International Journal of Heat and Mass Transfer 108 (2017) 2239–2252

Contents lists available at ScienceDirect

International Journal of Heat and Mass Transfer

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

Transitional flow regime heat transfer and pressure drop in an annulus with non-uniform wall temperatures

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ARTICLE INFO

Article history:
Received 4 July 2016
Received in revised form 12 December 2016
Accepted 5 January 2017

Keywords:
Heat exchanger
Annulus
Annular passage
Diameter ratio
Transition flow
Mixed convection
Developing flow
Wall temperature uniformity
Nusselt number
Colburn *j*-factor
Friction factor

ABSTRACT

An experimental investigation was carried out to determine the average heat transfer coefficients and friction factors in the transitional flow regime of a horizontal concentric annular passage. The flow was in the mixed convection flow regime and was both hydrodynamic and thermally developing. The annular diameter ratio was 0.483 with an inner passage wall diameter of 15.90 mm. The test facility, which consisted of a counter-flow heat exchanger having a heat transfer length of 5.08 m, was operated at different degrees of longitudinal wall temperature uniformity on the inner wall of the annular passage. Both heated and cooled flow applications with water as fluid (cold fluid and hot fluid respectively) were investigated using a conventional annular inlet geometry type. It was found that the degree of temperature uniformity on the inner surface of the annular passage had an influence on the transitional Reynolds number span, the heat transfer coefficients, and friction factors. Depending on the wall temperature uniformity it was found that the critical Reynolds number, based on the Nusselt number results, was approximately between 350 and 500 for a cooled annulus and between 430 and 510 for a heated annulus. The critical Reynolds numbers based on the friction factor results were different from those based on the Nusselt number and was found to be approximately 800 for isothermal flow, between 1000 and 1030 for a cooled annulus, and approximately 1000 for a heated annulus. Correlations for the prediction of the transitional regime Nusselt number and friction factor as a function of among others, the wall temperature uniformity are proposed. The Nusselt number correlation predicts all the experimental data within a ±6% error band for the heated and cooled annulus cases. The friction factor correlation also predicts all the data points, within a ±6% error band.

1. Introduction

Tube-in-tube type heat exchangers is one of the most common kinds of thermal exchange devices used. They are normally operated in a counter-flow configuration such that the fluid in the inner tube and the fluid in the annular space flow in opposite directions. Although a large body of literature is available for the heat transfer and pressure drop characteristics in the inner tube, less literature is available focusing on the annular flow passage. Among the limited literature for annuli, some correlations and calculation methods do exist for predicting the annular heat transfer coefficient in the turbulent flow regime, including those presented by Gnielinski [1], Swamee et al. [2] and Dirker and Meyer [3]. However, for the transitional flow regime, very little work has been done.

This could be so because most design guidelines advise designers to avoid operating heat exchangers in this flow regime. However, Meyer [4], lists several reasons why heat exchangers often operate in, or close to the transitional flow regime. These include changes in operating conditions, fouling, scaling, replacement of upstream or downstream equipment with different operating characteristics. The introduction of heat transfer enhancement geometries also plays a role. Even though these geometries increase the heat transfer coefficients, it unfortunately also increase the pressure drop. As a result, in many cases pumping power is reduced by decreasing, the mass flow rates (without a significant decrease in heat transfer) which may result in heat exchangers being operated in, or close to the transitional flow regime [5–10].

Because only limited work has been done focusing on the transitional flow regime characteristics in annular flow passages, a brief review of transitional flow regime literature for circular and other cross sectional flow passages need to be considered. In this context, noticeable work has been conducted by Ghajar and coworkers. For instance, for circular flow channels, Ghajar and Tam [11] and Ghajar and Madon [12] observed that the inlet configura-

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http://dx.doi.org/10.1016/j.ijheatmasstransfer.2017.01.022
0017-9310/

Nomenclature

a	annular diameter ratio, $a = D_1/D_o$
A_c	cross-sectional area
A_s	surface area of heat transfer
B_1, B_2	constant factor
C_1, C_2	coefficients
c_p	specific heat
D	diameter
E_1, E_2, E_3	constant factors
EB	energy balance (%)
F	factor to take into account the dependence on a
f	friction factor
g	gravity acceleration
Gr	Grashof number, $Gr = \frac{g\beta_o(\overline{T}_{iw}-\overline{T}_{b,o})D_h^3}{\nu_o^2}$
h	convection heat transfer coefficient
j	Colburn j factor
K	factor to take into account the temperature dependence of fluid properties
k	thermal conductivity
L_{dp}	pressure drop length
L_{hx}	length of heat transfer
$\dot{m}$	mass flow rate
m, n	exponent for τ
Nu	Nusselt number, $Nu = hD_h/k$
Δp	pressure drop
Pr	Prandtl number
$\dot{Q}$	heat transfer rate
Re	Reynolds number, $Re = \frac{4\dot{m}_o}{\pi\mu_o(D_0+D_1)}$
Ri	Richardson number, $Ri = \frac{Gr}{Re^2}$
T	temperature
$\overline{T}$	averaged temperature
u	exponent for Gr
v	exponent for Pr
V	average cross-sectional velocity, $V = \dot{m}/(\rho_o A_c)$
x	axial length
z_1	exponent for $GrPr/Re$
z_2	exponent for $GrPr$

Greek symbols

ρ	density
β	volume expansion coefficient
ν	kinematic viscosity
μ	dynamic viscosity
τ	degree of wall temperature uniformity

Subscripts

0	inner wall of outer tube
1	outer wall of inner tube
iso	isothermal
b	bulk property
ca	cooled annulus
d	diabatic
h	hydraulic
ha	heated annulus
i	inner flow passage
I	lower Reynolds number limit of transitional flow regime
II	upper Reynolds number limit of transitional flow regime
in	inlet
iw	on the outside wall of the inner tube
LMTD	logarithmic mean temperature difference
o	annular flow passage
out	outlet
ow	outer tube wall

tion of a flow channel can significantly influence the critical Reynolds number where transition is initiated. In one of their studies the effect of the inlet geometry on the developing and fully developed mixed convection for three different types of inlet geometries namely: reentrant, square edged and bell-mouth were investigated. They found that transition from laminar to turbulent flow under isothermal flow condition occurred at Reynolds numbers of 1980–2600, 2070–2840 and 2125–3200 for the reentrant, the square edged and the bell-mouth inlet, respectively. Further work by Tam and Ghajar [13] showed transition to occur at higher Reynolds numbers than those observed in previous studies. They found that transition from laminar to turbulent flow occurred at Reynolds numbers of 2900–3500, 3100–3700 and 5100–6100 for the reentrant, the square edged and the bell-mouth inlet, respectively. When different heat fluxes were applied to the test section the values of friction factor and the lower and upper limits of the transition boundaries increased with an increase in the heating rate for a fixed Reynolds number.

Olivier and Meyer [15] did a similar investigation as done by Ghajar and Tam [11] and Tam and Ghajar [13] with the same types of inlets. However, apart from smooth tubes they also investigated enhanced tubes. It was found for enhanced tubes that transition was affected not only by the type of inlet, but also the tube surface roughness. Transition started earlier for the re-entrant inlet type followed by the squared edged and bell-mouthed inlet types. Further investigation with heat transfer [15] showed that inlet disturbances had no effect on the critical Reynolds number. Dirker et al. [16] investigated the effects of different types of inlets in rectangular micro-channels. They considered sudden contraction, bell-mouth and swirl inlet types. They also found that the critical Reynolds number and the transitional behaviour in terms of heat transfer and friction factors were influenced significantly by the inlet flow condition. The transition for their bell-mouth inlet commenced at a Reynolds of 1050 for isothermal tests. This is much lower than the Reynolds number of 2125 found by Ghajar and Madon [12] for their bell-mouth inlet and could have been due to the differences in cross-section size, Grashof numbers, and shape of conduits that were used in the two investigations. Everts et al. [17] investigated the effects of surface roughness on transition in circular tubes. Heat transfer measurements at three different heat fluxes (5.6, 8.4 and 11.4 kW/m^2) were taken for water in a smooth and three roughened tubes with a relative roughness of 0.01, 0.02 and 0.04, respectively. It was found that transition occurred earlier with increasing surface roughness and the heat transfer increased as well, while the secondary flow effects decreased.

Some of the literature discussed so far, which did not consider inlet effects are for fully developed (hydrodynamically and thermally) flow regimes. Practically, many heat exchangers are compact and short in length and do not necessarily operate within the fully developed flow regime. In some cases free convection heat transfer may exist, which in turn creates secondary flow. In their handbook, Kakac et al. [18], in which they reviewed work by Hattori [19] and Nguyen et al. [20], indicated that for mixed convection in a horizontal concentric annulus the secondary flows can strongly distort the velocity and temperature profiles and consequently increase heat transfer coefficients. Mohammed et al. [21] carried out an experimental study on forced and free convective heat transfer in the thermal entry region of horizontal concentric annuli operated with air at a constant inner wall heat flux while