Case History	Self-Excited Vibration Caused by Labyrinth Seal of Steam Turbine Rotor	Rotating machinery (turbine & generator)
Self-excited Vibration		

Object Machine

Steam turbine for LNG cryogenic power generation (Fig.1)

Observed Phenomena

During the initial phase of commissioning on site, a vibration component of about one half (38 Hz) of the rotational speed (4,545 rpm) was observed, but the overall shaft vibration amplitude was as small as 25 μm PP. However, after performing a periodic inspection (shaft alignment adjustment and bearing disassembling & reassembling), the shaft vibration of the 38 Hz component suddenly increased up to 90 μm PP (maximum). In particular, the vibration depended to a large extent on the output, and started to develop at about 70% of the rated output, and the amplitude took its maximum value at the rated output. Fig.2 shows the result of frequency analysis of this shaft vibration (displacement).

Cause Presumed

This phenomenon was considered to be self-excited vibration because 38 Hz was close to the rotor natural frequency (first order). After examining several possible causes of the self-excited vibration in the machine, the cause estimation was made by elimination method.

Analysis and Data Processing

Since this machine used five-pad tilting-pad bearings, and there were almost no preceding examples, oil whirl was considered unlikely to occur. On the other hand, diaphragm couplings were used and turbine disks were blisks made by cutting work, so that the influence of internal friction was also considered unlikely. Judging from the remarkable characteristics of output dependency (Fig.3) of the shaft vibration, it was finally estimated that the most probable cause was a fluid induced self-excited vibration. In calculations without considering such fluid force, the shaft damping ratio at the rated rotational speed was as small as 0.01~0.015, which was sensitive to such destabilizing force. As for the flow induced self-excited vibration, the fluid force of labyrinth seals was specifically estimated to be the cause.

Countermeasures and Results

Countermeasures were taken for the following two points.

(1) Improvement of bearing damping characteristics (increase by about 0.1% of radial clearance, preload factor 0.8→0.0, pad layout change; bearing load direction was shifted to pad center (LOP: load on pad configuration) by rotating 20°; this aimed to gain damping ratio by increasing the anisotropic nature of spring coefficients $K_{xx} \neq K_{yy}$.)

(2) Installation of a swirl breaker (swirl flow prevention plate at the labyrinth seal entry)

The increase in radial gap and decrease in preload factor under (1) above had an effect of reducing the amplitude by nearly half, while the effect of the pad location change was significant, which decreased the amplitude to about 15 μm.

Furthermore by taking countermeasure (2), the self-excited vibration component has completely disappeared.

Although it is possible to qualitatively grasp unstable vibration caused by seals at present, it is hard to quantitatively assess such phenomenon. In this type of excitation force, the swirl flow at the seal entry has large effect on its destabilizing force, so that a swirl breaker (swirl brake) is installed as a stabilizing means.

References

Nothing in particular.

Lesson Learned

In many instances, it is hard to find out the causes for self-excited vibrations, thus it takes time to determine appropriate countermeasures. Consequently, as a cardinal rule, we are to take indirect countermeasures such as improvement of damping characteristics at the same time.

Keyword

flow induced vibration, labyrinth seal, swirl breaker, swirl brake, tilting pad bearings

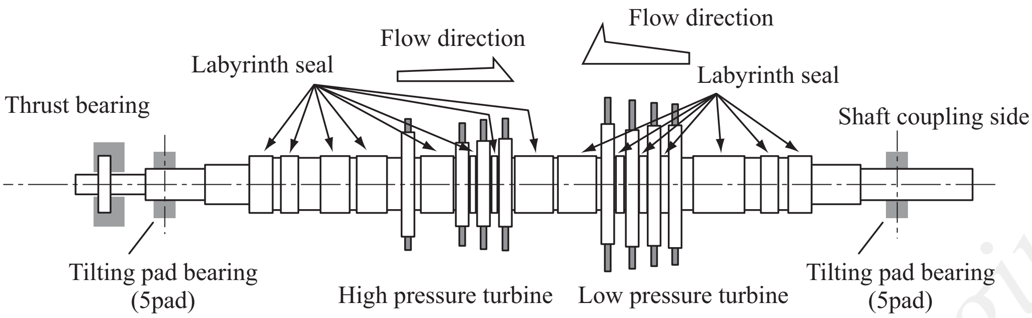

Fig.1 Schematic of steam turbine rotor for cryogenic power generation

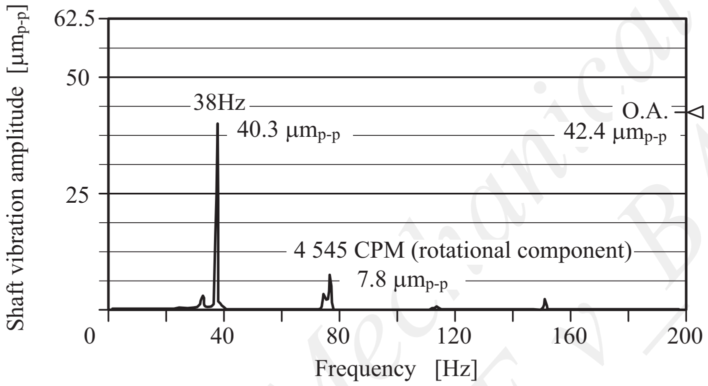

Fig.2 Example of frequency analysis result of the shaft vibration

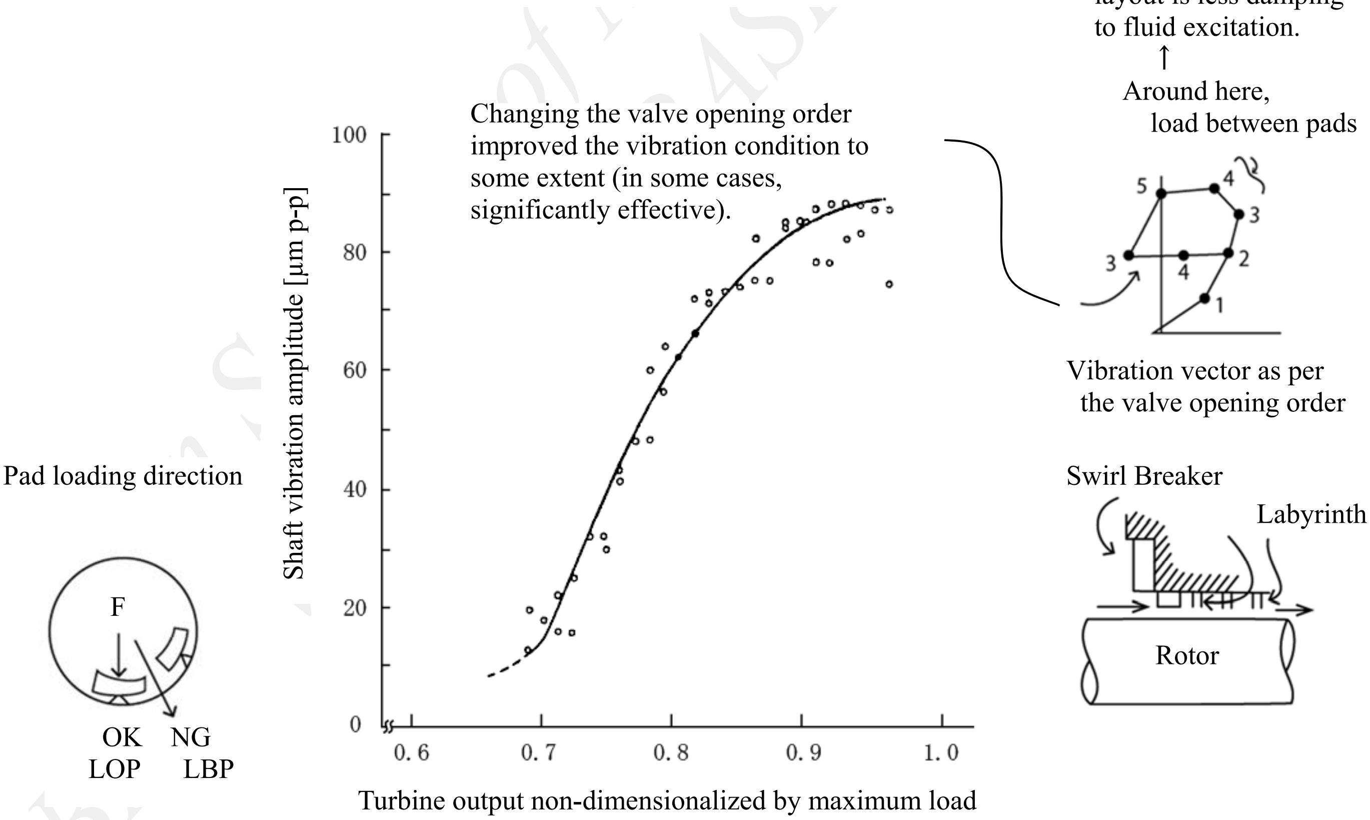

Fig.3 Shaft vibration change with changing turbine output

The Japan Society of Mechanical Engineers v_BASE v_BASE