Heat transfer and flow friction correlations for perforated plate matrix heat exchangers

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Abstract. Perforated plate matrix heat exchangers (MHE) are constructed of high conductivity perforated plates stacked alternately with low conductivity spacers. They are being increasingly used in many cryogenic applications including Claude cycle or Reversed Brayton cycle cryo-refrigerators and liqueifiers. Design of high NTU (number of (heat) transfer unit) cryogenic MHEs requires accurate heat transfer coefficient and flow friction factor. Thermo-hydraulic behaviour of perforated plates strongly depends on the geometrical parameters. Existing correlations, however, are mostly expressed as functions of Reynolds number only. This causes, for a given configuration, significant variations in coefficients from one correlation to the other. In this paper we present heat transfer and flow friction correlations as functions of all geometrical and other controlling variables. A Fluent™ based numerical model has been developed for heat transfer and pressure drop studies over a stack of alternately arranged perforated plates and spacers. The model is validated with the data from literature. Generalized correlations are obtained through regression analysis over a large number of computed data.

1. Introduction
A perforated plate heat exchanger (PPHE), schematically shown in figure 1, is constructed of a number of alternately arranged high conducting perforated plates and low conducting spacers [1-2]. Inter-fluid heat exchange takes place in lateral direction through the plates. The spacers help minimize axial conduction and flow mal-distribution. This type of heat exchanger, because of its high surface area density (>3000 m²/m³) and reduced axial conduction losses, can be designed with effectiveness as high as 99% or more. Being compact radiation heat load is also reduced. PPHEs are widely used in various types of cryogenic refrigerators and small capacity helium liquefiers.

Figure 1. Construction of a perforated plate heat exchanger
Heat transfer and flow friction characteristics of a PPHE are very complex. Flow velocity changes in a periodic manner as the fluid flows through the plates and spacers. The developing flow inside the holes leads to high heat transfer in the tubular regions. Flow impingement on the front face and flow separation at the back face of the plates also contribute significantly to the heat transfer. Thermo-hydraulic characteristics also depend on whether the holes of the successive plates are axially aligned (i.e. inline holes) or are arranged in an offset pattern (i.e. shifted holes).

Available correlations are mostly expressed in the form $Nu = c \times Re^a$. Venkatarathnam and Sarangi [3] derived analytical composite correlations using the available expressions for front face, tubular surface and back face of perforated plates. Hu et al. [4] presented separate correlations for front, tubular and back surfaces of the plates. They observed highest heat transfer in the tubular portion of the pores. Krishnakumar and Venkatarathnam [5] obtained expressions for Colburn factor and friction factor as functions of Re, p, t_s/tp and tp/d for some selected domains of the variables. Hayes et al. [6] considered perforated plates as porous medium and numerically obtained Nu as function of Re.

From parametric behavior point of view various authors have reported various findings. Mikulin et al. [7] observed marginal influence of spacer thickness on hydraulic resistance in the operating region $0.25 < t_s/d < 0.3$. However, in the region, $0.375 < t_s/d < 0.49$, Orlov et al. [8-9] observed decreasing hydraulic resistance with increasing $t_s$. They also observed higher hydraulic resistance with the increase of plate thickness. With porosity they found increasing Euler numbers at Re less than 10 and decreasing Euler numbers for Re in between 10 and 100. Using shifted holes arrangement, Mikulin et al. [7] observed insignificant variation in heat transfer when spacer thickness was varied in the region $0.18 < t_s/d < 0.667$. Guo et al. [10] reported insignificant influence of $t_p/d$ on heat transfer in the region $0.5 < t_p/d < 1.1$. On the contrary, Hubble and Cain [11] observed $t_p/d$ as an influencing parameter on both heat transfer and friction factor. Rodriguez and Mills [12] observed increased heat transfer with the increase of $t_s$ and $t_p$ for $1 < Re < 100$, $1.25 < d/t_p < 2.93$, $0.3 < t_s/t_p < 4$ and for shifted holes arrangement.

Above discussion shows that heat transfer and flow friction characteristics of perforated plates strongly depend on the geometrical parameters. Though a number of correlations are available in literature, significant variations in coefficients under given operating conditions can be observed from one correlation to the other. The reason behind this is that most of the correlations do not include all the geometrical parameters. Only a few correlations include some of the geometrical parameters such as $t_s/d$ [7], porosity [9] and hole arrangement [7].

In this paper heat transfer and flow friction correlations for perforated plates which include all geometrical parameters with their wide ranges of applicability are presented. A Fluent™ based numerical model has been developed for thermo-hydraulic studies over a stack of alternately arranged perforated plates and spacers. The model is validated with reported experimental data. A large number of parametric studies were conducted for both inline holes and shifted holes arrangements. Applying regression analyses on the computed data, correlations for Colburn factor and friction factor for wide range domains of the variables are presented.

2. The model

The model is built on a stack of seven circular perforated plates alternately arranged with spacers as shown in figure 2. Thickness of the spacer-2 is used as five times the thickness of the actual spacers. This is to gain some pressure from the large pressure drop that occurs after the first plate. It helps in reaching quick flow stability. Some authors [10] recommend that at least three plates should be there in the upstream of the stack. The fourth plate (P-4) and the fifth plate (P-5) in the stack are the test plates. The other plates are used for flow stabilization and simulating the flow pattern of a PPHE. In order to study the effect of inline and shifted holes two plates are used as the test plates. Major assumptions used in the model are (i) incompressible fluid, (ii) laminar flow, (iii) steady state and (iv) negligible external forces.

For a PPHE, unlike the case of flow through a tube or a channel, it is difficult to define flow as laminar or turbulent based on the Reynolds number. Existing literature also does not provide any information. Usually Reynolds number in practical PPHEs lies in the range of 100 to 1000. Especially,
for very high NTU heat exchangers Reynolds number is often restricted to 100 [13]. For computation and analysis purpose several authors [5,6,11,14-15] have considered laminar flow only. In view of these, in the proposed model laminar flow has been used. The governing equations for the fluid flow under the assumptions made above are

Continuity:
\[ \nabla \cdot \vec{v} = 0 \]  

Momentum:
\[ \rho (\vec{v} \cdot \nabla) \vec{v} = -\nabla P + \mu \nabla^2 \vec{v} \]  

Energy:
\[ \rho C_p (\vec{v} \cdot \nabla) T = k \nabla^2 T \]  

where the operators are

\[ \nabla = \frac{\partial \vec{V}}{\partial x} + \frac{\partial \vec{V}}{\partial y} + \frac{\partial \vec{V}}{\partial z}, \quad \nabla^2 = \frac{\partial^2}{\partial x^2} + \frac{\partial^2}{\partial y^2} + \frac{\partial^2}{\partial z^2} \]

and \( \vec{v} = u \vec{i} + v \vec{j} + w \vec{k} \)

Fluid at a given mass flow rate and at a temperature less than that of the stack is passed through the perforated plates. Nitrogen at 290 K and 0.9 MPa is used as the working fluid. Colburn factor and friction factor are computed from the heat transfer and pressure drop that occur across the test plates. The walls of the test plates and their holes which are in contact with the fluid are defined with a constant wall temperature of 300 K. The wall surfaces of the other plates and the spacers are assumed as adiabatic. This ensures that the heat transfer to take place only through the test plates.

Solutions of the governing equations (1-3) were done by using Fluent™ solver. After completion of the iterations Fluent creates a transcript file containing (i) the heat flow rates at the flow-inlet and flow-outlet and (ii) cell averaged static pressure of the fluid in each spacer volume. Heat transfer rate \( \dot{Q} \) is calculated from the difference between the heat flow at the inlet and at the outlet of the stack.

\[ \dot{Q} = \dot{Q}_{in} - \dot{Q}_{out} \]  

Convection heat transfer coefficient \( h \) is computed from the relation

\[ h = \frac{\dot{Q}}{A_q \, dT} \]  

where heat transfer area \( A_q \) is the sum of front, tubular and back surfaces of the test plates. Average temperature difference between the plates and the fluid \( dT \) is computed from

\[ dT = T_{wall} - T_{avg, fluid} \]  

where \( T_{wall} \) is fixed at a constant wall temperature of 300K.
\[ T_{\text{av, fluid}} = 0.5(T_{\text{in}} + T_{\text{out}}) \quad (7) \]

\[ T_{\text{out}} = T_{\text{in}} + \frac{Q}{(m \cdot C_p)} \quad (8) \]

Fanning friction factor \( f \) is computed from the pressure drop \( \Delta P \) across the test plates using

\[ f = \frac{2 \Delta P \cdot \rho \cdot d}{4 L_p \cdot G^2} \quad (9) \]

where \( \Delta P = P_4 - P_6 \). Pressures \( P_4 \) and \( P_6 \) are the volume averaged total static pressure in the 4th and 6th spacer region respectively. Assuming \( P_4 \) and \( P_6 \) to represent the pressure at the mid-point of their corresponding spacer, the length \( L_p \) is taken as \( L_p = 2(t_p + t_s) \).

3. Solution and model validation

Mesh file was created by using Gambit (ver 2.2.30) and Fluent (ver 6.2.16) was used for obtaining the solution. Three-dimensional pressure based solver with semi-Implicit Method for Pressure Linked Equations (SIMPLE) pressure-velocity coupling scheme was used in all computations. Second order upwind discretization method was used in both energy and momentum equations. A node size between 0.25 and 0.3 showed grid independency solutions and hence used.

![Comparison of predicted Colburn factor with experimental data [8].](image1)

**Figure 3.** Comparison of predicted Colburn factor with experimental data [8].

![Comparison of predicted friction factor with experimental data[8].](image2)

**Figure 5.** Comparison of predicted friction factor with experimental data[8].

![Comparison of predicted Colburn factor with experimental data [9].](image3)

**Figure 4.** Comparison of predicted Colburn factor with experimental data [9].

![Comparison of predicted friction factor with experimental data[9].](image4)

**Figure 6.** Comparison of predicted friction factor with experimental data[9].
The model is validated with literature data under identical parametric conditions. Figures 3 and 4 show the comparison for Colburn factors. Excellent matching in the both the figures shows the accuracy of the present model. Comparisons for friction factor are shown in figures 5 and 6 where good matching is also seen for most of the data. In the later part of this paper we have presented correlations under varying parametric conditions. Figures 7 and 8 compare the correlation data with the experimental data [13]. Deviations seen in these figures are mainly because of the error associated with the regression analyses while developing the correlations. In some cases such as in figures 5, 6 and 7, we find that in the region Re < 30 predicted data are as much as 30% less than the experimental data. This can be reasoned from the followings. For obtaining low Reynolds number usually mass flow rate is reduced in the same test stack. At low mass flow rates, because of low velocity, there is a possibility of flow mal distribution in the perforations of the plates. Also at higher NTU, because of low mass flow rate, data reduction using measurements at steady state can have significant errors [13].

4. Parametric studies

Heat transfer and flow friction studies on perforated plates under varying parametric conditions were conducted by using the proposed model. Domains of the variables are decided from the practical applicability point of view. Table 1 presents the levels of the variables used in the studies and table 2 shows the values of the geometric variables used in each case.

![Figure 7. Comparison of predicted Colburn factor with experimental data [13].](image1)

![Figure 8. Comparison of predicted friction factor with experimental data [13].](image2)

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**Table 1. Levels of variables used in heat transfer and flow friction studies**

| Variables | Levels |
|-----------|--------|
| Re        | 10,30,50,100,300,600,1000 |
| p         | 0.2, 0.28, 0.36 |
| t₀/p | 0.5, 0.9, 1.3 |
| t₀/t₀ | 0.4, 1.2, 2.0 |

**Table 2. Geometrical details of the variables used in the computations (porosity for the Set Nos. in each row refers to 0.2, 0.28 and 0.36 respectively).**

| Set Nos. | t₀/t₀ | t₀/p | t₀, t₀, d (all in mm) |
|----------|-------|------|-----------------------|
| 1, 2, 3  | 0.4   | 0.5  | 0.3, 0.75, 1.5        |
| 4, 5, 6  | 0.4   | 0.9  | 0.4, 1.1, 1.11        |
Correlations

Generalized correlations for Colburn factor and friction factor have been derived through regression analysis over the large number of data that were generated by running the program. Because of the large variation of data, a single correlation for Colburn or friction factor cannot cover the entire domain of the variables effectively. Therefore, separate correlations are developed for suitably identified domains. Data obtained by using these correlations are in general within ±10% of the predicted data. For convenience, the correlations for Colburn factor and friction factor are expressed in the forms:

\[
  j = a \text{Re}^b p^c (t_s/t_p)^d (t_p/d)^e \\
  f = a' \text{Re}^{b'} p^{c'} (t_s/t_p)^{d'} (t_p/d)^{e'}
\]

(10)

(11)

Coefficients and indices for equations (10) and (11) are given in table 3 and table 4 respectively. The correlations, for example, in the regions of 0.28<p<0.36, 0.5<t_p/d<0.9, 0.4<t_s/t_p<2.0, 10<\text{Re}<100 and for shifted holes can be obtained from the tables as

Colburn factor (j) \[ j = 0.19783 \text{Re}^{-0.17581} p^{0.22912} (t_s/t_p)^{-0.03123} (t_p/d)^{-0.46162} \]
Friction factor (f) \[ f = 2.28303 \text{Re}^{-0.63598} p^{-0.65193} (t_s/t_p)^{-0.87883} (t_p/d)^{-1.04855} \]

Table 3. Coefficient and indices for computing Colburn factor from equation (10).

| Domain | Coefficients |
|--------|--------------|
|        | a           | b            | c            |
| 0.2<p<0.28 | 0.89757 | -0.42872 | 0.8601 |
| 0.5<t_p/d<0.9 | 0.41777 | -0.10938 | 0.99175 |
| 0.4<t_s/t_p<2.0 | 3.54092 | -0.7415 | 1.10721 |
| 0.2<p<0.28 | 0.47963 | -0.30806 | 0.73899 |
| 0.9<t_p/d<1.3 | 0.25991 | -0.11799 | 0.8610 |
| 0.4<t_s/t_p<2.0 | 2.92036 | -0.74187 | 0.81267 |
| 0.28<p<0.36 | 0.80959 | -0.43783 | 0.86999 |
| 0.5<t_p/d<0.9 | 0.19783 | -0.17581 | 0.41777 |
| 0.4<t_s/t_p<2.0 | 3.42225 | -0.74434 | 0.80802 |
| 0.28<p<0.36 | 0.47192 | -0.31752 | 0.69446 |
| 0.9<t_p/d<1.3 | 0.26974 | -0.16144 | 0.54013 |
| 0.4<t_s/t_p<2.0 | 2.79911 | -0.74475 | 0.6365 |
| 0.28<p<0.36 | 1.05052 | -0.53929 | 0.28104 |
| 0.9<t_p/d<1.3 | 3.0225 | -0.75503 | 0.1375 |
| 0.4<t_s/t_p<2.0 | -0.11495 | -0.09384 |
Table 4. Coefficient and indices for computing friction factor from equation (11).

| Domain | Coefficients |
|--------|--------------|
| a      | b            | c            | e            | γ       |
| 0.2<ρ<0.28 | 10<Re<100   | Inline       | Shifted      | 9.02614 | -0.95224 | -0.04137 | -0.25485 | -0.10933 |
| 0.5<tp/d<0.9 | 100<Re<600  | Inline       | Shifted      | 9.09171 | -0.9393  | -0.02788 | -0.1671  | -0.0816 |
| 0.4<tp/t<2.0 | 600<Re<1000 | Inline       | Shifted      | 0.32525 | -0.14483 | -0.40971 | -0.81567 | -1.29798 |
| 0.2<ρ<0.28 | 10<Re<100   | Inline       | Shifted      | 18.40001 | -0.98507 | 0.15816  | 0.08328  | 0.31909 |
| 0.9<tp/d<1.3 | 100<Re<600  | Inline       | Shifted      | 0.1676  | -0.04411 | -0.42463 | -0.79592 | -1.34249 |
| 0.4<tp/t<2.0 | 600<Re<1000 | Inline       | Shifted      | 11.40017 | -0.84317 | 0.21937  | -0.03155 | 0.18184 |
| 0.28<ρ<0.36 | 10<Re<100   | Inline       | Shifted      | 9.17919 | -0.94343 | -0.00258 | -0.32659 | -0.07839 |
| 0.5<tp/d<0.9 | 100<Re<600  | Inline       | Shifted      | 4.07195 | -0.64513 | -0.22499 | -0.64139 | -0.79241 |
| 0.4<tp/t<2.0 | 600<Re<1000 | Inline       | Shifted      | 7.63551 | -0.86794 | 0.09176  | -0.16621 | -0.00222 |
| 0.28<ρ<0.36 | 10<Re<100   | Inline       | Shifted      | 7.79816 | -0.84317 | 0.21937  | -0.03155 | 0.18184 |
| 0.9<tp/d<1.3 | 100<Re<600  | Inline       | Shifted      | 0.18043 | -0.07277 | -0.51224 | -0.61859 | -1.2427 |
| 0.4<tp/t<2.0 | 600<Re<1000 | Inline       | Shifted      | 9.60726 | -0.93273 | 0.04514  | -0.21538 | -0.09067 |
| 0.28<ρ<0.36 | 10<Re<100   | Inline       | Shifted      | 2.28303 | -0.63598 | -0.65193 | -0.87883 | -1.04855 |
| 0.5<tp/d<0.9 | 100<Re<600  | Inline       | Shifted      | 11.64683 | -0.91693 | 0.1949  | -0.06336 | 0.11127 |
| 0.4<tp/t<2.0 | 600<Re<1000 | Inline       | Shifted      | 0.18539 | -0.16787 | -0.98182 | -0.78421 | -1.20422 |
| 0.28<ρ<0.36 | 10<Re<100   | Inline       | Shifted      | 13.10175 | -0.88147 | 0.41331  | 0.103    | 0.36155 |
| 0.9<tp/d<1.3 | 100<Re<600  | Inline       | Shifted      | 0.07531 | -0.05266 | -1.1392  | -0.78131 | -1.2824 |
| 0.4<tp/t<2.0 | 600<Re<1000 | Inline       | Shifted      | 8.79085 | -0.91662 | 0.01968  | -0.26871 | -0.10583 |
| 0.28<ρ<0.36 | 10<Re<100   | Inline       | Shifted      | 4.03619 | -0.67339 | -0.30584 | -0.61283 | -0.70604 |
| 0.9<tp/d<1.3 | 100<Re<600  | Inline       | Shifted      | 7.4524  | -0.83956 | 0.19494  | -0.1378  | -0.02804 |
| 0.4<tp/t<2.0 | 600<Re<1000 | Inline       | Shifted      | 0.37778 | -0.22983 | -0.67963 | -0.59997 | -1.03712 |
| 0.9<tp/d<1.3 | 100<Re<600  | Inline       | Shifted      | 5.51285 | -0.74786 | 0.44275  | 0.01927  | 0.15415 |
| 0.4<tp/t<2.0 | 600<Re<1000 | Inline       | Shifted      | 0.13392 | -0.09066 | -0.83932 | -0.60648 | -1.23125 |

6. Conclusion
In this paper we have studied thermo-hydraulic characteristics of perforated plates for using in matrix heat exchangers. The plates are arranged with a spacer in between two plates. A large number of studies were conducted numerically by varying the parameters such as plate thickness, spacer thickness, pore diameter, porosity, Reynolds number etc. Inline holes as well as shifted holes arrangement in axial direction was also considered in the study. The numerical model used in this work was validated with the reported experimental data under identical parametric conditions. Through regression analyses over the computed data, correlations for Colburn factor and friction factor for suitably identified domains of the controlling variables are presented.

7. Nomenclature

A_q : heat transfer area in test plates (m^2)  
A_s : flow area in spacers (m^2)  
C_p : specific heat of fluid at constant pressure (J/kgK)  
d : pore diameter (m)  
f : friction factor  
G : fluid mass velocity in perforations (kg/m^3)  
h : heat transfer coefficient(W/m^3K)  
Q : heat transfer (W)  
Re : Reynolds number, G d / μ  
T_{av, fluid} : average fluid temperature (K)  
t_p : plate thickness (m)  
t_s : spacer thickness (m)  
u : velocity in x direction (m/s)  
v : velocity in y direction (m/s)  
\vec{v} : velocity field vector
\( j \): Colburn factor, \( Nu / \left( Re Pr^{1/3} \right) \)  
\( k \): thermal conductivity of fluid (W/mK)  
\( m \): mass flow rate (kg/s)  
\( Nu \): Nusselt number, \( h_d/k_f \)  
\( n \): number of holes in perforated plate  
\( P_i \): fluid pressure in \( i \)th spacer region (Pa)  
\( \Delta P \): pressure drop (Pa)  
\( p \): porosity  

\( w \): velocity in \( z \) direction (m/s)  

Greek symbols  
\( \mu \): dynamic viscosity of fluid (Pa-s)  
\( \rho \): density (kg/m\(^3\))

Subscripts  
\( in \): inlet to the stack  
\( out \): outlet from the stack  
\( wall \): at the wall

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