Steam Water Pressure Drop under 15 MPa*

Wei LIU**, Tamai HIDESADA**, Kazuyuki TAKASE**, Hiroki HAYAFUNE**, Satoshi FUTAGAMI** and Naoyuki KISOHARA**
**Japan Atomic Energy Agency (JAEA),
Tokai, Ibaraki, Japan, 319-1195
E-mail: liu.wei@jaea.go.jp

Abstract

For a steam generator with straight double-walled heat transfer tubes that will be used in a sodium cooled faster breeder reactor, clarification of flow instability in heat transfer tubes is one of the most important research themes. As the first step of the research, thermal hydraulics experiments with water were performed under high pressure condition in JAEA with using a circular tube. Pressure drop, heat transfer coefficients and void fraction data were derived. This paper summarizes the pressure drop characteristics under 15MPa. Several two-phase flow multipliers were checked and then, it was found that both Chisholm two-phase flow multiplier and homogeneous model can predict the present experimental data in high accuracy. A sudden decrease of the pressure drop was observed when flow pattern shifts from bubbly and churn flows to annular flow. The reason for this decrease is tried to be interpreted.

Key words: Steam-Water Pressure Drop, High Pressure

1. Introduction

R&D activities have been starting at Japan Atomic Energy Agency (JAEA) for the commercialization of the next generation Faster Breeder Reactor (FBR) cycle. From the viewpoints of lowering breakage rate and easy maintenance, the FBR system adopts a Steam Generator (SG) with straight double-walled heat transfer tube. To reduce the cost, the SG is designed being big size. Figure 1 gives its general image. It is 38 m in height and 3.2 m in diameter. With using 7100 straight double-walled heat transfer tubes and is expected to have a total heat exchange of 1765 MW. In the SG, high temperature sodium flows the outside of the heat transfer tube from the upper to the lower part of the vessel. On the other hand, water flows the inside of the heat transfer tube from the bottom to the top. Water at the inlet is subcooling water. With the heat transferred from Na side, the water turns to water-vapor two-phase flow and becomes superheated vapor at the outlet. The effective heating length is 29 m.

In this SG, because a design strategy that no orifice is admitted at the inlet of each heat transfer tube is adopted, flow instability inside the heat transfer tube is of the most concern. Occurrence of flow instability in the parallel heat transfer tubes was expected to be suppressed by raising operation pressure. At present, the operation pressure is designed being 19.2 MPa. However, from the viewpoint of reduction of the operating cost, excessive pressure margin should be avoided. Therefore, it is important to clarify the flow instability with high accuracy. In order to achieve that, the verification and improvement of the existing thermal-hydraulic correlations for the SG thermal design, especially the pressure drop correlation, is very important.

As the thermal-hydraulic data for the SG thermal design under the high pressure condition over 15MPa is not enough, JAEA started the thermal-hydraulic tests under the
high pressure condition using a circular tube which simulates the SG heat transfer tube. In year 2008, we measured pressure drop in the simulated heat transfer tube under the condition of 18 MPa\(^2\) and it was reported that Chisholm two-phase multiplier will show the best prediction for the pressure drop at 18MPa. When we consider the start up and shut down processes in the operation of the SG, it is also necessary to predict the pressure drop under the pressure lower than the operation condition. In this paper, we focus on the research to the pressure drop under 15MPa. The objective of this study is to confirm whether the Chisholm two-phase multiplier, which gives the best prediction at the pressure condition of 18MPa, is still applicable under the condition of 15 MPa.

![Image of the SG with straight double-walled heat transfer tube for FBR](image)

2. Experiment

Images of test loop and test section are shown in Fig.2. In the loop, purified water (deionized and degassed water) was used as the working fluid. The water flows firstly through a preheater to the test section. In the test section, the water is heated up and becomes the water-vapor two-phase flow. After the water and vapor are separated by a separator, the water returns back to the loop through a cooler.

The test section is a circular tube with an inner diameter of 11.6 mm. The size simulates an inner diameter (12 mm) of the prototype SG heat transfer tube. Whole height of the test section is about 13.4 m, which simulates the middle 13 m's high two-phase flow region of the prototype SG heat transfer tube. The test section is axially divided into three levels. The three levels were heated separately with three radiation heaters, to assure the safety of the test section even in post Boiling Transition (BT) region. The heating lengths in the three levels are 3.3m, 3.6m and 3.3m, respectively.

Main measurement parameters are the flow rate \(\dot{w}\), exit pressure \(P_{ex}\), inlet temperature \(T_{in}\), power \(Q\) and pressure drop \(\Delta P\). In this research, the flow rate is measured with a venturi tube. The mass flow and power under steady state condition were calibrated before each experiment. The power was corrected using the results of the single phase flow experiment.
The error for power is within ±3% and the error for mass flow is within ±2%. The inlet water temperature is measured by high accuracy platinum resistance thermometers and then automatically controlled by the pre-heater within an error of ±0.5 K. The exit pressure of the test section is automatically controlled by a pressurizer and is measured by a high accuracy pressure transducer. For a steady exit pressure condition, the error is within ±0.3%.

The pressure drops were measured using differential pressure gauges. The measuring locations are shown in Fig. 2. Three differential pressure gauges (ΔP₁ - ΔP₃) measured partial pressure drops along three levels. Moreover, the pressure gauge (ΔPₜ) measured the total pressure drop in the whole test section. The error of each pressure gauge is within ±0.3%. The difference between the sum of the partial pressure drops and the total pressure drop in a two-phase flow condition is shown in Fig. 3 and it was expressed as:

$$\varepsilon = \frac{(\Delta P_1 + \Delta P_2 + \Delta P_3)}{\Delta P_T}$$  \hspace{1cm} (1)

Here, the average of ε is 1.001 and the standard deviation is 0.13%. The result indicates that the measurement system has high accuracy.

Before the experiments, dissolved gas was purged from the loop by degassing operation. Then, exit pressure, flow rate, inlet water temperature, and powers of three levels were controlled to be the values of the experimental conditions. After a stable condition was maintained for awhile, the pressure drop measurement was started. The experiment conditions are summarized in Table 1. In 2009, the tests were conducted at 15 MPa with the mass flow from 40 to 200 g/s, which covered the nominate flow rate (110 g/s) of the designed SG. The pressure drop data under both conditions of single and two-phase flow were derived. For the two-phase flow condition, data were taken under two kinds of heating condition. One is the condition that the all three levels give power outputs and the water is heated continuously in the whole test section. The other is the condition that the lower and middle levels give power outputs but the upper level is set to adiabatic. The pressure drop through the upper level is taken at a steady quality. The data are convenient in evaluating
two-phase multiplier correlations.

![Graph showing pressure drop data](image)

### 3. Analytical Method

Pressure drop, as shown in Eq.(2), is calculated as a sum of friction loss $\Delta P_F$, static head $\Delta P_H$ and acceleration loss $\Delta P_A$. The friction loss $\Delta P_F$ in two-phase flow condition is evaluated by a product of friction loss in single-phase flow $\Delta P_{FSP}$, and two-phase multiplier $\Phi_F^2$. In this paper, $\Delta P_{FSP}$ was calculated by Pfann’s correlation\(^3\).

\[
\Delta P = \Delta P_F + \Delta P_H + \Delta P_A
\]

\[
\Delta P_F = \Delta P_{FSP} \times \Phi_F^2
\]

The static head $\Delta P_H$ and the acceleration loss $\Delta P_A$ are calculated in Eqs. (4) and (5), respectively. They are basically the function of void fraction. The void fraction was calculated from drift flux model implemented in TRAC - BF1 code\(^6\). The model has been confirmed being of sufficient accuracy for prediction of void fraction under high pressure condition. The drift flux model is show in Eqs. (6) - (6p).

\[
\Delta P_H = \{\alpha \rho_g + (1 - \alpha) \rho_f \} g \Delta z
\]

\[
\Delta P_A = G^2 \left\{ \frac{\chi^2}{\alpha \rho_g} + \frac{(1 - \chi)^2}{(1 - \alpha) \rho_f} \right\}_{z+\Delta z} - \left[ \frac{\chi^2}{\alpha \rho_g} + \frac{(1 - \chi)^2}{(1 - \alpha) \rho_f} \right]_z
\]

\[
\alpha = \frac{\beta}{(c_0 + V_{gl}/j)}
\]

\[
\beta = \frac{\chi}{(\chi + \frac{(1 - \chi) \rho_g}{\rho_f})}
\]
\[ j = \frac{G\chi + G(1 - \chi)}{\rho_g + \rho_f} \]  

(6b)

In TRAC - BF1, \( C_\alpha \) is calculated as:

\[ C_0 = C_\infty - (C_\infty - 1)/\gamma \]  

(6c)

Where \( C_\infty = 1.0 + 0.2\left[ \frac{\rho_f}{G} \sqrt{G D h} \right]^{0.5}. \quad \gamma = \sqrt{\rho_f/\rho_g} \)

\( V_{gl} \) for bubbly flow region,

\[ V_{gl} = 1.53 \left[ \frac{\Delta \rho g}{\rho_f^2} \right]^{0.25} \]  

(6e)

\( V_{gl} \) for annular flow,

\[ V_{gl} = V_{ro} (1 - \alpha), \]  

(6f)

and \( V_{ro} = (1 - E\chi_2) V_{rba} + E\chi_2 V_{rd} \)

For drop flow, \( V_{gl} \) is calculated as:

\[ V_{gl} = V_{ro} (1 - \alpha), \]  

(6g)

with \( V_{ro} = V_{rd} \)

\[ V_{rba} = (1 - \chi_2) V_{rb} + \chi_2 V_{ra} \]  

(6h)

\[ V_{rb} = 1.53 \left[ \frac{\Delta \rho g}{\rho_f^2} \right]^{0.25} / (1 - \alpha) V_{ra} = \frac{D_{(r, min)}^2 g \Delta \rho}{3 \mu_l} \]  

(6i)

\[ V_{rd} = 0.5 R_d \alpha^{1.5} \left[ \frac{(\Delta \rho g)^{2}}{\rho_g \mu_g} \right]^{1/3} \]  

when \( R_d \leq R_{dc} \)  

(6j)

\[ V_{rd} = \sqrt{2} R_d \alpha^{1.5} \left[ \frac{(\Delta \rho g)^{2}}{\rho_g^2} \right]^{1/4} \alpha^{1.5} \]  

when \( R_d > R_{dc} \)  

(6k)

\[ E\chi_2 = \chi_2 ENT \]  

(6l)

\[ \chi_2 = (\alpha - \alpha_{tran}) / 0.25 \]  

(6m)

If \( X_E \leq 0.03 \), \( ENT = 0.0 \)  

(6n)

Else \( ENT = E + (1 - E)(1 - W_{wet}) \)  

(6o)

\[ E = (X_E - 0.03)[1 + (X_E + 0.1)^2]^{-0.5} \]  

(6p)

### 4. Results and Discussion

#### 4.1 Pressure Drop in Single Phase Flow

Pressure drop data in single phase flow was compared with that calculated values by Pfann’s correlation\(^3\), which is listed in Eqs. (7a - 7c). In the equations, \( e \) is relative roughness of surface and is defined in Eq.(7e). \( \delta \) in eq. (7e) is ten point roughness of the surface. \( D \) is the inner diameter of tube. Their definitions are schematically shown in Fig.4.

As the maximum Reynolds number at the present experimental condition is about 400,000, the effect of surface roughness on the wall inside the circular tube is considered. The definition to the Reynolds number is shown in eq.(7f). The measured ten point average roughness \( \delta \) is 10 \( \mu \)m. The verification results are shown in Fig.5. The Pfann’s correlation can predict the present single phase flow data within an error of \( \pm 2\% \).
Fig. 4 $\delta$ and D used in the definition of the relative roughness $e$

Fig. 5 Pressure drop in water single phase flow

$\begin{align*}
f &= \left( \frac{0.28}{\log Re - 0.82} \right)^2 \quad for \quad 2300 < Re \leq \frac{60}{e^{1.111}} \quad (7a) \\
f &= \left( \frac{0.25}{(3.393 - 0.8065g)g - 2.477 - \log e} \right)^2 \quad for \quad \frac{60}{e^{1.111}} < Re < 424 \frac{0.87 - \log e}{e} \quad (7b) \\
f &= \left( \frac{0.25}{0.87 - \log e} \right)^2 \quad for \quad Re \geq 424 \frac{0.87 - \log e}{e} \quad (7c)
\end{align*}$

Where

$g = \log \left( \frac{Re}{0.87 - \log e} \right) \quad (7d)$
4.2 Verification to Two-Phase Multiplier correlations

Data used for the evaluation of two-phase multiplier correlations were that taken under the heating condition that the lower and middle levels give power outputs but the upper level is kept adiabatic. The pressure drop through the upper level is taken under a steady quality. Therefore, the acceleration loss $\Delta P_A$ is 0 through the upper level.

Four correlations and one model are selected for evaluation; Martenlli-Nelson\(^5\), Hancox-Nicoll\(^6\), Friedel\(^7\) and Chisholm\(^8\) correlations and homogeneous model. Martenlli-Nelson and Hancox-Nicoll correlations were selected because they were reported being good accuracy under BWR condition. Friedel correlation was selected because it was reported being one of the most accuracy correlations\(^9\). Chisholm correlation was selected because it was based on high pressure data up to 14 MPa. Besides the above correlations, homogeneous model was also selected for evaluation. The homogeneous model calculates the friction pressure drop with the correlation for the single phase flow (Pfann's correlation) with only considering the two-phase flow effect on thermo-physical properties.

In the evaluation of each correlation and model, the void fraction was calculated using the drift flux model programmed in the TRAC-BF1. Moreover, the Martenlli-Nelson two-phase multiplier was calculated by interpolation of table values listed in\(^9\).

The evaluation results are shown in Figs.6-10. Here, we pay more attention to the prediction ability for the nominal flow rate condition ($w=110$ g/s) and lower flow rate condition ($w<110$ g/s), which may be encountered in a start up process of the SG.

The Martenlli-Nelson and Friedel correlations over predicted in comparison with the experimental data, especially under high quality and high mass flow rate condition. As for the Hancox-Nicoll correlation, the prediction accuracy was improved comparing with the Martenlli-Nelson and Friedel correlations. However, a small over prediction is still observed at high quality and low mass flow rate condition. On the other hand, the homogeneous model and Chisholm correlation show a good prediction to the experimental data. When we consider the prediction accuracy as the ratio of the predicted value to the experimental value, each ratio of four correlations and one model is shown in Table 2. From Table 2, it is found quantitatively that the homogeneous model and Chisholm correlation give the best predictions under the present high pressure condition.

As for the experimental data at $w=110$ and 150 g/s in Figs.6-10, a tendency of the pressure drop decrease at the quality of around 0.25 can be observed. This decreasing point was compared with a generation position of the slug-annular transition. The values of the quality and the void fraction at the slug-annular transition are shown in Table 3, which are calculated from the Eq. (8)\(^10\). The decreasing point of the pressure drop coincides well with the generating position of the slug-annular transition. Therefore, it would be judged that the reason of the pressure drop decrease depends on the two-phase flow pattern transfer from bubbly to annular flow. In the adiabatic condition, the entrainment and deposition rates in the two-phase annular flow should be an equilibrium condition. Therefore, the interface between liquid film and vapor core is quite smooth. This smooth interface causes a decrease in the friction loss and then the pressure drop decrease is led.

$$j_g^* = 0.4 + 0.6j_f$$

\[(8)\]
where $j_f^* = j_f \rho_f^{0.5} \left[ gD(\rho_f - \rho_g) \right]^{0.5}$ \hspace{1cm} (8a)

$j_g^* = j_g \rho_g^{0.5} \left[ gD(\rho_f - \rho_g) \right]^{0.5}$ \hspace{1cm} (8b)

Table 2 Prediction accuracies for two-phase multiplier correlations

<table>
<thead>
<tr>
<th>Correlations</th>
<th>M-N correlation</th>
<th>Friedel correlation</th>
<th>Hancox &amp; Nicoll correlation</th>
<th>Homogeneous model</th>
<th>Chisholm correlation</th>
</tr>
</thead>
<tbody>
<tr>
<td>Average</td>
<td>1.22</td>
<td>1.18</td>
<td>1.10</td>
<td>1.02</td>
<td>1.025</td>
</tr>
<tr>
<td>Standard Deviation</td>
<td>16.7%</td>
<td>15.9%</td>
<td>12.0%</td>
<td>7.5%</td>
<td>7.0%</td>
</tr>
</tbody>
</table>

Fig. 6 Verification to M-N correlation

Fig. 7 Verification to Friedel correlation
Fig. 8  Verification to Hancox & Nicoll correlation

Fig. 9  Verification to homogeneous model

Fig. 10  Verification to Chisholm correlation
The pressure drop decrease was observed obviously at \( w=110 \) and 150 g/s. However, that is not obvious under \( w=70 \) and 40 g/s. As for this, the following reason can be considered; since the friction loss in total pressure drop under the low flow rate condition is small, a decrease in the friction loss is also small, and therefore, the decrease in the pressure loss is small and becomes unremarkable.

The pressure drop in annular flow is a sum of the friction loss, static head and acceleration loss, as shown in Eq.(9). Here, because the present experimental data were taken under the adiabatic condition, the term of the acceleration loss \( \left( \frac{dP}{dz} \right)_{\text{a}} \) is 0. Therefore, Eq.(9) can be rewritten by Eq.(10). For annular flow, the friction loss \( \left( \frac{dP}{dz} \right)_{F} \) equals that on liquid film \( \left( \frac{dP}{dz} \right)_{f} \), which gives eq.(10a).

\[
\frac{dP}{dz} = \left( \frac{dP}{dz} \right)_{F} + \left( \frac{dP}{dz} \right)_{\text{a}} = \left( \frac{dP}{dz} \right)_{F} + \left( \frac{dP}{dz} \right)_{z} \quad (9)
\]

\[
\left( \frac{dP}{dz} \right)_{F} = \left( \frac{dP}{dz} \right)_{f} = \frac{2f_f \rho_f f_f^2}{D(1-\alpha)^2} \quad (10a)
\]

\[
\left( \frac{dP}{dz} \right)_{z} = g(\rho_f (1-\alpha) + \rho_g \alpha) \quad (10b)
\]

Here, \( f_f \) is the volumetric flux of the liquid film and \( f_f \) is the friction factor of that. By Dividing each term in Eq. (10) with \( [g(\rho_f - \rho_g)] \), a non-dimensional pressure drop equation as shown in Eq.(10) is obtained.

\[
\frac{dP^*}{dz} = \frac{dP}{dz} / [g(\rho_f - \rho_g)] = \frac{2f_f f_f^*}{(1-\alpha)^2} + \frac{\rho_f}{\rho_f - \rho_g} - \alpha \quad (11)
\]

Where \( f_f^* \) is defined in Eq.(8a). \( f_f^* \) is a function of mass velocity, quality and entrainment rate, and can be also expressed as,

\[
f_f^* = f(G, (1-\chi),(1-\text{ENT})) \quad (11a)
\]

When the pressure is 15 MPa, \( \frac{\rho_f}{\rho_f - \rho_g} \approx 1.19 \) and \( 2f_f \approx 0.01 \). Therefore, the Eq. (11) can be simplified as Eq. (12).

\[
\frac{dP^*}{dz} \approx \frac{0.01f_f^*}{(1-\alpha)^2} + (1.19 - \alpha) \quad (12)
\]

Fig.11 shows the relationship between non-dimensional pressure drop through the upper test section and void fraction in the annular flow region. Here, the solid lines represent the calculation results obtained from Eq.(12) and the solid circles represent the
experimental data at w=110 g/s. For the experimental data, the increase of quality leads to
the decrease of the liquid film flow rate, and then leads to the decrease of $J_f^*$ and $\frac{dP^*}{dz}$.
On the other hand, the increase of quality leads to the increase of the void fraction. Each
calculation result at a constant $J_f^*$ slightly decreases with increasing the void fraction and
reaches a minimum value, and then increases significantly. This characteristic is quite
similar to the tendency of the pressure drop data at w=110 and 150 g/s in Figs.6-10. In
annular flow region, an effect of the increase of the quality that tends to decrease the
pressure drop is significant, and then the pressure drop decrease. After that, an effect of the
void fraction that tends to increase the pressure drop becomes more and more significant
with increasing the void fraction. This causes the minimum value of the pressure drop.

Fig. 11  Effect of volumetric flux of liquid film and void fraction on pressure drop in annular flow

5. Concluding Remarks

For the verification of the evaluation methods on the steam - water pressure drop in
target SG heat transfer tube, steam - water pressure drops in a circular tube with the similar
inner diameter as that in the currently designed SG were measured under the conditions of
15 MPa in exit pressure and 40 to 200 g/s in flow rate. Prediction performance of two-phase
multiplier was investigated with Marletti-Nelson correlation, Hancox-Nicoll correlation,
Friedel correlation, Chisholm correlation and homogeneous model. From a series of the
experimental results, the following conclusions were derived:
1) The Chisholm correlation and homogeneous model give the best predictions.
2) The homogeneous model is used only for the friction loss calculation.
3) The void fraction which is necessary for the static head calculation is obtained by the
drift flux model
4) Under adiabatic condition, the pressure drop decrease was confirmed when two-phase
flow pattern shifts from bubbly and churn flows to annular flow.

Nomenclature

$D$: inner diameter, [m]
ENT: entrainment rate, [-]
e: relative roughness of surface, [-]
\( f_f \): friction factor of liquid film, [-]

\( G \): mass velocity, [kg/(m²s)]

\( j_l \): volumetric flux of liquid phase [m³/s]

\( j_v \): volumetric flux of vapor phase [m³/s]

\( j_{l*} \): non-dimensional volumetric flux of liquid film

\( j_{v*} \): non-dimensional volumetric flux of vapor core

\( P_{ex} \): pressure at the exit of test section, [MPa]

\( q_w \): wall heat flux, [MW/m²]

\( Q \): power input to test section, [kW]

\( T_{in} \): inlet temperature, [°C]

\( T_w \): wall temperature, [°C]

\( w \): flow rate, [g/s]

\( \Delta P \): pressure drop, [kPa]

\( \delta \): ten point average roughness, [m]

\( \chi \): quality, [-]

\( \rho_l \): density of liquid phase, [kg/m³]

\( \rho_v \): density of vapor phase, [kg/m³]

\( \alpha \): void fraction, [-]

References


