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Experimental investigation of circumferentially non-uniform heat flux on the heat transfer coefficient in a smooth horizontal tube with buoyancy driven secondary flow

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ABSTRACT

In this experimental investigation the influence of non-uniform heat flux distributions on the internal heat transfer coefficient in a horizontal circular tube was studied for liquid water. The tube had an inner diameter of 27.8 mm and a length to diameter ratio of 72. Different outer wall heat flux conditions were studied for Reynolds numbers ranging from 650 to 2600 at a Prandtl number of approximately 6.5. Heat flux distributions included fully uniform heating (which had a circumferential angle span of 360°) and different partial uniform heat flux distributions with angle spans of 180° or 90° at different circumferential positions. Depending on the angle span, local heat flux intensities ranging from 1658 W/m^2 to 6631 W/m^2 were tested. Results indicate that the average steady state Nusselt number is greatly influenced by the applied heat flux position and intensity. Highest average heat transfer coefficients were achieved for cases where the applied heat flux was positioned on the lower half (in terms of gravity) of the tube circumference, while the lowest heat transfer coefficients were achieved when the heating was applied to the upper half of the tube. Smaller angle spans produced lower heat transfer coefficients. The relative thermal performance of the different heating scenarios where characterised and described by means of newly developed heat transfer coefficient correlations for angle spans of 180° and 90° which correlated 92% and 96% of the data respectively within 3% of the measured Nusselt number.

1. Introduction

Several applications including, but not limited to, solar collectors and boiler systems contain circular tubes that are exposed to circumferential non-uniform heat flux or non-uniform wall temperature thermal boundary conditions. However, relatively little research has been performed to investigate such boundary conditions in terms of the effective convective heat transfer coefficient. This can lead to significant uncertainty regarding the convective heat transfer ability of circular flow passages, especially for conditions where buoyancy driven secondary flow is present. Such conditions, known as mixed convection are specifically prevalent at low Reynolds numbers if the thermal boundary condition has a significant influence on the fluid density distribution within the passage. In these cases, the impacts of both the mechanically forced flow component, as well as the natural convection component must be considered. Mixed convection can result in enhanced heat transfer or suppressed heat transfer depending on the directional components of the forced and buoyancy flow terms. In this sense, the thermal boundary is very important. It can be described either in terms of temperature (uniform or non-uniform), in terms of heat flux (uniform or non-uniform), or in terms of both temperature and heat flux.

Several studies have been conducted for uniform heat flux conditions. Yasuo et al. [1] experimentally investigated the buoyancy effects on air flowing in a horizontal tube under uniform heat flux conditions for a Reynolds number range of 100–13 000. They showed that the local Nusselt number is significantly influenced when the product of the Reynolds and Rayleigh numbers is higher than 1000. Kupper et al. [2] experimentally investigated mixed convection in a circular horizontal tube for Reynolds numbers ranging from 100 to 2000 and Grashof numbers ranging from 300 to 30 000. They showed that an increase in the Grashof or Reynolds numbers resulted in higher Nusselt numbers and that an entry length is needed in order to establish secondary flow. Bergles and Simonds [3] experimentally investigated the effects of mixed convection for a Reynolds number range of 460–720. They found that the Nusselt number can be greatly affected by the presence of secondary flow and that the heat transfer coefficient can be up to three or four times higher compared to predictions of traditional correlations for pure forced convection. Morcos and Bergles [4] continued the experimental investigation for distilled water and ethylene glycol. Their

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Nomenclature

A	area [m^2]
C	correlation coefficient [-]
c_p	specific heat [J/kg K]
D, D_0	diameter and outer diameter [m]
EB	energy balance error [%]
g	gravity [m/s^2]
Gr	Grashof number [-]
h	heat transfer coefficient [W/m^2 K]
h^*	hypothetically assumed heat transfer coefficient [W/m^2 K]
$\bar{h}$	average heat transfer coefficient [W/m^2 K]
I	electric current [A]
j	Colburn j-factor [-]
k	thermal conductivity [W/m K]
L	length [m]
M	correlation exponent [-]
m	measuring station index number [-]
$\dot{m}$	mass flow rate [kg/s]
N	correlation exponent [-]
n	thermocouple position
Nu, $\overline{\mathrm{Nu}}$	Nusselt number and average Nusselt number [-]
P	correlation exponent [-]
Pr	Prandtl number [-]
$\dot{Q}$	heat transfer rate [W]
$\dot{q}$	heat flux [W/m^2]
r, $\bar{r}$	radial position and average radial position [m]
r_1	inner radius of Insulation [m]
r_2	outer radius of insulation [m]
R	thermal resistance [K/W]
Re	Reynolds number [-]
Ri	Richardson number [-]
T, $\bar{T}$	temperature and average temperature [°C]
t	wall thickness [m]
V	voltage [V] or velocity [m/s]

Greek symbols

β	coefficient of thermal expansion [-]
ρ	density [kg/m^3]
μ	dynamic viscosity [kg/ms]
ν	kinematic viscosity [m^2/s]
φ	characteristic angular position of heated span [°]

Subscripts

B	bulk
ci	calculated value
DC	direct current
h	heated
i	inner
in	inlet/input
$insul$	insulation
$LMTD$	logarithmic mean temperature difference
$loss$	heat loss
m	measuring station position number
mid	midpoint
n	circumferential thermocouple position number
o	Outer
out	outlet/output
s	surface
TC	thermocouple
tot	total
w	wall
water	water

data indicated that the Nusselt number is influenced by the Rayleigh number, variations of the thermo-physical properties of the heat transfer fluid, and the radial conduction in the tubes wall. They proposed a Nusselt number correlation which can be used for a wide variety of fluid properties and flow conditions. Chou and Hwang [5] carried out a numerical analysis of the Graetz problem with the presence of natural convection for uniform heat flux. They used the vorticity-velocity method and coupled it to the Boussinesq approximation to model the temperature dependence of the fluid density. They showed that natural convection distorts the axial flow velocity and cross-sectional temperature profiles and that the highest fluid velocity and lowest fluid temperature can be found at the bottom of the tube. Ghajar and Tam [6] experimentally considered water and ethylene glycol as the working fluid in a horizontal circular tube for a large range of Reynolds numbers covering the laminar flow regime to the turbulent flow regime. They also showed the need for an entrance length in order to allow the development of the secondary flow. They proposed a Nusselt number correlation for the laminar flow regime in terms of the Reynolds number, Prandtl number, the length to diameter ratio, the Grashof number and a wall viscosity ratio. Mohammed and Salman [7] experimentally investigated mixed convection in developing flow for air in circular tubes with a uniform heat flux for a Reynolds number range of 400–1600. They found that secondary flow could have different effects on the Nusselt number depending on the Reynolds number. The Nusselt number decreased at lower Reynolds numbers and increased at higher Reynolds numbers, or when higher heat flux was applied. They correlated the average Nusselt number as a function of the Rayleigh and Richardson numbers.

Some work has also been done on flow passages with non-uniform heating. Lin and Lin [8] experimentally investigated the effect of non-uniform (bottom) heating on air flow in a horizontal rectangular duct for Reynolds numbers ranging from 9 to 186. They included flow visualization of the secondary flow vortices and showed that the onset of thermal instability, which enhances the heat transfer, occurs close to the duct entrance with increased Grashof numbers and decreased Reynolds numbers. Elatar and Siddiqui [9] experimentally investigated water flow in a horizontal square duct with bottom heating for Reynolds numbers ranging from 300 to 750 and Grashof numbers ranging from 6.37×10^6 to 3.86×10^7. They showed that the effect of secondary flow was dependent on the Reynolds and Grashof numbers and that turbulence intensity is increased due to rising plumes of warm fluid. Back flow along the top unheated wall, which increased in intensity with the lower wall temperature, was observed at higher Richardson numbers. Chang et al. [10] experimentally and numerically considered upper half heating of horizontal circular tubes for water under forced convection in the turbulent flow regime. They found that the well-known Dittus-Boelter correlation correctly predicted the average Nusselt number irrespective of heat flux distribution as can be expected from the absence of mixed convection. Okafor et al. [11,12] included mixed convection in their numerical investigation of circumferential non-uniform heating of a horizontal circular tube with water as working fluid. They considered a sinusoidal type heat flux distribution for a length to diameter ratio of 159 and a Reynolds number range of 180–2200. They used the Boussinesq approximation to model the temperature dependence of the fluid density. They found that the Nusselt number (average and local) are enhanced due to the presence of non-uniform heating occurring from below. Huang et al. [13] also performed a numerical study on the effects of non-uniform heat flux boundary conditions for fully developed flow in a circular tube. They considered upper and lower heating as well as fully uniform heating for