row in lower (second) pass, 4 – second tube row in lower pass; T'_w and T''_w – inlet and outlet water temperature, respectively, T'_{am} and T''_{am} – inlet and outlet air mean temperature, respectively, \dot{m}_a , \dot{m}_w – air and water mass flow rate at the inlet of the heat exchanger, respectively.

To calculate the overall heat transfer coefficient U , the heat transfer coefficient on the water h_w and the air side h_a are determined using the empirical correlations. Extensive heat transfer measurements under steady-state conditions were conducted to find the correlations for the air and water side Nusselt numbers, which enable calculation of heat transfer coefficients.

Based on 21 measurement series, the following correlations were identified:

$$\text{Nu}_a = 0.0112 \text{Re}_a^{1.0959} \text{Pr}_a^{1/3}, \quad 75 \leq \text{Re}_a \leq 350, \quad (12)$$

$$\text{Nu}_w = \frac{\frac{\xi}{8} (\text{Re}_w - 75) \text{Pr}_w}{1 + 33 \sqrt{\frac{\xi}{8}} (\text{Pr}_w^{2/3} - 1)} \left[1 + \left(\frac{d_r}{L_{ch}} \right)^{2/3} \right], \quad 1750 \leq \text{Re}_w \leq 7000, \quad (13)$$

where the friction factor ξ is given by

$$\xi = \frac{1}{(1.82 \log \text{Re}_w - 1.64)^2} = \frac{1}{(0.79 \ln \text{Re}_w - 1.64)^2}. \quad (14)$$

The water side Reynolds number $\text{Re}_w = w_w d_r / \nu_w$ is based on the hydraulic diameter $d_r = 4A_{in}/P_{in}$, where A_{in} denotes inside the cross section area of the oval tube. The hydraulic diameter for the investigated radiator is: $d_r = 7.06 \times 10^{-3}$ mm. The air-side Reynolds number is defined as $\text{Re}_a = w_{\max} d_h / \nu_a$, where w_{\max} is mean axial velocity of air in the minimum free flow area and $d_h = 1.42 \times 10^{-3}$ m is the air side hydraulic diameter [4].

The maximum air velocity w_{\max} occurring within the tube row is defined by

$$w_{\max} = \frac{sp_1}{(s - \delta_f)(p_1 - d_{\min})} \frac{\bar{T}_{am} + 273}{T'_{am} + 273} w_0, \quad (15)$$

where w_0 is the air velocity before the radiator. As the tubes in the radiator are set in line, w_{\max} is the air velocity in the passage between two tubes. The thermophysical properties of the water were determined at the mean temperature $\bar{T}_w = (T'_w + T''_w)/2$, where T'_w and T''_w denote the inlet and outlet temperatures. All properties appearing in the equation (9) for the air are also evaluated at the mean temperature $\bar{T}_{am} = (T'_{am} + T''_{am})/2$. The physical properties of air and water were approximated using simple functions. The method of prediction of heat transfer correlations (12) and (13) is described in detail in [4]. Based on the experimental results, the air velocity before the heat exchanger w_0 can be expressed as a function of the number of fan revolutions per minute n :

$$w_0 = 0.001566n - 0.0551, \quad 250 \leq n \leq 1440. \quad (16)$$

The measured water temperature $T''_{w, meas}$ at the outlet of the heat exchanger should be equal to a target value (set point). The control system based on the presented mathematical model is depicted in Fig. 5. The goal

of the heat exchanger control system is to adjust the speed of fan rotation n in order that the water temperature $T''_{w, meas}$ at the heat exchanger outlet is equal to a preset value $T''_{w, set}$. At the desired set point the heat transfer process will be at steady-state. Therefore the steady state form of the mathematical model of the heat exchanger is used to determine the speed of fan rotation n measured in revolutions per minute. The frequency of determining the number of revolutions per minute (rpm) depends on the time constant of the heat exchanger, time constants of the thermocouples measuring the water temperature at the inlet and outlet of the heat exchanger, and on the time constant of the fan.

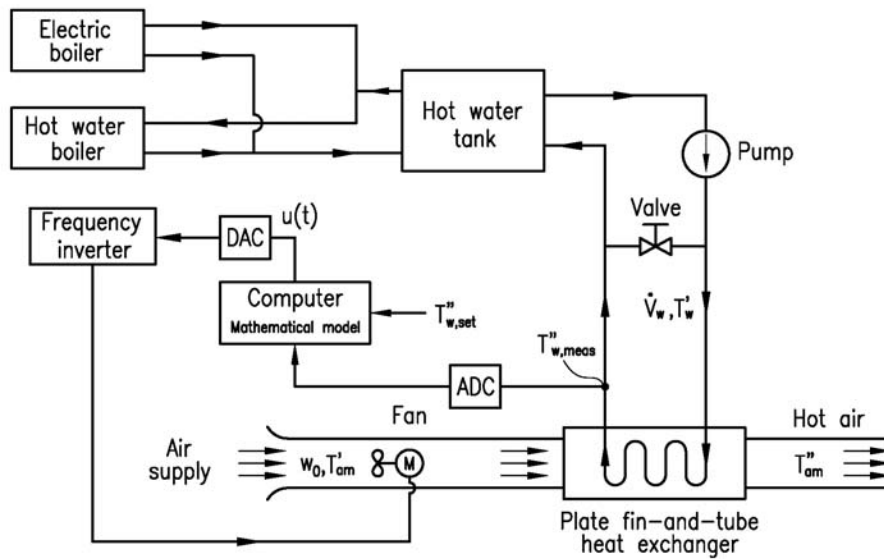


Figure 5. Block diagram of model based control loop to maintain preset water outlet temperature T''_w by adjusting the number of fan revolutions n (ADC and DAC denote respectively the analog and digital converters).

The speed of fan rotation n was determined sequentially with a time step Δt_c , which is at least five times greater than the time constant of the controlled process. In the present study, the time step was $\Delta t_c = 120$ s. The temperatures and mass flow rates of the water and air were measured every time interval $\Delta t = 10$ s. All the measured signals were pre-filtered before use as the input values to the mathematical model. Measured temperatures

and flow-rates were smoothed using a seven-point moving average filter [6]:

$$y_i = y(t_i) \approx \frac{(-2 f_{i-3} + 3 f_{i-2} + 6 f_{i-1} + 7 f_i + 6 f_{i+1} + 3 f_{i+2} - 2 f_{i+3})}{21}, \quad (17)$$

where f_{i-3}, \dots, f_{i+3} denote measured values in seven successive time points. The digital filter approach is very appropriate for on-line applications because it is much more computationally efficient than other signal smoothing methods. Equation (16) makes it possible partial elimination of random errors from measured signals.

The model based control system ensures perfect control provided that the model prediction is accurate.

3 Digital PID control

The most commonly used controller in the industry is a PID controller, e.g. a controller which output is the sum of proportional, integral and derivative mode outputs [7–11]:

$$u(t) = \bar{u} + K_p \left[e(t) + \frac{1}{\tau_i} \int e(t) dt + \tau_d \frac{de(t)}{dt} \right], \quad (18)$$

where: u – controller output, \bar{u} – steady-state controller output, e – control error which is the difference between the set point and process variable (process measurement), K_p – controller gain, τ_i – integral or reset time, τ_d – derivative time.

Output of the PID controller can be represented in the alternative form:

$$u(t) = \bar{u} + K_p \left[e(t) + K_i \int_0^t e(t) dt + K_d \frac{de(t)}{dt} \right], \quad (19)$$

where: $K_i = K_p / \tau_i$ – integral or reset gain, $K_d = K_p \tau_d$ – derivative gain.

Process data are measured at time intervals Δt and the PID difference equation can be obtained from Eq. (18) if the integral in Eq. (18) is calculated using the rectangle method and the derivative is approximated by a backward difference quotient to give:

$$u_k = \bar{u} + K_p \left(e_k + K_i \Delta t \sum_{i=1}^k e_i + K_d \frac{e_k - e_{k-1}}{\Delta t} \right), \quad k = 1, 2, \dots \quad (20)$$

$$u_{k-1} = \bar{u} + K_p \left(e_{k-1} + K_i \Delta t \sum_{i=1}^{k-1} e_i + K_d \frac{e_{k-1} - e_{k-2}}{\Delta t} \right), \quad (21)$$

where: $u_k = u(t_k)$, $e_k = e(t_k)$, $t_k = k\Delta t$, $k = 1, 2, \dots$. Taking into account Eqs. (20) and (19), the output change $u_k - u_{k-1}$ from one step to the next can be expressed as:

$$u_k - u_{k-1} = K_p \left(e_k - e_{k-1} + K_i \Delta t e_k + K_d \frac{e_k - 2e_{k-1} + e_{k-2}}{\Delta t} \right). \quad (22)$$

A simple transformation of Eq. (20) gives:

$$u_k = u_{k-1} + K_p \left(e_k - e_{k-1} + K_i \Delta t e_k + K_d \frac{e_k - 2e_{k-1} + e_{k-2}}{\Delta t} \right), \quad k = 1, 2, \dots \quad (23)$$

Equation (23) was implemented in the developed digital PID control system.

4 Experimental results

The proposed methods for determining the number of fan revolutions per minute n , which is required to maintain the constant water temperature T_w'' at the outlet of the heat exchanger, was verified experimentally. Equation (1) was solved iteratively using the secant method. The iteration tolerance ε , i.g. the difference between the last rpm $n^{(k)}$ and the penultimate rpm $n^{(k-1)}$ was assumed $\varepsilon = 0.001$ rpm. The measured values of T_w' , T_{am}' , \dot{V}_w' , and T_w'' were taken as the input data. The preset and measured water temperatures at the outlet of the heat exchanger are compared in Figures 6a and 6b. Inspection of the results shown in Fig. 6a indicates that the measured water temperature $T_{w,meas}''$ is very close to the preset temperature value $T_{w,set}''$. Only small discrepancies between measured and prescribed temperature are observed for sudden changes in volumetric flow rates (Fig. 6b).

To maintain the preset temperature of the water $T_{w,set}''$ at the heat exchanger outlet a digital PID controller was also used. The preset and measured values of the temperature T_w'' and the corresponding numbers of fan revolutions n , which were found by the PID controller, are shown in Fig. 7. The PID controller was investigated for two different sets of constants K_p , K_i , and K_d . It can be seen that after changing volumetric flow rate at time about 200 s, the controller became unstable. Continuous oscillations of the measured temperature $T_{w,meas}''$ are caused by the unstable operation of the

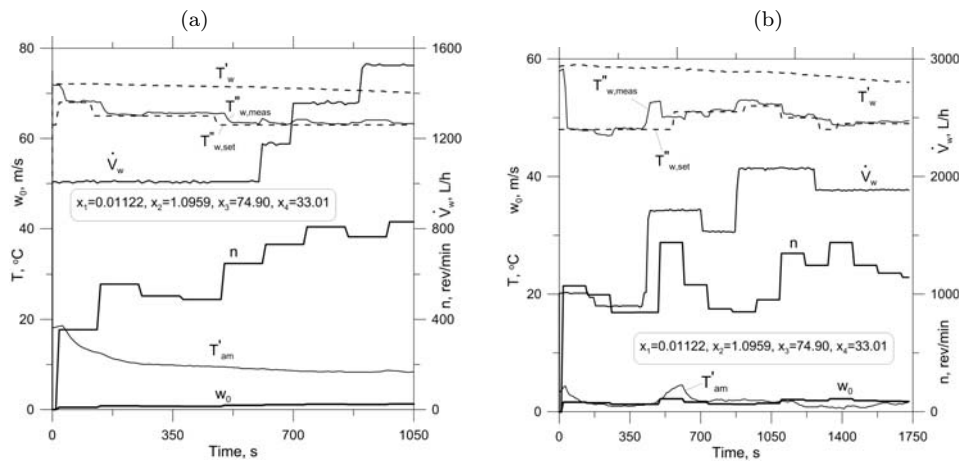


Figure 6. Experimental results of water outlet temperature control using the model based controller for two different time periods; (a) 0–1050 s, (b) 0–1750 s; n – number of fan revolutions per minute (rpm) set by the controller, w_0 – air velocity before heat exchanger, \dot{V}_w – water volumetric flow rate, T'_w and T'_{am} – water and air temperature at the inlet of heat exchanger, $T''_{w,set}$ and $T''_{w,meas}$ – set and measured value of water temperature at the outlet of the heat exchanger.

PID controller. Resetting volumetric flow rate to the previous value at time about 750 s makes again the controller stable (Fig. 7a). If controller constants are not adjusted precisely then the rpm setting time is long (Fig. 7b).

The model based control method has an advantage over the PID control of heat exchangers since the PID controller can become unstable if the water heat flow rate changes. After changing of the volumetric flow rate on the water side, the PID controller can become unstable. The fan was turned on and off with very high frequency (Fig. 7). Such operation of the fan can lead to its early wear out.

5 Conclusions

Two methods for control of fluid temperature in plate fin-and-tube heat exchangers were presented. The first method is based on the mathematical model of the heat exchanger while the second method represents a digital PID control. The measured water temperature at the outlet of the heat exchanger should be equal to a preset target value.

The model based control of the water exit temperature from the plate

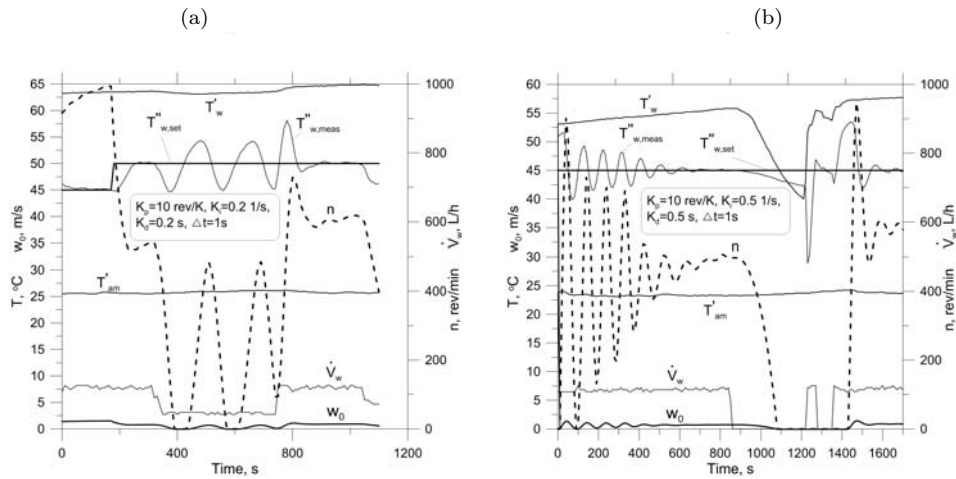


Figure 7. Experimental results of water outlet temperature control using digital PID controller for two different time periods; (a) 0–1200 s, (b) 0–1600 s; n – number of fan revolutions per minute (rpm) set by the controller, w_0 – air velocity before heat exchanger, \dot{V}_w – water volumetric flow rate, T_w' and T_{am}' – water and air temperature at the inlet of heat exchanger, T_w'' , $T_{w,meas}$ and $T_{w,set}$ – set end measured value of water temperature at the outlet of the heat exchanger.

fin-and-tube heat exchanger was proposed. To determine the number of fan revolutions a new mathematical model of the plate fin and tube heat exchanger was developed and implemented in the temperature control system. This kind of control allows the heat exchanger to achieve the steady-state much quicker in comparison to the classic PID control. In contrast to the PID control, the model based control system using the developed procedure for determining the rotational speed of the fan for the prescribed water temperature at the outlet of the heat exchanger is always stable and accurate provided that the numerical model of the heat exchanger is also accurate.

The advantage of the proposed digital PID controller is its simplicity. The mathematical modeling of the heat exchanger is not needed but transient mathematical model of the heat exchanger could be used for tuning PID controller.

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