Optimal Experimental Environment for Flow-induced High Frequency Vibration Test Facility

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1. Introduction

The coolant in a reactor transfers generated heat from the nuclear fuel assemblies to the coolant flow. The average velocity of the coolant is approximately 5 m/sec, and the coolant is affected by severe turbulence, with a value of hundreds of thousands of the Reynolds number. Nuclear fuel rods are exposed to coolant flow, and fuel rod vibration occurs due to this flow. Excessive vibration caused by turbulence induced fluctuating forces is the main source of fretting wear, which is a major cause of mechanical failure for fuel rods [1].

The grid structure in the fuel assembly has a large pressure drop, and disturbs the coolant flow, and increases vibration due to fluid-structure interaction. The incidence of vibration is dependent on the structure of the grid design. It can also be simultaneously affected by vortex and turbulence. Thus it is necessary to investigate the characteristics of grid assembly vibration in the development and design verification phase, because the grid assembly can cause fluid-elastic instability to the fuel assembly.

This paper presents an optimal experimental environment for a flow-induced high frequency vibration test facility.

2. Test facility for flow-induced vibration

Fig. 1 shows the schematic diagram of the test facility. Five 5x5 grid assemblies can be loaded into the test section of the facility, and the maximum flow velocity is up to 12 m/s.

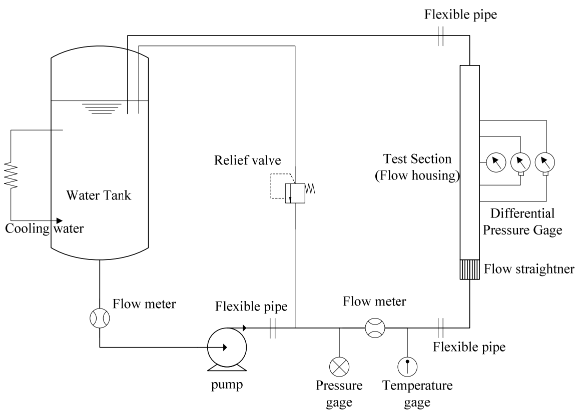

Fig. 1. Schematic diagram of the test facility

Test specimens in the test section are visible because the test section is made from transparent acrylic plates. Pressure taps are machined to measure any pressure drop. The vibration of the grid strap can be measured with an LDV (Laser Doppler Vibrometer).

If the vibration of the test section housing intervenes with the vibration of the grid strap, the LDV measures the mixed velocity of the housing and grid strap. Therefore, a special experimental environment is needed specifically for the measurement of flow-induced strap vibration.

3. Analysis of Experimental Environment and Improvement

The coolant is subjected to vibration from the entire loop, including the housing, because the coolant circulates throughout the loop. Therefore, an improved experimental environment is necessary to minimize the vibration of the housing into which the grid specimens are loaded. Characteristics of response are estimated through accelerometers attached on the housing surfaces. Fig. 2 shows the results of the housing modal testing through the LMS Test Lab. More than 10 modes of housing exist within a range up to 1,000 Hz, and several modes are located at higher frequency regions above 1,000 Hz. The mode shapes are typical bending shapes.

The response of housing due to the coolant, $\{X(\omega)\}$, is defined as the Eq. (1).

$$\{X(\omega)\} = [H(\omega)]\{F(\omega)\} \quad (1)$$

where,
$[H(\omega)]$: Transfer function of housing
$\{F(\omega)\}$: Exciting force of coolant

Coolant can be considered as white noise which is a random signal with a flat power spectral density, because the coolant has a severe turbulence with a very high flow velocity of approximately 5 m/sec. Therefore, the response of the housing can be expressed as Eq. (2) through the whole frequency.

$$\int_0^\infty \{X(\omega)\}^* \{X(\omega)\} d\omega \propto \int_0^\infty tr\left([H(\omega)][H(\omega)]^*\right) d\omega \quad (2)$$

The function can be rewritten as Eq. (3), because the amplitude of the flow-induced housing vibration is proportional to the magnitude of the transfer function.

$$J = tr\left([H][H]^*\right) \leq tr\left([U][U]^*\right) tr\left([\lambda]^2\right) \qquad (3)$$

where,
$[U]$: left side eigenvector according to singular value decomposition
$[\lambda]$: singular value matrix

Therefore, the position of vibration isolation plates can be determined by Eq. (3), to prevent vibration being delivered to the housing. The natural frequency of the unit cell grid strap is above 1,000 Hz, because the grid strap, compared to its own weight, is of a particularly stiff structure. Hence, the optimized locations of the vibration isolation plates can be determined within a 1/3 octave band, when the center frequencies are at 1,600 Hz and 2,000 Hz. Fig. 3 shows that the housing vibration decreases by 6 dB after the installation of the vibration isolation plates.

The LDV uses the principle of heterodyne interference to acquire characteristics of mechanical vibration. The LDV used in this test emits the He-Ne laser [2]. The emitted laser beams are firstly reflected onto the housing surface, and secondly reflected onto the strap surface. Thus the LDV measures not only the strap vibration, but also the housing vibration. To prevent this phenomenon, the reflected wave on the housing surface can be excluded using the law of reflection. In other words, when the laser is emitted with an optimal level of angle, the reflected wave from the housing surface can be removed. Because the reflected wave from the strap surface is scattered, the LDV is able to acquire measurements for only the strap vibrations. Fig. 4 shows the effect of the incidence angle. When the laser is emitted with the correct incidence angle, the housing vibration interference can be removed.

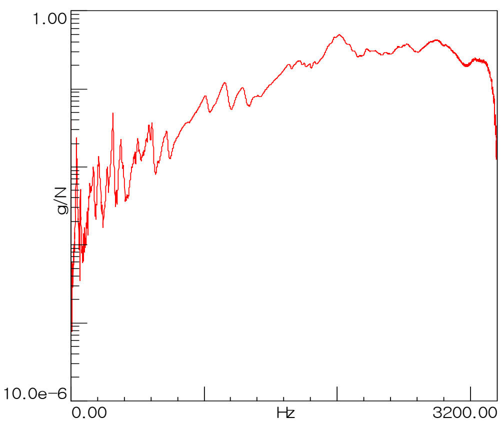

Fig.2 measured response function of housing

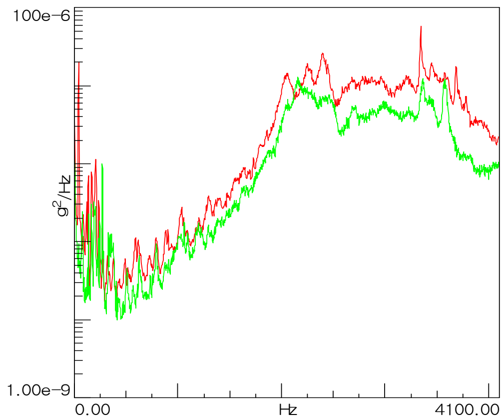

Fig.3 measured response before(red) and after modification (green)

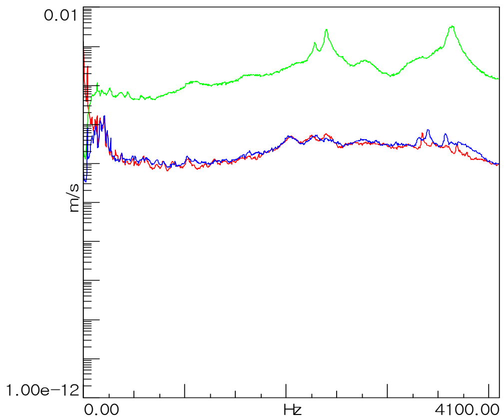

Fig.4 housing vibration(red and blue) and strap vibration(green)

4. Conclusions

Several tests were performed to determine the optimized test conditions for the measurement of flow-induced grid strap vibration.

The housing vibration can be minimized through the optimized location of vibration isolation plate, and the optimized LDV measuring conditions can be determined using the law of reflection. Therefore, through this study, the reliability of the flow-induced vibration measuring system can be improved.

REFERENCES

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[2] Polytec Vibrometer User's Manual