

Shigeki SUZUKI, Yoshitsugu NEKOMOTO, MITSUBISHI (Japan) **MHI experience on piping vibration issue and total planning** **of piping maintenance**

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Abstract

Vibration or vibration related high cycle fatigue of small bore piping has been one of the main issues for maintaining the integrity of nuclear piping. Mitsubishi Heavy Industries (MHI) has been involved in root cause analysis of a number of leakage events in Japan, and accumulated and developed technology for piping design procedure or preventive maintenance activities of existing plants, from those experiences over 20 years. The concept and the outline of these technologies such as vibration measurement evaluation procedure, selection of vibration sensitive location, structural modification of cantilever type small bore branch pipe and positive countermeasure device for suppressing vibration, are introduced in this paper.

For nuclear power plant piping, several other degradation mechanisms are recognized, such as high cycle thermal fatigue, SCC for austenitic stainless steel pipe, FAC for carbon steel pipe, etc. The procedure for integrating all the information concerning each degradation mechanism and programming total maintenance plan is also introduced.

1 Introduction

MHI is the only PWR supplier in Japan, and has been involved in the construction of 23 PWR plants in Japan, and piping remodeling for over 30 years. During this period, MHI has accumulated wide variety of lessons learned from PWR piping failure events, and has been contributed the reliability improvement of Japanese PWR plants, through the intense activities for root cause analysis against such events with mock up tests/laboratory examination, and studying, planning and implementation of countermeasures for such failures.

Among several kinds of piping failure mechanism, vibration of small bore piping is one of the key issues. Piping failure events has been reported for over 20 years, but the number of those events is relatively small due to appropriate preventive maintenance activities to all Japanese PWR plants. MHI has developed and modified the vibration evaluation procedure, and has applied it to actual piping design. Some of those examples are introduced.

The most sensitive location of piping for vibration is cantilever type configuration, which has been often applied to vent/drain lines. Butt weld or special insert weld has been applied for relatively recent plants and for retrofit of vintage plants, where socket weld were commonly used to those lines.

MHI has developed not only design procedure, but also the reliable evaluation procedure for vibration measurement. One of those procedures is to consider the welding defect (estimated conservatively) and its influence to fatigue strength for socket weld joint, in the course of setting the allowable stress for vibration measurement result.

As for vibration measurement, MHI has improved the vibration evaluation philosophy from rigidity measurement to acceleration measurement. Rigidity measurement is based on the thought that vibration failure can be prevented when resonance is avoided. However, even if resonance occurs, if the vibration energy source is small enough, the response can be limited to an acceptable level. And on the other hand, even if resonance is avoided, if the vibration energy source is big enough, the response can be significant to the level at which vibration failure may occur. Together with this philosophy change, MHI has extended the range of vibration measurement objects. In the early period, high energy pumps are the only target for vibration evaluation. But several failure events have let us learn that it is not sufficient. High energy flow or pressure fluctuation by decompression device (sometimes cavitation enhanced) without pump excitation can be a piping vibration failure cause. Therefore, very wide variety of piping is the vibration measurement object in Japanese PWR plants.

As for vibration evaluation technique, MHI has developed some advanced procedures based on field experiences. One of these is the natural frequency evaluation of small bore piping accounting for the antiplane rigidity of main pipe in the design stage. Without considering this rigidity, the estimated natural frequency at the design phase may differ from the natural frequency measured in the field.

Another procedure is hydraulic resonance evaluation. Each branch pipe with closed end has hydraulic natural frequency, and significant vibration can sometimes observed when it comes close to pump pressure pulsation frequency, or even vortex generation frequency at piping branch point.

As for the evaluation of vibration measurement for cantilever type small bore pipe, evaluation procedure is simple. However, the evaluation of complex piping system is not easy. MHI has developed a sophisticated evaluation procedure for complex piping system using transfer function with a calibration by tapping test result. This procedure has been applied both for vibration failure analysis and preventive vibration measurement.

Some positive countermeasures have also been studied and come to the applicable stage to actual plants. This device is a self-tuning dynamic absorber, and free from the natural frequency measurement of the object pipe. This device contains some small balls in it, and they rotate with the exciting frequency with some phase difference, which add meaningful damping to excited small bore piping.

For PWR piping, vibration is not the only degradation mechanism. High cycle thermal fatigue (cavity flow type or valve seat leakage type thermal stratification, thermal fluctuation at T-juncture etc.), IG-SCC for stagnant area, TG-SCC due to vinyl chloride tape or out door piping near the sea, FAC for carbon steel pipes, etc. MHI has accumulated deep knowledge for each mechanism and developed procedures with Japanese PWR utilities, which integrate

the knowledge for all the information on susceptible location and its level of susceptibility for all the degradation mechanisms, and establish the most rational and economical preventive maintenance plans for piping.

2 Preventive maintenance activities against vibration of small bore piping

2.1 Chronology of response to vibration issue

Table 1 shows the chronology of Japanese PWR piping leakage event due to vibration and countermeasure activities following those events. Almost all of the events occur at small bore piping with socket weld joint. Till 1990's, cantilever type structure has been the major player. However, in the late 90's and 2000's, the number of cantilever events has decreased (almost disappeared), and failures of complicated piping system configuration have started to arise instead. This tendency may be interpreted to be the result of elaborate countermeasures for cantilever type branch lines, such as systematic vibration measurement or configuration remodeling following its result.

Table 1 Chronology of response to piping vibration issue

	1979	1980	1981	1982	1983	1984		1989	1990	1991	1992	1993	1994	1995	1996	1997	1998	1999	2000	2001	2002
Event	Takahama-1 SIS Vent Line RCP Excessive Vibration	Oh-1 CVCS Charging Vent Line (Resonance with Reciprocating Charging Pump's Pulsation)		MSS Sampling Nozzle (Random Vortex Excitation)	Oh-1 MSS Sampling Nozzle (Random Vortex Excitation)						Takahama-1 MSS Vent Line (Selective-Resonance)	Takahama-1 Attached Pipe to RCP (Resonance)			Monju Thermo-well (Resonance with vortex)	Oh-2 CH/S-P Suction Vent Line (Selective-Resonance)	Oh-2 RHR-P Delivery Line Drain (Resonance with nZ)		Mihama-2 CH-P Delivery Line (Hydraulic Resonance)	Mihama-2 RCP Seal Injection Line Vent (Vibration due to Cavitation)	Mihama-3 RCP Seal Injection Line Vent (Vibration due to Cavitation)
Countermeasure		Vent & Drain Pipe Design Modification (Butt Weld Type)		Vent & Drain Pipe Design Modification (Special Insert Weld Type)							Design Manual for Small Bore Piping, Established	-Natural Frequency Evaluation Procedure Considering the Rigidity of Main Pipe -Vibration Measurement Procedure as Confirmation after Construction									
		-Confirmation of Natural Frequency ($\approx 40\text{Hz}$) by Calculation or Tapping for under 2B Conti-lever Type Branch Line -Vibration Measurement when Natural Frequency is under 40Hz -PT Implimentation.										-2B Added -Socket Weld Leg Dimension Measurement Added. -Confirmation of Resonance Avoidance with Pumps other than RCP (by tapping or calculation). => Vibration Measurement if Necessary									

2.2 Modification of socket weld configuration

All vibration fatigue failures in small bore piping at Japanese PWRs are for socket weld configuration. A possible reason of this fact is attributed to stress concentration at the root or the toe of socket weld. As a matter of fact, according to METI Code 501, which is

comparable to ASME Code Sec.III, the stress indices of socket weld joint for moment loading K_2C_2 is 4.2, and it is approximately three of four times bigger than that of butt weld. Another reason could be that socket weld has relatively high probability to contain weld defect compared to butt weld, which contributes significant fatigue strength reduction. This effect is clearly perceived and quantified in Fig. 1.

The best answer to small bore piping vibration issue is the design change from socket weld to butt weld. However, it is necessary to modify the branch pipe together with the main pipe when this design change is applied to existing plants, because the branch pipe is connected to main pipe with socket weld boss, and when the boss is changed, the main pipe has to have modification. Therefore, "special insert weld" is applied to existing plants as the second best answer to vibration issue. In this case, socket weld can be eliminated while the socket weld boss is re-used and no need for main pipe remodeling. The concept of modification of socket weld configuration is shown in Fig. 2.

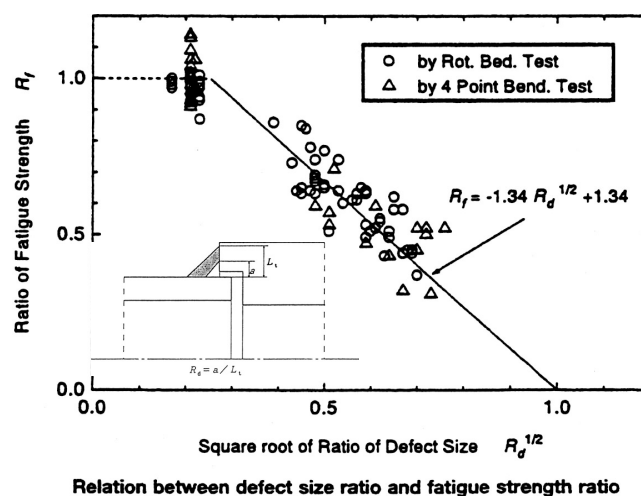


Fig.1 Effect of weld defect on fatigue strength

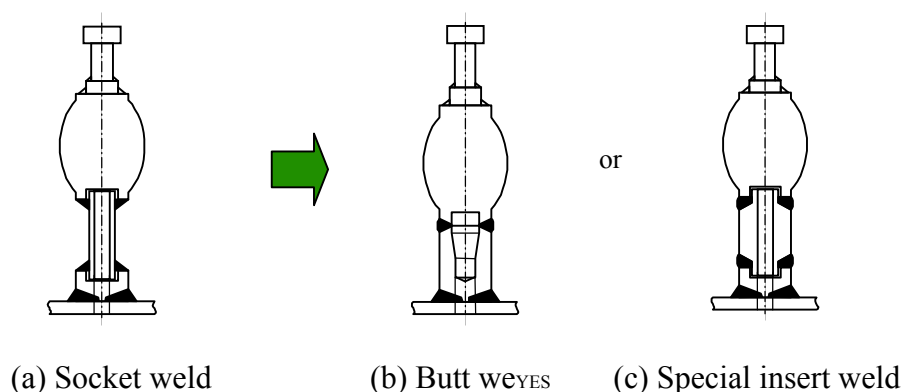


Fig.2 Design change of vent/drain line

2.3 Vibration measurement for cantilever type branch line

Systematic vibration measurement for cantilever type small bore branch pipe has been implemented. The key issue is how to deal with the amount of quantity, which are approximately a thousand per one PWR plant. The most basic idea to limit the vibration measurement scope is to select the small bore piping which is placed in a region where vibration energy is transmitted. However, the object of vibration measurement has been expanded based on the lessons learned from field experiences. In 1980s, cyclic pump pressure pulsation (nZ) or mechanical vibration (n) by rotating shaft imbalance used to be the only vibration energy source. However, in 1990s, vibration failures due to random excitation by high energy fluid flow such as main steam flow, or by high pressure decompression device such as let down orifice have been observed, followed by expansion of vibration measurement coverage.

Even though limiting the scope of vibration measurement by excitation source, there still remains a good amount. Therefore, in case of implementation of the measurement, priority ordering has been conducted based on excitation energy level, operating hours, whether the source has a specific dominant frequency or not, possibility of resonance, etc.

As a result of vibration measurement, when the branch pipe is judged to be susceptible to high cycle fatigue failure, modification described in 2.2 is carried out, usually in the next refueling outage, and these activities probably contribute to the low failure rate of Japanese PWR due to vibration.

Table 2 Vibration energy source to small bore piping

Exiting force		Evaluation Object
Pump Mechanical Vibration (n component)		
Pump Pressure Pulsation (nZ component)		
Random Exitatio	Pump Pressure	
	Pressure exited by high energy flow	
	Pressure fluctuation by decompression equipment (sometimes cavitation enhanced)	

2.4 Natural frequency evaluation considering antiplane rigidity of main pipe

To avoid resonance with pressure pulsation from pumps, it is very crucial to estimate the accurate natural frequency of branch pipe at design stage. When the rigidity of main pipe is high enough, relatively simple procedure such as hand calculation using a handbook will do. However, as the main pipe bore becomes bigger and the wall thickness thinner, the effect of main pipe rigidity becomes significant. If this effect is counted incorrectly, unexpected resonance may occur, which lead to remarkable vibration amplification ratio and fatigue failure.

Corresponding to this complexity of evaluation procedure and the amount of quantity of small bore branch pipe, a simplified design procedure has developed. The outline of this procedure is shown in Fig. 3. The first step is an ordinary way to generate a multi-mass point analysis model. As a boundary condition at the base of branch pipe, infinite rigidity is used to be set in traditional way. However, finite rigidity is recognized to be counted, and rigidity evaluation charts have been prepared. These charts are generated by a series of 3D-FEM analysis and organized using Bijlaad parameters, $\beta = 0.875 \cdot r/R$ and $\gamma = R/t$. The FEM parameters are selected in much wider range than Bijlaad method's usage restriction; $\beta \leq 0.25$, so these charts provide much wider applicability to piping designers as the second step of this procedure. Finally, eigenvalue analysis is carried out using the multi-mass point analysis model with the finite rigidity as a boundary condition. An example of 3D-FE_model and rigidity evaluation chart is shown in Fig. 4.

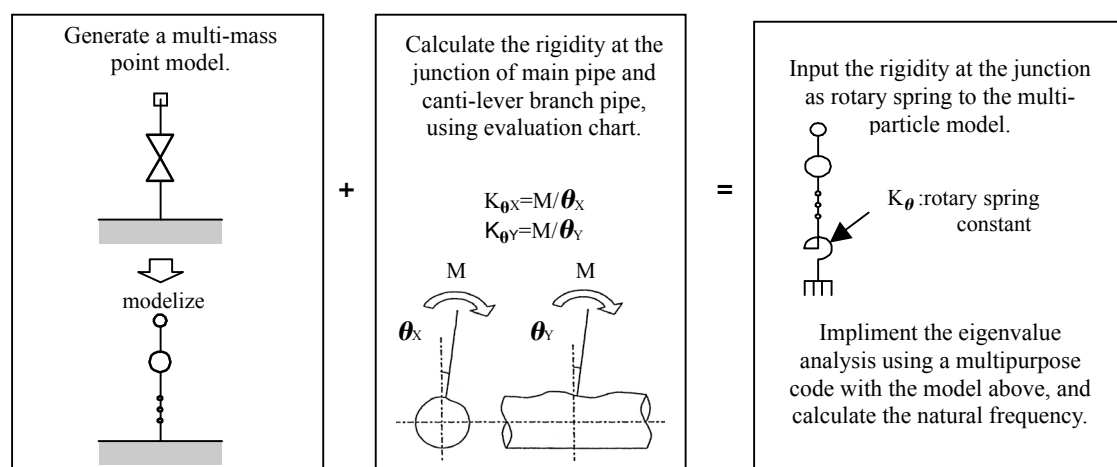


Fig. 3 Outline of main pipe rigidity evaluation

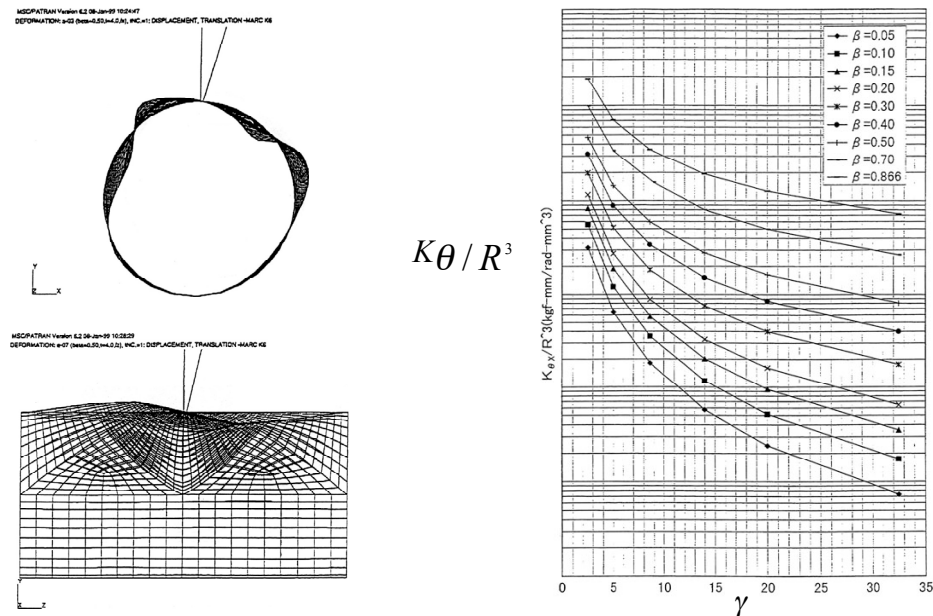


Fig. 4 Example of 3-D FE Model and rigidity evaluation chart

2.5 Hydraulic resonance evaluation

Occasionally, unexpected pressure pulsation at a specific frequency can be observed, conducting a root cause analysis of piping vibration fatigue failure in thermal, nuclear or any other processing plants. This phenomenon is attributed to hydraulic resonance. As shown in Fig. 5, any piping segment has a specific hydraulic natural frequency f_h depending on its length L and acoustic velocity C in the fluid. When some excitation frequency f_s accords with f_h , significant amplification ratio appears even though the source level of excitation is small, and as a result, considerable pressure pulsation may arise, which may result in a vibration fatigue failure.

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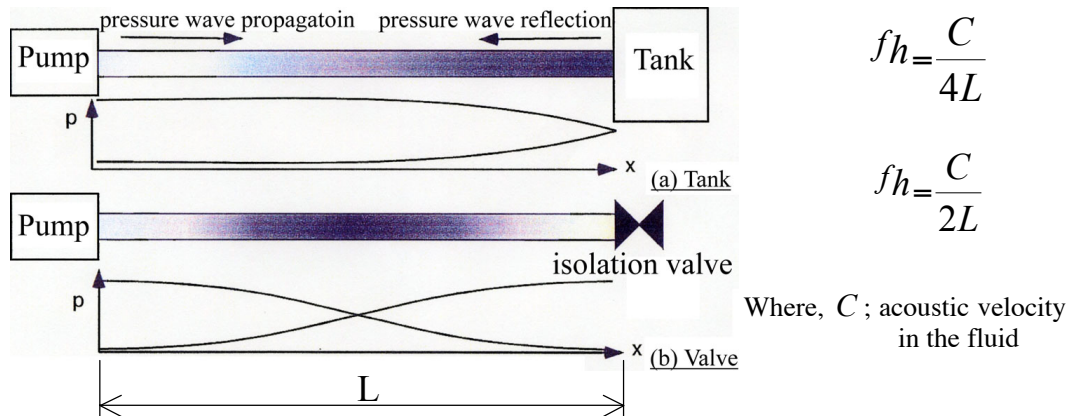


Fig. 5 Hydraulic natural frequency

One typical example that hydraulic resonance is taken into account at design stage is vortex excitation at the corner of stagnant branch pipe from MCP. Vortex is generated induced by MCP flow with the frequency of $f_v = S(U/d)$, where S is Strouhal No., U is flow velocity of MCP and d is inner diameter of branch pipe. Although the excitation energy source level is normally low, it can be amplified by hydraulic resonance to be significant. The only adjustable parameter is branch pipe length L . Therefore, L is adjusted at piping routing design to avoid hydraulic resonance to occur. Otherwise, cantilever type small bore pipes around the branch line may suffer great pressure pulsation, which may cause considerable vibration.

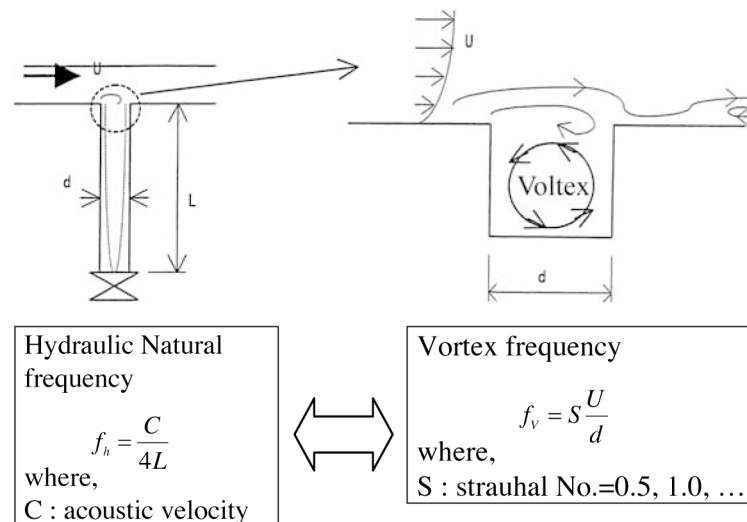


Fig. 6 Hydraulic resonance of stagnant branch line with vortex

Another example is amplification of high order pressure pulsation of reciprocating pump. Reciprocating pump has several specific frequencies, $1n, 2n, 3n, \dots$ in pressure fluctuation. The energy level of the first order pulsation is the highest, and as the order increases, the energy level decreases drastically. In traditional piping design, it was recognized that high order of pressure pulsation, such as 4th order ($4n$) or higher, is negligible. However, from the lessons learned, it is necessary to avoid hydraulic resonance even for high order of pressure pulsation. Fig. 7 shows vibration evaluation flow chart counting on hydraulic resonance. Piping system shall have higher natural frequency than 4th order of pressure pulsation to avoid normal resonance with the low order of pulsation, and design check is to be done to confirm that hydraulic resonance is avoided for high order of pulsation. If resonance is anticipated, design change of piping rout or piping support shall be carried out. Pressure pulsation analysis or vibration measurement may be conducted as a pessimistic option.

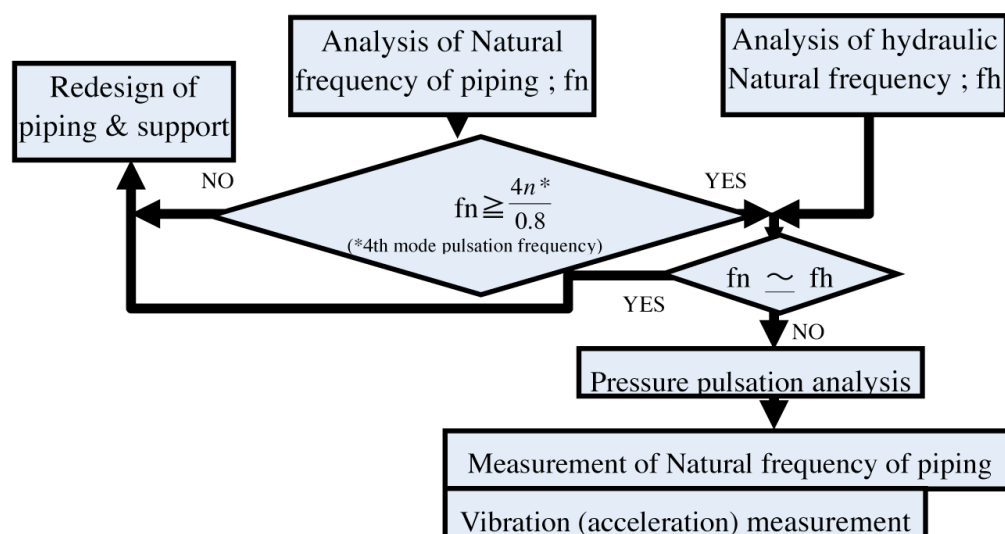


Fig. 7 vibration evaluation flow chart

2.6 Vibration measurement and its evaluation for complex piping system

As indicated in Table 1, most vibration failures used to occur at canti-lever type small bore pipes till the end of 1990s. However, rather complex piping system is becoming a major player recent years as the decrease of canti-lever failure. This can be interpreted that systematic preventive maintenance activities such as vibration measurement or configuration modification to butt weld for canti-lever type have worked well.

Based on the trend described above, the importance of vibration measurement evaluation accuracy of complex piping systems is growing, and a sophisticated procedure using a transfer function has been developed and applied to actual plants' evaluation. Fig. 8 shows the evaluation flow chart. The first step is to generate a multi-mass point analysis model. If the tapping test result is available, the analysis model is modified to have the same natural

frequency as the one obtained by the test. This procedure puts more accuracy to the analysis result. Conducting a sweep response analysis using the modified analysis model, piping system vibration response characteristic can be derived in the form of transfer function. Once the transfer function is acquired, vibration stress at focused point in the piping system can easily be translated inputting vibration measurement data to it.

Fig. 9 describes a little more detail of the transfer function. Conducting a sweep response analysis using unit load at any frequency, transfer function from load to acceleration at a vibration measuring point is generated. And at the same time, another transfer function from load to stress at a focused point can also be prepared. Mixing these two transfer functions, transfer function from acceleration at a vibration measuring point to stress at a focused point is generated. Selecting a vulnerable point to vibration excitation as a measuring point, vibration stress at arbitrary point can be evaluated preparing a transfer function for that point.

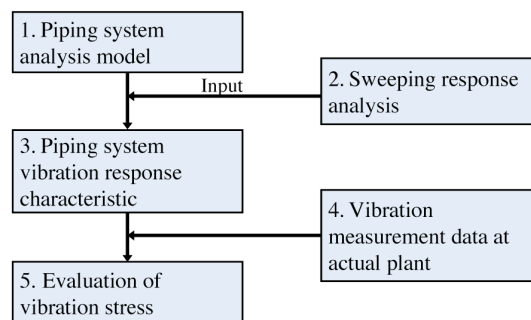


Fig.8 Evaluation flow chart for complex piping

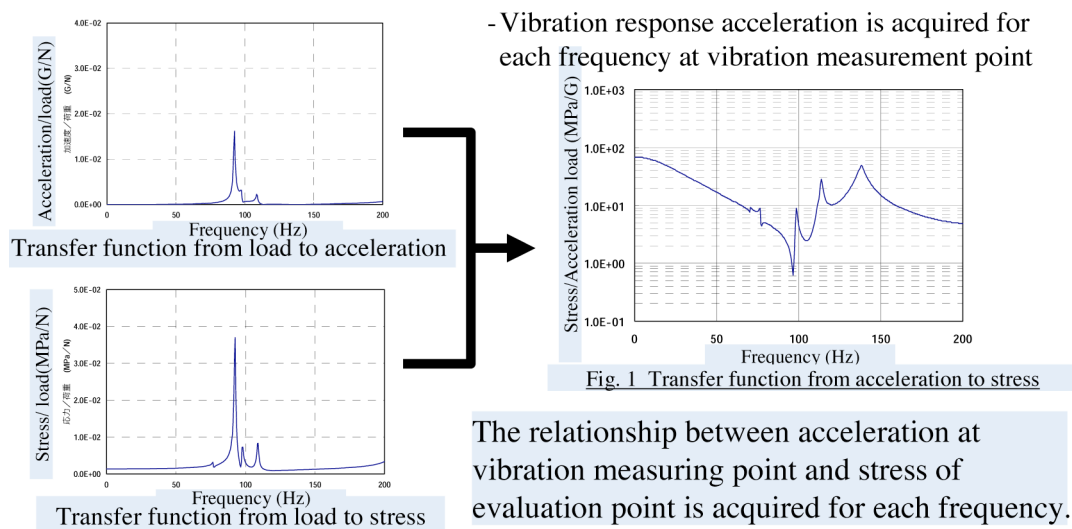


Fig.9 Generation of transfer function

2.7 Development of self-tuning dynamic vibration absorber

As a countermeasure against fatigue failure of small bore piping, a self-tuning dynamic vibration absorber has been developed. This absorber can be used as a temporary measure till the next refueling outage when notable vibration is observed during plant operation, or can be applied to a little space location where remodeling execution is difficult.

This absorber has an outstanding advantage compared to traditional dynamic vibration absorber, which needs manual frequency adjustment. Fig. 10 shows the structural concept of this absorber. Some balls are built into a ball case, which is clumped to vibrating small bore piping, with small gaps. The balls rotate synchronized with the piping vibration frequency with significant phase difference, and this phase difference and rotating friction between the balls and ball case adds damping effect. The synchronization occurs at any vibration frequency, so manual adjustment is not necessary.

The characteristics of this device can be summarized as follows.

- simple structure and applicability for high temperature piping
- no need for preliminary vibration measurement and frequency adjustment
- easy installation without any treatment to the pipe

The damping effect of this device is formulated using an engineering model shown in fig. 11, and verified by theory and experiments. The performance of this absorber is shown in Fig. 12. Remarkable damping effect is observed in wide range of frequency.

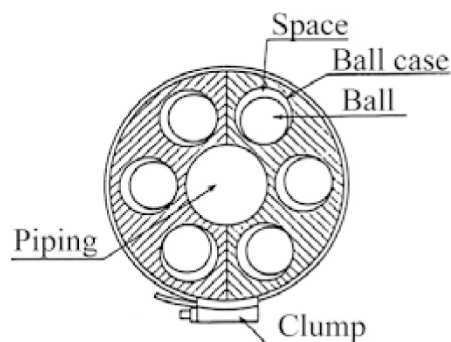


Fig. 10 Structural concept

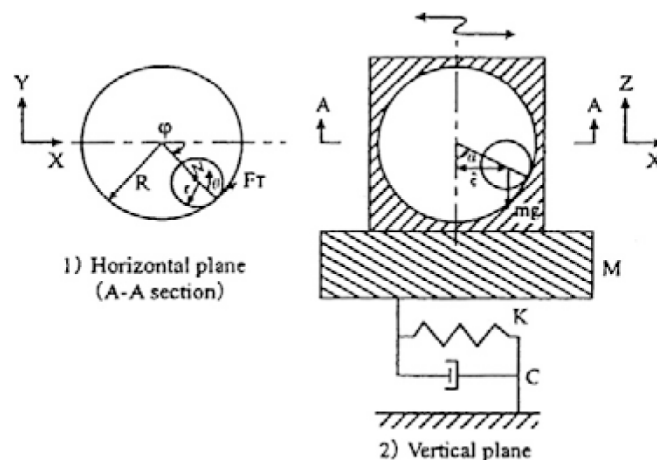


Fig. 11 Engineering model

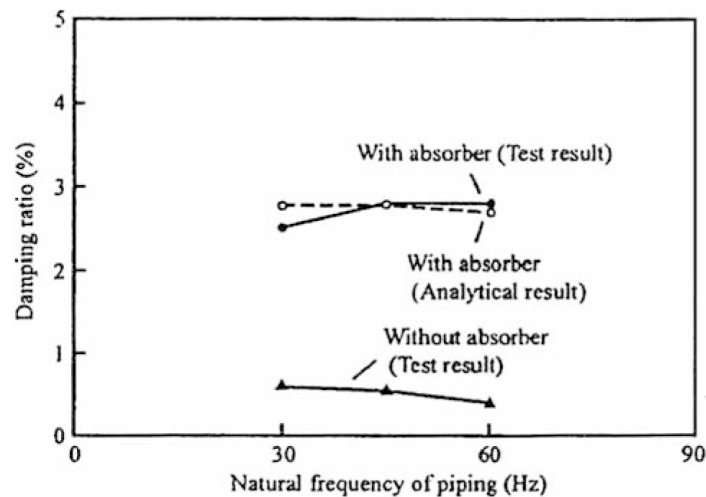


Fig. 12 Damping performance

3 Total planning of piping maintenance

Fig. 13 shows the cause analysis of unscheduled shut-down at Japanese PWRs. In 1980s and 1990s, heavy components used to be dominant, but Steam generators and Reactor vessels have been replaced. As a result, piping and valves are becoming champion, and effective and efficient preventive maintenance procedure is required to maintain the integrity of piping and valves pressure boundary material.

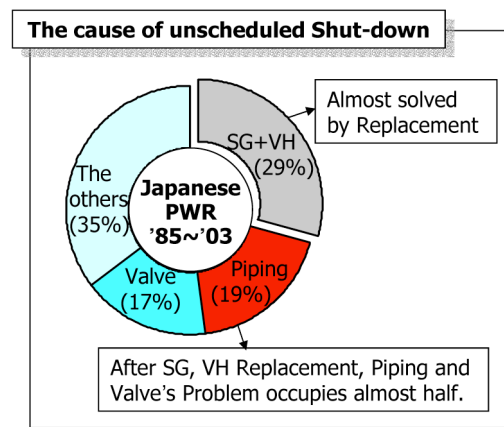


Fig. 13 Analysis of unscheduled shut-down

When considering preventive maintenance of piping and valves, it is inevitable to deal with the huge amount of quantity and the wide variety of degradation mechanisms. Therefore, "integration" and "prioritization" are the key words for it. Specialization and segmentalization has been developed in modern science and technology world, but this tendency is not always

preferable to nuclear power plant's piping maintenance. Vibration authority sometimes tends to think of vibration issue only, and plans modification of the branch pipe after vibration measurement. After the implementation of the plan, another SCC or thermal fatigue specialist may consider piping rout change to let the cavity flow front stays at vertical line and make the isolation valve cold. If the branch pipe just modified is located at the main line where thermal fatigue is susceptible, re-work becomes reality. This type of activities is completely the opposite side of rational, effective and efficient preventive maintenance. To avoid this kind of inefficiency, all the knowledge and experience of each degradation mechanism should be integrated, and superior preventive maintenance actions to be taken. We call this kind of activities "total planning of piping maintenance", whose necessity and concept are illustrated in Fig. 14 and 15.

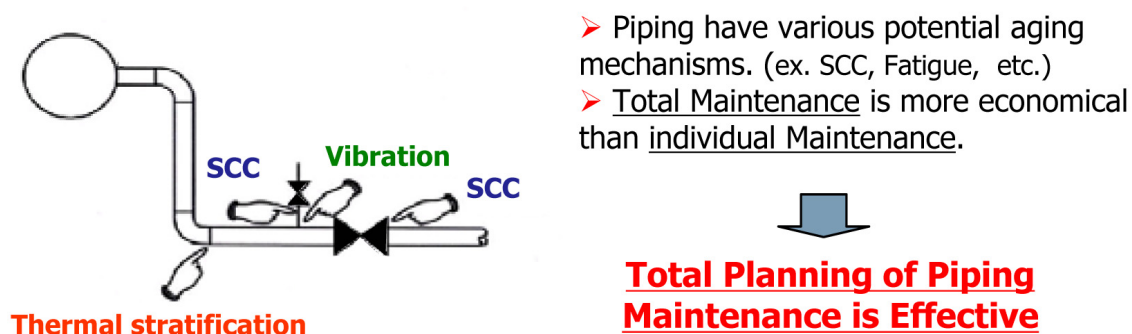


Fig. 14 Necessity of total planning of piping maintenance

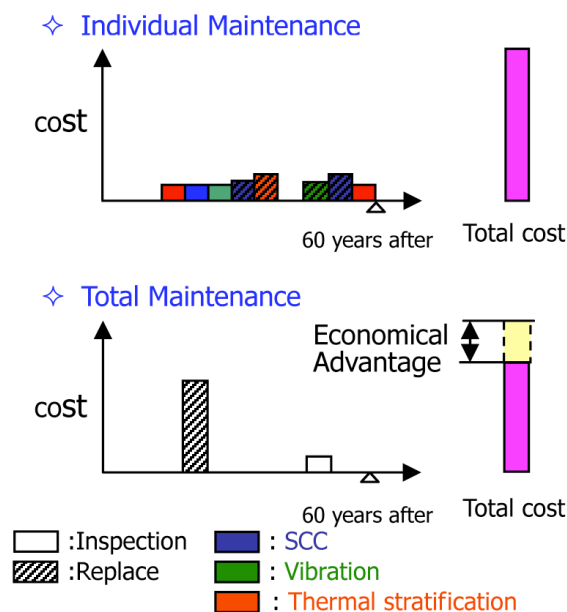


Fig. 15 Concept of total planning of piping maintenance

The first step to achieve this goal is organizing a number of degradation mechanisms which utilities and engineers have encountered. The image of organization is indicated in fig. 16. Not only mechanism itself, but the conditions on which each degradation mechanism is susceptible from the viewpoint of material, environment, stressor, operation conditions, service duration, etc. has to be well organized.

The next step is identification and prioritization of the potential area which is susceptible to each degradation mechanism. This procedure is conducted base on the knowledge which was collected and organized in the first step.

Finally, total maintenance plan is built up, integrating all the information organized till the previous steps. These plans shall be based on a broad range of piping lines, degradation mechanisms, priorities, the choice of maintenance method, modification experience, vibration or temperature measurement experience, etc. Therefore, colored isometric drawings, which contain any information mentioned above, are effective and put to practical use. Fig. 17 shows an example of such isometric drawing.

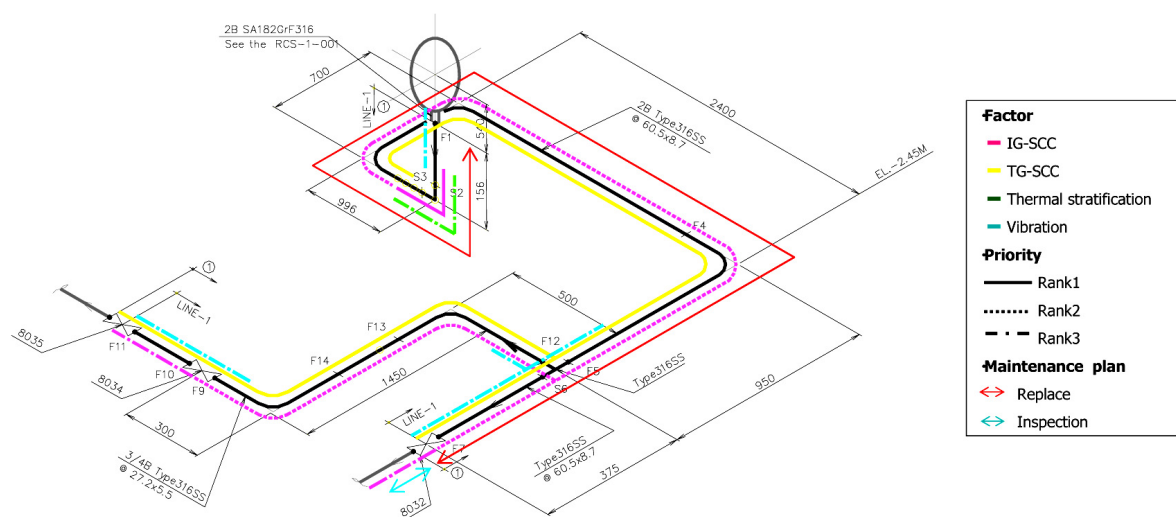


Fig. 17 Example of total planning isometric drawing

4 Conclusion

High cycle fatigue of small bore piping is one of the key issues of nuclear power plant piping, presumably in most of the countries in the world. MHI has gone through wide variety of field experiences in Japan, maintenance activities such as vibration measurement and its evaluation, reforming of vent/drain structure, and developing unique countermeasure device, etc. The authors would be happy if the utilities and safety authorities in the world find this paper informative, and provide them a clue to maintain the integrity of piping and its safety relate functions.