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# EXPERIMENTAL STUDIES ON PERFORATED PLATE MATRIX HEAT EXCHANGER SURFACES WITH DIFFERENT PLATE THICKNESS AND SPACER THICKNESS USING MODIFIED MAXIMUM SLOPE METHOD

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## ABSTRACT

The design of perforated-plate heat exchanger is combined with a high thermal conductance in transverse direction with a low thermal conductance in axial direction. Perforated plate matrix heat exchangers (MHE) essentially consist of a stack of perforated plates made of high thermal conductivity material alternating with spacers made of low thermal conductivity material. The single blow transient test is most appropriate for determining the heat transfer coefficients of perforated plate MHEs. Dimensionless temperature and dimensionless time are determined from the exit temperature history of the single blow transient test. The slope is determined from the dimensionless temperature and time plot. The maximum slope and time at which the maximum slope occurs is used in this method for the determination of ntu and longitudinal heat conduction factor simultaneously. The experimentation in this work is to examine the effect of plate thickness to perforation diameter ratio (l/d) and spacer thickness to plate thickness ratio (s/l) on the heat transfer characteristics. The heat transfer and the flow friction characteristics data are presented in the form of Colburn factor (j) and fanning friction factor (f) vs. Reynolds number (Re) respectively. The j and f increase as the l/d value increases. The value of j and f decrease as the s/l value of the plate increases for MHE.

## NOMENCLATURE

| | | |
|---|---|---|
| $A$ | [m$^2$] | Heat transfer area |
| $A_c$ | [m$^2$] | Cross sectional area |
| $C$ | [J/kgK] | Specific heat of the matrix |
| $c_p$ | [J/kgK] | Specific heat of fluid |
| $d$ | [m] | Perforation diameter |
| $f$ | [-] | Fanning friction factor |
| $j$ | [-] | Colburn factor |
| $j/f$ | [-] | Area goodness factor |
| $M$ | [kg] | Mass of the matrix |
| $m$ | [kg/s] | Fluid flow rate |
| ntu | [-] | Number of Transfer Units of plate |
| $G$ | [kg/m$^2$s] | Mass velocity of the fluid in the perforation |
| $L$ | [m] | Length of the heat exchangers |
| Pr | [-] | Prandtl number ($\mu\ c_p$ / k) |
| Re | [-] | Reynolds Number |
| $s$ | [m] | Thickness of spacer |
| $T$ | [K] | Temperature of the matrix |
| $T_0$ | [K] | Temperature of test section at time, $\xi = 0$ |
| $t$ | [K] | Temperature of the fluid |
| $t_{in}$ | [K] | Fluid temperature maintained at the entry of the bed |
| $\Delta p$ | [Pa] | Fluid static pressure drop |
| $\rho$ | [kg/m$^3$] | Fluid density |

**Subscripts**

| | |
|---|---|
| p | Plate |
| f | Fluid |
| in | Inlet |

## INTRODUCTION

The performance of refrigerators, liquefiers and separation units is strongly dependent on the effectiveness of the heat exchangers used. But, if the effectiveness of the heat exchanger is below a certain critical value, [1] most cryogenic processes would cease to function. The low values (0.1 - 0.25) of attainable coefficient of performance (COP) [2] and the resulting high cost of refrigeration make it economically sensible to use a more effective and expensive heat transfer equipment. At low performance levels, heat exchanger effectiveness is largely determined by the heat transfer film coefficients. The design of compact and high-effectiveness cryogenic heat exchangers operating at very low temperature requires consideration of axial heat conduction along the direction of flow. In cryogenic applications, axial conduction through the separating surfaces of the two streams can result in severe performance deterioration. Axial conduction acts to raise the average wall temperature and thus reduces the overall heat exchanger effectiveness [3, 4]. The compactness of plate-fin heat exchangers results in relatively short conduction lengths between the warm and cold ends of the heat exchanger. Therefore, the design of compact and high-effectiveness cryogenic heat exchangers requires careful consideration of axial heat conduction. Longitudinal heat conduction degrades the performance of the heat exchangers significantly for short flow length exchangers designed for the conditions of high effectiveness [5]. Apart from having a high effectiveness, cryogenic heat exchangers also need to be very compact, i.e. they must accommodate a large amount of surface area in a small volume. This helps in controlling heat exchange with the surroundings by reducing exposed surface area. Besides, a small mass means a smaller cooling load and a faster cooling time for refrigerators. This requirement is particularly important for small refrigerators operating at liquid helium temperature.

The necessity of attaining high effectiveness and high degree of compactness together in one unit led to the invention of matrix heat exchangers (MHEs) [6]. Subsequent developments in matrix heat exchangers have largely been directed towards use in helium liquefiers and, more recently, towards low power cryorefrigerators based on Claude and reverse Brayton cycles. Perforated plate matrix heat exchangers (MHE) essentially consist of a stack of perforated plates made of high thermal conductivity material such as copper alternating with spacers made of low thermal conductivity material such as stainless steel [7], plastic etc. as shown in figure 1. The figure 1 [8] shows perforated plate matrix heat exchanger test section. This has only one stream of fluid going through for conducting single blow transient test on the exchanger surfaces. The single blow transient test is used to find the heat transfer coefficient of the plate surfaces. The stack of alternate high and low thermal conductivity materials is bonded to form a monolithic block. The gaps in between the plates ensure uniform flow distribution (by continuous reheadering) and create turbulence which enhances heat transfer. Small perforations (diameter ranging from 1.5 mm to less than 0.4 mm) are made into the plates so that a large heat transfer coefficient and high surface area density (up to 6000 /m) is achieved. The ratio of the plate thickness (length of the hole in the plate) to the diameter of the hole is on the order of 0.75, therefore, the thermal and hydrodynamic boundary layers do not become fully developed within the perforations, which results in high heat transfer coefficients and correspondingly high friction factors [9]. 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 $500 < Re < 1500$ [10]. The single blow transient test experiments are conducted in the laminar region where the characteristic dimension is pore diameter. The correlation for the heat transfer coefficient within the small perforations for laminar flow within tubes is provided by Hubbell and Cain [11]. The spacers, being of low thermal conductivity material, also help in reducing axial conduction and consequent deterioration of performance. The spacers perform multiple roles such as reducing the longitudinal heat conduction through the walls, reducing the flow maldistribution by reheadering the flow in each spacer, interrupting the boundary layer and thus enhancing the heat transfer coefficients. A compact heat exchanger is generally defined as one which incorporates a heat transfer surface having a high area density. It possesses a high ratio of heat transfer surface area to volume. Shah [12] arbitrarily defines a compact heat exchange surface as one that has an area density (β) greater than 700 $m^2$ / $m^3$. A high β decreases volume. The motivation for using compact surfaces is to gain specified heat exchanger performance within acceptably low mass and box volume constraints. The heat exchanger performance depends upon mean temperature difference, and the overall heat transfer heat transfer coefficient U, resulting in a smaller volume. As compact surfaces can achieve structural stability and strength with thinner-gauge material, the gain in a lower exchanger mass is even more pronounced than the gain in a smaller volume. The performance goal for a matrix heat exchanger is to gain maximum heat transfer with minimum pumping power and heat exchanger size. Therefore, for judging and comparing the effectiveness of heat exchanger surfaces that can provide heat transfer in compact heat exchangers, different nomenclatures are used for heat exchanger analysis. The basic performance data for a matrix heat exchanger are often shown as curves of the Colburn factor ($j = St.Pr^{2/3}$), and the Fanning friction factor (f), plotted versus Reynolds number. Kays & London [13] present j and f vs. Re for a large number of compact surfaces, in one of the first comprehensive collections of data on enhanced surfaces for compact heat exchangers. Since that time, j and f curves have become a customary tool for presenting performance data for heat transfer surface geometries. The area goodness method actually makes a direct comparison between j and f values, since it consists of plotting $\frac{j}{f}$ vs. Re. For fully developed laminar flow of a specified fluid, $\frac{j}{f}$ is constant for a given surface, regardless of Reynolds number. Higher $\frac{j}{f}$ is considered desirable because it will require a lower free flow area and hence a lower frontal area for the exchanger [14].

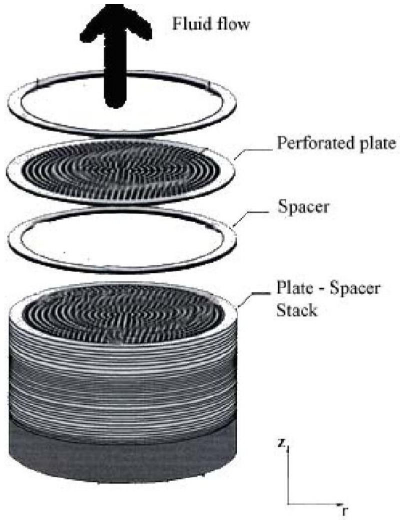


**Figure 1** Perforated plate matrix heat exchanger test section

The formal introduction of the ε -NTU method for the heat exchanger analysis to simplify heat exchanger design problems was introduced in 1942 by London and Seban [15]. But conventional effectiveness NTU approach methods cannot be used for MHEs because of the discontinuity in the heat transfer surface in the axial (longitudinal) direction [16]. New rating and sizing methods have been developed that treat the MHE as a distinct set of plate - spacer pairs. The convective heat transfer in MHE has been considered by a number of authors [17, 18]. The single blow transient test is most appropriate for determining the heat transfer coefficients of perforated plate MHEs. Lot of methods have been developed in literature for calculating the heat transfer coefficients from the single blow test data and to take into account the non-ideal conditions that exist during the testing. The single blow transient test consists of making a flow through a wind tunnel and varying its flow parameters to observe the changes in the flow. In this kind of test usually only one fluid is allowed through the wind tunnel or channel and hence the named as single blow transient test and the characteristics of plate surfaces of the heat exchanger is studied. The convective heat transfer and flow friction characteristics are strongly depending on a number of geometric parameters such as plate porosity, spacer thickness plate thickness, perforation diameter, shape of perforations etc. The porosity of the perforated plate is an important factor that controls the heat transfer coefficient in matrix heat exchangers. It is defined as ratio of fluid flow area to the frontal area of the plate. The overall heat transfer coefficient depends on the convective heat transfer area, which consists of (1) unperforated region of any plate facing the flow (2) the tubular portion of each plate and (3) the unperforated region at the back of each plate, in the wake region of the flow as referred in figure 2. Many researchers [19 - 21], have expressed the heat transfer coefficient as a function of porosity of matrix alone.

The experiments in this work is to examine the effect of plate thickness to perforation diameter ratio (l/d) and spacer thickness to plate thickness ratio (s/l) on the heat transfer characteristics using transient single blow tests are presented. An improved maximum slope method has been used to reduce the test data. Generalized correlations for j and f are presented as a function of Re and different geometric parameters.

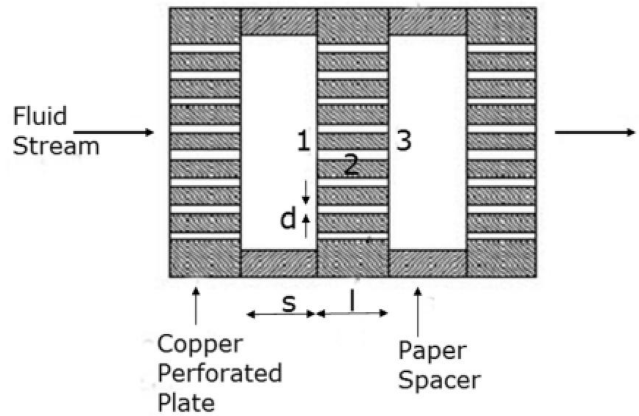


**Figure 2** Convective heat transfer area of perforated plate

## EXPERIMENTAL DESCRIPTION

The experimentations were carried out in an open circuit wind tunnel particularly designed for measuring the heat transfer and flow friction characteristics of perforated plate matrix heat exchanger surface. The schematic diagram of the investigational system is shown in the figure 3. The wind tunnel of 50 mm diameter is coupled to the suction side of a centrifugal blower. A bell mouth is used at the inlet of the tunnel for distortion free suction. The flow straightners compose the flow uniform. The flow from beginning to end of the tunnel is controlled by means of a butterfly valve. For uniform heating the matrix, spirally coiled nichrome coil is used as heater, which is controlled by an autotransformer. The rate of flow is measured by orifice meter using Fluke 992 air flow meter. The temperature at the inlet and outlet of the test matrix is measured using T- type thermocouples of size 32 gauge. The temperature difference between the values measured by different thermocouples is within the measurement accuracy (0.1K). A data acquisition system, Agilent 34970A was used to record the temperature history for the period of transient testing method. The plates were detained jointly by mechanical pressure, and sealed on the outside by means of adhesive tapes to put a stop to air leakage. The entire stack was insulated with a 2 mm thick polyurethane insulation. The plates were stacked randomly.

## EXPERIMENTAL METHODOLOGY

The experiment consists of varying the temperature of air inflowing to the test section abruptly and observing the thermal response at the outlet of the test section (perforated plate matrix heat exchanger). A sudden temperature perturbation is provided at the inlet section of the fluid flow and the temperature history of the single blow transient test method is obtained using a data acquisition system. The velocity profile was noted and it shows that the flow is fully developed in the upstream of the test section. This experiment method normally use single fluid flow throughout the wind tunnel and therefore known as single blow transient test method. The heat transfer coefficient is obtained

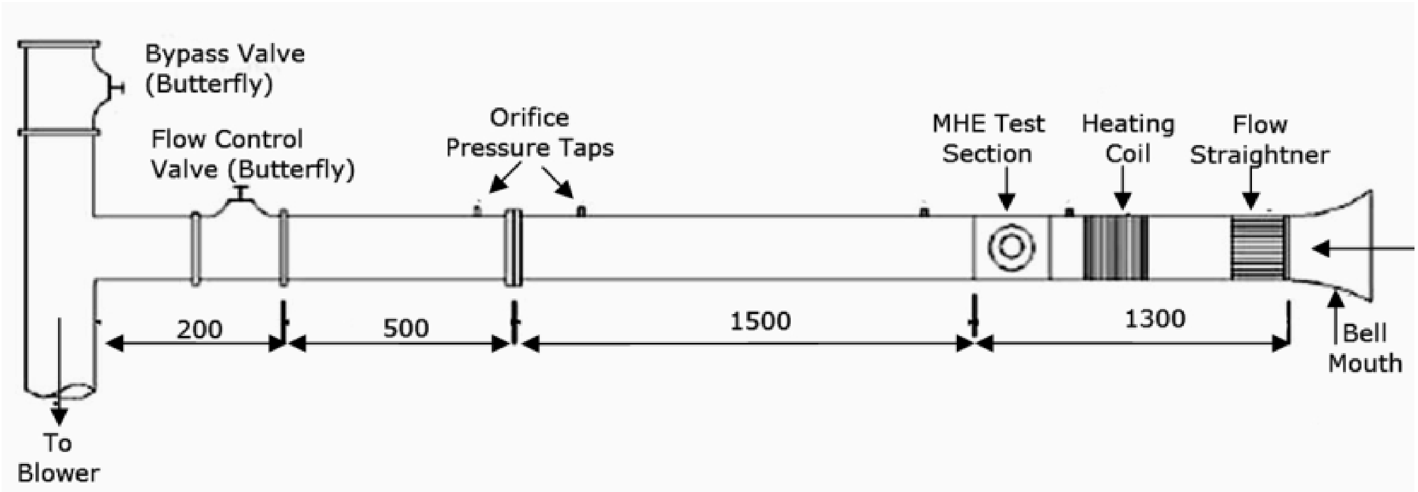


**Figure 3** Experimental setup for single blow transient test

from the exit temperature response. Dimensionless temperature and dimensionless time are determined from the exit temperature history of the single blow transient test. The slope is determined from the dimensionless temperature and time plot. The maximum slope and time at which the maximum slope occurs is noted. In the current work, the modified maximum slope method in which both the maximum slope and the time at which maximum slope is used concurrently to forecast the heat transfer coefficient as well as the longitudinal heat conduction parameter experimentally. The longitudinal heat conduction parameter is experimentally forecasted in this work. The contact resistance between the plate and spacers will differ from experiment to experiment. Heat transfer coefficient of the test core is forecasted by matching the slope and time at maximum slope of the theoretical and exit temperature variations. The experiment in this work is to examine the effect of plate thickness to perforation diameter ratio (l/d) and spacer thickness to plate thickness ratio (s/l) on the heat transfer characteristics. Plate porosity of 0.3 is used in the experimentation. The perforated plates used were 50mm diameter and made of copper using photo chemical milling process.

Paper spacers was used the paper spacers were made using die punch sets. Thirty number of plate - spacer pair is stacked together to form the test matrix. The experiment was conducted for Reynolds number in the range of 50 to 1300. Reynolds number is based on the perforation diameter. A varying input was provided at the inlet section and the exit temperature response curve was obtained. The slope of this curve was calculated and the maximum slope and corresponding time at the maximum slope was noted. In this experiment, modified maximum slope method by Krishnakumar and Venkatarathnam [22] was used to estimate the heat transfer coefficients. Based on the modified maximum slope method the numbers of transfer (ntu) values were obtained. The corresponding Colburn factor (j) was calculated from ntu using governing equations (1-6). The governing equations for the solid and fluid can be obtained in any plate by an energy balance across a control volume as follows, where n denotes no. of plates and j denotes $j^{th}$ plate (equation 1-7):

For n=1,

$$ntu_f \frac{\partial \tau_1}{\partial \varsigma} + \lambda_p (\tau_1 - \tau_2) = \theta_1 - \theta_2 \tag{1}$$

$$\theta_2 = \theta_1 - \varepsilon_p (\theta_1 - \tau_1) \tag{2}$$

For $2 \le j \le n$,

$$ntu_f \frac{\partial \tau_j}{\partial \varsigma} + \lambda_p (2\tau_j - \tau_{j-1} - \tau_{j+1}) = \theta_j - \theta_{j+1} \tag{3}$$

$$\theta_{j+1} = \theta_j - \varepsilon_p (\theta_j - \tau_j) \tag{4}$$

For j=n,

$$ntu_f \frac{\partial \tau_n}{\partial \varsigma} + \lambda_p (\tau_n - \tau_{n-1}) = \theta_n - \theta_{n+1} \tag{5}$$

$$\theta_{n+1} = \theta_n - \varepsilon_p (\theta_n - \tau_n) \tag{6}$$

where $\tau = \dfrac{(T - t_{in})}{(T_0 - t_{in})}$, $\theta = \dfrac{(t - t_{in})}{(T_0 - t_{in})}$,

$$\lambda_p = \frac{k_s A_c}{smc_p},\ \varsigma = \frac{(mc_p)}{(MC)}\xi,\ \varepsilon_p = 1 - \exp(-ntu_f)$$

$$\&\ ntu_f = \frac{(hA)}{(mc_p)} \tag{7}$$

The basic performance data for a perforated plate matrix heat exchanger surface are often shown as curves of the Colburn factor ($j = St.Pr^{2/3}$), and the Fanning friction factor (f), plotted versus Reynolds number. The exit temperature response is used for the experimental determination of ntu and λ. Dimensionless temperature and dimensionless time are determined. The slope curve is determined from the dimensionless temperature and time plot. The max slope and time at which the maximum slope occurs is used in this method for the determination of ntu and longitudinal heat conduction factor.

The heat transfer characteristics data is presented in the form of Colburn factor (j), vs. Reynolds number (Re) and the flow friction data is presented in the form of fanning friction factor (f), vs. Reynolds number (Re). The fanning factor (f) was determined from isothermal pressure drop data and both factors (j and f) were determined from the following expressions (equation 8):

$$j = ntu\frac{Ac}{A}\Pr^{\frac{2}{3}} \text{ and } f = \frac{2\Delta p\rho d}{4LG^2} \quad (8)$$

**EXPERIMENTAL UNCERTAINITIES**

Based on the uncertainties in the measurements, the uncertainty in the estimation of j was 10% and for friction factor, 2.4 %. This was calculated by the Kline and McClintock method [23]. In this work, velocity (± 1%), static pressure (± 1%), fluid temperature (±0.3%) etc. are the measured quantities and Reynolds number (± 1.5 %) and friction factor (± 2.4%) are the calculated quantities. The error in this work is due to the uncertainty in the measurements and the uncertainty in the value of physical properties. The uncertainty bands that may be placed on the reduced data (j, f and Re) are functions of the uncertainties in the measurements made during the experiments.

## RESULTS AND DISCUSSION

Figures 4 and 5 represents j and f vs. Re for copper plate with circular perforation having porosity of 0.3 and l/d of 0.3 but with different spacer thickness to plate thickness ratio (s/l). The j and f decrease as the spacer thickness to plate thickness ratio (s/l) increase. Jet impingement heat transfer decreases with an increase in the distance between the spacer to impingement surface ratio. In the present case, the flow approaching a plate will be compressed and expanded through the holes and impinges on the next plate. Lytle et al. [24] provided a correlation for average local heat transfer coefficients as

$$Nu = 0.424\, Re^{0.57}(\tfrac{s}{l})^{-0.33}.$$

The value of j decreases as the spacing between the plate increases for MHE. An increase in spacer thickness results in the increase in overall volume without any increase in heat transfer area. An increase in spacer thickness leads to an increase in weight of the heat exchanger and a decrease in its compactness. But the overall longitudinal conduction decreases with an increase in spacer thickness. The f value decreases as spacer thickness increases.

Figures 6 and 7 represents j and f vs. Re for copper plate with circular perforation having porosity of 0.3, but with different plate thickness to perforation diameter ratio (l/d). The j and f increase as the plate thickness increases. In MHE, as the thickness of the plate increases, the pore length increases and this leads to an increase in the surface area of perforations, while the front and rear side remains the same. Hence an increase in the plate thickness results in the overall surface area available for heat transfer and leads to a higher value for area averaged heat transfer coefficient. It is also found that f value also increases with l/d ratio.

The values of j as well as f are found to be decreasing as Reynolds number increases. As the values of Re increases, the residence time available for the fluid inside the matrix for transferring heat to the plates reduces and hence the heat transfer coefficient decreases. As the velocity increases, since the friction factor is inversely proportion to the square of the velocity, the friction reduces with increase in Reynolds number. It is found that the area goodness factor, j/f increases with Re as in figure 8 and 9. This shows that at higher Re, performance of the heat exchanger increases. At the lowest Reynolds number region, the flow might be laminar and in the remaining range of Reynolds number, j/f increases as j is more than that of friction factor, f. The area goodness factor (j/f) is inversely proportional to the square of flow area. Since area goodness factor is just a ratio of j and f, there might be variation in the curve depending on the value of j and f. An increase in j/f indicates a decrease in flow area and this leads to a more compactness for the heat exchanger. This aspect is highly desirable for a heat exchanger.

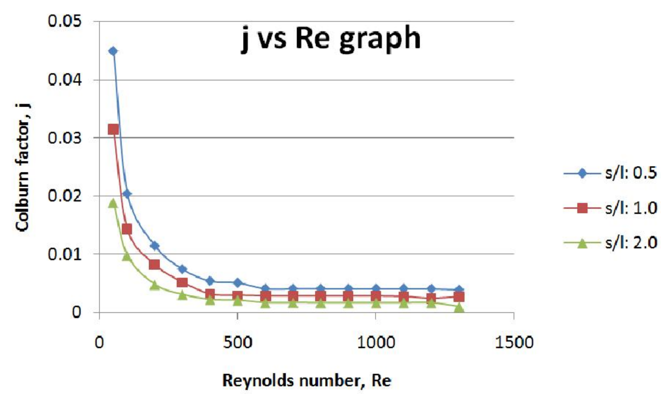


**Figure 4** j vs Re graph for various s/l ratios at constant porosity 0.3

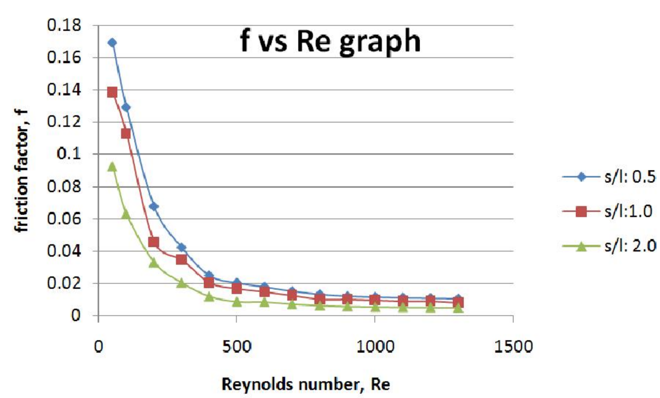


**Figure 5** f vs Re graph for various s/l ratios at constant porosity 0.3

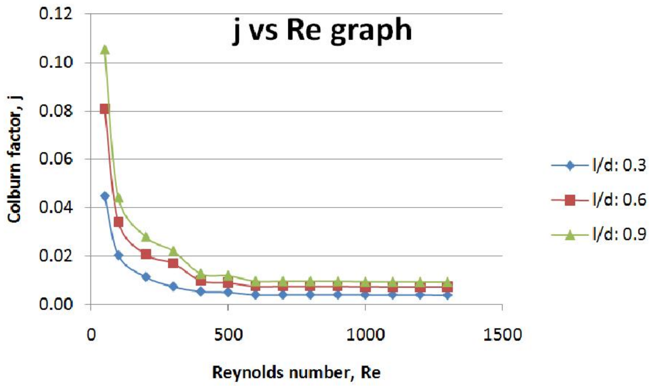


**Figure 6** j vs Re graph for various l/d ratios at constant porosity 0.3

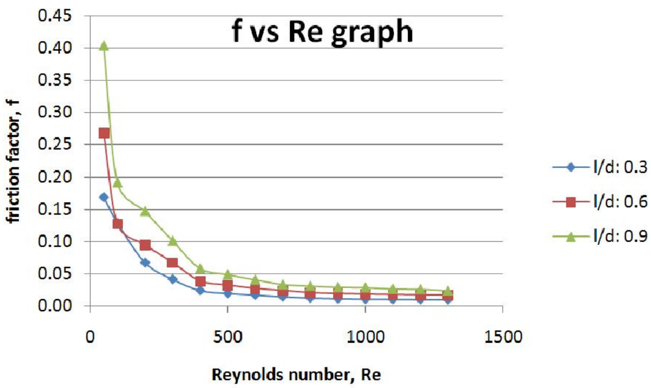


**Figure 7** f vs Re graph for various l/d ratios at constant porosity 0.3

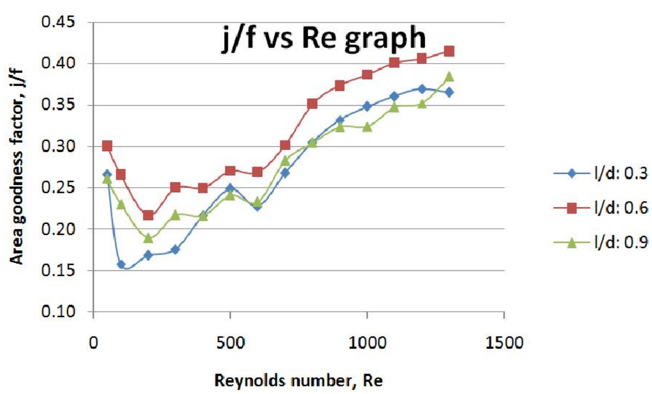


**Figure 8** j/f vs Re graph for various l/d ratios at constant porosity 0.3

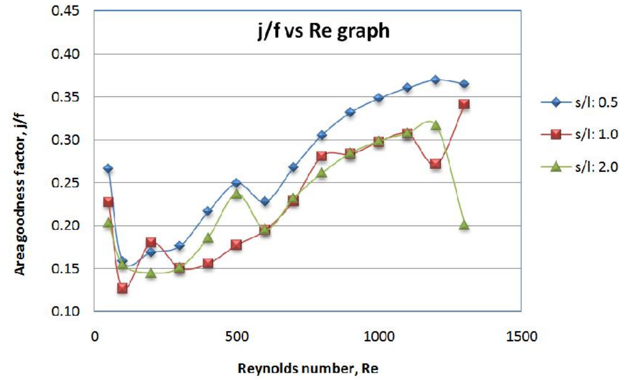


**Figure 9** j/f vs Re graph for various s/l ratios at constant porosity 0.3

The results obtained in this work is compared with that of Krishnakumar et al. [25] and shown in figure 10.

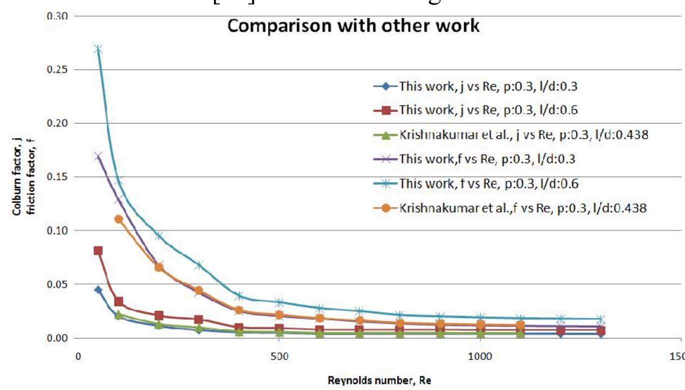


**Figure 10** Comparison of present results of j and f vs Re graph constant porosity 0.3

## CONCLUSION

The effect of s/l and l/d ratios on heat transfer and flow friction characteristics is studied for constant porosity.

1. It is observed that the Colburn factor, j and friction factor, f decrease as the spacer thickness to plate thickness ratio (s/l) increase.
2. As Reynolds number increases, j decreases as the residence time for the heat transfer decreases.
3. The j and f increase as the plate thickness to perforation diameter ratio (l/d) increases.
4. An increase in j/f indicates decreases in flow area and this leads to a more compactness for the heat exchanger.

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