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Measurement of the heat transfer coefficient in plate heat exchangers using a temperature oscillation technique

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Abstract—Thermal parameters of plate type heat exchangers are experimentally evaluated using a temperature oscillation technique. A mathematical model with axial dispersion has been utilised to evaluate heat transfer coefficient and dispersion coefficients characterised by *NTU* and Péclet number, respectively. Special reference has been made to the deviation from plug flow due to the phase lag effect in a U-type plate heat exchanger. A mathematical model for correcting the thermal penetration effect in plate edgings and thick end plates has been presented. The experimental results obtained by using the temperature oscillation technique have been compared with those obtained by traditional steady-state experiment. A series of developments have been suggested to make the method more suitable for plate type heat exchangers.

INTRODUCTION

PLATE heat exchangers (PHE) were originally devised for hygienic applications such as the dairy or brewing industries primarily because of their ease of cleaning and maintenance. Over the last two decades the use of plate heat exchangers has spread to chemical process industries like paper, pharmaceutical or petrochemical and in particular for process heating or cooling applications. The reason for the widespread application of plate exchangers in industry today is not only the ease of maintenance but also the fact that it is possible to achieve higher turbulence at a lower flow rate compared to a shell and tube heat exchanger. It is also important to note that with plate heat exchangers, heat can be recovered with comparatively lower temperature difference between the fluids.

A large number of papers [1–3] are available dealing with the effectiveness of plate heat exchangers considering a plug flow in both the fluids. This induces considerable deviation from reality since the effect of flow maldistribution is rather prominent as observed by Amooie-Foumeny [4], Haseler *et al.* [5] and many others. Along with this the other nonideality which affects the transient behaviour of the heat exchanger in particular, is the phase lag effect. This will be discussed later in the paper. Recently efforts have been made to take care of deviation from plug flow in shell and tube heat exchangers by introducing an axial dispersion term in the energy equation [6, 7]. Since the flow maldistribution is also severe in plate heat exchangers it appears to be essential to consider such dispersion in these heat exchangers as well.

The heat transfer characteristics of heat exchangers are traditionally measured by a steady-state technique described by Kays and London [8]. Though this technique is reliable and accurate for simpler geometries it becomes difficult to apply the same to complex geometries encountered in compact heat exchangers where an exact analytical solution for temperature distribution is not available. Such heat exchangers in particular and all others in general can be tested for heat transfer characteristics by transient test techniques. One of the transient test techniques is the temperature oscillation method which was first initiated by Hausen [9]. His investigation was followed by a series of developments [10–12] which has established such methods as a viable alternative to the steady-state technique. Apart from other advantages described by Stang and Bush [13], the transient technique has the additional advantage of determining both heat transfer coefficient and dispersion coefficient from a single experiment as described in a recent study [14].

The purpose of the present investigation is to apply the temperature oscillation technique to evaluate the heat transfer and dispersion coefficients of plate heat exchangers. The high degree of flow maldistribution and deviation from plug flow model demands that such heat exchangers be dealt with the concept of axial dispersion, especially when transient processes are considered. In this paper first the dispersion model is described in the light of temperature oscillation technique with special reference to dispersion in plate heat exchanger. Based on this model experiments are carried out with the temperature oscillation technique.

NOMENCLATURE

A_e	amplitude of outlet fluid temperature oscillation [K]
A_n	amplitude of the nth harmonic [K]
A_0	amplitude of temperature oscillation at the adiabatic end of the thick plate [K]
A_∞	amplitude of inlet fluid temperature oscillation [K]
a	thermal diffusivity of solid wall [$m^2\,s^{-1}$]
a_f	free flow area for the fluid [m^2]
a_h	total heat transfer area of the plates wetted by the test fluid [m^2]
a_k	complex coefficients, equation (13)
a_s	total plate cross-sectional area for wall longitudinal conduction [m^2]
B	dimensionless parameter, $M_f c_f/M_s c_s$
C	constant, equation (28)
c_f	heat capacity of fluid [$J\,kg^{-1}\,K^{-1}$]
c_s	heat capacity of solid wall [$J\,kg^{-1}\,K^{-1}$]
D	dispersion coefficient [$W\,m^{-1}\,K^{-1}$]
D_k	complex coefficients of equations (10) and (11)
F_n	complex Fourier coefficients
h	corrugation depth [m]
i	square root of -1
L	flow length in the channel [m]
l	characteristic length, equation (23) [m]
l_i	path traversed by fluid particles before channel i [m]
M_f	mass of fluid inside the plate heat exchanger during transient test [kg]
M_s	mass of the plate heat exchanger [kg]
m	exponent equation (28)
$\dot{m}_f$	mass flow rate [$kg\,s^{-1}$]
NTU	number of heat transfer units, $\alpha\, a_h/\dot{m}_f c_f$
Nu	Nusselt number, $\alpha l/\lambda$
P	period of temperature oscillation [s]
p	dimensionless period, $(\dot{m}_f c_f/M_s c_s)P$
Pe	axial dispersive Péclet number, $\dot{m}_f c_f L/a_f D$
Pr	Prandtl number
Re	Reynolds number, $(\dot{m}_f/A_f)\, d/\mu$
r_k	complex eigenvalues of equation (12)
T	temperature [K]
ΔT	temperature difference [K]
ΔT_{lm}	log mean temperature difference [K]
t	time [s]
t_i	time travelled by fluid particle to enter the ith channel [s]
U	overall heat transfer coefficient [$W\,m^{-2}\,K^{-1}$]
w	fluid velocity equation (24) [$m\,s^{-1}$]
x	coordinate [m]
z	coordinate along the width of thick walls [m]
$\tilde{z}$	dimensionless coordinate, $z\sqrt{(\tilde{\omega}/2a)}$.

Greek symbols

α	heat transfer coefficient [$W\,m^{-2}\,K^{-1}$]
γ_f	parameter, $a_f D/\dot{m}_f c_f L$
γ_s	parameter, $a_s \lambda_s/\dot{m}_f c_f L$
δ	solid wall thickness [m]
δ^*	equivalent solid wall thickness [m]
η	amplitude attenuation, A_e/A_∞
λ	thermal conductivity of fluid [$W\,m^{-1}\,K^{-1}$]
λ_s	thermal conductivity of wall [$W\,m^{-1}\,K^{-1}$]
μ	dynamic viscosity [$N\,m^{-2}$]
ξ	dimensionless coordinate, x/L
ρ	fluid density [$kg\,m^{-3}$]
τ	dimensionless time, $(\dot{m}_f c_f/M_s c_s)t$
$\tilde{\tau}$	dimensionless time, $\tilde{\omega} t$
ϕ	phase shift
$\tilde{\phi}$	phase shift of oscillation within thick wall
ψ	wall thickness correction factor, equation (21)
ω	circular frequency, $2\pi/p$
$\tilde{\omega}$	circular frequency, $2\pi/P$.

Subscripts

a	fluid a
b	fluid b
e	outlet
exp	experimental
f	fluid
n	nth harmonic
s	solid plate wall
th	theoretical
∞	inlet.

Finally, the heat transfer data obtained with this method are compared with the results obtained from steady-state tests.

TEMPERATURE OSCILLATION TECHNIQUE WITH DISPERSION IN FLUID

Hausen's original idea of using temperature oscillation for the determination of the heat transfer coefficient of thermal regenerators and other heat exchangers was further developed by his students Glaser [10], Langhans [11] and Roetzel [12]. This method applies a temperature oscillation at a particular section of the heat exchanger and observes its effect at another fixed point. Because of heat exchange with the heat exchanger walls (tubes, plates or porous solid) this propagating temperature wave encounters an amplitude attenuation and a phase shift. The heat transfer coefficient can be calculated from either of these two parameters. This method was utilised by Kast [15] to determine the heat transfer coefficient of fixed beds using sinusoidal temperature oscillation.

Stang and Bush [13] suggested that the use of either amplitude attenuation or phase shift should be chosen according to the range of *NTU* under consideration. Matulla and Orlicek [16] applied sinusoidal oscillations to a double pipe heat exchanger.

All the test methods described above suffer from the limitation that a plug flow has been assumed for the fluid which cannot be achieved in reality. The deviation from plug flow can be taken into account by introducing a dispersion term in the energy equation which takes care of the flow maldistribution popularly referred to as 'back mixing'. The dispersion coefficient D introduced for this purpose requires to be evaluated for different thermal hydraulic parameters and configurations. Recently, a unique method of determining both heat transfer and dispersion coefficient from a single experiment [14] has been presented. In this method under the assumption of constant thermal properties, heat transfer and dispersion coefficients the following governing partial differential equations were obtained by energy balance,

$$B\frac{\partial T_f}{\partial \tau}+\frac{\partial T_f}{\partial \xi}-\gamma_f\frac{\partial^2 T_f}{\partial \xi^2} = NTU(T_s-T_f) \quad (1)$$

$$\frac{\partial T_s}{\partial \tau}-\gamma_s\frac{\partial^2 T_s}{\partial \xi^2} = NTU(T_f-T_s). \quad (2)$$

The boundary conditions for these equations can be obtained from the results of Danckwerts [17] as,

$$\xi = 0: T_f-\gamma_f\frac{\partial T_f}{\partial \xi} = T_{f,\infty}, \frac{\partial T_s}{\partial \xi} = 0 \quad (3)$$

$$\xi = 1: \gamma_f\frac{\partial T_f}{\partial \xi} = 0, \frac{\partial T_s}{\partial \xi} = 0. \quad (4)$$

For steady temperature oscillation, the inlet temperature becomes a periodic function which can be described by

$$T_{f,\infty}(\tau+p) = T_{f,\infty}(\tau). \quad (5)$$

The solution to the above equations can be obtained by using complex Fourier transform [14] and reducing the equations to an ordinary differential equation of higher order. It is important to note that the system of partial differential equations is linear and hence for an approximate sinusoidal temperature wave at inlet, only the fundamental frequency of the inlet and outlet temperatures have to be considered. This is because any periodic function can be considered as a superposition of several sinusoids and it leads to the advantage that it is not necessary to have an exact sinusoidal temperature at entry. The solution can be expressed as

$$T_{f,1}(\xi,\tau) = A_{f,1}(\xi)\sin[\omega\tau+\phi_{f,1}(\xi)] \quad (6)$$

$$T_{s,1}(\xi,\tau) = A_{s,1}(\xi)\sin[\omega\tau+\phi_{s,1}(\xi)] \quad (7)$$

where the suffix 1 indicates fundamental oscillation. The numerical values of amplitude A_f and phase shift ϕ_f can be expressed as [14],

$$A_{f,1} = \{[\mathrm{Re}(F_1+F_{-1})]^2+[\mathrm{Im}(F_1-F_{-1})]^2\}^{1/2} \quad (8)$$

$$\phi_{f,1} = \arctan\left[\frac{\mathrm{Re}(F_1+F_{-1})}{\mathrm{Im}(F_1-F_{-1})}\right] \quad (9)$$

where

$$F_{f,n}(\xi) = \sum_{k=1}^{4} D_{f,k}\exp(r_k\xi) \quad (10)$$

$$F_{s,n}(\xi) = \sum_{k=1}^{4} D_{s,k}\exp(r_k\xi) \quad (11)$$

r_k ($k = 1, 2, 3, 4$) being the four complex eigenvalues satisfying the eigenfunction

$$a_0+a_1r+a_2r^2+a_3r^3+a_4r^4 = 0 \quad (12)$$

a_ns being the coefficients of the reduced ordinary differential equation,

$$\begin{aligned} a_0 &= -(n\omega)^2B+in\omega\,NTU(1+B)\\ a_1 &= NTU+in\omega\\ a_2 &= -NTU(\gamma_f+\gamma_s)-in\omega(\gamma_f+B\gamma_s)\\ a_3 &= -\gamma_s\\ a_4 &= \gamma_f\gamma_s. \end{aligned} \quad (13)$$

The constants $D_{f,k}$ can be determined by applying equations (10) and (11) to the boundary conditions (3) and (4).

The method utilises the fact that there are two measurable quantities A_f and ϕ_f which can be used to evaluate two coefficients, namely, heat transfer coefficient characterised by *NTU* and dispersion coefficient characterised by Péclet number.

SPECIAL SITUATION IN COUNTERFLOW PLATE HEAT EXCHANGERS

Plate heat exchangers can be treated as pure cocurrent or countercurrent heat exchangers in principle if end effects are neglected. This makes it possible to use the transient test technique described in the preceding section. One important fact in the single pass plate heat exchanger is that all the streams do not travel equal distance within the U-type heat exchanger. This is shown in Fig. 1(a), where a U-type plate heat exchanger is shown schematically. It is important to note that, when a temperature oscillation is created at the entry point, point 1, of the fluid, it enters the flow passages 1′, 2′, 3′ ..... with a phase lag of ϕ_i, given by

$$\phi_i = \frac{\omega l_i}{w} \quad (14)$$

when the flow velocity w is considered to be equal in all the channels. However, this effect may get cancelled in a Z-type plate heat exchanger (Fig. 1(b)) because the outlet fluids from channels 1′, 2′, 3′,... travel for decreasing length of time after exit from flow passages to arrive at exit point 2. For a U-type plate exchanger it is just the opposite and the phase lag is increased at the outlet because of increasing travel time of fluids

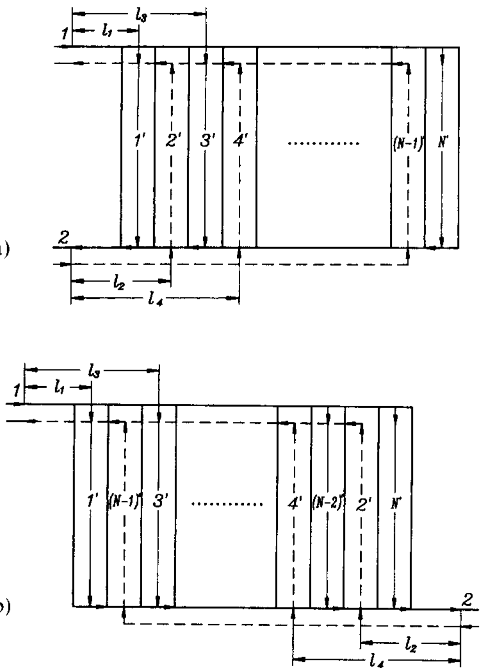

FIG. 1. (a) Different flow lengths in different flow channels in a counterflow U-type plate heat exchanger. (b) Equal flow lengths between inlet and outlet in all the flow channels in a counterflow Z-type plate heat exchanger.

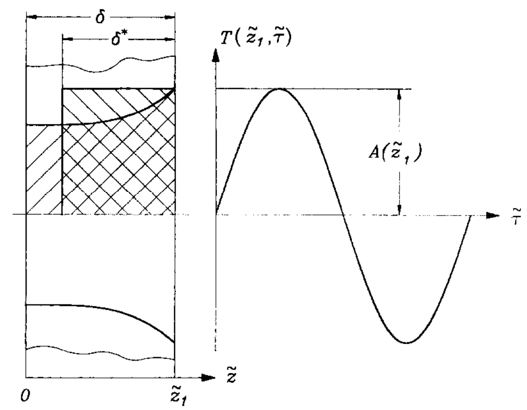

FIG. 2. Temperature amplitude profile in a thick wall and its equivalent thickness.

in channels 1′, 2′, 3′, ..., etc. This can be termed as 'phase lag effect'. The incorporation of this effect in the analytical solution makes it very complex. Hence an alternative approach can be resorted to by assuming that this effect is incorporated in the dispersion coefficient described earlier. This essentially means that the plate heat exchanger can be treated as a 'black box' with heat transfer and axial fluid dispersion taking place in it. This will result the values of the Péclet number different from that of a flow between channels only, but the data will be readily usable for the purpose of design and performance evaluation of plate heat exchangers.

The other phenomenon in testing plate heat exchangers is the temperature penetration in plate edgings and thick end plates. For a small period of oscillation, the temperature oscillation does not get enough time to penetrate through the entire thickness of the walls, so only a part of the thermal capacitances of the plate edgings and thick end plates should be added to the thermal capacitance of the exchanger core. The larger the period, the more is the effective thermal capacitance of the exchanger. This effective thermal capacitance can be calculated by defining an equivalent thickness of the wall δ^* which offers no thermal resistance in the lateral direction and will give the same temperature amplitude as that at the surface of the original plate. This is illustrated in Fig. 2, where the area under the real amplitude curve is equal to the area under the uniform amplitude curve of the wall of equivalent thickness δ^*. Transient heat conduction in a thick wall can be described by the following dimensionless differential equation:

$$2\frac{\partial T}{\partial \tilde{\tau}} = \frac{\partial^2 T}{\partial \tilde{z}^2} \tag{15}$$

with boundary conditions

$$\tilde{z} = 0: \quad \frac{\partial T}{\partial \tilde{z}} = 0$$

$$\tilde{z} = 0: \quad T = A_0 \sin \tilde{\omega}\tilde{\tau}. \tag{16}$$

Solving equation (15) together with the boundary conditions, equation (16), one obtains

$$T = \frac{A_0}{2}[\mathrm{e}^{\tilde{z}} \sin(\tilde{\tau}+\tilde{z}) + \mathrm{e}^{-\tilde{z}} \sin(\tilde{\tau}-\tilde{z})]$$

$$= A \sin(\tilde{\tau}+\tilde{\phi}) \tag{17}$$

where

$$A = A_0\sqrt{(\sinh^2\tilde{z}\sin^2\tilde{z}+\cosh^2\tilde{z}\cos^2\tilde{z})} \tag{18}$$

$$\tilde{\phi} = \arctan(\tan\tilde{z}\tanh\tilde{z}). \tag{19}$$

Hence the equivalent thickness of the plate can be evaluated as

$$\delta^* = \psi\delta \tag{20}$$

where

$$\psi = \frac{1}{\tilde{z}_1 A(\tilde{z}_1)}\int_0^{\tilde{z}_1} A(\tilde{z})\mathrm{d}\tilde{z}. \tag{21}$$

Substituting equation (18) into equation (21) one obtains

$$\psi(\tilde{z}_1) = \frac{\int_0^{\tilde{z}_1}\sqrt{(\sinh^2\tilde{z}\sin^2\tilde{z}+\cosh^2\tilde{z}\cos^2\tilde{z})}\,\mathrm{d}\tilde{z}}{\tilde{z}_1\sqrt{(\sinh^2\tilde{z}_1\sin^2\tilde{z}_1+\cosh^2\tilde{z}_1\cos^2\tilde{z}_1)}}. \tag{22}$$

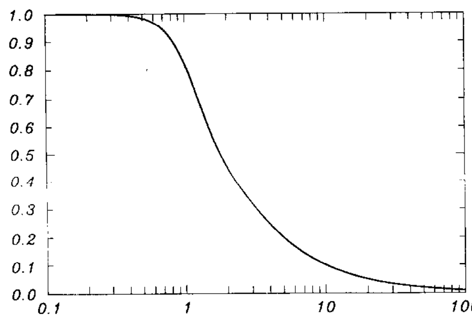

FIG. 3. Wall thickness correction factor

The correction factor ψ is given in Fig. 3 as a function of the dimensionless thickness $\tilde{z}_l$.

Experiments have been conducted to find the heat transfer coefficients and dispersion coefficients using the temperature oscillation technique with the axial dispersion model. The experimental results have been corrected for the thermal penetration effect described above.

EXPERIMENTS

Test plate heat exchanger

Tests were carried out in two plate heat exchangers with identical dimensions but different plate numbers. The first one (referred to as PHE 1) has 14 effective plates excluding the end plates with 8 channels on one side and 7 on the other. The second one (referred to as PHE 2) has 28 effective plates with 15 and 14 channels, respectively. The plates are standard chevron or herring bone type with a corrugation depth of 2 mm and corrugation pitch of 7 mm. The chevron angle is 70°. The inlet and exit port from the channels exists in the same half of the plates for ease of fabrication. The plates are made of stainless steel with a thin layer of copper on it. The average depth of the plates is 0.5 mm. They are 71 mm wide and the distance between the centre of the inlet and exit port is 176.5 mm.

For evaluating the quality of measurement, the results were compared to a conventional steady-state experiment which was carried out with counterflow arrangement with equal Reynolds number on both sides. During the steady-state measurements, the cold fluid was always kept on the side having more channels in the tests. The same channels were used for the transient test.

Experimental set up

The overall experimental set up for the temperature oscillation technique is shown schematically in Fig. 4. The assembly contains a cold and a hot water loop. The two water streams were alternately regulated towards the heat exchanger with the help of pneumatic valves controlled by a HP3852A data logger system interfaced with an IBM AT-80386 computer. The measurements were also recorded by the same data acquisition unit. Both the loops consist of pumps (2, 9) with digital speed control regulator. This enabled the online control of the flow rates of two loops which were measured by a turbine flowmeter (14). The fluid used for the measurement was distilled water which was heated by three heaters of maximum heating capacity of 22.5 kW with a temperature controller. The loops were connected to individual reservoirs (1, 10) to avoid overpressure. For cooling the cold water loop two heat exchangers of higher capacity were used. The NiCr–Ni thermocouples were formed in the shape of spirals so as to get rid of the error from overhanging. This was done to ensure that the thermocouples measure the temperature at the exact entry point of the heat exchanger. The whole assembly was insulated thermally from the environment.

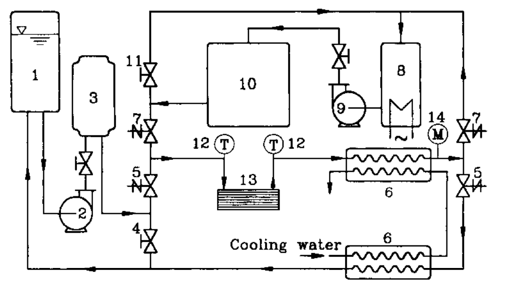

FIG. 4. Schematic diagram of the experimental setup. 1. Tank, 2. pump, 3. auxiliary tank, 4. bypass valve, 5. pneumatic valve, 6. heat exchanger, 7. pneumatic valve, 8. heater, 9. pump, 10. tank, 11. bypass valve, 12. thermocouple, 13. tested plate heat exchanger, 14. turbine flowmeter.

Data reduction

From the measured values of temperatures and flow rates the dimensionless parameters like Reynolds number and Nusselt number were calculated on the basis of characteristic length l and characteristic velocity w given by

$$l = 2h \tag{23}$$

$$w = \frac{\dot{m}_f}{\rho a_f} \tag{24}$$

so as to have

$$Nu = \frac{\alpha l}{\lambda}$$

$$Re = \frac{\rho w l}{\mu}$$

$$Pe = \frac{\dot{m}_f c_f L}{a_f D}. \tag{25}$$

For the steady-state experiment the usual LMTD relation was used in the form

$$\Delta T_{lm} = \frac{\Delta T_{fa} - \Delta T_{fb}}{\ln(\Delta T_{fa} - \Delta T_{fb})} \tag{26}$$

and

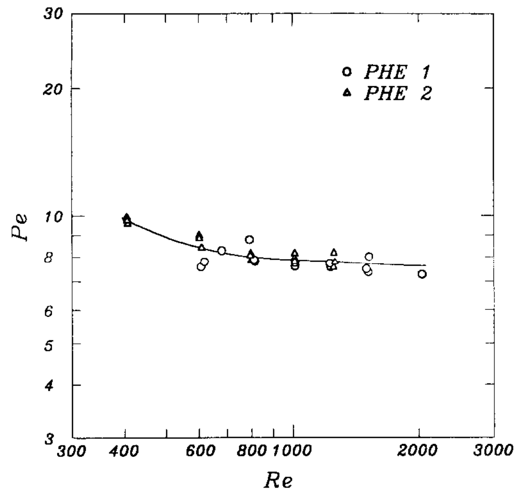

FIG. 5. Experimental data for dispersion characteristics and the optimum fitting correlation.

$$U = \frac{\dot{m}_f c_f (T_{a,\infty} - T_{a,e})}{a_h \Delta T_{lm}}. \tag{27}$$

From overall heat transfer coefficient U, the heat transfer coefficient of the two sides are evaluated by using correlation

$$Nu = C\,Re^m Pr^{1/3} \tag{28}$$

and achieving the same Reynolds number on both the sides it can be reduced to

$$\frac{Nu_1}{Nu_2} = \left(\frac{Pr_1}{Pr_2}\right)^{1/3}. \tag{29}$$

From this the individual heat transfer coefficients are calculated by taking care of the resistance due to the solid wall in between.

Results

Experimental results for PHE 1 and PHE 2 are presented in Figs. 5 and 6. From the measured

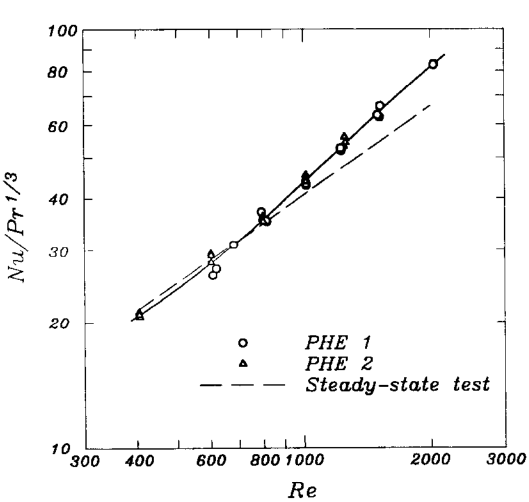

FIG. 6. Experimental data for heat transfer characteristics and its comparison with steady-state test.

oscillations, the dispersion coefficients and heat transfer coefficients are determined by minimising the error between theoretical amplitude attenuation along with phase shift and their measured values. The respective characteristic non dimensional parameters NTU and Pe can be calculated from the non-linear equation system

$$\eta_{th}(Pe, NTU) - \eta_{exp} = 0 \tag{30}$$

$$\phi_{th}(Pe, NTU) - \phi_{exp} = 0. \tag{31}$$

The resulting values of Péclet number are shown in Fig. 5 as a function of Reynolds number. Because of the limits of flow rates, PHE 1 and PHE 2 were tested for the range of $600 \leqslant Re \leqslant 2000$ and $400 \leqslant Re \leqslant 1200$, respectively. The measured heat transfer coefficients are plotted in the form of $Nu/Pr^{1/3}$ in Fig. 6. The relative standard errors of the dispersion parameter Pe and the heat transfer parameter $Nu/Pr^{1/3}$ are 3.9 and 2.9%, respectively.

Discussions

The results presented above bring out the fact that a strong dispersion phenomenon exists in the plate heat exchangers which can be observed from the relative lower order of magnitude (order 10) for the Péclet number. Under such conditions the transient test technique seems to be an ideal alternative to the conventional steady-state technique. The present test technique is not only an improvement over the previous transient methods because of its ability to measure dispersion in fluid but also because it takes into account the fluid heat capacity and 'phase lag effect' which is a special phenomenon in a U-type plate heat exchanger. The dispersion in general and in transient state in particular plays a major role in determining the behaviour of the heat exchanger which can be very well accounted for by this rapid test method.

It should also be mentioned here that the errors encountered in the method are mainly due to the flow fluctuation between the two loops of fluid. This manifests itself as an error in the parameters measured. It should be noted that the method is found to be quite sensitive to the dimensionless period chosen for measurement. The choice of optimal dimensionless period depends on the sensitivity to the measurement error which is kept at its minimum subject to the experimental constraints. In the present test a dimensionless period of 13–14 has been chosen by trial and error.

Comparison with steady-state test data

In order to judge the quality of the present test method the heat transfer data in the form of $Nu/Pr^{1/3}$ as a function of Re have been plotted. The steady-state test resulted in the following equation for heat transfer coefficient for the given configuration of the plate heat exchanger

$$Nu = 0.317\,Re^{0.703} Pr^{1/3}. \tag{32}$$

In Fig. 6 these steady-state data have been compared with the data achieved by the transient technique. It is interesting to note that in the lower range of *Re* good agreement between the two methods is observed while in the higher range the values begin to diverge from each other. This can be attributed to the fact that at higher Reynolds number the flow rate is higher and hence to achieve similar dimensionless period, a smaller real period is to be chosen. At shorter oscillation period the flow fluctuation effect and the time delay of pneumatic valves become comparable with the period of oscillation and this propagates error in measurement. It is also important to note that the heat transfer coefficient measured by the transient technique is on the higher side of the steady-state data. However, it should be kept in mind that the conventional steady-state experiment does not consider dispersion in fluid and thus it tends to incorporate the effect of dispersion in the value of heat transfer coefficient hence there should be always a difference between the two test data. The solution to this seems to be determination of steady-state heat transfer data with axial dispersion taken into account.

CONCLUSION

The present work applies a temperature oscillation technique for the determination of the heat transfer coefficients of plate heat exchangers with U-type flow arrangement. The heat transfer is considered with an axial dispersion in the fluid which takes care of the flow maldistribution and 'phase lag effect' of the heat exchanger. The results confirm that a strong axial dispersion phenomenon exists in the fluid which makes it considerably different from the conventional plug flow model. It is also important to carry out a correction on the thermal capacity of the thicker plates where penetration of temperature oscillation is poor at higher frequency. The experimental results are compared with the steady-state test results carried out with the same heat exchangers. On the basis of results obtained, the following recommendations are made for further development.

(i) Since the method is sensitive to dimensionless period p, a theoretical analysis of the error sensitivity with p may be carried out to improve the method.

(ii) In future the results should be compared to steady-state tests which take dispersions into account.

(iii) The test data can be used to predict the dynamic behaviour of the heat exchangers under consideration which can be readily verified by experimentation on dynamic response with different types of temperature variations at inlet.

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REFERENCES

1. A. Cooper and J. D. Usher, Plate heat exchangers. In *Heat Exchanger Design Handbook*, (Editor-in-chief E. U. Schlünder), Vol. 3. Hemisphere, Washington, D.C. (1983).
2. B. W. Jackson and R. A. Troupe, Plate heat exchanger design by ε–*NTU* method, *Chem. Engng. Prog. Symp. Series* **62**(64), 185–190 (1966).
3. S. G. Kandlikar, Performance curves for different plate heat exchanger configurations, ASME PAPER No. 84-HT-26 (1984).
4. M. M. Amooie-Foumeny, Flow distribution in plate heat exchangers, Ph.D. Thesis, University of Bradford (1975).
5. L. E. Haseler, V. V. Wadekar and R. H. Clarke, Flow distribution effects in a plate and frame heat exchanger, *ICWEME Symp. Ser.* **129,** 361–367 (1992).
6. W. Roetzel, Transient analysis in heat exchangers, presented at the *ICHMT International Symposium on New Development in Heat Exchangers*, Lisbon, Portugal, 6–9 Sept. (1993).
7. W. Roetzel and Y. Xuan, Analysis of transient behaviour of multipass shell and tube heat exchangers with the dispersion model, *Int. J. Heat Mass Transfer* **35,** 2953–2962 (1992).
8. W. M. Kays and A. L. London, Heat transfer and flow friction characteristics of some compact heat exchanger surfaces, Part I—Test system and procedure, *Trans ASME* **72,** 1076 (1950).
9. H. Hausen, *Wärmeübertragung im Gegenstrom, Gleichstrom und Kreuzstrom* (2nd Edn). Springer, Berlin (1976).
10. H. Glaser, Der Wärmeübergang in Regeneratoren, *Z. VDI, Beiheft Verfahrenstechnik* **4,** 112–125 (1938).
11. W. U. Langhans, Wärmeübergang und Druckverlust in Regeneratoren mit rostgitterartiger Speichermasse, *Arch. Eisenhüttenw.* **33,** 347–353 and 441–451 (1952).
12. W. Roetzel, Measurement of heat transfer coefficients in tubes by temperature oscillation analysis, *Chem. Engng Technol.* **12,** 379–387 (1989).
13. J. H. Stang and J. E. Bush, The periodic method for testing compact heat exchanger surfaces, *J. Engng Power* **96A**(2), 87–94 (1974).
14. W. Roetzel, X. Luo and Y. Xuan, Measurement of heat transfer coefficient using temperature oscillations, *Proceedings of the 3rd World Conference on Experimental Heat Transfer, Fluid Mechanics and Thermodynamics*, Honolulu, Hawaii, 31 Oct–5 Nov (1993). Also accepted for publication in *Experimental Thermal and Fluid Science*.
15. W. Kast, Messung des Wärmeüberganges in Haufwerken mit Hilfe einer temperaturmodulierten Strömmung, *Allg. Wärmetech.* **12**(6/7), 119–125 (1965).
16. H. Matulla and A. F. Orlicek, Bestimmung der Wärmeübergangscoeffizielten in einem Doppelrohrwärmeaustauscher durch Frequenzganganalyse, *Chemie-Ing.-Tech.* **43**(20), 1127–1130 (1971).
17. P. V. Danckwerts, Continuous flow systems—distribution of residence times, *Chem. Engng Sci.* **2,** 1–13 (1953).