



25th International Cryogenic Engineering Conference and the International Cryogenic Materials Conference in 2014, ICEC 25–ICMC 2014

## Film boiling heat transfer properties of liquid hydrogen in natural convection

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### Abstract

Film boiling heat transfer properties of LH<sub>2</sub> for various pressures and subcooling conditions were measured by applying electric current to give an exponential heat input to a PtCo wire with a diameter of 1.2 mm submerged in LH<sub>2</sub>. The heated wire was set to be horizontal to the ground. The heat transfer coefficient in the film boiling region was higher for higher pressure and higher subcooling. The experimental results are compared with the equation of pool film boiling heat transfer. It is confirmed that the pool film boiling heat transfer coefficients in LH<sub>2</sub> can be expressed by this equation.

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Peer-review under responsibility of the organizing committee of ICEC 25-ICMC 2014

*Keywords:* Liquid hydrogen; Heat transfer; Film boiling; Natural convection; Horizontal wire;

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### 1. Introduction

Liquid hydrogen (LH<sub>2</sub>) has excellent properties as a coolant for HTS conductors. It has lower viscosity than liquid nitrogen and larger latent heat of evaporation than liquid helium. Also the HTS conductors cooled by LH<sub>2</sub> have high

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critical current and specific heat around LH<sub>2</sub> temperature. So, LH<sub>2</sub> is expected to be used as a coolant to make the performance of HTS conductors better and more stable.

In operating superconducting devices cooled by LH<sub>2</sub>, joule heating caused by normal conducting transition sometimes results in film boiling which prevents the effective cooling and leads to the rapid temperature rise of superconducting devices. Knowledge of heat transfer in film boiling is important for using superconducting devices cooled by LH<sub>2</sub> safely.

In this paper, we report the film boiling heat transfer properties of LH<sub>2</sub> in natural convection as a preliminary step to a forced flow film boiling experiment. We have obtained the experimental data of pool film boiling heat transfer from a horizontal PtCo wire for various pressures, liquid-subcoolings, and surface superheats. The experimental data of the heat transfer coefficient in the range of film boiling is compared with the equation of pool film boiling heat transfer presented by Sakurai et al. [1].

### Nomenclature

$A, B, C, E$	non-dimensional quantity in Eq.(3)	$Q$	heat input, W
$c_p$	specific heat capacity, J/(kg K)	$S$	surface area of PtCo wire, m <sup>2</sup>
$d$	diameter of PtCo wire, m	$T_{ave}$	average temperature of PtCo wire, K
$D'$	non-dimensional cylinder diameter	$V$	volume of PtCo wire, m <sup>3</sup>
$Gr_v$	Grashoff number for horizontal cylinder	$\Delta T_L$	the excess heater surface temperature beyond the bulk LH <sub>2</sub> temperature, K
$g$	acceleration due to gravity, m/s <sup>2</sup>	$\Delta T_{sat}$	the excess heater surface temperature beyond saturated temperature, K
$h$	heat transfer coefficient, W/(m <sup>2</sup> K)	$\Delta T_{sub}$	liquid subcooling, K
$L$	length of PtCo wire between potential taps, mm	$\mu$	viscosity, kg/(s m)
$L_h$	latent heat of vaporization, J/kg	$\rho$	density, kg/m <sup>3</sup>
$Nu_v$	average Nusselt number	$\sigma$	surface tension, N/m
$Pr$	Prandtl number		
$q$	heat flux, W/m <sup>2</sup>		

### Subscripts

$l$	liquid	$tw$	PtCo test wire
$v$	vapour		

## 2. Experimental apparatus and method

### 2.1. Experimental system

We have developed a thermal-hydraulics experimental system for LH<sub>2</sub> in order to investigate the heat transfer characteristics for a wide range of subcoolings and pressures [2-4]. A schematic of the experimental system is shown in Fig. 1. This system consists of a main cryogenic tank, a sub cryogenic tank, a control valve, a hydrogen gas feed line coming from hydrogen gas clustered cylinders, a vent line, and a power lead. The main tank is a vacuum insulated cylindrical stainless steel vessel with the inner diameter of 406 mm and the height of 1495 mm. The maximum inventory of LH<sub>2</sub> is 50 L. The main tank is pressurized to a desired pressure value by hydrogen gas. The designed pressure of the main tank is 2.1 MPa. The temperature of LH<sub>2</sub> in the main tank is controlled by a sheath heater situated at the bottom of the tank. Its temperature is measured by Cernox sensors.

### 2.2. Test heater

The test heater is a PtCo wire with the diameter of 1.2 mm. Two 100- $\mu$ m diameter platinum potential taps are spot welded at around 10 mm from each end of the test heater. The effective length of the heater between the potential taps is 101.8 mm. The test heater is set to be horizontal to ground in the main tank. The heater's electrical resistance is calibrated for various temperatures before using it in the experiment.

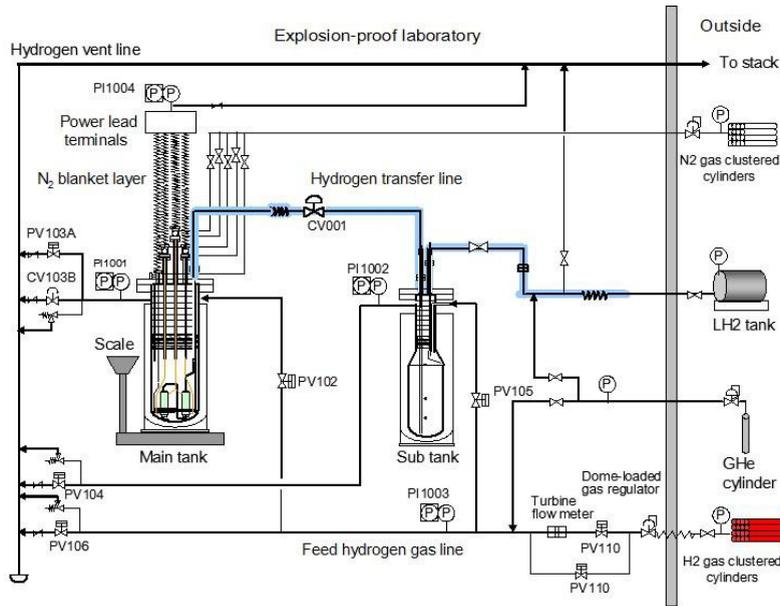


Fig. 1. Overview of experimental system.

### 2.3. Experimental method and condition

The heater is submerged in LH<sub>2</sub>. The pressure in the main tank is set to a desired value and the bulk temperature is controlled by using the sheath heater. After the pressure and temperature are set, the test heater is heated by applying an electric current to give an exponential heat input expressed by the following equation:

$$Q = Q_0 \exp\left(\frac{t}{\tau}\right), \quad (1)$$

where  $\tau$  is the exponential period ( $\tau = 10$  s). It is confirmed experimentally that the heat transfer phenomenon by this heat input generation rate is able to be regarded as a continuous sequence of steady state. In the previous experiments of heat transfer for LH<sub>2</sub>, the heat transfer curves with  $\tau$  larger than 2 s were almost the same. Additionally, it was confirmed that even when the heat input increasing with  $\tau = 10$  s was stopped and kept constant at a certain level in the middle of increasing, the  $q-\Delta T_L$  point stayed exactly on the heat transfer curve, that is assumed to be the steady state heat transfer curve.

The heat input is increased exponentially until the temperature of the test heater rises by about 400 K and then reduced exponentially to zero.

All systems such as valve control, current control, and measuring system are remotely operated by a computer connected via an optical network. In emergency situation, the interlock system shuts down all heater power sources and opens all the vent valves immediately.

Signal of heating current and voltage between potential taps of the test heater are amplified, simultaneously sampled at the interval of 30 ms and stored in a 16 bit memory system of a computer. The average temperature of the heater  $T_{ave}$  is obtained by a resistance thermometry using the relation between the electrical resistance of the test heater and its temperature calibrated previously.

The surface heat flux  $q$  is calculated by Eq. (2) from the difference between the heat input rate per unit surface area and the rate of energy storage in the heater.

The surface temperature of the heater is calculated from the measured average temperature and the heat flux by solving the one-dimensional thermal conduction equation in the radial direction of the heater. Experiments of film

Table 1. Experimental conditions.

System pressure	0.1, 0.4, 0.7, 1.1 MPa
LH <sub>2</sub> temperature	21 ~ 32 K
Maximum superheat	400 K

boiling on the horizontal test heater are performed for various pressures, bulk temperatures, and test heater surface superheats. Experimental conditions are shown in Table 1.

$$q(t) = \frac{Q - \rho_{tw} c_{ptw} V \frac{\partial T_{ave}}{\partial t}}{S} \quad (2)$$

### 3. Results and discussion

#### 3.1. Typical boiling heat transfer process

The typical result of boiling curves is shown in Fig. 2. The vertical axis is the heat flux  $q$  and the horizontal axis is the excess heater surface temperature beyond the bulk LH<sub>2</sub> temperature  $\Delta T_L$ .

As shown in Fig. 2, with increase of heat input, the heat flux at non-boiling region gradually increases along the curve predicted by the natural convection equation presented by McAdams [5] ( $Nu_v = 0.53(Gr_v Pr)^{1/4}$ ). Then, when the heater surface temperature reaches the saturation temperature, nucleate boiling starts. In nucleate boiling region,  $\Delta T_L$  does not increase very much while the heat flux becomes higher. When the heat flux reaches Critical Heat Flux, the heater temperature rapidly increases with a decrease of heat flux to film boiling regime. Heat input is further increased until the heater temperature reaches around 400 K. Then the heat input is decreased exponentially and film boiling heat transfer coefficients are measured down to the minimum heat flux point ( $\Delta T_L \cong 20$  K).

#### 3.2. Effect of pressure on film boiling heat transfer coefficient

Fig. 3 shows film boiling heat transfer coefficients ( $h = q / \Delta T_{sat}$ ) under saturated condition with pressure as a parameter. The horizontal axis is the excess heater surface temperature beyond saturation temperature  $\Delta T_{sat}$ . The value of  $h$  for each pressure decreases with the decrease of wall superheat and takes the minimum at around  $\Delta T_{sat} = 80$  K. Then it increases significantly with the decrease of wall temperature. As shown in Fig. 3, the heat transfer coefficient  $h$  in film boiling region is higher for higher pressure.

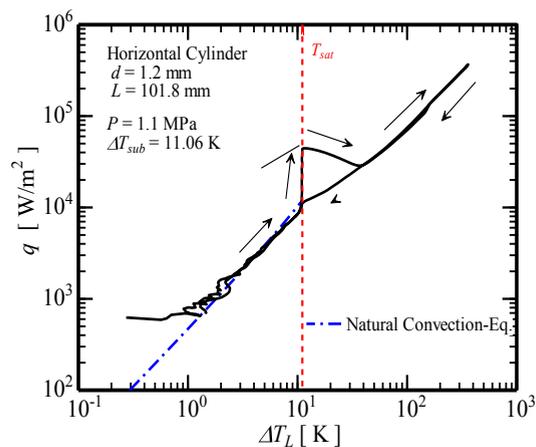


Fig. 2. Typical result of boiling curve in natural convection for the PtCo wire.

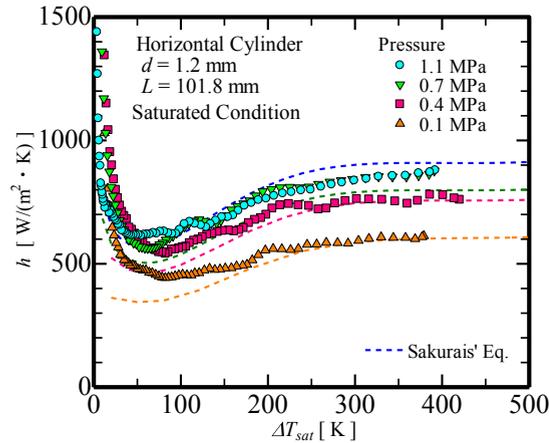


Fig. 3. Saturated film boiling heat transfer coefficients against surface superheat with pressure as a parameter.

### 3.3. Effect of subcooling on film boiling heat transfer coefficient

Fig. 4 shows the values of  $h$  versus  $\Delta T_{sat}$  with subcooling as a parameter at pressure of (a) 1.1 MPa, (b) 0.7 MPa, and (c) 0.4 MPa respectively. Trend of dependence on  $\Delta T_{sat}$  is similar to that for the saturated condition mentioned above. The values of  $h$  are higher for higher subcooling. The effect of subcooling is comparatively smaller than that of pressure because it is difficult to realize the sufficiently high subcooling for LH<sub>2</sub>.

## 4. Comparison with conventional correlation

Sakurai et al. [1] presented the following correlation of pool film boiling heat transfer from a horizontal cylinder by slightly modifying their solution of two-phase boundary layer film boiling model based on experimental data for various liquids:

$$Nu_v / (1 + 2/Nu_v) = K(D') [M^*]^{1/4}, \tag{3}$$

$$\begin{aligned}
 K(D') &= 0.415 D'^{1/4} && \text{for } D' > 6.6, \\
 K(D') &= 2.1 D' / (1 + 3.0 D') && \text{for } 1.25 \leq D' \leq 6.6, \\
 K(D') &= 0.75 / (1 + 0.28 D') && \text{for } 0.14 \leq D' \leq 1.25,
 \end{aligned}$$

where

$$\begin{aligned}
 M^* &= [Gr_v / (Sp)] [E^3 / \{1 + E / (Sp Pr_l)\}] / (R Pr_l Sp)^2, & E &= (A + C\sqrt{B})^{1/3} + (A - C\sqrt{B})^{1/3} + 1/3 Sc^* \\
 A &= (1/27) Sc^{*3} + (1/3) R^2 Sp Pr_l Sc^* + (1/4) R^2 Sp^2 Pr_l^2 \\
 B &= (-4/27) Sc^{*2} + (2/3) Sp Pr_l Sc^* - (32/27) Sp Pr_l R^2 + (1/4) Sp^2 Pr_l^2 + (2/27) Sc^{*3} / R^2 \\
 C &= (1/2) R^2 Sp Pr_l, & D' &= d [g (\rho_l - \rho_v) / \sigma]^{1/2}, & L' &= L_h + 0.5 c_{pv} \Delta T_{sat} \\
 R &= [\rho_v \mu_v / \rho_l \mu_l]^{1/2}, & Sp &= c_{pv} \Delta T_{sat} / (L' Pr_v), & Sc^* &= 0.93 Pr_l^{0.22} Sc.
 \end{aligned}$$

The experimental results were compared with the equation of pool film boiling heat transfer presented by Sakurai et al. [1]. The curves predicted by Eq. (3) are shown in Fig. 3 and Fig. 4 as broken lines. The values of  $D'$  for 1.2 mm diameter wire in LH<sub>2</sub> are 0.7, 0.9, 1.2, and 2.1 for the pressures of 0.1, 0.4, 0.7, and 1.1 MPa, respectively. The experimental data agrees with the values calculated from the Sakurai's equation within -15 to +10 percent error as shown in these figures, although the data for the range of  $\Delta T_{sat} < 100$  K is 20~40 percent different from the value predicted by Eq. (3). This reason is probably that the thermal conductivity of vapor hydrogen varies largely depending on temperature for this range. It is necessary to confirm whether the thermal conductivity of vapor hydrogen for this range can be calculated from the average temperature of the vapor with numerical analysis.

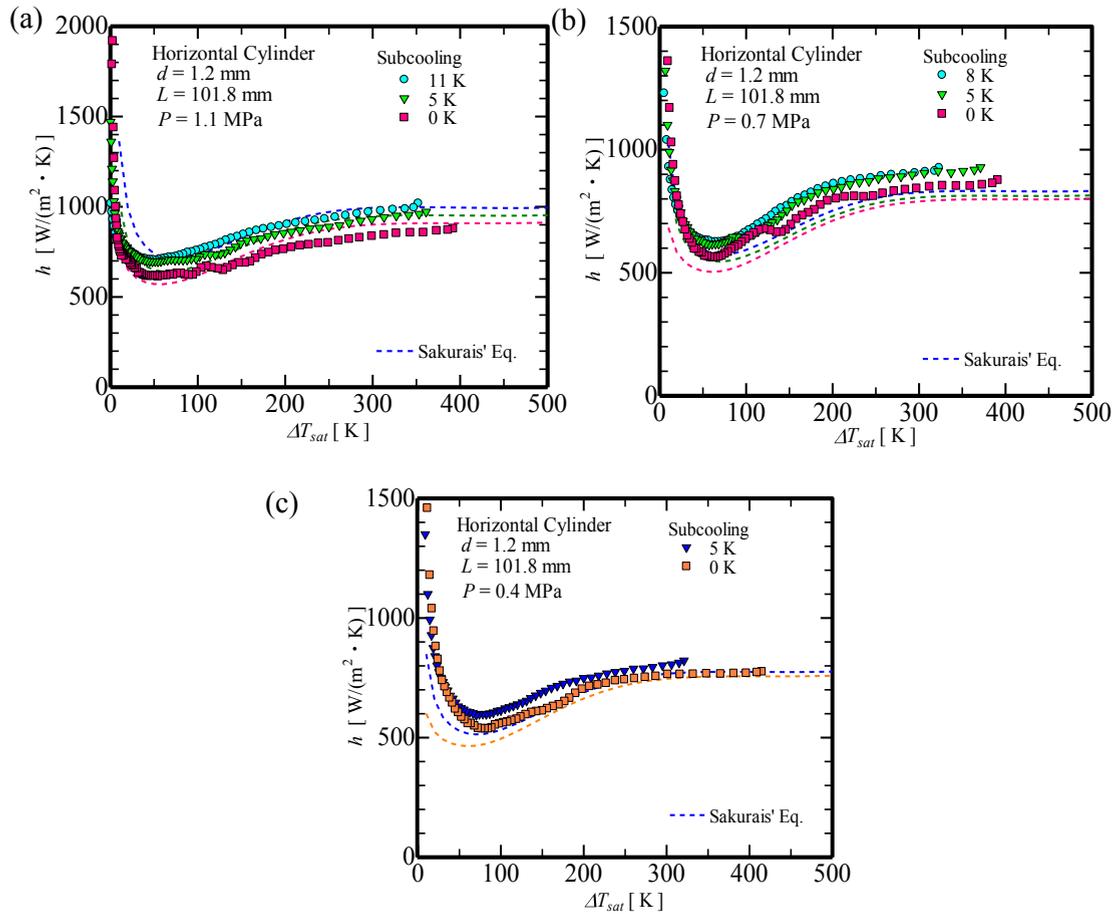


Fig. 4. Film boiling heat transfer coefficients against surface superheat with subcooling as a parameter.  
(a)  $P = 1.1$  MPa, (b)  $P = 0.7$  MPa, (c)  $P = 0.4$  MPa.

## 5. Conclusion

Film boiling heat transfer coefficients for  $\text{LH}_2$  in natural convection on horizontal wire were obtained for various system pressures and bulk temperatures.

The heat transfer coefficient  $h$  in pool film boiling region was higher for higher pressure and higher subcooling.

It is confirmed that the pool film boiling heat transfer coefficients in  $\text{LH}_2$  can be expressed well by the equation presented by Sakurai et al., although the data for the range of  $\Delta T_{sat} < 100$  K is 20~40 percent different from the value predicted by Eq. (3).

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