Twelfth Australasian Fluid Mechanics Conference
The University of Sydney, Australia 1995

VERIFICATION TEST FOR THE FLOW-INDUCED VIBRATION OF A NEW STEAM GENERATOR

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ABSTRACT

The results of the verification tests for the flow-induced vibration of the tube bundle of a new type of steam generator are shown. Partial tube bundle model of the half size mock-up is used in a water loop. It is proved that the responses of the tubes are small enough.

1. INTRODUCTION

A new type of steam generator for the PWR type nuclear power plant is now being planned in Japan. It is called "a high performance steam generator." It is composed of heat-exchanger tubes of U-bend type and an economizer in the inlet region of the feed water as shown in Fig.1.

There are some PWR power plants in Japan, where steam generators are used to get the steam flow for driving the turbine. These steam generators, however, have no economizer. Then it is required to prove the effect and the safety of the new design for introducing into Japanese power plants. The economizer adds an additional fluid flow in the inlet region of the secondary water, that is estimated to give an additional effect on the heat-exchanging capacity. On the other hand, it causes another inverse effect on the stand point of the flow-induced vibration of tubes.

This paper shows some results of the verification test for the flow-induced vibration of the above economizer type of steam generator. All test plan and the partial results are shown here.

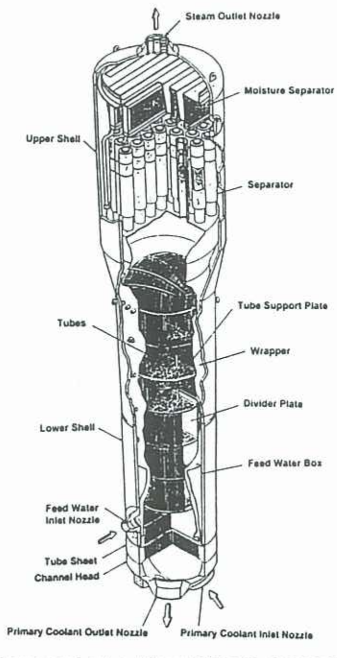


Fig.1 Concept of a New Steam Generator

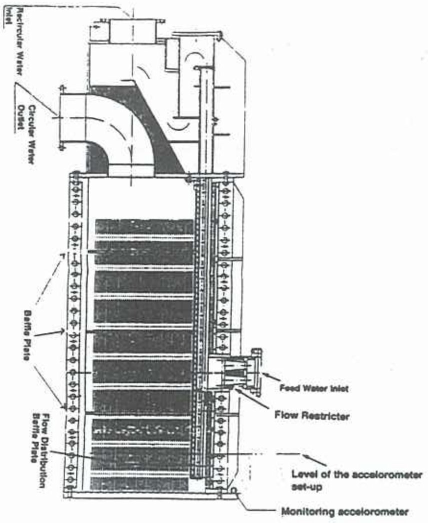
2. TEST PLAN

2.1 Similarity Law

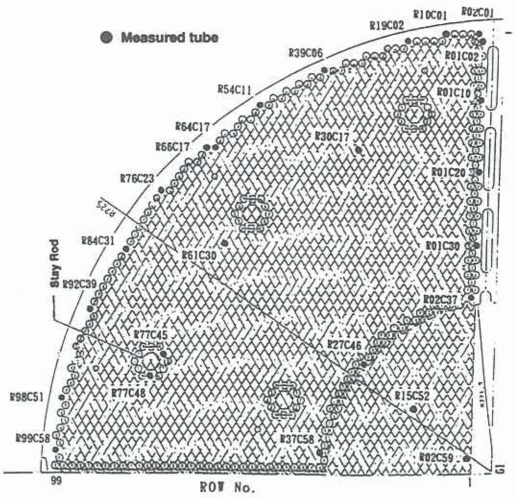
The tests have been performed using the water loop in Takasago R&D Center of Mitsubishi Heavy Industries. The test model have some limitations; the test should be done in almost room temperature and in an atmospheric pressure condition, the test model should be the part of the steam generator, and the size of the model should be smaller than that of the practical one to get enough flow velocity.

The size of the model has been fixed as the half of the practical tube bundle, including the diameter and the thickness of tubes. The model was a quarter part of the practical one as shown in Fig.2.

It is regarded for the problem here to be three kinds of the flow-induced vibrations, such as the resonance of the tube with the vortex shedding, the occurrence of the fluid-elastic instability and the intensity by the



(a) Mock-up



(b) Layout of measured tubes

Fig.2 Test Facility

turbulent buffeting force by the fluid flow.

The similarity of the test model is set for the same flow velocity as shown in Table 1. The fluid is water in the atmospheric condition, then the Reynolds number is one-twentieth of the practical one. It is not clear that this mock-up is similar to the practical one for the vortex shedding, but it is in the "sub-critical" region both for the model and for the practical one if it were a single tube. If the Strouhal number is the same as for the practical one, the frequency for the vortex shedding flow velocity, have the same ratio to the natural frequency of the tubes. Then the similarity for the vortex shedding resonance is satisfied. However, the response of the tubes has been followed while the flow velocity is increased to check the peak amplitude.

As for the fluid-elastic instability, the density of the fluid in the test is a bit heavier than that of the practical one because of the difference of the temperature. The critical flow velocity Uc in the test, that is supposed to be the following formula by Connors (1969), is lower than that of the practical one;

$$\frac{Uc}{fD}=\beta\left(\frac{m_T\delta}{\rho D^2}\right)^{\frac{1}{2}}. \qquad (1)$$

Finally, the responses of test tubes by the random force from the turbulence of the fluid-flow should be slightly larger than those of the practical tubes because the buffeting force F_b is assumed to have the following relation,

$$F_b \propto \rho U^2 DL. \qquad (2)$$

And the response of the tube can be expressed as the following,

$$X \propto \frac{F_b}{K}. \qquad (3)$$

Then the amplitude of the tube may be slightly greater than the half of the practical one because of the difference of the density of the fluid.

Table 1 Scaling Law

Parameter	Signature	Dimension	Model/Practical	Note
Diameter of tubes	D	L	0.5	Basic parameters
Density of fluid	ρ	$F\cdot T^2/L^4$	1.3 (×1)	
Flow velocity	U	L/T	1	
Vortex shedding frequency	f_K	l/T	2	
Natural frequency of tube	f_T	l/T	2	
Rigidity of tube	K	F/L	0.5	
Mass of tube per unit length	m_T	$F\cdot T^2/L^2$	0.25	
Length of tube	L	L	0.5	
Response of tube	X	L	0.5	
Nondimensional flow velocity	U/fD	—	1	Nondimensional parameters
Reynolds number	UD/2ν	—	$6\times10^4/1.2\times10^6$	
Scruton number	$m_T\delta/\rho D^2$	—	0.77	

2.2 Test facility

As shown in Fig.2 (a), the tube bundle is composed of the straight tubes with three spans, but the real test section is only the first span from the tube sheet to the first baffle plate. Flow Distribution Baffle plate (FDB) has larger holes for the tubes that keep the tubes are free from the plate. The tube is supported at the first baffle plate, where the tubes have 0.2mm gap-clearance for the plate in diameter, that is half of the practical one.

Fig.2 (b) shows the position of the measured tubes. The response of these selected tubes are measured with the bi-axial accelerometers set into the middle point of the span of the tubes.

The tubes are fixed on the tube sheet and have small gaps in the support buffle plate, which is regarded to be the pinned condition at the baffle plate. After a long period of the plant-operation, the heat-exchanging tubes are likely to become fixed condition at the support baffle plate because of the contamination in the small gap clearance. Then eight tubes of the measured tubes including the surrounding tubes are set to be fixed at the first baffle plate after the first test case has been completed.

However, it is anticipated to be hard to get the margin for the instability by the fluid elastic vibration. Then as additional test has been planned to get the instability, where the eight measured tubes have been cut at just above the tube sheet. The eight tubes become very flexible in this case. (The test results for the later two cases would not be introduced in this paper. They will be appeared in another method.)

2.3 Test Items

The economizer type steam generator can change the operating condition with the volume of the feed water from the economizer. It will usually be operated in 100% load, but it sometimes shifts to 50% load where the volume of the feed water from the economizer is less than the one in 100% load although the re-circular water is greater than 100% load. In this test, the balance of the volume of the both input flows is kept constant, but the volume of the flow is changed. The responses of the measurement tubes have been monitored while the flow volume was changed from one condition to the other, because vortex shedding can exist in narrow flow velocity range.

The natural frequencies and the damping ratios of the measurement tubes are obtained by tapping of the tubes, sinusoidal wave-sweep test, or tapping of the test mock-up itself. The vibration of the mock-up is measured while the test is done, in the out-side of the mock-up as shown in Fig.2 (a).

3. TEST RESULTS

3.1 Natural Frequency and Damping

The vibration characteristics of the measured tubes are estimated twice; before the test and after. Tubes have natural frequencies between 90Hz and 139Hz in air, and between 71Hz and 137Hz in water. The damping ratios are between 0.65% and 3.7% in air, and between 0.8% and 4.0% in water.

3.2 Response by Fluid Flow

Fig.3 shows the root-mean-square acceleration value of the response of the measured tubes. Fig.3 (a) shows the responses of the tubes in the out-side region of the tube bundle. "αU^2" in the figures means the relation where the response of the tube is linear to the second power of the flow velocity.

The responses in the central flow region, where there is a channel come from the center of the U-bend tubes, also shows that the responses of the tube have almost linear relation to the second power of the flow velocity,

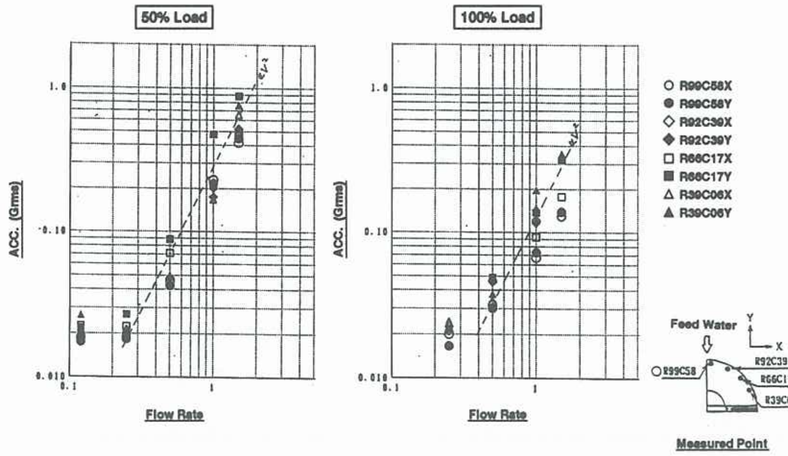


Fig.3 (a) Response of Tubes vs Flow Rate

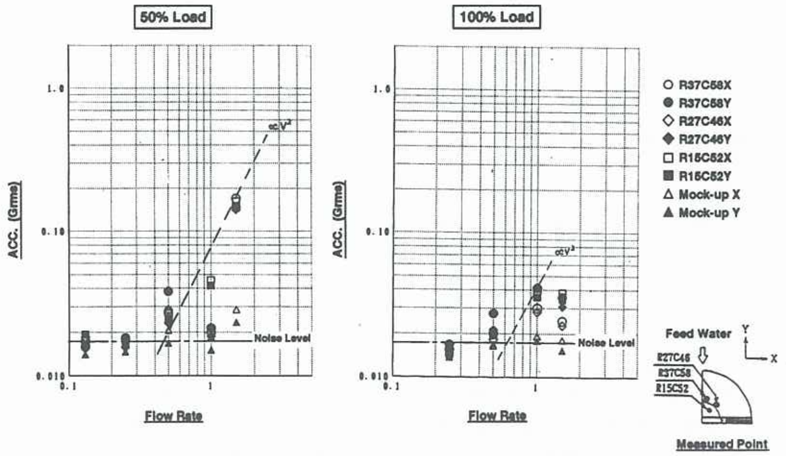


Fig.3 (b) Response of Tubes vs Flow Rate where the flow rate is theoretically linear to the flow velocity. The response in the smaller flow rate less than 30% are estimated to be a mechanical noise. This can be found in Fig.3 (b), that shows the response of the mock-up itself in out-side of the tube bundle. The other measured tubes in the outer-side region and in the central region show a similar trend of the RMS value of the response. The results are not shown here.

There are interesting facts to be revealed here, that the responses of the measured tubes in the middle of the tube bundle do not obviously show the linear relation to the second power of the flow velocity as shown in Fig.3 (b). As a result, all measured tubes in the center of the tube bundle show this kind of strange phenomenon.

The frequency spectrum of the responses of the measured tubes also show that the response of the tubes are only random vibration although there are some peaks in the natural frequencies of the tubes.

In addition to the above mentioned data, there has not been observed any peak of the response when the flow rate has changed. Then no resonance to the vortex shedding and no fluid-elastic instability have been observed.

4. DISCUSSION

The responses of the measured tubes indicate that the tubes are oscillated by the random buffetting force of the fluid flow in the former section. There are a couple of items that remain to be clarified: the first one is to estimate the vibration level is not excessive as a new design, and the second is to get some insight for the reason why the response of the tubes in the central part of the tube bundle do not show a linear relation to the flow rate.

At first, it will be required to estimate the stress of the tube that the response of the tubes are expressed in the form of the amplitude of the displacement instead of the acceleration. This amplitude corresponds to the one at the center of the span in each tube, then the maximum stress for each tubes is estimated with the relation between the displacement and the stress based on the analysis for the first mode of the span. The maximum peak displacement is 194μm for the tube R01C20 at the 100% flow rate of 50% load condition. Then the maximum stress can be estimated to be 1.52kgf/mm^2 for the test model, and it is the same for the practical plant by the scaling law. This is much smaller value compared with the usual stress limitation. Then the safety of the new design has been verified.

Secondly, we have to discuss for the non-linear relation to the flow rate in the tubes in the central region of the tube bundle. Fig.4 shows the distribution of the measured flow velocity along the tube axis. It is natural that the flow velocity inside the tube bundle is smaller than that of the outside one. And there is supposed to exist a large vortex, like a circuit, both horizontally and vertically. If the kind of vortices change their scale and their intensity, it would cause the local shift of the flow velocity for each tubes, where the distribution of the flow velocity depends on the vortex. This explanation, however, has a weak point that all measured tubes in the center region show the similar phenomenon. There may be some exceptions. Then, we had better give another idea. The key point is that 100% load condition shows the clear trend of the non-linearity, but that 50% load one does not. The difference of the two conditions in the balance of the feed water and the re-circular water. 50% load requires much volume flow for the re-circulating water. Then, it would be thought that in 1.5 time flow rate in 50% load the vibration of tubes in the central region shows a linear relation to the flow velocity and they can be small enough for the design-margin. Other data are in the range of mechanical noise.

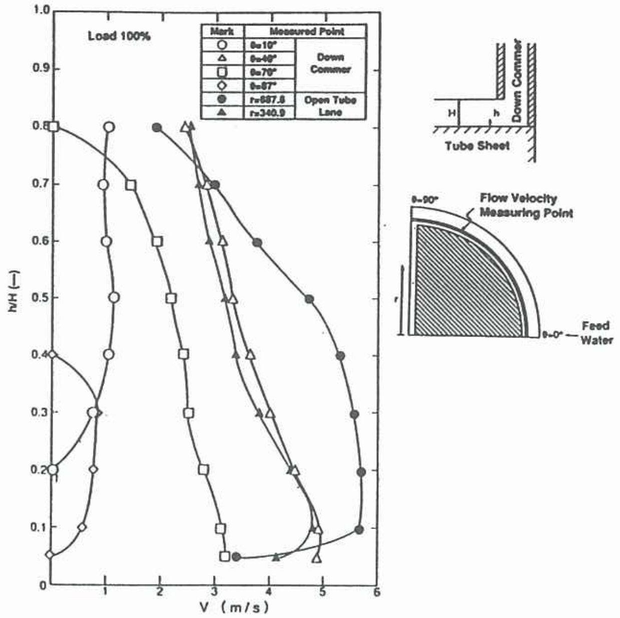


Fig.4 Flow Velocity Distribution

5. CONCLUSIONS

The new design of the steam generator of PWR plant is verified to have a small stress level for the flow-induced vibration. It is estimated to be a random vibration by turbulent fluid-flow. No instability has been observed. However, there are some local shift of the distribution of the flow obtained, although it is restricted in the area of smaller flow velocity and the smaller responses of the tubes.

6. ACKNOWLEDGMENT

This test has been performed under the sponsorship of MITI (Ministry of International Trade and Industry). The authors express their acknowledgment to those who concern the test.

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