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The single-blow transient testing technique considering longitudinal core conduction and fluid dispersion

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Abstract

Single-blow transient testing technique has been widely used to measure the thermal performance of compact heat exchangers. For compact heat exchangers with short length the effect of longitudinal core conduction can usually not be neglected. Furthermore, the flow nonuniformity in a heat exchanger has also a significant influence on its temperature response. A new conduction/dispersion model for the single-blow transient testing technique is developed to include the effects of the longitudinal core conduction and fluid dispersion. Because the axial dispersion coefficient depends on the flow pattern in the heat exchanger which is usually unknown, both the heat transfer coefficient and the axial dispersion coefficient are determined with the whole curve matching simultaneously. The experiments are conducted in an open circuit wind tunnel. Comparison is made between the experimental results and data available in the literature. The software TAIHE (Transient Analysis In Heat Exchangers) developed by the authors is applied to the data analysis to evaluate heat transfer coefficients and axial dispersion coefficients. The results show that the pulse testing technique developed in the present investigation can easily be carried out and gives good evaluated results.

Keywords: Heat exchangers; Measurement techniques; Transient

1. Introduction

Single-blow transient testing technique is a very efficient experimental method for the determination of heat transfer coefficients in compact heat exchangers and regenerators. The experiment uses one fluid stream flowing through the heat transfer surface to be measured, which is considered as a uniformly distributed porous matrix. Heat exchange occurs only between the heat transfer surface to be measured and the fluid flowing through it. Therefore, with single-blow technique one can directly obtain the convective heat transfer coefficient between the heat transfer surface and the fluid. If the single-blow technique is applied to a real heat exchanger, the flow passage for the other fluid is considered to be adiabatic and should be evacuated. Usually it is filled with stationary air without violating the adiabatic boundary condition. The experimental system and procedure are relatively simple: a fluid flows steadily through the porous matrix

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Nomenclature

A_f	test core free-flow area, m^2
A_{cw}	average cross-sectional area of the solid wall, m^2
B	fluid/wall heat capacity ratio in the test core, $B = M_f c_f / M_w c_w$, dimensionless
c_f	heat capacity of fluid, J/kg K
c_w	heat capacity of solid wall, J/kg K
D	axial dispersion coefficient, W/m K
d_h	hydraulic diameter of flow passage, m
F	heat transfer area, m^2
j	Colburn j-factor, $j = Nu\, Re^{-1} Pr^{-1/3}$, dimensionless
k	heat conductivity, W/m K
K_w	dimensionless axial heat conductivity of solid wall, $K_w = A_{cw} k_w / \dot{m}_f c_f L$, dimensionless
L	length of test core, m
$\dot{m}_f$	mass flow rate, kg/s
M_f	mass of fluid in the test core, kg
M_w	mass of solid wall, kg
NTU	number of transfer units, $\mathrm{NTU} = \alpha F / \dot{m}_f c_f$, dimensionless
Pe	axial dispersive Peclet number, $Pe = \dot{m}_f c_f L / D A_f$, dimensionless
Pr	Prandtl number, dimensionless
Re	Reynolds number, $Re = (\dot{m}_f / A_f) d_h / \mu$, dimensionless
s	complex parameter in the Laplace transform, dimensionless
T	temperature, K
T_0	initial temperature, K
T_{ref}	reference temperature, K
x	coordinate, m
$\bar{x}$	dimensionless coordinate, $\bar{x} = x/L$, dimensionless

Greek symbols

α	heat transfer coefficient, W/m^2 K
θ	dimensionless temperature, $\theta = (T - T_0)/(T_{ref} - T_0)$, dimensionless
μ	dynamic viscosity, Ns/m^2
τ	time, s
$\bar{\tau}$	dimensionless time, $\bar{\tau} = (\dot{m}_f c_f / M_w c_w)\tau$, dimensionless

Subscripts

f	fluid
w	solid wall
in	inlet

and the local fluid temperature histories are measured continuously. These data are then compared with the theoretical result to obtain the corresponding heat transfer coefficient between the test core and the fluid. The accuracy of a transient testing technique depends on whether the mathematical model can correctly describe the real situation of the experiment and whether the outlet temperature response of the mathematical model is sensitive to the parameters to be determined. The efficiency of the transient testing technique depends on the conditions that the given inlet temperature variation of the fluid can be realized and that the dynamic characteristics of the outlet fluid temperature can be calculated rapidly.

The original mathematical model and its analytical solutions of the single-blow problem are provided in Refs. [1–4]. Schumann's solution was first used as the basis for a transient technique by Furnas in 1932 [5]. In his work, the outlet fluid temperature was measured and compared with Schumann's theoretical temperature curves. From the theoretical curve which fitted the experimental data in the best way the value of NTU was determined. Other dynamic characteristics such as the maximum slope and the first moment of exit fluid temperature were also developed to evaluate the heat transfer coefficient [6–10]. Liang and Yang [11] obtained the analytical temperature response solution for the exponential change of inlet fluid temperature. This solution is very useful because in most cases the actual profile of the inlet fluid temperature can be regarded as an exponential function. Based on the numerical results using finite differential method, Cai et al. [12] extended Liang and Yang's analysis with an empirical formula to include the effect of the longitudinal heat conduction in the test core. The influences of non-zero inlet temperature time constant, longitudinal core conduction, variable heat transfer coefficient and heat capacity error on the evaluated results were illustrated and discussed by Loehrke [13]. Mullisen and Loehrke [14] as well as Heggs and Burns [15] made brief reviews and discussions about the transient testing techniques. The main efforts which have been undertaken to make the theoretical model more conformable to the real experimental situation concerned two aspects: more reasonable inlet temperature description and the effect of longitudinal heat conduction in the wall.

In the previous investigations the essential assumption of the traditional methods is that the fluid flowing along the heat transfer surface has an overall uniform velocity distribution. If the construction of the exchanger is complex, severe maldistribution caused by bypassing, dead zones, recirculating currents and other nonuniformities of fluid flow may play an important part in the transient behaviour of the outlet temperature response. The maldistribution can be taken into account with the axial dispersion model which introduces an axial dispersion term in the energy equation of the fluid, as has been investigated theoretically and experimentally by Roetzel [16]. The results show that the longitudinal dispersion in exchangers has a recognizable influence on the transient outlet temperature response even when the dispersive Peclet number is as large as 50. Luo [17] pointed out that the transient testing techniques using the step or exponential inlet condition would give wrong results if flow maldistribution occurs in the exchanger test core and the corresponding axial dispersion coefficient is unknown. An extended testing technique which combines the single-blow transient testing technique and the temperature oscillation method was proposed by Roetzel and Luo [18,19], in which a sinusoidal inlet temperature variation was realized with a computer controlled compact electric heater. Recently, the pulse testing technique applying an arbitrary temperature pulse as inlet condition was developed by Zhou et al. [20] to determine the heat transfer coefficient and the axial dispersion coefficient. More details about the single-blow transient testing techniques can be found in the literature [21].

In the present paper, a new conduction/dispersion model is applied to the single-blow transient testing technique with the arbitrary pulse inlet condition. This model takes both the axial heat conduction in the wall material and the axial heat dispersion in the fluid into account and is therefore more suitable for short heat exchangers with non-uniformly distributed flow.

The software TAIHE (Transient Analysis In Heat Exchangers) has been developed by the authors to solve the mathematical model using the Laplace transform and numerical inverse algorithm and to evaluate the heat transfer coefficients and axial dispersion coefficients simultaneously.

2. Mathematical description

Consider a fluid flowing steadily through a heat transfer surface matrix, whose temperature at the entrance to the matrix is initially constant and then varies with time when the test begins. It is assumed that the heat transfer coefficient, specific heat capacities of the fluid and the wall material and the axial dispersion coefficient are constant; there is no heat conduction resistance perpendicular to the wall surface and no heat loss to the environment. The mathematical description can then be obtained from energy balances of the fluid and solid wall. The resulting system of dimensionless partial differential equations is summarized below together with the initial and boundary conditions:

$$B\frac{\partial \theta_\mathrm{f}}{\partial \bar{\tau}} + \frac{\partial \theta_\mathrm{f}}{\partial \bar{x}} - \frac{1}{Pe}\frac{\partial^2 \theta_\mathrm{f}}{\partial \bar{x}^2} = \mathrm{NTU}(\theta_\mathrm{w} - \theta_\mathrm{f}) \tag{1}$$

$$\frac{\partial \theta_\mathrm{w}}{\partial \bar{\tau}} - K_\mathrm{w}\frac{\partial^2 \theta_\mathrm{w}}{\partial \bar{x}^2} = \mathrm{NTU}(\theta_\mathrm{f} - \theta_\mathrm{w}) \tag{2}$$

$$\bar{\tau} = 0: \qquad \theta_\mathrm{f} = \theta_\mathrm{w} = 0 \tag{3}$$

$$\bar{x} = 0: \qquad \theta_\mathrm{f} - \frac{1}{Pe}\frac{\partial \theta_\mathrm{f}}{\partial \bar{x}} = \theta_{\mathrm{f_{in}}}(\bar{\tau}), \qquad \frac{\partial \theta_\mathrm{w}}{\partial \bar{x}} = 0 \tag{4}$$

$$\bar{x} = 1: \qquad \frac{\partial \theta_\mathrm{f}}{\partial \bar{x}} = \frac{\partial \theta_\mathrm{w}}{\partial \bar{x}} = 0 \tag{5}$$

where $\theta_{\mathrm{f_{in}}}(\bar{\tau})$ is an arbitrary function of time. The axial dispersive Peclet number in Eq. (1) represents the flow maldistribution which must either be measured or deduced from a similar system. The Danckwerts' boundary condition [22] (4) results from the conservation law of energy. It indicates an additional assumption that there is no axial dispersion in front of the inlet section and behind the outlet section of the apparatus being investigated.

Eqs. (1)–(5) together with the user-supplied inlet variation can be solved by means of Laplace transform, which yields an ordinary differential equation system in the Laplace plane as follows:

$$sB\tilde{\theta}_\mathrm{f} + \frac{\mathrm{d}\tilde{\theta}_\mathrm{f}}{\mathrm{d}\bar{x}} - \frac{1}{Pe}\frac{\mathrm{d}^2\tilde{\theta}_\mathrm{f}}{\mathrm{d}\bar{x}^2} = \mathrm{NTU}\big(\tilde{\theta}_\mathrm{w} - \tilde{\theta}_\mathrm{f}\big) \tag{6}$$

$$s\tilde{\theta}_\mathrm{w} - K_\mathrm{w}\frac{\mathrm{d}^2\tilde{\theta}_\mathrm{w}}{\mathrm{d}\bar{x}^2} = \mathrm{NTU}\big(\tilde{\theta}_\mathrm{f} - \tilde{\theta}_\mathrm{w}\big) \tag{7}$$

$$\bar{x} = 0: \qquad \tilde{\theta}_\mathrm{f} - \frac{1}{Pe}\frac{\mathrm{d}\tilde{\theta}_\mathrm{f}}{\mathrm{d}\bar{x}} = \tilde{\theta}_{\mathrm{f_{in}}}(s), \qquad \frac{\mathrm{d}\tilde{\theta}_\mathrm{w}}{\mathrm{d}\bar{x}} = 0 \tag{8}$$

$$\bar{x} = 1: \qquad \frac{\mathrm{d}\tilde{\theta}_\mathrm{f}}{\mathrm{d}\bar{x}} = \frac{\mathrm{d}\tilde{\theta}_\mathrm{w}}{\mathrm{d}\bar{x}} = 0 \tag{9}$$

In general, the solution of Eqs. (6)–(9) in the Laplace plane can be expressed in the matrix form

$$\mathbf{T} = \mathbf{U}e^{\Lambda\bar{x}}\mathbf{R} \tag{10}$$

where $\mathbf{T} = (\tilde{\theta}_\mathrm{f},\ \tilde{\theta}_\mathrm{w},\ \mathrm{d}\tilde{\theta}_\mathrm{f}/\mathrm{d}\bar{x},\ \mathrm{d}\tilde{\theta}_\mathrm{w}/\mathrm{d}\bar{x})^\mathrm{T}$, $\mathbf{U} = \{\boldsymbol{u}_1,\ \boldsymbol{u}_2,\ \boldsymbol{u}_3,\ \boldsymbol{u}_4\}$, $\Lambda = \mathrm{diag}\{\lambda_1,\ \lambda_2,\ \lambda_3,\ \lambda_4\}$. λ_i and $\boldsymbol{u}_i$ are the eigenvalues and the corresponding eigenvectors of the coefficient matrix of Eqs. (6) and (7),

$$A = \begin{bmatrix} 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \\ Pe(sB + \mathrm{NTU}) & -Pe\mathrm{NTU} & Pe & 0 \\ -\mathrm{NTU}/K_\mathrm{w} & (s + \mathrm{NTU})/K_\mathrm{w} & 0 & 0 \end{bmatrix}, \quad (11)$$

respectively. **R** is determined by the boundary conditions. The substitution of Eq. (10) into the boundary conditions (8) and (9) yields

$$\mathbf{WR} = \mathbf{F} \quad \text{or}\ \mathbf{R} = \mathbf{W}^{-1}\mathbf{F} \quad (12)$$

where

$$\mathbf{W} = \begin{bmatrix} u_{11} - u_{31}/Pe & u_{12} - u_{32}/Pe & u_{13} - u_{33}/Pe & u_{14} - u_{34}/Pe \\ u_{41} & u_{42} & u_{43} & u_{44} \\ u_{31}e^{\lambda_1} & u_{32}e^{\lambda_2} & u_{33}e^{\lambda_3} & u_{34}e^{\lambda_4} \\ u_{41}e^{\lambda_1} & u_{42}e^{\lambda_2} & u_{43}e^{\lambda_3} & u_{44}e^{\lambda_4} \end{bmatrix}$$

$$\mathbf{F} = \left(\tilde{\theta}_{\mathrm{f_{in}}}, 0, 0, 0\right)^\mathrm{T} \quad (14)$$

Obviously for such a solution it is difficult or even impossible to find its inverse transformation analytically. Therefore the numerical inverse techniques have to be used to obtain the response in the real time domain.

Roetzel [16] suggested two algorithms for the inverse transform: Gaver–Stehfest algorithm [23,24] and FFT algorithm [25,26]. Both of them have successfully been applied to the transient analysis in heat exchangers. The Gaver–Stehfest algorithm requires very little computer time. However, it is valid only if there are no oscillatory components and no discontinuities in the region $\bar{\tau} > 0$. Comparing with the Gaver–Stehfest algorithm, the FFT algorithm has no such restrictions and can provide the whole outlet temperature history simultaneously. Therefore, it is more suitable for the whole curve matching. However, if there are discontinuity points (e.g., for the case of an inlet step change together with $1/Pe = K_\mathrm{w} = 0$ and $B > 0$), the FFT algorithm will produce additional oscillations near the discontinuity point $\bar{\tau} = B$. The maximum magnitude of these additional components could reach about 9% of the magnitude of the real step change [27]. This phenomenon is called Gibss' phenomenon and cannot be eliminated by increasing the number of calculating points. For such a case, Luo and Niemeyer [28] suggested to introduce a very large dispersive Peclet number (e.g., $Pe = 10^6$) into the governing equation system to eliminate the discontinuity point.

Fig. 1(a) and (b) shows the effect of axial dispersion in the fluid and heat conduction in the wall, respectively, for the step change in the fluid temperature at the entrance to the heat exchanger. These show that the axial dispersion in the fluid has a similar effect as the longitudinal heat conduction in the wall. Either of them will result in a higher temperature rise at the outlet of the heat exchanger. Therefore, the negligence of the axial dispersion/conduction would yield a lower value of NTU. Similar diagrams of the temperature response to the exponential inlet condition are available in the literature for the influence of Pe [17] or for the influence of K_w [12].

However, the step or exponential change inlet condition is suitable only if the axial dispersion coefficient is known. As has been pointed out by Luo [17], the outlet temperature response to the step change in inlet fluid temperature for one pair of NTU and Pe can

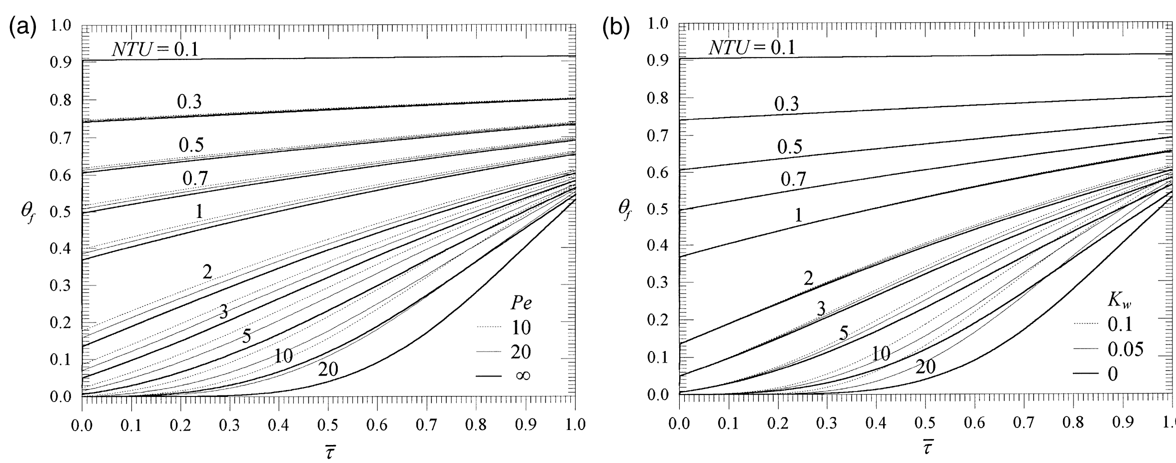

Fig. 1. The outlet fluid temperature response to a step change in the inlet fluid temperature: (a) $B = 0$, $K_\mathrm{w} = 0$; (b) $B = 0$, $Pe = \infty$.

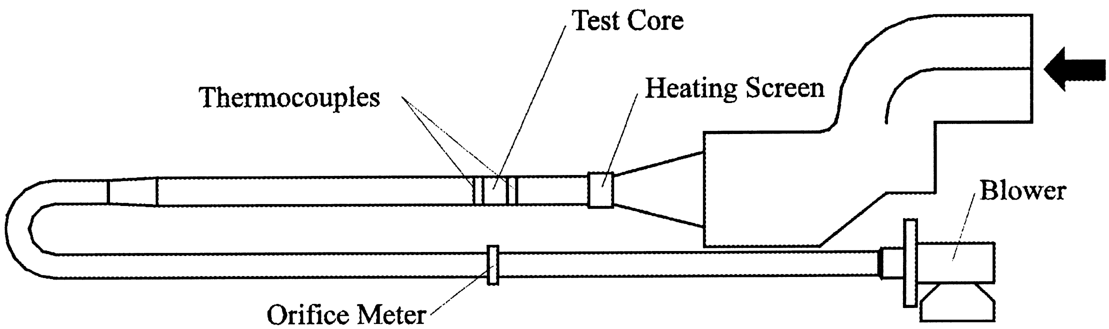

Fig. 2. Test apparatus.

also be matched with the curves for other pairs of NTU and *Pe* with very small deviation. If both the heat transfer coefficient and axial dispersion coefficient are unknown, the transient testing techniques using the step or exponential inlet condition would give unreliable results. To overcome the difficulty in evaluating both unknown parameters, Roetzel and Luo [18,19] proposed a new technique which uses a sinusoidal inlet temperature variation. In this technique, the inlet temperature variation consists of a rapid oscillation component and a step change component. The outlet temperature response therefore contains distinguishable behaviours of these two components, therefore it is possible to obtain NTU and *Pe* simultaneously. The disadvantage of their technique is mainly due to the difficulty in obtaining the inlet sinusoidal with high accuracy. As an alternative, Zhou et al. [20] used an arbitrary pulse inlet condition instead of sinusoidal variation. Such a pulse contains a rapid rise and is then followed by a rapid drop in inlet temperature. The form of the pulse is not important because in the data reduction the measured inlet temperature variation will be taken as an arbitrary function of time. According to Duhamel's theorem one obtains the outlet temperature response as,

$$\theta_{\mathrm{f}} = \int_0^{\bar{\tau}} \theta_{\mathrm{f}}^S(\bar{\tau} - z)\theta'_{\mathrm{f_{in}}}(z)\,\mathrm{d}z + \theta_{\mathrm{f}}^S(\bar{\tau})\theta_{\mathrm{f_{in}}}(0) \tag{15}$$

where $\theta_{\mathrm{f}}^S(\bar{\tau})$ is the outlet temperature response to a unit step change of inlet fluid temperature.

3. Test apparatus

The experiments were carried out in an open circuit wind tunnel located at the University of the Federal Armed Forces Hamburg, as shown in Fig. 2. The test apparatus consists of a blower, a heating element, a test segment and devices for measurement. Air is induced to flow through the wind tunnel by a centrifugal air blower. With nine layers of 80 × 80 mesh stainless steel screens installed in the inlet cross-section of the tunnel a uniform velocity distribution is obtained. In the experiment, the accuracy of the temperature and mass flow rate measurement is very important for the data evaluation. The inlet and outlet temperatures are measured with thermocouple grids which consist of 25 thermocouples joined in series and uniformly arranged across the inlet and outlet cross sections of the test segment, respectively. The wire diameter of the thermocouples is 0.1 mm. The voltage signals of the thermocouples are amplified by 5B modules and measured with a 16 bit multichannel AD/DA measurement board AT-MIO-16XE-50. The uncertainty in temperature measurement is less than 0.1 K and the inlet fluid temperature fluctuation is less than 0.2 K. The mass flow rate is measured with an orifice meter. An absolute pressure sensor and a pressure difference sensor with an accuracy of 0.15% are used for the orifice meter. The test core shown in Fig. 3 consists of 48 copper plates in the size of 1 mm (thickness) × 300 mm (width) × 500 mm (length in flow direction) to form 49 uniformly spaced parallel channels and has a total height of 300 mm. The total thermal capacity of the plates is 25.26 kJ/K.

The inlet temperature pulse of the air flow is realized with a computer controlled compact electric heater. First, the heater is turned off. After the inlet and outlet temperatures of the fluid passing through the test core

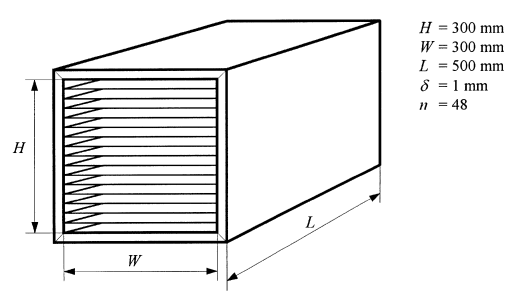

Fig. 3. Parallel plate test core.

are the same and keep steady at the initial value, the predetermined power control voltage is applied on the electric power source. The inlet and outlet temperatures as well as the mass flow rate are measured continuously in a time interval of 0.1 s and saved for evaluating the heat transfer coefficients and dispersion coefficients. The average time needed for one test run is about 20 min.

4. Data reduction and discussion

In the present work the whole curve matching technique is used in data reduction in order to reduce the influence of the temperature fluctuation. After a test run the theoretical outlet temperature response to the measured inlet temperature variation is calculated and the measured outlet temperature variation is then compared with the theoretical curve. Heat transfer coefficient and axial dispersion coefficient are determined with the criterion that the standard error of the dimensionless temperature deviation between the measured values and the theoretical results reaches the minimum. The measured temperatures with an arbitrary pulse inlet condition and a sinusoidal pulse inlet condition are shown in Figs. 4 and 5, respectively. All the matching curves of the theoretical outlet temperature responses are almost identical with the measured outlet temperature variations.

Fig. 6 shows the experimental values of the Colburn *j*-factor $j = Nu\,Re^{-1}Pr^{-1/3}$ and the axial dispersive Peclet number *Pe* versus *Re*. The relative standard errors of *j* correlation and *Pe* correlation are 2.7% and 14%, respectively. The correlation curve of the experimental data from Cai et al. [12] is also shown in the figure. The present results of *j*-factor agree well with their experimental data. However, the deviation in *Pe* is relative large. As has been pointed out by Zhou et al. [20], the measured outlet temperature profile can always be approximately matched with different pairs of NTU and *Pe*, which only yields a small deviation between the theoretical curve and the measured data.

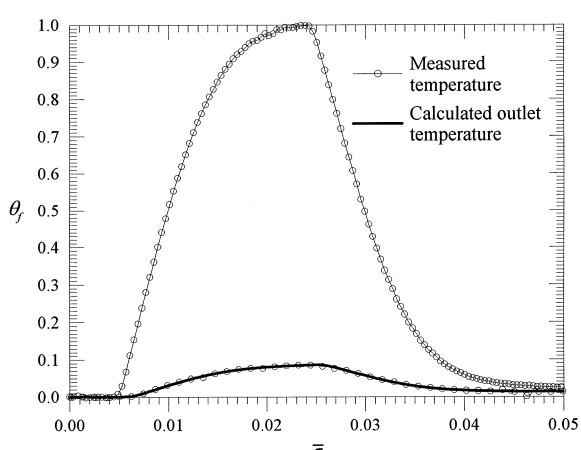

Fig. 4. Outlet temperature response to an arbitrary pulse at inlet. Re = 790, B = 0.00188, K_w = 0.05244. The calculated outlet temperature for NTU = 3.172 and $1/Pe$ = 0.1103 is almost identical with the measured one ($\sigma_n = 0.0022$).

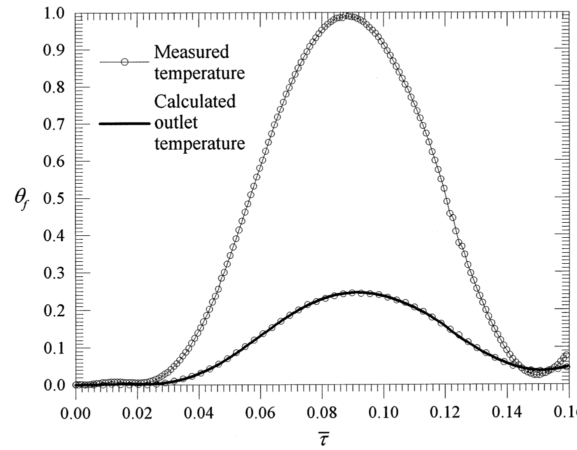

Fig. 5. Outlet temperature response to a sinusoidal pulse at inlet. Re = 1500, B = 0.00185, K_w = 0.02748. The calculated outlet temperature for NTU = 1.662 and $1/Pe$ = 0.1042 is almost identical with the measured one ($\sigma_n = 0.0013$).

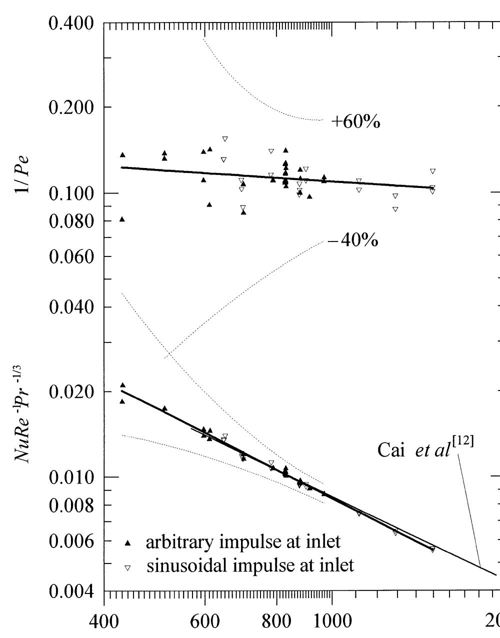

Fig. 6. Colburn *j*-factor and dispersive Peclet number as functions of Reynolds number.

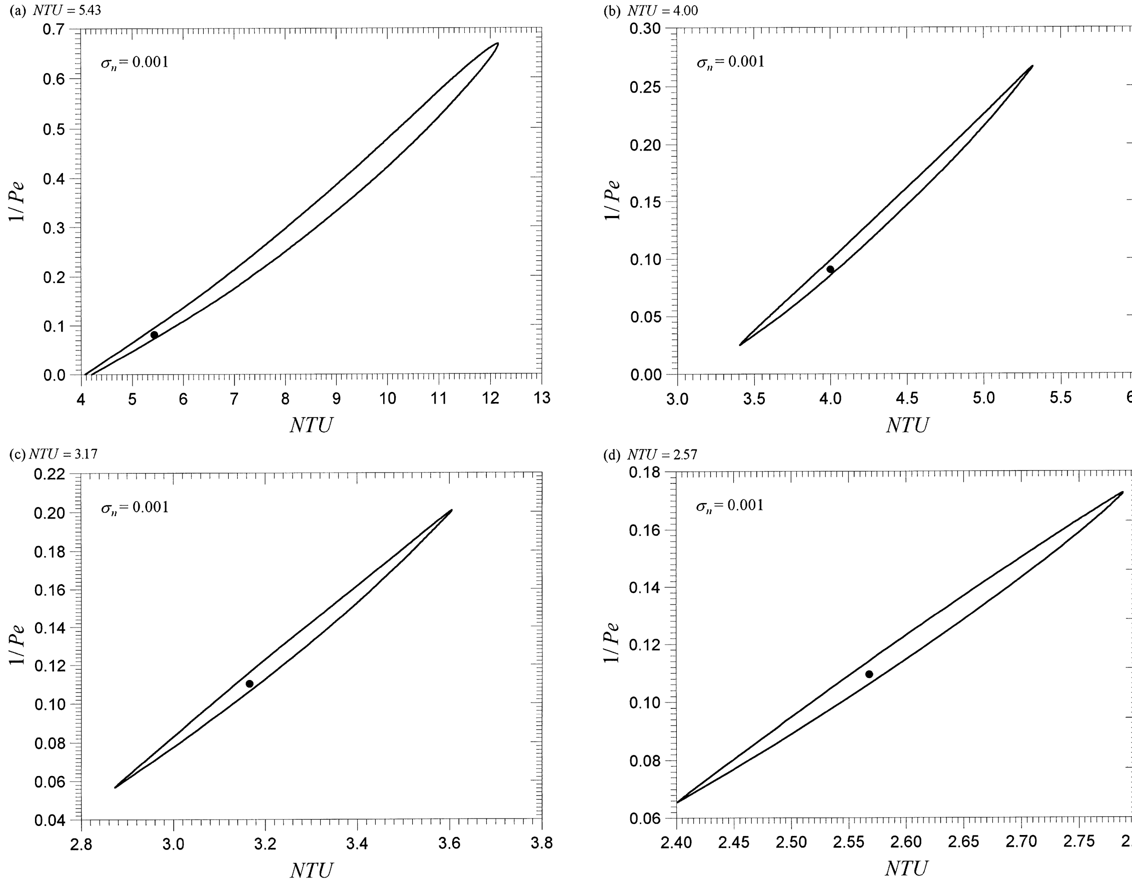

Fig. 7. Uncertainty region of NTU and *Pe* if both of them are unknown. The points indicate the pair of NTU and *Pe* which best fit the measured outlet temperature profiles: (a) NTU = 5.43; (b) NTU = 4.00; (c) NTU = 3.17; (d) NTU = 2.57.

If the maximum allowed standard error between the measured and calculated dimensionless outlet temperatures is given according to the uncertainty in temperature measurement, e.g., $\sigma_{\rm n} = 0.001$, there is an uncertainty region of evaluated values of NTU and *Pe*, as shown in Fig. 7. Any pair of NTU and *Pe* in this region will yield a calculated outlet temperature profile which fits the measured one with a standard error less than 0.001. This uncertainty region narrows when the value of NTU decreases.

The sensitivity analysis of the experiment shows that the uncertainties in NTU and *Pe* depend on the value of NTU. The uncertainty regions of NTU and *Pe* are shown in Fig. 6 with dotted lines. It indicates that the present transient testing technique for unknown NTU and *Pe* is more suitable for the region of high Reynolds numbers, i.e., for low values of NTU. Sensitivity analysis shows further that the sinusoidal inlet condition (using only one period) enlarges the uncertainty region of NTU and *Pe* because the sinusoidal temperature variation is obviously not as rapid as the pulse change which is obtained by simple on-off heating. To enhance the experimental accuracy, especially the accuracy of the axial dispersion Peclet number, it is suggested to use an inlet temperature pulse which is as close to a rectangular pulse as possible. This can be realized by using fewer layers of screens to heat the air flow and applying a rectangular power supply to the heater. To determine the axial dispersion coefficient more accurately, one can use the transient concentration measurement technique developed by Balzereit and Roetzel [29]. Their technique is also suitable to quantify weak dispersion, i.e., large values of *Pe*, with high precision.

5. Conclusions

A conduction/dispersion model for the single-blow transient testing techniques is proposed, in which the effects of the longitudinal heat conduction in the wall material and the longitudinal thermal dispersion in the fluid are considered. The outlet temperature response to an arbitrary inlet temperature pulse is solved be means of the Laplace transform. Matching the measured outlet temperature change with the theoretical model, the heat transfer coefficient and axial dispersion coefficient are determined simultaneously for each test run.

It is known that the step or exponential inlet condition would give a poor estimation of NTU and *Pe*. In fact, under the step or exponential inlet condition, in a wide range of *Pe* one can find corresponding pairs of *Pe* and NTU to get a theoretical curve very close to the measured one. If the axial dispersive Peclet number is unknown, this technique would yield significant errors in NTU. The sinusoidal inlet condition should give a better evaluation in *Pe* but it is difficult to generate the sinusoidal inlet temperature variation with sufficient accuracy. Furthermore, such a sinusoidal temperature rise is always slower than that of the pulse obtained by a rectangular electric power supply and it might yield a larger uncertainty in NTU and *Pe*. These disadvantages are overcome with the use of the pulse testing technique.

The present experiments were performed using the pulse testing technique. The software TAIHE developed by the authors was applied for the data analysis to determine heat transfer coefficients as well as axial dispersion coefficients. The experimental results agree well with those from the literature.

The software TAIHE is originally part of the selected computer programs developed by the authors in connection with the recently published book, *Dynamic Behaviour of Heat Exchangers* [21]. TAIHE calculates outlet temperature responses to arbitrary inlet temperature changes in multipass shell-and-tube heat exchangers, cross-flow heat exchangers and multistream plate-fin heat exchangers. Maldistribution is taken into account with the parabolic dispersion model. An analytical procedure is used for rapid calculations. If temperature dependent properties of the fluid and/or core material are considered, the partial differential equation system is no longer linear and the Laplace transform cannot be applied. In such cases, a numerical method is required. For such general applications, TAIHE offers a suitable finite-difference method with moving grid technique, which provides high accuracy even if the outlet temperature undergoes sudden changes or discontinuity points. However, TAIHE has not been published yet. More information about TAIHE is available from the authors.

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