Experimental investigation of forced convective heat transfer in cylindrical pipe flow

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I. Introduction

In recent years, forced convective heat transfer in cylindrical pipe flow plays an important role in many technical cooling systems. Nusselt number (Nu) is a dimensionless number which represents the ratio of convective (h) and conductive heat transfer (k), as expressed in Equation (1).

$$Nu = \frac{h \cdot L}{k} \tag{1}$$

From general dimensional analysis, Nusselt number represents function of Reynolds number (Re) times Prandtl number (Pr) as Equation (2).

$$Nu = \alpha \cdot Re^{\pi_\beta} \cdot Pr^{\pi_\gamma} \tag{2}$$

Here, factors α, β and γ are constant values that depend on flow regime and are calculated from an experimental result. Nusselt number is one of the most important numbers for forced convective heat transfer, and are calculated from Equation (1) and (2).

Many studies have pointed out that a heat transfer coefficient varies depending on the type of flow: laminar, transition and turbulent. Gnienlinski [2] showed a calculation method for the laminar heat transfer coefficient of two kinds of boundary conditions. (I) Constant wall temperature (UWT) and (Ⅱ) Constant heat flux (UHF). Petukhvov and Kirillov [3] showed calculation method for turbulent flows. There has been very scarce experimental data of laminar-to-turbulent transitional region. Bertsche et al, [4] focused on reliable prediction of heat transfer coefficient for transitional flows. In their study, Bertsche et al, showed experimental heat transfer coefficients for Reynolds number $500 < Re < 23000$ and Prandtl number $7 < Pr < 41$.

Much remains to be studied for providing experimental data except water and glycole as operation fluids. In this study, I focused on forced convective heat transfer in flow of water and glycole in a cylindrical pipe. A 50/50vol% mixture of water and glycole, which is a typical liquid coolant in automotive applications was used as a operating fluid. The experiment was carried out by considering a board range of Reynolds numbers, spanning from a laminar to fully turbulent flow. Moreover, the measurement of wall friction coefficients were also performed in this study. The experimental data were compared with other sources as well as computational results obtained from already existing numerical simulations (CFD) by Christphan [5].

II. Experimental setup

Figure. 1 shows experimental loop. The experimental loop consists of heat exchanger, pump, coriolis mass flow rate, welder, reservoir, and test section basically. Heat exchenger keep thermal stationary condition in flow pipe. Mass flow rate is controlled by pump and baypass valve C which is located in a parallel. The pipe is thermal isolated, surrounded with glass wool.

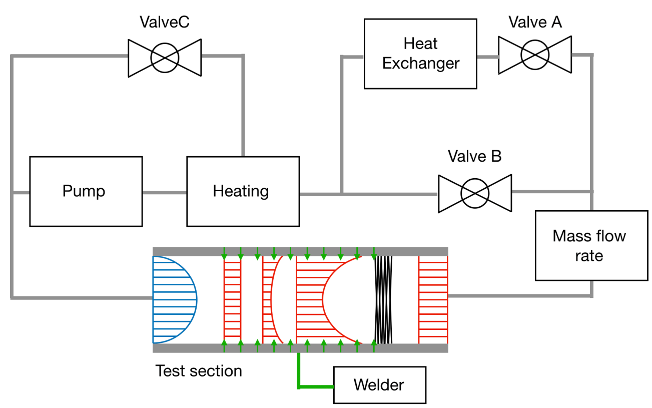

Fig. 1. Process flow diagram of the test facilities including test section.

Figure. 2 shows velocity and thermal boundary layer development vary with horizontal axis in a test section. Velocity and thermal profile are shown blue and red color, respectively. The test section is made of stainless steel (1.4301) with an inner diameter di=12mm and outer diameter do= 15mm. Highly accurate resistance thermall probes (PT-100) are used to find out the inlet and outlet bulk temperature (Tib , Tob) and wall temperature Tw. Moreover, thermocouple 'Type-K' are used to take temperature gradient in flow direction.

The test section consist of entrance, heated and thermal equalized part.

1) Entrance part
The first part of test section is 1.2 [m] length entrance part which is sufficiently long to ensure dynamically developed flow condition at the exist. The bulk temperature (Tb0) at this section were measured by PT-100.
2) Heated part
The second part of test section is 2 [m] length heated part which is sufficiently long to ensure thermaly fully developed flow condition at the exist. The tube wall were heated electrically by welder which provide high current and low voltage to keep the uniform heat flux condition in a inner pipe flow. Convective heat transfer is independent with horizontal axis in fully developed flow, constant heat flux condition. The wall temperature (Tw) at the exist of this section were measured by PT-100.
3) Thermal equalized part
The third part of test section is thermall equalized part which is including static mixture. Static mixture forms turbulent and vortex. Then, the thermal profile of heated exist mix together. At the end, the bulk temperature(Tb1) are measured.

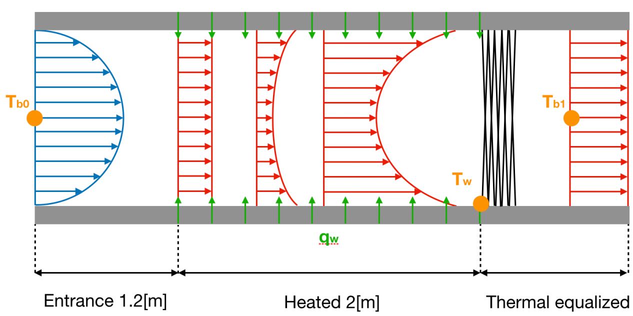

Fig. 2. Velocity (blue) and thermal (red) boundary layer development vary with horizontal axis in a test section.

III. Calucuration Flow

Material properties are all temperature-dependent function. At first, material properties varies with temperature were taken. Next, I move to experimental facilities and measured temperature differences, pressure differences and mass flow rates. Finaly, Nusselt, Prandtl, Reynolds numbers and friction coefficients were calcurated by post-proccesing, LabView and MATLAB.

IV. Results and Discussion

The Density rho, heat conductivity k, specific heat transfer Cp, kinetic viscosity nu, dynamic viscosity mu, Pradtl number Pr are all varies with temperature. It is difficult to keep high Pramdlt number and transitional Reynolds number. For example, as enhance cooling, temperature decrease, statics viscosity increase. As a result, Pr increase and Re decrease. An Experimental data and correlations were compared. In correlations, the Prandtl number was assumed to be constant. However, the experimental Prandtl number is not constant because the fluid properties vary with temperature. The aim is to set Prandlt number level and vary the Reynolds number. The avaraged Prandlt numbers were taken to plot Nusselt and Reynolds number.
Figure. 3 shows heat transfer coefficients for $500 < Re < 4000$ for $Pr = 26$, which shows good agreement with calcuration method showed by Gnienski of each flow regime, laminar, transitional and turbulent.
200 experimental results ($500 < Re < 4000$ and $10 < Pr < 30$) has been compared with calcurations method for predictiong the laminar, transitional and turbulent heat transfer coefficiets.

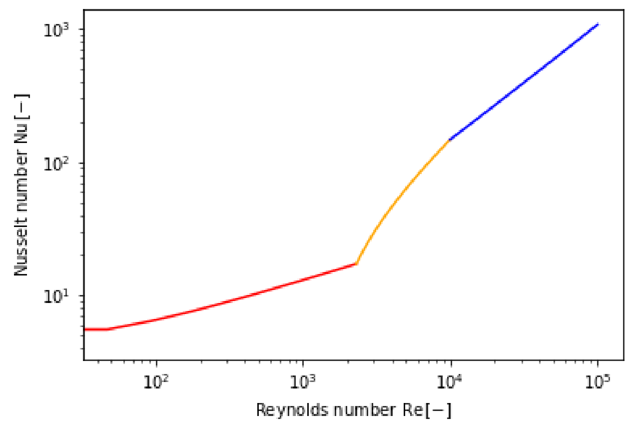

Fig. 3. Dimensionless heat transfer coefficients compared to literature data for Pr = 26. The red, yellow and blue lines are Gnielinski correlations for laminar, transitional and turbulent, respectively.

V. Conclusion

Forced convective heat transfer in cylindrical pipe flow has been investigated experimentally. Not so many data were available for transitional regime and high Prandtl number. Therefore, in this study, 200 experimental results ($500 < Re < 4000$, $10 < Pr < 30$) has been checked with calcuration method showed by Gnielinski [2] and it showed good agreement.

VI. Acknowledgement

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