Heat Transfer Engineering, 22:17–25, 2001

0145-7632/01 $12.00 + .00

Application of a New Analogy for Predicting Heat Transfer to Cross Rod Bundle Heat Exchanger Surfaces

PRASANT NANDA

Department of Mechanical Engineering, Indira Gandhi Institute of Technology, Sarang, Orissa, India

SARIT KUMAR DAS

Heat Transfer and Thermal Power Lab, Department of Mechanical Engineering, Indian Institute of Technology, Madras, India

HOLGER MARTIN

Institut für Thermische Verfahrenstechnik, Universität Karlsruhe (TH), Karlsruhe, Germany

A new technique for prediction and measurement of compact heat exchanger surfaces has been established. The method utilizes an analogy between heat transfer and fluid friction proposed by a French scientist in 1928. The applicability of this analogy to plate heat exchanger surfaces was examined recently. The present work proposes to use the same analogy for crossed rod bundles used in regenerators. For this purpose, the analogy equation has been modified to include a geometric factor which takes care of the fraction of the pressure drop arising out of skin friction. The proposed form of equation is found to be more general, applicable to crossed rod structures of any geometric configuration. The proposition was formulated based on standard data from the literature and also by conducting independent experiments. The result of the transient experiment indicate excellent adherence to the proposed analogical equation. The experimental evidence clearly indicates the possibility of predicting heat transfer characteristics of such compact surfaces by measuring pressure drop alone across the packed bed formed by the surface under consideration.

The use of compact heat exchangers has been on the rise in recent times. The primary reasons contributing to the phenomenal increase in the use of compact heat exchangers are the diversification of heat transfer studies in emerging areas of technology and the unprecedented growth of materials and manufacturing technology. Today, heat transfer studies are concentrated in areas such as aerospace, heat transfer at the nano scale in electronic chips, biomedical and biotechnological applications, renewable energy resources, and environment-friendly energy conversion processes, to name a few. All these applications call for very efficient,

Address correspondence to Dr. Sarit Kumar Das, Head, Heat Transfer and Thermal Power Lab, Department of Mechanical Engineering, Indian Institute of Technology, Madras 600 036, India. E-mail: sarit_das@hotmail.com

unique, reliable, and innovative design of heat exchangers ranging from a size of a few millimeters to a few meters and handling thermal loads between a few watts and megawatts. Incidentally, almost all the innovations for these special applications are in the area of compact heat exchangers. For example, the wave which appeared in the 1980s in Europe [1] was the invasion of plate and spiral plate heat exchangers. Even a newer generation of shell-and-tube heat exchangers was built to achieve higher efficiency through complex flow patterns like that in microminiature heat exchangers [2] and helically baffled heat exchangers [3]. The developments in the purely compact regime, such as multistream plate-fin heat exchangers [4] and micro-strip-channel heat exchangers [2], have also made heat transfer analysis complex due to their typical construction and flow configuration.

In all these examples, one common feature that can be noted is the very complex flow pattern aimed at augmenting the heat transfer primarily by using surfaces and obstructions which act as turbulence promoters or vortex generators. The major challenge before a designer of such heat exchangers is to determine accurately the heat transfer coefficient of that side of the heat exchanger which has a complex flow structure. It is impossible at present to derive any analytical correlation for heat transfer coefficient over such compact surfaces. Numerical solution is too expensive and time consuming, as supported by the fact that a detailed numerical study of even the most commonly used surfaces such as herringbone plates are hardly available in the literature. The only way seems to be experimental studies such as those presented by Kays and London [5]. It goes without saying that such experiments are not only time consuming but also need to be backed up by proper analytical modeling if they are to be determined by transient tests like those described in [6–8]. Such tests are particularly unavoidable for surfaces used for regenerative heat exchangers, and similar tests are gaining popularity for transfer types of heat exchangers [9] as well.

To date, the manufacturers of compact heat exchangers have carried out design practices based on test data which are proprietory in nature. This makes it extremely difficult to suggest a generalized correlation for compact surfaces used in heat exchangers. A theoretical approach in this direction was made by the third author of this article (Holger Martin), initially for chevron plates. In predicting flow friction characteristics, he has taken a logical approach in analyzing the different flow patterns and has developed a theoretical correlation for fluid friction based on experimental data and physical reasoning. For predicting heat transfer characteristic, he used an old study by Lévêque [10], originally proposed for thermally developing laminar flow over a short heated length. It was pointed out that the same analogy can be used in the turbulent regime and the corresponding equation was named the generalized Lévêque equation [11]:

$$\mathrm{Nu} = 0.404\left(f\,\mathrm{Re}^2\,\mathrm{Pr}\,\frac{d_h}{L_c}\right)^{1/3} \tag{1}$$

where d_h is the hydraulic diameter and L_c is the characteristic length, which was taken as the distance between two crossing points in a plate heat exchanger [11] and is a function of the wavelength and the chevron angle of the corrugation pattern.

Preliminary calculations have shown that the same approach is applicable for packed beds of particles, for tube bundles, and possibly for many other systems with periodic flow structures.

The present work is aimed at a similar approach of using the Lévêque analogy for cross rod bundles. Some fraction of the total pressure drop should be used for evaluating the heat transfer coefficient, because only the skin friction contribution of the pressure drop can be related to heat transfer and not the form drag contribution. It has been found that this fraction of Δp came out to be 0.5 for plate heat exchangers and tube bundles.

In the present comparison with data for cross rod bundles from the literature and from experiments of the first and second authors, we find that the fraction of Δp, to be used in the modified Lévêque equation, depends on a geometric parameter, the lateral pitch ratio, and can be easily correlated. The dependence of the fraction of Δp to be used in the Lévêque equation on lateral pitch is physically explained while discussing the results.

The Lévêque Analogy

Transport of heat and momentum are similar in nature. The classical studies concerning the transfer of heat and momentum in fully developed temperature and velocity profiles in turbulent flow were presented by Osborne Reynolds, Ludwig Prandtl, Theodore von Kármán, and many other investigators. These analogy models, starting with the simplest proportional correlation given by Reynolds for turbulent flow over a flat plate, relate the friction factor to the heat transfer coefficient. The Reynolds analogy proposes a linear relationship between heat and momentum transfer coefficients in the form

$$\mathrm{St} = \frac{\mathrm{Nu}}{\mathrm{Re\,Pr}} = \frac{f}{8} \tag{2}$$

This analogy corresponds to a Prandtl number of nearly one and hence can be applied without much error to ideal gases. For fluids differing widely from ideal gas, Ludwig Prandtl and many others have developed more refined forms of analogies (see also Schlünder [12]).

It can be observed readily that the Lévêque equation, in reality, is in the form of an analogy, too, where apart from the friction factor a geometric scale factor is also incorporated. The reason for inclusion of this factor is that the Lévêque equation was originally derived for a thermal boundary layer in a hydrodynamically developed laminar flow in a duct. This is also the reason that even after being completely aware of this theory for more than half a century, researchers working with practical heat transfer equipment did not consider it to be very important, as the flow in almost all equipment is turbulent in nature. Schlünder [12] expressed his opinion in an article entitled "The Historical Development and Present State of Scientific Theory of Heat Transfer" (in German) about the Lévêque equation which was translated in [11] as follows:

> ...and there might probably be one single case of turbulent heat transfer only for which an equation can be derived from the classical differential equations for viscous flow which are bound to no modeling concept whatsoever in the mechanism of turbulent flow and therefore may be taken as rigorous in the classical sense. It is only valid, however, for extremely short heated lengths of tube with a developed turbulent flow, and therefore it is more of an academic than of practical value.

It took the rigorous exercise of analyzing available experimental data on chevron plates from the literature in support of his proposition and found that they in fact do agree with the analogy.

APPLICATION OF THE ANALOGY FOR WIRE MESH AND CROSS ROD BUNDLES USED AS REGENERATOR PACKINGS

It is quite logical from the earlier studies [10, 11] that the analogy can be applied to heat transfer problems where small repetitive heated lengths are involved. The reason why the analogy has been found to be successful for plate heat exchangers is due to the zigzag wavy flow path in chevron plates. In compact heat exchangers, similar flow structures are also encountered, and hence the logical conclusion that the same can be applied to the said surfaces is quite obvious.

As an example, the crossed rod bundle used for regenerative heat exchangers can be studied. The different kinds of crossed rod bundles used for the analysis are oriented staggered, inline, and randomly packed with the assumption that rods are packed tightly, which leaves no space between the different layers of matrix material. Using the flow friction characteristic for these geometries from Kays and London (Figures 10-98 to 10-100 of [5]), the theoretical prediction is made according to Lévêque's equation. In this equation, the length L_c is taken as the longitudinal pitch between repeated flow structure as shown in Figure 1. However, for closely packed stacking, this length reduces to $L_c = d/2$, where d is the rod diameter.

A selected number of results are plotted (Figures 2–5) to bring out the most important features of the analogy. In all cases, the experimental data for flow friction are taken from the literature (Figures 10-98 to 10-100 of [5]) and after application of the Lévêque analogy, they are compared with the heat transfer data for identical geometries from the same reference [5]. Originally, calculation was done and plots were made for all the geometries of Figures 10-98, 10-99, and 10-100 of [5]. However, it was interesting to observe that the final form of the Leveque analogy suggested [Eq. (3), given later] does not depend on the arrangement (i.e., inline, staggered, or random). Due to limited space here we have presented in Figures 2–5 some selected values of results for four different piches—Figures 2 and 3 are for inline arrangement (1I and 3I), Figures 4 and 5 are staggered arrangement (5S and 7S) as given in Table 10-10 and Figures 10-98 and 10-99 of Kays and London [5]. Figures 2–5 show the comparison of measured heat transfer coefficients against the theoretical prediction using the generalized Lévêque equation [Eq. (1)]. On the same Figures (2–5) is also shown the predicted heat transfer characteristic from the modified Lévêque equation (to be presented later).

It can be observed from the curves that the value of $\mathrm{St}\,\mathrm{Pr}^{2/3}$ calculated from the generalized Lévêque equation deviates from the experimental result almost by a constant factor over the entire range of the Reynolds number. This virtually means that for each geometry there remains a constant multiplicity factor to match

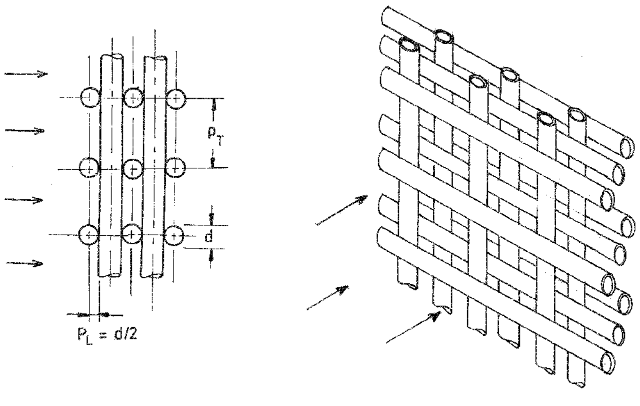

Figure 1 Schematic of wire mesh cross section for inline arrangement.

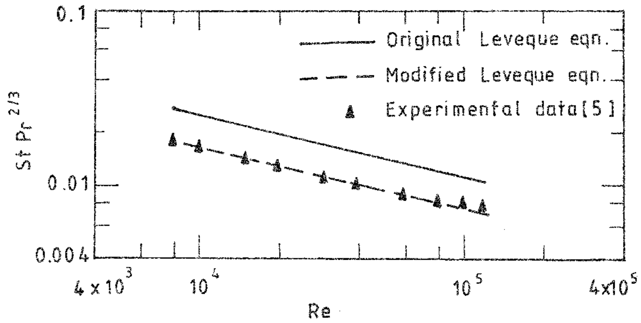

Figure 2 Comparison of original and modified Lévêque equation with experimental data from (1I of Figure 10-98 in [5]) (X_t = 4.675).

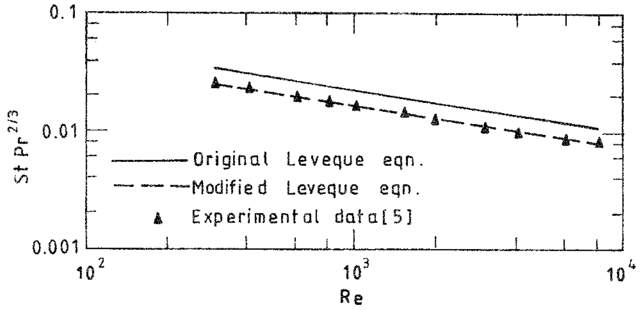

Figure 3 Comparison of original and modified Lévêque equation with experimental data from (3I of Figure 10-98 in [5]) (X_t = 3.356).

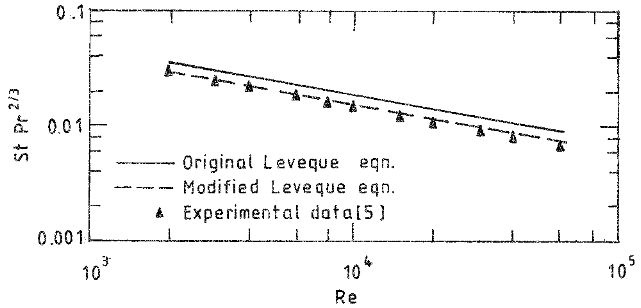

Figure 4 Comparison of original and modified Lévêque equation with experimental data from (5S of Figure 10-99 in [5]) (X_t = 2.417).

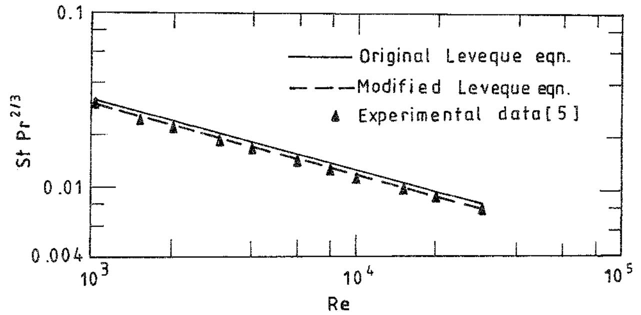

Figure 5 Comparison of original and modified Lévêque equation with data from (7S of Figure 10-99 in [5]) ($X_t = 1.571$).

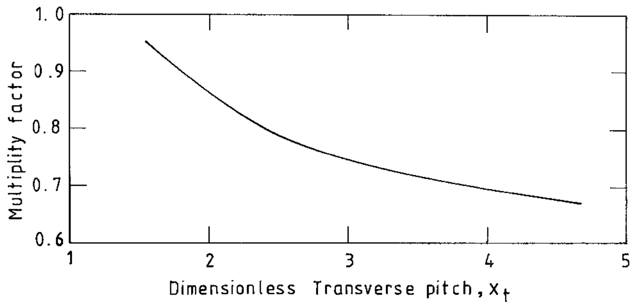

Figure 6 Variation of multiplicity factor for pressure drop with dimensionless transverse pitch.

with the experimental result. This is the factor which takes care of the fraction of the total pressure drop resulting from the skin friction alone. As a consequence, this multiplicity factor is found to be fairly independent of the arrangement of the wire mesh (i.e., inline, staggered, or random). Figure 6 shows the variation of this multiplicity factor with the transverse pitch, X_t. It should be mentioned here that the multiplicity factor indicated here is for the Nusselt number, since Nu$\alpha(\Delta p)^{1/3}$, the fraction of pressure drop to be used, is the cube of this multiplicity factor.

These results show that the original generalized Lévêque equation gives considerable deviation from the actual measured heat transfer coefficient as given by Kays and London [5]. It is also important to note (Figures 2–5) that the amount of deviation is neither random nor constant, but is found to vary with dimensionless transverse pitch. For an accurate prediction of heat transfer coefficient from the analogy, only a fraction of the pressure drop resulting from skin friction should be taken into consideration. Figure 7 shows such an exercise, in which it is found that 30% of the total pressure drop predicts the heat transfer accurately. However, the approach is empirical in nature. Since the deviation also varies with X_t, it is difficult to ascertain what fraction of the pressure drop is due to the skin friction that should to be taken for the calculation of the heat transfer coefficient.

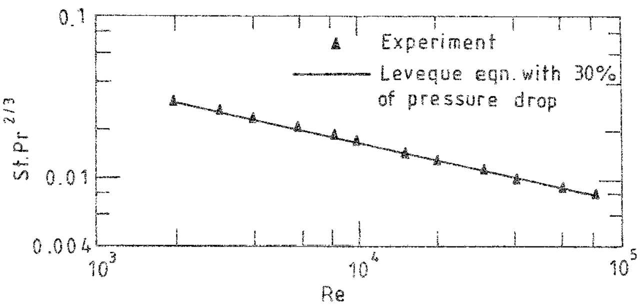

Figure 7 Comparison of experimental observation from [5] with original Lévêque equation using 30% of the pressure drop.

Therefore, looking at the trends of the deviation, we suggest that the nondimensional transverse pitch should appear in the analogy correlation. This will take care of the task of elimination of the form drag creeping into the equation (d being constant for all the cases). After a series of studies it is proposed that the same may be incorporated in the following form:

$$\mathrm{Nu}\,\mathrm{Pr}^{-1/3} = 0.44\left(f\frac{4r_h}{d/2}\frac{\mathrm{Re}^2}{X_t}\right)^{1/3} \tag{3}$$

which may be called a modified Lévêque analogy for cross rod bundles. It is obvious from the nature of the variation of the multiplicity factor that the part of total pressure drop accounting for skin friction depends on the transverse pitch of the wire mesh. This can be physically explained by the fact that when the pitch is small, the entire amount of the fluid is covered by the boundary layer and there is very little freedom for the fluid to turn in order to increase the form drag. On the other hand, when the pitch is large, the fluid at the center of each crossed rod square is virtually unaffected by fluid friction and is free to turn, giving more pressure drop due to form drag. Thus, as the transverse pitch increases, less and less of the total pressure drop comes from skin friction, lowering the multiplicity factor. The plots in Figures 2–5 clearly show that the prediction of heat transfer coefficient from the modified Lévêque analogy is in excellent agreement with the experimental results. The predictions are within a standard deviation of 5% from the experimental results.

EXPERIMENTAL VALIDATION OF MODIFIED LEVEQUE ANALOGY

Further strong foundation of the proposed modified Lévêque analogy has been laid by conducting independent experiments to measure the pressure drop and heat transfer coefficient during the same test. These tests also indicate the practical application of the analogy. It is found that the pressure drop measurement is good enough to predict the heat transfer characteristics, thus avoiding more complex transient tests, which are not only difficult but for which the accuracy is critically dependent on the theoretical model chosen for data reduction.

Experimental Setup

The schematic of the experimental setup is shown in Figure 8. The test section is a small bed of tightly packed wafers of stainless steel woven wire mesh screen. The woven wire mesh screen has been chosen for the validation because it simulates identical heat transfer and fluid frictional data as crossed rod matrices as indicated in Chapter 7 of [5]. The bed is kept inside a galvanized iron pipe. There are two valves, one of which (V_1) serves as a bypass valve. The blower is run and the air is heated by electrical heaters. At this stage, air is not allowed to pass over the bed, until the air temperature attains a steady-state value. During this phase of operation, to avoid increase in temperature of the bed due to heat conduction through the wall of the tube, the bed is continuously cooled by compressed air. The bed

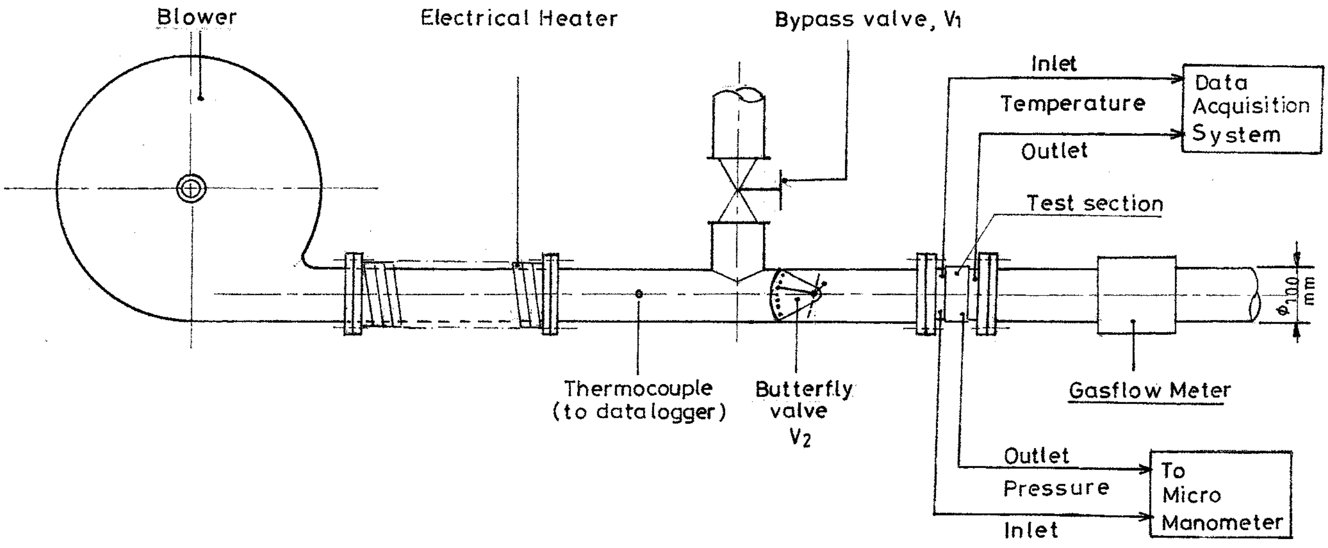

Figure 8 Schematic of the experimental setup.

temperature throughout this phase of operation remains fairly constant over the entire bed. Subsequently, valve V_2 is opened to allow the hot air to pass over the bed. The bed temperature increases and after some time the outlet air temperature approaches the inlet one. Flow rate is measured by the gas flow meter, and the pressure drop across the bed is measured by a digital micromanometer. The valves are readjusted for a different flow rate and the experiment is repeated. The temperature–time history at the inlet and outlet of the bed are continuously recorded by four thermocouples (two at each section). Another thermocouple is placed before the bypass valve to ensure that steady-state temperature has been reached for bed inlet temperature. Between two experiments, sufficient time is given for the bed to cool down to atmospheric temperature. Details of the test section are given below.

16 × 16 stainless steel wire mesh	$M = 0.74$ kg
d (wire diameter) $= 0.2$ mm	$C = 480$ J/kg K
X_t (dimensionless transverse pitch) $= 8.9375$	Tube diameter $= 10$ cm
$d_h = 2.075$ mm	$L = 4$ cm

Error Estimates

In transient tests, it is important to choose the proper thermocouple to match the allowable response lag. The present Philips thermocoax thermocouples have a time constant of 40–50 ms, which is less than the measurement time. The accuracy of the thermocouples is 0.01 K, which is less than 0.5% of the temperature difference measured during the experiment. In the present experiment the error in flow measurement is ±2.5%. The data reduction shows a sensitivity of less than 0.5% of the temperature difference measured during the experiment. It also shows a sensitivity of 3% in the heat transfer coefficient at the maximum error of flow measurement.

Data Reduction

The most important part of a transient test is the analysis of experimental outputs. In the present data reduction, the usual definition of the Reynolds number in the wire mesh has been used.

$$\mathrm{Re} = \frac{4 r_h G}{\mu} \tag{4}$$

where $G = \dot{m}/A_c$ is the mass velocity of the fluid.

The Darcy friction factor f is calculated as

$$f = \frac{2\rho\, \Delta p\, d_h}{L G^2} \tag{5}$$

The evaluation of heat transfer coefficient is based on the simple theoretical modeling of a regenerator bed. The underlying assumptions for this modeling are:

1. The entire bed is insulated from the surrounding.
2. The temperatures of both fluid and matrix are uniform over any cross section of the bed, i.e., the thermal conductivity along the radial direction in the bed is infinite.
3. The axial thermal conductivity of the bed and the fluid along the flow direction are neglected for the range of Reynolds number ($\mathrm{Re} > 150$) associated with the experiment.
4. The effect of holdup fluid heat capacity is negligible.
5. Depending on the above assumptions, the nondimensional governing equation for the bed fluid system can be given by

$$(\text{gas})\ \frac{\partial T_g}{\partial x_D} = N_{\text{tu}}(T_m - T_g) \tag{6}$$

$$(\text{bed})\ \frac{\partial T_m}{\partial x_D} = N_{\text{tu}}(T_g - T_m) \tag{7}$$

$$(\text{gas})\ T_g(x_D = 0, \theta) = T_{g,i} \tag{8}$$

Temperature distribution of the gas and matrix for the above conditions are given by [13]

$$\frac{T_g - T_{g,i}}{T_o - T_{g,i}} = G_o(\eta, \xi) \tag{9}$$

$$\frac{T_m - T_{g,i}}{T_o - T_{g,i}} = F_o(\eta, \xi) \tag{10}$$

where G and F are special functions known as Anzelius-Schumann functions. They are detailed in the Appendix.

The transfer function for the single-blow packed-bed system in the Laplace domain is $s\bar{G}_o(s, \xi)$. In the present case, the experimental input is not an exact step function. A typical recorded input temperature history is shown in Figure 9. To obtain the output, a fitting of the following nature has been used, which is also shown in Figure 9.

$$\frac{T_{g,\text{in}} - T_{\text{min}}}{T_{\text{max}} - T_{\text{min}}} = a(1 - e^{-ct}) + b \tag{11}$$

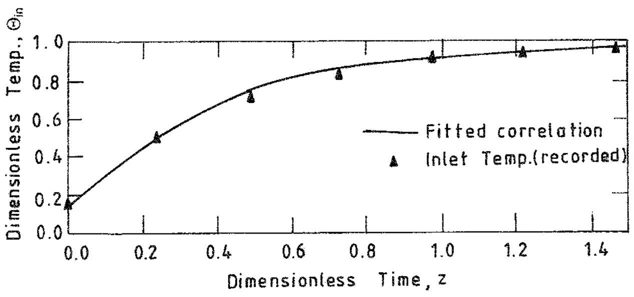

Figure 9 Fitted input temperature used for computation.

The output in the Laplace domain is obtained by multiplying the Laplace transform of the above function by the transfer function. The resulting equation is inversed back numerically to the time domain based on Crump's [14] algorithm using fast Fourier transform. The temperature–time response is compared with the experimental response (Figure 10) for the determination of the heat transfer coefficient.

Results of the Experiment

Even though the analysis of data from Kays and London [5] proved conclusively the applicability of the Lévêque analogy to crossed rod bundles, it was important to ensure that this match was not by chance and all similar flow structures can be modeled through the same analogy. Moreover, the experiment would demonstrate how the heat transfer behavior of an unknown compact surface can be determined directly from the analogy concept. To do this, the prediction based on pressure drop data has been plotted (Figure 11). The figure shows the experimental results for heat transfer measurements along with the prediction from the Lévêque analogy (modified) based on measured friction factor. It demonstrates excellent agreement in the entire range of Re from 300 to 650. The deviation is less than 5%, which is even smaller than the accuracy of standard experimental data.

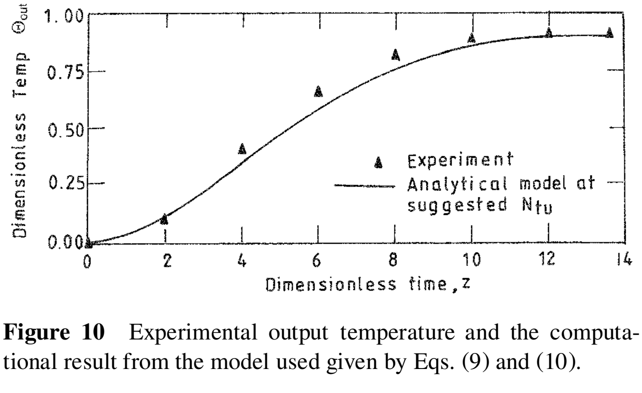

Figure 10 Experimental output temperature and the computational result from the model used given by Eqs. (9) and (10).

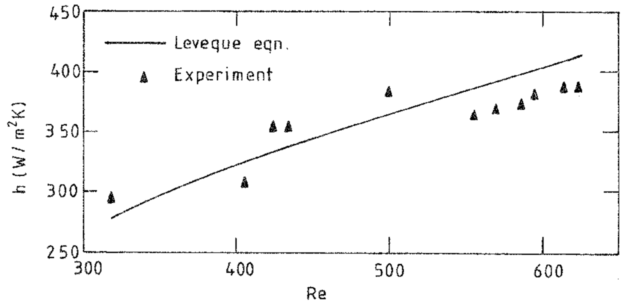

Figure 11 Comparison of present experimental heat transfer data with modified Lévêque equation.

CONCLUSION

The present work brings out a new look at an analogy which remained unused for more than half a century. The suggested form of analogy, which can be termed a modified Lévêque analogy, compares excellently with the experimental data available in the literature for crossed rod bundles. This fact has been utilized to suggest a method for the prediction of heat transfer coefficient based on pressure drop measurement, and the proposition has been founded by conducting experiments on wire mesh regenerators. It is interesting to note that a unique equation can be utilized for both crossed rod and wire mesh matrices without any necessity to change the constants and the exponent involved in the equation. This makes the present form of analogy more general and directly usable for similar compact surfaces. It is further suggested that experiments can be conducted with compact surfaces with similar flow structures which are of practical importance for heat transfer engineers.

NOMENCLATURE

A	total heat transfer area of the matrix, m^2
A_c	free flow area of the matrix, m^2
C	specific heat of the solid/matrix, J/kg K
C_p	specific heat of the fluid, J/kg K
d	diameter of the wire/crossed rod, m
d_h	hydraulic diameter, m
f	Darcy (skin) friction factor
G	mass velocity in the bed $\dot{m}/A_c$, kg/m^2 s
$G_n(\eta, \xi)$	a special function (given in Appendix)
$\bar{G}_n(s, \xi)$	transformed $G_n(\eta, \xi)$
h	convective heat transfer coefficient, W/m^2 K
k	thermal conductivity of the fluid, W/m K
L	length of the regenerator bed, m
L_c	characteristic length, m
$\dot{m}$	mass flow rate of the fluid, kg/s
M	total mass of the matrix solid, kg

Nu	Nusselt number ($= hd_h/k$)
N_{tu}	number of transfer units ($= hA/\dot{m}C_p$)
P_L	longitudinal pitch, m
P_T	transverse pitch, m
Pr	Prandtl number ($= \mu C_p/k$)
r_h	hydraulic radius ($= d_h/4$), m
Re	Reynolds number ($= Gd_h/\mu$)
s	transformed time variable in frequency domain
St	Stanton number ($= h/GC_p$)
t	time coordinate, s
T_m	temperature of the matrix, K
T_g	temperature of the fluid (gas), K
$T_{g,i}$	gas inlet temperature to the packed bed, K
$T_{g,\mathrm{in}}$	instantaneous inlet temperature, K
T_o	ambient temperature, K
$T_{\min}$	minimum of the inlet temperature transient, K
$T_{\max}$	maximum of the inlet temperature transient, K
$T_{g,\mathrm{out}}$	instantaneous outlet temperature, K
x	distance coordinate along the axis, m
x_D	dimensionless distance ($= x/L$)
X_t	dimensionless transverse pitch ($= P_T/d$)
z	dimensionless time (= real time/residence time)
Δp	pressure drop, N/m^2
η	dimensionless charging time ($= \theta N_{tu}$)
θ	dimensionless charging time ($= \dot{m}C_p t/MC$)
Θ_{in}	dimensionless inlet temperature [$= (T_{g,\mathrm{in}} - T_{\min})/(T_{\max} - T_{\min})$]
Θ_{out}	dimensionless outlet temperature [$= (T_{g,\mathrm{out}} - T_{\min})/(T_{\max} - T_{\min})$]
μ	viscosity of the fluid, kg/ms
ρ	density, kg/m^3
ξ	dimensionless distance $= x_D N_{tu}$

REFERENCES

[1] Samdani, G., Fouhly, K., and Moore, S., Heat Exchanger: The Next Wave, *Chem. Eng.*, June, pp. 30–37, 1993.

[2] Das, S. K., Advances in Heat Exchanger Technology, the New Route, *Proc. Workshop on Heat Exchangers*, IIT Madras, 17 August 1996.

[3] Krat, D., Stehlik, P., Van der Prolog, H. J., and Bashir I. Master, Helical Baffles in Shell and Tube Heat Exchanger, Part I: Experimental Verification, *Heat Transfer Eng.*, vol. 17, pp. 93–100, 1996.

[4] Prasad, B. S. V., The Thermal Design and Rating of Multi-stream Plate-Fin Heat Exchangers, in R. K. Shah (ed.), *Compact Heat Exchangers for the Process Industries*, pp. 79–100, Begell House, New York, 1997.

[5] Kays, W. M., and London, A. L., *Compact Heat Exchangers*, 3rd ed., McGraw-Hill, New York, 1984.

[6] Mullisen, R. S., and Loehrke, R. I., A Transient Heat Exchanger Evaluation Test for Arbitrary Fluid Inlet Temperature Variation and Longitudinal Core Conduction, *J. Heat Transfer*, vol. 108, pp. 370–376, 1986.

[7] Heggs, P. J., and Burns D., Single-Blow Experimental Prediction of Heat Transfer Coefficients—A Comparison of Four Commonly Used Techniques, *Exp. Thermal Fluid Sci.*, vol. 1, pp. 243–251, 1988.

[8] Stang, J. H., and Bush, J. E., The Periodic Method for Testing Compact Heat Exchanger Surfaces, *ASME J. Eng. Power*, vol. 96A, no. 2, pp. 87–94, 1974.

[9] Roetzel, W., Luo, X., and Xuan, Y., Measurements of Heat Transfer Coefficient and Axial Dispersion Coefficient Using Temperature Oscillations, *Exp. Thermal Fluid Sci.*, vol. 7, pp. 345–353, 1993.

[10] Lévêque, A., Les jois de la transmission de chaleur par convection, *Ann. Mines. Ser.*, vol. 12, no. 13, pp. 201–415, 1928.

[11] Martin, H., A Theoretical Approach to Predict the Performance of Chevron-Type Plate Heat Exchanger, *Chem. Eng. Processing*, vol. 35, pp. 301–310, 1996.

[12] Schlünder, E. U., Analogy between Heat and Momentum Transfer, *Chem. Eng. Processing*, vol. 37, pp. 103–107, 1998.

[13] Das, S. K., and Sahoo, R. K., Thermodynamic Optimization of Regenerators, *Cryogenics*, vol. 31, pp. 862–868, 1991.

[14] Crump, K. S., Numerical Inversion of Laplace Transforms Using a Fourier Series Approximation, *J. Assoc. Commer. Machinery*, vol. 23, pp. 89–96, 1976.

APPENDIX

Special Functions

The special function $G_n(X, Y)$ known as the Anzelius-Schumann function is given by

$$G_n(X, Y) = e^{-X+Y} \sum_{r=0}^{\infty} \frac{Y^{(n+r+1)}}{(n+r+1)!}$$

$$\sum_{p=0}^{r} \left[\frac{(n+r-p)!}{(r-p)!}\right] \frac{x^p}{p!} \qquad \text{for } n \geq 0$$

The F_n function can be expressed as

$$F_n = e^{-Y} \frac{\partial}{\partial Y} \left(e^Y G_n\right)$$

However, it is easier to evaluate F_n from the recurrence relations,

$$F_n(X, Y) = G_n(X, Y) + G_{n-1}(X, Y)$$

For $n = -1$, G_n can be expressed as

$$G_{-1}(X, Y) = e^{-(X+Y)} \sum_{r=0}^{\infty} \left(\frac{X^r}{r!}\right) \cdot \left(\frac{Y^r}{r!}\right)$$

From the above the functions, $G_o(X, Y)$ and $F_o(X, Y)$ can be expressed as

$$G_o(X, Y) = e^{-(X+Y)} \sum_{q=0}^{\infty} \frac{Y^{q+1}}{q+1} \sum_{p=0}^{q} \frac{X^p}{p!}$$

$$F_o = G_o(X, Y) + G_{-1}(X, Y)$$

Prasant Nanda is a lecturer at Indira Gandhi Institute of Technology—Sarang, Orissa (India). He completed his master's degree at the Heat Transfer and Thermal Power Laboratory of the Indian Institute of Technology—Madras in 1998. The present work is part of his dissertation for Master of Technology in Mechanical Engineering. His research interests include numerical simulation of heat transfer processes, heat transfer analogy, and application of artificial neural network to heat transfer problems.

Sarit Kumar Das is head of the Heat Transfer and Thermal Power Section of the Mechanical Engineering Department at the Indian Institute of Technology—Madras. He received his B.E. and M.E. degrees from Jadavpur University, Calcutta (India), in 1984 and 1987, respectively. He received his Ph.D. degree from Sambalpur University in 1994. He has published one book and more than 40 technical papers in various journals and conference proceedings. He had been a guest professor at the University of Federal Armed Forces, Hamburg (Germany). He is a member of the editorial board of the *International Journal of Heat Exchangers*. His research interests include heat exchangers, dynamic analysis of heat exchangers, nuclear heat transfer, thermal storage, exergy analysis of heat transfer processes, computational fluid dynamics and heat transfer, analogies in heat transfer, and liquid metal heat transfer. He is a Humboldt Fellow for 2000–2001.

Holger Martin has been a professor of thermal process engineering at the Universitaet Karlsruhe (TH), Germany, since 1980. His research interests are in the fields of drying, heat and mass transfer, and heat exchangers. He holds a Dr.-Ing. and a Dr.-Ing. habil.-degree from Karlsruhe University. Since 1980 he has also been the scientific director of the International Seminar (IS) for Teaching and Research in Chemical Engineering and Physical Chemistry at the Universitaet Karlsruhe. Numerous of his original papers on the mentioned research topics have been published in professional journals. Book contributions in *Advances in Heat Transfer, Vol. 13* (1977) on impinging jet flow heat and mass transfer, in the *Heat Exchanger Design Handbook* on transient conduction, fluidized beds, and impinging jets, and in the VDI Waermeatlas on the same topics as well as on plate heat exchangers. Books include *Waermeuebertrager (Heat Exchangers)* (in German), 1988; *Heat Exchangers* (in English), 1992; and, with Ernst-Ulrich Schluender, *Einfuehrung in die Waermeuebertragung (Introduction to Heat Transfer)* (in German), 1995. In 1980 he was awarded the Arnold-Eucken-Preis from GVC-VDI in Strasbourg, and in 1984 he obtained the French-German Alexander-von-Humboldt-Preis, which enabled him a stay of several months as a guest researcher at the Laboratoire des Sciences du Génie Chimique (LSGC) at Nancy, France, which belongs to the French National Scientific Research Centre (CNRS). He also has been a guest professor at the Indian Institute of Technology (IIT) in Madras. Some years ago, he discovered the "Lévêque analogy," a method to calculate heat or mass transfer rates from pressure drop in periodic structures such as the cross-corrugated channels of chevron-type plate heat exchangers, pached beds, tube bundles, etc.