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Dynamics of plate heat exchangers subject to flow variations

Anil Kumar Dwivedi, Sarit Kumar Das *

Heat Transfer and Thermal Power Laboratory, Department of Mechanical Engineering, Indian Institute of Technology Madras, Chennai 600 036, India

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Abstract

A predictive model has been presented to suggest the transient response of plate heat exchangers, subjected to a step flow variation. The work also brings out the effect of the port to channel maldistribution on the performance of plate heat exchangers under the condition of flow variation. The results indicate that flow maldistribution affects the performance of the plate heat exchangers in the transient regime. A wide range of the parametric study has been presented which brings out the effects of NTU and heat capacity rate ratio on the response of the plate heat exchanger, subjected flow perturbation.

To verify the presented theoretical model, appropriate experiments have been carried out. Experiments include the responses of the outlet temperatures subjected to inlet temperature transient in the circuit followed by a sudden change in flow rate in one of the fluids. Simulated performance has been compared to the performance measured in the experiments. Comparisons indicate that theoretical model developed for flow transient is capable of predicting the transient performance of the plate heat exchangers satisfactorily, under the given conditions of changed flow rates.

Keywords: Pate heat exchanger; Flow variations; Transient response of heat exchangers; Flow maldistribution; Control of heat exchangers

1. Introduction

Plate heat exchangers are important components of process and power industry today. Initially, use of the plate heat exchangers was limited to hygienic industries such as food processing, pharmaceuticals and dairy industries primarily due to their ease of clearing. Now a days they are finding increasing usage over wide variety of applications because of the advantages such as flexibility, higher heat transfer, ease of maintenance, compactness, lower rates of fouling, less effect of flow induced vibration and better controllability.

It is very important to know about the behaviour of plate heat exchangers, when it is subject to flow transient, not only due to possible flow perturbation in the process loop but also to know about the flow variation required to impart control on the plate heat exchanger when a temperature transient takes place. Dynamic analysis of heat exchangers provides information about transient responses subjected to various disturbances. There are many cases in which the system dynamic behaviour is a prime design consideration. To asses this behaviour accurately, one needs to look at flow distribution in heat exchangers. In reality the flow is distributed non uniformly from the port to the channels affecting both thermal and hydraulic performance of plate heat exchangers. Therefore for a better design there is a need for better knowledge of flow distribution and the effect of this distribution on thermal performance.

In literature considerably less attention has been paid to flow maldistribution in plate heat exchangers. Some studies are available, for the case of uniform flow distribution on thermal performance of plate heat exchangers for steady state. Datta and Majumdar [1] extended the method of Majumdar [2] for calculating the flow distributions in U and Z type manifolds. Although their results compared well with experimental data of Bajura and Jones [3], the

* Corresponding author. Tel.: +91 44 2257 4655; fax: +91 44 2257 0545.
E-mail address: skdas@iitm.ac.in (S.K. Das).

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Nomenclature

a	cross-sectional area of the intake, m^2
A	heat transfer area for effective plate, m^2
A_c	channel free flow area, m^2
A_p	port free flow area, m^2
C	heat capacity of fluid, $J\,K^{-1}$
C_p	isobaric specific heat of the fluid, $J\,kg^{-1}\,K^{-1}$
C_w	heat capacity of the plate material, $J\,K^{-1}$
D	channel hydraulic diameter, m^2
h	heat transfer coefficient, $W\,m^{-2}\,K^{-1}$
L	fluid flow length in a channel, m
$\dot{m}$	mass flow rate, $kg\,s^{-1}$
m^2	flow distribution parameter
n	exponent of Re in equation
N	number of channels
NTU	number of transfer units $= (UA)/(\dot{m}C_p)_{\min}$
Nu	Nusselt number
Pr	Prandtl number
Re	Reynolds number$= u_i d_e/v$
r_h	ratio of velocity
r_u	ratio of heat transfer coefficients
R_τ	ratio of residential times, $\frac{\tau_{ra2}}{\tau_{ra1}}$
R_c	wall heat capacity ratio, $\frac{C_W}{C_1}$
R_N	$U_{a(2)}/U_{a(1)}$
R_2	capacity rate ratio
t	non-dimensional temperature of fluids $= \frac{T-T_{1,in}}{T_{2,in}-T_{1,in}}$
T	temperature of fluid, K
$T_{1,in}$	inlet temperature of the fluid (1), K
$T_{1,out}$	outlet temperature of the fluid (1), K
$T_{2,\ in}$	inlet temperature of the fluid (2), K
U_c	velocity of fluid, m/s
$U_{1(2)}$	$(hA/\dot{w})_{1(2)}$
u	dimensionless velocity of fluid in the channel, u_i/U
u_c	dimensionless channel velocity, $\frac{U_c}{U_m}$
u_i	channel velocity, m/s
U_m	mean channel velocity, m/s
v_c	dimensionless volume flow rate in the channels
$\dot{w}_i$	thermal capacity rates of the fluids in channels, $W\,K^{-1}$
W	velocity in the intake conduit
W_c	axial component of the intake flow velocity at the channel inlet
X	space coordinate for the fluid flow, m
x	non-dimensional flow path coordinate $= x/L$
z_c	dimensionless flow axial coordinate in the port, $\frac{z}{L_p}$
z	dimensionless time

Greek symbols

v	kinematic viscosity of the fluid, $m^2\,s^{-1}$
ζ	channel hydraulic resistance = friction factor X (D/L)
τ	time, s
τ_r	characteristic time, $C_1/\dot{w}$, s
ε	effectiveness of plate heat exchanger $= \frac{(\dot{m}C_p)_1(T_{1,out}-T_{1,in})}{(\dot{m}C_p)_{\min}(T_{2,in}-T_{1,in})}$

Subscripts

a	average
1	fluid in odd channel
2	fluid in even channel
ch	channel
i	ith channel
j	jth node in the channel
	uniform case of uniform flow distribution amongst channels
W	plate

prediction procedure still needed empirical momentum-transport correction coefficients in the header momentum equations to account for the turning effect. They have analysed the effect of unequal distribution of fluid inside the channels using a numerical technique. An analytical study has been made by Bassiouny and Martin [4] to calculate the axial velocity and pressure distribution in both the intake and exhaust conduits of plate heat exchangers, the flow distribution in the channels between the plates and the total pressure drop. From the analysis a general flow characteristic parameter (m^2) has been derived from the mass and momentum formulation for inlet and exit port flows for U and Z type plate heat exchangers, which determines the flow behaviour. Prabhakara Rao and Das [5] studied the effect of flow maldistribution from channel to channel on thermal performance of PHE comprehensively using the Bassiouny and Martin's [4] model for flow distribution.

There is a limited number of studies on the transient analysis of plate heat exchangers. Khan et al. [6] presented theoretical and experimental analyses of the dynamic characteristics of plate heat exchangers. The first and second order models with dead time was proposed and checked against results obtained by experimental sinusoidal and pulse testing. Das and Roetzel [7] presented an extensive analysis for plate heat exchangers, which takes into consideration the deviation from ideal plug flow through an axial dispersion model to predict the response due to temperature transients. The results lead to conclusion that the effect of flow maldistribution and phase lag plays an important role in the response. Das et al. [8] carried out experiments on the transient behaviour of two welded plate heat exchangers with identical constructions but different umber of the plates, under different operating conditions. The model takes the effects of both back mixing and flow maldistribution into account by introducing a dispersion term in the energy equation. To predict the transient response of multipass plate heat exchanger Das and Murugeasan [9] analysed 1-2 and 2-2 pass PHEs with the axial heat dispersion model in the fluid which takes deviation from ideal plug flow into consideration. However, Roetzel and Na Ranong [10] showed that maldistribution should be treated separately and it is more appropriate to assign only back mixing to axial dispersion. Using this concept Srihari et al. [11] carried out transient study due to step increase in inlet temperature using both axial dispersion and flow maldistribution.

Compared to temperature transients, number of studies on flow transients is fewer. A versatile and efficient method was developed by Xuan and Roetzel [12] for predicting dynamic performances of parallel and counter flow heat exchangers subjected to arbitrary temperature variations and step flow disturbances, including the effect of flow maldistribution (lumped into the dispersion term) and influence of the heat capacities of both the fluids, shell wall and tube bank as well as non zero initial temperatures. Abdelghani-Idrissi et al. [13] investigated temperature transient response along a tubular counter flow heat exchanger, when the mass flow rate is subject to sudden change. Dynamic behaviour is approximated by a first order response with the time constant. The hot fluid subjected to the flow rate steps presents a decreasing linear time constant. Transient temperature response of cross flow heat exchangers having finite wall capacitance with both fluids unmixed was investigated numerically by Mishra et al. [14], for perturbations provided in both temperature and flow.

Thus, the above survey of literature shows that the accurate modeling of the effect of flow transient in plate heat exchangers in presence of maldistribution has not yet been carried out which is supported by proper experimental studies. The present work aims to develop a model which predicts the transient responses of the U type plate heat exchanger under step change in inlet flow rates. In the present analysis port to channel maldistribution is modeled accurately in order to indicate its effect on the transient performance of the plate heat exchanger. Experiments have been carried out showing the responses of the plate heat exchangers, subjected to step changes in the inlet flow rates and thus the theoretical model developed have been validated against experimental results.

2. Mathematical formulation

In the present model first steady state temperature distribution of a U type plate heat exchanger is obtained. This becomes the initial condition for the flow transient in the heat exchanger. Further transient response of the plate heat exchanger is analysed when mass flow rate at inlet is subjected to sudden change. The plate heat exchanger is thermally modelled with unequal flow in channels taking the distribution as suggested by Baussiouny and Martin [4]. To formulate the governing equations following assumptions are made.

1. The flow distribution inside the channel is taken to be uniform giving a ‘plug flow’ of fluid inside each channel. It can be justified as the channel gap is small.
2. The flow maldistribution from channel to channel has been taken into account through the Baussiouny model [4].
3. As a result of flow maldistribution, heat transfer coefficient is considered to be variable from channel to channel.
4. The plates are considered to be thin enough so that axial conduction in them in the direction of flow can be neglected.
5. The thermophysical properties of the fluids are considered to be independent of temperature and pressure.
6. Heat transfer is assumed to take place only between the channels and not between the channel and the ports or through the seals and gaskets.
7. The heat exchanger is assumed to be insulated from the surroundings.
8. The projected area of plate is taken as heat transfer area.

Above mentioned assumptions are taken into consideration and a mathematical model is developed. The solution of model can be split up into two steps. First step is to attain temperatures distributions across the heat exchanger while the heat exchanger is in steady state. Subsequently this state is taken as initial condition of the next step of the model, which is the flow transient case. In the transient case step flow variation of the inlet fluids (either increase or decrease of flow) is taken into account, which leads to another steady state. For both steps of the model a small control volume of fluid inside the channel and control volumes of solid plate (shown in Fig. 1) are taken into consideration. Energy balances across control volumes (Fig. 1) is carried out for the transient case. It gives the following partial differential equations: For fluid 1,

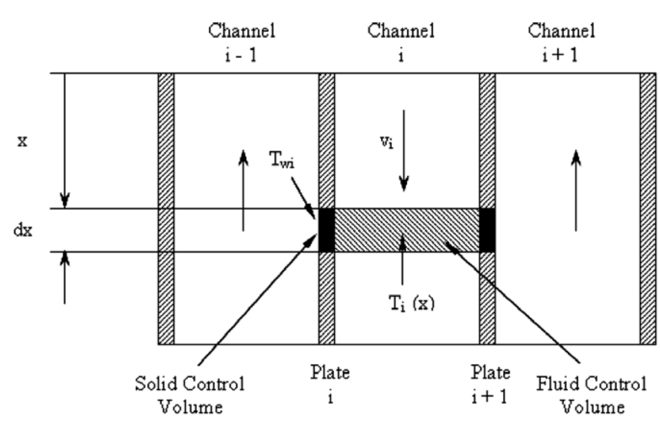

Fig. 1. Control volume of the fluid inside the channel and solid control volume in the wall.

$$\frac{C_1}{L}\frac{\partial T_i}{\partial \tau} = -(-1)^{i-1}\dot{w}_i\frac{\partial T_i}{\partial X} + \frac{(hA)_i}{2L}(T_{Wi} + T_{Wi+1} - 2T_i) \quad (\text{for } i = 1,3,5,\ldots,N) \tag{1}$$

For fluid 2,

$$\frac{C_2}{L}\frac{\partial T_i}{\partial \tau} = -(-1)^{i-1}\dot{w}_i\frac{\partial T_i}{\partial X} + \frac{(hA)_i}{2L}(T_{Wi} + T_{Wi+1} - 2T_i) \quad (\text{for } i = 2,4,6,\ldots,N-1) \tag{2}$$

For the solid wall, the first and last plates have different governing equations than the intermediate plates since they are in contact with only one fluid. Governing equations for the plates are:

For intermediate plates:

$$\frac{C_W}{L}\frac{\partial T_{Wi}}{\partial \tau} = \frac{(hA)_{i-1}}{2L}(T_{i-1} - T_{Wi}) + \frac{(hA)_i}{2L}(T_i - T_{Wi}) \quad (\text{for } i = 2,3,4,\ldots,N) \tag{3}$$

For first plate:

$$\frac{C_W}{L}\frac{\partial T_{W1}}{\partial \tau} = \frac{(hA)_1}{2L}(T_1 - T_{W1}) \quad (\text{for plate } 1) \tag{4}$$

For last plate:

$$\frac{C_W}{L}\frac{\partial T_{W,N+1}}{\partial \tau} = \frac{(hA)_N}{2L}(T_N - T_{W,N+1}) \quad (\text{for plate } N+1) \tag{5}$$

Above governing equations are non-dimensionalized with the help of following non-dimensional terms:

$$x = \frac{X}{L}, \quad t = \frac{T - T_{1,\text{in}}}{T_{2,\text{in}} - T_{1,\text{in}}}, \quad \text{NTU}_i = \frac{hA}{2(\dot{m}C_p)}$$

$$r_v(i) = \frac{u_{\text{uniform}}}{u_i}, \quad r_h(i) = \frac{h_i}{h_{\text{uniform}}} = \left(\frac{u_i}{u_{\text{uniform}}}\right)^n$$

$$\tau_{\text{ra}1} = \frac{C_1}{\dot{w}_{\text{a}1}}, \quad \tau_{\text{ra}2} = \frac{C_2}{\dot{w}_{\text{a}2}}, \quad z = \frac{\tau}{\tau_{\text{ra}}}, \quad R_\tau = \frac{\tau_{\text{ra}2}}{\tau_{\text{ra}1}}$$

$$R_2 = \frac{\dot{w}_{\text{a}2}}{\dot{w}_{\text{a}1}}, \quad R_C = \frac{C_W}{C_1}, \quad R_N = \frac{U_{\text{a}2}}{U_{\text{a}1}}$$

where $U_a = \frac{hA}{\dot{w}_a}$.

For dependence of the heat transfer coefficient on flow rate, the usual heat transfer correlation is given as

$$Nu = CRe^n Pr^r$$

In the present case the constant and exponents are taken as $C = 0.218$, $n = 0.65$ and $r = 0.33$, as found for the same heat exchanger (used for the experimental validation) by Prabhakara Rao et al. [15].

Prandtl number does not affect 'h' since property variation with temperature is not taken into account in model. NTU in each channel is given as

$$\text{NTU}_i = U_{\text{a}1} \cdot r_h(i) \cdot r_v(i).$$

Dimensionless forms of the Eqs. (1)–(5) are following: For fluids,

$$\frac{(R_\tau)^{m_i+1}}{R_{Wi}}\frac{\partial t_i}{\partial z} = -(-1)^{i-1}\frac{\partial t_i}{\partial x} + \frac{\text{NTU}_i}{2}(R_N)^{m_i+1}(t_{Wi} + t_{Wi+1} - 2t_i) \tag{6}$$

For plates,

(a) For intermediate plates:

$$R_C\frac{\partial t_{Wi}}{\partial z} = \frac{\text{NTU}_{i-1}}{2}(R_N R_2)^{m_i}(t_{i-1} - t_{Wi}) + \frac{\text{NTU}_{i-1}}{2}(R_N R_2)^{m_{i+1}}(t_i - t_{Wi}) \tag{7}$$

(b) For first plate:

$$R_C\frac{\partial t_{W1}}{\partial z} = \frac{\text{NTU}_{i-1}}{2}(R_N R_2)^{m_i}(t_1 - t_{W1}) \tag{8}$$

(c) For last plate:

$$R_C\frac{\partial t_{W,N+1}}{\partial z} = \frac{\text{NTU}_N}{2}(R_N R_2)^{m_{i+1}}(t_N - t_{W,N+1}) \tag{9}$$

Here variable m_i is introduced such that, for various values of number of channel i, ($i = 1,2,3,4,\ldots$) it is defined as $0,1,0,1,\ldots$ respectively.

2.1. Boundary conditions

For the above governing equations, the boundary conditions can be set as,

$$\left.\begin{array}{ll} \text{at } x = 0, t_i = 0 & \text{for } i = 1,3,5,\ldots,N \\ \text{at } x = 1, t_i = 1 & \text{for } i = 2,4,6,\ldots,N-1 \end{array}\right\} \tag{10}$$

2.2. Solution procedure for steady state

Bassiouny and Martin [4] gave a proper distribution of fluid in the channels from the port. They presented the flow formulation for normal geometries where the volumetric flow rate decreases along the flow direction in the entrance port for U-type flow arrangements as

$$u_c = \frac{m\cosh m(1 - z_c)}{\sinh m} \tag{11}$$

where m^2 is the flow distribution parameter giver by $\frac{1}{\zeta}\left(\frac{NA_c}{A_p}\right)^2$.

Above depicted velocity distributions are applied to the Eqs. (6)–(9). For steady state analysis, the time dependent terms of these equations on the left hand side are dropped which converts the system of partial differential equations to ordinary differential equations. This system of ordinary differential equations of first order with boundary values are solved numerically using finite difference technique. Each channel (say $i = 1,2,3,\ldots,N$) is distributed in k number of nodes ($j = 1,2,3,\ldots,k$). In the present analysis thirty numbers of nodes in each channel has been taken which is found to be sufficient for grid independence of result. So a nodal network of $N \times k$ is aggregated as shown in Fig. 2.

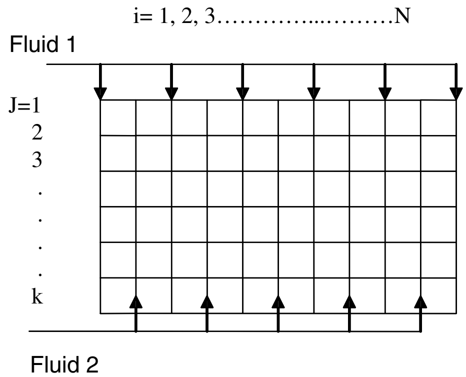

Fig. 2. The grid structure used in the finite difference analysis.

Temperatures at these discrete points are denoted as $t_{i,j}$. For the finite difference method Gauss-Jordan scheme is used for point by point iteration. The method is stopped when specified convergence criterion (0.0001) for dimensionless temperature is satisfied.

2.3. Initial conditions

To solve the above differential Eqs. (6)–(9) in the transient regime the initial conditions can be set as

At $z = 0$,

$$t_i|_z = t_i \quad \text{for } i = 1, 2, 3, \ldots, N$$
$$(t_i \text{ is obtained by solving Eq. (6) in steady state}) \tag{12}$$
$$t_w|_z = t_w \quad \text{for } i = 1, 2, 3, \ldots, N+1$$
$$(t_w \text{ is obtained from Eqs. (7)–(9) in steady state}) \tag{13}$$

at $z = 0^-$, $R_2 = 1$ and at $z = 0^+$, $R_2 = R_2'$ (R_2' is the heat capacity rate ratio after the introduction of flow perturbation).

2.4. Solution procedure for transient state

System of partial differential Eqs. (6)–(9) with boundary and initial conditions as well as the change in heat capacity rate ratio indicating step change in one of the flow rates can be solved, using numerical technique. Explicit scheme with respect to time using finite difference approach has been used here to solve these partial differential equations.

Outlet temperatures of the heat exchanger can be calculated by assuming the adiabatic mixing of the fluid. It can be shown as

$$T_{\text{out}} = \frac{\sum m_i T_i}{\sum m_i} \quad i = 1, 3, \ldots, N \quad \text{for cold fluid} \tag{14}$$

$$T_{\text{out}} = \frac{\sum m_i T_i}{\sum m_i} \quad i = 2, 4, \ldots, N-1 \quad \text{for hot fluid} \tag{15}$$

The convergence of the solution has been checked by varying the number of the space grids and size of the time step. For the present analysis numbers of the nodes in each channel has been taken as 30 and the dimensionless time step is 0.05. The solution gives the one-dimensional temperature distribution for each channel fluid as well as each plate. The method is stopped when specified convergence criterion (0.0001) of dimensionless temperature is satisfied.

3. Experiments

3.1. Experimental setup

The test plate heat exchanger is manufactured by Alfa Laval company and the plates are made of stainless steel with Nitrile rubber gaskets. It contains 40 corrugated stainless steel plates and their geometrical features are shown in Fig. 3a. The plate heat exchanger is arranged for U-type flow configuration.

Schematic diagram of the transient experimental test facility is shown in Fig. 3b. The experimental setup has been built such that it is able to produce the sudden rise in hot fluid temperature and measure the responses at the outlet of both the fluids. Two circuits: cold water circuit and hot water circuit as shown in Fig. 3b are made to circulate both the fluids through the heat exchanger. Cold fluid is supplied to the heat exchanger where it receives heat from the hot fluid across the plates. It is then sent to the cooling tower where the temperature comes down to the inlet condition. The hot fluid is sent to the plate heat exchanger and fed back to the hot water tank which is kept at a constant temperature using a temperature controller. In addition, by-pass lines are connected in the main pipelines because, during the initial period of the experiments cold water is allowed to flow in both cold and hot channels of the exchanger. Electro-pneumatic valves are provided to obtain the required flow directions during the experiments. These are ON/OFF type valves operated by electrical switches. Hot water is obtained by using the submerged electrical heaters of 42 kW capacity provided to the hot water tank. The flow rates can be adjusted by the control valves. Flow rates are measured with the orifice flow meters which are located before the fluids enter into the heat exchanger. Both hot and cold side orifice meters have been calibrated and compared with standard ASME charts. T-type thermocouples are connected at the entrance and exit pipelines of both cold and hot sides of the exchanger to measure the exchanger response for every one second. These responses are recorded with the help of the data acquisition system (HP 34970A). These thermocouples are grounded type, which are suitable for fast response. Fast response is required for the transient temperature measurement to capture the changes in time domain accurately. All the thermocouples are calibrated over entire range of interest, using a precision thermometer and a constant temperature bath. Least count of the thermocouple is 0.1 °C. It has been estimated that the time constant is 150 ms. To get flow disturbances two spring operated butterfly valves are used.

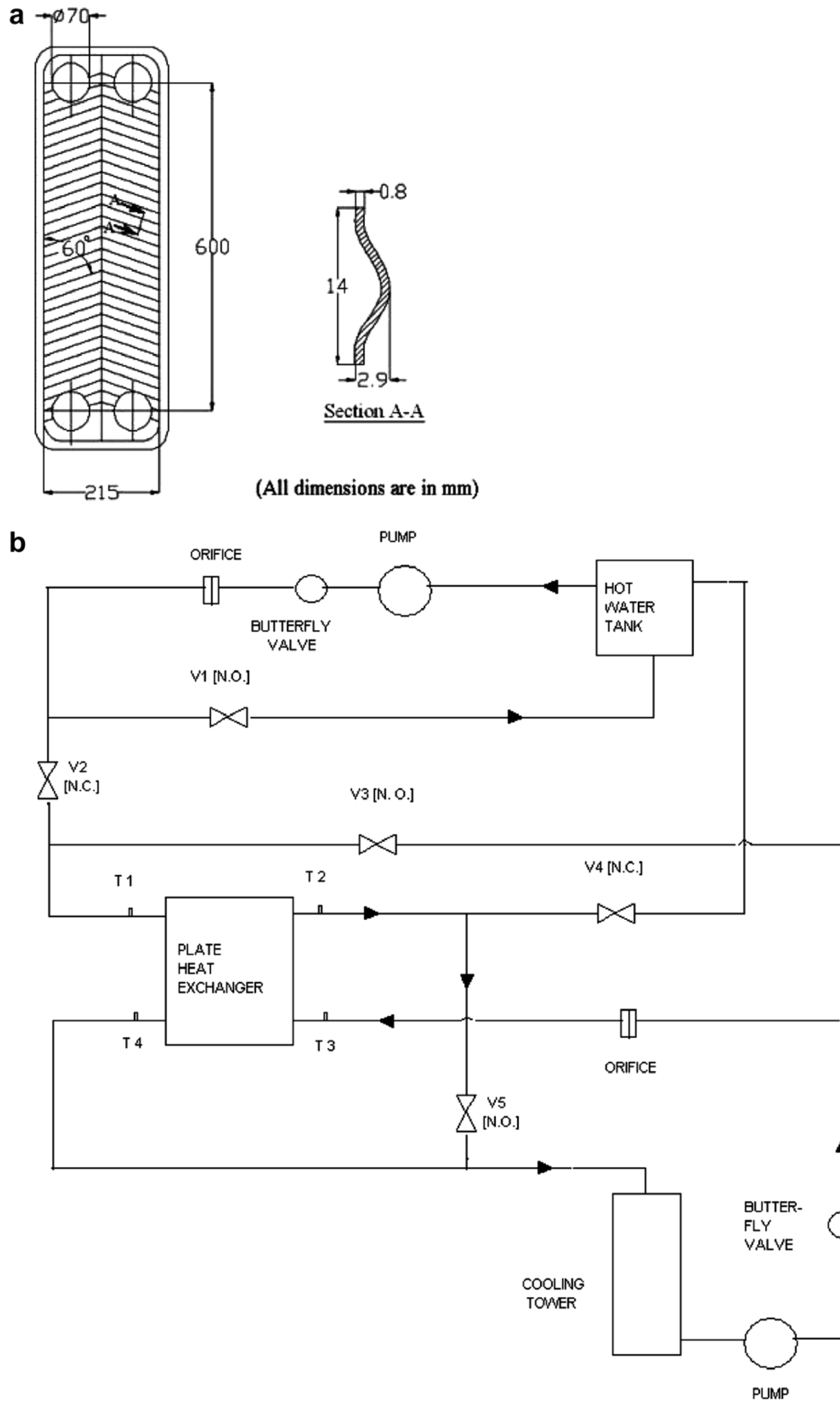

Fig. 3. Flow diagram of the experimental facility.

3.2. Experimental procedures

The test plate heat exchanger is used to conduct the transient experiments for U-type flow configurations. The heat exchanger has 31 channels out of which 16 carries cold fluid and 15 carries hot fluid. Cold fluid is always allowed to flow on the side having more number of channels to minimize the heat loss from the end plates. The experimental test setup is designed to obtain the step rise in hot fluid temperature. Electro-pneumatic valves are connected to the pipe lines as per the requirement and as per the valve position i.e. normally open (NO)/normally closed (NC).

At the beginning of the experiments, valves V2 and V4 are closed and valves V1, V3, and V5 are opened as shown in Fig. 3b. Initially hot water flows in bypass line and back to the hot water tank as the valve V2 is closed and valve V1 is opened. Cold water flows in cold water circuit and some part of it is bypassed to hot water inlet pipe as the valve V3 is opened. At this condition only cold water circulates in both the sides and both the streams flow back to the cooling tower as the valve V5 is open. Then the recording of fluid temperatures is started before the hot water is sent into the hot channels. Because cold water only flows in both the sides heat transfer will not takes place in the heat exchanger. Initially all the fluid temperatures have the same value as the cold inlet temperature. Electro-pneumatic valves are connected to single main switch which is switched ON after the hot water attains the set steady temperature. As the valve V2 is opened and valves V1 and V3 are closed, hot water pushes the existing cold water into the hot water line at the entrance of the exchanger. It produces the sudden rise in hot inlet temperature and then heat transfer takes place inside the heat exchanger. As the valve V4 is opened and valve V5 closed the hot water coming from the exchanger goes back to the hot water tank. The accompanying variation of flow rates on both sides during this starting from cold state does not affect the test because heat transfer takes place only after the step change in temperature takes place. The inlet and outlet temperatures of the fluids are recorded till the steady state is reached. The temperatures on cold outlet and hot outlet sides gradually increase and then remain constant which is the steady state condition. Transient response of the particular configuration for the given flow rate is observed by analysing the recorded data of temperatures at regular intervals of time. The procedure has been repeated for different flow rates. Now, after getting the steady state, experiment for flow transient was carried out. Inlet flow rates were changed through the butterfly valves. Flow rates were measured before and after flow perturbation.

3.3. Data reduction and error estimates

Logarithmic mean temperature difference for counter flow plate heat exchanger was calculated as

$$\Delta T_{\mathrm{lm}} = \frac{\Delta T_1 - \Delta T_1}{\ln \frac{\Delta T_1}{\Delta T_1}} \tag{16}$$

where $\Delta T_1 = T_{h,\mathrm{in}} - T_{\mathrm{c,o}}$ and $\Delta T_2 = T_{\mathrm{h,o}} - T_{\mathrm{c,in}}$.

Reynolds number was calculated based on hydraulic diameter (for plate heat exchangers it is $2b$ as $H \gg b$).

Based on the projected area suggested by Shah and Wanniarachchhi [16] Reynolds number was calculated as

$$Re = \frac{U_{\mathrm{c}}(2b)}{\nu} \tag{17}$$

Fluid properties are evaluated at mean temperature given by

$$T_{\mathrm{m}} = \frac{T_{\mathrm{c,in}} - T_{\mathrm{c,o}}}{2} \quad \text{or} \quad \frac{T_{\mathrm{h,in}} - T_{\mathrm{h,o}}}{2}$$

The over all heat transfer coefficient was calculated as

$$U = \frac{mc_p(T_{\mathrm{c,o}} - T_{\mathrm{c,in}})}{A\Delta T_{\mathrm{lm}}} \tag{18}$$

Using this value of U and from the measured value of mass flow rates, the experimental value of NTU was calculated as

$$\mathrm{NTU} = \frac{UA}{(mC_p)_{\min}} \tag{19}$$

Residence time for any fluid circulating through N numbers of channels with a volume flow rate of V can be calculated as

$$\tau_{\mathrm{r}} = \frac{NA_{\mathrm{c}}L}{V}$$

The over all energy balance between hot and cold fluids as follows

$$Q = (mc_p)_{\mathrm{h}}(T_{\mathrm{h,in}} - T_{\mathrm{h,o}}) = (mc_p)_{\mathrm{c}}(T_{\mathrm{c,o}} - T_{\mathrm{c,in}}) \tag{20}$$

Effectiveness ε was calculated as

$$\varepsilon = \frac{(mc_p)_{\mathrm{c}}(T_{\mathrm{c,o}} - T_{\mathrm{c,in}})}{(mc_p)_{\min}(T_{\mathrm{h,in}} - T_{\mathrm{c,in}})} \tag{21}$$

The uncertainty analysis for all the measured and derived quantities has been carried out. The uncertainty in the measurement of temperature was ±0.5%. Time constant (time required to reach 63.2% of steady state temperature) of the thermocouples is around 100 ms. Error in pressure measurement across the orifice plates was ±2%. Error in discharge coefficient was ±0.5%. Based on these values of measured quantities maximum uncertainty in values of mass flow rate and Reynolds number were calculated as ±5.107% and ±1.007% respectively.

3.4. Results and discussion

Experiments have been carried out for flow transients in U-type configuration of plate heat exchangers. Transient response of plate heat exchanger with 31 number of channels, is shown in Fig. 4. It can be observed that at the beginning all the temperatures (cold and hot inlets, cold and hot exit temperatures) across the PHE are same. In all the channels fluid with total flow rate of 1.318 l/s ($Re = 1244$) is flowing. It can be seen that later there is increase in hot inlet, cold and hot outlet temperatures, after introducing hot fluid in hot channels with an equal flow rate. As depicted in the figure, PHE with specific heat capacity rate ratio of unity reaches a steady state with time, which is disturbed further by step increment in the hot inlet

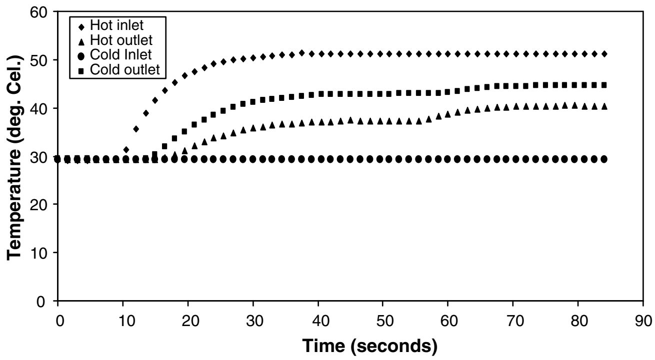

Fig. 4. Responses of temperatures when hot fluid flow rate is increased (at time = 50 s) from 1.318 to 1.86 l/s. Cold fluid is at 1.318 l/s. $R = 1.41$.

flow rate. Hot flow rate is increased to 1.86 l/s ($Re = 1756$). As it is shown in this figure, it leads to once again rise in both the outlet temperatures. Both outlet temperatures reach to a new steady with time, as shown in the figure.

In Fig. 5 all channels are initially supplied cold water with a flow rate of 0.93 l/s ($Re = 877$). The initial part of the figure is similar to that described in Fig. 4. But this time hot water flow rate is kept constant and cold water flow rate is increased to 1.318 l/s ($Re = 1244$). It can be clearly observed that in the later part of the figure that represents the flow transient, both the outlet temperatures are decreased. They go on decreasing till a new steady state is achieved. However during both the steady states energy balance is checked and is found to agree within 2%.

To check the validity of the numerical scheme the results of the numerical model have been compared with the experiments carried out. Fig. 6 shows the comparison of ε–NTU curve for steady state operation of the plate heat exchanger. These experiments have been carried out for 31 numbers of the channels (16 cold fluid and 15 hot fluid channels) and the NTU from the experiments were calculated using Eqs. (18) and (19). It can be observed from the figure that there is good agreement between model

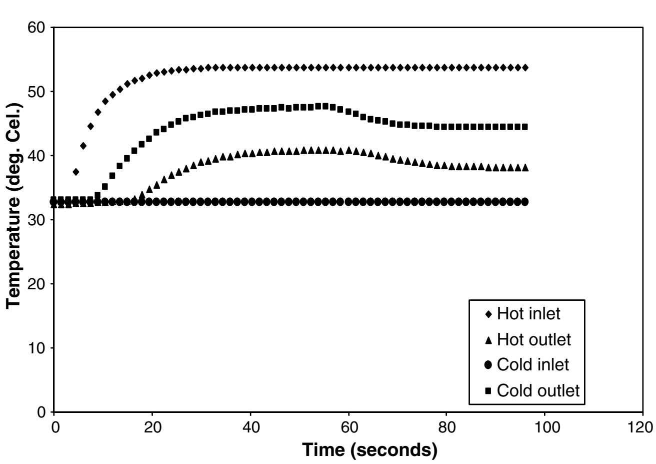

Fig. 5. Responses of temperatures when cold fluid flow rate is increased (at time = 50 s) from 0.93 to 1.318 l/s. Hot fluid is at 0.93 l/s. $R = 0.7$.

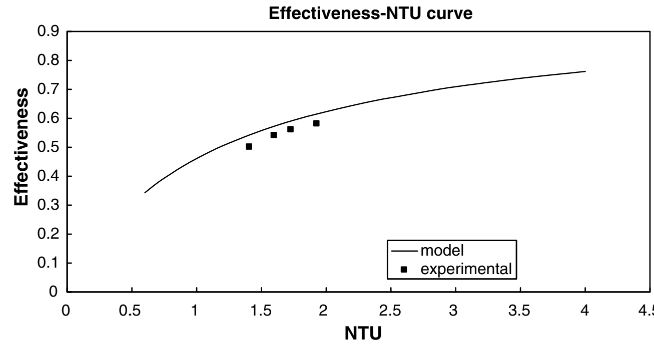

Fig. 6. Comparison of model and experimental results for steady state.

and experimental result. For flow transient case also, the comparisons between the calculated and measured temperature histories have been carried out. Figs. 7 and 8 show temperature profiles of PHE for flow transient with step disturbance in inlet flow rates. It is important to observe that our focus is flow transient and hence the theoretical model was developed with the steady state as initial condition. Hence, during flow transient only the later part of the

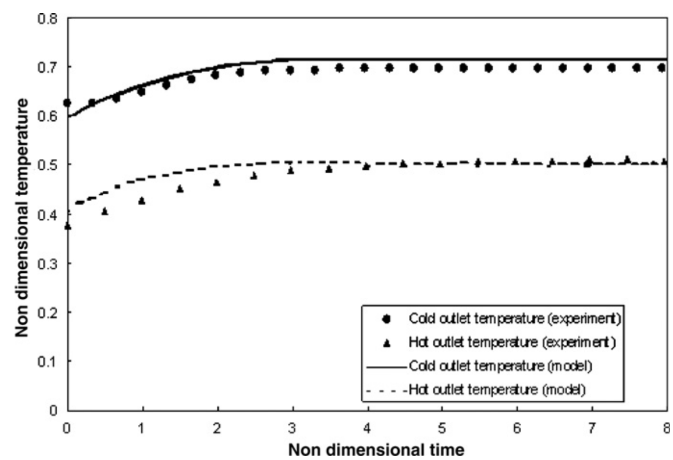

Fig. 7. Comparison of model with experimental results. For $R = 1.41$, NTU = 1.82, $N = 31$.

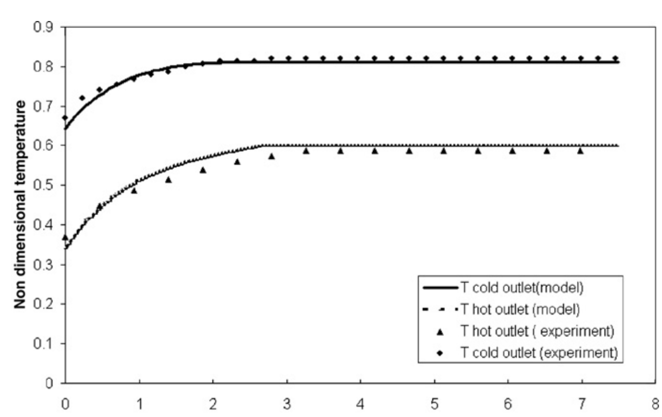

Fig. 8. Comparison of model with experimental result. For $R = 2$, NTU = 1.91, $N = 31$.

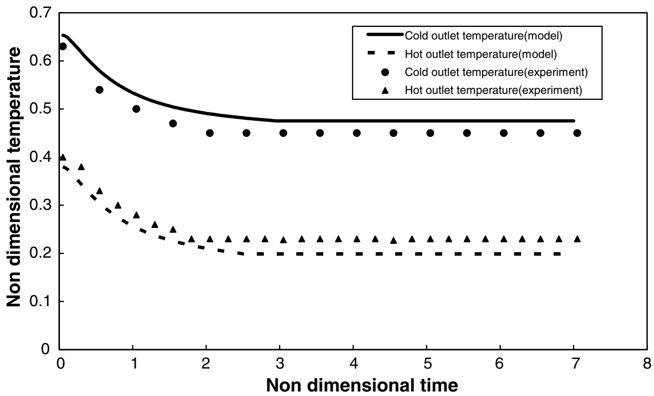

Fig. 9. Comparison of model with experimental result. For $R = 0.57$, NTU = 2.4, $N = 31$.

transient experiment curves (Figs. 4 and 5) has been used for comparison. For both the cases hot inlet flow rates are suddenly increased in order to achieve flow perturbations. Experimental results are expressed in non-dimensional terms in order to compare it with numerical model. In Fig. 7 the temperature response for the flow transient when the heat capacity rate ratio has been increased from 1 to 1.41 by increasing the hot fluid flow rate. For Fig. 8 the same has been increased from 1 to 2. It is clearly depicted in the figures that such cases lead to rise in outlet temperatures. Fig. 9 shows the comparison when cold inlet flow rate is increased and hot flow rate is kept constant giving the heat capacity rate ratio of 0.57. In all the cases even with a number of simplified assumptions the present model is found to replicate the experimental results quite accurately. The small deviation between the model and experiment can be attributed to the heat loss through the edges, end plated and gaskets since the mismatch was found to be of the same order as the difference in energy balance of the two sides in the experiment.

With this validation further parametric studies have been carried out with the model developed here. Fig. 10 shows the response of outlet temperatures of PHE subjected to flow transient achieved through different routes U-type PHE with 31 number of channels, NTU = 2 has been analysed here. The PHE initially running under steady state with a heat capacity rate ratio of unity is suddenly subject to step flow changes at inlet. Non-dimensional cold outlet and hot outlet temperatures variations with non-dimensional time are shown in the figure. The figure corresponds to the variations when heat capacity rate ratio is made double. It is clearly indicated that both fluid stream's temperatures increase. Heat capacity rate ratio of 2 can be achieved in two ways either by increasing hot inlet flow rate or by decreasing cold water flow rate. Figure shows both the possibilities. It is interesting to note that both of these possibilities do not yield identical responses in the transient regime although the initial and final steady states are identical. In one case the temperature rise in truly exponential whereas in the other case it is close to linear. This can be attributed to different ranges of flow rates of the fluids and the resulting heat transfer coefficients of the channels due to flow maldistribution under these two different conditions of flow.

In Fig. 11 it is shown that both the temperatures decrease when heat capacity rate ratio is made half (i.e., initial heat capacity rate ratio is changed from unity to 0.5). The initial states correspond to previous steady state which exactly satisfies the energy balance. When the cold fluid flow rate is raised, decrease in hot outlet temperature can be justified as the flow disturbance increases the heat removal capacity of the cold fluid. Alternatively heat capacity rate ratio can be made half by decreasing the hot fluid flow rate. Difference in the response can be justified since for both cases as heat transfer coefficient as well as residence times undergo significant changes under the two conditions.

So far in the discussion flow inside PHE is assumed as uniformly distributed or m^2 (maldistribution parameter) is taken as zero. Figs. 12–15 depict the transient performance of the PHE with maldistribution. For Figs. 12–14 heat capacity rate ratio is changed from unity to 2.0 for different values of NTUs. In Fig. 15 heat capacity rate ratio is made half. These figures clearly show the deterioration in the performance of the heat exchanger with the increase

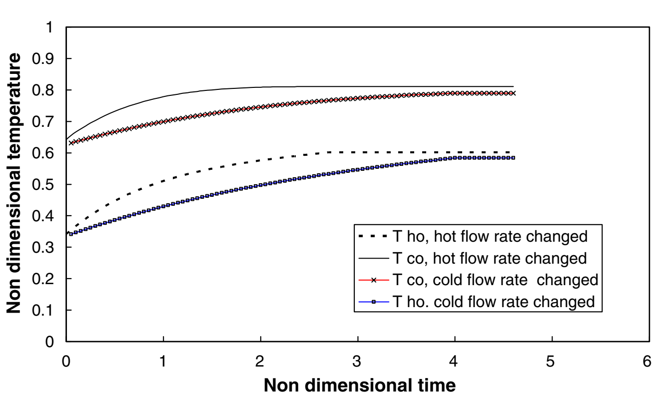

Fig. 10. Changes in outlet temperatures for $R = 2$ (for 31 channels, NTU = 2).

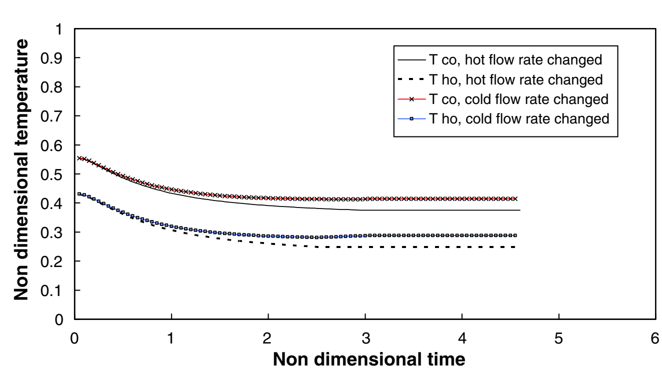

Fig. 11. Changes in outlet temperatures for $R = 0.5$ (for 31 channels, NTU = 2).

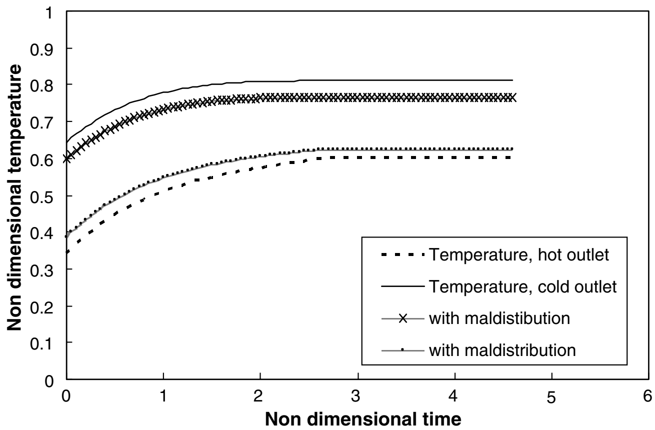

Fig. 12. Changes in outlet temperatures when hot flow rate is doubled and comparison with response considering maldistribution. (NTU = 2, $R = 2$, $N = 31$, $m^2 = 9$).

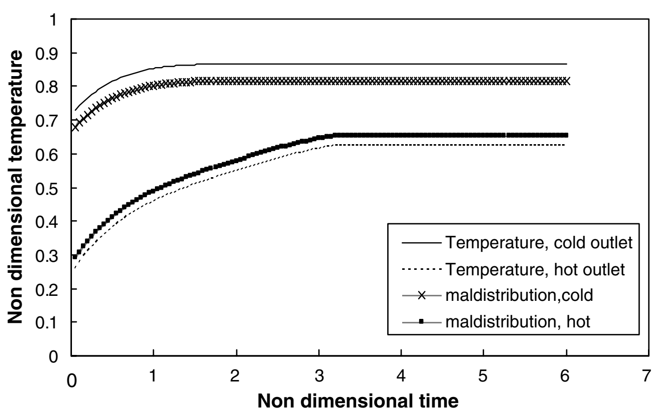

Fig. 13. Changes in outlet temperatures when hot flow rate is doubled and comparison with response considering maldistribution. (NTU = 3, $R = 2$, $N = 31$, $m^2 = 9$).

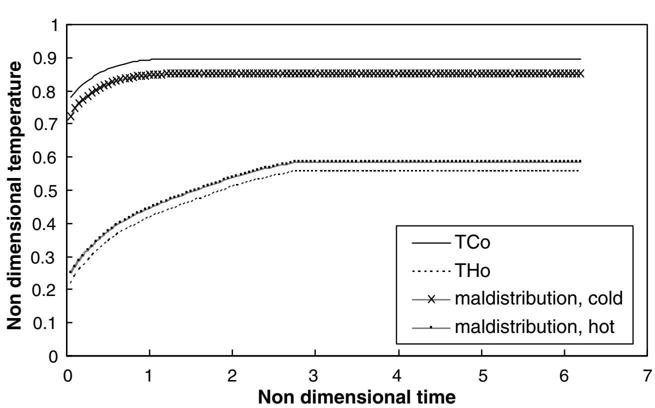

Fig. 14. Changes in outlet temperatures when hot flow rate is doubled and comparison with response considering maldistribution. (NTU = 4, $R = 2$, N = 31, $m^2 = 9$).

in flow maldistribution. Here it is important to mention that, in the initial state (steady state) itself there are differences of outlet fluid temperatures between the cases of

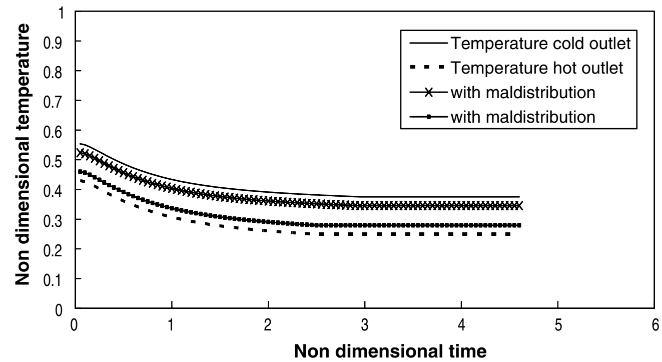

Fig. 15. Changes in outlet temperatures when hot flow rate is made half and comparison with response considering maldistribution. (NTU = 2, $R = 0.5$, N = 31, $m^2 = 9$).

$m^2 = 0$ and $m^2 = 9$. For all the cases, with and without maldistribution the energy balance is satisfied after getting new steady state. It can be observed from the Figs. 12–14 that by changing heat capacity rate ratio unity to 2 the rise in cold outlet temperature decreases with increasing NTU values.

4. Conclusions

In the present work, transient performance of the plate heat exchanger subjected to step flow disturbances has been investigated. A model has been presented, which predicts the performance of PHEs, subjected to perturbation in the inlet flow rates from a steady state operation. Port to channel flow maldistribution has been taken care, in the model. Finite difference technique is used to solve the governing equations in order to get the steady state temperature distribution across the PHE. Taking this temperature distribution as initial condition, governing equations for the flow transient have been solved. For solving the finite difference equations explicit time marching method is used. Results have been presented for a wide range of parameters like NTU and heat capacity rate ratio.

Experiments have been carried out for both steady and transient operations. For the transient case, temperature perturbation experiments were followed by flow transient experiments. Experiments have been executed for various possibilities of step flow transients. To check the validity of the numerical scheme the results of the numerical model have been compared to the experiments carried out. The comparison indicates an excellent agreement, which validates the suggested theoretical model for flow transient in plate heat exchangers. A further parametric study shows that it is not only the changes in heat capacity rate but the way this change is achieved (i.e., by changing the hot or the cold flow rate) makes a difference in response in the transient regime. The flow maldistribution is also found to influence the transient response considerably. The present analysis suggests the scope of the control system required to regulate the outlet temperatures of the plate heat exchangers subjected to dynamic state. Results also indicate the allowable time duration required for the control system to bring back a plate heat exchanger to steady state.

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