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Film boiling heat transfer from a round wire to liquid hydrogen flowing upward in a concentric annulus

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Abstract

Hydrogen film boiling heat transfer coefficients were measured for the heater surface superheats up to 400 K under pressures from 400 to 1100 kPa, liquid subcoolings from 0 to 11 K and flow velocities up to 7 m/s. The test wire used was 1.2 mm in diameter and 120 mm in length made of PtCo (0.5 wt.%) alloy, which was located at the center of 8 mm diameter conduit made of FRP (Fiber Reinforced Plastics). The heat transfer coefficients were higher for higher pressure, higher subcooling and higher flow velocity. The heat transfer coefficients were about 1.6 times higher than those predicted by Shiotsu-Hama equation for forced flow film boiling in a wide channel. Discussions were made on the mechanism of difference between them.

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Keywords: Film boiling, Forced convection, Liquid hydrogen, Annulus

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1. Introduction

Knowledge of film boiling heat transfer from a heated wire to forced flow of liquid hydrogen in a narrow gap is important for conductor design and quench analysis of superconducting magnets cooled by liquid hydrogen. However there have been few experimental data as far as we know. Recently, we have performed a series of experimental studies on liquid hydrogen cooling. An experimental system without using a pump was developed (Tatsumoto et al.(2010)). Boiling heat transfer characteristics in a pool was reported by Shirai et al. (2010). Forced flow boiling heat transfer and its Departure from Nucleate Boiling (DNB) heat flux for wide range of pressures below the critical pressure was reported by Shirai et al. (2011) .

The purpose of this study is twofold. The first aim is to obtain the experimental data of forced convection heat transfer from a round wire to liquid hydrogen flowing upward in concentric annulus with a narrow gap. The second aim is to clarify whether the experimental data can be described by conventional correlation.

Nomenclature

Gr	Grashof number	T_{sat}	saturation temperature (K)
h	boiling heat transfer coefficient ($Wm^{-2}K^{-1}$)	T_w	heater surface temperature (K)
$\overline{Nu}$	average Nusselt number	z	test heater length (m)
P	pressure (kPa)	ΔT_L	$T_w - T_B$, surface temperature increase from
Pr	Prandtl number		inlet liquid temperature (K)
Q	heat generation rate (Wm^{-3})	ΔT_{sat}	$T_w - T_{sat}$, heater surface superheat (K)
q	heat flux (Wm^{-2})	ΔT_{sub}	$T_{sat} - T_B$, liquid subcooling (K)
Re	Reynolds number	ρ	density (kgm^{-3}) μ viscosity ($kg\ s^{-1}m^{-1}$)
Sc	non-dimensional subcooling	σ	surface tension (Nm^{-1}) τ exponential period (s)
Sp	non-dimensional superheat		
T_{av}	average heater temperature (K)	<u>subscripts</u>	
T_B	inlet liquid temperature (K)	l liquid; v vapor ; z based on z	

2. Experimental apparatus and methods

Fig. 1 shows a schematic of the experimental system, whose details have already been presented by Tatsumoto et al. (2010). It consists 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. The test heater block is located at one end of the transfer tube in the main tank. Forced flow is achieved by the pressure difference between the cryostat and sub tank and the valve opening. The mass flow rate is estimated by the flow rate of the feed hydrogen gas and the weight change of the main tank, which is put on a scale (MettlerToledo WMHC 300 s) which can measure up to 400 kg within 0.002 kg resolution. The bath temperature, T_B, are measured by Cernox temperature sensors with the accuracy of 10 mK. The hydrogen temperature is controlled by a sheathed heater with the maximum power of 500 W that is wounded around the bottom of the main tank.

Fig. 2 shows the test heater made of PtCo (0.5 wt.%) alloy wire, 1.2 mm in diameter, 120 mm in length supported at the center of 8 mm diameter conduit in a block made of fiber reinforced plastic (FRP). The heating current to the test heater is supplied by a power amplifier (max. 400 A at a power level of 4.8 kW). The input signal of the power amplifier is controlled so that the heat generation rate in the test heater agreed with a desired value. In this study, exponential heat generation rate of $Q = Q_0 e^{t/\tau}$ with $\tau = 10.0$ s up to a certain value $Q = Q_m$ at $t = t_m$ (at point E in Fig. 3) and $Q = Q_m e^{(t_m - t)}$ for $t > t_m$ are applied to the test wire. As the heat transfer characteristics and values of DNB heat fluxes were not affected by the variation of τ from 3 s to 10 s, the heat transfer phenomenon for the heat generation rate with $\tau = 10.0$ s could be regarded as a continuous sequence of steady-state. Average temperature of the test heater T_{av} was measured by resistance thermometry using a double bridge circuit including the test heater as a branch of the bridge. The double bridge circuit is first balanced at the bath temperature. The output voltages of the bridge circuit due to the deviation of the heater resistance during the current heating are amplified, simultaneously sampled at the interval of 30 ms and stored in a 16 bit digital memory system. The surface heat flux q was obtained as the difference between the heat generation rate Q and the rate of change of energy storage in the test heater.

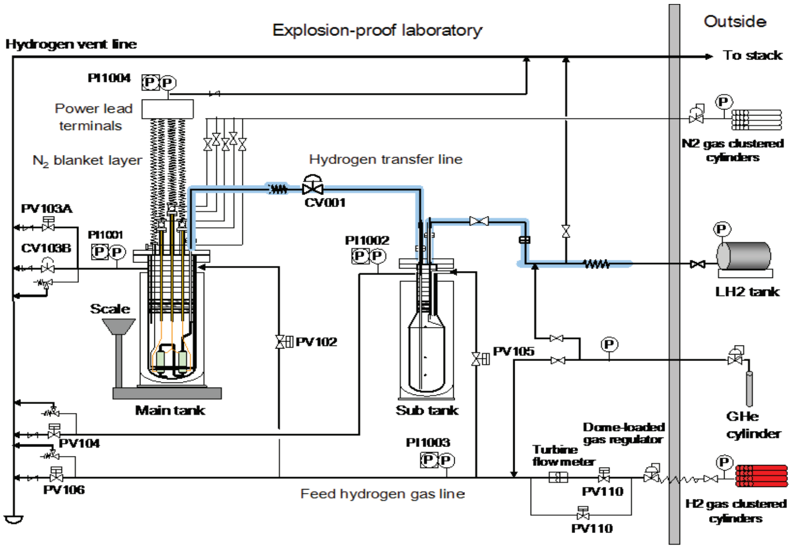

Fig. 1 Schematic of the experimental apparatus.

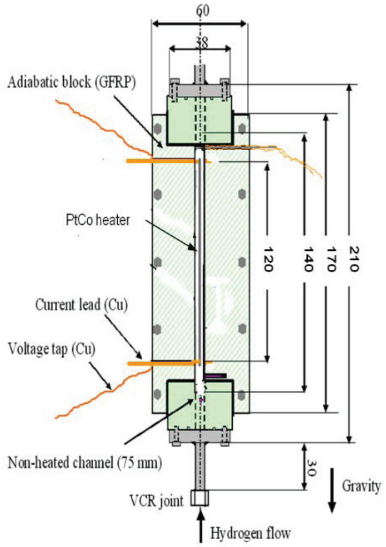

Fig. 2 Schematic of test heater block.

3. Results and discussion

Film boiling heat transfer coefficients were measured for the heater surface superheats up to 400 K under pressures from 0.4 to 1.1 MPa, the liquid is subcooled between 0 and 11 K and with flow velocities up to 7 m/s. Fig.3 shows a typical process to measure film boiling heat transfer without too much thermal shock to the test heater. Longitudinal axis is heat flux q and abscissa is wall temperature increase from inlet temperature. Firstly the heat generation rate was gradually increased for a low flow rate (0.9 m/s). Boiling initiates at point A. The process from A to B is nucleate boiling regime. When the heat flux reaches the DNB heat flux (point B), heater temperature jumps to film boiling for 0.9 m/s (point C). Transient heat transfer process during the jump is shown in the figure. The process from B to the bottom of heat flux is the transition boiling and the bottom to the point C is film boiling. Then flow velocity is increased to a desired value (here 7.23 m/s) while heating current is continuously increased to the heater temperature around 400 K. Then the heating current is decreased exponentially and film boiling heat transfer coefficients are measured.

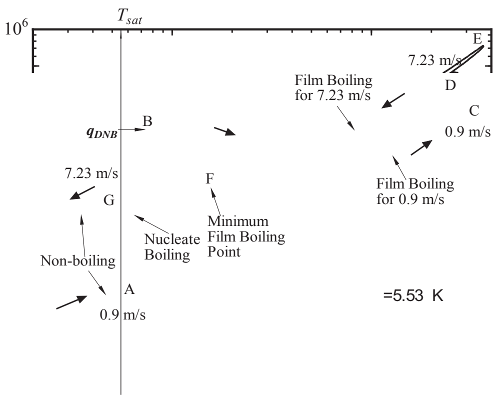

Fig. 3 Heat transfer process to measure film boiling heat transfer.

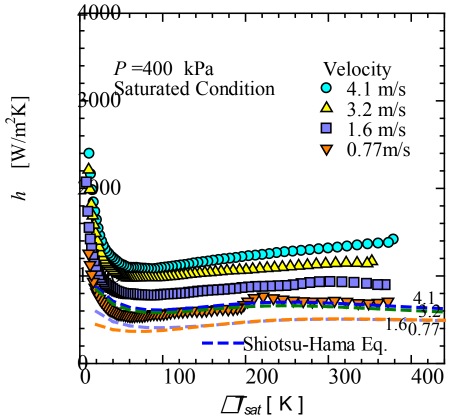

Fig. 4. Film boiling heat transfer coefficients for $\Delta T_{sub} = 0$ K at 400 kPa.

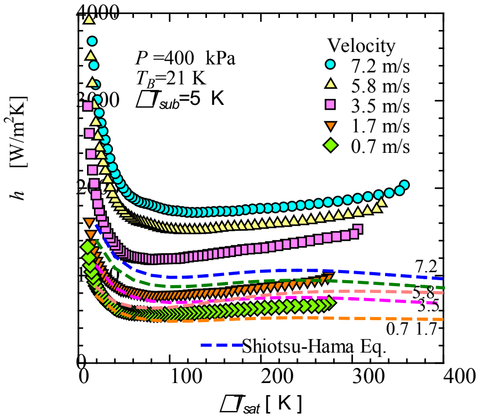

Fig. 5. Film boiling heat transfer coefficients for $\Delta T_{sub} = 5$ K at 400 kPa.

Fig.4 shows the relation between film boiling heat transfer coefficient $h = q / \Delta T_{sat}$ versus wall superheat for *P*=400 kPa under saturated condition with flow velocity as a parameter. We can see from the figure that film boiling heat transfer coefficients for each flow velocity gradually decreases or is kept almost constant for ΔT_{sat} down to about 80 K. It becomes significantly higher with the decrease of ΔT_{sat} from the value. This would be because the thickness of the vapor film becomes very thin with the decrease of wall superheat. The film boiling heat transfer coefficients are higher for higher flow velocity. For the slow velocity under saturated condition, small fluctuation of heat transfer coefficient was observed.

Fig. 5 shows the relation between film boiling heat transfer coefficient and ΔT_{sat} for *P*=400 kPa and a liquid subcooling of 5 K with flow velocity as a parameter. Trend of the dependence on ΔT_{sat} and flow velocity is similar to that for saturated condition mentioned above. By comparing the data for 4.1 m/s in Fig. 4 with those for 3.5 m/s in Fig. 5, we can see that the heat transfer coefficients are higher for higher subcooling.

Figs. 6 and 7 show the relation between film boiling heat transfer coefficient versus ΔT_{sat} for *P*=700 kPa under saturated condition and subcooling of 8 K, respectively. And Figs. 8 and 9 show the relation between film boiling heat transfer coefficient and ΔT_{sat} for *P*=1100 kPa under saturated condition and subcooling of 11 K. By comparing the data in these figures, we can see that the heat transfer coefficients are higher for higher pressure and subcooling.

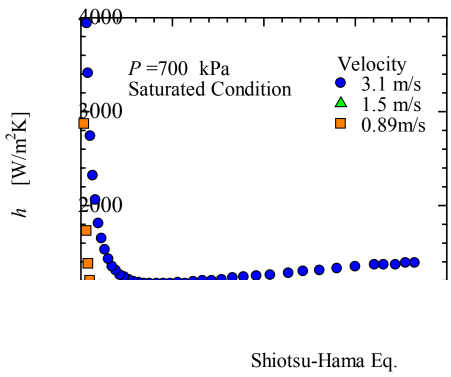

Fig. 6. Film boiling heat transfer coefficients for $\Delta T_{sub} = 0$ K at 700 kPa.

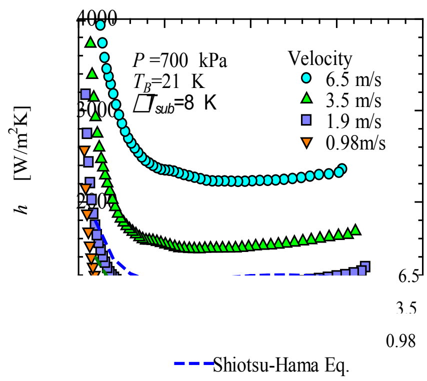

Fig. 7. Film boiling heat transfer coefficients for $\Delta T_{sub} = 8$ K at 700 kPa.

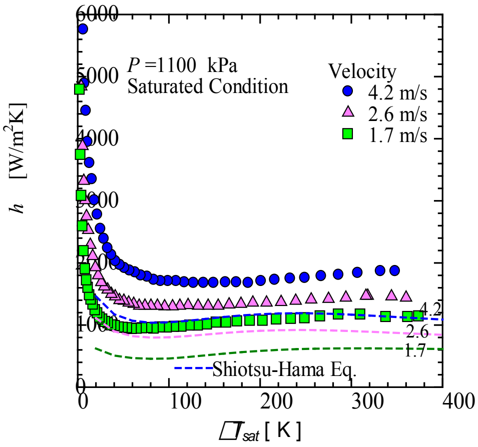

Fig. 8. Film boiling heat transfer coefficients for $\Delta T_{sub} = 0$ K at 1100 kPa.

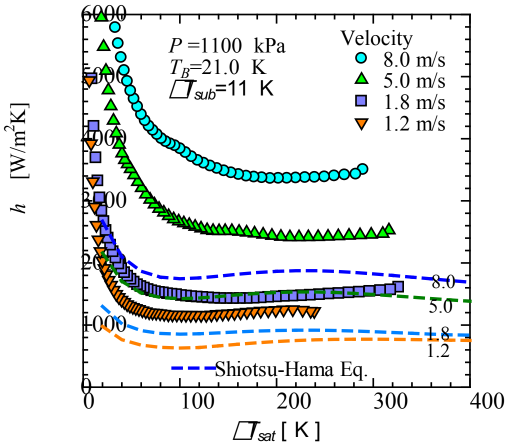

Fig. 9. Film boiling heat transfer coefficients for $\Delta T_{sub} = 11$ K at 1100 kPa.

At the pressure of 1100 kPa and liquid subcooling of 11 K for instance, the film boiling heat transfer coefficients for velocities are about 70 % higher than those for nearly the same flow velocities at the pressure of 400 kPa.

To see the effect of flow velocity more clearly, the heat transfer coefficients at $\Delta T_{sat} = 200$ K are shown in Fig. 10 versus flow velocity. It can be seen that the heat transfer coefficients at each pressure and subcooling increase proportional to square root of the flow velocity.

4. Comparison with conventional correlation

There have been few experimental data and correlation of forced flow film boiling heat transfer in liquid hydrogen. Shiotsu and Hama (2000) studied the saturated and subcooled film boiling heat transfer for a vertical cylinder in forced flow of water and R113 in 40 mm dia. cylindrical conduit. They derived the correlation of forced convection film boiling heat transfer by modifying an approximate analytical solution based on their experimental data (see Appendix A).

The values predicted by the equation are shown in Figs. 4 to 9 for comparison. The experimental data are about 1.7 times higher than the predicted values, although the trend of dependence on flow velocity (proportional to square root of flow velocity, namely square root of Reynolds number) is similar to that predicted by the equation. Shiotsu-Hama equation is based on the experimental data for wide conduit as mentioned above. Vapor film layer around the wire heater may be made thinner by a very narrow gap of 3.4 mm in this study.

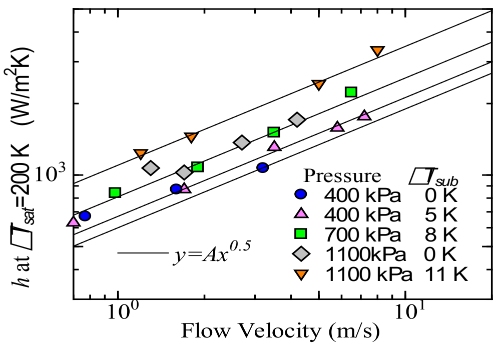

Fig. 10. Film boiling heat transfer coefficients at $\Delta T_{sat} = 200$ K versus flow velocity.

5. Conclusion

Film boiling heat transfer coefficients were measured for the heater surface superheats up to 400 K under pressures from 400 to 1100 kPa, liquid subcoolings from 0 to 11 K and flow velocities up to 7 m/s. The experimental results lead to the following conclusion. Film boiling heat transfer coefficients are higher for higher pressure, flow velocity and subcooling. The heat transfer coefficients at each pressure and subcooling increase proportional to the square root of the flow velocity. Film boiling heat transfer coefficients are about 1.7 times higher than those predicted by Shiotsu and Hama equation based on approximate solution of two phase layer model and data for water and R113 in 40 mm dia. cylindrical conduit. The trend of dependence on flow velocity (proportional to square root of flow velocity) is similar to that predicted by the equation. Vapor film layer around the wire heater may be made thinner by a very narrow gap of 3.4 mm in this study.

Acknowledgments

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Appendix A

Forced flow film boiling correlation by Shiotsu and Hama (2000):

$$\frac{\bar{N}u}{\sqrt{\mathrm{Re}_z}}\frac{\mu_v}{\mu_l} = 0.53\left[\frac{Sp}{R^2}\left(1+\frac{E_2}{2Pr_l Sp}\right)\left(1-0.7\frac{Sc}{E_2}\right)\right]^{-1/3} z'^{1/4} \qquad \text{for} \quad Re_z \geq F_v \tag{1}$$

$$\overline{Nu} = 0.52 z'^{1/4} M_z^{1/4} \qquad \text{for} \quad Re_z < F_v \tag{2}$$

where $F_v = 0.96 M_z^{1/2}\left(\mu_v/\mu_l\right)^2\left[\left(Sp/R^2\right)\left\{1+E_2/\left(2\,\mathrm{Pr}_l\,Sp\right)\right\}\left(1-0.7Sc/E_2\right)\right]^{2/3}$, $z' = z\left[g(\rho_l-\rho_v/\sigma\right]^{1/2}$, $R = (\rho_v\mu_v/\rho_l\mu_l)^{1/2}$. E_2 is a positive root of the following equation

$$E_2^3 + (5\,\mathrm{Pr}_l\,S_p - S_c)E_2^2 - 5\,\mathrm{Pr}_l\,S_p S_c E_2 - (15/2)\mathrm{Pr}_l^2\,S_p^2 R^2 = 0 \; ; \tag{3}$$

$$M_z = \left[Gr_z\,\mathrm{Pr}_v/Sp\right]\left[E^3/\left\{1+E/\left(Sp\,\mathrm{Pr}_l\right)\right\}\right]/\left(R\,\mathrm{Pr}_l\,Sp\right)^2 ;$$

$$E = \left(A+C\sqrt{B}\right)^{1/3} + \left(A-C\sqrt{B}\right)^{1/3} + (1/3)Sc^* ; \quad A = (1/27)Sc^{*3} + (1/3)R^2 Sp\,\mathrm{Pr}_l\,Sc^* + (1/4)R^2 Sp^2\,\mathrm{Pr}_l^2 ;$$

$$B = (-4/27)Sc^{*2} + (2/3)Sp\,\mathrm{Pr}_l\,Sc^* - (32/27)Sp\,\mathrm{Pr}_l\,R^2 + (1/4)Sp^2\,\mathrm{Pr}_l^2 + (2/27)Sc^{*3}/R^2 ;$$

$$C = (1/2)R^2 Sp Pr_l ; \quad Sc^* = 0.93 Pr_l^{0.22} Sc .$$

Equation (2) is the pool film boiling equation presented by Sakurai et al. (1992).