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Forced flow heat transfer from a round wire in a vertically-mounted pipe to supercritical hydrogen

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Abstract. Forced flow heat transfer of hydrogen from a round wire in a vertically-mounted pipe was measured at pressure of 1.5 MPa and temperature of 21 K by applying electrical current to give an exponential heat input ($Q=Q_0\exp(t/\tau), \tau=10$ s) to the round wire. Two round wire heaters, which were made of Pt-Co alloy, with a diameter of 1.2 mm and lengths of 54.5 and 120 mm were set on the central axis of a flow channel made of FRP with inner diameter of 5.7 and 8.0 mm, respectively.   Supercritical hydrogen flowed upward in the channel. Flow velocities were varied from 1 to 12.5 m/s. The heat transfer coefficients of supercritical hydrogen were compared with the conventional correlation presented by Shiotsu et al. It was confirmed that the heat transfer coefficients for a round wire were expressed well by the correlation using the hydraulic equivalent diameter.

1. Introduction

When a superconducting coil wound by cable in conduit conductor (CICC) is cooled by hydrogen under a supercritical pressure, an extraordinary heat at a quench will be cooled down smoothly without a jump to film boiling. For a design and safety evaluation of such a superconducting coil, experimental results of forced flow heat transfer under supercritical pressure with an inlet fluid temperature lower than the critical temperature is necessary.

Forced flow heat transfer of low temperature hydrogen under supercritical condition has been studied for a cooling design of rocket engines [1, 2]. However, these studies were mainly aimed for cooling systems operating near the temperature limitation of metals.

Shiotsu et al [3] studied the heat transfer from inner wall of a vertical tube to forced flow of low temperature hydrogen under supercritical pressures for the test tubes with various inner diameters and lengths. Their experimental results were compared with an equation of forced flow heat transfer under supercritical pressures presented beforehand [4] based on experimental data of helium and conventional data for non-cryogenic fluids. It was confirmed that the equation can be applicable to forced flow of hydrogen in heated tubes with wide ranges of diameter and length.

In case of CICC, heat is transferred not from inner side of a tube but from a round wire to outer flowing fluid.   The purpose of this study is firstly to obtain the experimental data of forced convection heat transfer from a round wire to supercritical hydrogen, and secondly to confirm the applicability of

the equation with the use of equivalent diameter.

2. Apparatus and method

Figure 1 shows a schematic of the experimental system, whose detail has been already presented in other paper [3]. It comprises of a main cryostat, a sub tank (receiver tank), a connecting transfer tube with a control valve, a feed hydrogen gas line from clustered cylinders and vent lines. Two test heater blocks are located in series at one end of the transfer tube in the main tank. They are used one by one.   Liquid hydrogen in the main tank is forced to flow into the transfer tube by the pressure difference between the cryostat and sub tank. The mass flow rate is set by the pressure difference and the valve opening (CV001). Liquid hydrogen flows upward through the concentric conduit of the test heater blocks. The main tank is pressurized to a desired pressure by pure hydrogen gas (99.999 %) controlled by a dome-loaded gas regulator, while the sub tank is maintained to be atmospheric pressure by opening the release valve (PV105).The mass flow rate is estimated by the weight change of the main tank, which is put on a scale (MettlerToledo WMHC 300s) that can measure up to 400 kg within 0.002 kg resolution.   The feed hydrogen gas is controlled so that the pressure is kept constant during the test.   The mass flow rate estimated above is calibrated with the flow rate of the feeding gas.   Flow measurement error is estimated to be within 0.1 g/s, if the flow rate is assumed to be constant during the test period of 60 s, for example.

The liquid temperature of the main tank and the inlet temperature T_{in} of the conduit are measured by Cernox temperature sensor. The bath temperature is controlled by a sheathed heater coil with the maximum power of 500 W which is set at the bottom of the main tank.
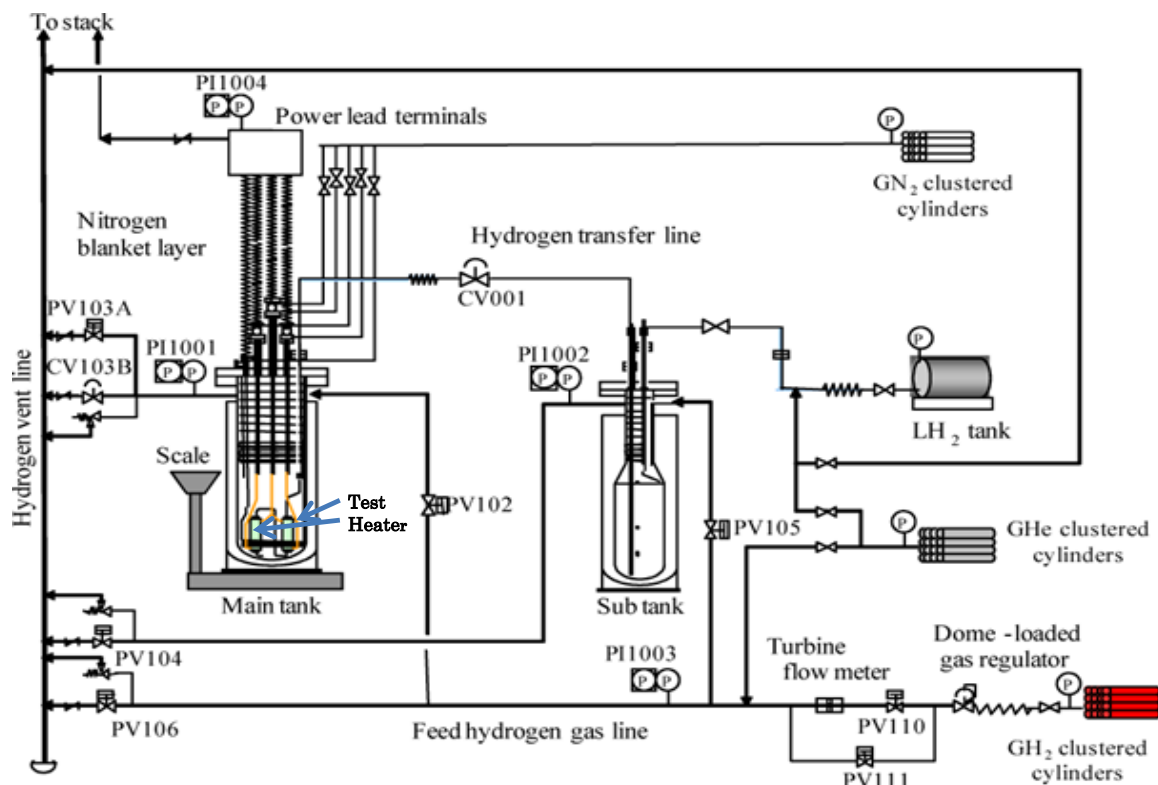

FIGURE 1.   Schematic of the experimental apparatus.

Two types of test heater block are used. Figure 2 shows schematic of the heater blocks.   Type A has a test wire made of PtCo (0.5 wt. %) alloy, 1.2 mm in diameter D_2, 54.5 mm in length L supported at the center of D = 5.7 mm conduit in a block made of fiber reinforced plastic (FRP).   Hydraulic equivalent diameter $D_e (= D - D_2)$ and the ratio of $L D_e^{-1}$ for the heater block are 4.5 mm and 13.3,
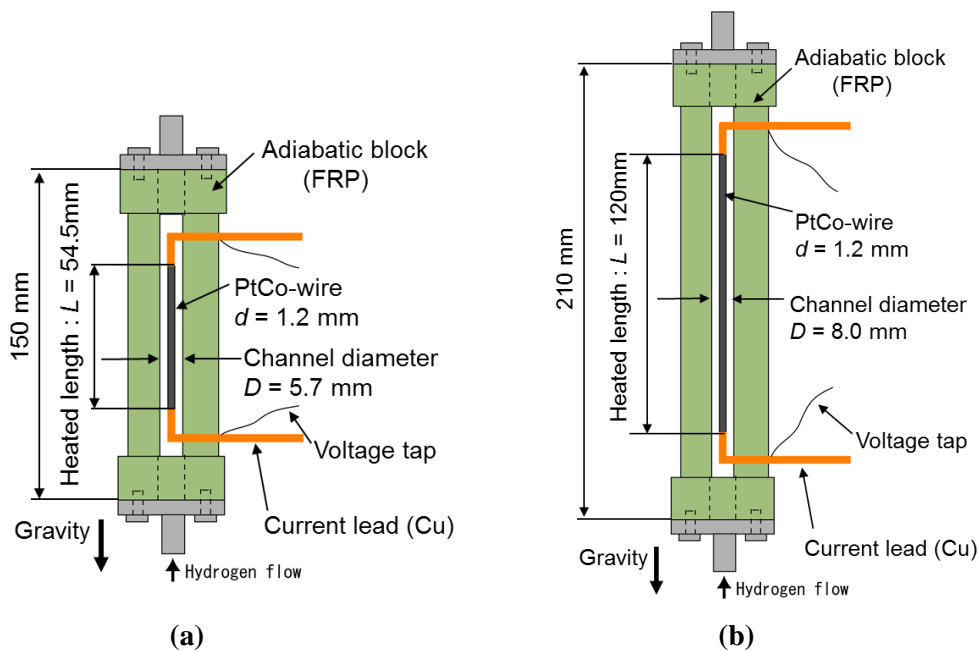
FIGURE 2. Schematic of the test heater blocks (a) Type A and (b) Type B.

respectively. Type B has the test wire made of PtCo alloy, $D_2 = 1.2$ mm, $L = 120$ mm supported at the center of $D = 8$ mm conduit. The values of D_e and LD_e^{-1} for the heater block are 6.8 mm and 17.6. Both ends of each heater wire are electrically insulated from the hydrogen transfer line by the FRP channel, whose diameter is equal to D. Each heater has an entrance length corresponding to over ten times its diameter.

The heating current to the test wire is supplied by a power amplifier (max. 400 A at a power level of 4.8 kW). The electric resistance of the heater was measured using a double-bridge circuit. The output voltage of the bridge circuit together with the voltage across a standard resistance was amplified and passed to a 16 bit digital memory system (Yokogawa WE7000). The voltage drops across the potential taps of the test heater and the output signal of a strain gauge pressure sensor were also measured. These signals are simultaneously sampled at an interval of 30 ms. The average temperature of the heated wire, T_{av}, was estimated using its electrical resistance. Temperature characteristics of the resistance had been obtained previously. The surface heat flux q was given as the difference between the heat generation rate Q and the time rate of energy storage. The average temperature on the heated surface T_w was calculated by solving steady-state conduction equation in a radial direction of the wire using T_{av} and Q (that is T_w is given as the boundary condition that satisfies measured T_{av} for Q).

The double-bridge circuit for measurement of the wire heater resistance has an accuracy of 1×10^{-4}, and a temperature deviation of about 0.1 K can be measured by the bridge. The inlet temperature was measured by a Cernox sensor with an accuracy of 10 mK and amplified by a precision amplifier (Yokogawa 3131). Calibration of the measurement circuit is performed before a series of experiment by using a standard volatage-current generator and an approved pressure gauge. Experimental error is estimated to be within 0.2 K for T_w and 3 % for q, and 0.1 K for T_{in}.

Forced convection heat transfer from the vertically-mounted heated wire was measured with a quasi-steady increase of the heat generation rate of $Q_0 e^{t/\tau}$ with $\tau = 10.0$ s at a pressure P of 1.5 MPa. It had been already confirmed that the heat transfer phenomena could be regarded as a continuous series of steady-states with $\tau = 10$ s. The inlet temperatures T_{in} is 21 K and the flow velocity is increased to 12.6 m/s.

The para-hydrogen is considered to be more than 99% although the liquid hydrogen temperature is

temporarily increased in the experiment. This is because it was liquefied using an ortho-para hydrogen converter and it takes a long time to change from para-hydrogen to ortho-hydrogen [5].

3. Results and discussion

3.1 *Boiling curve under supercritical pressur*e

Typical boiling curves for the heaters of Type A and Type B under a supercritical pressure of 1.5 MPa are shown in figure 3 and figure 4 respectively. They are shown with flow velocity as a parameter on the heat flux q versus wall temperature increase from the inlet temperature ΔT_L graph.   The heat transfer curves on each figure shift upward with the increase in the flow velocity. The nucleate boiling and transition to film boiling do not occur as shown in these figures.

The heat transfer curve for each flow velocity consists of a region with a higher gradient and that with a lower gradient. The highest temperature limit of the former region is slightly higher than the pseudo-critical temperature T'_{cr} at which the specific heat takes a maximum peak at the pressure. When ΔT_L exceed that point, the heat transfer coefficient h starts to decrease with an increase of ΔT_L.

It is well known that thermo-physical properties of fluids remarkably change for the fluid temperature higher than T'_{cr}. Low density (gas-like) flow exists in thermal boundary layer surrounding the heater wire and high density (liquid-like) fluid is flowing in the gap between the surface of the boundary layer and the inner surface of the conduit. The boundary layer would grow along the inner surface of the test wire with the increase in the wall temperature beyond T_{cr}'. The growth of the boundary layer would be the cause of the degradation of heat transfer from the former region. Thickness of the boundary layer would become thicker and mutual action between the main flow and the boundary layer would become significant with the increase in the wall temperature. We can see in figures 3 and 4 that heat transfer for each flow velocity tends to oscillate and becomes better at the ΔT_L of approximately 100 K. This may be due to the partial collapse of the boundary layer by the mutual action between them.
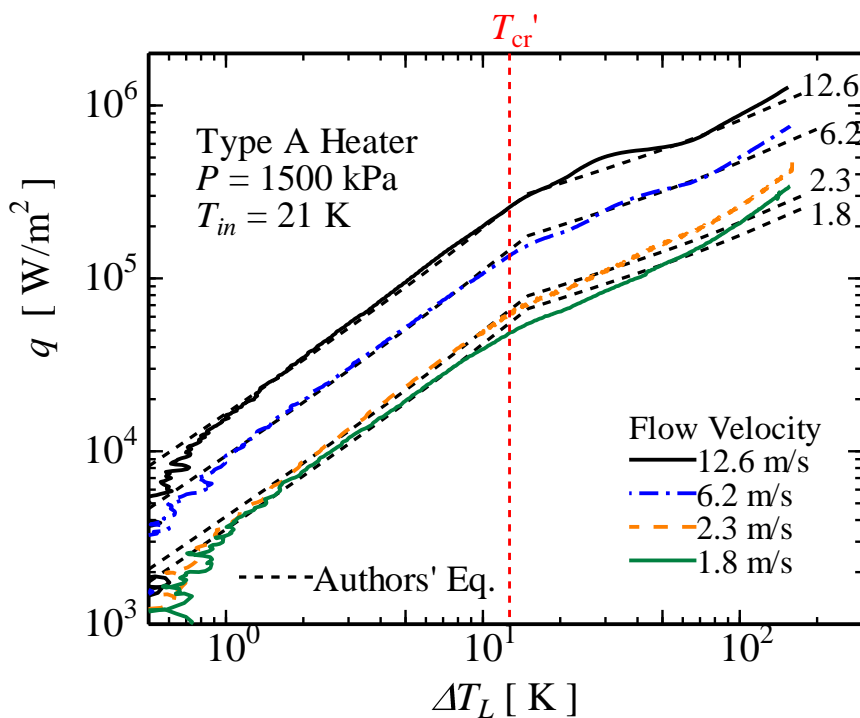

FIGURE 3. Heat transfer curves for Type A heater with flow velocity as a parameter.
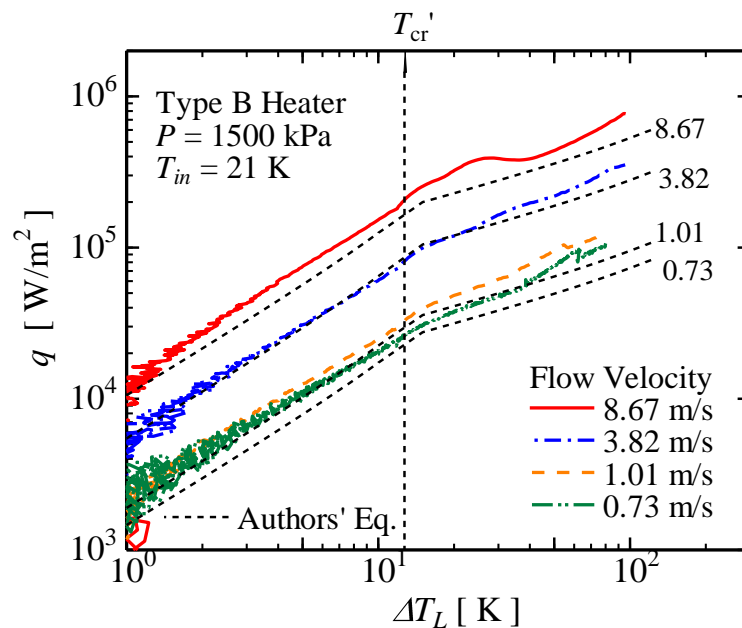
FIGURE 4. Heat transfer curves for the Type B heater with flow velocity as a parameter.

3.2 Comparison of heat transfer under supercritical pressure with that under a pressure lower than P_{cr}
The boiling curve with the same flow velocity at a pressure of 1.1 MPa, which is slightly lower than the critical pressure, are shown in figure 5 and figure 6 for the Type A and Type B heaters respectively. In the region where ΔT_L is lower than the pseudo-critical temperature T'_{cr}, the heat flux under supercritical pressure increases with an increase of heat input along with the curve predicted by Dittus-Boelter equation [6], where hydraulic equivalent diameter $D_e (= D_1 - D_2)$ was used as a typical length. The heat transfer at supercritical pressure agrees well with that at below critical pressure. Then, in the regions where ΔT_L is higher than T'_{cr}, the heat flux at the supercritical pressure is higher than that at below critical pressure.
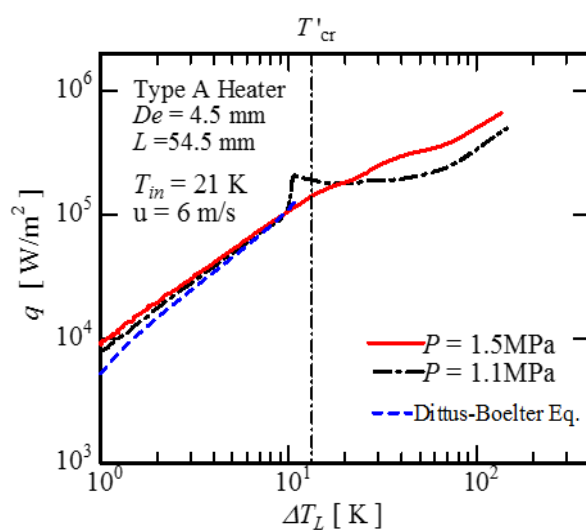

FIGURE 5. Typical heat transfer curve of supercritical hydrogen for the Type A heater in comparison with that below critical pressure.
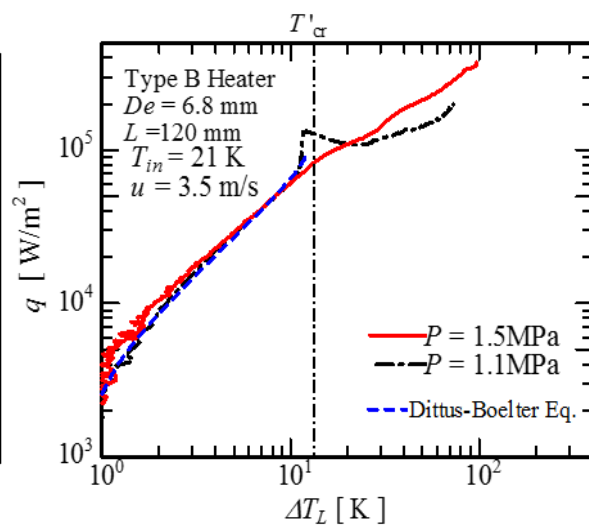
FIGURE 6. Typical heat transfer curve of supercritical hydrogen for the Type B heater in comparison with that below critical pressure.

4. Heat transfer correlation

Shiotsu et al. [4] presented the following correlation for forced flow heat transfer to supercritical helium based on their experimental data for a flat plate heater pasted on inner side of a rectangular duct. This correlation consists of the Dittus-Boelter equation with a correction factor F_c to express the degradation of heat transfer due to the growth of thermal boundary layer.

$$Nu_{De} = 0.023 Re_{De}^{0.8} \overline{Pr}^{0.4} F_c \qquad (1)$$

$$F_c = [1.0 + 108.7(L/D_e)^{-2}]^{0.25}[1 + 0.002(\Delta T_L / T_{cr}')](\rho_w / \rho_{in})^{0.34}(\mu_{in} / \mu_w)^{0.17} \qquad (2)$$

where Nu_{De} and Re_{De} are Nusselt and Reynolds numbers, $\overline{Pr} = \overline{c_p}\mu_{in}/\lambda_{in}$, $\overline{c_p} = (h_w - h_{in})/(T_w - T_{in})$, h is the enthalpy and suffixes w and in mean wall and inlet. The D_e is equivalent diameter, L is the heated length, the ρ, μ are density and viscosity.

Then we studied the heat transfer from inner wall of a vertical tube to forced flow of hydrogen under a supercritical pressure for the test tubes with various inner diameters and lengths [3]. We compared the experimental results with the equation (1). The thermophysical properties of para-hydrogen are given using the computer software 'GASPACK'. It was confirmed that the equation can be applicable for wide ranges of tube diameter and length.

The curves predicted by the equation (1) are shown in figures 3 and 4 for comparison.The experimental data almost agree with the predicted curves although fluctuation around the curves can be seen.   The fluctuation may be due to the growth and collapse of the boundary layer mentioned above and liquid-like fluid outside is compressible. This would be the characteristic of the undeveloped supercritical forced flow heat transfer.

To see the applicability of our correlation in more detail, all the experimental data are shown on $Nu_{De} \cdot \overline{Pr}^{-0.4} \cdot F_c^{-1}$ vs. Re_{De} graph in figure 7 in comparison with the equation (1). These data are for the Reynolds numbers ranging from about 1.8×10^4 to 2.9×10^5. Most of the data are within +35 % and -30 % of the equation. Though the error band is relatively wide due to the fluctuation, this correlation would be useful for a cooling design of HTC systems where relatively low temperature range is important.
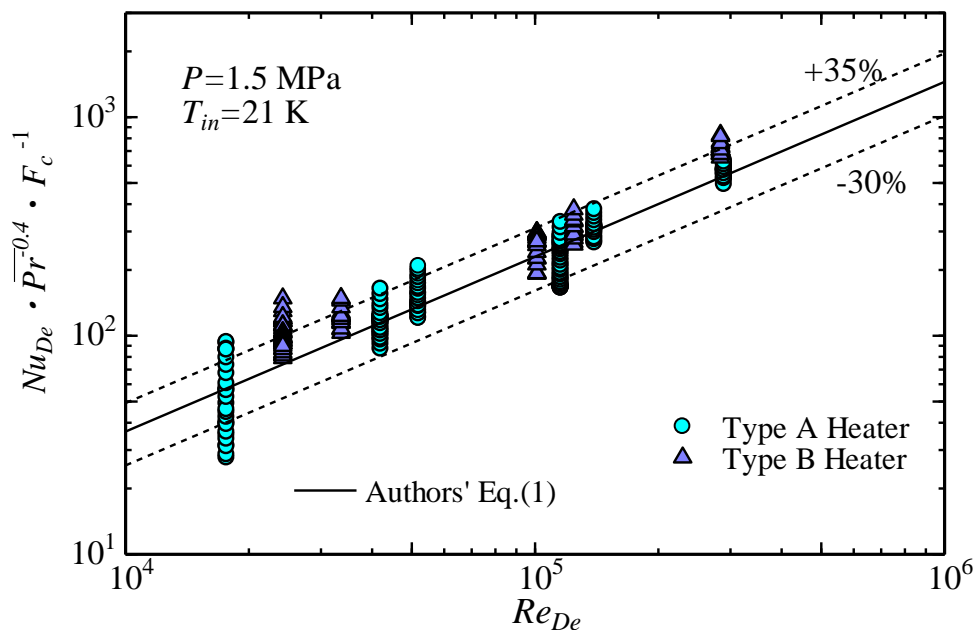

FIGURE 7. Comparison of the correlation with the experimental data.

5. Conclusions

Heat transfer from a PtCo wire in a vertically-mounted conduit to flowing supercritical hydrogen was measured for two different sized heater blocks.   The measurement was made at a pressure of 1.5 MPa for an inlet fluid temperature of 21 K and flow velocities from 0.5 to 12 m/s. Experimental results lead to the following conclusions:

   The heat transfer coefficients are higher for higher flow velocity. The heat transfer curve for each flow velocity consists of a region with a higher gradient and that with a lower gradient. The highest temperature limit of the former region is slightly higher than T'_{cr} of the fluid at the pressure.

   The heat transfer in the former region agrees well with the Dittus-Boelter equation and becomes lower than the equation with further increase in wall temperature.

   The experimental results were compared with the authors' equation of forced flow heat transfer under supercritical pressure. Most of the data for Re_{De} from 1.8×10^4 to 2.9×10^5 are within +35 % and -30 % of the equation.

   It was confirmed that the equation can be applicable for a central wire heater in a conduit. This would be useful for a cooling design of HTC systems where relatively low temperature range is important.

6. Acknowledgements

7. References

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8. Nomenclature

$\overline{c_p}$ $:(= (h_w - h_{in})(T_w - T_{in})^{-1})$, specific heat   (Jkg^{-1}K)

D :inner diameter of conduit   (m)

D_e $:(= D - D_2)$, equivalent diameter,   (m)

D_2 :heater diameter   (m)

$\overline{h}$ $:(= q / \Delta T_L)$, heat transfer coefficient   (Wm^{-2}K^{-1})

h :enthalpy   (Jkg^{-1})

L :test heater length   (m)

$\overline{Nu_{D_e}}$ $:(= \overline{h} D_e / \lambda_m)$,   average Nusselt number

P :pressure   (kPa)

P_{cr} :critical pressure   (kPa)

$\overline{Pr}$ $:(= \overline{c_p} \mu_{in} / \lambda_{in})$, Prandtl number

Q :heat generation rate   (Wm^{-3})

q :heat flux   (Wm^{-2})

Re_{D_e} $: (= uD_e / \nu_{in})$,   Reynolds number

T_{av} :average heater temperature   (K)

T_{in} :inlet liquid temperature   (K)

T_{cr}' :pseudo-critical temperature   (K)

T_w :heater surface temperature   (K)

u :velocity (ms^{-1})

ΔT_L $: (= T_w - T_{in})$ (K)

λ :thermal conductivity $(\text{Wm}^{-1}\text{K}^{-1})$

μ :viscosity $(\text{kg s}^{-1}\text{m}^{-1})$

ρ :density (kgm^{-3})

τ :exponential period   (s)

Subscripts

in: at inlet temperature,   *w*: at wall temperature