International Journal of Thermal Sciences 120 (2017) 459–468

Contents lists available at ScienceDirect

International Journal of Thermal Sciences

journal homepage: www.elsevier.com/locate/ijts

New design equations for turbulent forced convection heat transfer and pressure loss in pillow-plate channels

M. Piper [a], A. Zibart [a], E.Y. Kenig [a, b, *]

[a] Chair of Fluid Process Engineering, Faculty of Mechanical Engineering, University of Paderborn, Paderborn, Germany
[b] Gubkin Russian State University of Oil and Gas, Moscow, Russian Federation

ARTICLE INFO

Article history:
Received 10 February 2017
Received in revised form
5 May 2017
Accepted 14 June 2017
Available online 10 July 2017

Keywords:
Pillow-plate heat exchanger
Design methods
Pressure loss
Heat transfer

ABSTRACT

Pillow-plate heat exchangers (PPHE) represent a stack of pillow plates characterized by a wavy surface and fully welded construction. The benefits of PPHE over conventional heat exchangers have made them increasingly attractive for numerous applications in the process industry. However, the lack of verified design methods and reference applications for PPHE hinders their widespread application. This work aims at overcoming this "bottleneck" by providing new equations for the thermo-hydraulic design of pillow plates.

In particular, design methods for pressure loss and heat transfer are proposed for turbulent forced convection in pillow-plate channels over a wide range of Reynolds numbers ($1000 \leq Re \leq 8000$), Prandtl numbers ($1 \leq Pr \leq 150$) and characteristic geometry parameters (e.g., welding spot arrangement, inflation height). For the determination of heat transfer coefficients, two different approaches are presented. The first approach is based on the method typically suggested in literature, whereby the well-known Dittus-Boelter type power law function (cf. [1]) for the Nusselt number is applied and fitted to numerical data largely obtained in Ref. [2]. The second approach is based on the characteristic flow pattern in pillow plates suggested in Ref. [2]. In this pattern, the initial complex flow is broken down into two simpler flows, which can then be modeled separately. For both methods, the relative deviations between the numerical data obtained in Ref. [2] and in this study are less than 15%.

1. Introduction

Pillow-plate heat exchangers (PPHE) have gained increased attention in the process industry as a promising alternative to conventional heat transfer equipment, such as shell-and-tube heat exchangers (STHE). The benefits of PPHE include the fully-welded construction (high leak tightness), light weight, large operating pressure and temperature ranges, high geometrical flexibility and adaptability, high overall heat transfer coefficients (values higher than in STHE and close to those of plate heat exchangers) and easy manufacturing. PPHE are commonly encountered as top condensers in distillation columns, but they are also used as gas coolers and evaporators. The implementation of innovative equipment in production plants is commonly a lengthy process. Reference applications demonstrating advantages and limitations of the new technology are required. Furthermore, verified design methods must be available. At the moment, however, such design methods for PPHE are still lacking in literature, and this hinders their widespread application.

Mitrovic and Peterson [3] were the first to publish experimental results on forced convection heat transfer in a pillow-plate channel. Based on the measured data, they developed an empirical Dittus-Boelter type correlation for heat transfer coefficients. This correlation rested on measurements of only one pillow-plate geometry. However, variability of the characteristic geometry parameters of pillow plates shown in Fig. 1 is practically unlimited.

Mitrovic and Maletic [4] tried to develop more universal design methods for pillow plates and to gain a more detailed understanding of fluid dynamics and heat transfer in pillow-plate channels. They performed a comprehensive CFD study over a wide range of pillow-plate geometries. Similar to [3], they developed an empirical Dittus-Boelter type Nusselt correlation, which was able to approximate their numerical data with an accuracy of ±10%. However, they used certain simplifications regarding both the geometrical representation of the wavy pillow-plate surface based

* Corresponding author. Chair of Fluid Process Engineering, Faculty of Mechanical Engineering, University of Paderborn, Paderborn, Germany.
E-mail address: eugeny.kenig@upb.de (E.Y. Kenig).

http://dx.doi.org/10.1016/j.ijthermalsci.2017.06.012
1290-0729/

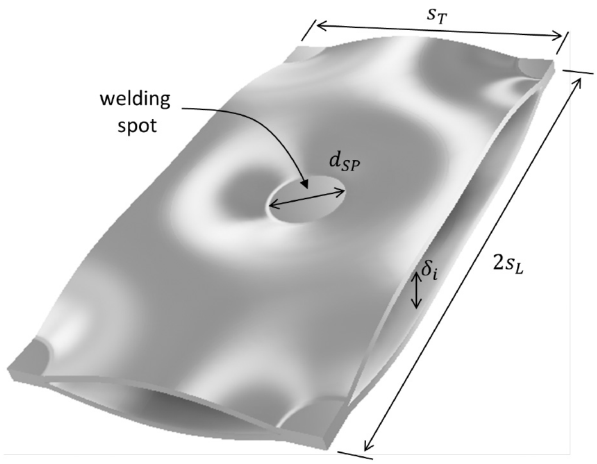

Fig. 1. Periodic section of a pillow plate with characteristic geometry parameters used for representing the pillow-plate geometry.

on trigonometric functions and flow description. Meanwhile an alternative approach based on forming simulations has proven to yield a more accurate reproduction of the real pillow-plate surface (Piper et al. [5]).

Furthermore, in Ref. [4] a laminar model was used for the investigation of fluid flow and heat transfer, even at Reynolds numbers at which the flow was turbulent. As mentioned by Maletic [6], the use of the laminar model resulted in an underestimation of pressure loss and especially heat transfer coefficients, as compared to the measurements carried out in Ref. [3]. Maletic [6] also performed simulations with the standard $k-\varepsilon$ turbulence model; however, this resulted in a significant over-prediction of heat transfer coefficients. Moreover, the Prandtl number was kept constant in the simulation studies in Ref. [4], and thus, the dependency of the Nusselt number on the Prandtl number in pillow-plate channels could not be captured.

In Ref. [2], a comprehensive CFD study of turbulent forced convection heat transfer and fluid flow in pillow-plate channels was performed for a wide range of pillow-plate geometries. Forming simulations were used to generate the real wavy pillow-plate surface, while Reynolds stresses in the turbulent flow were computed using the realizable $k-\varepsilon$ turbulence model of Shih et al. [7]. The CFD simulations were then successfully validated against experiment. The deviation between calculated and measured values was less than 5% for pressure loss and less than 15% for heat transfer coefficients. In Ref. [2], the Reynolds number was varied between 1000 and 8000 and the Prandtl number between 1 and 150. This allowed the dependency of the Nusselt number on both the Reynolds and Prandtl numbers to be determined.

In this work, new equations are proposed for the thermo-hydraulic design of pillow plates. These equations were developed using results obtained by CFD simulations in Ref. [2] complemented by further numerical data generated in this work (using the methods described in Ref. [2]), in order to extend the range of applicability to a larger variety of characteristic geometry parameters $2s_L$, s_T, δ_i and d_{SP}. The additional geometries investigated here are summarized in Table 1.

For the determination of heat transfer coefficients, two different methods are proposed. In the first method, a typical Dittus-Boelter type power law function [1] for the Nusselt number is adopted. The second approach is based on the analysis of the characteristic flow pattern in pillow-plate channels.

It is worth noting that the terminology used in this article is largely the same as in Ref. [2]. Therefore, it is advisable to keep Ref. [2] handy or to have read it first.

Table 1
List of pillow-plate geometries investigated in this study. The parameter d_h represents the hydraulic diameter of the pillow-plate channel (cf. [5]).

$2s_L$	s_T	δ_i	d_{SP}	d_h
mm	mm	mm	mm	mm
42	72	6	7.2	9.13
42	72	6	8.6	8.89
42	72	4.5	10	6.55
42	42	6	7.2	7.18
72	42	6	7.2	9.13
72	42	6	8.6	8.89
72	42	4.5	10	6.55

2. Design equation for pressure loss

Pressure loss is commonly represented by the dimensionless Darcy friction factor $\zeta_{\Delta p}$ [8], which is defined by the following equation:

$$\zeta_{\Delta p} = \frac{2d_h}{\rho u_m^2}\frac{\Delta p}{L} \tag{1}$$

In Eq. (1), d_h denotes the hydraulic diameter of the channel, while u_m represents the mean stream velocity. Often, $\zeta_{\Delta p}$ is a function of the Reynolds number Re and is represented by a curve with quasi-asymptotic behavior, as shown in Fig. 2. This curve is valid for turbulent forced convection in a typical pillow-plate channel, with the characteristic flow in it shown in Fig. 3.

Such a curve can be approximated with a power-law function:

$$\zeta_{\Delta p} = n_1 Re^{n_2} \tag{2}$$

This quasi-asymptotic behavior can be explained, if different contributions of the two constitutes of the Darcy friction factor, namely the friction drag coefficient and the form drag coefficient, are considered. While the coefficient of form drag depends only weakly on Re in most pillow plate geometries [2], the friction drag coefficient shows an asymptotic behavior resulting in an asymptotic behavior of $\zeta_{\Delta p}$ (cf. Eq. (2)).

Form drag arises from recirculation zones appearing in the wake of the welding spots (cf. Fig. 3). Typically, the size of these zones

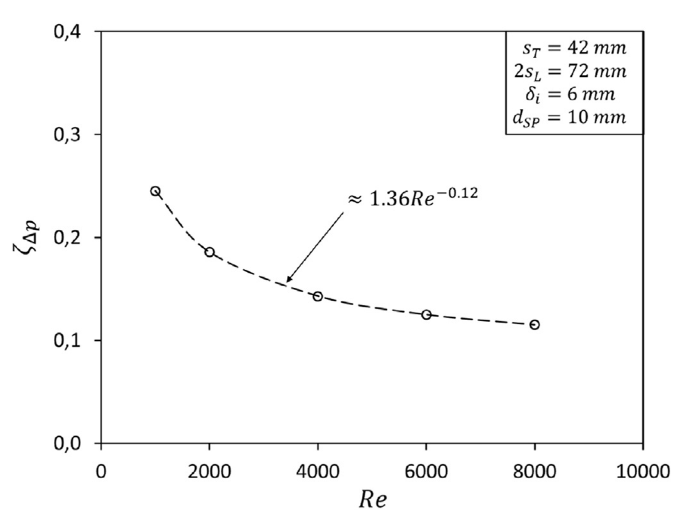

Fig. 2. Darcy friction factor $\zeta_{\Delta p}$ as a function of the Reynolds number for turbulent, single-phase flow in a longitudinal-type pillow plate (cf. [2]).