Evaluation of Thermocouple Fin Effect in Cladding Surface Temperature Measurement during Film Boiling

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Thermocouple fin effect on surface temperature measurement of a fuel rod has been studied at elevated wall temperatures under film boiling condition in a reactivity initiated accident (RIA) situation. This paper presents an analytical equation to evaluate temperature drops caused by the thermocouple wires attached to cladding surface. The equation yielded the local temperature drop at measuring point depending on thermocouple diameter, cladding temperature, coolant flow condition and vapor film thickness. The temperature drops by the evaluating equation were shown in cases of free and forced convection conditions. The analytical results were compared with the measured data for various thermocouple sizes, and also with the estimated maximum cladding temperature based on the oxidation layer thickness in the cladding outer surface.

It was concluded that the temperature drops at above 1,000°C in cladding temperature were around 120 and 150°C for 0.2 and 0.3 mm diameter Pt-PtRh thermocouples, respectively, under a stagnant coolant condition. The fin effect increases with the decrease of vapor film thickness such as under forced flow cooling or at near the quenching point.

KEYWORDS: thermocouple fin effect, temperature measurement, cladding temperature, film boiling, RIA, NSRR, vapor film thickness, two-phase flow, quench temperature, cladding oxidation, fuel pellet-cladding eccentricity, heat transfer

I. INTRODUCTION

In-reactor experimental research on fuel behavior under reactivity initiated accident (RIA) conditions is being conducted in the Nuclear Safety Research Reactor (NSRR) Program at Tokai Research Establishment, Japan Atomic Energy Research Institute. The threshold energy depositions for fuel failure and the consequences of the fuel failure have been investigated for unirradiated light water reactor fuels under RIA conditions. The effects of various fuel design and cooling condition parameters on the fuel behaviors are also being investigated.

In these experiments, it is very important to measure the cladding temperature accurately, because the cladding temperature, especially that at maximum, is one of the dominant factors governing the fuel failure. In the RIA conditions, the cladding temperature begins to increase very rapidly just after the power burst with the occurrence of departure from nucleate boiling (DNB), and reaches its maximum after around 1 s. Then, the cladding surface is kept at elevated temperature for about 10 s in steady film boiling regime until quenching initiation. In order to attain a responsive measurement of cladding surface temperature in the RIA transient, fine Pt and Pt-13%Rh bare-wire thermocouples, 0.2 or 0.3 mm in diameter, are attached directly on the cladding surface by spot-welding. Local
cooling effect by the fine thermocouple wire, however, may not be negligibly small, since the temperature difference between the cladding wall and ambient fluid becomes very large under low heat transfer capability of film boiling. This effect was revealed by the metallurgical examination of irradiated fuel rods as a local decrease of oxidation layer thickness around a thermocouple. The local temperature drop was estimated to be as large as 300°C by the oxidation layer thickness variation.

In the present paper, the fin effect of thermocouple wire is evaluated analytically and the sensitivity of the effect for wire diameter, vapor film thickness and coolant flow rate are discussed. The results of this analysis are also compared with the measured data by thermocouples of various sizes and with the estimated values based on the metallurgical examination.

II. ANALYSIS

There exist many analyses on thermal conduction along a fin of various shapes on heating wall. However, most of these are for the single phase, and not applicable to the present film boiling situation. Consequently, two layer model has to be introduced for the present situation.

1. Physical Model and Fundamental Equations

The physical model of the system is shown in Fig. 1. The assumptions are as follows:

(1) Cladding is assumed as an infinite flat plate, since its diameter is much larger than that of thermocouple attached vertically on it.

(2) Temperature drop across the cladding wall is neglected in comparison with the temperature difference in tangential direction caused by the thermocouple, because the temperature difference between the outer and inner surfaces of cladding is evaluated at about 20°C during the steady film boiling while the temperature drop around the thermocouple reaches 300°C. Thus, the cladding temperature is a function of the distance from the thermocouple along the cladding surface.

(3) Vapor film on the cladding surface is a flat and stationary layer. A fixed film thickness \( a_g \) represents its average value neglecting the oscillation.

(4) Heat transfer coefficients at thermocouple surface in vapor film and liquid layer are taken to be uniform. Also, the bulk temperatures of the both phases are uniform.

(5) Gap heat transfer coefficient is invariable along the cladding surface and all the phenomena are treated as steady state.

In the analysis of the fin effect of a single wire, heat transfer calculation for the fin and that for the cladding were made separately. The part of fin is analyzed with the coordinate system shown in Fig. 2(a). The equations of heat balance for gas phase and liquid phase regions are described as,
in gas-phase region \((0 \leq z \leq a_g)\),
\[
\frac{d^2(T_i - T_g)}{dz^2} - \frac{4h_g}{k_i \cdot d}(T_i - T_g) = 0,
\]
and in the liquid region \((z \geq a_g)\),
\[
\frac{d^2(T_i - T_l)}{dz^2} - \frac{4h_l}{k_i \cdot d}(T_i - T_l) = 0.
\]

Boundary conditions are

1. \(z = 0, \quad T_i = T_o\)
2. \(z = a_g, \quad T_i|_{z=a_g} = T_i|_{z=a_g}^+\)
3. \(d(T_i/dz)|_{z=a_g} = (dT_i/dz)|_{z=a_g}^+\)
4. \(z \to \infty, \quad T_i \text{ is finite.}\)

The temperature \(T_o\) is to be determined with the combination of the equations for the cladding part.

For the part of cladding, the effect of fin was treated as a cylindrical heat sink of diameter \(d\) as shown in Fig. 2(b). Heat balance equation for an infinitesimal ring at radius \(r\) is
\[
\dot{q}_{gap} - \dot{q}_{film} - \frac{\delta}{r} \frac{d}{dr}(r \dot{q}_r) = 0.
\]

The following simple equations are used to describe heat fluxes \(\dot{q}_{gap}\) from fuel pellets, \(\dot{q}_{film}\) into the ambient fluid by film boiling and \(\dot{q}_r\) by heat conduction along the cladding:
\[
\dot{q}_{gap} = h_{gap}(T_p - T_c),
\]
\[
\dot{q}_{film} = h_{film}(T_c - T_{sat}),
\]
\[
\dot{q}_r = -k_c \frac{dT_c}{dr}.
\]

Equations (3)~(6) give the following Bessel differential equation of 0th-order.
\[
r^2 \frac{d^4T_c}{dr^4} + r \frac{d^3T_c}{dr^3} \left( \frac{h_{film} + h_{gap}}{k_i \cdot \delta} r^2(T_c - T^*) \right) = 0,
\]
where \(T^* = \frac{h_{film} T_{sat} + h_{gap} T_p}{h_{film} + h_{gap}}\).

Equations (1), (2) and (7) can be described as follows with dimensionless parameters by introducing \(Z = z/(d/2)\) and \(R = r/\delta\):
\[
\frac{d^2(T_i - T_g)}{dZ^2} - \lambda_i^o(T_i - T_g) = 0, \quad (0 \leq Z \leq A_g = a_g/(d/2)),
\]
\[
\frac{d^2(T_i - T_l)}{dZ^2} - (\gamma \lambda_i^o) T_i - T_l = 0, \quad (Z \geq A_g),
\]
\[
R^2 \frac{d^2(T_c - T^*)}{dR^2} + R \frac{d(T_c - T^*)}{dR} - \lambda_i^2 R^2(T_c - T^*) = 0,
\]
where \(\lambda_i^o\) and \(\lambda_i^2\) are the Biot numbers defined as \(\lambda_i^o = h_g \cdot d/k_i\) and \(\lambda_i^2 = (h_{film} + h_{gap}) \cdot \delta/k_c\), respectively, and \(\gamma^\circ\) is the ratio of \(h_i\) to \(h_g\). The solutions for Eqs. (8) and (9) are
\[
T_i - T_g = A_g e^{\lambda_i^2 Z} + A_g e^{-\lambda_i^2 Z}, \quad (0 \leq Z \leq A_g),
\]
\[
T_i - T_l = A_l e^{-\lambda_i^2 Z}, \quad (Z \geq A_g),
\]
\[
T_i - T_c = A_c e^{\lambda_i^2 Z} + A_c e^{-\lambda_i^2 Z}, \quad (0 \leq Z \leq A_g),
\]
\[
T_i - T_l = A_l e^{-\lambda_i^2 Z}, \quad (Z \geq A_g),
\]
where \( A_1 = \frac{\gamma (T_l - T_g) + (1 - \gamma) (T_0 - T_g) e^{-\lambda t A_g}}{2 \cosh \lambda t A_g + \gamma \sinh \lambda t A_g} \)

\( A_2 = -\frac{\gamma (T_l - T_g) + (1 + \gamma) (T_0 - T_g) e^{\lambda t A_g}}{2 \cosh \lambda t A_g + \gamma \sinh \lambda t A_g} \)

\( A_3 = \frac{(T_0 - T_g) - (T_l - T_g) \cosh \lambda t A_g}{\cosh \lambda t A_g + \gamma \sinh \lambda t A_g} e^{\lambda t A_g} \).

Equation (10) has the solution of

\[ T_0 - T^* = \frac{-\hat{Q}_s}{\frac{\pi d k_c \lambda_c}{K_0(\lambda_c R)}} \frac{K_0(\lambda_c R)}{K_0(\lambda_c D/2)}, \quad (D \equiv d/\delta), \]

where \( K_0 \) and \( K_1 \) are the modified Bessel function of the second kind of order zero and that of order one, and \( \hat{Q}_s \) is the heat flow rate at \( R = D/2 \) into the heat sink by thermal conduction. The temperature distribution along the thermocouple wire is determined from Eqs. (11) and (12), and the temperature distribution around the heat sink from Eq. (13).

The heat balance at the thermocouple base is considered to combine the two heat transfer calculations, i.e. the calculation for the thermocouple and that for the cladding. The heat balance equation at the thermocouple base in Fig. 2(b) is described as

\[ Q = -\hat{Q}_s + \hat{Q}_{gap,s}. \]

The first term of right-hand side of Eq. (14) can be written from Eq. (13) as

\[ -\hat{Q}_s = \frac{\pi d k_c \lambda_c}{K_0(\lambda_c D/2)} \frac{K_0(\lambda_c R)}{K_0(\lambda_c D/2)} (T^* - T_0). \]

The second term, heat flow rate through the fuel-cladding gap is

\[ \hat{Q}_{gap,s} = \frac{\pi d^2}{4} h_{gap}(T_p - T_0). \]

Then, the ratio of these two terms becomes

\[ \frac{\hat{Q}_{gap,s}}{-\hat{Q}_s} = \frac{d h_{gap}}{4 k_c} \frac{K_0(\lambda_c D/2)}{\lambda_c K_0(\lambda_c D/2)} \frac{T_p - T_0}{T^* - T_0}. \]

The order of magnitude of this ratio is 100th because the orders of following terms are

\[ \frac{d h_{gap}}{4 k_c} \sim 10^{-5}, \quad \frac{K_0(\lambda_c D/2)}{\lambda_c K_0(\lambda_c D/2)} \sim 1, \quad \frac{T_p - T_0}{T^* - T_0} \sim 1. \]

Then, \( \hat{Q}_{gap,s} \) can be neglected in comparison with \( -\hat{Q}_s \) in Eq. (14). The right-hand side of Eq. (14), heat transferred to the thermocouple wire \( \hat{Q} \), is given from Eq. (11) as

\[ \hat{Q} = -\frac{\pi d^2}{4} \frac{k_c}{k_f} \frac{dT}{dZ} \bigg|_{\lambda = 0} = \frac{\pi d^2}{4} k_c \lambda_c \xi (T_0 - T_g + \xi (T_g - T_l)) \]

\[ \xi = \frac{\sinh \lambda_c A_g + \gamma \cosh \lambda_c A_g}{\cosh \lambda_c A_g + \gamma \sinh \lambda_c A_g} \]

\[ \zeta = \frac{\gamma}{\sinh \lambda_c A_g + \gamma \cosh \lambda_c A_g} \]

The parameters \( \xi \) and \( \zeta \) represent the effect of vapor film thickness. When the ambient coolant is single phase of gas \((A_g \rightarrow \infty)\), the values of \( \xi \) and \( \zeta \) become unity and zero, respectively. On the other hand, in the case of liquid phase the values of \( \xi \) and \( \zeta \) are equal to \( \gamma \) and unity. By equating \( \hat{Q} \) and \( -\hat{Q}_s \), fundamental equation to evaluate the temperature drop caused by the thermocouple fin effect is obtained as follows:
The equation has six parameters; \( \lambda_c, \lambda_t, \gamma, T_t, T_g \) and \( A_g \). Of these parameters, \( \lambda_c, \lambda_t \) and \( \gamma \) depend upon the surface heat transfer coefficient and also upon the physical properties of the cladding and the thermocouple.

2. Application to NSRR Experiments

In the NSRR experiment PWR type fuel rods were used as test fuel rods and they were instrumented with some pairs of thermocouples for the measurement of cladding surface temperature. The major characteristics of the test rods and the thermocouples are listed in Table 1. The test rods have 10% enriched \( \text{UO}_2 \) pellets of nominal 9.29 mm in diameter, contained in a Zircaloy-4 cladding of 0.62 mm thickness. The thermocouple consists of a pure-Pt wire and a Pt-Rh (13%) one.

In the present analysis, heat transfer coefficient at thermocouple surface is estimated from heat transfer coefficients for film boiling in the vapor film and for forced convection heat transfer around a cylinder in the liquid. The bulk temperatures of gas and liquid phases were assumed to be the saturation temperature of the coolant and the subcooled bulk coolant temperature, respectively. The vapor film thickness was changed as a parameter. For these conditions, Eq. (19) can be written as

\[
\frac{T^* - T_o}{T^* - T_{sat}} = \frac{1}{1 + \frac{2}{\xi} \frac{k_c}{k_t} \frac{\lambda_c}{\lambda_t} \frac{K_s(\lambda_c D/2)}{K_g(\lambda_t D/2)}} \left\{1 + \frac{\Delta T_{sub}}{T^* - T_{sat}}\right\}. \tag{20}
\]

Equations (13) and (16) give the equation

\[
\frac{T^* - T_e}{T^* - T_o} = \frac{K_g(\lambda_c R)}{K_s(\lambda_c D/2)}. \tag{21}
\]

Substituting Eq. (20) into Eq. (21), the equation for the cladding temperature distribution is obtained,

\[
T^* - T_e = \frac{K_g(\lambda_c R)}{K_s(\lambda_c D/2)} \frac{(T^* - T_{sat} + \xi \Delta T_{sub})}{1 + \frac{2}{\xi} \frac{k_c}{k_t} \frac{\lambda_c}{\lambda_t} \frac{K_s(\lambda_c D/2)}{K_g(\lambda_t D/2)}}. \tag{22}
\]

These equations are for the case of a single wire. In NSRR experiments, however, since two wires are attached on the cladding surface at an interval of about 1 mm, the cladding temperature around each wire may be affected by each other. This effect was approximated by superposing the temperature drops obtained for a single wire. This approximation will not result in a large error since the temperature drop by the thermocouple fin
effect is not very sensitive to the cladding temperature as to be shown in the next section. The temperature obtained by thermocouple $T_{tc}$ was assumed to be the arithmetic mean of the temperatures at the bases of two wires. An example of this procedure is shown in Fig. 3. The cladding temperature without the fin effect is 1,200°C in this case, and the subcooling of the stagnant cooling water is 75°C. The diameter of the wire is 0.3 mm. A value of 1.0 mm is assumed for vapor film thickness. The other parameters in the calculation are listed in Table 2. Some of these values assumed in the calculation are based on the results from transient heat transfer analysis by utilizing the computer code NSR-77.

Figure 3 shows an example of the calculation. Two dotted lines are the results for the individual wires by Eq. (22). The solid line is the superposition of the two dotted lines.

### Table 2 Typical data for calculation of thermocouple fin effect

<table>
<thead>
<tr>
<th>Thermocouple diameter</th>
<th>$d$</th>
<th>0.30 mm</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cladding thickness</td>
<td>$\delta$</td>
<td>0.62 mm</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Thermal conductivity of clad $k_1$</th>
<th>$3.13 \times 10^{-2}$ kWs/m-K</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pt $k_2$</td>
<td>$8.96 \times 10^{-2}$ kWs/m-K</td>
</tr>
<tr>
<td>Pt·13%Rh $k_{2e}$</td>
<td>$6.0 \times 10^{-2}$ kWs/m-K</td>
</tr>
<tr>
<td>Gap gas temperature $T_p$</td>
<td>1,440°C</td>
</tr>
<tr>
<td>Saturation temperature $T_{sat}$</td>
<td>100°C</td>
</tr>
<tr>
<td>Water temperature $T_w$</td>
<td>25°C</td>
</tr>
<tr>
<td>Film boiling heat transfer coefficient $h_{film}$</td>
<td>1.18 kW/m²-K</td>
</tr>
<tr>
<td>Gap heat transfer coefficient $h_{gap}$</td>
<td>5.43 kW/m²-K</td>
</tr>
<tr>
<td>Heat transfer coefficient in liquid phase $h_l$</td>
<td>10.63 kW/m²-K</td>
</tr>
</tbody>
</table>

Data from MATPRO(4)
Data from TPRC Data Series(7)
Data from NSR-77(4)

By performing this superposition, the temperature difference between thermocouple reading $T_{tc}$ and the temperature at unaffected area $T^*$ can be given by the equation

$$
T^* - T_{tc} = \frac{1}{2} \left\{ 1 + \frac{K_1(\lambda D)}{K_0(\lambda D/2)} \right\} \left\{ \frac{(T^* - T_{sat}) + \zeta_1 \Delta T_{sub}}{1 + \chi_1} + \frac{(T^* - T_{sat}) + \zeta_2 \Delta T_{sub}}{1 + \chi_2} \right\}
$$

where the subscripts 1 and 2 of $\chi$ and $\zeta$ mean each thermocouple wire.

### 3. Effects of Thermocouple Diameter and Cladding

**Temperature on Temperature Drop**

The effects of thermocouple diameter and cladding temperature on the temperature drop due to fin effect of thermocouple are studied based on Eq. (23) under free and forced convection conditions. The vapor film thickness is varied as a parameter to evaluate its effect.

Figure 4 shows the effects of thermocouple diameter and vapor film thickness on the temperature drop under a stagnant condition. The temperature drop increases with the thermocouple diameter nearly in proportion to it. The thinner film thickness produces the larger drop for the same thermocouple diameter. The effect of the vapor film thickness becomes significant when the thickness is less than around 1 mm. These results indicate the major characteristics of the fin effect under the present situation. The increase of thermocouple diameter gives larger cross section for heat conduction, larger surface area...
for heat transfer to the ambient fluid and larger area for heat sink on the cladding. All of these effects enhance the temperature drop. Above finding that the increase of the temperature drop is in proportion to the thermocouple diameter suggests that the peripheral effects such as the increase of heat transfer area and the peripheral length of heat sink are dominant under the present condition. Regarding the effect of film thickness, the calculational result indicates that the effect increases with the decrease of the distance to the liquid coolant from the thermocouple base, since the heat transfer to the liquid is the major portion of the heat flow from the wire. The effect increases nearly exponentially with the film thickness because the temperature distribution along the thermocouple is parabolic.

The influence of cladding temperature on the thermocouple fin effect is shown in Fig. 5. This is for the case of the thermocouple of 0.3 mm diameter under a stagnant condition. The temperature drop increases with the cladding temperature, but the degree of increase is not so conspicuous when the cladding temperature exceeds 1,000°C. This tendency is caused by the fact that we adopt the film boiling heat transfer coefficient at thermocouple surface in vapor film, which is not so sensitive to the cladding temperature above 1,000°C, and that the surface heat transfer coefficient of thermocouple in liquid layer is independent of the cladding temperature in our calculation.

Figures 6(a) and (b) show the results for the forced convection cooling at the velocity of 1.8 m/s. (Each figure corresponds to Figs. 4 and 5 for the stagnant condition.) The temperature drop is generally larger than that in the case of stagnant condition reflecting the larger heat transfer coefficient in the liquid phase. Both effects of thermocouple diameter and vapor film thickness are much enhanced than under stagnant conditions. These results suggests that the consideration of the thermocouple fin effect becomes more important in the forced convection experiments.

III. EXPERIMENT AND DISCUSSION

1. Experiment

An experiment was carried out to confirm the fin effect of thermocouples on the temperature measurement for different thermocouple diameters. The test fuel used in this experiment was a PWR type rod as illustrated in Fig. 7. The major characteristics of the rod are listed in Table 1. The cladding surface temperatures were measured at six points
as indicated in Fig. 7. At the axial midplane of the fuel pellet stack, 0.1 and 0.2 mm diameter Pt-Pt-Rh (13%) thermocouples were attached at opposite sides. Similarly, 0.5 and 0.1 mm diameter thermocouples and 0.3 and 0.1 mm ones were attached at the axial levels of 33 mm above and below the midplane, respectively.

![Figure 6](image1.png)  
**Fig. 6** Calculated temperature drops for forced convection condition (1.8 m/s)

![Figure 7](image2.png)  
**Fig. 7** Schematic drawing of test fuel rod and location of thermocouples

The test fuel rod was fixed vertically at the center of a test capsule as shown in Fig. 8. The test capsule was 120 mm in inner diameter and 800 mm in height and was filled with distilled water as the coolant. The thermal capacity of the coolant is so large as compared with the energy generation in the fuel rod that the temperature rise of bulk coolant is within 1°C. Simulating the RIA conditions, the test fuel rod was subjected to a rapid energy deposition in the NSRR test cavity, in pulse operation of the reactor.
In the present test for the thermocouple fin
effect verification, the energy of 203 cal/g \cdot \text{UO}_2 was
deposited in the test fuel rod. Transient histories
of cladding temperature were obtained with the
thermocouples, and the circumferential variation of
the maximum cladding temperature was evaluated
by the measurement of oxidation layer thickness
of cladding surface.

2. Transient Histories of
Cladding Temperature

Measured histories of cladding temperature at
three axial levels were compared in Fig. 9. These
thermocouple signals exhibit the effects of thermo-
couple diameter clearly except that by the failed
thermocouple of 0.3 mm diameter at lower level.
The temperature measured with the thermocouple
of 0.1 mm diameter was the highest at each meas-
urement level, and the temperature with the ther-
mocouple of 0.5 mm diameter was much lower than
that with the thermocouple of 0.2 mm diameter.
This tendency is just consistent with the analytical
result discussed in Chap. II that the temperature
drop increases with the increase of ther-
mocouple diameter. Further, the tempera-
ture difference of two thermocouples at the
same level increases with time, and becomes
maximum at the time of quenching. As it
is known by high speed cinematographic
study of the fuel behavior\(^2\) that the vapor
film thickness decreases with time and be-
comes very thin just before quenching. The
above observation indicates the effects of
the vapor film thickness on the temperature
drop. The increase of the temperature drop
with time also agrees with the analytical
results on the effect of film thickness quali-
tatively. But the experimental temperature
difference of each thermocouple seems to be
somewhat larger than that in the calculation.
The reason is that the measured data include
not only the fin effect but also the tempera-
ture variation caused by the un-uniformity
of fuel pellet-cladding contact which is
unavoidable in the in-pile experiment with
an actual LWR fuel rod. Consequently,
the quantitative comparison has to be done

\begin{figure}[h]
\centering
\includegraphics[width=\textwidth]{fig8}
\caption{Experimental capsule}
\end{figure}

\begin{figure}[h]
\centering
\includegraphics[width=\textwidth]{fig9}
\caption{Comparison of cladding temperature histo-
ries for different diameter of thermocouples}
\end{figure}
among the reduced data in which the additional effect of the pellet eccentricity is considered.

3. Maximum Cladding Temperature Estimation from Oxidation Layer Thickness

At elevated temperatures, cladding is oxidized by Zircaloy-water reaction and oxidation layer is produced at the cladding surface. Shiozawa et al.\(^{(1)}\) proposed a correlation between the oxidation layer thickness and the maximum cladding temperature for the NSRR experiment conditions. By making use of this correlation, the circumferential distributions of the maximum cladding temperature were estimated from the oxidation layer thickness measurement.

The post-test fuel rod was cut at three axial levels where the thermocouples were attached. **Photo. 1** Macrophotograph of transverse section of the test fuel rod, showing fuel-cladding eccentricity and thermocouples location.

**Fig. 10** Azimuthal variation in gap width, oxidation layer thickness and evaluated maximum cladding temperature of fuel rod subjected to 203 cal/g·UO\(_2\)
measured by microscopic observation. The scale in the right-hand side in this figure shows the maximum temperature attained corresponding to the oxidation layer thickness. Sharp dips of oxidation layer thickness are observed at each thermocouple location. In this figure, the maximum temperatures from the thermocouple readings in Fig. 9 are symbolized by the black dots, and these values are well consistent with the estimated temperatures at thermocouple junctions from the oxidation layer thickness. Then, we can use the temperature estimated from the oxidation layer instead of the thermocouple reading. It can be understood from Fig. 10 that the temperature distribution is composed of two types of variations, i.e. sharp dips at thermocouple locations and broader variation. Since the latter is consistent with the variation of gap width, it should be due to the eccentricity of the fuel pellet. The former should be due to the thermocouple fin effect, but it was influenced by the pellet eccentricity. To subtract the effect of pellet eccentricity for the quantitative verification of the analysis for the fin effect, the temperature variation due to pellet eccentricity was estimated as the dotted line in Fig. 10 by considering the variation of gap width. That is, the temperature drop by the fin effect was evaluated to be the deviation from the dotted line.

Figure 11 shows the comparison of the temperature drops estimated from the oxidation layer thickness as explained above with the calculation. The experimental data are well consistent with the analytical results at around 1~2 mm thick vapor film. The comparison in the case of forced convection is shown in Fig. 12. The experimental data in this figure are obtained from the metallurgical examination of fuel rods irradiated under the condition of forced convection at the coolant velocity of 1.8 m/s. We can see most of data lying in the range of 0.6~1.0 mm of vapor film thickness in the calculated results.

**Fig. 11** Comparison of experimental data with calculated temperature drop at thermocouple

**Fig. 12** Comparison of experimental data from forced convection tests with calculated temperature drop at thermocouple

It is necessary to examine the vapor film thickness in order to complete the quantitative comparison of analytical results with the experiments. Transient fuel behavior and vapor
film motion were filmed with a high speed camera at the rate of 250 frames per second. The thickness of vapor film layer were measured at a fixed axial level of the fuel rod. Figure 13 presents the transient histories of vapor film thickness and cladding surface temperature from the visualization experiments performed in NSRR under the stagnant water and the forced convection condition. Energy depositions of each experiment were about 300 cal/g·UO₂ in a stagnant water and about 250 cal/g·UO₂ in forced flow cooling at 1.8 m/s. The thick vapor film is generated intermittently. In our model, the thickness of film covering the thermocouple constantly was considered as the average thickness of vapor film. It can be concluded from this viewpoint that the thicknesses of vapor film at the maximum temperature are around 1~2 mm in the stagnant condition and around 0.8~1.5 mm in the forced cooling condition. These values are in good agreement with the results of above comparison between the analysis and metallurgical estimation.

IV. CONCLUSIONS

An analysis model has been developed for estimating the temperature drop due to heat conduction along the thermocouple wire in the measurement of cladding surface temperature under film boiling condition. The model includes the effects of vapor film thickness, thermocouple diameter, cladding temperature and coolant flow conditions. By comparing the analysis with the NSRR experiments, the following conclusions have been derived:

1. The good agreement of the calculation with the NSRR experiment indicated that the proposed analytical model can be one of the effective approaches in the evaluation of thermocouple fin effects under film boiling condition.

2. The temperature drop increases with the decrease of the vapor film thickness. Consequently, the fin effect becomes significant when the vapor film is very thin as in the case of forced flow cooling and near the quench point.

3. The temperature drop increases with the increase of the cladding temperature. However, this tendency does not become so conspicuous when the cladding temperature exceeds 1,000°C in a stagnant condition.

4. The temperature drop is influenced by the thermocouple diameter, too. The larger diameter of thermocouple gives the larger temperature drop. The temperature drops due to the attachment of thermocouples of 0.2 and 0.3 mm diameter were about 120 and 150°C, respectively, at the cladding temperature above 1,000°C in a stagnant condition.
[NOMENCLATURE]

\( \bar{a} \): Average vapor film thickness (mm or m)
\( d \): Thermocouple diameter (mm or m)
\( h_{\text{film}} \): Film boiling heat transfer coefficient (kW/m²·K)
\( h_{\text{gap}} \): Gap heat transfer coefficient (kW/m²·K)
\( h_{\text{f}} \): Surface heat transfer coefficient of fin in gas phase (kW/m²·K)
\( h_{\text{l}} \): Surface heat transfer coefficient of fin in liquid phase (kW/m²·K)
\( k_{\text{c}} \): Thermal conductivity of cladding (kW/m·K)
\( k_{\text{t}} \): Thermal conductivity of thermocouple (kW/m·K)
\( K_{0} \): Zeroth-order modified Bessel function of second kind
\( K_{1} \): First-order modified Bessel function of second kind
\( l \): Distance between each base of thermocouple wires (mm or m)
\( \dot{q}_{\text{film}} \): Heat flux at outer surface of cladding (kW/m²)
\( \dot{q}_{\text{gap}} \): Heat flux at inner surface of cladding (kW/m²)
\( \dot{q}_{\text{r}} \): Heat flux by heat conduction along cladding (kW/m²)
\( \dot{Q} \): Heat flow rate into thermocouple wire (kW)
\( \dot{Q}_{s} \): Heat flow rate into heat sink along cladding (kW)
\( \dot{Q}_{\text{gap,s}} \): Heat flow rate into heat sink from fuel-cladding gap (kW)
\( T_{\text{c}} \): Temperature of cladding (°C)
\( T_{\text{g}} \): Temperature of gas phase (°C)
\( T_{\text{l}} \): Temperature of liquid phase (°C)
\( T_{\text{p}} \): Cladding temperature at a junction of thermocouple wire (°C)
\( T_{\text{sat}} \): Saturation temperature (°C)
\( T_{\text{tp}} \): Temperature of fuel pellet (°C)
\( T_{\text{t}} \): Temperature of thermocouple wire (°C)
\( \Delta T \): Coolant subcooling (°C)

\( \delta \): Cladding thickness (mm or m)
\( \lambda_{b} \): Biot number defined as \( \lambda_{b} = \left( h_{\text{film}} + h_{\text{gap}} \right) / k_{\text{c}} \)
\( \lambda_{1} \): Biot number defined as \( \lambda_{1} = h_{\text{f}}d / h_{\text{f}} \)
\( \gamma \): Ratio of \( h_{\text{f}} \) to \( h_{\text{l}} \) (i.e. \( \gamma = h_{\text{f}} / h_{\text{l}} \))
\( \xi \): Parameter defined by Eq. (18)
\( \zeta \): Parameter defined by Eq. (18)
\( \chi \): Parameter defined by Eq. (23)

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