Experimental investigation of forced convective heat transfer in cylindrical pipe flow

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I. Introduction

Forced convective heat transfer in cylindrical pipe flow plays an important role in many technical cooling systems. These coolant technology is used wide varaety of coolant applications such as electric devices, automotive, and plant factry. Considering heat transfer issues, heat transfer coefficient is one of the most important numbers.

Much remains to be studied for providing experimental data for high Pramblt number and laminar-to-turbulent transitional regime. In this paper, we focus on forced convective heat transfer in cylindrical pipe flow in particular high Pramblt number and transitional regime. A 50/50vol% mixture of water and glycole which is a typical liquid coolant in automotive applications were used as a operating fluid. This coolant liquid is normally used in low temperature. However, it's difficult to maintain the cylindical pipe at low temperature. Therefore, as a 1st approach, we take a data for high temperature and then compared with Direct numerical simulation (DNS). If there is a good agreement between experimental data for high temperature and DNS, the result in a low temperature would be predicted by DNS. Moreover, the investigation shall also include the measurement of wall friction coefficient. The engineeres is frequently interested in pressure drop which is related to determine pump or fan power equipments. The experimental data compared with some correlations and other sources as well as computational results obtained from already existing numerical simulations (CFD) by Rorenzo [**?**].

II. The phenomena

III. Experimental setup

The experimental setup is already existing facilities by Christphan2018 figure.() . The experimental setup 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 pipe which is located parallel.

Figure shows the tube in a test-sction area. The test section is made of stainless steel (1.4301) with an inner diameter di=12mm and outer diameter do= 15mm. The test section consist of entrance, heated region and thermal equalisation.

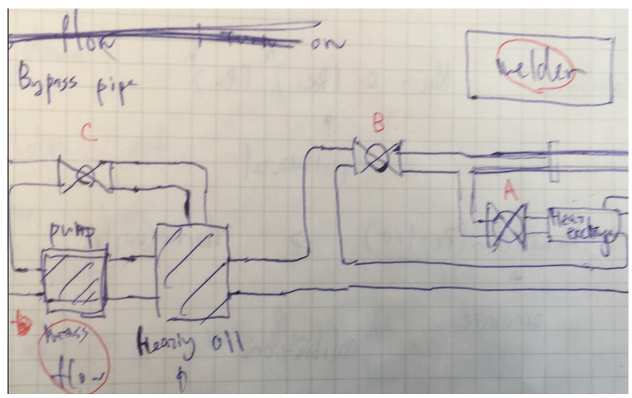

Fig. 1. Experimental setup.

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. The wall temperature(Tw) at the exist of this section were measured by PT-100.
3) Thermal equalization part
The third part of test section is thermall equalisation part which is including static mixture. Static mixture forms turbulent and vortex. Then, the thermal profile of heated exist mix tigether. At the end, the bulk temperature(Tb1) are measured.(Assumed that the pipe is thermal isolated, surrounded with glass wool.)

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. However, those thermocouples aren't directly related to calculete four parameters.(Re,Nu,Pr,Cf)

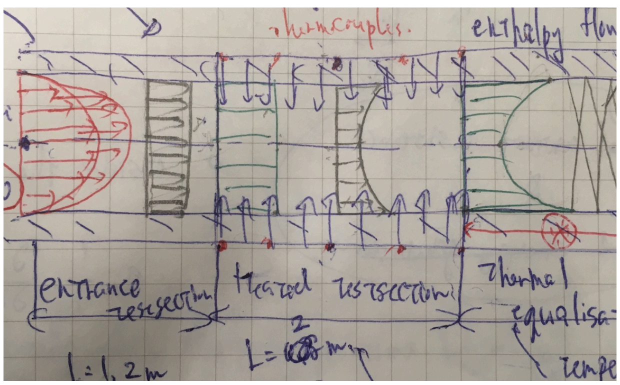

Fig. 2. Thermal boundary layer development

There are 3 ways to control Re, Pr. It is hard to keep high Pr and low Re. Figure shows material properties of Shell Heat Transfer Oil S2.

1) Cooling water
   As opening valve A and closing valve B, decreasing flow temperature. Then, viscosity increase and Re decrease and Pr increase.
2) Bypass pipe
   As opening bypass pipe C, decreasing mass flow rate then, Re decrease. As decreasing mass flow rate, Pr decrease because the number is related to heat transfer convection from the wall. (Heat is provided from the wall.) In order to decrease mass flow rate, controlling bypass pipe is stronger than pump machine.
3) Welder and mass pump machine
   As decreasing mass flow rate (pump machine) and welder, Re decrease and Pr nearly constant. Pr is proposonal to Nu which is, convection devided by conduction. Controlling both welder and mass flow rate, the ratio is maintained constant value.

IV. Calucuration Flow

Material properties are temperature-dependent function. Therefore, those were fitted to the data.

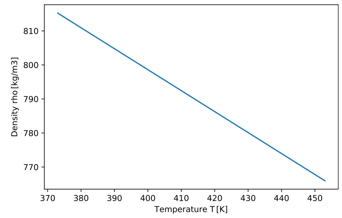

Fig. 3. Density rho, $\mathrm{kg/m^3}$

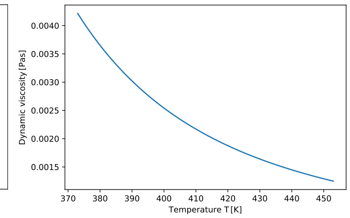

Fig. 4. Dynamic viscosity mu, Pas

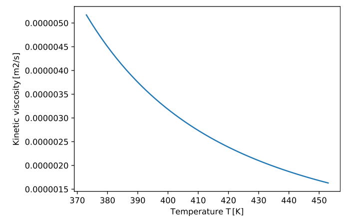

Fig. 5. Kinetic viscosity nu, $\mathrm{m^2/s}$

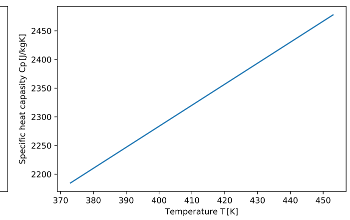

Fig. 6. Heat capacity Cp, J/kgK

V. Correlations

Skinf friction coefficient for laminar flow is descrived following equation.

$$C_{f,lam} = \frac{16}{Re_b} \tag{1}$$

Konakov(1954) showed skin friction coefficient for turbulent flow.

$$C_{f,turb} = 0.25(1.8log(Re_b) - 1.5)^{-2} \tag{2}$$

Note that these skin friction coefficient just suitable for no-heating condition, constant fluid properties. In this thesis, we provide heat to the pipe. Therefore, the fluid properties change depend on the temperature.

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

$$Nu = \alpha \cdot Re^{\pi_\beta} \cdot Pr^{\pi_\gamma} \tag{3}$$

Here, factors α, β and γ are constant value depend on flow regime and calcurated from numerical experimental results. Gunienski [**?**] showed correlations for each flow regime laminar and turbulent, respectively. Gunienski [**?**] showed calculation method for laminar flow.

$$Nu_{lam} = (3.66^3 + 0.7^3 + (1.615(Re_b Pr_b \frac{d_i}{L})^{1/3})^3)^{1/3} \tag{4}$$

Gunienski [**?**] showed calculation method for turbulent flow.

$$Nu_{turb} = \frac{\frac{C_f}{2Re \cdot Pr_b}}{1 + 12.7\sqrt{\frac{C_f}{2}}(Pr_b^{2/3} - 1)} \cdot (\frac{Pr_b}{Pr_w})^{0.11} \tag{5}$$

The range is

$$0.1 << Pr_b << 1000, 10^4 << Re_b << 10^6. \tag{6}$$

He presented transitional flow as a liner interpolation between turbulent and laminar flow.

$$Nu_m = (1-r)Nu_{m,lam} + rNu_{m,turb} \tag{7}$$

$$r = \frac{Re_b - 2300}{10^4 - 2300} \tag{8}$$

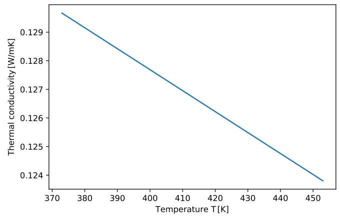

Fig. 7. Thermal conductivity, $\mathrm{W/mk}$

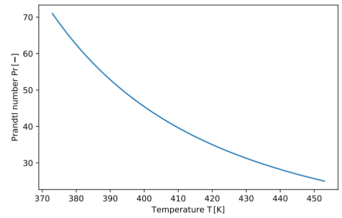

Fig. 8. Prandtl number Pr, -

VI. State-of-the-art

The presentaly considerd oil is Shell Heat Transfer oil S2. The result show good agreement with correlations.

VII. Scientific problems and interests issues

1) Fluid properties
   Density, heat conductivity, specific heat transfer Cp, nu, mu, 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. The ways to control Pr and Re are pointed above. Pr and Re are related each other and it it difficult to maintain the value that we want to measure.
2) Temperature calibration
   There are two kinds of temperature measurement, Pt-100 and thermocouples. Thermocouples are calibrated by Pt-100 because Pt-100 is accurate enough. When we calibrate, the thermocouple temperature is always lower than Pt-100.
3) Fully developed length

4) 3 parameters(Nu,Re,Pr)
   The aim is to set Prandlt number level and vary the Reynolds number. There are only two parameters to descrive the correlation between Nusselt number and Reynolds number. The only choice is to take average Prandlt number.
5) As increasing pump power, increasing flow speed and decreasing Pr number vary with heat transfer rate.
   Increasing pump power = Flow speed(Re) + Heat transfer rate(Pr) + Enthalpy
   Calcurating enthalpy, I guess it would be easy to control pump power.
6) Thermal contact resistant of the capton tape which is connected with pipe and thermocouple.
7) Turbulent intencity
8) Slliped Knudsen number
9) Corrilation between enthalpy and temperature profile
10) Oil properties depend on long-distance time
11) Outside temperature unceirtenty
12) Pipe surface roughness
13) Low temperature cooling
14) Temperature genelation(round?)

Acknowledgment

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