Numerical simulation of thermal stress fluctuation at a mixing tee for thermal fatigue problems

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Abstract
Thermal fatigue cracks have been found at mixing tees in nuclear power plants. The mixing flow of high and low temperature fluids causes temperature and stress fluctuations in the pipe wall and these result in fatigue crack initiation. The authors have conducted a fluid-structure coupled simulation to estimate the fluid and pipe wall temperatures in a mixing tee in their previous study. In the present study, the authors simulated thermal stress using the previous simulation results of the pipe wall temperature. The simulated thermal stress was validated using the stress obtained from the temperature on the pipe inner surface measured by mock-up tests. The test section of the tee pipe was made of stainless steel and consisted of a horizontal main pipe with a diameter of 150 mm and a T-junction connected to a vertical pipe with a diameter of 50 mm. The ranges of the large temperature and stress fluctuation areas on the pipe inner surface calculated by the fluid-structure coupled simulation were narrower in the axial direction of main pipe compared with the results of test. On the other hand, the profiles of the circumferential direction were reproduced by the fluid-structure coupled simulation. The maximum values of the temperature and stress fluctuation ranges were overestimated. The stress fluctuation obtained from the measured temperature showed the equi-biaxial behavior where the axial and circumferential stresses had a proportional relationship. Such characteristics of stress fluctuation were reproduced well by the numerical simulation. Not only the stress fluctuation range but also the number of cycles for the stress amplitude were estimated using the time history of the thermal stress and the rain-flow counting method. The distribution of the number of cycles for the stress amplitude estimated by the simulation was similar to that obtained from the measured temperatures.

Keywords: Thermal fatigue, Mixing tee pipe, Temperature fluctuation, Thermal stress, Numerical simulation

1. Introduction

Fatigue cracks have been found at mixing tees in nuclear power plants (Chapuliot, et al., 2005; McDevitt, et al., 2015). The mixing of fluids with different temperatures causes temperature fluctuations in the pipe wall and fatigue cracks are initiated (this is referred to as thermal striping). The degree of fatigue damage at the mixing tee depends not only on the temperature difference of the mixing fluids but also on piping geometry and flow conditions (Kawamura, et al., 2003; Noguchi, et al., 2003; Wakamatsu, et al., 2003). The Japan Society of Mechanical Engineers (JSME) has issued a guideline for assessing the fatigue damage at mixing tees (JSME, 2003). The possibility of fatigue crack initiation is predicted using flow and pipe conditions. The JSME guideline is, however, based on mock-up tests for several piping configurations and flow conditions. Therefore, the applicable conditions are limited to the scope of the test data. Since it is expensive to conduct additional tests to extend the applicable scope, a numerical simulation method should be developed to predict fatigue damage for various conditions. In addition, it has been reported that the assessment procedure of the JSME guideline is conservative (Kasahara, et al., 2014), because the evaluation method ignores the effect of the heat transfer attenuation due to the frequency of the temperature fluctuations in order to simplify the procedure. The numerical simulation method would be helpful to quantify the conservativeness and estimate the appropriate thermal stress.
Some numerical simulation studies for mixing phenomena in a tee pipe have been performed (Höhne, 2014; Howard and Pasutto, 2009; Nakamura, et al., 2009, 2015; Qian, et al., 2012, 2015; Tanaka, et al., 2010; Tanaka and Miyake, 2015; Utanohara, et al., 2016). A comparison of fluid and wall temperatures between the numerical simulations and the test results has been discussed. It is difficult to simulate the wall temperature fluctuations accurately due to the complexity of the fluid phenomena. Kamaya and Nakamura (2011) simulated thermal stress distributions using the results of the fluid-structure coupled numerical simulations which were conducted by Nakamura et al. (2009). The simulation results of wall temperature and thermal stress, however, have not been validated by test data sufficiently.

A consistent numerical simulation method, which can predict thermal stress in the pipe wall using inlet conditions of fluid, has to be developed in order to address thermal fatigue problems at a mixing tee. However, such a numerical simulation method has never been developed and validated by test data. One of the difficulties in the development and validation is that reliable thermal stress have not been obtained for the mixing tee. The authors have developed a novel test section, which has 148 thermocouples to measure the temperature distribution at the pipe inner surface (Miyoshi, et al., 2014a). The characteristics of the thermal stress that caused fatigue damage were then discussed using measured temperatures (Miyoshi, et al., 2016). In this study, the thermal stress obtained by the fluid-structure coupled simulation was validated using the wall temperature identified using the wall temperatures measured by mock-up tests. First, the thermal stress was estimated from the wall temperature measured by the test in order to obtain the validation data. Then, the thermal stress was estimated from the wall temperature obtained by the fluid-structure coupled simulation which was reported in the authors’ previous paper (Utanohara, et al., 2018). Finally, the validity of the fluid-structure coupled simulation was assessed by comparison of stress fluctuation characteristics.

2. Estimation of thermal stress using temperature measured by the T-cubic test
2.1 Test loop and test section

The thermal stress in the mixing tee can be estimated from the pipe wall temperature. In this study, the pipe wall temperature was estimated from the temperature on the pipe inner surface measured by the mock-up test called T-Cubic (Miyoshi, et al., 2014a). Figure 1 shows a schematic drawing of the T-Cubic test loop. Hot water was mixed with cold water in the test section. The flow straighteners were set at 18\(D_m\) (\(D_m\) = inner diameter of main pipe) upstream from the inlet of the horizontal pipe and at 56\(D_b\) (\(D_b\) = inner diameter of branch pipe) upstream from the inlet of the vertical pipe. The turbulence caused by elbows and valves upstream was reduced by these flow straighteners. It was confirmed that the measured velocity distributions were close to the fully developed flow distributions (Miyoshi, et al., 2014a).

Figure 2 shows a schematic drawing of the test section for temperature measurement. This test section was made of stainless steel pipes: a horizontal main pipe with an inner diameter of 150 mm and a vertical branch pipe with an inner diameter of 50 mm. The wall thicknesses of the main pipe and the branch pipe were 7.6 mm and 5.3 mm, respectively. The pipe was covered with heat insulator. Sheathed thermocouples (diameter, 0.5 mm) were installed near the inner surface to measure the temperature on the pipe inner surface. A cross-sectional diagram showing the region around a thermocouple installed in the pipe is shown in Fig. 3. The non-grounded type thermocouple was brazed into a groove 0.7 mm deep and 0.6 mm wide on the pipe inner surface. The brazed inner surface was faired by polishing. Figure 4 shows the 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 \(\theta = 0°\) to 60°. Two thermocouples were installed at \(z = -150\), 600 mm and \(\theta = 0°\). Here, \(z\) is the position in the axial direction of the main pipe and \(\theta\) is the azimuthal angle in the main pipe (see Fig. 2). The origin is the intersection of the axes of the main and branch pipes. Since the accuracy of the stress amplitude depends on the interval of the measuring points in the axial and circumferential directions by the thermocouples, the numerical simulation was performed to decide the optimum location of thermocouples. The detailed analysis results have been described previously (Miyoshi et al., 2014b).
Fig. 1 Schematic diagram of T-cubic test loop. The test loop consisted of a hot water tank, a cold water tank, a mixing water tank and two pumps. Hot water from above and cold water from the side mixed in the test section and flowed into the mixing water tank.

Fig. 2 Schematic drawing of test section for temperature measurement. The test section was made of stainless steel pipe; the main pipe had an inner diameter of 150 mm and the branch pipe had an inner diameter of 50 mm. The hot water in the branch pipe and the cold water in the main pipe mixed at the tee pipe.

Fig. 3 Cross-sectional diagram of the region around a thermocouple installed in the pipe wall. A sheathed thermocouple (diameter, 0.5 mm) was installed near the inner surface to measure the temperature on the pipe inner surface. The thermocouple was brazed into a groove 0.7 mm deep and 0.6 mm wide on the pipe.
2.2 Test conditions

Table 1 summarizes test conditions. The mean cross-sectional velocity was the flow rate divided by the cross-sectional area of the pipe. The fluid temperatures in the inlets were measured with the thermocouples which were installed in the upstream pipe of the flow straighteners. Kamide, et al. (2009) reported that flow patterns in the mixing tee are categorized as impinging jet, deflecting jet, and wall jet based on the inlet flow rate ratio. They clarified that the flow pattern of the wall jet, where the jet from the branch pipe was bent by the main pipe flow and made to flow along the pipe wall, caused a higher fluctuation intensity of the fluid temperature near the pipe wall than in other flow patterns. The mean cross-sectional velocity in the inlets was set under the condition of the wall jet flow in this study, because that flow pattern seemed to be the severest for thermal fatigue.

Table 1 Test conditions

|                        | Main pipe | Branch pipe |
|------------------------|-----------|-------------|
| Fluid temperature in the inlet [°C] | 25.7      | 59.8        |
| Mean cross-sectional velocity in the inlet [m/s] | 0.99      | 0.66        |
| Reynolds number [-]    | 1.7×10⁵   | 6.9×10⁴     |

2.3 Stress analysis method

2.3.1 Analysis procedure

Time history of the thermal stress was derived by heat conduction analysis and stress analysis. The commercial finite element analysis code Abaqus (ver.6.14) was employed to conduct the heat conduction and the stress analyses. The mesh division used for the analyses is shown in Fig. 5. The analysis region was a half of the tee pipe, which was separated at the y-z plane. The element widths in the axial direction and the circumferential direction were 12.5 mm and 2.5°, respectively. These widths were smaller than the interval between the measurement points. The mesh thickness near the inner surface was 0.1 mm. This thickness should be small, because the stress fluctuation might be underestimated due to the attenuation of the heat conduction in the thickness direction. The total number of mesh elements was 81,856. The element types of DC3D8 (three dimension and 8 node) and C3D8 (three dimension, 8 node and complete integral) were employed for heat conduction analysis and stress analysis, respectively (Dassault systemes, 2014a). The transient heat conduction was calculated by backward Euler method (Dassault systemes, 2014b). The stress analysis was calculated by linear static analysis procedure (Dassault systemes, 2014c).

2.3.2 Heat conduction analysis conditions

The temperature on the pipe inner surface is needed as the boundary condition in order to conduct the heat conduction
analysis. The data measured by the thermocouples were not assumed as 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. 3. The inner surface temperatures were estimated from the measured temperatures according to the procedure developed previously (Miyoshi, et al., 2014b). As the inner surface boundary condition, the time histories of the temperatures during 160 s obtained by above procedure were given to the nodes at the position of the measurement points shown in Fig. 4. The temperatures at the other mesh nodes on the inner surface of the main pipe were linearly interpolated with the measured data.

The outer boundary of the pipe was adiabatic. The fluid temperature measured in the inlet of the branch pipe was uniformly given to the inner surface of the branch pipe at all times. The time averaged temperature at the position of \( z = -150 \text{ mm}, \theta = 0^\circ \) was set to the initial temperature at all the nodes. The time step of heat conduction analysis was 0.02 s which was the same as the sampling period of the temperature measurement. Material properties used for the heat conduction analysis are listed in Table 2.

### 2.3.3 Stress analysis conditions

The stress was calculated using the temperature in the pipe obtained by the heat conduction analysis. Only the temperature data from 60 s to 160 s were employed for the analysis. The data before 60 s were not used for the stress analysis in order to take the heat conduction from the inner surface to the outer surface into account. After 60 s the temperatures of the outer surface on the main pipe were in a quasi-steady state. The time step of stress analysis was 0.02 s and that was the same as for the heat conduction analysis. The displacement in the \( x \) direction and the rotation around the \( z \)-axis were constrained at the nodes of the symmetry plane. Material properties used for the stress analysis are listed in Table 2.

![Fig. 5 Computational mesh for heat conduction and stress analyses](image)

These analyses were conducted in order to obtain thermal stress from wall temperature measured by tests.

| Table 2 Material properties for the heat conduction and stress analyses |
|---------------------------|---------------------------|
| Density [kg/m³]           | 7920 (at 20°C)*           |
|                           | 7910 (at 50°C)            |
| Young’s modulus [GPa]     | 195 (at 20°C) *           |
|                           | 193 (at 50°C)             |
| Thermal conductivity [W/(mK)] | 16.0 (at 20°C) *   |
|                           | 16.1 (at 50°C)            |
| Poisson’s ratio [-]       | 0.30                      |
| Specific heat [J/(kgK)]   | 498 (at 20°C) *           |
|                           | 502 (at 50°C)             |
| Thermal expansion coefficient [1/K] | 1.52×10⁻⁵ (at 20°C)*      |
|                           | 1.55×10⁻⁵ (at 50°C)       |

*Material properties were given by linearly interpolating values at 20°C and 50°C
3. Estimation of thermal stress by fluid-structure coupled simulation

3.1 Wall temperature obtained by fluid-structure coupled simulation

The authors conducted the fluid-structure coupled simulation for the T-cubic test shown in Fig. 1 and Fig. 2. The wall temperature was simulated for the test conditions shown in Table 1. The results of numerical simulation were validated using the test data and reported elsewhere (Utanohara, et al., 2018). Numerical simulations were carried out using the commercial CFD software FLUENT (ver.15.0). The LES dynamic Smagorinsky-Lilly model (ANSYS, 2009) was selected as a turbulence model. As the boundary conditions at inlets of main and branch pipes, velocity profiles (Miyoshi, et al., 2014a) measured by laser Doppler velocimetry (LDV) were applied. The velocity fluctuations were also considered using the vortex method (ANSYS, 2009) and the velocity fluctuation intensity measured by LDV. The detailed conditions of the fluid-coupled simulation were described elsewhere (Utanohara, et al., 2018). In this study, the wall temperature obtained by this fluid-structure coupled simulation was used for stress analysis.

3.2 Stress analysis method

The commercial finite element analysis code Abaqus (ver.6.14) was employed to conduct the stress analysis. The mesh division used for the stress analysis is shown in Fig. 6. The total number of mesh elements was 321,120. This computational mesh was the same as the structure domain of the fluid-structure coupled simulation. The simulated wall temperatures every 0.02 s were used as the boundary condition of the stress analysis. The wall temperature of 91 s (10 s to 101 s) was employed for the stress analysis, because the wall temperature distribution reached the quasi-steady state after 10 s (Utanohara, et al., 2018). Material properties used for the stress analysis are listed in Table 2. No deformation constraint was considered as the boundary condition. The element type and analysis procedure were the same as those described in Sec. 2.3.1.

![Computational mesh for stress analyses using results of the fluid-structure coupled simulation](image-url)
4. Validation of thermal stress obtained by fluid-structure coupled simulation

4.1 Temperature distribution on the pipe inner surface

Figures 7(a) and (b) show the instantaneous temperature distributions on the pipe inner surface calculated by the fluid-structure coupled simulation. The times of 74.44 s and 87.94 s correspond to the maximum and minimum temperature peaks at the position $z = 75$ mm, $\theta = 30^\circ$ which is indicated by the x-mark. The hot jet flow from the outlet of the branch pipe heated the inner surface of the main pipe. The temperature was relatively high downstream from the T-junction along the line of $\theta = 0^\circ$. These figures also show the size of the hot spot, which was heated by the hot jet flow, changed with time. The instantaneous temperature distributions on the pipe inner surface measured by the T-Cubic test are shown in Fig. 8. The times of 86.28 s and 97.46 s correspond to the minimum and maximum temperature peaks at the position $z = 75$ mm, $\theta = 30^\circ$ which is indicated by the x-mark. The hot spot was observed along the line of $\theta = 0^\circ$ and changed with time. These characteristics were similar to those of the temperature distributions calculated by the fluid-structure coupled simulation.

![Fig. 7 Instantaneous temperature distributions on the pipe inner surface at the times of 74.44 s and 87.94 s (fluid-structure coupled simulation).](image1)

(a) 74.44 s  
(b) 87.94 s

Fig. 7 Instantaneous temperature distributions on the pipe inner surface at the times of 74.44 s and 87.94 s (fluid-structure coupled simulation). The times of 74.44 s and 87.94 s correspond to the maximum and minimum temperature peaks at the position $z = 75$ mm, $\theta = 30^\circ$ which is indicated by the x-mark. The hot jet flow from the outlet of the branch pipe heated the inner surface of the main pipe. The size of the hot spot, which is heated by the hot jet flow, changes with time.

![Fig. 8 Instantaneous temperature distributions on the pipe inner surface at the times of 86.28 s and 97.46 s (T-Cubic test).](image2)

(a) 86.28 s  
(b) 97.46 s

Fig. 8 Instantaneous temperature distributions on the pipe inner surface at the times of 86.28 s and 97.46 s (T-Cubic test). The times of 86.28 s and 97.46 s correspond to the minimum and maximum temperature peaks at the position $z = 75$ mm, $\theta = 30^\circ$ which is indicated by the x-mark. The size of the hot spot, which is heated by the hot jet flow, changes with time.

Figure 9 shows the distribution of the temperature fluctuation range on the pipe inner surface obtained by the fluid-structure coupled simulation, which was defined as the difference between the maximum and the minimum of the time history during 91 s. The root mean square of wall temperature was compared for the validation of the previous numerical simulation (Utanohara, et al., 2018). However, not the root mean square of temperature but temperature fluctuation range is shown here, because fluctuation range is important for fatigue damage assessment. The temperature fluctuation range was relatively large in the region from the outlet of the branch pipe to $z = 75$ mm and along the lines of $\theta = \pm 30^\circ$. The distribution of the temperature fluctuation range during 100 s on the pipe inner surface measured by the T-Cubic test is
shown in Fig. 10. It should be noted that the color ranges are different between Fig. 9 and Fig. 10, because the profiles of distribution become fuzzy and difficult to compare if the color range of Fig. 10 is the same as that of Fig. 9. The temperature fluctuation range was relatively large in the region from the outlet of the branch pipe to \( z = 150 \text{ mm} \) and along the line of \( \theta = 30^\circ \). The simulated temperature fluctuation range was overall larger than that of T-Cubic test. The range of the large fluctuation area was narrower in the \( z \) direction. On the other hand, the profile of the \( \theta \) direction was reproduced by the fluid-structure coupled simulation.

![Fig. 9 Temperature fluctuation range on the pipe inner surface (fluid-structure coupled simulation).](image)

The temperature fluctuation range was relatively large in the region from the outlet of the branch pipe to \( z = 75 \text{ mm} \) and along the lines of \( \theta = \pm 30^\circ \).

![Fig. 10 Temperature fluctuation range on the pipe inner surface (T-Cubic test).](image)

The temperature fluctuation range was relatively large in the region from the outlet of the branch pipe to \( z = 150 \text{ mm} \) and along the line of \( \theta = 30^\circ \).

Figure 11(a) shows the temperature fluctuation range distributions on the pipe inner surface at \( z = 75 \text{ mm} \). The horizontal axis denotes the circumferential angle defined in Fig. 2 and Fig. 6. The temperature fluctuation range measured by the T-Cubic test had the peak at about \( 30^\circ \). The position of the peak was reproduced well by the simulation. The maximum value of the simulation was overestimated by 22% relative to the test data. Figure 11(b) shows the temperature fluctuation range distributions on the pipe inner surface at \( z = 150 \text{ mm} \). Although the temperature fluctuation range measured by the T-Cubic test had the peak at about \( 40^\circ \), the peak was not reproduced by the simulation. On the other hand, the simulated maximum value was close to the maximum value of the test data for \( z = 150 \text{ mm} \).

As described above, the simulated temperature fluctuation range was overall larger than that obtained by the T-Cubic test. Similar trend was observed for the fluctuation intensity (= root mean square), which was reported in the previous paper (Utanohara, et al., 2018). This was due to the overestimation of the fluid temperature fluctuation near the pipe wall. Since the axial range of the large fluctuation area obtained by the simulation was narrower than that obtained by the test, it was deduced that the mixing of hot and cold fluids progressed quickly at upstream. The reason is not clear at this time, but the employed model of the temperature diffusion in LES is one of the possible causes. The predictive performance of fluid temperature fluctuations near the wall should to be improved.
4.2 Instantaneous stress distribution on the pipe inner surface

Figure 12 shows the simulation results of the instantaneous axial stress $\sigma_z$ distributions on the pipe inner surface. The times of 74.44 s and 87.94 s correspond to the minimum and maximum $\sigma_z$ peaks at the position $z = 75$ mm, $\theta = 30^\circ$ which is indicated by the x-mark. The contour figure was colored only in the tensile region of $\sigma_z \geq 0$ for better visualization. The compressive stress region was formed in the hot spot region which was heated by hot jet flow. In contrast, the tensile stress region was formed outside the hot spot region. These figures show the contour line of $\sigma_z = 0$ was moving in the circumferential direction $\theta$ with time. The test results of the instantaneous axial stress $\sigma_z$ distributions on the pipe inner surface are shown in Fig. 13. The times of 86.28 s and 97.44 s correspond to the maximum and minimum $\sigma_z$ peaks at the position $z = 75$ mm, $\theta = 30^\circ$ which is indicated by the x-mark. The compressive stress region was formed in the hot spot region and moved in the circumferential direction $\theta$ with time. These characteristics were similar to those of stress distributions obtained by the fluid-structure coupled simulation.

4.3 Stress fluctuation range distribution on the pipe inner surface

Figure 14 shows the simulation results of the stress fluctuation range distributions on the pipe inner surface. The fluctuation ranges of axial stress $\Delta\sigma_z$ and circumferential stress $\Delta\sigma_\theta$ were defined as the difference between the maximum and the minimum of the time history during 91 s. Stress fluctuation range $\Delta\sigma$ is important for fatigue damage assessment, because the maximum stress amplitude is compared with the fatigue limit of the pipe materials. The stress fluctuation ranges were relatively large in the region from the outlet of the branch pipe to $z = 75$ mm and along the lines of $\theta = \pm 30^\circ$. No significant difference was found between the axial (a) and circumferential (b) stress distributions. In addition, these distributions were similar to that of the temperature fluctuation range shown in Fig. 9.

The test results of the stress fluctuation range distributions during 100 s on the pipe inner surface are shown in Fig. 15. It should be note that the color ranges are different between Fig. 14 and Fig. 15, because the profiles of distribution become fuzzy and difficult to compare if the color ranges of Fig. 15 are the same as those of Fig. 14. The stress fluctuation ranges were relatively large in the region from the outlet of the branch pipe to $z = 150$ mm and along the lines of $\theta = \pm 30^\circ$. No significant difference was found between the axial (a) and circumferential (b) stress distributions. These distributions were also similar to that of the temperature fluctuation range shown in Fig. 10. The simulated stress fluctuation range was overall larger than that of T-Cubic test. The range of the large fluctuation area was narrower in the $z$ direction. On the other hand, the profile of the $\theta$ direction was reproduced by the fluid-structure coupled simulation.

Figure 16 shows the axial and circumferential stress fluctuation range distributions along $\theta$ on the pipe inner surface at $z = 75$ mm. The stress fluctuation ranges $\Delta\sigma_z$, $\Delta\sigma_\theta$ for the T-Cubic test had the peaks at about $30^\circ$. The $\Delta\sigma_z$ and $\Delta\sigma_\theta$
obtained from the results of the fluid-structure coupled simulation reproduced well the peak at about 30°. However, the maximum values of $\Delta \sigma_z$ and $\Delta \sigma_\theta$ at $z = 75$ mm were overestimated by 56% and 57%, respectively.

Figure 17 shows the axial and circumferential stress fluctuation range distributions along $\theta$ on the pipe inner surface at $z = 150$ mm in the circumferential direction. While the $\Delta \sigma_z$, $\Delta \sigma_\theta$ for the T-Cubic test had the peaks at about 40°, similar peaks were not observed in the simulation. The maximum values of $\Delta \sigma_z$ and $\Delta \sigma_\theta$ at $z = 150$ mm were overestimated by 14% and 16% respectively.

As described above, the stress fluctuation range was overestimated by the fluid-structure coupled simulation. These results seem to be caused by the discrepancy of the simulated temperature fluctuation range as described in Sec. 4.1.

4.4 Time history of stress fluctuation on the pipe inner surface

Figure 18 shows time history of the axial and circumferential stress on the pipe inner at the position $z = 75$ mm and $\theta = 30^\circ$. Figures 18(a) and (b) show stresses obtained from the fluid-structure coupled simulation and from the measured data of the T-Cubic test, respectively. These figures show that the biaxial stress was loaded on the pipe inner surface. Fatigue damage under biaxial loading depends on phase difference of each stress component. Therefore, the relationship between the axial and circumferential stresses was confirmed. The correlation between the axial and circumferential stresses is shown in Fig. 19. This figure showed $\sigma_z$ and $\sigma_\theta$ had a proportional relationship. This meant that the stress fluctuations were caused by equibiaxial thermal load. Hence, the axial and circumferential stress fluctuation ranges had similar distributions as shown in Fig. 14 and Fig. 15. The characteristic of the equibiaxial stress fluctuation was reproduced well by the numerical simulation.
Fig. 14 Stress fluctuation range on the pipe inner surface (fluid-structure coupled simulation). The fluctuation ranges of axial stress $\Delta\sigma_z$ and circumferential stress $\Delta\sigma_\theta$ were defined as the difference between the maximum and the minimum of the time history during 91 s. The stress fluctuation ranges were relatively large in the region from the outlet of the branch pipe to $z = 75$ mm and along the lines of $\theta = \pm 30^\circ$.

Fig. 15 Stress fluctuation range on the pipe inner surface (T-Cubic test). The fluctuation ranges of axial stress $\Delta\sigma_z$ and circumferential stress $\Delta\sigma_\theta$ were defined as the difference between the maximum and the minimum of the time history during 100 s. The stress fluctuation ranges were relatively large in the region from the outlet of the branch pipe to $z = 150$ mm and along the lines of $\theta = \pm 30^\circ$. 

Fig. 16 Axial and circumferential stress fluctuation range on the pipe inner surface at $z = 75\,\text{mm}$. The black and red lines denote the simulation and test results, respectively. The $\Delta\sigma_z$ and $\Delta\sigma_\theta$ obtained by the T-Cubic test had the peaks at about $30^\circ$ and that was reproduced well by the simulation. These maximum values were overestimated by the simulation.

Fig. 17 Axial and circumferential stress fluctuation range on the pipe inner surface at $z = 150\,\text{mm}$. The black and red lines denote the simulation and test results, respectively. While the $\Delta\sigma_z$ and $\Delta\sigma_\theta$ obtained by tests had the peaks at about $40^\circ$, similar peaks were not reproduced in the simulation. The maximum values of $\Delta\sigma_z$ and $\Delta\sigma_\theta$ were overestimated by the simulation.

Fig. 18 Time history of stress on the pipe inner surface at the position of $z = 75\,\text{mm}$, $\theta = 30^\circ$. The black and red lines denote the axial stress and circumferential stress, respectively.
4.5 Fatigue damage

In order to predict the thermal fatigue adequately, not only the stress fluctuation but also the fatigue damage should be discussed. The fatigue damage depends not only on the stress amplitude but also on the number of cycles. The fatigue damage is calculated according to the linear damage accumulation rule which is used in the JSME guideline (JSME, 2003). In this rule, it is expected the fatigue failure will occur when accumulated fatigue damage \( U_f \) become 1. The \( U_f \) is calculated by the following equation:

\[
U_f = \sum_i \frac{N_i}{N_{if}}
\]  

(1)

where \( N_i \) and \( N_{if} \) are the number of cycles of the \( i \)th stress amplitude and the number of cycles of fatigue failure obtained by the constant amplitude test for the \( i \)th stress amplitude, respectively. The stress amplitude \( S_a \) and the number of cycles \( N \) are estimated using the rain-flow counting method (ASTM International, 2002) for the time series data of stress. The number of cycles for failure \( N_f \) is estimated using the fatigue life curve for material of pipe.

The T-Cubic test was performed under the temperature difference of 34.1 K in the inlet. Therefore, the estimated stress amplitude was smaller than the fatigue limit \( S_{limit} \) of the stainless steel pipe. Since the \( N_f \) for \( S_a < S_{limit} \) reaches an infinite value, the value of \( U_f \) becomes zero. In this study, only the distribution of \( N_f \) for each stress amplitude was compared. Figure 20 shows the number of cycles for the stress amplitude at the position of \( \theta = 30^\circ \), \( z = 75 \) mm. The stress amplitudes extracted by the rain-flow method were normalized by the stress fluctuation range \( \Delta S_a \), \( \Delta S_\theta \) (= the maximum stress amplitude). The distributions of the number of cycles for the stress amplitude estimated by the numerical simulation which was employed in the authors’ previous paper (Utanohara, et al., 2018) were similar to those obtained by the T-Cubic test.

The number of cycles for the normalized stress amplitude was compared for validation of the numerical simulation. The number of cycles for fatigue failure \( N_f \), however, depends on the stress amplitude. Therefore, the accumulated fatigue damage is also overestimated when the stress amplitude is overestimated by numerical simulations. The stress fluctuation range should be reproduced well in order to reduce the conservativeness of the fatigue life predicted by the numerical simulations.
5. Conclusion

Thermal stress at a mixing tee was calculated using the results of the fluid-structure coupled simulation which has been reported previously by the authors (Utanohara, et al., 2018). The simulated thermal stress was validated using the thermal stress estimated from the wall temperatures measured by the T-Cubic mock-up test (Miyoshi, et al., 2014a). The following conclusions were obtained.

(1) The range of the large temperature fluctuation area on the pipe inner surface calculated by the fluid-structure coupled simulation was narrower in the $z$ direction compared with the result of test. On the other hand, the profile of the $\theta$ direction was reproduced by the fluid-structure coupled simulation. The maximum value of the simulation at $z = 75$ mm was overestimated by 22% relative to the test data.

(2) The range of the large stress fluctuation area on the pipe inner surface calculated using the results of the fluid-structure coupled simulation was narrower in the $z$ direction compared with the result of test. On the other hand, the profile of the $\theta$ direction was reproduced by the fluid-structure coupled simulation. The calculated maximum values at $z = 75$ mm of the axial and the circumferential stress fluctuation ranges were overestimated by 56% and 57% relative to those obtained from test data.

(3) The axial stress and the circumferential stress had a proportional relationship at the position where the stress fluctuation range was relatively large. The characteristic of such equibiaxial stress fluctuation was reproduced well by the numerical simulation.

(4) The number of cycles of the stress amplitude was obtained using the rain-flow counting method. The distribution of the number of cycles for the stress amplitude estimated by the numerical simulation was similar to that obtained from the test data.

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