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Effect of end plates on heat transfer of plate heat exchanger

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ABSTRACT

Steady-state heat transfer data for single-phase (water) in both frame-and-plate (FPHE) and brazed plate heat exchangers (BPHE) are presented with various number of chevron plates in U-type flow arrangement. Analysis of the experimental results indicates that the end plates, instead of being adiabatic, function as fins due to the contact between adjacent plates. The experimental data is used to validate a thermal conduction model in ANSYS, which indicates that the end plates fin efficiency is a function of fluid convective heat transfer coefficient and conductive thermal resistance. In the FPHE, the pressing force of the frame may affect the contact thermal resistance, thus change the fin efficiency. In BPHE, the fin efficiency is much higher due to the larger contact area and higher conductivity of the brazing material. Although the effect of end plates is quickly diluted by the increased number of plates in real applications, it could be significant when plate number is small, as is often the case in laboratory settings for the development of heat transfer correlations.

1. Introduction

Plate heat exchangers are typically categorized in three types: frame-and plate, brazed plate and shell-and-plate. Frame-and-plate heat exchanger (FPHE) is commonly used for their ease of cleaning, simple adjustment of heat transfer area, compactness and excellent thermal-hydraulic performance [1]. It essentially consists of multiple thin metal plates that are stamped with a wavy chevron or herringbone pattern. Fluid channels are formed by pressing the plates with opposite chevron direction together. The alternating flows are directed and sealed by the gaskets in between. The contact points between crests and troughs of two adjacent plates subdivide the fluid path into an array of interconnected unitary cells, which turbulate the flow and enhance heat transfer.

Early applications of FPHE are mainly for liquid-liquid heat transfer in the lower pressure range (usually below 1.6 MPa), including dairy, pulp and paper industries for their hygiene requirements [1]. With the introduction of brazed plate heat exchanger (BPHE), such plates could withstand higher pressure and later on found its increasing application as condenser and evaporator in air-conditioning and refrigeration systems.

Numerous studies have been carried out to measure single-phase and two-phase flow heat transfer, as summarized in review articles [2] and textbooks [3]. However, only a few have investigated the effect of end plates, which are referred to as the "two outer plates" and "ideally do not transfer heat" in most of the open literature [1,3,4]. Meanwhile most manufacturers only count the interior plates, known as thermal plates, as active heat transfer area.

Nevertheless, the effect of end plates is not always trivial. For instance, Heggs and Scheidat [5] recommended 19 plates for the end plates effect to be less than 2.5%. To characterize and compensate such effect, most work in open literature have taken the method of adding a correction factor on log mean temperature difference (LMTD) or plot ε-NTU for different configurations and operating conditions. In 1961, Buonopan et al. [6] experimentally determined the correction factor F for 1pass-1pass flow arrangement with up to 17 thermal plates and multi-pass series flow arrangements with up to 11 thermal plates. In a similar manner, Foote [7], Usher [8] and Marriott [9] presented the F factor as a function of thermal plates for various configurations in the late 1960s. Jackson and Troupe [10], and Kandlikar [11] used numerical method to analyze the ε-NTU relationship in various number of plates. In 1988, a more comprehensive study was carried out by Kandilikar and Shah [4], who investigated the influence of the number of thermal plates on plate heat exchanger performance through numerical analysis for 1pass-1pass, 2pass-1pass and 3pass-1pass flow arrangement. The correction factor F for LMTD

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Nomenclature

A	area (m^2)
b	plate thickness (mm)
C	heat capacitance (kW K)
Cp	specific heat (kJ kg^{-1} K^{-1})
f	Fanning friction factor
D_e	equivalent diameter (m)
G	mass flux (kg m^{-2} s^{-1})
h	heat transfer coefficient (W m^{-2} K^{-1})
k	conductivity (W m^{-2} K^{-1})
L	length (mm)
$\dot{m}$	mass flow rate (kg s^{-1})
N	number of plates
Nu	Nusselt number
NTU	number of transfer unit
P	pressure (kPa)
Pr	Prandtl number
Q	capacity (kW)
Re	Reynolds number

Subscripts

avg	average
b	bulk
c	cold
cs	cross section
f	fin
h	hot
i	inlet
o	outlet

Greek letters

ε	effectiveness
ρ	density (kg m^{-3})
μ	dynamic viscosity (kg m^{-1} s^{-1})
φ	corrugation angle (°)

was tabulated as a function of number of capacitance ratio R, heat transfer unit (NTU), temperature effectiveness P and number of plates N. As a result, the authors concluded that for 1pass-1pass heat exchanger, even versus odd number of thermal plates have a strong influence on the correction factor F and a negligible influence for $N > 40$. Polley and Abu-Khader [12] followed the same path but simplified the process with a bypass model, which covered a wider range of heat exchanger capacity.

The approach of using correction factor has provided a good guideline for most practical purposes. Yet its idealized assumption renders it insufficient under certain circumstances. For example, the approach assumes uniform flow distribution, thus is inapplicable in two-phase flow where maldistribution is non-negligible even with small number of plates. At occasions with maldistribution excluded, such as a 1pass-1pass 3-channel setup ($N = 4$) with two-phase flow in the center channel, as is often the case in two-phase heat transfer test [13,14], such method does not cover the situation of capacitance ratio R being infinity.

This paper provides a new explanation that end plates, instead of being treated conventionally as adiabatic, function as fins due to the contact between the corrugated surfaces of adjacent plates. Steady-state heat transfer data for single-phase water in both FPHE and BPHE with various number of plates are presented. The experimental data is used to validate a thermal conduction model in ANSYS, which incorporates plate geometries, materials and operating conditions. It indicates that the end plate fin efficiency is a function of fluid convective heat transfer coefficient and conductive thermal resistance.

2. Experimental apparatus and procedure

2.1. Experimental apparatus

The schematics of the experimental apparatus is shown in Fig. 1. It consists of three independent loops: a hot water loop, a cold water loop and a water-glycol loop. Electrical heaters are used in the two water loops for heating and chiller is used in the water-glycol loop for cooling. The two water loops are charged with deionized water. Upon charging, the system is first evacuated. Water is added through expansion tanks, which are placed at the highest location of each loop. The charging rate is adjusted such that liquid level is maintained in the transparent expansion tank. As a result, the system is held vacuum until fully charged so that no pocket of air is trapped inside. Two magnetic driven pumps are used to circulate the water to exclude any mixing of oil. The flow rate is controlled through variable frequency drives and bypass valves.

Coriolis type flow meter (Micromotion CMF25), absolute and differential pressure transducers (Rosemount), and type T (copper-constantan) thermocouples (Omega) are installed at locations as indicated in Fig. 1. Their range and uncertainty after calibration are listed in Table 1. As a result, the experimental uncertainty for *Re*, *f* and *Nu* are calculated through error propagation rule, with their maximum value also listed in Table 1.

National Instrument SCXI1000 chassis is used for data acquisition. It is connected to a desktop computer through PCI-MIO-16e-1 and used in conjunction with LabVIEW software. The modules and terminal blocks used in the data logger are SCXI1102-SCXI1303 for input measurement and SCXI1124-SCXI1325 for output control. All data are obtained under steady state conditions for about 20 min.

The test section is well insulated, with heat loss calibrated so that the energy balance (measured heat load between hot and cold stream) is within ±3%, in accordance with ANSI/ASHRAE standard 181-2014 [15].

The geometries of the two types of heat exchangers tested (frame-and-plate and brazed plate) are depicted in Fig. 2. They are both of 1pass-1pass U-type configuration. The parameters of their geometry are summarized in Table 2. In the case of frame-and-plate heat exchanger, the same frame was used with a different number of identical plates and for two torques of tightening (resulting in two different platages), while for the brazed plate heat exchangers special samples were manufactured, cutting one large heat exchanger to appropriate sizes and brazing the connections. The rest of the heat exchanger was used to provide additional information about contact area and geometry as shown in Fig. 11.

2.2. Data reduction

The primary measurements consist of the flow rates of each fluid stream, their inlet and outlet temperatures, and the pressure drop. Following the method outlined by Muley and Manglik [16], equivalent diameter D_e (=2*b*) is used for calculation with all rele-