Determination of Effective Thermal Conductivity of Perforated Plates with Low Porosities Used as Matrix Heat Exchanger Core Surfaces

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Abstract—Perforated plate matrix heat exchangers (MHE) are known for providing high effectiveness and high compactness in a single unit. It consist of high thermal conductivity plates separated with low thermal conductivity spacers stacked and bonded together to form passages for working fluid. All perforations in the plates align to form a large number of passages hence providing high heat transfer area per unit volume. The spacers helps in reducing axial conduction there by reducing losses.

The effective thermal conductivity of perforated plates with low porosities are discussed in this work. Plates with Porosity 0.16, 0.22 and 0.30 are considered. The effective thermal conductivity of plates are numerically determined and experimentally validated. These values has been used in the modified maximum slope method used to obtain the heat transfer coefficients of the plate surfaces. Variation of heat transfer coefficient with Reynolds number is obtained for each porosity.

Keywords—low porosity, effective thermal conductivity, single blow transient test, colburn factor

NOMENCLATURE

q	= Heat flux (W/m^2)
K_{eff}	= Effective thermal conductivity (W/mK)
K_p	= Thermal conductivity of plate(W/mK)
T	= Temperature of test section (C)
x	= Distance (m)
λ	= Effective thermal conductivity ratio
p	= Porosity
j	= Colburn factor
NTU	= Number of Transfer Units
Ac	= Total cross sectional area (m)
A	= Total conduction area (m)
Pr	= Prandtl number
Re	= Reynolds number
ρ	= Density (kg/m^2)
μ	= Dynamic viscosity (kg/ms)
v	= Velocity (m/s)
d	= Perforation diameter (m)
ζ	= Non dimensional time
θ	= Dimensionless temperature of fluid at exit section
m	= mass flow rate (kg/s)
C_p	= Specific heat at constant pressure of fluid (J/kgK)
C	= Heat capacity of test section material (J/kgK)
M	= Mass of the test section (kg)
T_0	= Temperature of test section at time ξ=0 (C)
ξ	= Time (s)
t_i	= Temperature of fluid at inlet (C)
t	= Temperature of fluid at any given time (C)

I. INTRODUCTION

Heat exchangers are used for transferring enthalpy between two fluids of different temperature and in thermal contact. The developments in industrial fields has increased the demand for heat exchangers which can provide high effectiveness along with a compact structure. This has lead to the development of perforated plate Matrix Heat Exchangers (MHE). MHEs comes under the classification of compact heat exchangers with surface area density in range of 2000m^2/m^3-6000m^2/m^3. They provide high effectiveness value (90% and above). This is attained by using high thermal conductivity perforated plates separated with low thermal conductivity spacers. The plates and spacers are stacked together to form a single unit and the perforations in the plate together forms a number of flow passage for the fluid as shown in Fig. 1. The plates used are 0.1mm-0.3mm thick and entire unit will have length of 100mm-300mm. Thus high effectiveness and compactness are attained in a single unit. The spacer plates helps in reducing longitudinal heat conduction and provide constant reheadering between plates. The flow within the small holes in the perforated plate is generally laminar. Due to their small hydraulic diameter and the low density of gases, the surfaces are usually operated in the Reynolds number range 500 < Re < 1500.

Many research activities have been conducted on MHEs since it was first introduced in 1949. The sequential development of MHEs have been explained in detail by

Venkataratnam[1]. Nilles et. al.[2] provided a numerical heat exchanger equations along with designing and fabrication of MHE using parameters of application, property matrix material and working fluid parameters.

The performance of heat exchangers are usually studied by effectiveness and NTU method. This convectional method fails in case of MHE due to discontinuity along the longitudinal direction[3]. Single blow transient technique is widely used for determination of heat transfer coefficients in MHEs. In this method a perturbation is given to the inlet temperature and the temperature time response at the outlet is noted. The NTU of MHE is obtained by comparing this response with an appropriate mathematical model. The accuracy of result from this test depends on the accuracy of numerical model. The numerical model should take into account various losses occurring in MHEs.
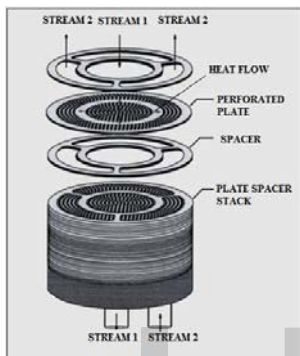

Fig. 1. Schematic of a perforated plate matrix heat exchanger[4]

Schumann [5] obtained an analytical solution for the transient problem. Locke [6] showed that for a step change in the inlet fluid temperature, NTU is a function of the maximum slope of the temperature response curve of the fluid at the outlet without considering longitudinal heat conduction effect. Pucci et. al. [7] improved Locke's analysis by including longitudinal heat conduction. Of the various method used to obtain NTU from transient test data the most widely used is the maximum slope method which expresses NTU as function of maximum slope of response curve. Krishnakumar et. al[8] showed that maximum slope and time at maximum slope of response curve can be used to predict the NTU and the longitudinal conduction parameter simultaneously. This method was used to study how various parameters like porosity, plate thickness, spacer thickness, plate hole diameter etc.. affects the performance of MHE. A detailed review on single blow transient technique has been provided by Anish et.al.[10]

In the present work effective thermal conductivity of plates with low porosities are presented. The effective thermal conductivity of the plates are numerically investigated and experimentally validated. Then single blow transient test was conducted for three porosities with 30 plate spacer pairs. The heat transfer coefficients values in this study were obtained by using the modified maximum slope method presented by K.

Krishnakumar[8]. The effective thermal conductivity values were utilized in the method.

II. DETERMINATION OF EFFECTIVE THERMAL CONDUCTIVITY OF PLATES

A. Numerical Simulation

The numerical simulation was carried out using commercial finite element software ANSYS 14.
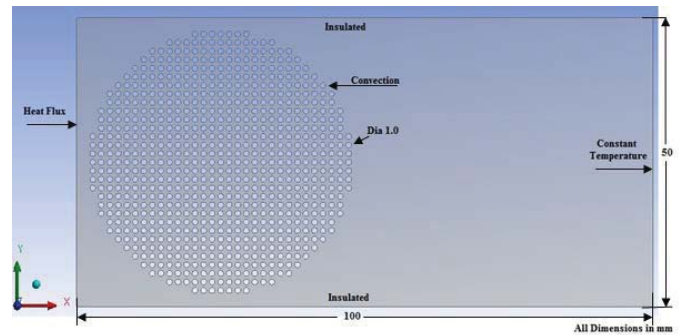

Fig. 2. Geometry

The 3D geometry of plate was inscribed in a rectangle to obtain the geometry for simulation which is shown in fig. 2. The geometry was meshed using solid87 element. After grid sensitivity study a value of 25700 was set for number of elements for meshing. Fig. 3 shows the mesh for porosity 0.3.

A constant heat flux is given at one side and a constant temperature is maintained at the opposite side. The other two sides are insulated as shown in Fig. 2. Convection corresponding to stagnant air is provided inside the perforations. Temperature profile for plates are obtained from numerical simulation.
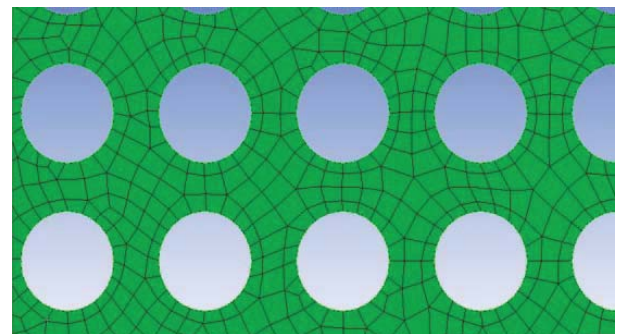

Fig. 3. Mesh

The effective thermal conductivity of plates are obtained by using the temperature readings in the Fourier heat conduction equation given by (1) .

$$q = -K_{eff} \frac{dT}{dx} \tag{1}$$

Here K_{eff} is the effective thermal conductivity, q id the heat flux and dT/dx is the temperature gradient across the plate.

B. Experimental validation

The experimental arrangement is shown in Fig. 4. The circular test specimen is placed in between two rectangular extensions with semicircular ends of same radius. Care is taken to avoid any air resistance between the contact surfaces. The one end of this plate arrangement is wounded with heating coil which is connected to an auto transformer which provides the heat input. The other end of the arrangement is in contact with a cooling water supply which maintain a constant temperature at that end.
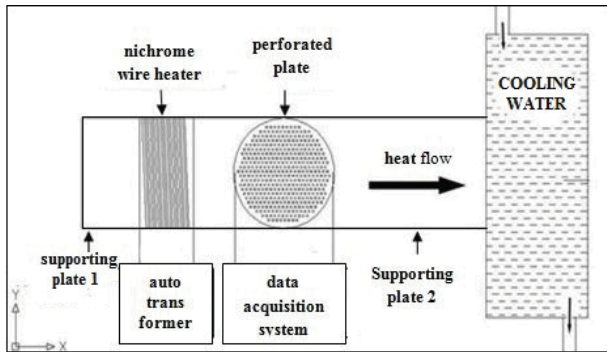

Fig. 4. Experimental arrangement

Two J type thermocouples of size 32 gage are used to measure the temperature at two sections of perforated plate along the diameter in the longitudinal direction. A data acquisition system, Agilent 34970A, was used to record the temperature variation. Asbestos insulation is provided to facilitate heat flow along one direction. The temperature across the plate obtained is used in Fourier heat conduction equation to obtain the effective thermal conductivity of plates.

C. Results

Fig. 5 shows the graph showing effective thermal conductivity ratio value for the two porosities. The equations for thermal conductivity ratio and porosity are given by (2) and (3).

Effective thermal conductivity ratio (λ)

$$\lambda = \frac{K_{eff}}{K_p} \tag{2}$$

K_p is the thermal conductivity of the plate

Porosity (p)

$$p = \frac{void\ volume}{Total\ volume} \tag{3}$$

The values obtained numerically and experimentally are plotted against porosity as shown in Fig. 5. The results show no significant difference between the experimental and numerical results. The values are also in agreement with those obtained from Nilles[2]. The uncertainties in the estimation of effective thermal conductivity was found to be 13.13%.
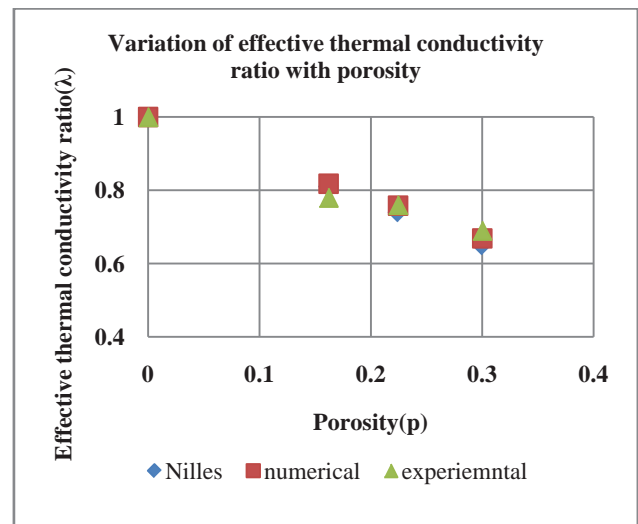

Fig. 5. Effective thermal conductivity ratio for the three porosities.

III. SINGLE BLOW TRANSIENT TEST

A. Experimental setup

Experimental investigations were carried out in an open circuit wind tunnel. The wind tunnel of 50 mm internal diameter is connected to the suction side of a centrifugal blower. A bell mouth is used at the inlet of the tunnel. The flow is made uniform by using flow straightners. The flow through the tunnel is controlled using two butterfly valves. For heating the test matrix, nichrome heater of wire mesh construction are used. The heater is connected to an autotransformer. The rate of flow is measured using a calibrated orifice meter, designed as per ASME standard, which is located downstream of the test matrix. The pressure drop across the test matrix and that across the orifice meter is measured using a micromanometer. The micromanometer has an accuracy of 0.01 mm of methanol. The temperature at the inlet and outlet of the test matrix is measured using K type thermocouples of size 32 gage. A data acquisition system, Agilent 34970A, was used to record the temperature variation. The 30 perforated plates separated with paper spacers of thickness 0.15mm forms the test section which was held together by mechanical pressure, and sealed on the outside using adhesive tapes to prevent air leakage. Fig. 6 shows the experimental setup for single blow transient test.
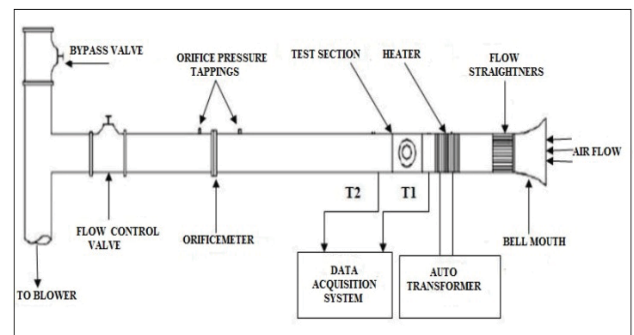

Fig. 6. Experimental setup for transient testing

B. Experimental procedure

The test procedure consists of changing the temperature of the air entering the test section suddenly, and observing the

thermal response at the exit of the test section (heat exchanger). The heat transfer coefficient is obtained from the outlet temperature response. In the present work, a modified maximum slope method, in which both the maximum slope and the time at which maximum slope occurs is used simultaneously to predict NTU experimentally. Experiment was repeated for different Reynolds numbers and porosities.

C. Results and Discussions

The exit temperature of the air and the corresponding time are expressed in dimensionless form using initial conditions and mass flow rate respectively as shown in (4) and (5).

Dimensionless temperature of fluid at outlet(θ)

$$\theta = \frac{t - t_i}{T_0 - t_i} \qquad (4)$$

Dimensionless time(ζ)

$$\zeta = \frac{mC_p}{MC}\xi \qquad (5)$$

Where, t is temperature of fluid at outlet, t_i is the temperature of the fluid at inlet and T_0 is the initial temperature of test section. C_p is the specific heat capacity of air at constant pressure, m is the mass flow rate of air, M is mass of the test section and C is the heat capacity of the test section and ξ is the time.

The NTU can be determined by using the maximum slope and the time at which the maximum slope is obtained using the modified maximum slope method [8]. Heat transfer data for matrix heat exchanger is presented in the dimensionless form of Colburn factor, j, and its variation with different Reynolds number (Re) is obtained. Variation of Colburn factor (j) with Reynolds Number (Re) for different plate porosities are shown in fig. 8.

Reynolds Number (Re)

$$Re = \frac{\rho v d}{\mu} \qquad (7)$$

Here v is velocity of the air, d is the diameter of perforation, ρ is the density of air, μ is the dynamic viscosity of air.
Colburn Factor (j)

$$j = NTU \frac{A_c}{A} Pr^{2/3} \qquad (6)$$

NTU stands for Number of Transfer Units, A_c is the cross section area, A is the conduction area and Pr is the Prandtl number. As Reynolds number increases, the residence time of fluid for heat transfer reduces due to high velocity, the effective heat transfer rate thus decreases and, hence, the 'j'

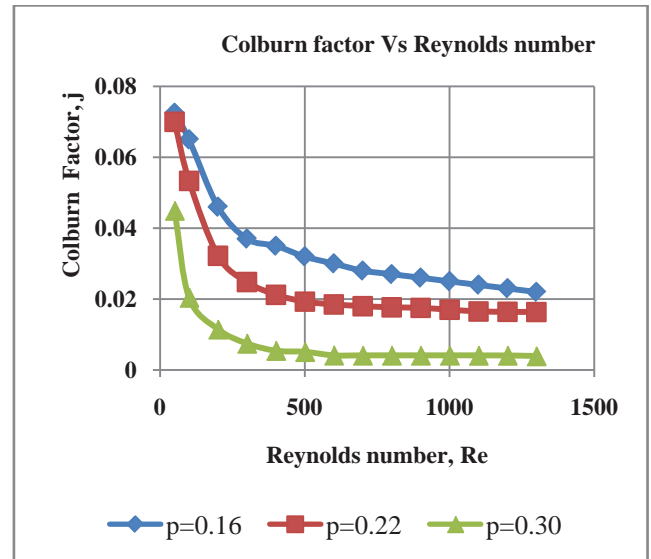value decreases. The uncertainties in estimation of NTU was 10%.

Fig. 8. Variation of colburn factor for different Reynolds number for porosities 0.16, 0.22 and 0.30

IV. CONCLUSIONS

The effective thermal conductivities of the plates were numerically determined and experimentally validated. The single blow transient test were conducted for the three porosity values 0.16, 0.22 and 0.30 with 30 plate separated with paper spacers of 0.15mm. The NTU values were obtained from the exit temperature response for different Reynolds number. The graph of colburn factor versus Reynolds number was plotted.

The results are summarized below:

- The effective thermal conductivity was seen to decrease with increasing porosity. This is because of the reduction in metal area of the plate with increasing porosity.
- The plot of colburn factor versus Reynolds number shows a continuous decrease in colburn factor with increasing Reynolds number. With increase in Reynolds number there is corresponding increase in the velocity of air flow and at higher velocities the residence time for heat transfer was less which reduce the heat transfer coefficients.
- The heat transfer coefficient for plate with porosity 0.16 was higher than that for plate with porosity 0.22 which was higher than that of 0.30. The increase in porosity results in decrease in total heat transfer area of the plate thereby reducing the heat transfer coefficient.

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