Vibration / Resonance	Vibration of Water Turbine Runner due to Interference Stator and Blade	Rotating Machinery

Object Machine

High-head pump turbine runner (head: 500m class)

Observed Phenomena

As shown in Fig.1, the target pump turbine consists of a spiral casing, stay vanes, guide vanes, runner and draft tube, and functions to rotate the runner in the pump rotating direction and in the turbine rotating direction, to perform water pumping and power generation, respectively. In order to enhance the economic efficiency, the head of pump turbines is becoming higher and higher year by year. 500m class high-head pump turbine runners manufactured in the 1970s experienced many high cyclic fatigue cracks at the blade roots on the turbine inlet side, and some runners broke after 2 years of operation.

Cause Estimation

Because of high-speed operation associated with increasingly higher heads, the excitation frequency due to interference between the rotating runner and stationary guide vanes, and the peculiar natural frequency of the runner vibration mode excited by the former come closer to each other, resulting in the generation of a large variable stress. The measurement results of variable stress of the actual equipment are shown in Fig.5, which indicates that the resonance point is close to the rated rotating speed where a large variable stress was generated.

Analysis and Data Processing

Judging by the results of an actual head model test in consideration of a hydro-elastic similarity law with the actual equipment, and a simplified circular plate model has been conducted. As well as other investigations, it has been found that hydraulic pulsations occur due to interference between the rotating runner and stationary guide vanes that have a peculiar pattern determined by the combination of the number of blades of the runner and the guide vanes; that the runner is also excited by a vibration mode corresponding to the above pattern; and that the natural frequency also decreases to a greater extent compared to a phenomenon in an air medium due to the added mass of water around the runner (refer to Fig.4). The above is the outbreak mechanism of runner vibrations.
The condition for an excitation mode to occur due to interference between the rotating runner and stationary guide vanes is theoretically given in the following equation:

$$n \cdot Zg \pm k = m \cdot Zr \qquad (1)$$

where n, m are arbitrary integers, and k is the number of excited nodal diameter, and "+" sign indicates that the mode has the forward direction equal to the runner rotational direction, while "–" sign indicates the reverse direction. Fig.2 represents as an example a pattern of interference between the rotating and stationary blades wherein the numbers of runner blades and guide vanes are 6 and 20, respectively. In case of this combination of the numbers of blades and vanes, a vibration mode as in Fig.3 is excited wherein the runner nodal diameter 2 is combined with nodal diameter 4.

Countermeasures and Results

The fundamental countermeasure is to avoid peculiar resonance of runner for an excitation mode due to the interference between the rotating runner and stationary guide vanes, that is, (1) adjustment of natural frequency by means of the shape of runner outer circumference, (2) change in the combination of the number of blades of runner and guide vanes. Fig.5 shows an example applied to the actual equipment after confirming the vibration characteristics by modifying the shape of runner outer circumference according to an actual head model test. As shown clearly from this diagram, the vibration stress decreased substantially and no problem has occurred at all since then.

Lesson

At that time, almost no knowledge was available as to resonance phenomena of circular runners. Encountered with this event under such a circumstance, the above measures were taken to overcome the problem, thus leading to the development and commercial realization of super high-head equipment in excess of 700m.

References

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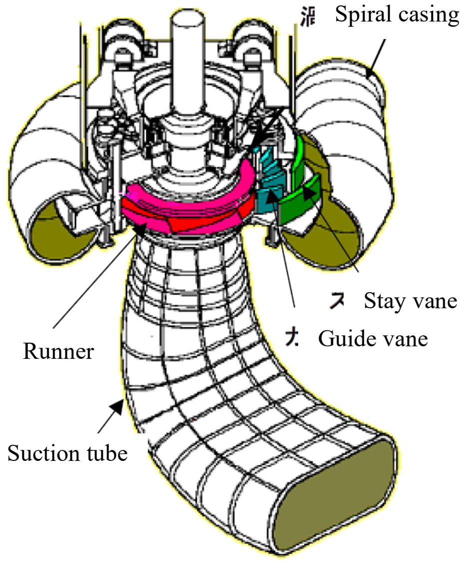
(2) Tanaka, H., Special Book, U2, IAHR Symposium 1990

Keywords

Fluid elastic vibration, interference between rotating and stationary blades, resonance, nodal diameter vibration, centrifugal type fluid machinery, pump turbine, runner, underwater reduction rate



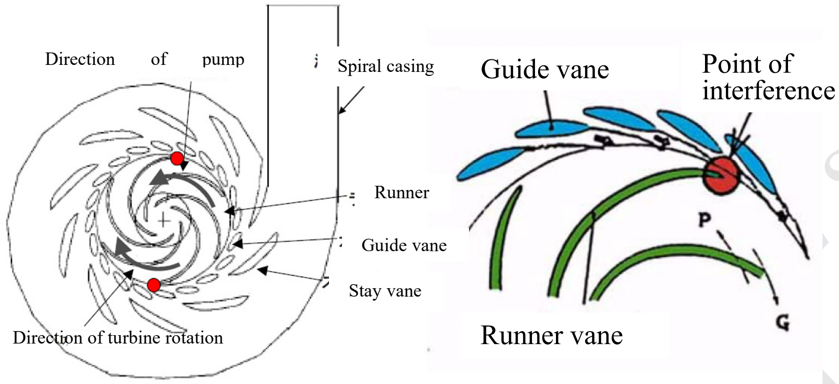
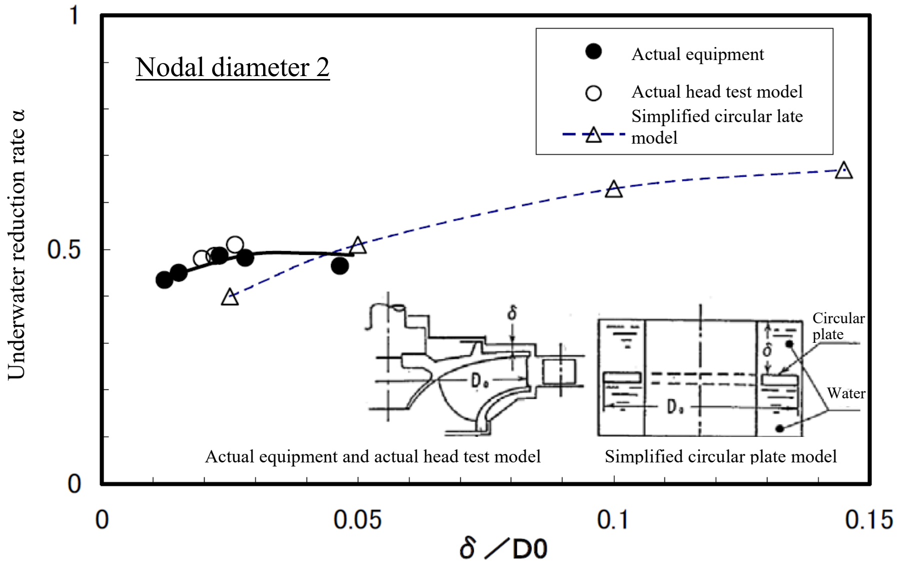


Fig.2 Rotating and stationary blades interference between runner and guide vane

Fig.1 Schematic diagram of pump turbine

Fig.3 Example of excitation mode of pump turbine runner



fna : Natural frequency in air
fnw : Natural frequency in water
α : Underwater reduction rate of natural frequency = fnw/fna

Fig.4 Underwater reduction rate of runner natural frequency

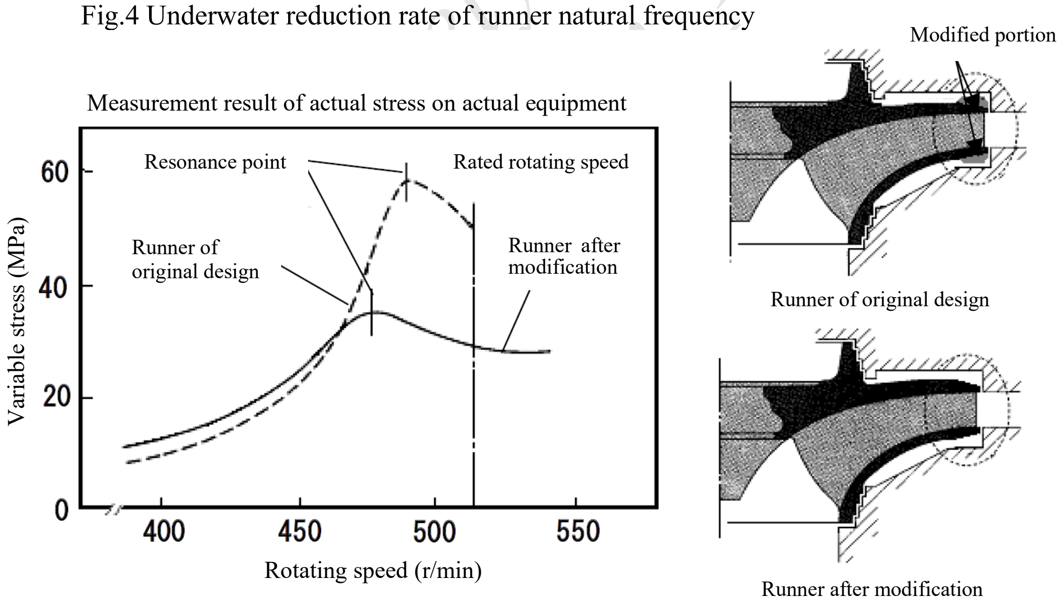


Fig.5 Example of adjustment of runner vibration characteristics