특 별 강 연

Review of Fatigue Failures in LWR Plants in Japan

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A review was made of fatigue failures of nuclear power plant components in Japan, which were experienced in service and during periodical inspection. No case has been recently reported of a service fatigue failure of a reactor pressure vessel itself, excluding nozzle corner cracks, that occurred many years ago. But, service fatigue failures have been occasionally experienced in piping systems, pumps, and valves, on which fatigue design seems to have been inadequately applied.

The causes of fatigue failures can be divided into two categories: mechanical-vibration-induced fatigue and thermal-fluctuation-induced fatigue. Vibration-induced fatigue failure occurs more frequently than is generally thought. The lesson gleaned from the present survey is a recognition that a service fatigue failure may occur due to any one or a combination of the following factors: (1) lack of communication between designers and fabrication engineers, (2) lack of knowledge about a possibility of fatigue failure and poor consideration about the effects of residual stresses, (3) lack of consideration on possible vibration in the design and fabrication stages, and (4) lack of fusion or poor penetration in a welded joint.

1. Introduction

As shown in table 1, it was 24 years ago that the first commercial nuclear power plant, Tokai Plant No. 1, a gas-cooling-type reactor, began operation. Four years later, the first BWR-type nuclear plant, Tsuruga Plant No. 1, and the first PWR-type nuclear plant, Mihama Plant No. 1, started their commercial operations. The electric power generated by these early stage reactors is approximately 350 MWe. In 1979 the electric power generated per reactor was increased to 1,100 MWe. As of August 1990, 38 commercial nuclear power plants were being operated in Japan, and 2 plants are now under trial operation. Additionally 10 plants are under construction. The total electric power generated by the nuclear power plants has ranged between 23 and 29% of the total electric power consumed in Japan in recent years. The capacity factor of the nuclear power plants has been in the range of 68% to 77%. The number of accidents and troubles has shown no remarkable decrease in recent years. In other words, it seems that the average incidence per year has settled to a fairly stable value, between 1.0 and 1.5. Here, the average incidence means the total number of incidences per year divided by the total number of nuclear power reactors.

If an electric utility experiences any trouble, deficiency, or accident during service operation or in the course of in-service inspection, the utility has to report the related details to the MITI (Ministry of International Trade and Industry) [1]. This is a legal requirement. If the nature of such trouble is not so simple as to be understood by the MITI staff such that they can give instructions and guidance to the utility, the MITI asks the Technical Advisory Committee for Nuclear

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Table 1
Nuclear power plants in Japan (as of August 1990)

Operation started		Name and type of plant	Power (MWe)
July	1966	Tokai (GCR)	166
March	1970	Tsuruga-1 (BWR)	357
November	1970	Mihama-1 (PWR)	340
⋮		⋮	⋮
October	1979	2nd Fukushima-1 (BWR)	1,100
March	1979	Ohoi-1 (PWR)	1,175
		Total 38 plants	29,280

Power Plant Operation for helpful advice and guidance to explain the trouble and provide proper countermeasures to be given to the utility. The Advisory Committee, which is organized by university professors and specialists from national research institutes, is independent of the MITI. The cases of trouble and accident, which are brought up for discussion in the Technical Advisory Committee, are generally required to be investigated by experimental and theoretical studies in order to explain the failure mechanism and to establish countermeasures to prevent recurrence of the same or similar troubles and accidents. The MITI has published every year an annual report of service troubles and incidents experienced in the previous year at the commercial nuclear power plants [2].

Notwithstanding the general situation that structural components ought to be carefully designed and fabricated so as to prevent a possible service failure, in fact many kinds of service fatigue failure have been experienced in structural components in piping systems, pumps, and valves. Most of the failures seem to have occurred due to factors unexpected in the design stage, but in some cases an imperfection of a welding fabrication triggered a fatigue failure. At this point it may be worthy to note that there have been some cases where blankly careless mistakes in fabrication, maintenance work, and operation resulted in service failures. This is an unwelcome tendency, but this problem seems unfortunately to be growing slowly but steady. It must be true that an analysis of a service failure and feedback of the knowledge obtained from it to the designers, fabricators, and operators should be very useful for the development of new techniques in design, fabrication, and safe operation.

The following deals with a review of typical service fatigue failures experienced in Japan in the hope that many valuable lessons gleaned from the comprehension of the cases can be helpful to prevent the same and similar failures.

2. Classification of fatigue failures

What structural components in a nuclear power plant are most liable to suffer from fatigue failure? The answer is piping components, nozzles, valves, and pumps in the order of the number of cases. Especially the number of fatigue failures that have occurred in piping systems is the largest and most outstanding.

The causes of fatigue failure may be mainly divided into two categories. The first is mechanical vibration due to Kárman vortex and forced vibration induced by a pump, non-stationary turbulent flow in a pipe, etc. The number of cycles to failure is not known in most cases, but it is assumed to be on the order of 10^7 cycles and more. It is pertinent to mention that the relevant stress amplitude in such cases is not very high, probably on the order of about 30 to 100 MPa, and welding residual stresses seem to have effectively contributed in many cases. Often discussed in this regard is the effect of welding residual stresses on fatigue crack initiation and propagation. In such discussion, however, no consideration is generally given to the subject of relaxation of welding residual stresses by an applied external load. The residual stresses may not remain as they existed in the as-welded condition, if a welded joint is subjected to static or cyclic loading. It is a well-known fact that a partial relaxation of the welding residual stresses will occur by one cycle static loading or the first cycle in cyclic loading. And furthermore, in the case of cyclic loading, relaxation of the welding residual stresses may be continued, more or less depending on the value of the applied stress.

The structural components, that have suffered from vibration induced fatigue are as follows:

(1) Heat exchanger tube in steam raising unit
(2) Splitter plate in elbow
(3) Hydrostatic bearing ring in primary re-circulation pump
(4) Socket welded joint
(5) Instrument piping for main steam line
(6) Mechanical seal line for high pressure condensate pump
(7) Instrument line for primary re-circulation line
(8) Vent line for suction or discharge valve of PLR pump
(9) Instrument line for PLR pump motor upper bearing oil level
(10) Drain line for hand operated valve of RHR system
(11) Seal water return line for feed water pump.

The second category consists of fatigue failures due to thermal fluctuation. Structural components fatigued by thermal cycling are:

(1) Feed water nozzle corner and sparger opening in BWR
(2) Control rod driving (CRD) return nozzle
(3) Connection of feed water system with reactor water clean up system – recombination tee
(4) Butt joint of horizontal pipe in RHR system.

Such failures generally occur by the mechanism of mixing of cold water into hot water, and in another case by cyclic thermal stratification. The number of cases of such failure is not high. It may be said that the above-mentioned cases (1), (2), and (3) are past problems, because they were found around 1977, and no cases have been reported in recent years. On the other hand, case (4) is a very interesting phenomenon that occurred in 1988, even though it is only one case.

3. Fatigue failure due to Kárman vortex

The first case of service fatigue failure in the history of Japanese nuclear power plant operation was experienced in 1966 in Tokai No. 1 Nuclear Plant [3]. This is the first plant constructed in Japan, and even today it is the only gas-cooling-type reactor. One month after the start of commercial operation the plant had to be shut down because of steam leakage from low-pressure super heater tubes in all four steam raising units. Figure 1 shows the vertical section of a steam raising unit. A high-speed flow of CO_2 gas comes down from the top of the unit for the purpose of heat exchange. The fatigue failure occurred in tubes located in the low-pressure super heater unit.

As shown in fig. 2, tubes of low carbon steel were supported by struts in three groups with a span of 2.2 m and 2.5 m. Because of the requirement for anti-oscillation in an earthquake condition, a semicylindrical cover plate of 7 mm thick was fillet-welded on a strut to fix the tube, which was inlaid in a groove on the strut, as shown in fig. 3. Finally circumferential fillet welding was executed on both edge faces of the strut and the cover plate.

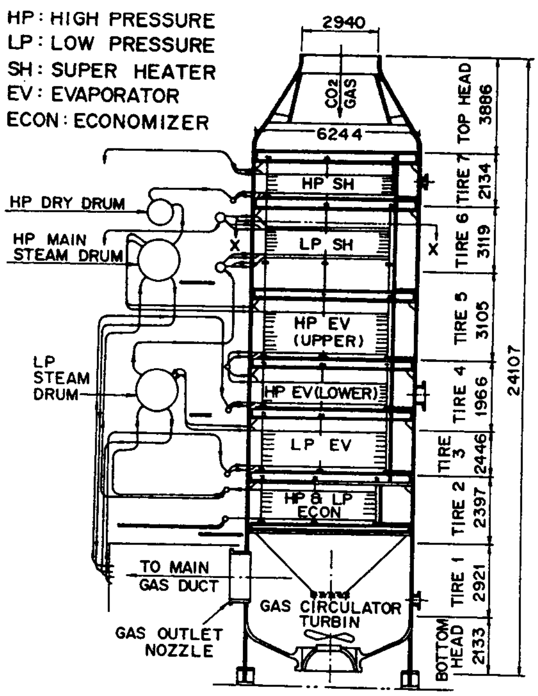


Fig. 1. Vertical section of steam raising unit.

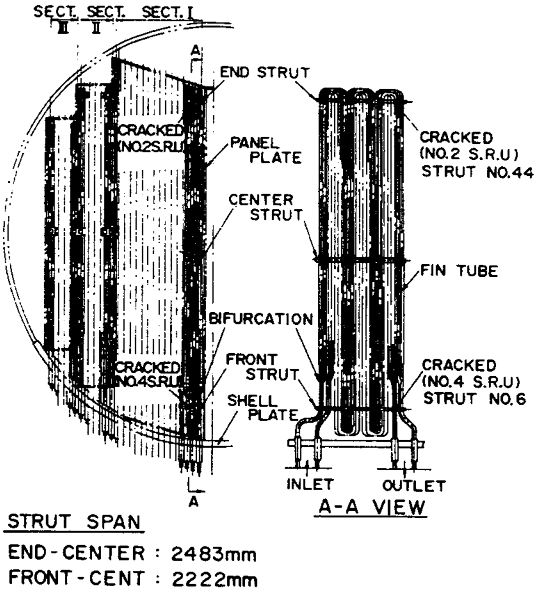


Fig. 2. Horizontal section (X–X in fig. 1) of steam raising unit.

Fatigue cracks were found in a certain range of fillet toes, which is a localized area perpendicular to the coolant gas flow direction. This observation was considered to afford an evidence of fatigue cracking caused by bending vibration in the horizontal plane. Dynamic displacement measurement was carried out to prove the bending vibration due to Kárman vortex generated by the coolant flow. The fracture surfaces were examined by micro-fractography. Although most

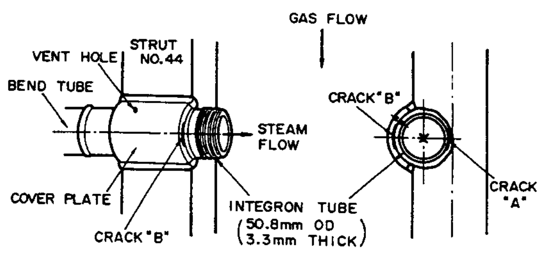
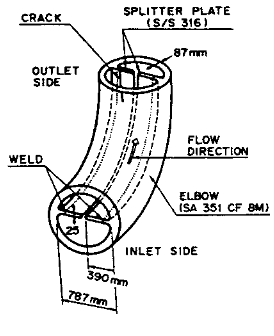


Fig. 3. Fatigue cracks at fillet weld toes.

of the fracture surface was covered by heavy scale, it was successful to find, at some spots of the fracture surface, well-defined striations, which propagated from the inside to the outside of the tube. In order to obtain information about the striation spacing versus fatigue crack propagation rate relation for the material of the tube, displacement-controlled fatigue tests of full-size tubular specimens of the same material as that of the damaged tube were carried out in a laboratory. As a result of comparative examination of the fracture surfaces by service failure and by fatigue experiment, it was concluded that the repeated load at a relatively high frequency caused the service fatigue failure. High frequency bending loading could be caused by resonance when the frequency of the Kárman vortex generated by the gas flow matched with the natural frequency of the supported tube. Superimposed on this loading might be low-cycle thermal stress cycling, which was possibly caused from the restraint imposed by the rigidity of the supporting structure.

To prevent the resonant vibration of the low-pressure superheater tubes, chains with weight at the end were hung through the gaps of the tube bundles and the tubes were fixed. The restraint of the tubes by the supporting structure was released a little, and thermal stress cycling on the tubes was reduced by making them movable in the axial direction.

The next example of fatigue failure due to resonance vibration is fatigue cracks that occurred in splitter plates in pipe elbows in the primary coolant system in PWR plants [4]. As shown in fig. 4, splitter plates were butt-welded to extruded ribs on the inside surface of an elbow, which was assembled just before the



Sketch of Elbow with Splitter Plate

Example	Plate	Crack Length (mm) Upper Face	Crack Length (mm) Lower Face	Average Crack Length (mm)
I	A	190	15	103
	B	360	9	185
II	A	0	0	0
	B	320	220	270
III	A	0	0	0
	B	340	50	195

Crack Length in Splitter Plate

Fig. 4. Fatigue crack initiated at toe of fillet welds joining splitter plate and elbow.

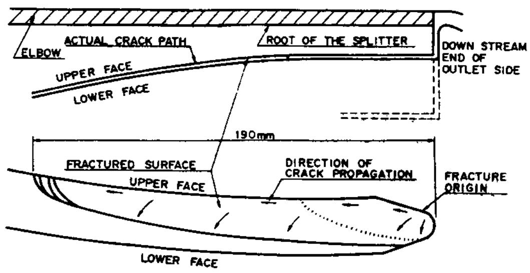


Fig. 5. Schematic view of fracture surface of splitter plate.

primary circulation pump. The main purpose of the splitter plate is to regulate possible turbulence of water flow, which goes into a pump.

The splitter plate is generally made of a single, solid and curved plate, of which both edges are welded to each inside surface of the elbow. But it consisted of two separate curved plates in the case where service fatigue failures were experienced. This variation was probably made under an intention of easier fitting work of the plate. The problem was a decrease in natural frequency because of weaker bending rigidity. No attention to possible vibration of the tail of the plate due to a decrease in the natural frequency was paid in the design stage, because of no experience of fatigue cracks in the previously constructed solid one-plate-type structure.

The splitter plates and the elbows were respectively made of stainless steel of type 316 and cast stainless steel of type 316. The table in fig. 4 lists examples of fatigue crack lengths in plate A and plate B, which make a pair in an elbow. Fatigue cracks of this kind were observed in almost all splitter plates of the similar design. In the worst case a fractured fragment plunged into the pump. Figure 5 shows a sketch of the typical fracture appearance. In all cases, the fatigue crack started at a toe of the welds and propagated mainly in the longitudinal direction, with a tendency of curving to the centre side free edge of the splitter plate. Microfractographic examination revealed evidence of crack propagation in a very high cycle fatigue regime.

In order to simulate the hydraulic condition, hydraulic experiments were conducted, setting a 1/5 scale splitter model in a U-shaped long water tunnel to confirm the fluctuating stress at several points close to the fillet weld toe due to resonance vibration of the splitter plate end, and to obtain the relation between resonance frequency and flow velocity, the vibration stress in resonance condition, and others. Figure 6 shows the results of measurements of the fluctuating

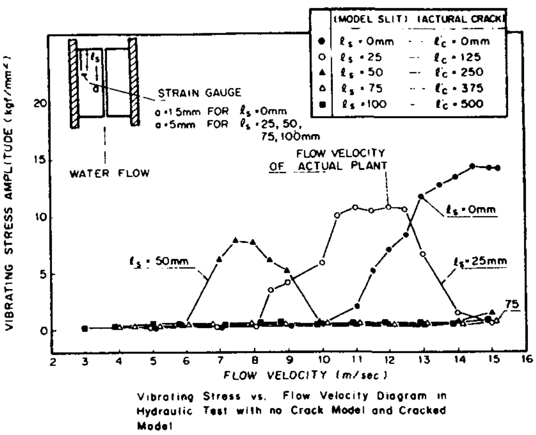


Fig. 6. Vibrating stress amplitude against flow velocity.

stress due to vibration of the tail of the splitter plate. The experiment was carried out with two kinds of models, one free from cracks and the other with a crack-like slit along the fillet toe. As shown in the figure, the splitter plate with a slit of 25 mm in length, which corresponds to 125 mm long crack in the actual case, became apparently into the resonance condition if the flow velocity reached 11 to 12 m/s, which roughly correspond to the real flow velocity in an actual elbow. In the case of no slit condition, the resonance phenomenon may be observed at the flow velocity of approximately 15 m/s.

The fluctuating stress at a point close to the crack tip changes as a function of flow velocity and slit length. If the splitter has no slit, the value of the fluctuating stress is about 150 MPa, which is high enough to initiate a fatigue crack [5]. The results of the experiment indicated that the crack in the splitter might be caused by the resonance with von Kármán's vortex street occurring in the downstream edge of the splitter, and that if the crack propagates beyond a certain length, the vibrating stress decreases considerably, because the natural frequency of the splitter plate begins to decrease and it gets out of the resonance range.

Extensive experimental and analytical studies, such as fatigue crack propagation tests of curved plates, FEM fracture mechanics analysis on crack path, and so on, provided information available to explain the behaviour of fatigue failure of splitter plates. Based on further experimental results that showed little advantage of the splitter plate for the suction efficiency of the pump, it was decided to remove all the splitter plates from elbows in the primary circulation loop. For this purpose a remote-operating cutting and removing machine was developed, and removing work was executed.

4. Fatigue failure due to forced vibration

A vibration fatigue failure not by Kárman vortex, but by forced vibration due to impeller revolution, was first experienced in 1984 in a BWR plant. In the course of the start-up run after a periodic exchange of mechanical sealing, a re-circulation pump produced abnormal noise from the inside. The pump was immediately dismantled, revealing that about one-half of the bearing ring plate welded to the hydrostatic bearing housing had been removed, and it rested on the upper shroud of the impeller. A schematic view of the pump is shown in fig. 7. As shown in fig. 8, it is the purpose of the bearing ring plate to form an orifice, a narrow gap between the bearing ring plate edge and the inside surface of the pump casing. Due to the passage resistance at the orifice, the amplitude of fluctuation of hydraulic pressure caused by the revolution of the impeller blades is reduced considerably, and the hydraulic pressure with little pulsation works as hydrostatic bearing action. The bearing ring of ASME A-240-type 304 stainless steel is fillet-welded on the cir-

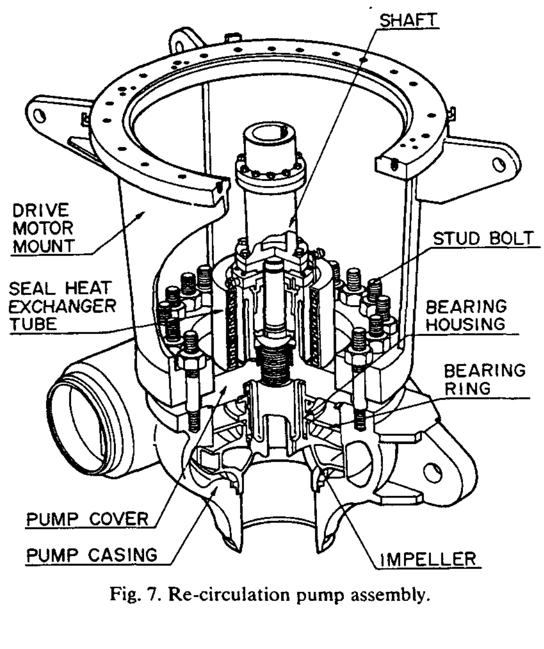
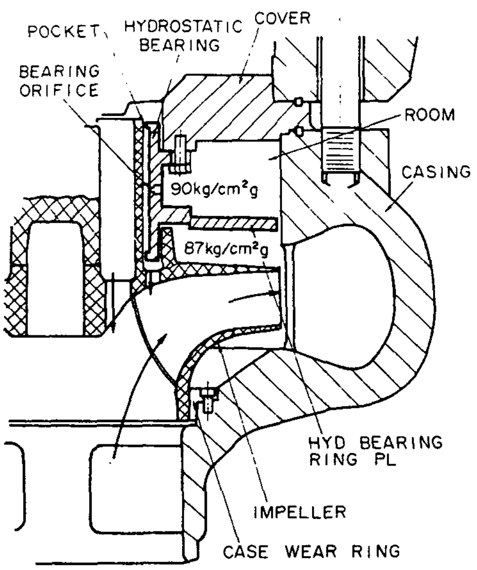


Fig. 7. Re-circulation pump assembly.



PUMP INLET : 72 kg/cm² g
OUTLET : 89 kg/cm² g

Fig. 8. Vertical section of recirculation pump.

cumferential edge of the solid rib of the bearing housing of ASME A-351, Gr. CF3R.

It was assumed from the observation of the fracture surface and fracture path that a crack might have started at one or multiple points located in the centre portion of the cracked circumferential welded joint, and propagated in both clockwise and counterclockwise circumferential directions along the joint until the critical point where the crack was forced to curve out to the radial direction, as shown in fig. 9.

Figure 10 shows the cross-sectional weld shape at several sections. Despite the welding requirement of full penetration in upper and lower edge preparations, as shown in fig. 11, lack of penetration is obviously found in the greater part of the girth length of the fractured welded joint. In particular, very poor penetration, approximately one-half depth penetration, was observed in the range of about 30 degrees of the angle at the circumference. These joints were welded by manual arc welding. A quantitative examination on the percentage of lack of penetration, which is the ratio of the lack of penetration depth to the designed penetration depth, was made, showing the following results: 50% at a maximum and 23% on average for the upper side fillet joint, and 3% at a maximum and 1% on average for the lower side fillet joint. Microfractographic examination was not successful on the fracture surface of the welded joint due to the difficulty of removing the oxide tightly adhered to the fracture surface, but it was successful on the fracture surface of

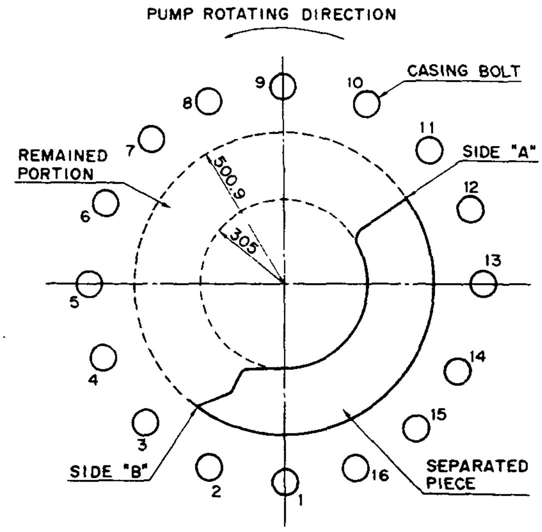


Fig. 9. Sketch of removed bearing ring plate.

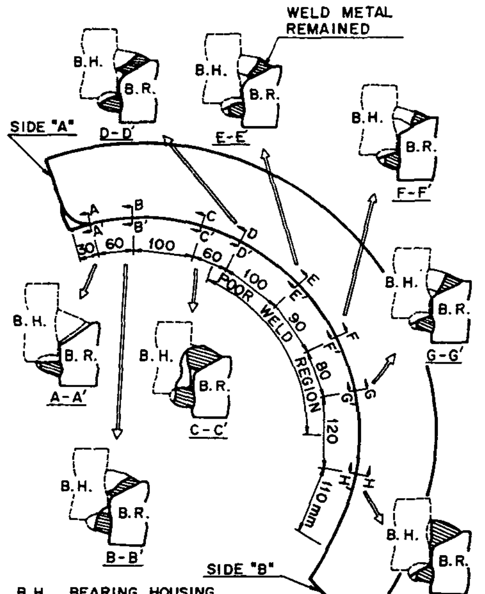


Fig. 10. Cross-sections of bearing ring welds.

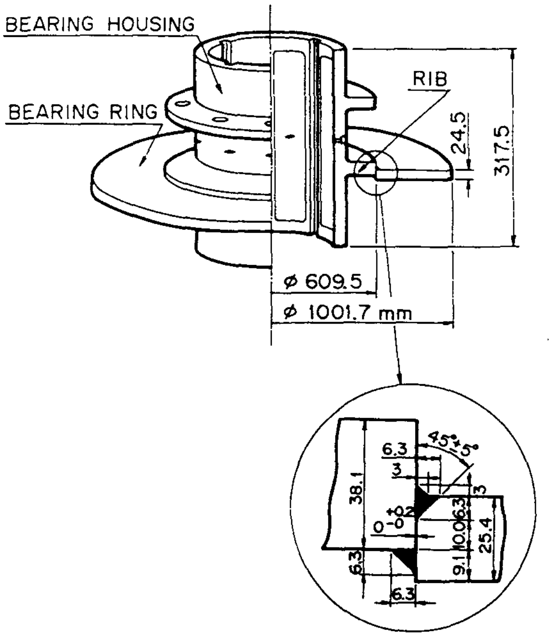


Fig. 11. Welding design of fillet welded joint in hydrostatic bearing.

cracks radially propagated in the base metal, exhibiting well-defined striation except on the scratched surfaces. No problem was confirmed with regard to chemical contents and hardness of the weld metal, HAZ and base metal.

An extensive investigation was made further to show that the stationary plus pulsating stresses under the operating condition are too low to initiate fatigue cracks at the root of the weld metal, if the upper and lower fillet weld joints are fully penetrated as designed. Thus, the conclusion was that the fatigue failure occurred because of the considerable amount of lack of penetration in the fillet welded joint. The assembly of the hydrostatic bearing was newly fabricated, and the new bearing ring was carefully welded by automatic welding.

Three years and eight months after the experience of fatigue failure of the hydrostatic bearing ring mentioned above, quite similar cracks were found in the course of a periodic in-service inspection of the same hydrostatic bearing ring welds. Fortunately the bearing ring did not remove off from the bearing housing in this case, but one-third of the circumference of the upper side fillet welds was cracked, and the lower side fillet-welded joint was fully cracked along the total girth length of the circumference, as shown in fig. 12. An examination of the percentage of lack of penetration again revealed poor penetration. The results were 31% at a maximum and 26% on average for the upper side fillet joint, and 7% at a maximum and 4% on average for the lower side fillet joint. Fractographic examination of the fracture surfaces showed striations that propagated from the weld root to the outside of the weld metal.

The third case of similar fatigue cracking in a hydrostatic bearing ring plate occurred in January 1989 at another BWR plant. The bearing ring was completely removed from the bearing housing, and was separated into two parts, approximately 4/5 of the circumference ring and 1/5 of the circumference ring. A serious problem in this case was that fragments of the ring, bolts, washers, and a great quantity of metal powder, produced mainly by abrasion between the removed bearing rings and the upper shroud of the impeller, were transported by the re-circulation coolant flow into the inside of the reactor pressure vessel. The total weight of such transported powder and fragments was assumed to reach about 30 kg.

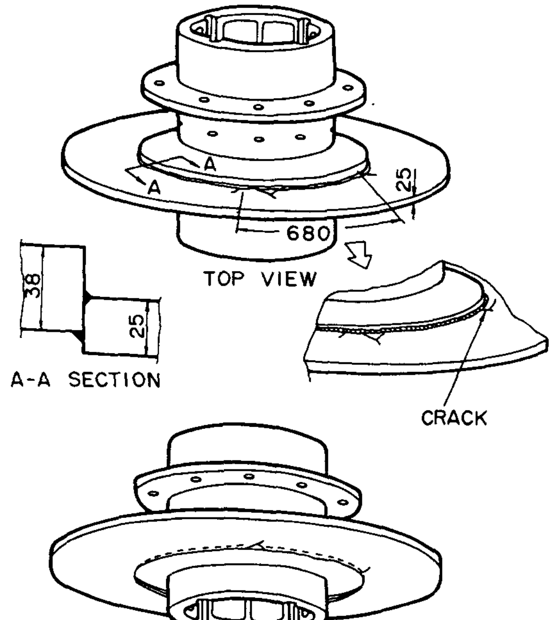


Fig. 12. Fatigue cracks in welds of bearing ring plate.

The basic cause of all three cases of fatigue failure of the hydrostatic bearing ring welds was the lack of penetration in the fillet welded joints. The problem is a possibility of leaving poor penetration, if the welding conditions are not appropriate, and if the welder's skill or the setting and performance of the welding machine are inadequate. It may be worthy of mention that the fillet joints of the first case were welded by manual arc welding, and the fillet welds in the second and third cases were welded by automatic machine welding. The reason why a remarkable lack of penetration was found only in the range of about 1/4 of the circumference of the fillet joint in the first case may be due to unstable scatter of skillfullness of the welder. For the second and third cases, it can be said that the results are quite contrary to the preconceived idea that machine welding should be much better than manual welding in providing uniform quality of welds. As mentioned above, the lack of penetration was localized in the first case, and on the contrary the lack of penetration was not localized but uniformly distributed in the upper side fillet welds in the second and third cases. In other words, a lack of penetration of roughly equal depth was observed over the total circumference of the upper side fillet welds. According to the investigation of the causes of the lack of penetration, it was proved that such uniformly distributed lack of penetration might be formed by a higher feeding speed of welding wire, incorrect setting of the welding machine against the edge preparation, and an over-snipped corner edge of the root of edge preparation.

A lesson obtained from the failures mentioned above is that use of a welded joint, whose soundness is difficult to check by non-destructive inspection, should be avoided in a strength member. Following this philosophy, all hydrostatic bearing rings in all BWR plants were replaced, as a countermeasure for recurrence prevention, either by full penetration butt welded type or monoblock casting type, shown in fig. 13.

It may be generally said that the piping system is usually subjected to vibration induced by the connected motor, self-exited vibration, parasitic vibration, and others. Often experienced is resonance-induced fatigue failure of a short pipe with a top-heavy element. One example of such phenomenon was experienced in a PWR plant, which was under service operation at the rated output. During the service, the rise of water level was found abnormal in a sump pit, suggesting possible leakage of the coolant in the container vessel. A leakage occurred from a crack at a toe of welds joining a vent pipe to a charging line pipe in the chemical and volume control system, as shown in fig. 14 and fig. 15.

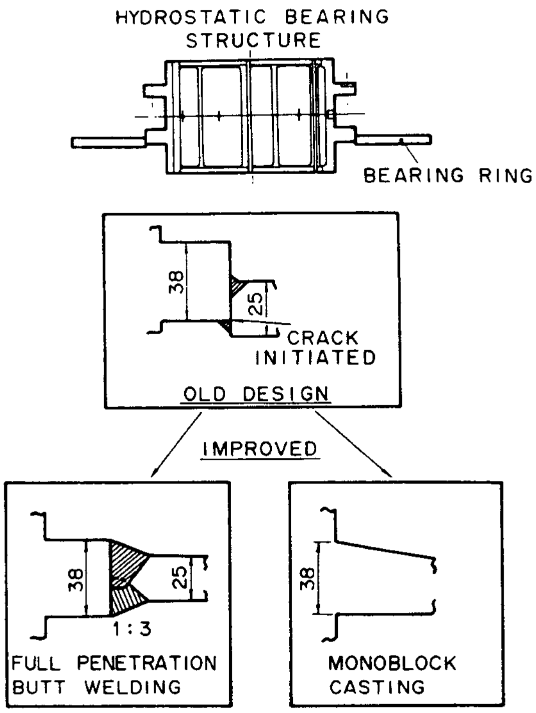


Fig. 13. Old and improved designs of hydrostatic bearing structure.

Figure 16 shows a detailed sketch of the valve and the pipe. The crack was extended along the weld toe up to a length of 35 mm. The cracked location is roughly the opposite side of the vent valve. The leg length of the socket welds was 7.5 to 8.5 mm, which is a

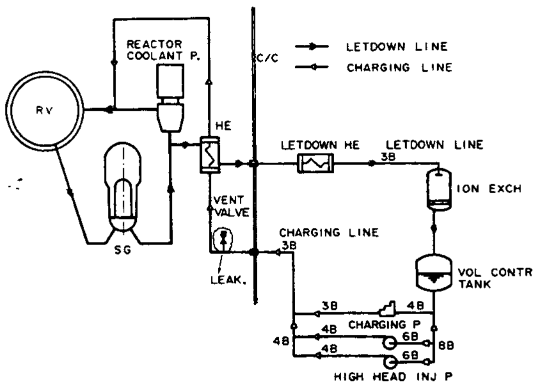


Fig. 14. Chemical and volume control system.

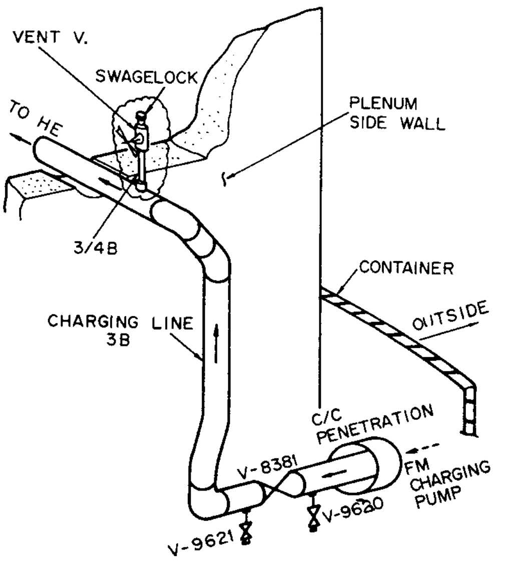


Fig. 15. Leakage from vent valve in charging line in chemical and volume control system.

normal size. The surface of the welds was ground smooth, and unsoundness, such as undercut and pits, was not observed on the welds. Fractography revealed that two main cracks started at two points on the weld toe line and propagated in thickness direction by forming shell marking. These two cracks coalesced, and finally penetrated to the inside surface of the pipe. Well-defined striations were observed on some spots close to the crack arresting contour line. As far as the starting and propagating zones are concerned, well-defined striations were not found, showing only non-characteristic and smooth topography. It has been recognized that such a non-characteristic and smooth fracture surface can be formed by very high cycle fatigue with very low fluctuating stress.

The length of the pipe was initially designed to be 180 mm. But, after fabrication a field engineer found that the handle operation of the valve was not easy due to the level of footing. Then, without consultation or discussion with the design engineer, the pipe length was extended by the field engineer on his own judgement. It seems that he paid no attention about a decrease in resonance frequency due to lengthening of the pipe. This case of failure seems to provide us a valuable lesson that a lack of communication or discussion between design and fabrication engineers may result in a possibility of unexpected failure. Finally the straight pipe was shortened as shown in fig. 17.

Figure 18 shows another example of service fatigue failure that occurred in a different pressure measuring instrument line for a primary re-circulation line due to the resonance vibration of the piping system. As shown in the dotted line frame labelled "initial," the structural component was top-heavy and the rigidity of the horizontal pipe branched from the tee was not high enough to prevent oscillation of the pipe-valve system. Details of the weld design of the set-on and socket welded joints are shown in fig. 19. Leakage occurred

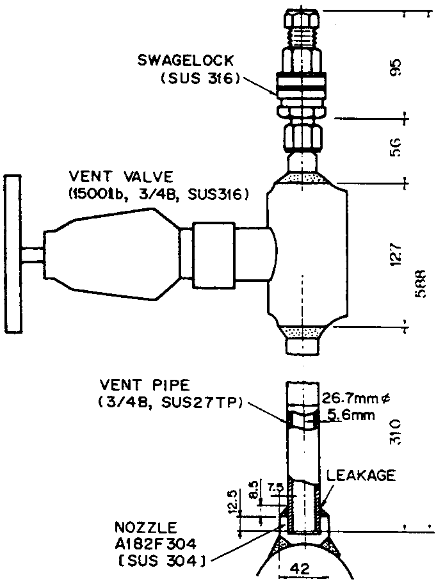


Fig. 16. Vent valve and connecting pipe.

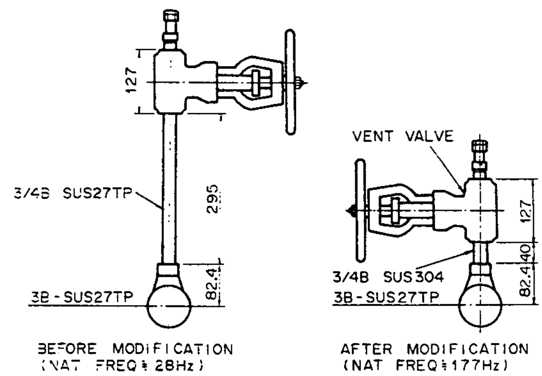


Fig. 17. Vent valve piping before and after modification.

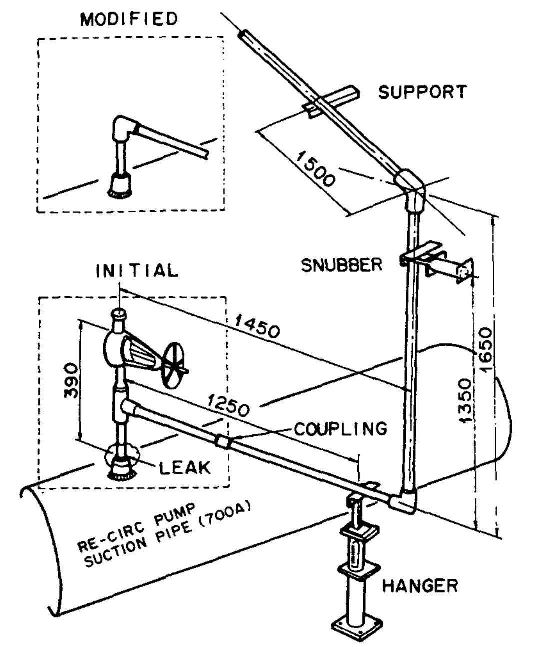


Fig. 18. Fatigue failure in different pressure measuring instrument lines.

from a crack, which initiated from a toe of the welds. Microfractographic examination showed well-defined striation on the fracture surface. The valve was temporarily assembled only for the trial operation. Although the valve was not used after the trial operation, it was left as it was, while the opening of the pipe downstream of the valve was stopped by a welded plug.

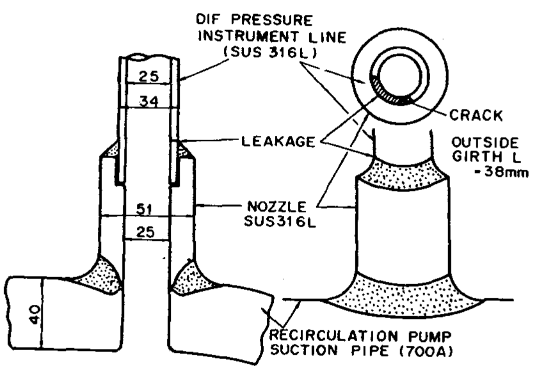


Fig. 19. Details of welded joints.

OD (inch)	THICK (mm)	MATERIAL
3/4B	3.9	SUS316LTP

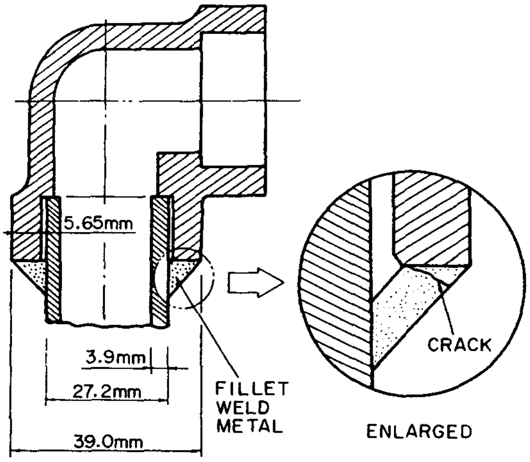


Fig. 20. Fatigue crack started from root of socket welds.

As a countermeasure to prevent recurrence, the assembly was modified, as shown in the left upper dotted line frame labelled "modified" in fig. 18, by removing the valve.

Several cases of fatigue failure that started from the root or the toe of fillet welds are further presented in the following. An example of a fatigue crack that started from the root of socket welds is shown in fig. 20. The elbow was welded tò a discharge valve of a PLR pump. Striations were found on the fracture surface. Figure 21 illustrates a fatigue crack initiated at the toe of the fillet weld in a mechanical seal line for a high-pressure condensate pump. In this case the recognition of insufficient fatigue strength of a 27.2 mm outer diameter tube resulted into a modification, shown in fig. 22. An example of a vibration-induced fatigue crack that started at the root of a socket weldjoint is shown in fig. 23. It may be noteworthy that the structure, where the fatigue crack was experienced, is also top-heavy, and with low bending rigidity, which means low natural frequency. In the case of a top-heavy structure of a piping, it may be better to restrict bending deflection by supports or to increase bending rigidity in order to increase the natural frequency higher enough than the global natural frequency of the piping, approximately 15 to 30 cycles.

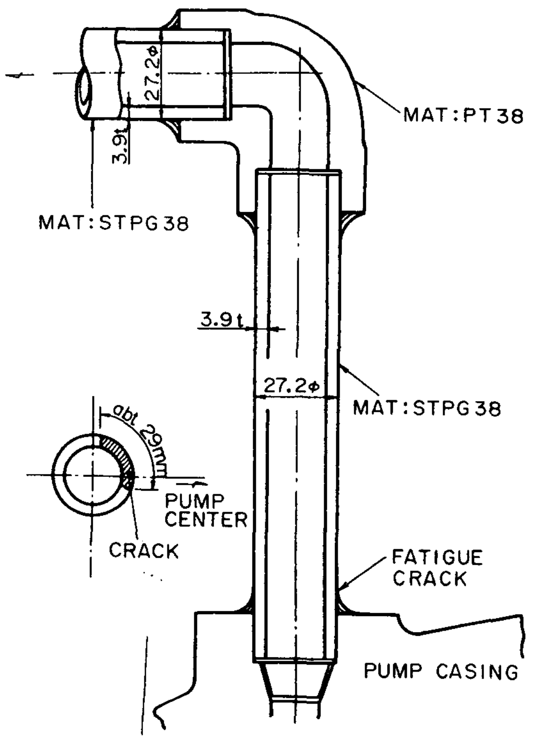


Fig. 21. Fatigue crack occurred in high pressure condensate pump mechanical seal water line.

Figure 24 illustrates fatigue failure of a valve component. A fatigue crack started from a corner of a cut-out in the yoke. Fatigue failure of a valve stem rod is shown in figs. 25 and 26. Such fatigue failures are due to vibration of a valve disk, which is forced to vibrate by by-path turbulent flow, when the disk is pulled up.

The final example of vibration-induced fatigue failure is shown in fig. 27 and fig. 28. In this case a lack of root penetration of the socket welds contributed to the initiation of the fatigue crack.

5. Fatigue failures due to thermal fluctuation

It was around 1977 when thermal fatigue cracks were experienced at the crotch corner of a feed water nozzle and at edges of circular holes in a feed water sparger in BWR reactor vessels (see fig. 29). The former crack occurred by the following mechanism:

(1) Leakage of low-temperature feed water through a gap between the feed water sparger nozzle and the thermal sleeve.

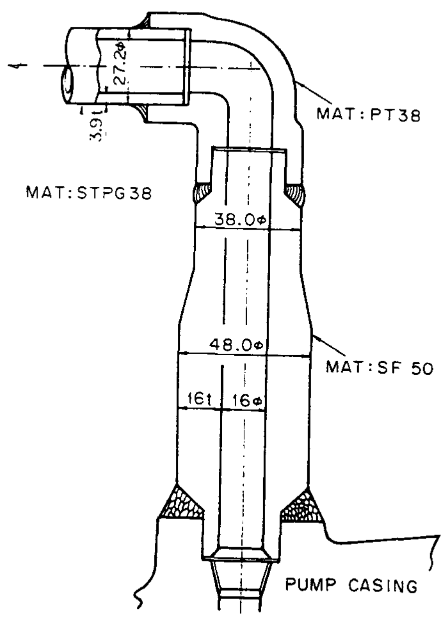


Fig. 22. Repaired high pressure condensate pump mechanical seal water line.

(2) Mixing of cold feed water with hot RPV water at the feed water nozzle crotch corner.
(3) Fluctuation of thermal stresses at the nozzle crotch corner.
(4) Initiation of thermal fatigue cracks.
(5) Crack growth due to global stress fluctuation during RPV start-up and shut-down.

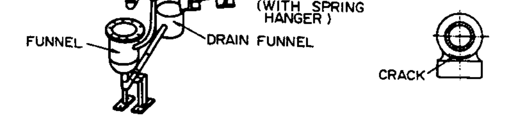


Fig. 23. Vent piping for re-circulation pump discharge valve.

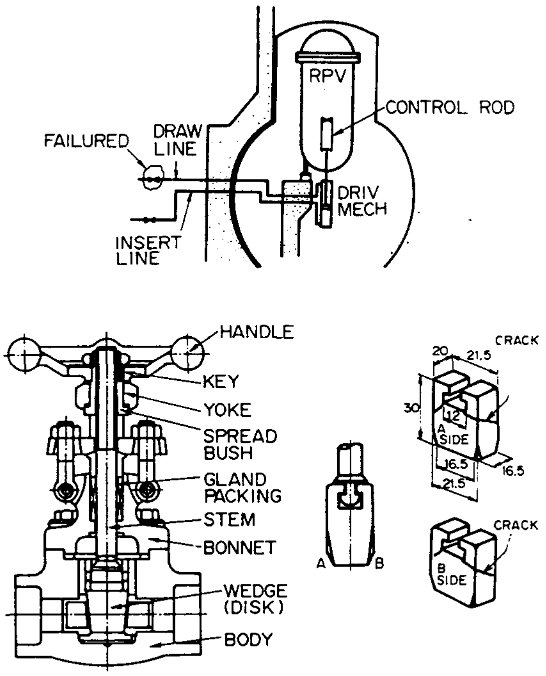


Fig. 24. Fatigue crack in yoke of valve disk (CRD hydraulic system).

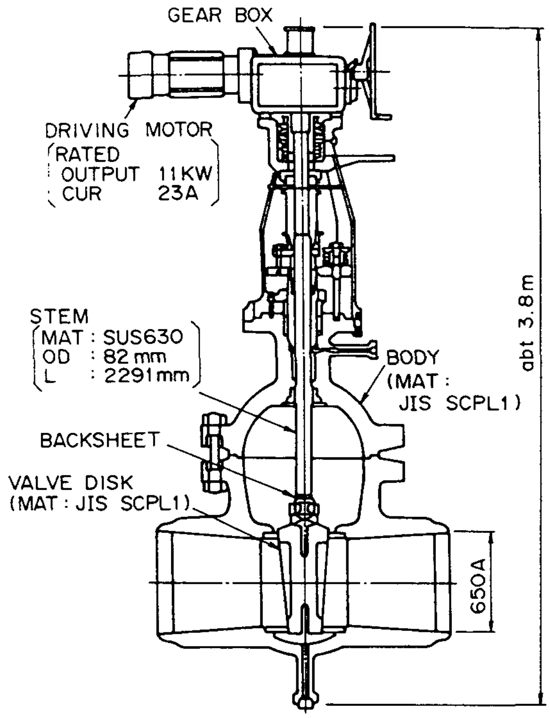


Fig. 25. Section of main steam stop valve.

Cracks at the edge of the circular hole in the sparger result from the thermal fluctuation due to mixing of cold water and hot water at the circumferential edge of the hole.

Improvement was made as shown in fig. 30. The fitting was changed from a loose fitting to a tight fitting so as to allow no leakage of low-temperature feed water. The opening of the sparger was improved from a flat circular hole to an elbow nozzle, so as to jet out cold water.

6. Fatigue failure due to thermal stratification

Recently an interesting and extraordinary fatigue failure, which was experienced for the first time in Japan, occurred in a PWR plant. Because of an increase in flow rate to the containment sump pit, the plant was brought to a shut-down for inspection. Leakage was found at the butt welded joint between an elbow and a straight pipe, which was welded to the first isolation valve in the A-loop residual heat removal system piping, as shown in fig. 31. A sketch of the

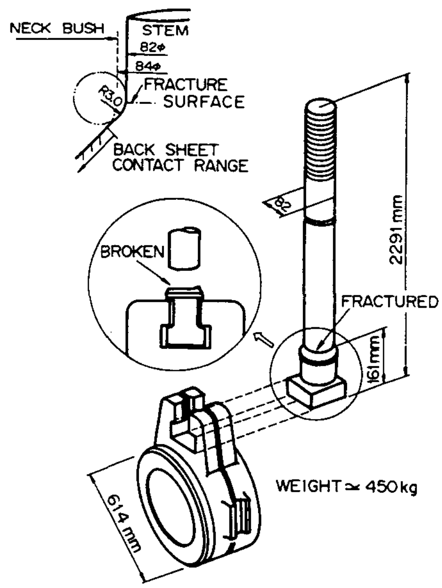


Fig. 26. Details of broken stem and disk.

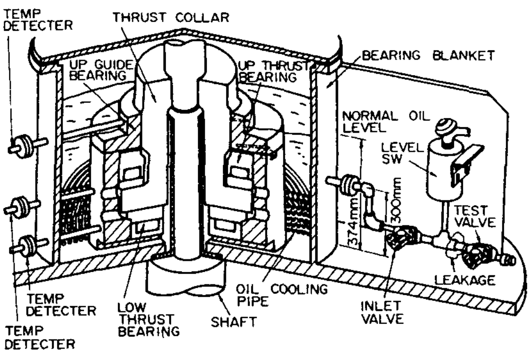


Fig. 27. Bearings and oil level gauge piping.

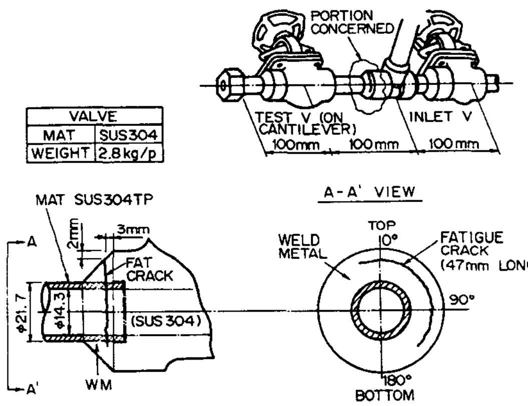


Fig. 28. Fatigue crack started from root of socket welds.

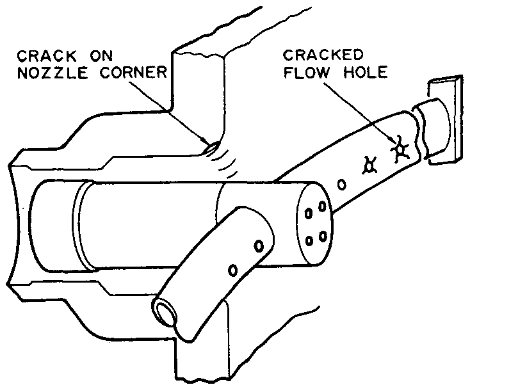


Fig. 29. Thermal fatigue cracks on nozzle corner and sparger hole edge.

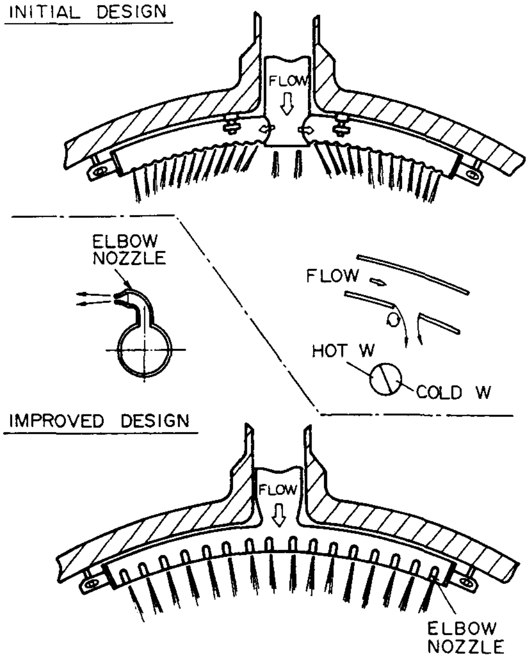


Fig. 30. Initial and improved designs of feedwater sparger nozzle.

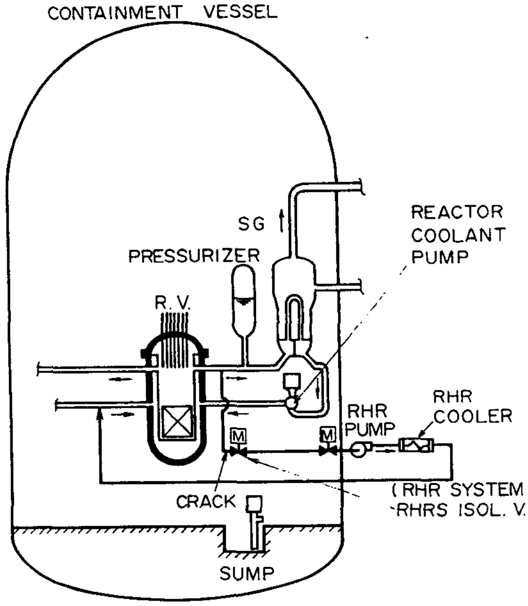


Fig. 31. Location of cracking in RHR piping.

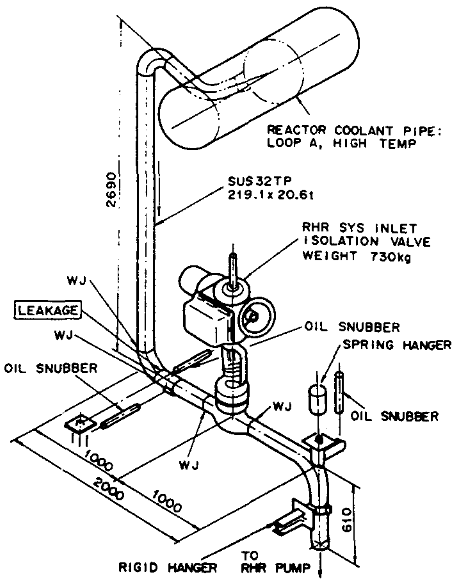


Fig. 32. Residual heat removal system piping.

layout of the pipes and valve is shown in fig. 32. An examination showed, as shown in fig. 33, that multi-site cracks initiated on the elbow side toe line of the welded joint, and propagated through the weld metal

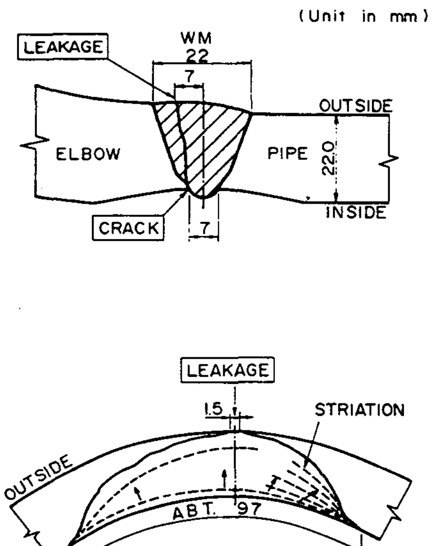
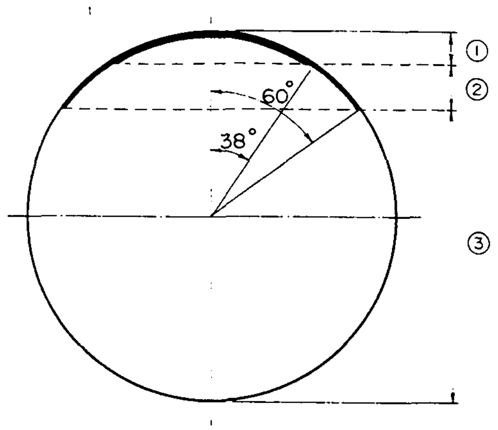


Fig. 33. Details of welded joint and fatigue crack shape.



REG.	COLOR	THICK.(μm)
1	DARK BROWN	2.8
2	BROWN	0.3~1.0
3	LIGHT BROWN	≦ 0.5

Fig. 34. Oxide on inner surface of straight pipe.

toward the outer surface by coalescing to form a semi-elliptical crack shape. The crack length was about 97 mm in circumference on the inner surface and 1.5 mm on the outer surface. The leaked position, the final opening point of the crack, was located just on the top generatrix, which is an extension of the generatrix passing the crotch point of the elbow concerned. Fractographic examination proved the existence of striations of spacing 0.3 to 1.0 μm.

As a result of observation of the inner surface of the elbows and the horizontal straight pipes illustrated in fig. 32, it was found that the inner surface of the straight pipe, which was welded both to the upstream side flange housing of the isolation valve and to the cracked elbow, was covered by oxide, showing classification into three regions depending on colour and oxide thickness, as shown in fig. 34.

In order to get information about the vibrating stress amplitude at the elbow concerned under the operation of the A-RHR pump and reactor coolant pump, vibration measurement was made to calculate the vibrating stress. The results were that the stress amplitude was very low, less than 1 MPa, even if the piping system was brought into resonance condition by fluid-induced vibration.

The first and the second isolation valves in the A-RHR line, and the first isolation valve in the B-RHR line, all of which are 8 inch, motoroperated, wedge gate type and limit switch controlled valves, were disassembled and examined to find the following, which indicated leakage through the gland packing:

(1) Deposit of oxide on the upstream side surface of the disk. The appearance was classified just same as the inner surface of the straight pipe illustrated in fig. 34: dark brown on the upper side, and a normal appearance of the downstream side surface of the disk.
(2) Stripping marks of boric acid on the outer surface of the lantern ring installed in the gland packing section.
(3) Wet and rusted appearance of the inner surface of the leak-off piping.

No abnormality was found by Vickers hardness distribution measurements and chemical analysis. Finally it was provisionally concluded that the fatigue failure concerned was caused not by fluid-induced mechanical vibration, but probably by thermal cycling. This conjecture was supported by the fact of a difference in colour and thickness of the oxide deposited on the inner surface of the straight pipe and traces of the leakage through the gland in the isolation valve. To confirm this conjecture, two kinds of thermal stratification hydraulic tests were conducted:

(1) Thermal stratification proving tests at ambient condition using a 1/1.7 scale model of the RHR piping system, which was made of transparent acrylic resin. The leaking valve was modelled by a throttle valve in a leak-off pipe.
(2) High temperature full scale model tests using a real-scale mock-up assembly of the RHR piping system, including the isolation valve. The valve gland was modelled in the same manner as the ambient temperature tests. The main purpose was to measure the temperature distribution through the horizontal pipe as a function of the leak rate.

Thermal stratification was obviously observed. Also found were the thermal stress range at the top generatrix of the pipe, which was enough amount to cause high cycle fatigue, the relation between the leak rate and the cycling speed of repetition of thermal stratification, and others. The thermal stratification was concluded to be produced by the mechanism described below (see fig. 35).

(1) Initial gap between the disk upper edge and the valve sheet of the upstream side.

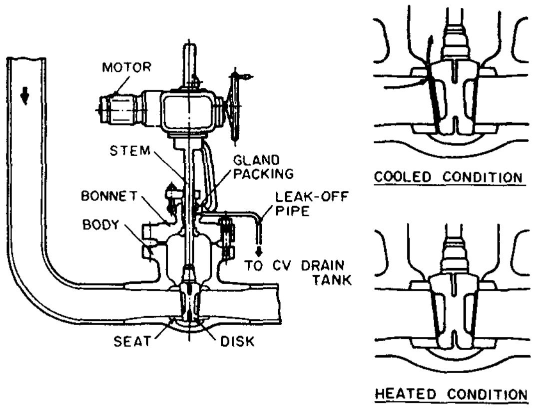


Fig. 35. Mechanism of repeated thermal stratification.

(2) High temperature primary coolant leak through the upper gap.
(3) Expansion of heated disk and closure of the gap. Leakage stops.
(4) Cooling down of the stagnant coolant due to radiation, and shrinkage of the expanded disk.
(5) Restart of coolant leakage through the gap. Return to step 3.

7. Concluding remarks

Service fatigue failures in commercial light water reactor plants in Japan were reviewed, presenting typical and interesting cases. Many valuable things can be learned from the analysis of these failures. The following four items may be the most noteworthy factors in the development of a service fatigue failure.

(1) Lack of communication and discussion between designers and fabrication engineers.
(2) Lack of knowledge of possible fatigue failure and poor consideration about the effects of welding residual stresses.
(3) Lack of consideration on possible vibration in the design and fabrication stages.
(4) Lack of fusion at the root of fillet welds (incl. socket welds).

With regard to the second item, systematic research activity on the subject of cyclic relaxation of welding residual stresses due to fatigue loading should be necessary for realistic explanation of the fatigue failure mechanism and for reasonable fatigue design of welded joints.

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