621.224.7:534.141

Paper No. 195-10

Characteristics of Radial Force Fluctuations on Runners of Francis-type Pump-turbines*

(Report 1 : Experiments on General Characteristics of
Radial Force Fluctuations and Effects of
Structural Factors)

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The hydraulic, radial force fluctuation acting on the runner of a Francis-type pump-turbine was investigated experimentally. Five model pump-turbines of different specific speeds were tested. In pumping operation, the amplitude of the fluctuation increases as discharge decreases. In generating operation, the maximum amplitude appears with the draft surge frequency during partial load operation. The runner speed frequency component always appears. Components with 0.5 to 1.0 times the runner speed frequency become dominant when maximum amplitude fluctuation occurs during transient operations.

1. Introduction

Most studies on shaft vibration in pump-turbine and generator-motor systems have been concerned with response analyses of the systems⁽¹⁾⁻⁽³⁾. There has been only limited information collected on the radial force fluctuation, i.e. the hydraulic force that causes displacement fluctuation of the runner in the radial direction. Most of the studies on radial force in hydraulic machinery have focused on pumps and considered only static components⁽⁴⁾⁻⁽⁸⁾. Imaichi et al.⁽⁹⁾, investigating both static and dynamic components (radial force fluctuation) of radial force in a pump, state that the latter component is not necessarily smaller than the former one, and that both components increase with the growth of cavitation. They also showed that the dynamic component consists of the runner speed frequency component, its harmonics, and random components.

Radial force fluctuation is generally supposed to be caused by the following: (1) pressure fluctuation in runner periphery (generation of radial force fluctuation due to variation of pressure distribution in outer periphery of the runner, generation of moment fluctuation due to variation of pressure acting on the runner crown and the runner band); (2) pressure fluctuation in gaps between the runner and the stationary part; and (3) asymmetry and fluctuation of hydraulic reaction force acting on runner blades.

The main purpose of this study is to make clear various characteristics of radial force fluctuation acting on a pump-turbine runner. Using five models of Francis-type pump-turbines with pump specific speeds ranging from 28 to 60 (rpm, m, m³/s), the authors experimentally investigated the general characteristics of amplitude and frequency of radial force fluctuation both in pumping and generating operations. Evaluation was made of how this radial force fluctuation relates to structural factors (runner sealing gap, clearance between two blades) and to cavitation factor.

2. Nomenclature

B : height of runner inlet
D₁ : diameter of runner inlet
F : amplitude of radial force fluctuation
GVO : gate opening
H : total pump head or effective head
H₁₀₀₀ : H converted to 1000 rpm
ΔH : amplitude of pressure fluctuation
nondimensional value of F,
K = F/γD₁BH
K₀ : K at η_{max}
K₀' : K₀ of model with n_{sp} 28
P : pump input or turbine output
P₁₀₀₀ : P converted to 1000 rpm
Q : discharge
Q₁ : Q per unit head = Q/√H
Q₁₀₀₀ : Q converted to 1000 rpm
f : frequency
n : runner speed or frequency of runner speed
n₁ : runner speed per unit head = n/√H
n_s : specific speed
q : nondimensional discharge =
Q₁₀₀₀/(Q₁₀₀₀ at η_{max})
γ : specific weight of water
η : efficiency
σ : cavitation factor

* Received 4th October, 1979.

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to tensile load. Influence of axial thrust on the detector was also checked in the calibration test. It was found that the x-axis gauges do not respond at all to the axial thrust. The y-axis gauges respond as if a radial force of about 10 N acted on the runner, as against the axial thrust assumed to act on the runner in the test with about 30 m head (in the case of medium- n_s model). From these results, we decided to take the measurable range of this detector as over 30 N. Also we determined the rigidity of the detector in the radial direction and obtained the maximum radial load which is limited by the geometry of the gap between the runner and the stationary part.

3.2.2 Frequency characteristics

The frequency characteristics of the detector were investigated by hammering and by using vibration exciter. The natural frequency of the detector, depending on the mass of the test runner, ranged from 40 Hz to 68 Hz in air and was about 15 % lower in water because of the virtual mass effect of water. The frequency range in which the effect of resonance of the detector disappeared was lower than about 75 % of the detector's natural frequency.

3.2.3 Mechanical vibration of test device and detector

Adjustments were made to minimize the mechanical vibration due to unbalance weight in the rotor, misalignment and so on. The amplitude of the mechanical vibration was from one-fourth to one-half of the minimum amplitude of radial force fluctuation measured, depending on the test model involved.

In order to investigate the relation between radial force fluctuation and pressure fluctuation in the runner periphery, pressure transducers were mounted as shown in Fig. 1(c).

3.3 Processing of experimental results

In this paper, amplitudes of radial force fluctuation are nondimensionalized by the following equation :

$$K = \frac{F}{\gamma D_1 B H}$$

4. Experimental results and considerations of the results

4.1 Characteristics in pumping operation

4.1.1 Characteristics in steady-state operation

Figure 2 shows pump performance curves and the ratios of nondimensional amplitudes of radial force fluctuations, K/K_0 for the medium- n_s model. Similar results were obtained with the high- n_s and low- n_s models. The amplitude of the radial force fluctuation has the following features. (1) The amplitude is larger in $H_{\max}$ operation than in H_{nor} or $H_{\min}$ operation. (2) The amplitude increases as pump discharge decreases. (3) The amplitude

increases abruptly when gate opening exceeds an optimum value (which is a gate opening at maximum pump discharge under constant head operation).

Figure 3 shows K/K_0 versus cavitation factor σ in operating conditions where the amplitude of radial force fluctuation is near the maximum, relatively small, or in between. Also shown in Fig. 3 are σ_c values measured under a constant head operation. The term K/K_0 is constant in $\sigma > \sigma_c$, increases in $\sigma < \sigma_c$, and after reaching a maximum value, decreases as degree of cavitation further intensifies. As mentioned above, amplitude of radial force fluctuation tends to increase as discharge decreases.

Figure 4 shows K/K_0' as a function of n_{sp} and q . The ratio K/K_0' is small in low- n_s pump-turbines. From the curves in Fig. 4, one can predict, although roughly, the amplitude of radial force fluctuation in pumping operation of a pump-turbine of a given n_s value.

Figure 5 shows spectra of radial force fluctuation in pumping operation. Frequency analysis of radial force fluctuation shows that the n component (which is the runner speed component) is dominant, and amplitudes of the random components increase considerably during over gate opening operation (in Fig. 5(c)).

4.1.2 After input rejection

In a prototype pump-turbine, pump head and runner speed vary in the condition from input rejection to no discharge operation. In the present study, a locus of operating point during the transient state was predicted by numerical analysis. Figure 6 shows the amplitudes of radial force fluctuation at selected operating points on the locus. Amplitude of radial force fluctuation increases in the reverse pumping zone and then decreases. The main frequency components of radial force fluctuation at (A) in Fig. 6 are the n component and a component with a frequency slightly lower than n .

4.2 Characteristics in generating operation

4.2.1 Characteristics in steady-state operation

Figure 7 shows a contour map of K/K_0 and efficiency ratio measured in the medium- n_s model. Amplitude of radial force fluctuation is a minimum around the maximum efficiency point. The region where K/K_0 is a maximum is found in the partial load operation region (near $n_1 = 230 \text{ rpm}/\sqrt{m}$, $Q_1 = 0.01 \text{ m}^3/\text{s}/\sqrt{m}$). Beyond this region, as Q_1 decreases, K/K_0 decreases. A similar tendency was also observed in tests with other n_s models.

Figure 8 shows K/K_0 values of the medium- n_s model at various σ values. In the generating operation of the medium- n_s model, amplitude of radial force fluctuation was largely constant within the range of σ values used in the test, and there was little effect of cavitation. With the high- n_s model, on the other hand, there were cases where pressure fluctuation in the draft pipe during partial load opera-

Fig. 2 Pump performance curves and amplitudes of radial force fluctuation (medium- n_s)

Fig. 3 Amplitude of radial force fluctuation versus cavitation factor (medium- n_s , pumping operation)

Fig. 4 Amplitude of radial force fluctuation as functions of specific speed and discharge
(): quoted data⁽⁷⁾

Fig. 5 Spectra of radial force fluctuation
(medium- n_s , pumping operation, $H_{1000}=26.5\text{m}$, $n=1278\text{rpm}$, $\sigma=0.16$)

Fig. 6 Radial force fluctuation after input rejection (low- n_s)

Fig. 7 Turbine characteristic curves and distribution of amplitude of radial force fluctuation (medium- n_s)

Fig. 8 Amplitude of radial force fluctuation versus cavitation factor (medium- n_s , generating operation)

Fig. 9 Spectra of radial force fluctuation (medium- n_s , generating operation, $n_1=221\text{rpm}/\sqrt{m}$, $n=1296\text{rpm}$)

tion increased with a variation of σ , and, as a result, the amplitude of radial force fluctuation changed.

Figure 9 shows the spectra of radial force fluctuation in the generating operation of the medium- n_s model. It may be noted that the main frequency components of radial force fluctuation in the generating operation are the n component and $n/3$ component (which is the surging cycle during partial load operation (Fig. 9(b)). In the case (Fig. 9(a)) where the amplitude of radial force fluctuation is the maximum, the surging cycle increases from $n/3$ to about n , and the amplitude of n component appearing also in (b) and (c) increases. At the same time, the waveform of fluctuation disintegrates and random

components in the entire frequency range increase. As shown in Fig. 10, the $n/3$ component in partial load operation is significant in high- n_s models. The amplitude of radial force fluctuation in partial load operation is strongly influenced by pressure fluctuation due to draft surge.

Now, radial force fluctuation in generating operation, as shown in Fig. 7, varies widely depending on n_1 (or head) and Q_1 (or turbine output), so that the trend of the radial force fluctuation cannot be expressed simply as a function of specific speed and discharge as it can in the case of pumping operation (see Fig. 4). The values of the maximum amplitude of radial force fluctuation in a practical operation range are presented in Fig. 11. The values represent values in partial load operation in $H_{\min}$ conditions. There is no significant relationship between K/K_0' and n_s in contrast to the relationship in Fig. 4 for pumping operation. Only when the ratio of K/K_0' versus $\Delta H/H$ is used, can such a trend as shown in Fig. 11 be obtained. This may be attributed to no significant relationship between n_s and pressure fluctuation in draft pipe in partial load operation. Therefore, in order to predict the amplitude of radial force fluctuation in partial load operation, it is necessary that the pressure fluctuation in the runner periphery be measured.

4.2.2 After load rejection

A locus of operating point after load rejection was predicted by numerical analysis. Figure 12 shows the amplitudes of radial force fluctuation at selected operating points on the locus. Amplitude of radial force fluctuation after load rejection increases gradually in the region from about runaway speed to the turbine brake region and becomes a maximum near the point of zero discharge, thereafter it decreases toward the reverse pumping zone. As operating point once enters the reverse pumping zone and then returns toward the point of zero discharge, amplitude increases again. A special feature of radial force fluctuation after load rejection is that it varies widely with a slight deviation of n_1 . Figure 13 shows the spectra of radial force fluctuation and pressure fluctuation at point (B) in Fig. 12. It is seen that the component of frequency (11.6 Hz) lower than n (19.2 Hz) is dominant in both fluctuations. It is evident that the runner is excited by the pressure fluctuation in the runner periphery just as in the case of partial load operation.

Fig. 10 Spectra of fluctuations (high- n_s , generating operation, GVO=30%, $n_1=255\text{rpm}/\sqrt{m}$, $n=1272\text{rpm}$, $\sigma=0.14$)

Fig. 11 Specific speed and amplitudes of radial force fluctuation in partial load operation

Fig. 12 Radial force fluctuation after load rejection (low- n_s)

4.3 Effects of structural components of runner and its periphery on radial force fluctuation

To investigate the changes of radial force fluctuation that may be caused by changes in model tests and in structures of test devices, tests were made on the following two items.

4.3.1 Effect of runner sealing gap

Sealing gap used here means the gap between the discharge wearing ring and rotating wearing ring, as illustrated in Fig. 14(a). Compared with the prototype pump-turbine, the model has a small sealing gap in terms of absolute value. And the gap is likely to vary during tests due to abrasion or the like, particularly when the model is operated over a long period of time under unfavorable conditions, as in radial force fluctuation tests. If radial force fluctuation is strongly influenced by the sealing gap, reliability of the values measured in model pump-turbines would be reduced. Some researchers have reported that sealing gap size is related to the self-exciting vibration of the turbine⁽¹⁰⁾.

Tests were made with three sealing gap sizes of 0.15, 0.25 and 0.50 mm. The results are presented in Fig. 14(b) and (c). Amplitude of radial force fluctuation increased slightly in both pumping and generating operations as the sealing gap is enlarged. However, the data do not show a distinguishable difference in amplitude during over gate opening pumping operation and partial load operation in which the amplitude reaches a maximum level. Widening of the sealing gap due to abrasion would amount to no more than 0.2 mm, which is not large enough to affect the accuracies in model tests. Enlargement of the gap did not bring about any changes in the frequency of radial force fluctuation.

(a) Radial force fluctuation
(b) Pressure fluctuation in the periphery of runner inlet
(c) Pressure fluctuation in draft pipe

Fig. 13 Spectra of fluctuations at point B in Fig. 12 ($n_1=230\text{rpm}/\sqrt{m}$, $n=1152\text{rpm}$)

(a) Details of gap

(b) Pumping operation
($H_{1000}=26\text{m}$, $\sigma=0.16$)

(c) Generating operation
($n_1=227\text{rpm}/\sqrt{m}$, $\sigma=0.4$)

Fig. 14 Sealing gap and radial force fluctuation (medium- n_s)

Fig. 15 Runner clearance

4.3.2 Effect of tolerance in runner blade dimensions

Unless the canals between runner blades, in particular, the runner outlet clearances, are uniform, discharge would vary in each canal. As a result, hydraulic reaction force acting on the blades would be imbalanced and radial force fluctuation could be produced. A low- n_s model with a runner outlet clearance difference of 5 % was assumed. By calculating the difference of flow momentum in the canals between runner blades caused by the above-mentioned assumption and converting this to the amplitude of radial force fluctuation, we got the amplitude of 9.8 N in generating operation of the model. This value is extremely small compared with the amplitude actually measured and it is thought that radial force fluctuation due to non-uniform clearance is very small. However, since calculations involved many assumptions, a test was made for confirmation. One runner outlet clearance (Fig. 15) was made 5 % smaller than the others. It was found that such a reduction had little effect on the amplitude, frequency, and phase shift between x and y components of radial force fluctuation.

5. Conclusions

Radial force fluctuation acting on the runner of a Francis-type pump-turbine was experimentally investigated in all its operation regions. Also relationships with hydraulic and structural factors were studied. The findings may be summarized as follows.

(1) In steady-state pumping operation and pump starting: In a small discharge operation, amplitude of radial force fluctuation is large. The largest occurs in the early stage of pump starting. The main frequency component of radial force fluctuation is the runner speed component. Nondimensional amplitude of radial force fluctuation increases as the specific speed increases. The amplitude of radial force fluctuation in pumping operation will be able to be predicted by accumulated data in this paper.

(2) In steady-state generating operation: In partial load operation, radial force fluctuation is large. The main frequency component is the runner speed component. Furthermore, in partial load operation, the frequency component of the pressure fluctuation in draft pipe is added. To predict the amplitude of radial force fluctuation in partial load operation, it is necessary that the amplitude of

pressure fluctuation in the runner periphery be measured.

(3) In transient operation: The amplitude of radial force fluctuation is considerably larger than the one in steady-state operation. The frequency of maximum radial force fluctuation after input rejection and load rejection is the same frequency or lower than the runner speed. Amplitude of radial force fluctuation after load rejection is large in the region from runaway speed to zero discharge.

(4) In operation with a cavitation factor larger than a critical cavitation factor, cavitation does not affect radial force fluctuation (in pumping operation of medium specific speed model). Cavitation has hardly any effect of radial force fluctuation (in generating operation of medium specific speed model).

(5) With a large sealing gap, runner vibration increases, but the amplitude of vibration is not large enough to introduce inaccuracies in model tests.

(6) A non-uniformity in the range of 5 % in the canals between runner blades has very little effect of radial force fluctuation.

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