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Experimental investigation on heat transfer in laminar, transitional and turbulent circular pipe flow

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ABSTRACT

Experimental investigations on heat transfer in a circular pipe for laminar, transitional and turbulent flow are presented. Reliable prediction of heat transfer coefficients for transitional flows is still a challenging task. While methods for predicting laminar and turbulent heat transfer coefficients are widely established in literature, the proposed methods and the validity for calculating according values for the transition region and even the definition of this region itself remains a field of ongoing development. One aspect in this picture is the scarce availability of experimental data on heat transfer coefficients in circular pipe flow for Reynolds numbers in the range of $1000 < Re < 4000$, especially for higher Prandtl numbers, i.e. $10 < Pr < 90$. Thus, this paper focuses on providing experimental results for heat transfer in circular pipe flow for Reynolds (Re) numbers in the range of $500 < Re < 23000$ and $7 < Pr < 41$. The test fluid is a water–glycol mixture with a mass fraction of water of $x_m = 0.477$. The results presented in this paper show good agreement with the widely used calculation methods proposed by Gnielinski in 2013 for laminar, transitional and turbulent flow. The results also confirm the presence of the transition region to occur between $2300 < Re < 4000$. 184 data points in the range of $1000 < Re < 4000$ are shown, since in this range scarce data are available in literature. In summary 261 heat transfer coefficients for $500 < Re < 23000$ and $7 < Pr < 41$ are presented, which show good agreement (80.8% are within ±15%) to the cited literature.

1. Introduction

In the meanwhile calculating heat transfer in circular pipes is a standard procedure. Since the work of Pethukov and Kirillov [1] and Gnielinski [2] it is possible to calculate laminar as well as turbulent heat transfer coefficients under different thermal and hydrodynamic conditions for various media. As the heat transfer coefficient depends on the flow regime, the calculation method for laminar and turbulent flow differs. Laminar heat transfer is calculated by the following Eq. (1) as shown by Gnielinski [2].

$$Nu_{lam} = \sqrt[3]{Nu_1^3 + b^3 + (Nu_2 - b)^3 + Nu_3^3} \quad (1)$$

The factor b depends on the geometry of the pipe or channel, in which the heat is transferred to or from the fluid. For a circular pipe the factor b is equal to $b = 0.3$. Another important influence on the heat transfer is given by the velocity and temperature profile of the flow. For fully developed thermal and hydrodynamic flow the heat transfer coefficient, i.e. the dimensionless Nusselt number, Nu, is constant and thus, independent of the Reynolds and Prandtl number. This is represented in Nu_1 (Eq. (2)):

$$Nu_1 = 3.66 \quad (2)$$

If the flow in the pipe is hydrodynamically developed, but still thermal developing, the heat transfer is enhanced compared to a fully developed velocity and temperature profile due to a reduced thermal boundary layer thickness. The influence of this enhancement, depending on different thermal and hydrodynamic conditions in the pipe, represented by the Graetz number, Gz, is described by Eq. (3) for Nu_2.

$$Nu_2 = 1.615 \cdot \sqrt[3]{Gz} \quad (3)$$

In the case of a hydrodynamically and thermal developing flow, the heat transfer in the laminar flow regime is enhanced by reducing the thermal boundary layer as well as by enhanced velocity components in the near-wall region. This influence is described by the following Eq. (4).

$$Nu_3 = \sqrt[6]{\frac{2}{1 + 22 \cdot Pr}} \cdot Gz \quad (4)$$

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SYMBOLS

Latin symbols

A	Heat transfer surface (m^2)
A_c	Cross-section of the measuring channel (m^2)
a	Geometrical distance of pipe center to isothermal plane (m)
b	Geometry factor
c_p	Specific heat capacity ($J\,kg^{-1}\,K^{-1}$)
d	Diameter (m)
d_h	Hydraulic diameter (m)
GUM	Guide to the expression of Uncertainty in Measurement
$\Delta\dot{H}$	Enthalpy flow difference (W)
kA	Heat transmission coefficient ($W\,K^{-1}$)
l	Length (m)
$\dot{M}$	Mass flow ($kg\,s^{-1}$)
$\dot{Q}$	Heat flow (W)
R	Heat transfer resistance ($W\,K^{-1}$)
R_e	Electrical resistance (Ω)
ΔT	Temperature difference (K)
V	Voltage (V)
x	Fraction (–)
X	Various parameter effecting the Nusselt number

Greek symbols

α	Heat transfer coefficient ($W\,m^2\,K^{-1}$)
λ	Thermal conductivity ($W\,m^{-1}\,K^{-1}$)
ξ	Friction factor (–)

Subscripts

Al	Aluminum
cold	Cold flow
const	Constant
hot	Hot flow
i	Control variable
in	Inlet
lam	Laminar
m	Mass
out	Outlet
turb	Turbulent
W	Wall

Dimensionless numbers

Nu	Nusselt number (–)
Pr	Prandtl number (–)
Re	Reynolds number (–)

Eq. (1) combines these scenarios for a laminar flow in a channel with constant cross section. The thermal boundary condition at the heat transferring surface has an influence on the fluid, especially on the velocity nearby the wall. Since different thermal conditions at the wall are possible, for two, theoretical solutions are derived in literature [2], namely the uniform heat flux (UHF) and the uniform wall temperature (UWT). Either a constant heat flux or a constant temperature along the heat transferring surface is present. The Eqs. (2)–(4) are used for calculating laminar heat transfer coefficients under a UWT condition. For a constant heat flux (UHF) over the heat transfer surface, Eqs. (2)–(4) have to be changed to the Eqs. (5)–(7) to take the effect of the thermal boundary condition at the heat transferring surface into account.

$$Nu_1 = 4.364 \tag{5}$$

$$Nu_2 = 1.953 \cdot \sqrt[3]{Gz} \tag{6}$$

$$Nu_3 = 0.924 \cdot Pr^{\frac{1}{3}} \cdot \sqrt{Re \cdot \frac{d_h}{l}} \tag{7}$$

Thus, laminar heat transfer coefficients can be determined under different thermal boundary conditions for various media in tube with different constant cross section and length with Eqs. (1)–(7). Heat transfer coefficients for turbulent flows are frequently calculated using the common Eq. (8) by Petukhov and Kirillov [1].

$$Nu = \frac{\frac{\xi}{8} \cdot Re \cdot Pr}{1 + 12.7 \cdot \left(\frac{\xi}{8}\right)^{0.5} \cdot \left(Pr^{\frac{2}{3}} - 1\right)} \cdot \left(1 + \left(\frac{d_h}{l}\right)^{\frac{2}{3}}\right) \tag{8}$$

$$\xi = (1.8\,log_{10}Re - 1.5)^{-2} \tag{9}$$

The first correlation for overall heat transfer coefficients in transitional flow which was paid attention to, was proposed by Hausen [4] in 1959. It covers Reynolds numbers ranging from $2300 < Re < 10^6$. Schlünder (1970) [5], Hufschmidt et al. (1966) [6] and many others found their experimental results on heat transfer in the transition region deviating from this equation. Gnielinski (1976) [7] and Churchill (1977) [8] proposed correlations that can be used to calculate heat transfer coefficients in the transition flow regime. The correlation of Churchill was developed for the entire flow regime, meaning laminar, transition and turbulent, in contrast to Gnielinski's correlation which is based on the ideas of Pethukov and Kirillov [1]. Gnielinski compared his correlation primarily with experimental values in the nearly fully turbulent flow regime ($Re > 8000$). This is not mentioned explicitly by Gnielinski, but obvious with a view to his experimental results [7] and also mentioned by Tam and Ghajar [3] in 2006. Nevertheless, the equation of Gnielinski [7], published in 1976, was one of the first that allowed the calculation of heat transfer coefficients for both, transition and turbulent flow regime. It is shown in [3] that the Gnielinski correlation [7] shows a better all-round performance for calculating heat transfer coefficients in forced convection transitional flow under different inlet geometries than the correlation of Churchill [8]. Since Gnielinski recognized that his correlation shows deviations to experimental results especially for transitional flows, (cf. Shah and Sekulic [9]) Gnielinski [10] proposed in 1995 a calculation method for calculating heat transfer coefficients in the transition flow regime. He proposed to use a linear interpolation between the Nusselt values for laminar flow Nu_{lam} (Eq. (1)) at $Re = 2300$ and turbulent flow Nu_{turb} (Eq. (8)) at $Re = 10^4$ for the given Reynolds number in the transition region ($2300 < Re < 10^4$). This method was checked against an enormous data set of experimental heat transfer coefficients (e.g. [11–13]) for different ratios of diameter to length as well as Reynolds and Prandtl number ranging from $2 < Pr < 190$, $2000 < Re < 10^6$. In the following years this method has become the standard procedure for the calculation of heat transfer in transitional flow.

The calculation methods for predicting heat transfer coefficients are based on numerous investigations cited in literature, which have been performed for mostly $Re > 4000$ with water or air. Stone et al. [14] provided experimental data for circular pipe flow with a diameter to length ratio of 0.001 for $4 < Pr < 25$ and $10^4 < Re < 10^6$. Hufschmidt and Burck [15] investigated the heat transfer for $2 < Pr < 6$ and $80 < Pr < 180$ with a Reynolds number range of $3000 < Re < 640000$ in circular pipe flow. For $1000 < Re < 4000$ only five data points are shown with Prandtl numbers within $80 < Pr < 180$. Churchill [8] reviewed many other