Vibration Analysis Technologies for High Head Pump-Turbine Runners

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1. Introduction

In the early 1970's, the head exceeded 500 meters in pumped storage power plants in Japan. But for a more economical construction, a higher head was pursued. Recently, higher rotational speed and more compact equipment have been adopted. Among the many technical subjects in the planning and design of high head pump-turbines, the estimation of dynamic stress on the runners is most important in realizing highly reliable runners. With higher speed and a higher head, not only static stress on the runner but also dynamic stress and its frequency increases. This is because the amplitude and frequency of the pressure pulsation around the runner increase. Therefore, in the design stage, the technology for prediction of dynamic stress acting on the runners is of major importance.

In accordance with these technical requirements, Fuji Electric has developed fluid-structure coupled vibration analysis technology and an actual head model test apparatus. In this paper, these developments will be briefly described.

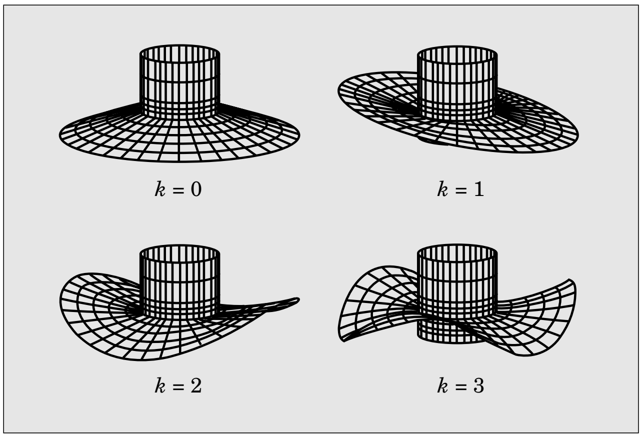
2. Vibration Characteristics of the Runner in Water

As the runner blades of a pump-turbine in generating mode pass one by one through the wakes of the guide vanes, velocity and pressure around the blades fluctuate periodically. In pumping mode, velocity and pressure around the blades also fluctuate as the blades approach and pass the guide vanes. Such pressure pulsation due to the interaction between stationary and rotating cascades varies according to the combination of the number of blades. This pressure pulsation generates a pressure field with diametrical nodes satisfying the equation;

$$n \cdot Z_G + k = m \cdot Z_R \quad \cdots\cdots (1)$$

Where Z_G and Z_R are the number of guide vanes and runner blades, respectively, and n and m are orders of excitation. The symbol k is the number of diametrical nodes which corresponds to the vibration mode. This is illustrated in Fig. 1.

Fig.1 Vibration modes of a disk with various diametrical nodes



The frequency of pressure pulsation acting on the rotating runner is;

$$f_R = n \cdot Z_G \cdot N / 60 \quad \cdots\cdots (2)$$

Where N denotes the rotating speed of the runner (r/min). Such a pressure field with diametrical nodes rotates with a speed of;

$$f_{PR} = f_R / k \quad \cdots\cdots (3)$$

Where positive k corresponds to a progressive wave, a wave rotating in the same direction as the runner, and negative k corresponds to a reverse wave, a wave moving in the opposite direction as the runner.

The frequency and the rotational speed of the pressure pulsation observed on the stationary parts are;

$$f_S = m \cdot Z_R \cdot N / 60 \quad \cdots\cdots (4)$$

and

$$f_{PS} = f_S / k \quad \cdots\cdots (5)$$

In addition, although there are infinite combinations of n, m and k satisfying equation (1), the specific mode of the pressure field which actually cause the vibration on the runners can be found. Taking into account the response of the runner excited by such a pressure pulsation, the following conditions can be drawn:

(1) Lower order excitation modes (smaller n) generally cause a larger response on the runners.
(2) Excitation modes with smaller k generally cause larger response on the runners because they excite

vibration with less diametrical nodes.

(3) A pump-turbine runner consists of two disks (crown and band) connected by blades, and thus responds easily to the excitation by which the displacement at blade roots can always be zero.

The first and second conditions, specify modes with $n = 1$ and smaller $|k|$. From the third condition, it can be concluded that the combination of two modes are specified, of which the difference in k is equal to the number of runner blades. Table 1 lists the combinations of excitation modes for various numbers of runner blades and guide vanes which are obtained by such an investigation. For example, a pump-turbine with 20 guide vanes and 6 runner blades is subject to excitation with 2 and 4 diametrical nodes. In this case, the pressure pulsations with frequencies of 3 and 4 times $Z_R \times N$ are observed on the head cover. Excitation of the first order on the runner has a frequency of $Z_G \times N$ in all cases.

When the resonance frequency of a pump-turbine runner is numerically analyzed, the above-mentioned excitation mode should be considered. Because the vibration caused on a runner by the interaction with guide vanes has a relatively high frequency, many natural frequencies are obtained below the modes in question. The result should be evaluated, taking into consideration that the mode with the same number of diametrical nodes as the excitation mode is selectively amplified.

3. Coupled Vibration Analysis in Water

3.1 Analysis method and calculation model

It is a well-known fact that the natural frequency of a structure in water generally varies from that in air. The natural frequency of a structure in extensive water is reduced by around 20%. This reduction in natural frequency is caused by the added mass effect, which is due to the reaction force (pressure) from the water adjacent to the structure. This reaction force is proportional to the acceleration of vibration.

In the case of a pump-turbine runner, there are small spaces between the crown and head cover and between the band and bottom cover. When the crown and the band vibrate, the water in these spaces must move peripherally or move out of the space through the runner seal and the outer periphery. The volume of this water is determined by the displacement of the crown and the band. Therefore, the reaction force from water, or the added mass effect, of the runner is much larger than in extensive water. Therefore, the natural frequency of pump-turbine runners in operation is reduced by around 50% that of air. Because this reduction rate is affected by the mode shape of vibration, the space between the head/bottom covers and the runner, the stiffness of the head cover, etc., numerical analyses are necessary in order to quantitatively predict the natural frequency in the design stage.

Table 1 Excitation mode on the runner (number of diametrical nodes)

Z_R \ Z_G	16	20	24
6	−4 / +2	−2 / +4	−6 / 0 / +6
7	−2 / +5	−6 / +1	−3 / +4
8	−8 / 0 / +8	−4 / +4	−8 / 0 / +8
9	−7 / +2	−2 / +7	−6 / +3
10	−6 / +4	−10 / 0 / +10	−4 / +6

Vibration analysis of the runner is carried out using the solver provided by MSC/NASTRAN, which calculates the added mass effect of fluid. As the boundary element method is used for the solution of the fluid property, the definition of the boundary is only required for the modeling of the fluid region. But a knowledge of the definition of the fluid boundary is necessary. In addition, it takes a long time for the solution of simultaneous equations. This is because a full matrix is produced for the fluid region due to the application of the boundary element method. As a result, modeling knowledge has been accumulated through the comparison between calculated and measured vibration modes based on such systems as a simple disk in a vessel.

Figure 2 shows the analytical model of a pump-turbine runner. To take into account the effect of the gap between the runner and the head/bottom covers and the stiffness of the head cover, the model includes the head and bottom covers.

3.2 Example of analysis results

Figure 3 shows an example of the vibration mode and stress distribution of a runner. Many natural frequencies with various vibration modes are obtained by the fluid-structure coupled vibration analysis solver in MSC/NASTRAN. As mentioned above, the degree of response is different for each mode because of the specific mode of the pressure pulsation acting on the runner in operation.

Hence, the modal response analysis is carried out

Fig.2 Finite element analysis model of a pump-turbine runner

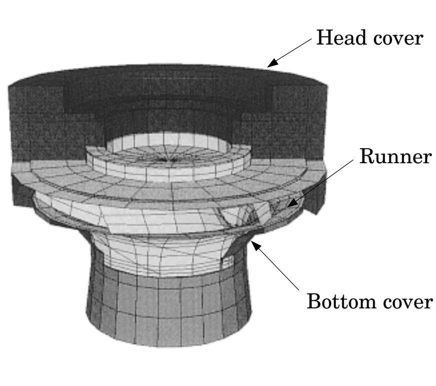


Fig.3 Vibration mode and stress distribution on the runner

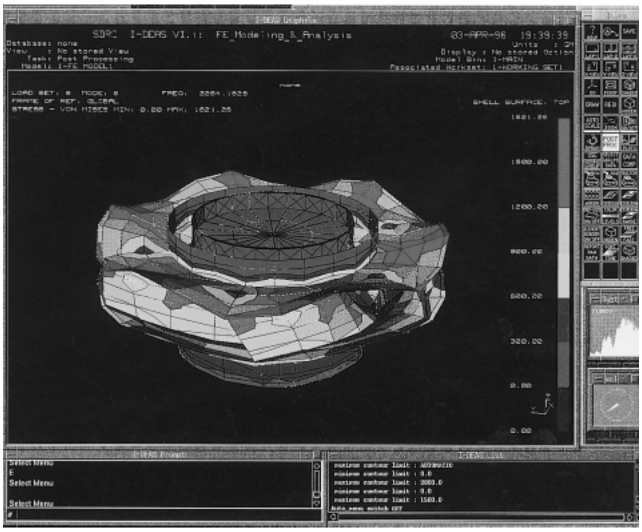


Fig.4 Flow chart of mode synthesize method program

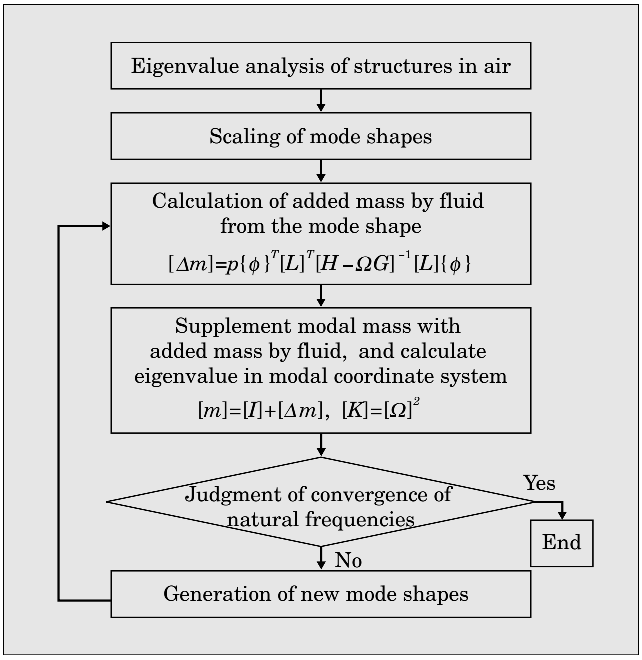
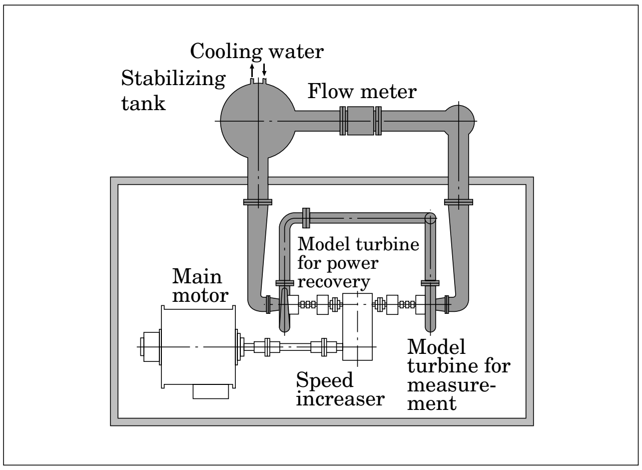


Fig.5 Actual head model test stand



by applying pressure pulsation on the runner, based on the result of fluid-structure coupled vibration analysis, in order to predict the actual natural frequency. Then, the level and location of dynamic stress can be quantitatively estimated from the stress analysis according to the amplitude of vibration. Because the absolute level of pressure pulsation as well as the dumping factor are difficult to directly calculate, the reliability of calculation is improved by feedback of the measured data in the power plants and the actual head model test stand.

3.3 Fluid-structure coupled vibration analysis method by mode synthesize method

As the fluid-structure coupled vibration solver in MSC/NASTRAN applies the boundary element method for the fluid region, it has the advantage of easy mesh generation. At the same time, its disadvantages induce long calculation time and structural modeling limited by the shell elements. In order to eliminate such defects, a new fluid-structure coupled vibration analysis code was developed using the mode synthesize method.

This method is based on the governing equations of unstable pressure distribution caused by small scale vibration of compressible fluid. The equation of motion for structures are coupled with fluid elements by introducing in the external force term the pressure generated by the vibration of the wall facing the fluid region. If the simultaneous equations are solved directly, a very large matrix with a combination of structures and fluid regions is generated. Thus, the algorithm shown in Fig. 4 is applied to reduce calculation time.

By this method, the eigenvalue analysis of the structures in air is carried out first. Then, the force acting on the fluid is calculated using the mode of vibration normal to the wall. Consequently, the pressure pulsation, generated by this external force, acts on the structure. As this pressure pulsation is proportional to the acceleration of the wall, it is equivalent to the added mass on the boundary wall. These added mass components are calculated by the mode synthesize method based on the vibration mode of the structure. These algorithms are iterated until the natural frequencies converge.

4. Dynamic Stress Measurement in Actual Head Model Test Stand

4.1 Actual head model test stand

As stated above, a pump-turbine runner rotating in water forms a coupled vibration system due to its interaction with the surrounding water and structures. In order to simulate such a coupled vibration phenomenon using a model, the hydroelastic similarity law must be satisfied. In other words, the ratio of hydraulic excitation frequency to the natural frequency of the runner should be equal for both the prototype and the model. In the case of a pump-turbine runner, this similarity can be satisfied by manufacturing the model runner with the same material as the prototype and by operating the model runner with the same peripheral speed, or with the same head, as the prototype. Under such conditions, vibration observed in the model has a frequency in inverse proportion to the scale ratio and the amplitude is nearly similar to the prototype, if a small discrepancy in damping due to the difference in the Reynolds number is overlooked.

The actual head model test stand is the facility for the above-mentioned purpose and is outlined in Fig. 5. The torque generated by the main motor is transmitted to the model turbine through the speed increaser. The maximum available speed of the model is more than 7,000 r/min. In spite of the small scale model, a few MW of input power is required for hundreds of meters of test head; thus, this facility is equipped with a pair of model turbines at both ends of the high speed shaft. When one of the models is operated in pumping mode, the other recovers the driving power through the generating mode operation. Figure 6 portrays the model test facility in operation.

4.2 Example of measurement in actual head test stand

Stress measurement was carried out in this test stand with the model designed for a 400m class pumped storage power plant. The principal specifications of the prototype are listed below.

(1) Type : Reversible Francis pump-turbine
(2) Net head : 400m
(3) Max. output : 360MW
(4) Speed : 360r/min
(5) No. of runner blades : 6
(6) No. of guide vanes : 20

The stress on the runner was measured by 12 strain gauges, each of which was monitored simultaneously by a 12 channel FM telemeter. Together with the stress, operating conditions such as the rotating speed of the runner, discharge and static pressure at the inlet and outlet of the model were also measured. Preceding the installation of the strain gauges, finite element analyses were carried out in order to determine the location of maximum stress for the assumed

Fig.6 Actual head model test stand in operation

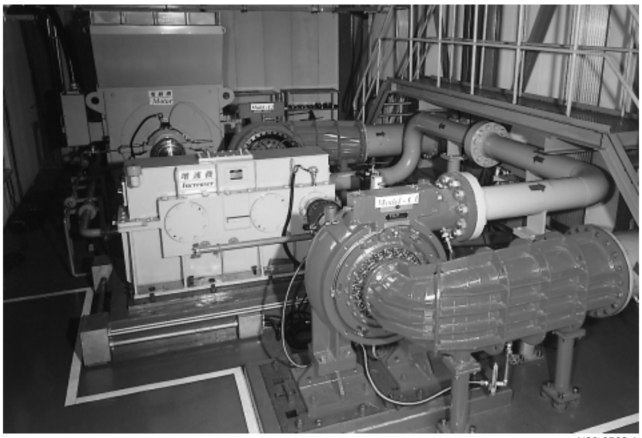


Fig.7 Measured stress amplitude on the runner in generating mode

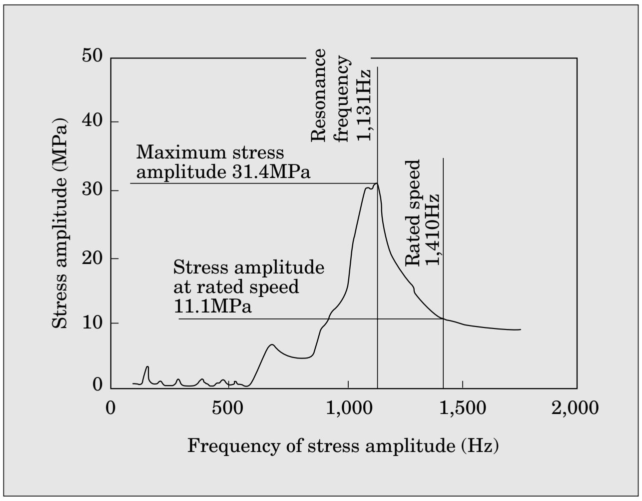


Fig.8 Measured stress amplitude on the runner in pumping mode

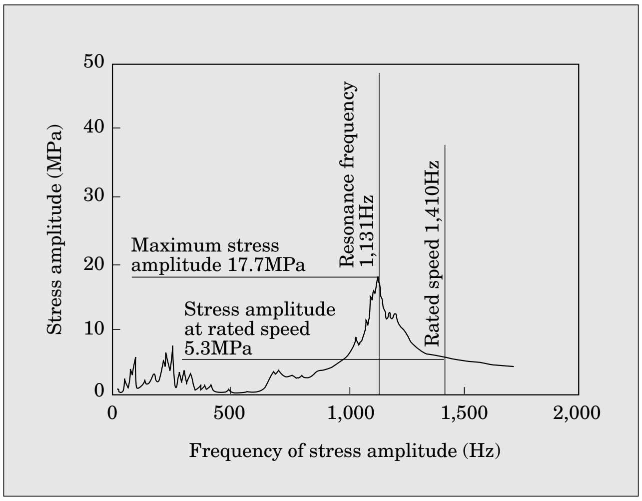
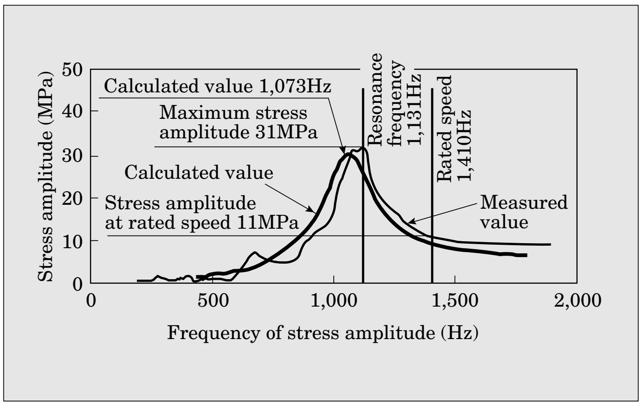


Fig.9 Comparison between analyzed and measured stress amplitude (generating mode)



vibration mode.

Figure 7 shows the amplitude of dynamic stress as a function of frequency in generating mode, which is measured by changing the rotational speed continuously. An obvious peak can be seen at 1,131Hz, and the maximum stress amplitude at the resonance point is 31.4MPa. The frequency of hydraulic excitation of the prototype rotating with the rated speed of 360 r/min corresponds to 1,410Hz on the model. Therefore, the prototype is operated at a frequency 20% different from the resonance frequency, where the stress amplitude is 11.1MPa, or 1/3 of peak value.

Figure 8 shows the stress amplitude measured during pumping operation. An obvious peak exists at 1,131Hz, the same as in the generating mode, but the level of the stress amplitude is about half that in the generating mode including the resonance point. This is because the hydraulic excitation due to the interaction between the guide vanes and runner blades is less than in generating mode.

From these measurements, it is confirmed that the level of the stress amplitude for the prototype's operating conditions is sufficiently low compared with the strength of the runner in both the generating and pumping modes.

4.3 Comparison of the results of coupled vibration analysis

Figure 9 shows the results of the modal response analysis in generating mode based on the fluid-structure coupled vibration analysis compared to measurements taken during testing with the actual head model. The resonance frequency of the runner obtained by the analysis is 1,073Hz, while the measured frequency is 1,131Hz, as stated above. The error of numerical prediction is about 5%, and accuracy is sufficient for practical use. In addition, simulated stress amplitude as a function of frequency also coincides well with measurements. It is confirmed that the stress amplitude acting on the runners can be predicted in the design stage with sufficient accuracy, and that a design that avoids resonance is possible.

5. Conclusion

In this paper, an outline of recent developments in the evaluation of vibration characteristics of pump-turbine runners in operation were reported. It was shown that the characteristics of dynamic stress on pump-turbine runners in operation can be estimated with sufficient accuracy before the manufacturing of prototypes. This was proven using both computer aided fluid-structure coupled vibration analysis technology and actual head model test technology.

The pumped storage power plants of the future will probably increase in both head and speed. Though the technical requirements become increasingly severe from the viewpoint of reliability for the vibration of the runners, we hope the technology presented here will contribute to further developments in the world of electric power generation.