Conjugate numerical simulation of wall temperature fluctuation at a T-junction pipe

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Abstract
Thermal fatigue cracks may occur in a T-junction pipe due to the mixing of hot and cold fluids. To develop an evaluation method for thermal fatigue, the authors previously performed a mixing tee experiment called the T-Cubic experiment. In this study, a fluid-structure coupled simulation for conjugate heat transfer was carried out to investigate the predictive performance of the flow and temperature fields and temperature fluctuation on the pipe inner surface at a mixing tee of the T-Cubic experiment. The computational domain included 304 type stainless steel pipe as well as the working fluid of water. Time-averaged velocity and temperature were reproduced well over the entire computational domain. Although velocity fluctuation intensity at a distance from the wall was relatively smaller than experimental data, the simulation could reproduce the trend of the experimental data, especially the velocity fluctuation intensity peak near the wall. The temperature fluctuation intensity was also larger than the experimental data, though the tendency could be reproduced by the simulation. The temperature fluctuation intensity on the pipe inner surface is the most important parameter for thermal fatigue and though it was 20% to 36% larger than the experimental data at its peak, the tendency was reproduced to a certain extent. The fluid temperature in the numerical simulation fluctuated at almost the same level from 0.1 Hz to 10 Hz, but high frequency components attenuated and low frequency components around 0.1 Hz remained on the pipe inner surface.

Keywords: Thermal fatigue, T-junction pipe, Heat transfer, Temperature fluctuation, Numerical simulation, Large eddy simulation

1. Introduction

Mixing tees are inevitable piping structures in fossil fuel and nuclear power plants. When high- and low-temperature flows are mixing in the mixing tee, high-cycle thermal fatigue may occur. Fatigue cracks have been often found downstream from the mixing tee, for example in the Civaux-1 (Chapuli et al., 2005) and more recently in U.S nuclear power plants (McDevitt et al., 2015). Hence thermal fatigue is one of the major degradation mechanisms that must be considered in nuclear power plant management.

The Japan Society of Mechanical Engineers (JSME) has issued a guideline for piping systems (JSME, 2003) to prevent thermal fatigue. The JSME guideline provides evaluation flowcharts of thermal fatigue, and it consists of a four-step assessment procedure. If one of the four steps is satisfied, the evaluation is finished. When the evaluation does not satisfy any steps, the “detailed evaluation” option remains in the evaluation flowchart. The JSME guideline allows substitution of another appropriate procedure as the “detailed evaluation” for the four-step assessment procedure. In addition, although the JSME guideline was based on extensive tests which cover various flow conditions (the inlet velocities and diameter ratios), not all conditions are covered in detail and the JSME guideline has a limitation for its application range. Moreover, the present procedure in the JSME guideline has conservativeness. Hence improvement of the JSME guideline based on experimental and numerical studies is desirable.

To understand the thermal fatigue mechanism, experiments under actual plant conditions are inevitable. The FATHER experiment (Braillard et al., 2007) measured fluid and pipe wall temperatures to observe the temperature...
fluctuation mechanism in the mixing tee. The fluid-structure-interaction (FSI) test facility (Kuschewski et al., 2013, Zhou et al., 2017) was used to measure pipe wall temperature when hot (265 °C) and cold (20 °C) flows were mixed and finally, cracking in the piping material was reproduced. The FATHER experiment and the FSI test facility measurements were carried out under the condition of a high temperature difference (over 150 °C). The Japan Atomic Energy Agency (JAEA) conducted a series of T-junction experiments called the WATLON experiment (Kamide et al., 2009). The test section was made of transparent acrylic resin and flow velocity and temperature distributions were measured. Though the WATLON experiment was conducted under a relatively low temperature difference (15 °C), flow velocity and temperature distributions were measured in the whole region of the T-junction to understand the mixing phenomena. Various researchers have spent considerable effort to develop a computational fluid dynamics (CFD) method to simulate the results of the WATLON experiment (Tanaka et al., 2010, Qian et al., 2012, 2015, Tanaka and Miyake, 2015).

The authors have also made numerical studies including CFD and structural analyses (Nakamura et al., 2009, 2010, 2012, 2015, Kamaya et al., 2011, Utanohara et al., 2016) to develop a method for “detailed evaluation” in the JSME guideline. The numerical method for the “detailed evaluation” is intended to cover CFD for temperature fluctuation in the local flow field, heat transfer from fluids to the pipe and heat conduction into the pipe structure, and then structural analysis for the thermal stress distribution. Hence, the fluid-structure coupled simulation for conjugate heat transfer is needed for this numerical simulation method. To confirm the validity of the prediction results, experimental data are needed. Since the WATLON experiment mainly measured the fluid temperature, only the predictive performance of the fluid temperature field has been investigated even if the conjugate numerical simulations were carried out. Although some experiments measured wall temperature at a mixing tee pipe (Braillard et al., 2007, Kimura et al., 2009), temperature was measured for only a few tens of points and the numbers of measurement points were not enough to validate the wall temperature distribution.

Hence, the authors performed a mixing tee experiment called the T-Cubic (Transient Temperature measurement equipment at a T-junction pipe) experiment (Miyoshi et al., 2014a, 2014b, 2014c, 2016). The T-Cubic experiment measured the temperature distributions on the wall using a test section with 148 thermocouples. These experimental data can be used as validation data for the conjugate numerical simulation. Additionally, the obtained wall temperature data can also be used to estimate the thermal stress distribution directly by finite element analysis (Miyoshi et al., 2016).

The final goal of the authors’ study is to develop a method for “detailed evaluation” in the JSME guideline, and to this end, in the present study, the predictive performance of the fluid-structure coupled simulation for the conjugate heat transfer was investigated through the comparison with the T-Cubic experiment data. Particularly, one of the important points for the thermal fatigue evaluation, the predictive performance of the temperature fluctuation intensity on the pipe inner surface was confirmed and the attenuation characteristics of the temperature fluctuation during heat transfer from the fluid to the pipe wall was investigated.

2. Numerical simulation

2.1 Simulation target

The simulation target was the T-Cubic experiment loop (Miyoshi et al., 2014b, 2014c, 2016). A schematic diagram of this loop is shown in Fig. 1. The loop consisted of a hot water tank, cold water tank, mixing water tank and two pumps. The disturbance in the flow was reduced by the flow straighteners placed upstream from the test section. The test section of the mixing tee is shown in Fig. 2 and it was made of 304 type stainless steel pipes. The inner diameters of the main and branch pipes $D_m$ and $D_b$ were 150 mm and 50 mm, respectively. The edge of the tee pipe was not rounded and the only burr at the edge was taken off by light-chamfering. The geometry of the test section was almost the same as that of the WATLON experiment (Kamide et al., 2009). The pipe was covered with heat insulator.

Experimental conditions are summarized in Table 1. Their details were described elsewhere (Miyoshi et al., 2014c). The flow pattern was classified as the wall jet (Kamide et al., 2009), in which the jet from the branch pipe was bent toward the main pipe wall due to the higher main flow velocity, and a larger temperature fluctuation was observed downstream from the junction.

2.2 Temperature measurement

Temperature distributions in fluid were measured by the thermocouple tree shown in Fig. 3. Thermocouples were
non-grounded K-type JIS class 1 and the diameter of their sheath was 0.25 mm. They were arranged in a row and 15 of them were placed from the center of the main pipe to \( r = 70 \) mm with a separation distance of 5 mm and the sixteenth was at \( r = 74 \) mm. In order to reinforce the structure of the thermocouples, each of them was inserted and brazed to a cover tube, 1 mm in outer diameter and 50 mm in length, with the thermocouple head of 5 mm length remaining exposed. The thermocouple tree could be rotated and moved in the axial direction of the main pipe.

The temperature of the pipe inner surface was measured by sheathed thermocouples (diameter, 0.5 mm) installed near the inner surface as shown in Fig. 4 (Miyoshi, et al., 2014b). Non-grounded K-type thermocouples were brazed into grooves 0.7 mm deep and 0.6 mm wide on the pipe inner surface. All the thermocouples were brazed by Ni in a high vacuum furnace, because some voids were observed at the corner of the groove when they were brazed at atmospheric pressure (Miyoshi, et al., 2014a). The brazed inner surface was fairied by polishing. Strictly speaking, the measured data were not assumed as the exact data on the pipe inner surface due to the distance from the inner surface. Hence, the measured temperature data were converted to the temperature on the pipe inner surface using the transfer function (Miyoshi, et al., 2014b) developed based on the temperature attenuation and phase delay. The detailed method was described in Sec. 2.3.1. Figure 5 shows 148 measurement points for wall temperature. The interval in the axial direction was 25 mm in the range from \( z = -50 \) to 225 mm. The interval in the circumferential direction was 5° in the range from \( \theta = 0^\circ \) to 60°. Two thermocouples were installed at \( z = -150, 600 \) mm and \( \theta = 0^\circ \). Here, \( z \) is the position in the axial direction of the main pipe and \( \theta \) is the azimuthal angle in the main pipe shown in Fig. 2.

Data from the thermocouples were recorded into a data logger (KYOWA, USB-500A, UCAM-500B), which had an A/D converter in each channel and allowed simultaneous sampling of multiple channels. After the beginning of the experiment, it was checked that the temperatures reached quasi-steady state at all measurement points, and then the statistical temperature values (time-averaged value and fluctuation intensity) were obtained during 160 s. Sampling frequency was 50 Hz. This was sufficient to measure the temperature fluctuation in the T-junction, because it has been reported that the dominant frequency was at most several Hz under the experimental condition in Table 1 (Miyoshi, et al., 2014b). The time difference between channels was 12.4 μs and negligible compared with the sampling period (0.02 s). The calibration formulas for all the thermocouples were derived with the platinum resistance thermometer under the static water condition to reduce the error of measurement. The errors were less than 0.12 °C in fluid temperature and 0.42 °C in the wall temperature.

### Table 1 Experimental Conditions

| Condition [unit] | Main pipe | Branch pipe | Remarks |
|------------------|-----------|-------------|---------|
| Mean velocity [m/s] | 1.0       | 0.65        | Wall jet condition |
| Temperature [°C]  | 25.7      | 59.8        |         |
| Inner diameter [m] | 0.15      | 0.05        |         |
| Reynolds number [-] | 1.7 x 10³ | 7.1 x 10⁴  |         |

Fig. 1 T-Cubic experiment loop which consisted of a hot water tank, a cold water tank, a mixing water tank and two pumps. The hot water from above and the cold water from the side mixed in the test section and flowed out to the mixing water tank.
Fig. 2 The 304 type stainless steel pipe test section of the mixing tee pipe; inner diameter of the main pipe was 150 mm and that of the branch pipe was 50 mm. The hot water in the branch pipe and the cold water in the main pipe mixed at the junction.

Fig. 3 Thermocouple tree to measure fluid temperature distributions. The tree consisted of 16 thermocouples; 15 of them were arranged in a row with the separation distance of 5 mm and the final one was at $r \approx 74$ mm from the center of the pipe. To reinforce the structure of the thermocouples, each of them was inserted and braze to a cover tube, 1 mm in outer diameter and 50 mm in length, with the thermocouple head of 5 mm length remaining exposed. The thermocouple tree could be rotated and moved in the axial direction.

Fig. 4 Sheathed thermocouples (diameter, 0.5 mm) were installed within the inner surface. Grooves 0.7 mm deep and 0.6 mm wide were made in the pipe wall. All the thermocouples were installed in these grooves and brazed by Ni.
Fig. 5 Measurement points for wall temperature are located: The interval in the axial direction was 25 mm in the range from \( z = -50 \) to 225 mm; the interval in the circumferential direction was 5° in the range from \( \theta = 0^\circ \) to 60°; and two thermocouples were installed at \( z = -150, 600 \) mm where \( \theta = 0^\circ \).

2.3 Data processing method

2.3.1 Data processing of the temperature on the pipe inner surface

The data measured with the thermocouples were not assumed as the exact data on the inner surface, because the distance between the thermal contact point and the inner surface was 0.45 mm as shown in the right photo of Fig. 4. Hence heat conduction analysis was conducted in order to estimate the attenuation of amplitude and the delay of phase between the inner surface and the measuring point. The temperature fluctuation at the measuring point was simulated by giving the sinusoidal temperature to the inner surface. The material property of the sheathed thermocouple including the insulator was also considered in the heat conduction analysis. Some kinds of frequencies of temperature fluctuation on the inner surface were simulated in order to obtain the transfer functions which are consisted of the attenuation of amplitude and the delay of phase. The steps to estimate the temperature on the pipe inner surface using the measured data and the transfer function were as follows.

1. Measure the temperature fluctuations with the thermocouples.
2. Convert the obtained data to the frequency response by Fast Fourier Transform (FFT) analysis.
3. Prepare the transfer function which estimates the data on the inner surface from the measured data, by heat conduction analysis.
4. Calculate the amplitude and the phase on the inner surface from the frequency response with the transfer function.
5. Convert the above data to the time response by inverse-FFT analysis.

The transfer function obtained in step (3) was as follows:

\[
a = \exp(-0.236 f^{0.628})
\]

(1)

\[
\varphi = 0.355 f^{0.602}
\]

(2)

where \( a \), \( \varphi \) and \( f \) are the amplitude ratio, the phase delay and the frequency, respectively. The detailed estimation method have been previously reported (Miyoshi, et al., 2014a and 2014b).

2.3.2 Data processing of the fluid temperature

Similar to the above, the thermocouple tree in Fig. 3 had a distance between measurement points and fluid around the sheath. As a result, measured data included errors due to the distance. Hence the same method as described in Sec. 2.3.1 was adopted to estimate the error (Miyoshi et al., 2014c).

The transfer function between fluid and the center of the thermocouple was obtained by heat conduction analysis using the commercial FEM software ABAQUS. The fluctuation frequency of the fluid temperature was treated as a parameter and the heat conduction analysis was conducted. The computational grid was shown in Fig. 6 and the material
properties used in the heat conduction analysis were listed in Table 2. The heat transfer coefficient between the fluid and the surface of the sheath was calculated using heat transfer correlation for a laminar flat plate.

\[
\text{Nu}_x = \frac{h_x x}{\lambda} = 0.332 \text{Re}^{1/2} \text{Pr}^{1/3}
\]  

(3)

Here \( \text{Nu}_x \) is the local Nusselt number at distance \( x \) from the head point of the sheath to the measurement point and \( x = 0.2 \) mm. Physical properties of fluid were values at 40 °C. From Eq.(3), the heat transfer coefficient was \( h_x = 29697 \) W/m²K. The temperature fluctuation of the fluid around the sheath was simulated by giving the sinusoidal wave and the obtained transfer function was as follows:

\[
a = \exp\left(-3.59 \times 10^{-4} f^{1.73}\right)
\]  

(4)

\[
\varphi = 0.0297 f^{0.936}
\]  

(5)

Compared with Eqs. (1) and (2), the damping ratio of the amplitude and the phase delay were small. This was because the diameter of the sheath thermocouple was small (\( D = 0.25 \) mm) and the influence of the heat capacity and the attenuation of the heat transfer also became small. To show the effect of the transfer function, Fig. 7 compares the processed temperature by the transfer function with the measured fluid temperature at \( z = 100 \) mm, \( \theta = 30^\circ \), \( r = 74 \) mm, at which temperature fluctuation intensity was relatively large among the measurement points. As shown in the figure, the fluid temperatures with and without the data processing were almost same with each other and the maximum error during 160 s was 0.10 °C and negligible. Hence, the measured fluid temperature was not processed and raw data was shown hereafter.

| Table 2 Material properties of the thermocouple for the heat conduction analysis |
|----------------------------------|--------|--------|
| Density [kg/m³]                  | SUS304 | MgO*   |
| Thermal conductivity [W/(mK)]    | 16.1   | 1.55   |
| Specific heat [J/(kgK)]          | 501    | 937    |

*a: The case when filling fraction was 95% (air 5% and MgO 95%)

Fig. 6 Computational grid for the heat conduction analysis of the thermocouple using the commercial FEM software ABAQUS. The red colored area denotes the sheath made of SUS304 and the blue colored area denotes the insulation material made of MgO.
Fig. 7 Comparison of the processed fluid temperature by the transfer function with the measured fluid temperature. Measurement point was at $z = 100$ mm, $\theta = 30^\circ$, $r = 74$ mm.

2.4 Velocity measurement

Fluid velocity distribution was measured using a laser Doppler velocimetry (LDV) system (KANOMAX, Model 8739-S). A schematic diagram of the test section was shown in Fig. 8. The test section was made of transparent acrylic resin. The inner diameter of the main pipe was 150 mm and that of the branch pipe was 50 mm. A square shaped water jacket surrounded the pipe to minimize refractions of the laser beam at the pipe boundary. Nylon resin particles were used as tracer particles, with a 4.1 μm mean diameter and 1020 kg/m$^3$ density. The number of samplings was about 30,000. Mean cross-sectional velocities were set to almost the same values as Table 1, 0.98 m/s in the main pipe and 0.66 m/s in the branch pipe, but the fluid temperatures in the main and the branch pipe were room temperature.

Fig. 8 Test section for fluid velocity measurement. The test section was made of transparent acrylic resin. The inner diameter of the main pipe was 150 mm and that of the branch pipe was 50 mm. A square shaped water jacket surrounded the pipe to minimize refractions of the laser beam at the pipe boundary.

2.5 Simulation conditions

The computational domain is shown in Fig. 9. The upstream lengths from the junction were $2D_m$ (main pipe) and $2D_b$ (branch pipe) and the downstream length was $5D_m$. The length of the computational domain was set as much shorter than that of the experiment test section in order to reduce the computational loads. The fluid-structure thermally coupled simulation was carried out; that is, the simulation included thermal conduction in the pipe structure as well as thermal
and flow fields in the water. The pipe thickness of the computational domain was the same as that of the T-Cubic experiment.

Simulation conditions are shown in Table 3. The numerical simulation was carried out using the commercial CFD software FLUENT 15.0. The LES dynamic Smagorinsky-Lilly model (ANSYS Inc., 2013) was selected as a turbulence model based on previous studies (Nakamura et al., 2010, Utanohara et al., 2016). As the boundary conditions at inlets of the main and branch pipes, velocity profiles obtained from the T-Cubic experiment (Miyoshi et al., 2014b) were applied as shown in Fig. 10. It should be noted that the simulation profile looked slightly different from the experimental data, but it was confirmed that the simulated flow rates at the inlet of the main and branch pipes were the same as the experiment values. Velocity fluctuation intensity $u'_{rms} / U_{m}$ was also considered at the inlet using the vortex method (ANSYS Inc., 2013). Here the root mean square (RMS) of the velocity $u'$ is derived from

$$u'_{rms} = \sqrt{\frac{\sum_{i=1}^{N} (u_i - u_{ave})^2}{N}},$$  

(6)

where $u_i$ is instantaneous velocity $u$ at the time step $i$, $u_{ave}$ is the time-averaged velocity and $N$ is the number of sampling. It should be noted that in ANSYS FLUENT the profile of the velocity fluctuation intensity cannot be directly applied as a boundary condition, but the profiles of the turbulence kinetic energy $k$ and turbulence dissipation rate $\varepsilon$ can be applied. Hence the profile of $u'_{rms}$ obtained from the experiment was turned into the profiles of $k$ and $\varepsilon$ using the following equations.

$$k = \frac{3}{2} u'_{rms},$$  

(7)

$$\varepsilon = \rho C_{\mu} \frac{k^3}{\mu_t},$$  

(8)

Here $\rho$ is density, $C_{\mu} (= 0.09)$ is one of the closure coefficients of the standard $k$-$\varepsilon$ turbulence model and $\mu_t$ is the turbulent viscosity. In Eq. (8) $\mu_t$ is unknown, so appropriate values in the fully developed region were obtained from a simulation separately conducted for straight pipes of 150 mm and 50 mm in diameter and $\mu_t = 0.2$ Pa s for the main pipe and $\mu_t = 0.05$ Pa s for the branch pipe. The profile of the velocity fluctuation intensity of the main pipe obtained from the T-Cubic experiment is shown in Fig. 11. The solid line is the simulated velocity fluctuation intensity obtained during 10 to 101 s. Though the simulated profile was smaller than the experimental data profile, the simulation could reproduce the tendency of the experimental profile well.

The wall function was applied for the wall boundary condition. There is a controversy regarding the application of the wall function to the flow field of the mixing tee, but here it was used to reduce the number of computational cells near the wall. The predictive performance of the temperature fluctuation intensity near the wall when using the wall function was confirmed by Tanaka et al. (2010) and the authors’ previous study (Nakamura et al., 2015). In the case of the fluid-structure coupled simulation, application of the wall function might degrade the heat transfer between the fluid and structure. The predictive performance of the heat transfer and the temperature fluctuation on the wall are discussed in Sec. 3.2.2 and 3.3. The wall boundary condition depends on $y^+$ value (ANSYS Inc., 2013) and if $y^+$ is larger than about 10, the log-law is employed and if not Newton’s law of viscosity is employed. The minimum value of the wall $y^+$ was about 5 upstream from the junction and maximum value was about 60 downstream from the junction.

The thermal boundary conditions on the pipe outer surfaces was adiabatic and it was consistent with the experiment because the pipe was covered with heat insulator in the experiment.

The computational grids are shown in Fig. 12. The number of grids was about 0.25 million cells in fluid part (water) and 0.32 million cells in solid part (pipe). The cell thickness of the first layer from the pipe inner surface was 0.5 mm in the fluid part and 0.1 mm in the solid part. The resolution of the solid part was made to be fine enough to be used for structural analysis in the future. The grid convergence of the computational grid in the fluid part has been checked in a previous study for simulations of the WATLON experiment (Utanohara et al., 2016). The grid convergence and uncertainty was estimated using four different resolution grids based on ASME V&V 20 standard (ASME, 2009), and the coarsest grid for which grid convergence was almost attained was selected. For the T-Cubic experiment, the test
section was made to the same size as the WATLON experiment to make a comparison easy between them. In addition, the test conditions of the T-Cubic experiment were almost the same as the WATLON experiment such as momentum ratio and Reynolds numbers in the main and branch pipes. Hence, the grid selected in the previous study was used again and the grid of the solid part was added. The initial condition was the result of the steady state simulation (standard $k$-$\varepsilon$ model) and the LES was continued until the flow field reached the quasi-steady state (10 s) and the statistical values such as time-averaged values and fluctuation intensities were obtained from about 90 s duration (10 s to 101 s).

| Table 3 Simulation conditions |
|--------------------------------|
| **Software** | FLUENT 15.0 |
| Fluid-structure coupled simulation for conjugate heat transfer |
| **Fluid (Water)** |  |
| Density | 991.0 kg/m³ at 42.75°C, Boussinesq approximation |
| Viscosity | Polynomial function |
| Specific heat | Polynomial function of temperature |
| Thermal conductivity |  |
| Thermal expansion coefficient | 3.99 x 10⁻⁴ 1/K at 42.75°C |
| **Solid (type 304 stainless steel)** |  |
| Density | 7916.3 kg/m³ at 42.75°C |
| Specific heat | Polynomial function |
| Thermal conductivity | Polynomial function of temperature |
| **Turbulence model** | LES dynamic (Smagorinsky-Lilly) |
| **Boundary conditions** |  |
| Inlet |  |
| Main pipe | $U_m = 1.0$ m/s, $T_m = 25.7$ °C |
| Branch pipe | $U_b = 0.65$ m/s, $T_b = 59.8$ °C |
| Velocity profile | Experimental data |
| Velocity fluctuation | Vortex method with turbulence intensity profile |
| Temperature profile | Uniform |
| Outlet | Pressure-outlet |
| Pipe inner surface |  |
| Momentum | Non-slip or log-law (depending on $y^+$) |
| Energy | Fourier's law or log-law (depending on $y^+$) |
| Pipe outer surface | Adiabatic |
| **Time** |  |
| Initial condition | Result of std. $k$-$\varepsilon$ |
| Statistics | 10 to 101 s |
| **Grid** |  |
| Fluid | 247,444 cells |
| Solid | 321,120 cells |
| Wall $y^+$ | Less than 60 |
Fig. 9 Computational domain of a mixing tee pipe. The upstream lengths from the junction were $2D_m$ (main pipe) and $2D_b$ (branch pipe) and the downstream length was $5D_m$. The pipe thickness was included in the computational domain and the fluid-structure thermally coupled simulation was carried out. The $x$, $y$ and $z$-coordinates were defined to be the span-wise, vertical and stream-wise directions, respectively, and the origin of the coordinate system was set at the crossing point of the main and branch pipe axes.

Fig. 10 The profile of the time-averaged flow-directional velocity $w_{ave}$ on the inlet boundary of the main pipe. The solid line is the boundary condition used that was obtained from the T-Cubic experiment. Data of the T-Cubic experiment (Miyoshi et al., 2014b) are also shown. It should be noted that the simulation profile looked slightly different from the experimental data, but it was confirmed that the simulated flow rates at the inlet of main and branch pipes were the same as the experiment values.
Fig. 11 The profile of the flow-directional velocity fluctuation intensity $w'_{\text{rms}} / U_m$ on the inlet boundary of the main pipe. Circles denote the experimental data of the velocity fluctuation intensity (Miyoshi et al., 2014b) and the solid line denotes the simulation results obtained during 10 to 101 s.

Fig. 12 Computational grids on pipe cross sections. The number of grids was about 0.25 million cells in the fluid part and 0.32 million cells in the solid part. The cell thickness of the first layer from the pipe inner surface was 0.5 mm in the fluid part and 0.1 mm in the solid part. The resolution of the solid part was made to be fine enough to be used for structural analysis.
3. Results and discussion

The \( x \), \( y \) and \( z \)-coordinates were defined to be the span-wise, vertical and stream-wise directions, respectively, as shown in Fig. 9. The origin of the coordinate system was set at the crossing point of the main and branch pipe axes.

3.1 Velocity distribution in fluid
3.1.1 Time-averaged velocity

Time-averaged velocity distribution is shown in Fig. 13 on the cross section vertically parallel to the flow direction. The plotted value is the time-averaged scalar velocity \( U_{\text{ave}} \) and normalized by the inlet velocity of the main pipe \( U_m \).

\[
U_{\text{ave}} = \left( \sqrt{u^2 + v^2 + w^2} \right)_{\text{ave}}
\]  

Here \( u \), \( v \) and \( w \) are the \( x \), \( y \) and \( z \) component of the flow velocity. The tendency of the distribution was almost the same as the previously known results (Utanohara et al., 2016), that is, main and branch flows with fully developed profiles entered and mixed in the junction and a stagnant region appeared on the branch-side wall just after the junction. Velocity increased locally after the junction due to the inflow from the branch pipe.

Fig. 14 shows the profile of the time-averaged stream-wise velocity \( w_{\text{ave}} \) along the vertical \( y \) direction. The figure compares the simulated profiles with experimental data. At 0.5 \( D_m \) downstream from the junction (Fig. 14 (a)), backflow appeared near the wall (\( y/R_m < -0.6 \), where \( R_m = D_m/2 \)) and the velocity increased toward the pipe center. At 1.0 \( D_m \) downstream (Fig. 14 (b)), the backflow region disappeared. Simulation results could reproduce the velocity profile of the T-Cubic experiment, particularly in the backflow region.

Fig. 13 Simulation results of the time-averaged velocity distribution on the cross section vertically parallel to the flow direction. A stagnant region appeared on the branch-side wall just after the junction.
3.1.2 Velocity fluctuation intensity

Fig. 15 shows simulation results of the flow-directional velocity fluctuation intensity $w'_{\text{rms}} / U_m$. The simulated velocity fluctuation increased in the stagnant region, particularly near the interface of the main and branch flows. This was because a free shear layer formed due to the velocity difference between main and branch flows. For a more quantitative comparison, simulated velocity fluctuation intensities were compared with those of experimental data in Fig. 16. At 0.5 $D_m$ downstream from the junction (Fig. 16 (a)), although the simulation could reproduce the trend of the experimental data, the predicted value was relatively smaller than the experimental data, in particular, 13.8% smaller at the peak ($y/R_m = -0.5$). The predicted value at 1.0 $D_m$ downstream relatively agreed with the experimental value, but the predicted values were smaller than the experimental data for the main flow side ($y/R_m > 0$).

The authors had previously examined the influence of the inlet boundary condition on the flow field of the T-junction (Utanohara et al., 2015) and it was found that the velocity fluctuation intensity on the inlet boundary was necessary for the accurate prediction of the velocity fluctuation intensity downstream from the T-junction. The present study also considered the velocity fluctuation intensity on the inlet boundary as shown in Fig. 11. Hence the reason of the discrepancy in Fig. 16 seemed not to be the influence of the inlet boundary condition and another factor of the simulation method might cause the discrepancy. As mentioned above, though the predicted value was relatively smaller than the experimental data for the main flow side, predictive performance was sufficiently good near the junction.

Fig. 15 Simulation results of the distribution of the flow-directional velocity fluctuation intensity $w'_{\text{rms}} / U_m$ on the cross section vertically parallel to the flow direction.
3.2 Temperature distribution in fluid

3.2.1 Time-averaged temperature

Fig. 17 shows time-averaged fluid temperature distributions. Experimental data (Fig. 17 (a)) were obtained by the thermocouple tree shown in Fig. 3. They were the distributions on cross sections at \( z = 0.17D_m, 0.5D_m \) and \( 1.0D_m \). The angle of the thermocouple tree was changed from 0° to 60° (15° to 60° at \( z = 0.17D_m \)) with 5° pitch in each cross section and there were 208 data points in each cross section (160 points at \( z = 0.17D_m \)). The experimental and simulation temperatures in Fig.12 were normalized values of \( (T - T_m) / (T_b - T_m) \). Hot flow from the branch pipe placed on the branch-side wall (\( y < -30 \) mm) and temperature gradually decreased along the flow direction due to mixing with cold flow in the main pipe. In the numerical simulation this tendency was well simulated as shown in Fig. 17 (b). For a more quantitative comparison, Fig. 18 compares the simulated time-averaged temperature profile with the experimental data profile. The profiles were along the radial segment 30° inclined from the vertical \( y \) axis, as shown in the figure. The simulation results could reproduce the experimental data well.
Fig. 17 Time-averaged fluid temperature distribution at each cross section downstream from the junction. Experimental data were from 208 data points measured by the thermocouple tree (Fig. 3). These temperatures were normalized values of $(T - T_m) / (T_b - T_m)$.

![Diagram](image-url)

(a) T-Cubic experimental data

(b) Simulation results

Fig. 18 Profiles of the time-averaged fluid temperature along the radial direction by $30^\circ$ from the vertical $y$ axis. Symbols denote the experimental data and lines denote the simulation results. The $x$ axis shows normalized values of $(T - T_m) / (T_b - T_m)$.

![Diagram](image-url)

(a) $z = 0.17D_m, 0.33D_m, 0.5D_m$

(b) $z = 0.67D_m, 0.83D_m, 1.0D_m$

3.2.2 Temperature fluctuation intensity

The distributions of temperature fluctuation intensity $T_{rms}^*$ are shown in Fig. 19 and Fig. 20. $T_{rms}^*$ is a non-dimensional value and derived from

![Diagram](image-url)
\[ T_{\text{rms}}^* = \frac{T_{\text{rms}}}{T_b - T_m}, \]  
(10)

\[ T_{\text{rms}} = \sqrt{\frac{\sum_{i=1}^{N} (T_i - T_{\text{ave}})^2}{N}}, \]  
(11)

Here \( T_i \) and \( T_{\text{ave}} \) are instantaneous and time-averaged temperatures, respectively. The region where \( T_{\text{rms}} \) was large was similar to that of the velocity fluctuation intensity (Fig. 15) and it was the interface of the hot and cold flows. This can be understood more clearly from Fig. 20 which shows distributions of \( T_{\text{rms}}^* \) at each cross section downstream from the junction. The region where \( T_{\text{rms}}^* \) was large covered the hot branch jet. The distribution of \( T_{\text{rms}}^* \) was almost the same as that of the simulation results of the WATLON experiment (Kamide et al., 2009) despite the fact that the setting of cold main and hot branch flows in the T-Cubic experiment was opposite to that in the WATLON experiment, that is, hot main and cold branch flows. Since the momentum ratio defined by Kamide et al. (2009) was the same between the WATLON and the T-Cubic experiments, the temperature fluctuation might also become the same distribution.

Fig. 20 compares the simulation results of \( T_{\text{rms}}^* \) with experimental data. Although the simulation could reproduce the distributions, the values were larger than the experimental data. For a more quantitative comparison, Fig. 21 compares the simulated \( T_{\text{rms}}^* \) with experimental data along the radial direction 30° inclined from the vertical \( y \) direction, analogous to Fig. 18. Though the tendency could be reproduced by the simulation, the simulated profile was larger than the experimental data profile, and in particular, 50% larger near the wall at \( z = 0.5D_m \) (Fig. 21(a)) and 57% larger near the wall at \( z = 1.0D_m \) (Fig. 21(b)). The reason for these overestimations is not clear at this stage and future work will need to improve the simulation.

Fig. 19 Simulation results of the distributions of the fluid temperature fluctuation intensity on the cross section vertically parallel to the flow direction.
Fig. 20 Distributions of the fluid temperature fluctuation intensity at each cross section downstream from the junction. Experimental data were from 208 data points measured by the thermocouple tree (Fig. 3). These temperature fluctuations are normalized values of $T_{\text{rms}} / (T_b - T_m)$.

Fig. 21 Profiles of the temperature fluctuation intensity along the radial direction by 30° from the vertical $y$ axis. Symbols denote the experimental data and lines denote the simulation results.

3.3 Temperature distribution on the pipe inner surface

3.3.1 Attenuation of the temperature fluctuation

The contours of instantaneous temperature near the wall are shown in Fig. 22. Fig. 22 (a) plots the fluid temperature 1 mm from the wall and Fig. 22 (b) plots the temperature on the pipe inner surface. These are overhead views, that is, they were directly viewed from the fluid to the pipe wall. Horizontal solid lines in the figures are 30° pitches and drawn...
to understand the spread width of hot fluid from the branch pipe. The hot flow from the branch pipe (hereafter called a hot spot) covered the region of about ±30° (until about 1.0Dm) and gradually spread wider up to ±90° (about 3.0Dm). Fluid temperature near the wall fluctuated strongly compared with the temperature on the pipe inner surface (Fig. 22 (b)). It was revealed by previous studies (Kamide et al., 2009; Nakamura et al., 2009; Tanaka et al., 2010) that the prominent fluctuation downstream from the junction was caused by horseshoe vortices and the fluctuation frequency was almost the same as that of the Karman vortex street, St = 0.2, where St is Strouhal number as given by

\[
St = \frac{fD_n}{U_m},
\]

(12)

and f is the vortex shedding frequency. A fluctuation like the Karman vortex street exists in Fig. 22 (a) and the similar thermal flow field to that of previous studies (Kamide et al., 2009, Miyoshi et al., 2014a) was able to be simulated.

On the other hand, the fluctuation on the pipe inner surface (Fig. 22 (b)) was not as strong as the fluctuation of the fluid temperature and the distribution looked like the time-averaged value. This indicated that relatively high-frequency fluctuations of the fluid temperature attenuated during the heat transfer and low-frequency fluctuations remained on the pipe inner surface. A more detailed discussion is made in Sec. 3.3.3.

![Instantaneous temperature distributions](image)

Fig. 22 Instantaneous temperature distributions (a) near the wall and (b) on the pipe inner surface. These are overhead views, and observed from the fluid to the pipe wall. Horizontal solid lines denote 30-degree pitches to allow understanding of the spread width of the hot fluid from the branch pipe. Gray colored regions in the upper and lower ends denote the pipe wall.

### 3.3.2 Predictive performance of the wall surface temperature

Profiles of the time-averaged temperature on the pipe inner surface are shown in Fig. 23. The horizontal axis denotes
the circumferential angle as defined in Fig. 9. Time-averaged temperature was the highest at 0°, on the symmetry line, and decreased as the angle increased. The simulation could reproduce the tendency of the experimental data, but the peak value of the simulation was 18% lower at 0°, \( z = 0.5D_m \) and the gradient was gentler than that of the experimental data.

The contours of the temperature fluctuation intensity on the pipe inner surface are shown in Fig. 24 (a) for the T-Cubic experimental data (Miyoshi et al., 2014b) and in Fig. 24 (b) for the simulation results. In the T-Cubic experiment, temperature fluctuation was large at around \( \theta = 20° \) to 30° and from the outlet of the branch pipe to \( z = 1.0D_m \). The qualitative tendency of the distribution was simulated as shown in Fig. 24 (b), but the range of the large fluctuation area was narrower compared with the experimental data. A more quantitative comparison was shown in Fig. 25. The horizontal axis denotes the circumferential angle as defined in Fig. 9. The simulation could reproduce the tendency of \( T_{\text{rms}^*} \), though the simulation results were larger than the experimental data. In Fig. 9 (a) (\( z = 0.17D_m \)) the peak value appeared at around 20° and the simulation result was 36% larger than the experimental one. In Fig. 9 (b) (\( z = 0.5D_m \)) the simulation result at 0° was about twice as high as the experimental one and the peak value at about 30° was 20% larger.

As described above, though the numerical simulation could reproduce the flow and temperature field to a certain extent, simulation results had some discrepancies with experimental data, particularly \( T_{\text{rms}^*} \) on the pipe inner surface. \( T_{\text{rms}^*} \) on the pipe inner surface is the most important parameter in this study because it directly causes the thermal stress fluctuation. The overestimation of \( T_{\text{rms}^*} \) on the pipe inner surface seemed to be related to \( T_{\text{rms}^*} \) in fluid shown in Fig. 21, in which the simulation value was 50% larger than the experimental data near the wall at \( z = 0.5D_m \). Compared with \( T_{\text{rms}^*} \) in fluid, the simulation value of \( T_{\text{rms}^*} \) on the pipe inner surface was decreased and 20% larger than the experimental data at \( z = 0.5D_m \). The possible reason for this difference was the use of the wall function as the wall boundary condition on the pipe inner surface. Originally, the wall function was used on the assumption of fully developed flow, but the velocity and temperature boundary layers downstream from the T-junction seemed not to be developed due to flow separation and mixing. As a result, temperature gradient near the wall might become sharper than that of the developed flow and heat transfer might also increase. However, the use of the wall function could not represent such a shaper temperature gradient near the wall. Hence, the wall function might reduce the temperature fluctuation intensity during the heat transfer process between fluid and solid. In any case, improvement of the predictive performance of \( T_{\text{rms}^*} \) in fluid should be addressed in the future. Moreover, the applicability of the wall function on the pipe inner surface should be investigated.

Previously, predictive performance of the temperature distribution on the pipe inner surface could not be verified due to the lack of experimental data. In this study, owing to availability of the T-Cubic experimental data, the predictive performance on the pipe inner surface could be confirmed. Though there remain some problems, a good prospect for development of an evaluation method for thermal fatigue using CFD was obtained. The simulated temperature distribution in the pipe structure can be used as the input data for the thermal stress analysis. The authors plan to carry out a finite element analysis using the simulated temperature distribution obtained in this study and to compare simulated thermal stresses with experimental data (Miyoshi et al., 2016).

![Fig. 23 Profiles of the time-averaged temperature on the pipe inner surface.](image)

Fig. 23 Profiles of the time-averaged temperature on the pipe inner surface. They are normalized values of \( (T - T_w) / (T_b - T_w) \). Horizontal axes denote the circumferential angle as defined in Fig. 9. Circles denote the experimental data and lines denote the simulation results.
Fig. 24 Temperature fluctuation intensity $T_{rms}^*$ distributions on the wall. (a) Experimental results were obtained by Miyoshi et al. (2014b). (b) Simulation results are shown as an overhead view, and observed from the fluid to the pipe wall. The x marks in (b) denote monitoring points of Fig. 26.

![Diagram of temperature fluctuation intensity](image)

Fig. 25 Profiles of the temperature fluctuation intensity $T_{rms}^*$ on the pipe inner surface. Horizontal axes denote the circumferential angle as defined in Fig. 9. Circles denote the experimental data and lines denote the simulation results.

![Profile of temperature fluctuation intensity](image)

3.3.3 Temperature fluctuation frequency

The large temperature fluctuation appeared around $\theta = 20^\circ$ to $30^\circ$. To analyze frequency characteristics of the temperature fluctuation in the numerical simulation, the power spectrum density (PSD) of the temperature was analyzed as shown in Fig. 26. The figure plots the PSD of the fluid temperature (3 mm from the wall) and the wall surface temperature monitored at two x-marked points in Fig. 24 (b), $z = 0.5D_m$ and $0.83D_m, \theta = 25^\circ$. The simulated temperatures were monitored every time step (5 kHz) during 10 to 101 s (91 s) and then 45,500 sampling data were obtained. To
clarify the peak frequency, the PSD was averaged by the following method. First, the 38 sets of $2^{38} = 262,144$ data (52.4 s duration) were extracted from 455,000 sampling data at intervals of 5000 data (1 s). Then the FFT was done for the $2^{38}$ extracted data to get each PSD as 38 FFT results, and these 38 results were averaged as the final PSD (Fig. 26).

Fig. 26 shows that PSD of the temperature fluctuation on the pipe inner surface was attenuated compared with the PSD of the fluid temperature fluctuation. The degree of the attenuation became larger as frequency was higher and fluctuations in the low frequency region did not attenuate as much as the high frequency region. At $z = 0.5D_m$ (Fig. 26 (a)) the fluid temperature fluctuation from 0.1 Hz to 10 Hz did not have any characteristic frequencies and the value was in the order of 1.0 °C²/Hz, but the frequencies in the low frequency region (about 0.2 Hz) remained on the pipe inner surface because of the attenuation of the high frequency fluctuations. At $z = 0.83D_m$ (Fig. 26 (b)) the fluid temperature fluctuation had a peak frequency at about 4 Hz, which corresponded to the fluctuation of $St = 0.2$ as described in Sec. 3.3.1. The fluctuation of $St = 0.2$ also remained on the pipe inner surface but attenuated to the weak value and the fluctuation in the low frequency region (about 0.3 Hz) remained more strongly.

Fig. 26 also shows the experimental data obtained in the T-Cubic experiment. The numerical simulation reproduced the PSD of the temperature fluctuation on the pipe inner surface to a certain extent and it indicated the predictive performance of this simulation. The peak frequency of $St = 0.2$, however, appeared at $z = 0.5D_m$ and $0.83D_m$ in the experiment but appeared only at $z = 0.83D_m$ in the numerical simulation. According to Miyoshi et al. (2014b), the frequency of $St = 0.2$ appeared for $z > 0.5D_m$ downstream from the junction in the experiment and this tendency was different in the numerical simulation.

As described above, the fluid temperature in the numerical simulation fluctuated at almost the same level (in the order of 1.0 °C²/Hz) from 0.1 Hz to 10 Hz, but high frequency components attenuated and low frequency components around 0.1 Hz remained on the pipe inner surface. It needs to be confirmed in the future if the experiment also follows this tendency. Additionally, the low frequency and long period fluctuation around 0.1 Hz was observed experimentally (Miyoshi et al., 2014b) and numerically (Utanohara et al., 2016), but the mechanism of the fluctuation is still not clear. This point also should be addressed as future work.

![Fig. 26](image-url)

**Fig. 26** Power spectrum density of the temperature fluctuation near the wall. Black and red lines are the simulation results of the fluid and pipe inner surface, respectively. Fluid temperature was measured at 3mm from the wall. Blue dash lines are the experimental results on the pipe inner surface.

4. Conclusion

The fluid-structure coupled simulation for conjugate heat transfer was carried out to reproduce a mixing tee experiment called the T-Cubic experiment and the predictive performance was investigated not only about flow and temperature fields in the fluid but also the temperature distribution in the solid structure.

In the fluid part, time-averaged velocity and temperature were reproduced well. Although the simulation result of velocity fluctuation intensity at a distance from the wall was relatively smaller than experimental data, the simulation could reproduce the trend of the experimental data, especially with respect to the peak near the wall. The temperature fluctuation intensity could be reproduced by the simulation, though the temperature fluctuation intensity was larger than
the experimental data.

On the pipe inner surface, time-averaged temperature and temperature fluctuation intensity could be reproduced by the numerical simulation to a certain extent. The temperature fluctuation intensity on the pipe inner surface is the most important parameter in this study for thermal fatigue assessment and the peak values at the cross sections were larger than experimental data by 36% ($z = 0.17D_m$) and 20% ($z = 0.5D_m$) and the range of the large fluctuation area was narrower compared with the experimental data. Owing to availability of the T-Cubic experimental data, the predictive performance of the temperature fluctuation intensity on the pipe inner surface could be confirmed for the first time.

In the numerical simulation, the fluid temperature near the wall fluctuated strongly compared with the temperature on the pipe inner surface. This indicated that relatively high-frequency fluctuations of fluid temperature attenuated during the heat transfer and low-frequency fluctuations remained on the pipe inner surface. The numerical simulation reproduced the PSD of the temperature fluctuation on the pipe inner surface to a certain extent. The numerical simulation indicated the important attenuation characteristics of the temperature fluctuation during heat transfer. Specifically, the fluid temperature in the numerical simulation fluctuated at almost the same level from 0.1 Hz to 10 Hz, but high frequency components attenuated and low frequency components around 0.1 Hz remained on the pipe inner surface. The mechanism of the fluctuation around 0.1 Hz is still not clear and should be addressed as future work.

References

ANSYS Inc., ANSYS FLUENT Theory Guide, Release 15.0 (2013), Chap. 4.14.2, Standard Wall Functions, ANSYS, Inc.

ASME (American Society of Mechanical Engineers), Standard for verification and validation in computational fluid dynamics and heat transfer. ASME V&V 20-2009 (2009).

Braillard, O., Quemere, P. and Lorch, V., Thermal fatigue in mixing tees impacted by turbulent flows at large gap of temperature: the FATHER experiment and the numerical simulation, Proceedings of the 15th International Conference on Nuclear Engineering (ICONE 15) (2007), Paper No. ICONE15-10805.

Chapuliot, S., Gourdin, C., Payen, T., Magnaud, J.P. and Monavon, A., Hydro-thermal-mechanical analysis of thermal fatigue in a mixing tee, Nuclear Engineering and Design, Vol. 235, Issue 5 (2005), pp. 575-596.

Japan Society of Mechanical Engineers, Guideline for Evaluation of High-Cycle Thermal Fatigue of a Pipe, JSME S 017-2003, JSME (2003) (in Japanese).

Kamaya, M., Utanohara, Y. and Nakamura, A., Thermal fatigue analysis at a mixing tee by a fluid-structural simulation, Proceeding of the ASME 2011 Pressure Vessels & Piping Division Conference (ASME-PVP) (2011), Paper No. PVP2011-57585.

Kamide, H., Igarashi, M., Kawashima, S., Kimura N. and Hayashi, K. Study on mixing behavior in a tee piping and numerical analyses for evaluation of thermal striping, Nuclear Engineering and Design, Vol. 239 (2009), pp. 58-67.

Kimura, N., Ono, A., Miyakoshi, H. and Kamide, H., Experimental study on high cycle thermal fatigue in T-junction - effect of local flow velocity on transfer of temperature fluctuation from fluid to structure -, Proceedings of the 13th International Topical Meeting on Nuclear Reactor Thermal Hydraulics (NURETH-13) (2009), Paper No. N13P1169.

Kuschewski, M., Kulenovic, R., Laurien, E., Experimental setup for the investigation of fluid–structure interactions in a T-junction, Nuclear Engineering and Design, Vol. 264 (2013), pp. 223-230.

McDevitt, M., Hoehn, M., Childress, T. and McGill, R., Analysis and impact of recent U.S. thermal fatigue operating experience, Fourth International Conference on Fatigue of Nuclear Reactor Components (2015), No. 27.

Miyoshi, K., Nakamura A. and Takenaka, N., Numerical evaluation of wall temperature measurement method developed to estimate thermal stress at T-junction pipe, Mechanical Engineering Journal (2014a), Vol. 1, No. 2, pp. tep0006.

Miyoshi, K., Nakamura, A. and Utanohara, Y., An investigation of wall temperature characteristics to evaluate thermal fatigue at a T-junction pipe, Mechanical Engineering Journal (2014b), Vol. 1, No. 5, pp. tep0050.

Miyoshi, K., Nakamura A. and Utanohara, Y., An investigation of characteristics of thermal stress caused by fluid temperature fluctuation at a T-junction pipe,” Journal of the Institute of Nuclear Safety System (INSS Journal) (2014c), Vol. 21, pp. 86-98 (in Japanese).

Miyoshi, K., Kamaya, M., Utanohara, Y. and Nakamura, A., An investigation of thermal stress characteristics by wall temperature measurements at a mixing tee, Nuclear Engineering and Design (2016), Vol. 298, pp. 109-120.

Nakamura, A., Oumaya, T. and Takenaka, N., Numerical investigation of thermal striping at a mixing tee using detached eddy simulation, Proceeding of the 13th International Topical Meeting on Nuclear Reactor Thermal Hydraulics
Nakamura, A., Ikeda, H., Qian, S., Tanaka, M. and Kasahara, N., Benchmark simulation of temperature fluctuation using CFD for the evaluation of the thermal load in a T-junction pipe, Proceeding of the Seventh Korea-Japan Symposium on Nuclear Thermal Hydraulics and Safety (NTHAS-7) (2010), Paper No. N7P-0011.

Nakamura, A., Utanohara, Y., Miyoshi, K. and Kasahara, N., Simulation of thermal striping at T-junction pipe using LES with Smagorinsky constants and temperature diffusion schemes, Proceeding of the Experiment al Validation and Application of CFD and CMFD Codes in Nuclear Reactor Technology (CFD4NRS-4) (2012), Paper No.PS2-P2#3.

Nakamura, A., Utanohara, Y., Miyoshi, K. and Kasahara, N., A review of evaluation methods developed for numerical simulation of the temperature fluctuation contributing to thermal fatigue of a T-junction pipe, E-Journal of Advanced Maintenance, Vol. 6, No. 4 (2015), pp.118-130.

Qian, S., Kanamaru, S. and Kasahara, N., High-accuracy analysis methods of fluid temperature fluctuations at T-junctions for thermal fatigue evaluation, Proceedings of the ASME 2012 Pressure Vessels & Piping Division Conference (ASME-PVP) (2012), Paper No. PVP2012-78159.

Qian, S., Kanamaru, S. and Kasahara, N., High-accuracy CFD prediction methods for fluid and structure temperature fluctuations at T-Junction for thermal fatigue evaluation, Nuclear. Engineering and Design, Vol. 288 (2015), pp. 98-109.

Tanaka, M. and Miyake, Y., Numerical simulation of thermal striping phenomena in a T-junction piping system for fundamental validation and uncertainty quantification by GCI estimation, Mechanical Engineering Journal, Vol. 2, No. 5 (2015), p. 15-00134.

Tanaka, M., Ohshima, H. and Monji, H., Thermal mixing in T-junction piping system concerned with high-cycle thermal fatigue in structure, Journal of Nuclear Science and Technology, Vol. 47, No. 9 (2010), pp.790-801.

Utanohara, Y., Nakamura, A., Miyoshi, K. and Kasahara, N., Numerical simulations of thermal striping at T-junction pipe - 3rd report: The effects of inlet boundary conditions on temperature fluctuation intensity -, Journal of the Institute of Nuclear Safety System (INSS Journal) (2015), Vol. 22, pp. 71-83 (in Japanese).

Utanohara, Y., Nakamura, A., Miyoshi, K. and Kasahara, N., Numerical simulation of long-period fluid temperature fluctuation at a mixing tee for the thermal fatigue problem, Nuclear Engineering and Design, Vol. 305 (2016), pp. 639-652.

Zhou, M., Kulenovic, R., Laurien, E., Kammerer, M. C., Schuler, X., Thermocouple measurements to investigate the thermal fatigue of a cyclic thermal mixing process near a dissimilar weld seam, Nuclear Engineering and Design, Vol. 320 (2017), pp. 77-87.