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THERMAL-HYDRAULIC ANALYSIS OF WATER-WATER HEAT EXCHANGER UNDER LOW FLOW CONDITIONS USING CFD CODE

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Abstract

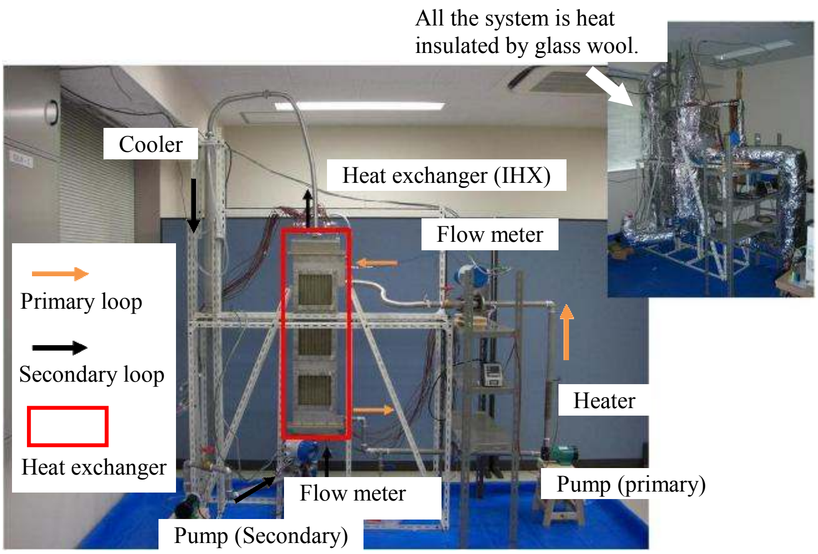
In order to establish the evaluation method of the local heat transfer in the intermediate heat exchanger (IHX) for a fast breeder reactor, a CFD analysis method has been applied to a heat exchanger with the primary and secondary water-loops. Analyses were conducted under the forced circulation and natural circulation conditions. For the forced circulation experiment with the Reynolds number at 10^4, a quasi-steady state condition is analyzed. For the natural circulation experiment, an analysis is also conducted for a quasi-steady state condition where the Reynolds number is approximately 10^2. The calculated heat transfer coefficients are converted into the Nu numbers and compared with the experimental results. Good agreement is obtained between the analytical results and the test results. Temperature distributions by the calculation results with the 1-dimensional NETFLOW++ code and CFD code are compared with the test results. For the natural circulation condition, it is clarified that there is almost no temperature distribution in radial direction, and the temperature is distributed only in axial direction. The flow on the primary-side seems to be rectified by the group of the heat transfer tubes and the turbulence is suppressed. For the forced circulation condition, the flow on the primary-side of the heat exchanger is stabilized also. The present CFD evaluation method can be applied to the IHX of the fast reactor with complex flow system.

Introduction

The present paper describes an analytical study of a water-to-water heat exchanger. The present analysis has been conducted on the basis of the experiment reported by Suzuki and Mochizuki[1]. The experimental system is shown in **Figure 1**. An intermediate heat exchanger (IHX) used in the experiment is a scaled model of the IHX at the 50MW steam generator facility. **Table 1** shows the comparison of dimensions between IHXs in different facilities. The 50 MW Steam Generator Facility (50MW SG) was built to investigate the heat transfer characteristics of the SG for the 'Monju' reactor. The IHX of the test loop has 35 heat transfer tubes that is 1m in length, 6mm in inner diameter and 8 mm in outer diameter. The heat transfer was conducted with the countercurrent water-water flows. The electrically heated water was circulated by a pump in the primary loop, and the water cooled by a water cooler in the secondary loop was circulated by another pump. A series of the experiments were conducted by cooling down the secondary loop after the establishment of the isothermal initial conditions for both loops. In the natural circulation test, both pumps were tripped at the same time of the initiation of the cooling. This heat exchanger is a scaled model of an IHX of a sodium-cooled facility. The IHX of a fast breeder reactor is important equipment which transfers heat from the core to the secondary heat transport system. Mochizuki and Takano[2] investigated heat transfer of the IHX and found out the Nu number was degraded under the low Péclet number conditions. Mochizuki[3] tried to explain this phenomenon by the heat conduction in flow direction. However, he found out the heat conduction in flow direction has almost no effect on the degradation. Analyses using CFD code were conducted under the forced circulation and natural circulation conditions.[4] In the present study, the velocity distribution and the temperature distribution in the IHX is calculated precisely. The local heat transfer coefficient is evaluated by the calculated temperature distribution and compared with the experimental results. The comparison between the calculated and experimental results is useful in terms of verification for the CFD evaluation of the sodium-sodium IHX with more than 3000 heat transfer tubes of the fast breeder reactor.

Table 1 Comparison of IHX terms between the 50MW SG and the present experimental apparatus

Terms	unit	50MW SG	Apparatus
Outer diameter of heat transfer tube (d_o)	mm	15.9	8.0
Inner diameter of heat transfer tube (d_i)	mm	13.5	6.0
Pitch of heat transfer tubes	mm	23.5	11.5
Effective heated length	m	3.75	1.0
Hydrodynamic equivalent diameter of shell side (D_h)	mm	28.53	24.17
Thermodynamic equivalent diameter of shell side (D_H)	mm	29.66	14.53
Flow area of shell side (A_s)	m^2	0.7573	5.316×10^{-3}
Total flow area of tube side (A_t)	m^2	0.2926	9.896×10^{-4}



Specification of Heat exchanger apparatus

	Type	Shell-and-tube
Shell	Size	Height:1020mm, length: 265mm, width: 26.7mm, wall thickness: 2mm
	Material	Stainless steel
Heat transfer tube	Numbers	35
	Size	O.D.: 8mm, I.D.: 6m, Length: 1m
	Material	Brass
Pump	Flow rate	38 l/min
	Head	5.4 m
Thermo-couple	Type	Sheathed, S35 T-type
	O.D.	1.6 mm
	Numbers	35
Heater	Type	Micro-heater-type
	Capacity	1 kW
	Length	6.12 m

Figure 1 Experimental apparatus of water-water heat exchanger with Primary and Secondary loops.

1. Experiment

1.1 Experimental procedure

Two kinds of experiment were conducted using the experimental apparatus, i.e., a forced circulation test and a natural circulation test. In the forced circulation test two pumps were operated during the transient. One of the tests is named F-10. In the natural circulation test, two pumps were tripped during the transient. One of the tests is named N-10. The operating procedures of these tests are illustrated in **Figure 2**. The experimental temperatures at the inlet and outlet of the heat exchanger for 10,000 sec are shown in **Figure 3** together with the analytical result [1] using the one–dimensional plant dynamics analysis code NETFLOW++. The analytical results for both cases trace the temperature trends very well. It is shown from the figure that the temperature after 8,000 sec would be in the quasi-steady state.

Experiment was conducted in the following procedures.

1 fill water to primary system, secondary system and cooler .
2 open the valve in primary and secondary system.
3 start up the primary pump and the secondary pump.
4 put on the switch with temperature control units at 75 ºC set point.
5 check the panel with temperature control units on steady state.

Procedure for forced circulation (F-10)

6 water is fed into the water cooler (10ℓ/min)
7 change the temperature set point from 75 ºC to 90 ºC
8 both flows are controlled at 10 ℓ/min

Procedure for natural circulation (N-10)

6 trip pumps in primary and secondary loops
7 water is fed into a cooler at 10ℓ/min
8 change the temperature set point from 75 ºC to 90 ºC

The experiment is finished after 10,000 seconds from the starting time of the transient.

Note: Procedures No. 6-8 have finished in less than 1minute.

Figure 2 Procedure of an experiment of a forced circulation test (F-10) and a natural circulation test (N-10).

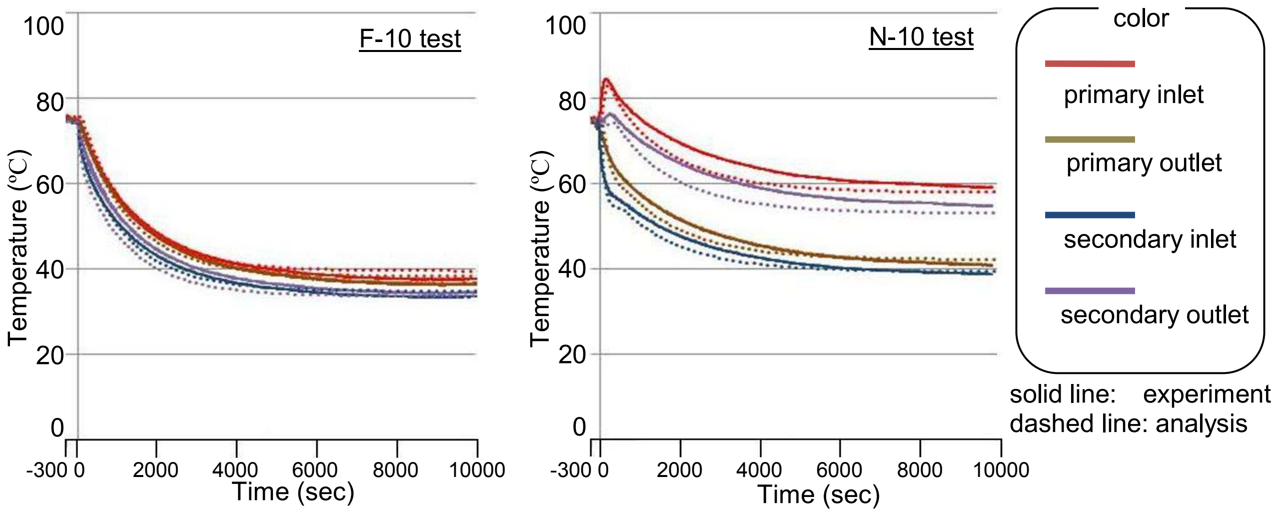


Figure 3 Experiment results of the forced circulation test (F-10) and the natural circulation test (N-10).

1.2 Measuring thermocouples points and results

An analysis is conducted for a quasi-steady state condition where is 9,000 sec from the initiation of the transient by the trip of the two pumps and cooling-down the secondary loop. In the 3D analysis, thermocouple locations are important to compare the results. The locations are shown in **Figure 4**. The experimental temperatures at 9,000 seconds of F10 and N10 are listed in **Table 2**.

Table 2 Measuring thermocouples points (F-10, N-10) of the IHX at 9,000 seconds.

Measuring thermocouples points (F-10) unit: ºC

primary-side	inlet	A-1 (0.9m)	A-2 (0.7m)	A-3 (0.5m)	A-4 (0.3m)	A-5 (0.1m)	outlet	A
	38.8	38.8	38.7	38.7	38.5	38.2	37.6	
	inlet	E-1 (0.9m)	E-2 (0.7m)	E-3 (0.5m)	E-4 (0.3m)	E-5 (0.1m)	outlet	E
	38.8	38.3	38.4	38.1	38.1	37.5	37.6	
secondary-side	1.0m(outlet)	0.9m	0.7m	0.5m	0.3m	0.1m	0m (inelt)	A
	36.0	36.9	36.4	36.2	35.8	35.8	34.5	
	1.0m(outlet)	0.9m	0.7m	0.5m	0.3m	0.1m	0m (inlet)	E
	35.8	36.7	36.2	35.9	35.7	35.2	34.5	

Measuring thermocouples points (N-10) unit: ºC

primary-side	inlet	A-1 (0.9m)	A-2 (0.7m)	A-3 (0.5m)	A-4 (0.3m)	A-5 (0.1m)	outlet	A
	59.6	58.0	53.7	50.3	47.1	44.2	41.3	
	inlet	E-1 (0.9m)	E-2 (0.7m)	E-3 (0.5m)	E-4 (0.3m)	E-5 (0.1m)	outlet	E
	59.6	57.6	53.5	50.2	47.1	44.1	41.3	
secondary-side	1.0m(outlet)	0.9m	0.7m	0.5m	0.3m	0.1m	0m (inlet)	A
	55.8	55.1	51.6	48.2	45.0	42.0	39.1	
	1.0m(outlet)	0.9m	0.7m	0.5m	0.3m	0.1m	0m (inlet)	E

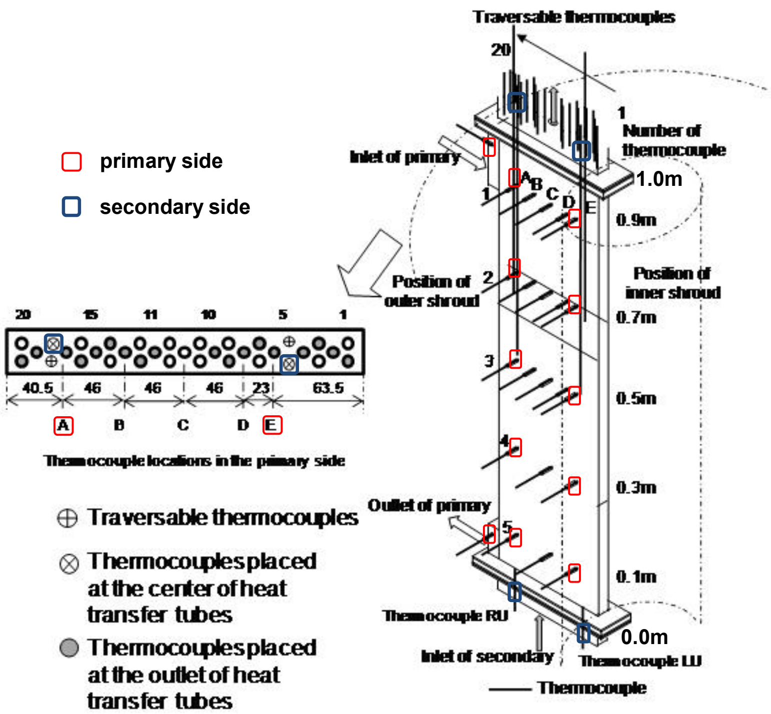


Figure 4 Thermocouples locations in the experimental apparatus

2. CFD calculation

2.1 CFD- modeling

The CFD modelling produced for the experimental apparatus includes the overall loops for the primary and the secondary systems. However, in the present study, the heat exchanger itself that is a part of the experimental apparatus loop is chosen for the analysis. **Figure 5** shows the part of the CFD analysis model for the apparatus.

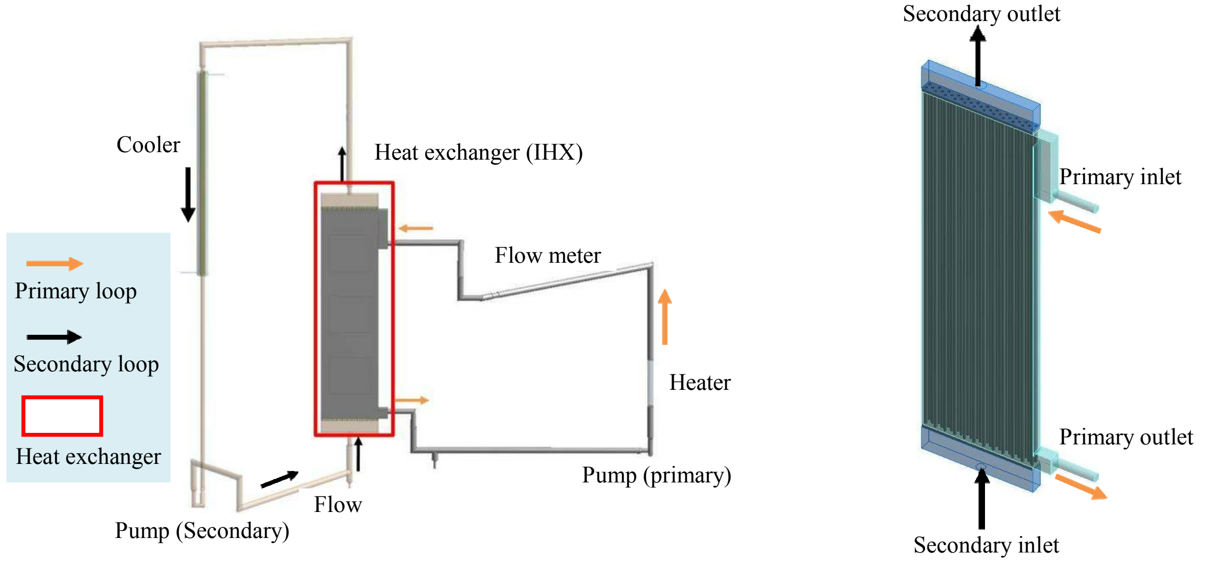


Figure 5 CFD-modeling of IHX with experimental apparatus

2.2 CFD-meshing

The calculation mesh is generated for the model explained above. The CFD-meshing are shown in **Figure 6**. The number of the total meshes is approximately 10,000,000 using tetrahedral lattices. The boundary layer is specified near the tube wall surface. This layer is arranged perpendicularly to the tube surface. The total thickness of 5th layers is 0.75 mm. The 1st layer thickness of 0.1mm from the surface near the wall is multiplied by 1.2 for the next layers. This mesh type is

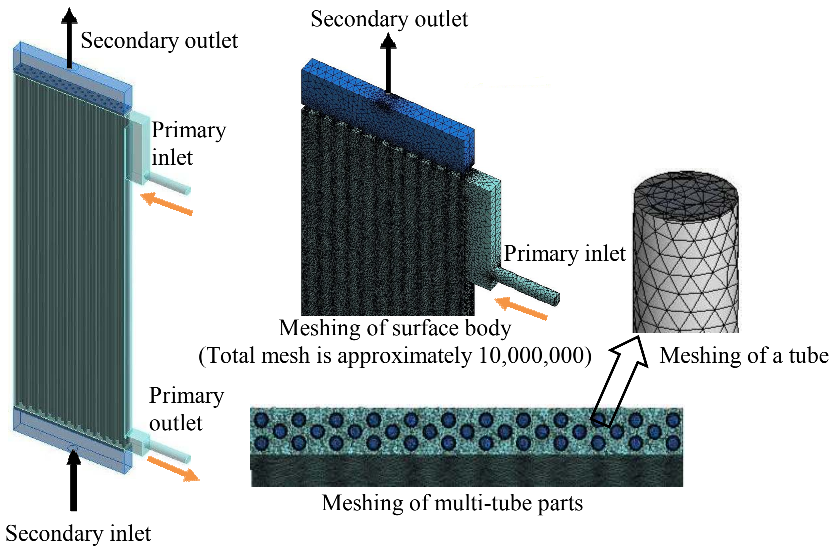


Figure 6 CFD-meshing of IHX with experimental apparatus

pentahedral lattices. Furthermore, the size of the tetrahedral mesh inside the tube is 1.5 mm on an average.

2.3 CFD-setting

The following physical properties are used in the CFD analysis. [5]

2.3.1 Properties of cooper tube (300<T<600K)

density (kg/m^3) $$D_{copper} = -0.33333 \times T + 8980 \quad (1)$$

specific heat (kJ/kg/K) $$Cp_{copper} = 0.00013 \times T + 0.347 \quad (2)$$

thermal conductivity (W/m/K) $$k_{copper} = -0.05 \times T + 413 \quad (3)$$

2.3.2 Properties of water (273.15<T<373 K)

density (kg/m^3)
$$D_{water} = -1.92 \times 10^{-7} T^4 + 2.620459 \times 10^{-4} T^3 - 0.13682 \times T^2 + 31.8024 \times T - 1750.3 \quad (4)$$

specific heat (kJ/kg/K)
$$Cp_{water} = 2.13333 \times 10^{-9} T^4 - 2.87488 \times 10^{-6} T^3 + 1.46027 \times 10^{-3} T^2 - 0.33082 \times T + 32.345 \quad (5)$$

viscosity (Pa*s)
$$\mu_{water} = \left(2.968533 \times 10^{-5} T^4 - 4.097553 \times 10^{-2} T^3 + 21.246 \times T^2 - 49.101034 \times 10^2 T + 427632\right) \times 10^{-6} \quad (6)$$

thermal conductivity (W/m/K)
$$k_{water} = \left(-1.28 \times 10^{-7} T^4 + 1.85719 \times 10^{-4} T^3 - 0.107767 \times T^2 + 29.8527 \times T - 2624.1\right) \times 10^{-3} \quad (7)$$

2.3.3 Conditions of CFD analyses

A turbulent scheme selected for the CFD calculation is a standard k-ε with scalable wall function. In addition, the up-wind differentiation with 2nd order is used for a convection term. The primary and secondary inlet mass flowrates and inlet temperatures are given to the code as the boundary conditions, and the pressure at the outlet is given as the boundary condition. The wall surface is modeled with the non-slip conditions. These CFD analysis conditions are summarized in **Table 3**. The wall surface of the outer shell is modeled with the non-slip conditions. In the present analysis, the heat transfer coefficient (HTC) from outer shell to the environment is assumed to be zero, because the thickness of heat insulator on the IHX is more than 30 cm. The HTC is also confirmed as negligible small by hand calculation.

Table 3 CFD-setting

Material	
Tube material: cooper (Cu)	
Fluid material: water (single phase)	
Pressure	0.1 MPa
CFD code	
ANSYS CFX version 12.0	
Solver-setting	
Turbulence model: Standard k-ε + scalable wall function	
(where, natural circulation is laminar)	
Convection term: 2nd order up-wind differencing	
Boundary conditions	
Inlet :Temperature (mass flowrate)	
Forced :primary	38.8 ºC (0.165 kg/s)
secondary	34.5 ºC (0.161 kg/s)
Natural :primary	59.6 ºC (0.0136 kg/s)
secondary	39.1 ºC (0.0146 kg/s)
Outlet: Pressure boundary	
Wall: Non-slip	

2.4 Calculation results using CFD code

The measured and calculated temperatures at the inlet and outlet of the heat exchanger at time 9000 seconds are listed in **Table 4**. Inlet temperatures on the primary and the secondary sides are not calculated results in the case of the CFD analysis but boundary conditions. The temperature boundary condition was not used in the case of the 1D-code NETFLOW++. The CFD contour maps of temperature and velocity in IHX of the experimental apparatus are shown in **Figure 7**. In the case of the forced circulation, velocity has a small local distribution. Temperature has also distribution according to the velocity distribution. In the case of the natural circulation, velocity in the shell shows a uniform distribution. Therefore, temperature distribution in the shell shows a uniform distribution in the radial direction. The axial temperature distribution in the shell of the IHX is shown in **Figure 8**. The calculation results of 3D-CFD and 1D-code were compared with the measured result. In regard to the temperature distribution on the shell-side, agreement of the 3D-CFD calculation result with the test result is better than that of 1D calculation. However, the calculated results in the heat transfer tube underestimate the measured temperature distribution. As a result of the comparison, it is estimated that the temperature distribution calculated by the 3D-CFD code could evaluate the temperature distribution in the complicated system.

Table 4 The measured temperatures at the inlet and outlet of IHX and calculation results using 1D-code and 3D-CFD code at 9000 seconds

		forced circulation (F-10)			natural circulation (N-10)		
(temperature, °C)		measured	1D-code	CFD	measured	1D-code	CFD
primary side	inlet	38.8	39.6	38.8	59.6	57.2	59.6
	outlet	37.6	38.4	37.9	41.3	42.4	42.0
secondary side	inlet	34.5	34.9	34.5	39.1	39.4	39.1
	outlet	35.9	33.7	35.4	55.7	53.2	54.0

	forced circulation (F-10)			natural circulation (N-10)		
(mass flowrate, kg/sec)	measured	1D-code	CFD	measured	1D-code	CFD
primary side	0.165	0.168	0.165	0.0136	0.0135	0.0136
secondary side	0.166	0.171	0.166	0.0146	0.0146	0.0146

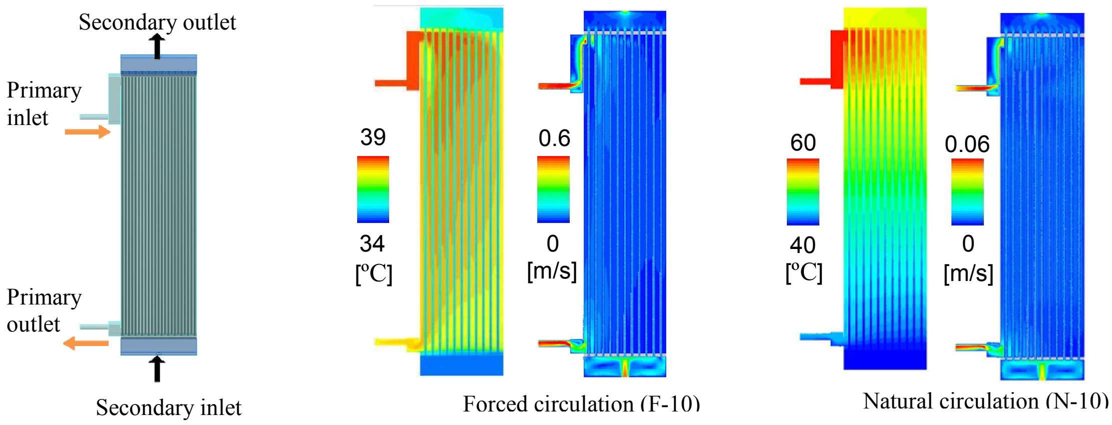


Figure 7 CFD contour maps of temperature and velocity in IHX of the experimental apparatus

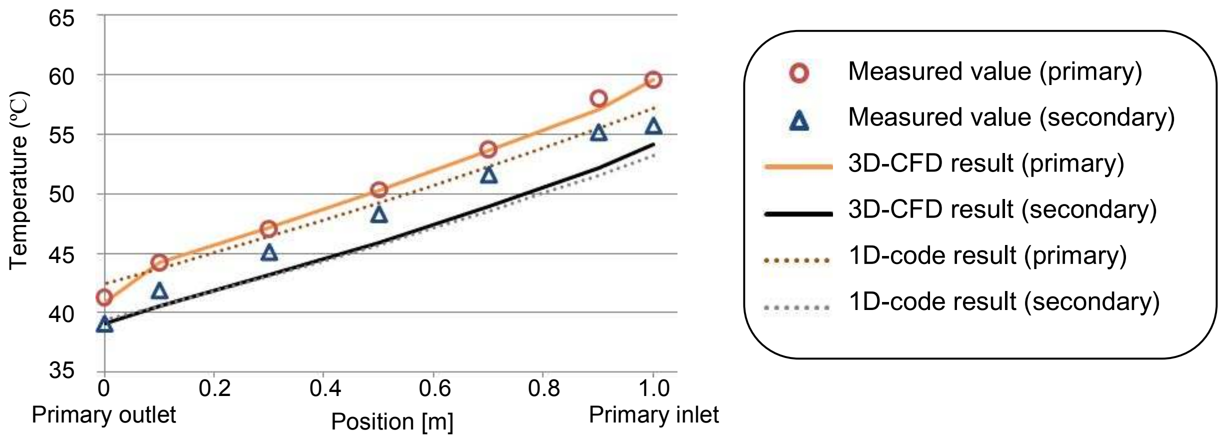


Figure 8　Comparison between measured and 1-D&3-D calculations for natural circulation test

3. Evaluation of local heat transfer under the low flow condition

3.1 Evaluation method of local heat transfer

The heat transfer rates of the outside and the inside of the heat transfer tube can be calculated using a three-dimensional calculation result.
inside

$$q_i = G_i \times (S_{i1} - S_{i2}) \tag{8}$$

outside

$$q_o = G_o \times (S_{o1} - S_{o2}) \tag{9}$$

where,
q_i : inside heat transfer rate (W), q_o : outside heat transfer rate (W), $G_i . G_o$: mass flowrate (kg/s),
$S_{i1}\ S_{i2}\ S_{o1}\ S_{o2}$: specific enthalpy (J/kg)
Moreover, the Nu number is calculated when the heat transfer coefficient is known. The heat transfer coefficient is expressed as follows.
inside

$$h_i = \lambda_i \frac{Nu}{d_i} = \frac{q_i}{(T_i - T_{wi})} \frac{1}{\pi d_i \ell} \frac{1}{n} \tag{10}$$,

outside

$$h_o = \lambda_o \frac{Nu}{d_o} = \frac{q_o}{(T_{wo} - T_o)} \frac{1}{\pi d_o \ell} \frac{1}{n} \tag{11}$$

where, h_i : inside heat transfer coefficient (W/m^{2}K), h_o : outside heat transfer coefficient (W/m^{2}K), λ_i : inside thermal conductivity (W/m^{2}K), λ_o : outside thermal conductivity (W/m^{2}K), q_i , q_o heat transfer rate (W), T_i : inside temperature (K), T_o : outside temperature (K), T_{wi} :inside wall temperature (K), T_{wo} : outside wall temperature (K), d_i :inner diameter (m), d_o : outer diameter (m), ℓ : axial length (m), n : number of tubes (-)

3.2 Evaluation results

The evaluated local heat transfer coefficient is expressed on the plain of the Nu number vs the Re number. The measured results are taken by Suzuki and Mochizuki[1]. The Dittus and Boelter correlation is usually applied to the flow in a pipe. They assumed that this correlation could be applied to the shell-side because the flow passages in between the heat transfer tubes were assumed as equivalent tubes. The evaluated Nu numbers from the experiment are scattered in the region higher than the Dittus and Boelter correlation over the low Re number (for 1,000 or less) region, i.e., laminar region, and turbulent region. However, the calculated result using CFD code shows the difference of the heat transfer coefficients between shell-side and tube-side.

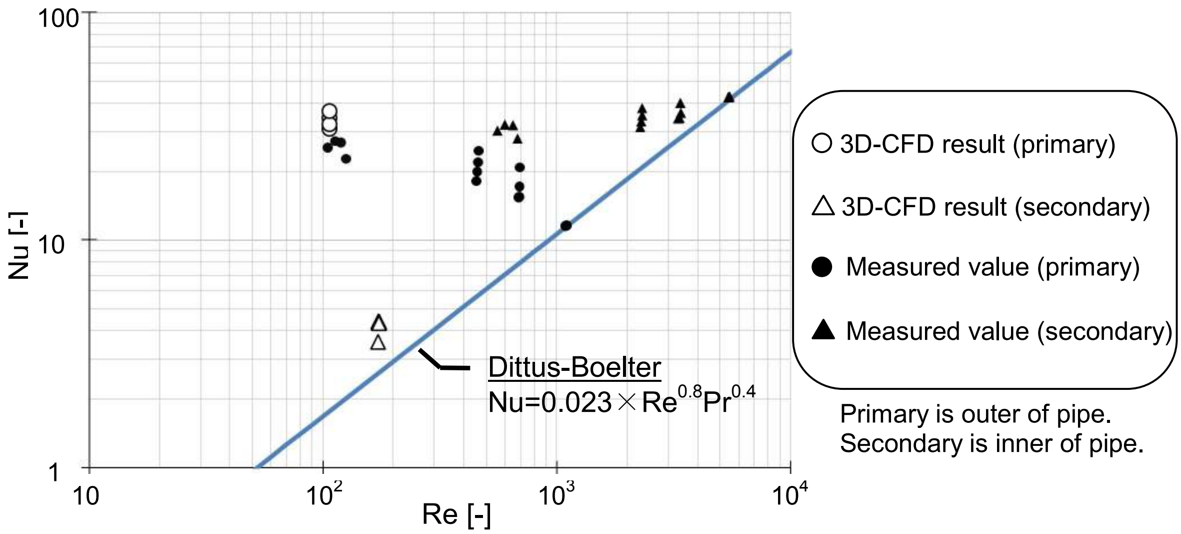


Figure 9 Comparison between Nu-Re formula results (Dittus-Boelter) and CFD calculation with natural circulation test

4. Conclusion

The following conclusions have been obtained by the present study.

(1) Temperature distributions in the IHX under forced and natural circulations have been calculated using 3D-CFD code. Good agreement has been obtained between the measured result and calculated result.

(2) The local heat transfer coefficient is calculated on the basis of the 3D-CFD calculation. The result shows that the heat transfer coefficients on the shell-side and the tube-side is different.

5. Acknowledgments

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6. References

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