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Friction Pressure Drop and Heat Transfer Coefficient of Two-Phase Flow in Helically Coiled Tube Once-Through Steam Generator for Integrated Type Marine Water Reactor

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TECHNICAL REPORT

Friction Pressure Drop and Heat Transfer Coefficient of Two-Phase Flow in Helically Coiled Tube Once-Through Steam Generator for Integrated Type Marine Water Reactor

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Two-phase friction pressure drop and heat transfer coefficients in a once-through steam generator with helically coiled tubes were investigated with the model test rig of an integrated type marine water reactor. As the dimensions of the heat transfer tubes and the thermal-fluid conditions are almost the same as those of real reactors, the data applicable directly to the real reactor design were obtained. As to the friction pressure drop, modified Kozeki's prediction which is based on the experimental data by Kozeki for coiled tubes, agreed the best with the experimental data. Modified Martinelli-Nelson's prediction which is based on Martinelli-Nelson's multiplier using Ito's equation for single-phase flow in coiled tube, agreed within 30%. The effect of coiled tube on the average heat transfer coefficients at boiling region were small, and the predictions for straight tube could also be applied to coiled tube. Schrock-Grossman's correlation agreed well with the experimental data at the pressures of lower than 3.5 MPa. It was suggested that dryout should be occurred at the quality of greater than 90% within the conditions of this report.

KEYWORDS: *friction pressure drop, heat transfer coefficient, two phase flow, coiled tube, once-through steam generator, integrated type marine water reactor, dryout, heat flux*

I. INTRODUCTION

An integrated type pressurized marine water reactor is expected to be one of the most promising marine reactor for future commercial use in Japan. The reactor has a helically coiled tube once-through steam generator inside the pressure vessel. A model test rig of an integrated type marine water reactor was constructed at the Ship Research Institute to investigate the thermo-hydrodynamic characteristics of the steam generator. We can obtain the test data at real reactor conditions with this model test rig, as the length and inside/outside diameters of heat transfer tubes, and the fluid conditions in the steam generator were fitted to those of the conceptual design reactor of the Shipbuilding Research Association in Japan⁽¹⁾. The steam generator has the following characters:

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- (1) It is once-through type, and the steam pressure is around 5 MPa, which is lower than those of sodium heated once-through steam generators.
- (2) It consists of helically coiled heat transfer tubes.
- (3) It is heated by pressurized high temperature water.
- (4) It has a long superheating region to keep the steam at superheated condition under the varieties of operational conditions. It needs the test data at wide ranges of parameters.

Investigations for such type steam generators have scarcely been done under the conditions with even one such character until now. As the works of NSR-4 committee at the Shipbuilding Research Association in Japan about 15 years ago, Kozeki *et al.*⁽²⁾ conducted the basic tests on heat transfer and pressure drop characteristics in helically coiled tube at steam pressures of 0.5~2.1 MPa heated by high temperature water. They derived the results that two-phase friction pressure drop data were larger than those predicted by Martinelli-Nelson's correlation for straight tube, and the differences were larger with smaller pressures and with larger flow rates. They also derived the result that the heat transfer coefficients along boiling region were about $2 \times 10^4 \text{ W/m}^2 \cdot \text{K}$. These results were explained qualitatively by secondary flow effects in helically coiled tubes.

Other researches concerning the two-phase thermo-hydraulics in coiled tubes are burn-out phenomena at high pressure (18 MPa) by Carver *et al.*⁽³⁾, heat transfer and friction pressure drop at atmospheric pressure by Owhadi-Bell⁽⁴⁾ and others⁽⁵⁾⁽⁶⁾. Single-phase flow in coiled tubes has well been investigated by Ito⁽⁷⁾ for friction pressure drop and Mori-Nakayama⁽⁸⁾ for heat transfer coefficients.

Present research is to obtain the thermo-hydraulic test data at real reactor conditions with a model test rig, and to discuss how to evaluate the friction pressure drop and heat transfer coefficients of two-phase flow in real steam generators.

II. EXPERIMENT

1. Test Rig and Test Conditions

Figure 1 shows the flowsheet of the model test rig of an integrated type marine water reactor. The pressure vessel is simulating a reactor pressure vessel, and has a 0.8 MW heater simulating a reactor core, a steam generator with helically coiled heat transfer tubes, and a cover gas portion filled with N_2 gas at the upperpart in the pressure vessel which is taking a role as a self-pressurization system. Figure 2 shows the configurations

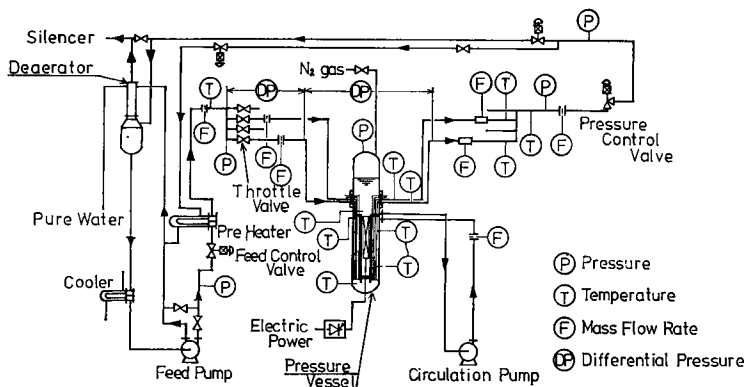


Fig. 1 Flow sheet of test rig

inside the pressure vessel. Primary water from circulation pump enters into pressure vessel, heats up the secondary water in steam generator, is heated at heater and returns to circulation pump. Secondary water is cooled in cooler, is pressurized by feed pump, and flows through feed water heater into heat transfer tubes at steam generator in pressure vessel to become superheated steam, and then goes out of pressure vessel. A part of the superheated steam is used to heat the feed water in feed water heater. Signs P, DP, F and T on the flowsheet in Fig. 1 indicate the measuring positions of pressures, pressure differences, flow rates and temperatures, respectively, and signs V and CV means the valves. **Figure 3** shows the dimensions of steam generator configurations in pressure vessel.

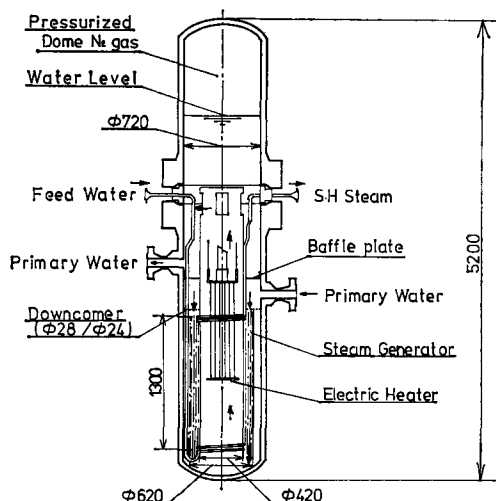


Fig. 2 Configurations inside pressure vessel

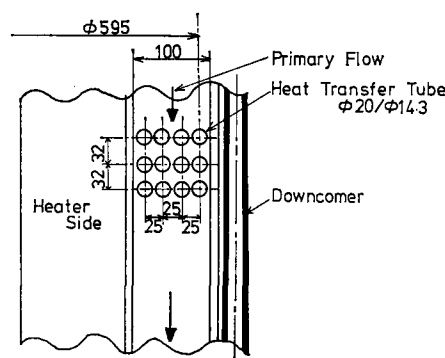


Fig. 3 Dimensions of steam generator configuration

The measurements were conducted with the fourth tube in the present experiment. The coil diameter and inside/outside diameters of the tube were measured to be 595 mm and 14.3/20.0 mm, respectively. The total length and the helical coil portion length of the heat transfer tube are about 61 m and 56 m, respectively. Among these dimensions, the length and inside/outside diameters of the tube are fitted to those of NSR-7 conceptually designed reactor. As the coil diameters of NSR-7 reactor are between 2,430 and 4,522 mm, which are larger than those of our test rig, present experimental results with smaller coil diameter may contain larger secondary flow effects than in the real reactor. Temperatures of secondary fluid were measured with sheathed thermocouples of 1.6ϕ at 15 points along heat transfer tube, from which we can obtain the heat transfer characteristics. Each thermocouple was penetrating the tube wall and hard soldered to it, to measure the temperature of the fluid at the center in the tube.

The 147 experiments were conducted with the experimental conditions shown in **Table 1**. Pressure drop and heat transfer characteristics of secondary flow were investigated under the conditions of three pressures (5, 3 and 2 MPa), and the wide ranges of mass flow rates ($150 \sim 850 \text{ kg/m}^2 \cdot \text{s}$).

2. Data Reduction of Pressure Drop and Heat Transfer Coefficients at Boiling Region

Figure 4 shows temperature distributions, pressure distribution and average heat flux distribution for subcooling (both downcomer and helical coil portions), boiling and superheating regions along tube length in the case of 397 kg/h secondary flow rate and 4.93

Table 1 Experimental conditions

(1) Primary water							
Pressure (MPa)	11						
Flow rate (t/h)	80						
Temperature (°C)	310	300	280	290	260	270	280
(2) Secondary fluid							
Feed water	194	114	114	114	44	114	114
subcooling (K)	184		44			44	
	114						
	44						
Steam exit pressure (MPa)	5		3		2		
Flow rate (kg/h)	100~500						

MPa exit steam pressure. Circles indicate measured values and crosses indicate the values estimated by the calculations. The lengths of subcooling, boiling and superheating regions were determined from the temperature distributions along heat transfer tube. The boundary between subcooling and boiling regions was determined as an intersecting point of two lines; one is a curve of secondary degree connecting three temperature records next to the boundary at subcooling region, and the other is a straight line connecting two temperature records next to the boundary at boiling region.

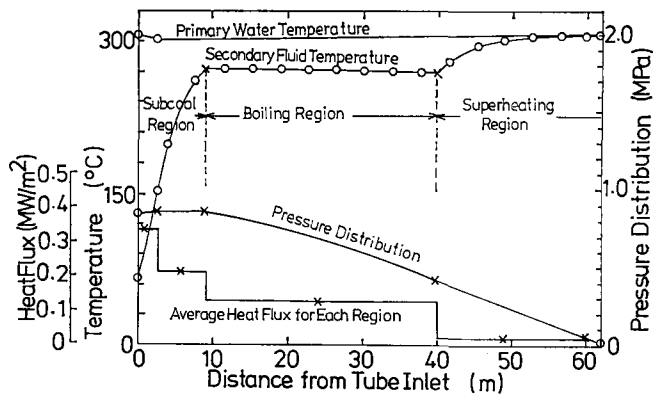


Fig. 4 Temperature, pressure and heat flux distributions along tube length in case of 397 kg/h secondary flow rate and 4.93 MPa exit steam pressure

The boundary between boiling and superheating regions was also determined in the same manner as that between subcooling and boiling regions, as an intersecting point of a straight line for boiling region and a curve of secondary degree for superheating region. The friction pressure drop ΔP_{TP} at boiling region were calculated from the equation

$$\Delta P_{TP} = \Delta P_T - \Delta P_{ac} - \Delta P_{head} - \Delta P_{Sub} - \Delta P_{SH}, \tag{1}$$

where ΔP_T is the pressure difference between inlet and outlet of heat transfer tube or total pressure drop. Friction pressure drop at subcooling region ΔP_{Sub} and that at superheating region ΔP_{SH} were calculated from the equation

$$\frac{\Delta P_{Sub}}{\Delta L} \text{ or } \frac{\Delta P_{SH}}{\Delta L} = f \frac{1}{d} \cdot \frac{\rho u^2}{2} = f \frac{G^2}{2d\rho}. \tag{2}$$

For straight tube, the coefficient of friction pressure drop f_s was determined by Colebrook's equation⁽⁹⁾

$$\frac{1}{\sqrt{f_s}} = 2 \log \left(\frac{\epsilon}{3.71d} + \frac{2.51}{Re\sqrt{f_s}} \right). \tag{3}$$

For coiled tube, the coefficient of friction pressure drop f_c was determined by Ito's equation for turbulent flow⁽⁷⁾:

$$f_c = f_s \left(Re \left(\frac{d}{D} \right)^2 \right)^{1/20} \quad (4)$$

In Eq. (3), surface roughness of heat transfer tube ε was determined as $\varepsilon=0.015$ mm ($\varepsilon/d=0.00105$) both by measurement of surface roughness and by preliminary experiment to compare the single-phase pressure drop data with Eqs. (2) and (3).

Average heat transfer coefficients at boiling region were determined as follows. The average heat flux q for subcooling, boiling and superheating regions were determined from the length of each region and the enthalpy increase in secondary fluid. Then, the average overall coefficients of heat transfer K defined by logarithmic-mean overall temperature difference were determined for each region. Average heat transfer coefficients inside the tube h_i was calculated by the equation

$$\frac{1}{h_i} = \frac{1}{K} - \frac{d_i \ln d_o/d_i}{2\lambda} - \frac{d_i}{h_o d_o}, \quad (5)$$

where h_o is the heat transfer coefficients outside the tube and λ the thermal conductivity of the tube wall material (SUS 304), and d_i and d_o are the inside and outside diameters of the tube, respectively. The λ 's were carefully determined basing on some data sources. They show linear dependency on temperatures. Typical values are 15.1 W/m·K at 300 K and 17.9 W/m·K at 500 K. Preliminary experiment was conducted to determine h_o for coiled tube bundle by flowing single phase water in secondary side. In the experiment, Dittus-Boelter's equation (6) for straight tube⁽¹⁰⁾ and Mori-Nakayama's equation (7) for coiled tube⁽⁸⁾ were assumed as secondary side heat transfer coefficients

$$Nu = 0.023 Re^{0.8} Pr^{0.4}, \quad (6)$$

$$Nu = \frac{1}{41} Re^{5/6} Pr^{0.4} \left(\frac{d}{D} \right)^{1/12} \left(1 + \frac{0.061}{(Re(d/D)^{2.5})^{1/6}} \right), \quad (7)$$

then, h_o value thus determined was 6,400 W/m²·K.

III. EXPERIMENTAL RESULTS

1. Friction Pressure Drop at Boiling Region

Figures 5(a)~(c) show the friction pressure drop at boiling region against flow rate for three steam pressures. From the figures, following results were reduced: (1) Pressure drop increases both with the increase of flow rates and with the decrease of steam pressure. (2) Pressure drop decreases with the increase of primary water temperature or with the increase of the heat flux. Average heat flux q at boiling region was in the ranges of $(0.7 \sim 1.8) \times 10^5$ W/m² in present experiments.

Figures 6(a)~(c) show the friction pressure drop per unit length of boiling region against flow rates for three steam pressures. Though they also increase both with the increase of flow rates and with the decrease of steam pressure, the effect of water temperature disappears. These facts mean that the increase of primary water temperature affects only on the decrease of the boiling region length, and that the effects of boiling phenomena on the friction pressure drop are not distinct in the fluid conditions of once-through steam generator of integrated type marine water reactors.

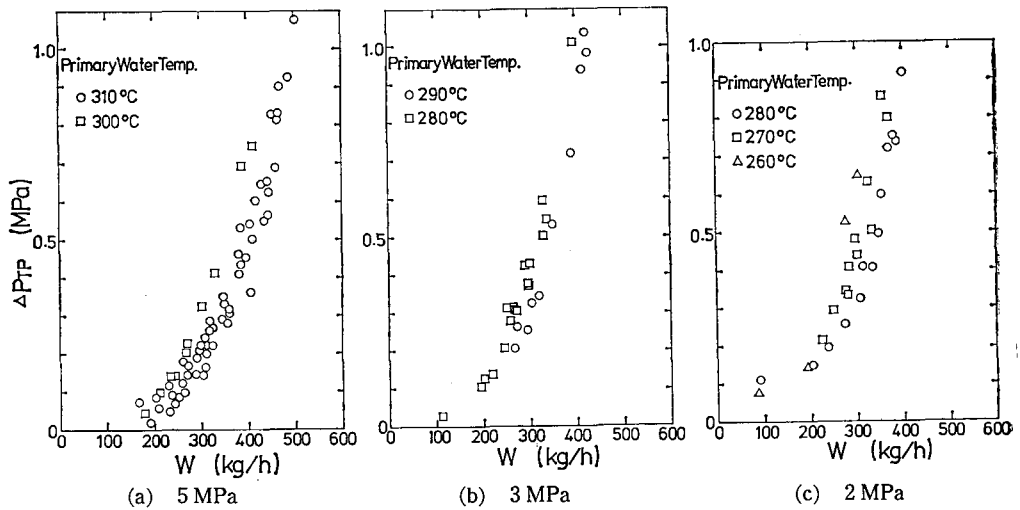


Fig. 5(a)~(c) Friction pressure drop at boiling region for three SG exit steam pressures

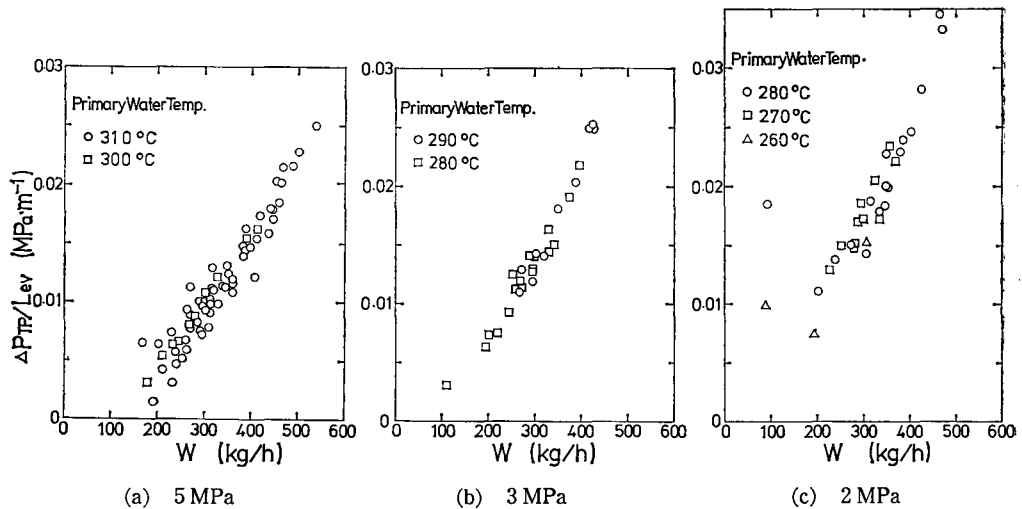
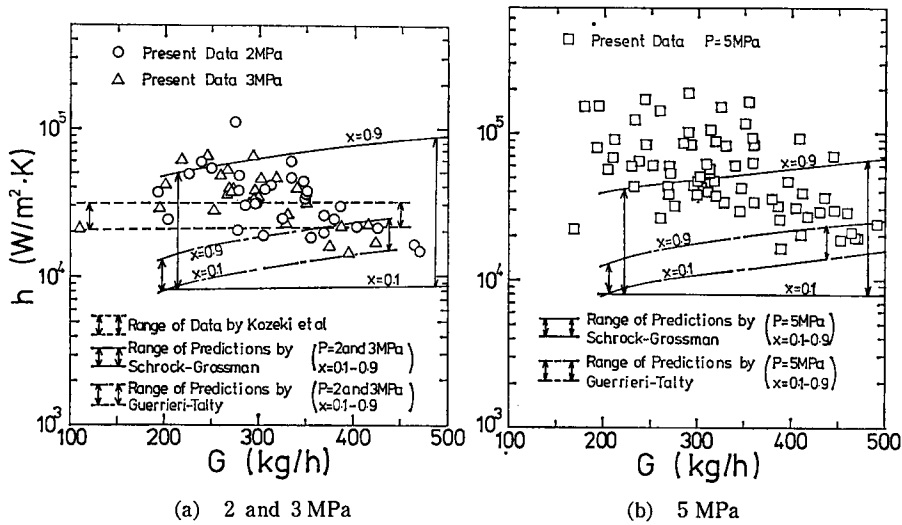


Fig. 6(a)~(c) Friction pressure drop per unit length of boiling region for three SG exit steam pressures

2. Heat Transfer Coefficients at Boiling Region

Figures 7(a) and (b) show the average heat transfer coefficients at boiling region against the flow rates. These values are only the average one, since the fluid experiences nucleate boiling, forced convection vaporization, dryout and mist flow at boiling region. Both figures show that even though the experimental data are scattering, the average heat transfer coefficients decrease with the increase of flow rates. The data at 5 MPa pressure condition are larger than those of 2 and 3 MPa pressures. The scattering of the data is caused by two main reasons. One is that the 63 data were recorded with a digital volt meter (5 points/s) in one run even though each data was fluctuating with time. Another reason is that h_i is obtained by Eq. (5), as a difference of a large value and another large value. Mean values of the scattered data are only meaningful in Fig. 7(a), (b).



(a) 2 and 3 MPa (b) 5 MPa
Fig. 7(a), (b) Average heat transfer coefficient at boiling region for three SG exit steam pressures

IV. DISCUSSION OF RESULTS

1. Friction Pressure Drop at Boiling Region

Friction pressure drop data at boiling region is including complex secondary flow effects in helically coiled tubes, and predictions of them are very difficult works. Discussions were conducted with some well-known correlations.

Martinelli-Nelson's prediction⁽¹¹⁾ of two-phase friction pressure drop in straight tube ΔP_{TP} is given by the single-phase friction pressure drop ΔP_{l0} assuming all the flow in water phase, multiplied by a coefficient R_{MN} .

$$\Delta P_{TP} = R_{MN} \Delta P_{l0}, \tag{8}$$

$$R_{MN} = (1-x)^{1.75} \Phi_{lt}^2 = \Phi_{l0}^2(P, x), \tag{9}$$

where x is quality and Φ_{l0}^2 is given as a function of both pressure P and quality x . Experimental values for Φ_{l0}^2 were given by Martinelli-Nelson in table. Ito's equation (4) was used for the evaluation of single-phase pressure drop ΔP_{l0} . Prediction by Eq. (8) and Ito's equation was named as modified Martinelli-Nelson's prediction for coiled tube. This means that the helical coil effects are completely included in single-phase pressure drop calculation.

Modified Kozeki's prediction named in present paper, is almost the same method as that of modified Martinelli-Nelson's prediction, except that it uses the experimental values by Kozeki for two-phase friction pressure drop at coiled tube under the pressure of 1.1 MPa⁽¹²⁾. First of all, parameter Φ_{gt}^2 was made to fit the experimental values so that $\Phi_{lt} \rightarrow 1$ when $x \rightarrow 0$ and $\Phi_{gt} \rightarrow 1$ when $x \rightarrow 1$.

$$\Phi_{gt}^2 = 0.895 + (X_{lt} + 0.076)^{0.875} + 1.21 \times 10^{(-0.334(\log X_{lt} + 0.668)^2)}. \tag{10}$$

The following relation exists between Φ_{lt} and Φ_{gt} :

$$\Phi_{lt} = \frac{\Phi_{gt}}{X_{lt}^{0.875}}. \tag{11}$$

In Eqs. (10) and (11), X_{lt} is known as Martinelli's parameter

$$X_{tt} = \left(\frac{\rho_g}{\rho_l}\right)^{0.571} \left(\frac{\mu_l}{\mu_g}\right)^{0.143} \left(\frac{1-x}{x}\right) \quad (12)$$

Figure 8 shows the relation between Eq. (10) and Kozeki's experimental values. Parameter $\Phi_{tt}(P)$ at arbitrary pressure P can be derived from Eqs. (10) and (11) by logarithmic interpolation between 1.1 MPa ($=P_0$) and critical pressure P_{cr} . At critical pressure, Eq. (13) exists⁽¹¹⁾

$$\Phi_{tt}(P_{cr}) = \left(\frac{1+X_{tt}}{X_{tt}}\right)^{0.875} \quad (13)$$

$$\log \Phi_{tt}(P) = \log \Phi_{tt}(P_0) - \left\{ \log \Phi_{tt}(P_0) - \log \Phi_{tt}(P_{cr}) \right\} \frac{P - P_0}{P_{cr} - P_0} \quad (14)$$

As the experimental values by Kozeki include the coiled tube effects, ΔP_{to} was calculated by the straight tube correlation. In Fig. 9, predictions by Eq. (14) was compared with the experimental values at 2.1 MPa pressure conducted at NSR-4 committee⁽²⁾. Agreements are good.

Figures 10(Ia)(Ib), (IIa)(IIb) and (IIIa)(IIIb) show the comparisons of both modified Kozeki's prediction and modified Martinelli-Nelson's prediction with our experimental data for frictional pressure drop at boiling region. The coordinate in each figure is the experimental data divided by each prediction, and the abscissa is the length of boiling region L_{ev} divided by total length of heat transfer tube L . The reason why L_{ev}/L was chosen as the abscissa is as follows: ΔP_{TP} was derived by Eq. (1). If the evaluation of each term in Eq. (1), especially that of ΔP_{SH} which is the main term in Eq. (1), is not correct, errors in the evaluation of ΔP_{TP} may become larger with the smaller L_{ev}/L or with the larger L_{SH} . As shown Fig. 10(I)~(III), distinguished errors are not seen at the smaller L_{ev}/L . Figures 10(Ia), (IIa) and (IIIa) show the comparisons with modified Kozeki's prediction, and Figs. 10(Ib), (IIb) and (IIIb) show the comparisons with modified Martinelli-Nelson's prediction. Following conclusions are deduced from these figures.

- (1) Though the predictions are in general smaller than the experimental data, the former are in accordance with the latter within 30%.
- (2) Modified Kozeki's prediction is better than modified Martinelli-Nelson's prediction.
- (3) The main reason why the scatters of the data are seen at smaller L_{ev}/L regions, might be reduced to the measurement errors, as the absolute values of pressure drop are small compared with the instrumentation accuracy at the small flow rates or small L_{ev} .
- (4) Predictions by Baroczy's⁽¹³⁾ or Thom's⁽¹⁴⁾ correlation were worse than modified Kozeki's and modified Martinelli-Nelson's predictions, though the figures of the comparisons were omitted in present report.

The reason why modified Kozeki's prediction is the best, is that the prediction is based on the experimental data with coiled tube at the similar configurations and experimental conditions to present experiment.

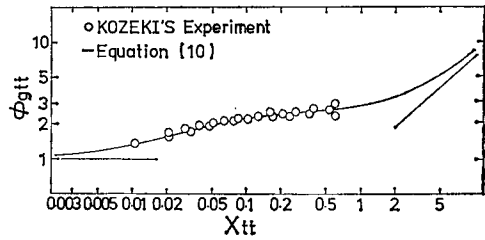


Fig. 8 Relation between Kozeki's data and Eq. (10)

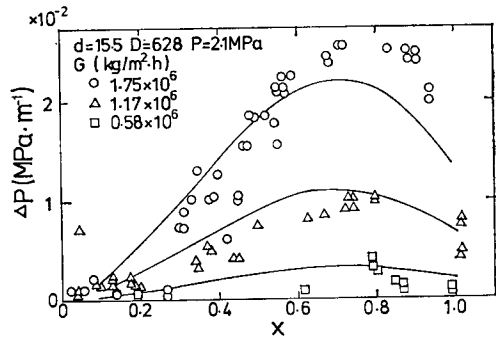


Fig. 9 Comparisons of Eq. (14) with experimental values at NSR-4 research

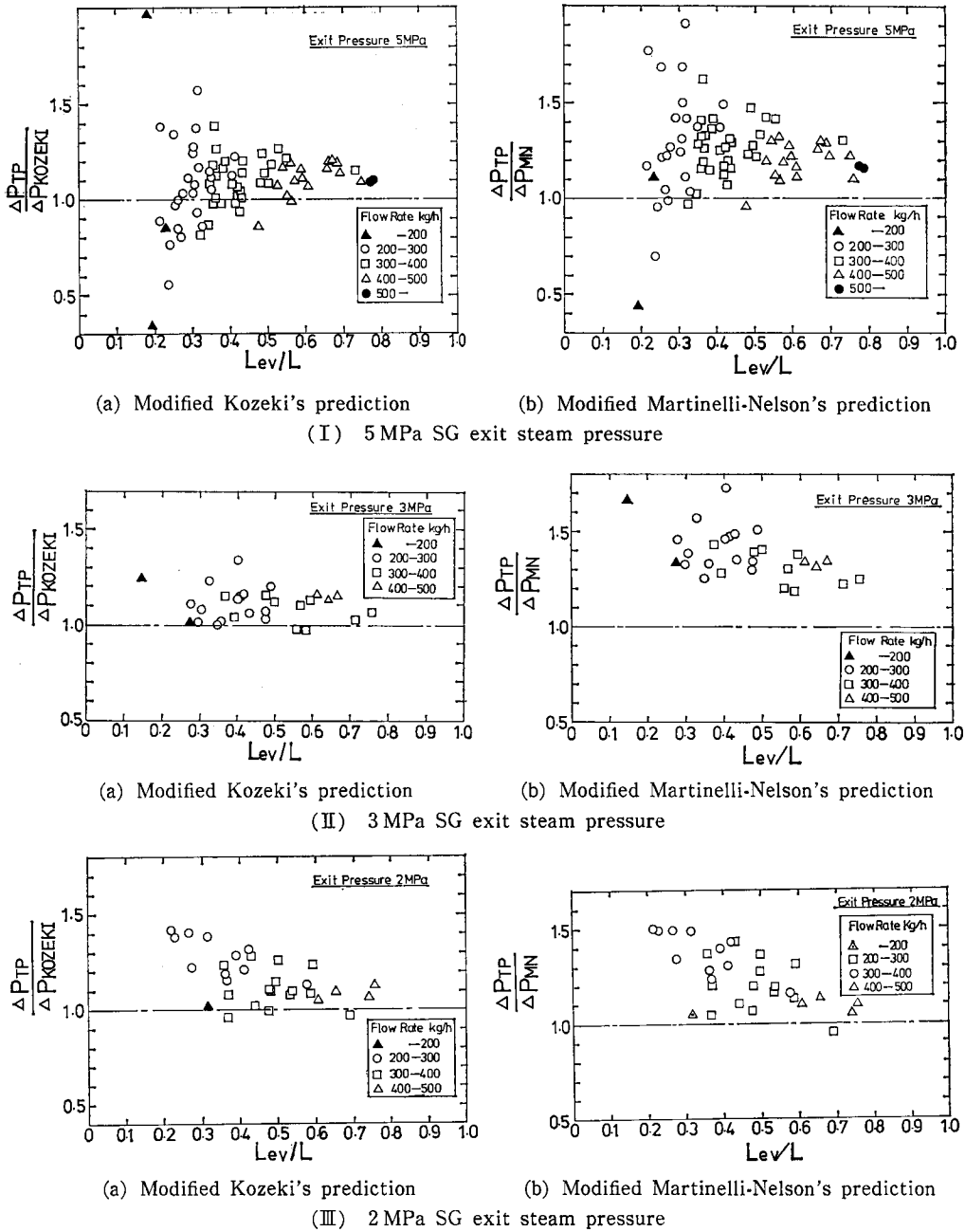


Fig. 10(I)~(III) Comparisons of predictions with experimental data in case of three SG exit steam pressures

2. Heat Transfer Coefficients at Boiling Region

Strictly speaking, heat transfer coefficients in helically coiled tube show locally the different values at outside and at inside of the radial direction of coil and also changes with quality and pressure. We treated, however, average values along whole boiling region, and tried to compare the average experimental values with two well-known correlation for straight tube and the experimental results of NSR-4 committee.

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Schrock-Grossman's correlation is one of the most famous one for the prediction of heat transfer coefficients at annular forced convection vaporization flow regime⁽¹⁵⁾

$$h_{TP} = 170 \frac{\lambda_l}{d} Re_{l0}^{0.8} Pr^{1/3} (Bo + 1.5 \times 10^{-4} X_{tt}^{-0.67}), \quad (15)$$

$$Bo = q/Gr, \quad (16)$$

$$Re_{l0} = dG/\mu_l, \quad (17)$$

$$\frac{1}{X_{tt}} = \left(\frac{x}{1-x} \right)^{0.9} \left(\frac{\rho_l}{\rho_g} \right)^{0.5} \left(\frac{\mu_g}{\mu_l} \right)^{0.1}. \quad (18)$$

Another famous correlation taking a little different form, is by Guerrieri-Talty for forced convection vaporization flow⁽¹⁶⁾

$$h_{TP} = 3.4 h_l \left(\frac{1}{1-x} \right)^{0.45}, \quad (19)$$

$$h_l = 0.023 \frac{\lambda_l}{d} Re_l^{0.8} Pr_l^{0.4}, \quad (20)$$

$$Re_l = \frac{dG(1-x)}{\mu_l}. \quad (21)$$

Equations (15) and (19) include quality x , and so the predictions of h_{TP} by them changes widely with it. As our experimental values are average one, we compared them with the values at $x=0.1$ and at $x=0.9$. The average one may be between them.

Kozeki *et al.* derived two-phase flow heat transfer coefficients in coiled tube under the pressures of 0.5~2.1 MPa at the NSR-4 research. The h_{TP} values derived there are around 2.3×10^4 W/m²·K for the wide ranges of qualities and flow rates.

Figure 7(a) shows the comparisons of experimental data at 2 and 3 MPa pressure with above-mentioned predictions. The predictions by Schrock-Grossman and Guerrieri-Talty increase with the flow rates, though the experimental data decrease. The experimental data, however, agree fairly well with both Kozeki *et al.*'s data and Schrock-Grossman's prediction.

Figure 7(b) is the case of 5 MPa pressure. The experimental values are larger than Schrock-Grossman's prediction at low flow rates. The following results are derived from both figures:

- (1) Average heat transfer coefficients at 2 to 3 MPa pressure are very close to the predictions by Schrock-Grossman's correlation. This means that the average heat transfer coefficients in coiled tube does not differ so much from those in straight tubes.
- (2) Schrock-Grossman's correlation was derived with the experimental data of steam-water flow at the pressure ranges of 0.5 to 3.5 MPa. This may be the main reason why our data did not agree with Schrock-Grossman's correlation at 5 MPa pressure.
- (3) Guerrieri-Talty's correlation bases on the experimental data for organic fluids. This may be the main cause of the disagreement with present experimental data.
- (4) Heat transfer coefficients decrease with the increase of flow rates. This is explained assuming that dryout occurs at smaller quality than 100% with large flow rates. We tried to estimate the dryout qualities, so that the average heat transfer coefficients at boiling region assuming the heat transfer coefficients for nucleate boiling by

Nishikawa *et al.*⁽¹⁷⁾, for forced convection vaporization by Schrock-Grossman, and for mist flow by Miropolskii⁽⁶⁾ show the same tendency to decrease with the increase of flow rate as in the experiments.

Figure 11 shows the relation between dryout qualities and flow rates for 13 typical data selected among whole the experimental conditions. As shown in the figure, dryout quality decreases with the increase of the flow rates. Average dryout qualities of more than 90% were also derived at NSR-4 experiment.

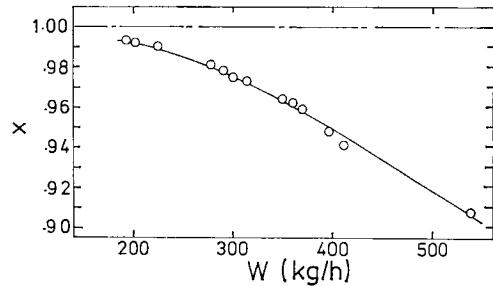


Fig. 11 Relation between dryout qualities and flow rates

V. CONCLUSION

Thermo-hydraulic characteristics in a once-through steam generator with helically coiled tubes, were investigated with the model test rig of an integrated type marine water reactor. As the experiments were conducted under the similar dimensions and fluid conditions to the real steam generator, almost all of the experimental data could be applied directly to the real reactor design. The effect of coiled tube on the friction pressure drop at boiling region should be taken into account. Modified Kozeki's and modified Martinelli-Nelson's predictions agree with the experimental values within 30%. The effect of coiled tube may, however, be smaller in real steam generators than that in present test rig, as the coil diameter of the former is larger than that of latter. The effect of coiled tube on the average heat transfer coefficients is small, and the predictions for straight tube can also be applied to coiled tube. Schrock-Grossman's correlation agrees well with the experimental values at the pressures of lower than 3.5 MPa. Dryout at more than 90% quality should be taken into account for the evaluation of heat transfer characteristics.

[NOMENCLATURE]

Bo : Boiling number	λ : Thermal conductivity (W/m·K)
D : Coil diameter (m)	μ : Viscosity (Pa·s)
d : Inside diameter of tube (m)	ρ : Density (kg/m ³)
f : Coefficient of friction pressure drop	Φ : Parameter
G : Mass flow rate (kg/m ² ·s or kg/m ² ·h)	(Subscript)
h : Heat transfer coefficient (W/m ² ·K)	ac : Acceleration
K : Overall coefficient of heat transfer (W/m ² ·K)	c : Coiled tube
L : Length (m)	cr : Critical condition
Nu : Nusselt number	ev : Boiling region
P : Pressure (Pa)	g : Steam
ΔP : Pressure drop (Pa)	gtt : Gas phase turbulent flow
Pr : Prandtl number	$head$: Head
q : Heat flux (W/m ²)	i : Inside the tube
Re : Reynolds number	l : Liquid
r : Latent heat of vaporization (J/kg)	lo : Total flow in liquid phase
u : Fluid velocity (m/s)	ltt : Liquid phase turbulent flow
W : Flow rate (kg/h)	MN : Martinelli-Nelson
X_{tt} : Martinelli parameter	o : Outside the tube
x : Quality	SH : Superheater region
ϵ : Surface roughness of heat transfer tube (m)	Sub : Subcooling region
	s : Straight tube
	T : Total, TP : Two phase

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