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OCCURRENCE OF SUB-SYNCHRONOUS VIBRATION IN A MULTISTAGE TURBINE PUMP AND ITS PREVENTION

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1. Introduction

It is evidently because of the critical importance the prevention of occurrence of vibration for high-load rotary machinery assumes in ensuring reliability of a plant as a whole that so many investigations and studies have been done on the matter. Regarding the sub-synchronous vibration with which we are concerned here, the self-excited vibration such as due to oil whipping, forced vibration such as due to propagating stall, and non-linear vibration such as due to sub-harmonic resonance have been studied quite well. However, we note that the case of self-excited vibration, whose frequency has been found to be generally less than one half the revolution number, is rather complex, because there can be any number of exceptions. For example, when the labyrinth seal is responsible for the self-excited vibration, its frequency is often entirely unrelatable to the machine revolution, and one of us (S. S.) has found for rotating hollow shafts that are partially filled with liquid that the frequency is over one half of the revolution[1)].

As for the pump vibration, on the other hand, the vibration due to non-steady fluid force that arises mainly in the partial flow domain[2,3)] and the kind that is induced by the fluid force acting within propagating stall are by no means uncommon.

In this paper, we intend to present and discuss a peculiar vibration that we have encountered in a multistage turbine pump. The pump in question was serving an active power plant in a part that was a veritable "heart" of the entire plant, and the major vibration component was of about 80% frequency of the revolution.

At first, the propagating stall was thought to be responsible, but the absence of higher harmonics made this presumption untenable. Or else, even though we were aware of previous reports that dealt with seemingly similar mechanical vibration troubles[4,5)], we found these investigations to offer no clear diagnosis nor suggest simple remedial measures.

It is for these reasons that we have attended the problem from the start. Through fundamental tests and experiments, we have been able to gain several insights into the nature of this anomalous vibration, determine whereat the fluid force that caused such a vibration was given rise to[6,7)], as well as devise countermeasures that turned out to be quite effective. These are, then, the subjects of the present paper.

2. Service History of the Pump

The pump that gave rise to anomalous vibration was a high pressure multistage turbine pump serving a seven-train, 1,000 MW combined cycle

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power plant. The duty of the pump was to transfer the feedwater from low pressure drum to high pressure drum for the high/low dual line steam generator of the gas turbine waste heat recovery unit. It is illustrated in Fig. 1 in a vertical cross sectional view, and its major specifications are given in Table 1.

Since it was commissioned in 1985, this power plant has served quite well as a "peak shaver" in the entire power network for its ease of rapid start-and-stopping and high efficiency over a wide load range. Therefore, the plant was often shut down when the total power demand was low, e.g., at nights and over weekends, so much so that, even though its annual operation time was no more than about 5,000 hours on average, the number of start-and-stopping amounted to about 100 an year.

This is a rather demanding service condition for any machine to operate under, and for this pump it is particularly so on starting, when the pump suction condition varies from 3 to 5 kgf/cm^2 and 100 to 130°C to 10 to 11 kgf/cm^2 and 180°C in only about one hour of time. To bear up to such a large and fast thermal expansion taking place within the pump, we had a highly skilled worker assemble the shaft seal using a precision formed gland packing made of Graphoil, yet a lifetime of one to one and a half year was all that could be achieved.

To ameliorate on the maintenance work, the shaft seal was changed to mechanical seal in 1988, whereupon the pump ran into a serious vibration trouble: the vibration, which was about 10 μm p-p on modification, became 156 μm p-p in ten days when the pump was forced to shut down. The accumulated operation time to stoppage was 130 h, in which the pump went through four start-and-stops.

The pump was restarted on temporary repair, which consisted in replacing substantially all the parts within the casing, including impellers, shaft, and wearing rings, with the reserves, but the vibration grew with time rapidly as shown in Fig. 2; the vibration level was deemed to have reached the danger lever in a week, after an accumulated running time of about 100 h and eight start-and-stops. The vibration situation at the time of this second shutdown is given in Table 2.

The vibration behaviors and the damage the pump sustained as found on opening inspection are as follows:

1) the main component of vibration was 40 to 42 Hz (i.e., about 80% of the revolution number), and its amplitude was far greater than that of the synchronous component (49.5 Hz);
2) no higher harmonics were found;
3) the amplitude increased with operation time;
4) it grew also with increasing flow rate, and the frequency, too, changed, though slightly; and
5) the stationary wearing rings were seen to have worn greatly, but only in the vicinity of their bottom parts, as illustrated in Fig. 3.

3. Fundamental Shop Testing

Trying to reproduce the accident, we have assembled a pump at a shop on a reserve pump of the identical dimensions with the damaged pump's impellers, shaft, wearing rings, etc., and run a series of vibration tests. From the histories of vibration changes and frequency analyses thus acquired, which are exemplified in Figs. 4 and 5, following observations were made:

1) the same anomalous vibration with about 80% of the synchronous frequency for the main component did also occur in this isolated pump;
2) even though the initial vibration amplitude observed at shop was smaller than that found on the site, this was thought ascribable to the better initial concentricity between the rotor and the casing realized at shop; this advantage was lost rapidly so that the vibration amplitude approached that which was experienced actually within a comparatively short period of time, however;
3) the vibration amplitude increased greatly with the rise of suction temperature, due most probably to aggravation of concentricity between rotor and casing; and
4) for the same suction temperature, the amplitude tended to grow with the elapse of operation time, due most probably to aggravating concentricity under progressive wear.

Based on these observations, we have deducted the following as the major mechanism of vibration:

1) the elimination of the gland packing, which used to work as a bearing to support the shaft also, has brought about increase in the natural deflection of the shaft, which in turn has brought about increase in the bearing pressure on the intermediate wear rings located in between the shaft bearings;
2) the wear rings that are used in the interstage seals are all provided with a groove so as to prevent seizing by foreign matter floating in the feedwater, so that its load bearing capacity was not large enough to start with for a job of hydraulic bearing;
3) for this reason, metallic contact took place at or near the wearing ring bottom to wear them according to the amount of shaft's natural deflection;
4) excessive wear of the wearing ring has resulted in excessive eccentricity between the stationary part and the rotating part; and
5) this brought about increase in the fluid force acting on the rotor, giving rise to self-excited vibration of the about 80% the synchronous frequency in question, which in its turn accelerated the wear, thereby constituting a vicious cycle of events.

The fact that the observed wear was mainly in the direction of static loading as shown clearly in Fig. 3 attests to this theory.

4. Remedial Measures and Their Effects

A) The First Countermeasure Taken

Having analyzed the cause of the anomalous vibration as described above, we have decided to form the wearing rings eccentrically so that their inner diameters should change in alignment with the natural deflection of the shaft, or, in other words, so that no eccentricity may arise between the rotating part and the stationary part at each interstage seal.

The performance of the pump thus modified is shown in Fig. 6-a, where we see that, although the vibration characteristics have been greatly improved as a whole, the 80% synchronous frequency still persists, though now with an amplitude that is much smaller than that of the synchronous component. Also, a tendency, even though slight, for the amplitude to increase with time was noted. For these reasons, we have judged this measure to be insufficient for the long-term operation expected of the pump on the site.

B) The Complementary Measure

It is known that in many cases what induces self-excited vibration for a rotating shaft is swirl of the fluid, and that the vibration caused by labyrinth seal can be prevented most effectively by preventing the swirl from flowing into the seal. Translated into the case at hand, this would mean that it is the swirling flow that leaks back into the suction direction from the tip of an impeller blade presumably at the accelerated peripheral velocity by the impeller shroud that should be reduced.

Thereupon, we have provided each impeller with a swirl breaker at the outer space of its shroud as depicted in Fig. 7. The result was rather remarkable: as shown in Fig. 6-b, the troublesome sub-synchronous vibration has disappeared completely. The pump modified thus and installed for the same duty as before has been working in the plant already for over two years now, giving off no sign of anomalous vibration of any kind.

Even then, we recognize one more point that remains to be clarified; according to Childs, who has analyzed the fluid forces generated outside of the shroud[6,7], such a force does not induce self-excited vibration directly, but his conclusion had the constant concentricity between the rotor and the stator as its base. This may mean that once the concentricity is lost, the same fluid force will become able to induce a self-excited vibration. In fact, we have observed during the shop testing that vibration did indeed grow when the concentricity was degraded on purpose. We intend to continue the study on this matter.

References

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2) P. Hergt and P. Krieger, "Radial Forces in Centrifugal Pumps with Guide Vanes," PIME, 1969-70, p. 101.

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4) H. F. Black, "Lateral Stability and Vibrations of High Speed Centrifugal Pump Rotors," IUTAM Symp. on Dynamics of Rotor, Lyngby, Denmark, Aug.., 1974.

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Table 1 Principal specifications of the pump

Type	Horizontally Split Type Turbine Pump
No.of Stages	9
Capacity	3.83 m3/min
Total Head	930 m
Rotaing Speed	2970 RPM
Moter Output	820 KW
Suction Pressure	2 -- 14 kg/cm2
Suction Temperature	Room Temp. -- 180 Deg.C
Pumping Liquid	Boiler Water
Shaft Seal	Gland Packing (GRAFOIL)

Table 2 Vibration characteristics of the as-damaged pump determined at the bearing housing

Pump Flow Rate (%)	Generator Output (MW per Train)	Pump's Bearing Housing Vibration			
		Synchronous Component		Sub-synchronous Component	
		Freq.(Hz)	Amp(μmP-P)	Freq.(Hz)	Amp.(μmP-P)
45	0	49.5	3.0	41.8	12.0
60	70	49.5	7.2	39.8	28.8
90	150	49.5	5.8	39.3	35.6
100	165	49.5	1.4	39.8	40.0

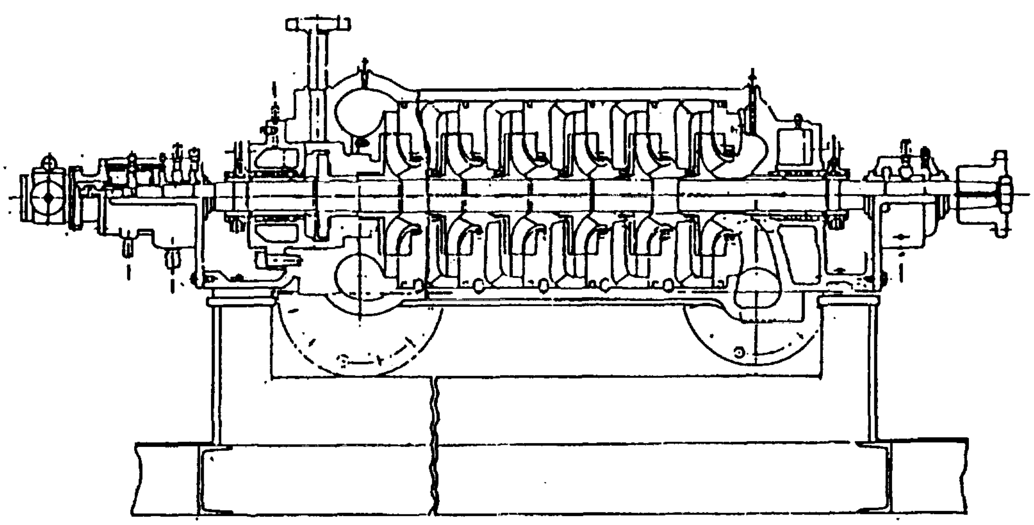


Fig. 1 Multistage turbine pump

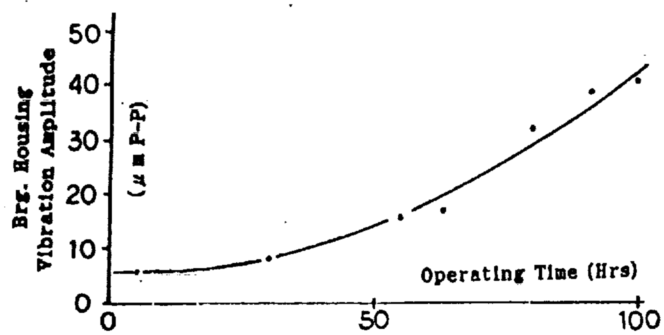


Fig. 2 Growth of vibration

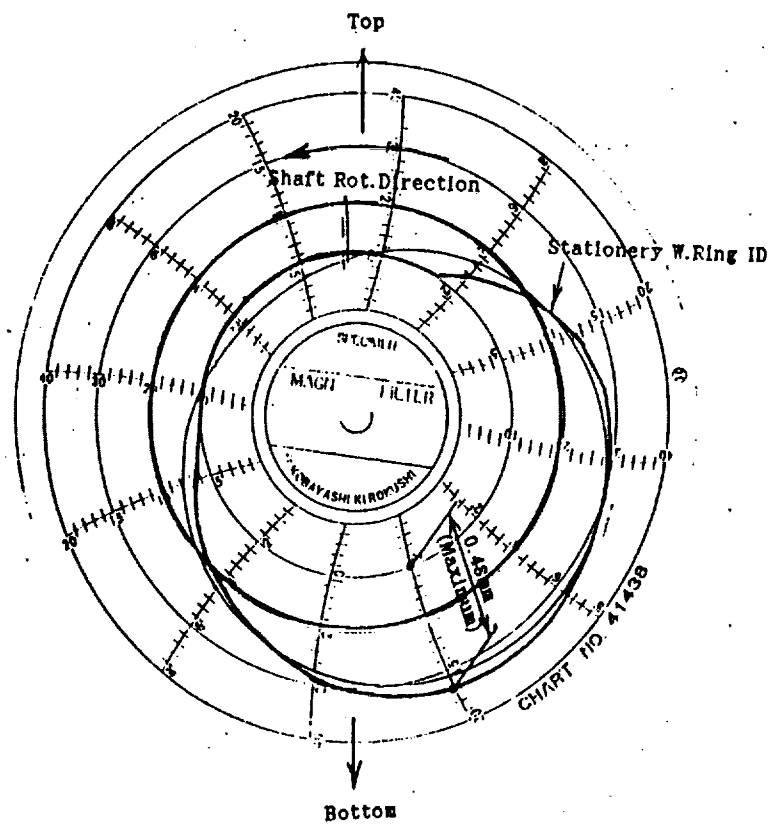


Fig. 3 Profile of a wearing ring as determined on pump shutdown

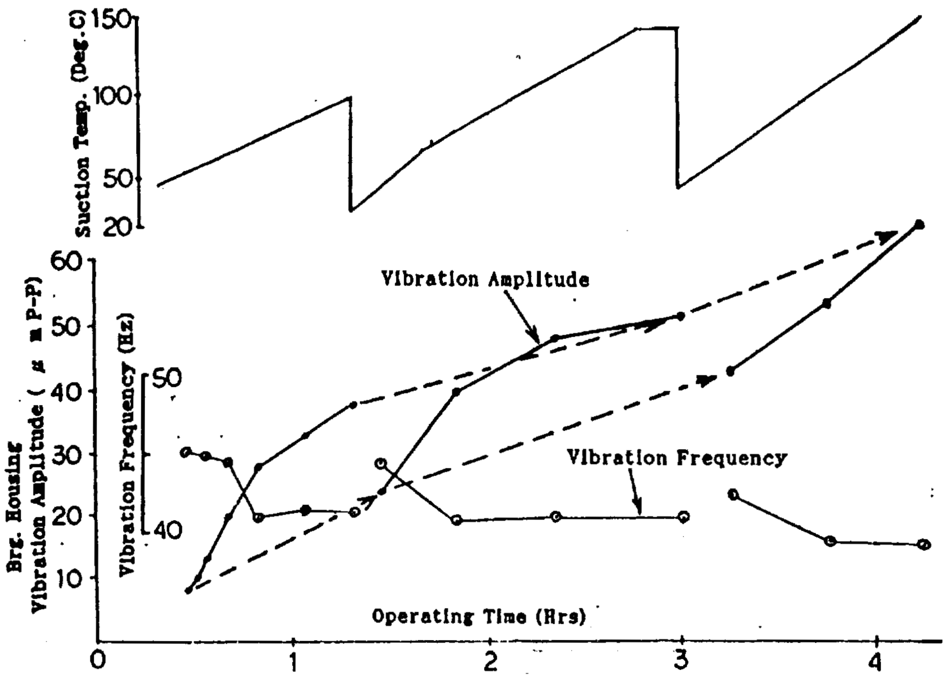


Fig. 4 Growth of sub-synchronous vibration

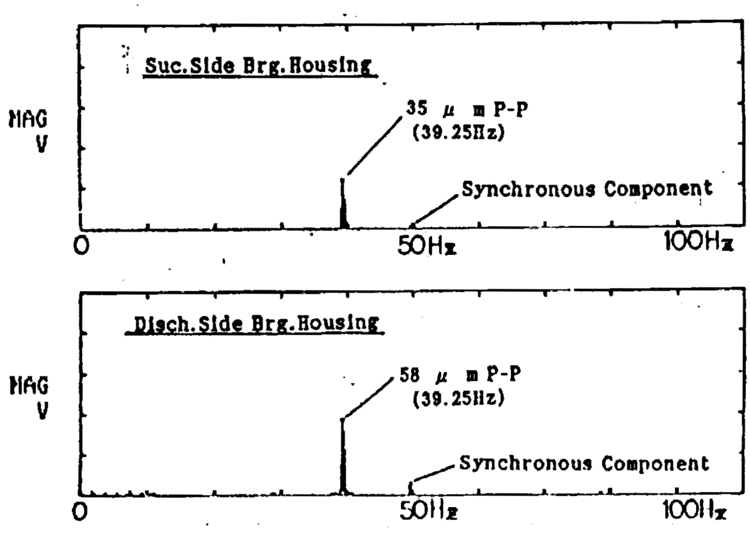
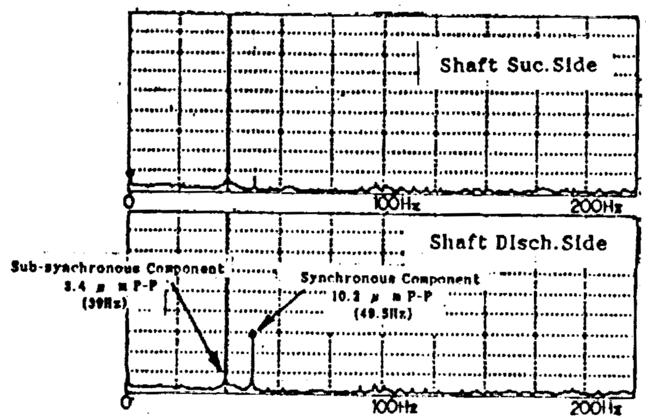
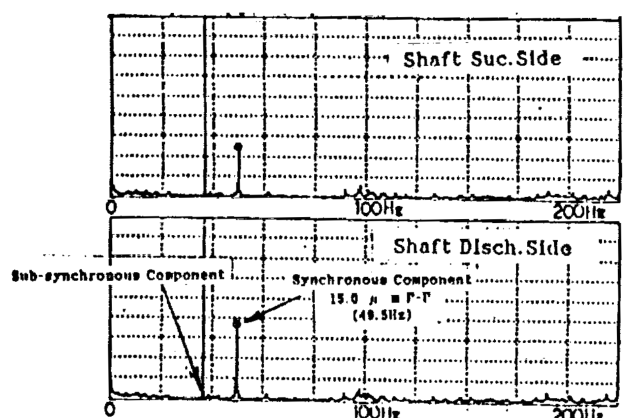


Fig. 5 Power spectrum of the sub-synchronous vibration



a. With aligned wearing rings



b. With aligned wearing rings and swirl breakers

Fig. 6 Successful suppression of the sub-synchronous vibration by aligning the wearing rings in accordance with the natural deflection of shaft and by providing a swirl breaker to each impeller blade

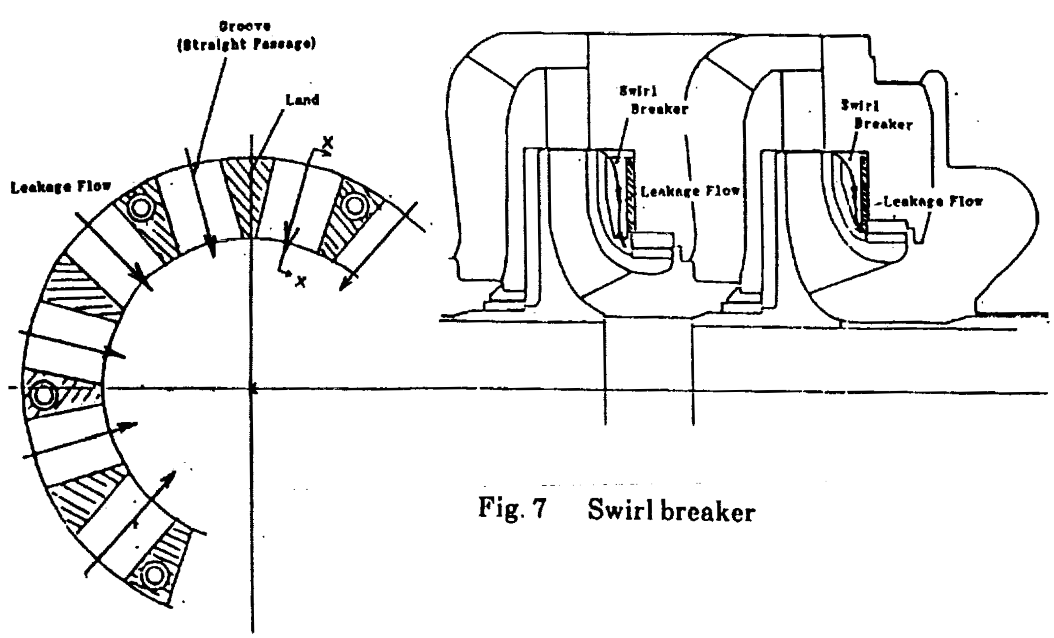


Fig. 7 Swirl breaker