Measurement on the Fluid Forces Induced by Rotor-Stator Interaction in a Centrifugal Pump*

Shijie GUO**, Hidenobu OKAMOTO*** and Yoshiyuki MARUTA***

The pressure fluctuations and the radial fluid forces induced by rotor-stator interaction in a centrifugal pump were measured and their relationship was investigated. Experiments were done for various guide vanes, flow rates, and rotating speeds. It was demonstrated that both the blade pressure fluctuations and the volute static pressures are non-uniform circumferentially (not axisymmetric) under off-design operating conditions and that the two have a strong relationship. At high flow rates, the interaction-induced blade pressure fluctuations are large in areas where the volute static pressure is low. The propagating directions of the pressure fluctuations, the whirling directions of the radial fluid forces acting on the impeller and the dominant frequency components of both the fluctuations and the fluid forces are discussed. When measuring the fluid forces in the rotating frame, other frequency components, in addition to those related to the products of the number of guide vanes and the rotating frequency, may occur due to the circumferential unevenness of the pressure fluctuations.

Key Words: Centrifugal Pump, Rotor-Stator Interaction, Pressure Fluctuation, Fluid Force, Vibration, Rotordynamics

1. Introduction

Investigation on the fluid forces acting on impellers in turbomachinery has been a key research topic for a long time since the fluid forces may cause not only fatigue failure of the impellers but also vibration of the shaft. The present research was concentrated on the fluid forces induced by rotor-stator interaction (interaction between the impeller blades and the guide vanes), especially under off-design operating conditions. The pressure fluctuations acting on the impeller and the rotordynamic fluid forces as a result of the integration of the fluctuations were investigated experimentally.

Many studies have been done on the rotordynamic fluid forces in centrifugal pumps. Kanki et al.⁽¹⁾ measured the fluid forces in various operating conditions. Ohashi et al.^{(2),(3)} investigated experimentally the destabilizing fluid forces to the rotor by whirling the impeller with a DC

motor and measuring the reaction forces for various ratios of whirling to rotating. They concluded that the fluid forces of a free impeller basically had a stabilizing effect, but that in the case of a vaned diffuser, the interaction forces might be destabilizing for a small ratio of whirling to rotating. Zhang et al.⁽⁴⁾ measured the radial fluid forces with strain gauges installed on the shaft and found that unexpected frequency components in addition to ZgN and its higher harmonics occurred under off-design operating conditions. A similar phenomenon was also observed in the present measurement. In the present paper, a physical explanation is given and an equation is presented for predicting the frequency components. On the other hand, from the view of fluid dynamics, lots of investigations have been reported on the inner flows and the pressure fluctuations inside the impeller, the diffuser as well as the volute to understand the characteristics and the mechanisms of the fluid forces. Kurokawa et al.⁽⁵⁾ studied the flow in a two-dimensional volute and demonstrated that the static pressures and the velocities in the volute were not uniform circumferentially, i.e. not axisymmetric under off-design operating conditions. Miner et al.⁽⁶⁾ measured the flow inside a centrifugal impeller by using a laser velocimeter. They demonstrated that the flow was not uniform circumferentially, even at design point, because of the interaction between the impeller and the vo-

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** Tokai Rubber Industries, Ltd., 1 Higashi 3-chome, Komaki-shi, Aichi 485–8550, Japan

*** Ebara Research Co., Ltd., 2–1 Honfujisawa 4-chome, Fujisawa-shi, Kanagawa 251–8502, Japan.
E-mail: okamoto.hidenobu@er.ebara.com

lute. As for pressure fluctuations, Iino and Kasai⁽⁷⁾ measured the blade pressure fluctuations in a centrifugal pump and found that the fluctuations depended on flow rates and blade/vane angles, and that the fundamental frequency component was ZgN inside the impeller. Arndt et al.^{(8),(9)} measured the pressure fluctuations on the blades of a two-dimensional centrifugal impeller. They reported that the largest blade pressure fluctuations occurred at the trailing edge, independently of the guide vane configuration, but that the magnitudes depended greatly on the vane number and vane angle. Tsukamoto et al.^{(10)–(14)} investigated the pressure fluctuations inside the diffusers of centrifugal pumps by both numerical analysis and experiments. They reported that the fluctuations on the suction side of the guide vane leading edge were large and that the fundamental frequency components were ZrN and its higher harmonics, while other frequency components were also observed under off-design operating conditions.

The above mentioned studies have given us a comprehensive understanding of the pressure fluctuations and the fluid forces induced by rotor-stator interaction. However, our understanding of the relationships among the volute pressures, the impeller pressures, and the radial fluid forces as a result of the integration of the impeller pressures is still insufficient. In the present study, the volute pressures, the impeller pressure fluctuations and the radial fluid forces acting on the impeller in a centrifugal, vaned-diffuser pump were measured simultaneously, and their relationships were investigated. The propagating directions of the pressure fluctuations, the whirling directions of the radial fluid forces, and the dominant frequency components of both the fluctuations and the fluid forces are discussed.

Nomenclature

Zr : Number of blades of the impeller

Zg : Number of guide vanes

m, n, h : Integers

N : Rotating angular frequency of the impeller [rad/s]

K_{dX}, K_{dY} : Radial fluid forces with respect to the fixed frame (the frame fixed to the casing), normalized by $D_2 b_2 H_0$

k_{dx}, k_{dy} : Radial fluid forces with respect to the rotating frame (the frame fixed to the impeller), normalized by $D_2 b_2 H_0$

$P_{m,n}$: Radial force acting on the impeller when blade m interacts with vane n

D_2 : Outlet diameter of the impeller [m]

b_2 : Outlet width of the impeller [m]

H_0 : Total head at BEP [Pa]

Q : Flow rate [m³/s]

Q_0 : Flow rate at BEP [m³/s]

t : Time [s]

T : Rotating period of the impeller, $T = 2\pi/N$ [s]

BEP : Best Efficiency Point

2. Experiment

Figure 1 shows a schematic diagram of the volute casing of the centrifugal pump used in the experiment. Ten pressure gauges (C1-C10) were installed on the casing to measure the volute pressures. The experiment was conducted in a recirculating water test loop with a suction tank. The pressure in the test loop could be controlled by a vacuum pump and a compressor. The pump was driven by a motor and the rotating speed was changed by an inverter. The flow rate was adjusted with a discharge valve and measured by using a Venturi meter. The centrifugal impeller was a three-dimensional closed one with 6 blades, which was designed based on an actual pump. The guide vanes were fixed to the casing by bolts and therefore were easily removable from the casing. Three kinds of two-dimensional guide vanes (GV11: $Zg = 11$, GV7: $Zg = 7$ and GV5: $Zg = 5$) were used. Table 1 lists the specifications of the impeller and the guide vanes. All of them were made by NC cutting.

Twelve pressure gauges were installed on the impeller to measure the impeller pressure fluctuations, with three on the pressure sides of the blade trailing edges (P1, P3, and P5), three on the suction sides (S1, S3, and S5), and six on the hub (D1, D3, D5, R1, R3, and R5), as shown in Fig. 2. The radial fluid forces acting on the impeller were measured with strain gauges installed on the shaft, as shown in Fig. 3. All the wires of both the pressure gauges and the strain gauges were picked up through the hollow shaft by a slip ring assembly. In the measurement of fluid forces by strain gauges, bridge circuits were made so as not to measure the torsional and axial vibrations of the shaft, but only the bending vibration. Although the

Fig. 1 Locations of pressure gauges (C1-C10) on the volute casing and the fixed frame O-XY (fixed to the casing)

Table 1 Specifications of the impeller and the guide vanes

Impeller	Closed
Number of blades	6
Outlet diameter D_2 /Inlet diameter	328/202 mm
Outlet width b_2	32 mm
Outlet blade angle	32 degree
Guide vane GV11	
Number of vanes	11
Inlet diameter/Outlet diameter	331.5/430 mm
Inlet vane angle	18 degree
Guide vane GV7	
Number of vanes	7
Inlet diameter/Outlet diameter	331.5/430 mm
Inlet vane angle	16.5 degree
Guide vane GV5	
Number of vanes	5
Inlet diameter/Outlet diameter	331.5/430 mm
Inlet vane angle	10 degree

Fig. 2 Locations of pressure gauges on the impeller and the rotating frame o - xy (fixed to the impeller)

Fig. 3 Locations of strain gauges for measuring fluid forces

bending from the radial fluid forces and that from the fluid moments were both included in the output of the strain gauges, it was assumed that the measured bending strains were all from the radial forces acting on the gravity center of the impeller. Since the radial fluid forces were measured in the rotating frame, the gravity and buoyancy acting on the impeller influenced the measurement results. This influence was removed from the measurement data by using the rotating signals of the shaft as measured by a photoelectric sensor.

The rotating frame was fixed to the impeller. The relative locations of the coordinate axes in relation to the impeller blades are shown in Fig. 2. The fixed frame was fixed to the stationary casing as shown in Fig. 1.

3. Pressure Fluctuations

The time history of the pressure fluctuations measured at the pressure side of the impeller blade trailing edge for various flow rates in the case of GV7 is shown in Fig. 4. In the figure, the horizontal axis represents the normalized time by the rotating period T , the vertical axis, the normalized pressure by the head at BEP. It was confirmed that the pressure fluctuations at other measurement points (S1-S5, D1-D5, and R1-R5) were relatively small compared with those at the pressure sides (P1-P5) of the impeller blades. Figure 4 demonstrates that the pressure pulses are upward (pressure increasing) at high flow rates, while downward (pressure decreasing) at partial flow rates. This may be a consequence of the flow

Fig. 4 Blade pressure fluctuations at P1 in the case of GV7

Fig. 5 Blade pressure fluctuations at P1 with the GV11 and GV5 guide vanes at high flow rates

patterns around the leading edges of the guide vanes. Furthermore, the magnitudes of the pressure pulses are not uniform circumferentially during one revolution of the impeller. This unevenness depends on flow rates. At a high flow rate, the pulses are large when the interaction occurs in the front of the volute exit at C7-C9 (see Fig. 1), while the opposite situation can be observed at partial flow rates, i.e. the pulses are small when the interaction occurs at C7-C9. The fluctuations become erose near the shut off condition. The circumferential unevenness of the fluctuations was also observed in the cases of guide vane GV11 and guide vane GV5. Figure 5 shows the pressure fluctuations measured on the blade trailing edge (pressure side, P1) at high flow rates with the GV11 and GV5 guide vanes. The unevenness is relatively significant and the maximum magnitude of the pulses is large in the case of GV11, while it is slight and the maximum magnitude is small in the case of GV5.

Figure 6 shows the frequency spectra of the blade pressure fluctuations at P1 (see Fig. 2) for the three guide vane cases. In addition to Z_gN and its higher harmonic peaks, many side peaks are generated. The frequency difference of two adjacent side peaks is N . The side peaks, which are caused by the circumferential unevenness of the pressure fluctuations, are large in the case of GV11, while small in the case of GV5. This agrees with the fact that the circumferential unevenness is significant in the case of GV11.

It is confirmed that the circumferential unevenness of the blade pressure fluctuations is strongly related to the distribution of the volute static pressure. Figure 7 shows the relationship between the blade pressure pulses at P1 and the volute static pressure during one revolution of the impeller, at a high flow rate in the case of GV7. Distribution of the volute static pressure is also uneven at a high flow rate. It is low near the volute exit⁽¹⁵⁾. The pressure pulses are large when the interaction occurs in areas where the static pressure is low. This may be attributed to the

Fig. 6 Frequency spectra of the blade pressure fluctuations at P1

Fig. 7 Relation between the volute static pressures and the blade pressure fluctuations at P1

high flow velocity in areas where the static pressure is low. Same results were also observed in the cases of GV11 and GV5.

Figures 8 and 9 show the frequency spectra of the pressure fluctuations in the volute in the cases of GV7 and GV11, measured at C9 on the casing (see Fig. 1). Although the most dominant frequency component of the pressure fluctuations acting on the rotating impeller is Z_gN for all the three guide vane cases, that of the pressure fluctuations in the volute depends on guide vanes. The $Z_rN = 6N$ component is the most dominant one in the

Fig. 8 Frequency spectra of the pressure fluctuations measured at C9 on the volute casing in the case of GV7

Fig. 9 Frequency spectra of the pressure fluctuations measured at C9 on the volute casing in the case of GV11

Fig. 10 Phases of the pressure fluctuations measured at P1-P5 on the blades of the impeller

cases of GV7 and GV5, while $2ZrN = 12N$ is the most dominant component in the case of GV11. The circumferential propagation of the pressure fluctuations both acting on the impeller and acting on the casing was investigated by the phases at various measurement points. Figure 10 shows the phases of the blade fluctuations based on the phase at point P1 (ZgN component). Figure 11 gives the

Fig. 11 Phases of the pressure fluctuations measured at C1-C9 on the volute casing

phases of the volute fluctuations based on the phase at point C1 (ZrN component in the cases of GV5 and GV7, $2ZrN$ component in the case of GV11). The two figures demonstrate that both the blade fluctuations and the volute fluctuations propagate circumferentially. They propagate in the same direction as the rotating impeller (*forward*) in the cases of GV11 and GV5, in both the rotating and the fixed frames, while they propagate in the opposite direction in the case of GV7 (*backward*). The most dominant frequency and the propagating direction were found not to be dependent on the flow rate but to depend on the combination of the impeller blades and the guide vanes. This issue will be further discussed in a subsequent section.

4. Radial Fluid Forces

Frequency spectra of the radial fluid forces in the rotating frame are given in Fig. 12 (GV11), Fig. 13 (GV7) and Fig. 14 (GV5). The forces plotted in Figs. 12–14 are those in the x direction in the rotating frame. It is confirmed that the forces in the y direction are almost the same. The forces are shown to be relatively small near the BEP flow rate, while large at high or partial flow rates. As for frequency components, although ZgN is the most dominant one in all the three guide vane cases, other components can also be observed: N and $13N$ in the case of GV11; N and $5N$ in the case of GV7; N and $7N$ in the case of GV5. Zhang et al.⁽⁴⁾, who measured the combination of ($Zg = 7$, $Zr = 6$) and ($Zg = 5$, $Zr = 6$), reported similar results. The reason is believed to be the circumferential unevenness of the pressure fluctuations. This will be further discussed in the next section.

Lissajous patterns of various frequency components of the radial fluid forces in the rotating frame are shown in Fig. 15. The N component whirls in a direction opposite from that of the rotating impeller. This is because the casing rotates in the opposite direction when it is viewed from the rotating frame. The ZgN components whirl in the same directions as the propagating pressure fluctuations.

Fig. 12 Radial fluid forces in the rotating frame (GV11)

Fig. 13 Radial fluid forces in the rotating frame (GV7)

Fig. 14 Radial fluid forces in the rotating frame (GV5)

The whirling directions of other components (13N in the case of GV11, 5N in the case of GV7 and 7N in the case of GV5) are opposite from those of the Z_gN components.

(a) GV11 ($Z_g = 11$, 600 rpm, $Q/Q_0 = 168\%$) (b) GV7 ($Z_g = 7$, 600 rpm, $Q/Q_0 = 157\%$) (c) GV5 ($Z_g = 5$, 600 rpm, $Q/Q_0 = 167\%$)

Fig. 15 Lissajous patterns of the radial fluid forces in the rotating frame

Fig. 16 Radial fluid forces in the fixed frame (GV11)

The radial fluid forces in the rotating frame were transformed into the fixed frame and the frequency spectra thus obtained are shown in Fig. 16 (GV11), Fig. 17 (GV7), and Fig. 18 (GV5). The dominant frequency components

are the same as those of the pressure fluctuations in the fixed frame: $2ZrN = 12N$ is the most dominant one in the case of GV11, while $ZrN = 6N$ is the most dominant one in the cases of GV7 and GV5. Lissajous patterns of the fluid forces in the fixed frame are shown in Fig. 19. They whirl in the same directions as the pressure fluctuations propagate in even in the fixed frame. They give a backward excitation to the shaft in the case of GV7, but a forward excitation in the cases of GV11 and GV5.

Fig. 17 Radial fluid forces in the fixed frame (GV7)

Fig. 18 Radial fluid forces in the fixed frame (GV5)

Fig. 19 Lissajous patterns of the radial fluid forces in the fixed frame

5. Discussions on Frequency Components

The propagating directions of the pressure fluctuations, the whirling directions of the fluid forces, as well as their frequency components will be discussed herein. Figure 20 is a schematic of the interaction between the impeller blades and the guide vanes. If there is no common measure between the number of the blades and that of the vanes, interaction occurs once the impeller rotates at an angle of $2\pi/(ZrZg)$, or in other words, once the time passes $\Delta t = 2\pi/(NZrZg)$. We number the impeller blades and the guide vanes 0, 1, 2, ... The numbers take positive values if they are counted in the rotating direction of the impeller, while they take negative values if they are counted in the opposite direction. We assume that blade 0 is passing through vane 0 at $t = 0$, and at time $t = \Delta t = 2\pi/(NZrZg)$, blade m and vane n will be interacting; then

$$n \frac{2\pi}{Zg} - m \frac{2\pi}{Zr} = \frac{2\pi}{ZgZr} \quad (1)$$

Let the integers m and n take positive values, we get

$$nZr - mZg = \pm 1 \quad (2)$$

In the fixed frame, the pressure fluctuations propagate circumferentially at an angle of $n \cdot 2\pi/Zg$ once the impeller rotates $2\pi/(ZrZg)$. The propagating speed of the pressure fluctuations is nZr times the rotating speed of the impeller, so $nZrN$ will be the most dominant frequency component in the fixed frame. Accordingly, we can conclude that $mZgN$ is the most dominant component in the rotating frame. That is why $2ZrN$ is the most dominant component in the fixed frame in the case of GV11, while ZrN is the most dominant one in the cases of GV7 and GV5. This is demonstrated in Eq. (2). In experiments or field measurements, it is well observed that higher harmonic components are more dominant than ZrN or ZgN components⁽¹⁰⁾. Equation (2) gives a physical explanation in many cases. The pressure fluctuations propagate in the same direction as the rotating impeller in the case of a "+" sign in Eq. (2) (*Forward*), while they propagate in the op-

Fig. 20 Schematic of rotor-stator interaction

Table 2 m, n satisfying $nZr - mZg = \pm 1$ and the propagating directions of the pressure fluctuations estimated by the equation

Zr	Zg	m	n	Sign	Propagating direction
6	11	1	2	+	Forward
6	7	1	1	-	Backward
6	5	1	1	+	Forward

posite direction in the case of a “-” sign (*Backward*), regardless of whether they are measured in the rotating or the fixed frames. Table 2 gives a list of the propagating directions of the pressure fluctuations estimated by Eq. (2) for the three guide vane cases. The estimation agrees with the measured results.

Let us here consider the frequency components of the radial fluid forces measured in the rotating frame. The radial fluid forces are the results of the integration of the impeller pressure fluctuations. We suppose here that the radial force, as a vector, acts towards the gravity center of the impeller (the direction of the force is not important in the discussion of frequency components). As discussed in the above paragraph, if blade m and vane n are interacting at time $t = \Delta t = 2\pi/(NZrZg)$ and a force $\mathbf{P}_{m,n}$ is generated, then, when the h th interaction occurs at time $t = h\Delta t = 2\pi h/(NZrZg)$, blade hm and vane hn will be interacting and a force $\mathbf{P}_{hm,hn}$ will be generated (the impeller blades and the guide vanes are supposed to be counted endlessly with periods Zr and Zg , respectively). Analogically, the force in the x -direction in the rotating frame can be expressed as

$$f_x(t) = \mathbf{P}_{hm,hn} \cos(mZgNt) \quad (3)$$

If the magnitude of $\mathbf{P}_{hm,hn}$ is constant—i.e. the pressure fluctuations acting on the impeller are uniform circumferentially—the most dominant component of the fluid forces in the rotating frame will be $mZgN$, where m is an integer satisfying Eq. (2). However, the fluctuations are not uniform and $\mathbf{P}_{hm,hn}$ is not constant under off-design operating conditions, as shown in Figs. 4 and 5. It was confirmed in the experiment that $\mathbf{P}_{hm,hn}$ depends only on hn , the position in the volute. It does not depend on hm , the impeller blades. Although $\mathbf{P}_{hm,hn}$ is not a harmonic function of time, it is a periodic function with the same period as the rotating impeller. Express it as an even function by a Fourier series, we get

$$\begin{aligned} \mathbf{P}_{hm,hn} &= a_0 + a_1 \cos(hn \cdot 2\pi/Zg) \\ &\quad + a_2 \cos(2hn \cdot 2\pi/Zg) + \cdots \\ &= a_0 + a_1 \cos(nZrNt) + a_2 \cos(2nZrNt) + \cdots \end{aligned} \quad (4)$$

where $a_0, a_1, a_2, \dots$ are integration constants. Disregarding items higher than the second order and substituting Eq. (4) into Eq. (3) we have

Table 3 m, n satisfying $nZr - mZg = \pm 1$ and the dominant frequencies of the fluid forces in the rotating frame

Zr	Zg	m	n	Dominant frequency
6	11	1	2	$(nZr - mZg)N = (2Zr - Zg)N = 1N$
				$mZgN = ZgN = 11N$
				$(2nZr - mZg)N = (4Zr - Zg)N = 13N$
				$(nZr + mZg)N = (2Zr + Zg)N = 23N$
				$(2nZr + mZg)N = (4Zr + Zg)N = 35N$
6	7	1	1	$(nZr - mZg)N = (Zr - Zg)N = 1N$
				$(2nZr - mZg)N = (2Zr - Zg)N = 5N$
				$mZgN = ZgN = 7N$
				$(nZr + mZg)N = (Zr + Zg)N = 13N$
				$(2nZr + mZg)N = (2Zr + Zg)N = 19N$
6	5	1	1	$(nZr - mZg)N = (Zr - Zg)N = 1N$
				$mZgN = ZgN = 5N$
				$(2nZr - mZg)N = (2Zr - Zg)N = 7N$
				$(nZr + mZg)N = (Zr + Zg)N = 11N$
				$(2nZr + mZg)N = (2Zr + Zg)N = 17N$

$$\begin{aligned} f_x(t) &= a_0 \cos(mZgNt) + a_1 \cos(nZrNt) \cdot \cos(mZgNt) \\ &\quad + a_2 \cos(2nZrNt) \cdot \cos(mZgNt) + \cdots \end{aligned} \quad (5)$$

The frequency component of the first item in Eq. (5) is $mZgN$; the components of the second item are $(nZr + mZg)N$ and $(nZr - mZg)N = 1N$; those of the third item are $(2nZr + mZg)N$ and $(2nZr - mZg)N$. These are what we observed in the experiment. Table 3 is a list of the frequency components estimated by Eq. (5). The estimation agrees with the measurement shown in Figs. 12–14. Equation (5) demonstrates that the frequency components, in addition to the higher harmonic ones of $mZgN$, are caused by the circumferential unevenness of the pressure fluctuations.

Since the radial fluid forces are generated from the impeller pressure fluctuations, the whirling directions of the fluid forces are the same as the propagating directions of the fluctuations. When the propagating direction of the fluctuations is the same as that of the rotating impeller, the fluid forces will be a forward whirling excitation to the shaft; otherwise, they will be a backward whirling excitation.

6. Conclusions

The pressure fluctuations, the radial fluid forces acting on the impeller, and the pressures in the volute in a centrifugal pump with vaned diffusers were measured simultaneously, and their relationships were investigated. The experiment was done for various guide vanes, flow rates, and rotating speeds. It was found that:

(1) The propagating/whirling directions and the dominant frequency components of both the pressure fluctuations and the radial fluid forces can be estimated by $nZr - mZg = \pm 1$ if there is no common measure between the number of impeller blades Zr and the number of guide

vaned Zg. In the case of a “+” sign, the propagating direction of the pressure fluctuations is forward and the radial fluid forces are a forward-whirling excitation to the shaft in both the rotating frame and the fixed frame, while it is backward and the radial fluid forces are a backward-whirling excitation in the case of a “-” sign.

(2) The pressure fluctuations acting on the impeller are large on the blade pressure side. They are uneven circumferentially under off-design operating conditions. The unevenness has a strong relationship with the volute static pressure, which is also uneven circumferentially. At a high flow rate, the blade pressure fluctuations are large in areas where the volute static pressure is low. The unevenness causes not only sidebands in frequency spectra of the pressure fluctuations but also other harmonic components in addition to $mZgN$ in frequency spectra of the fluid forces in the rotating frame. The components, which are predictable, are $(nZr + mZg)N$, $(nZr - mZg)N = 1N$, $(2nZr - mZg)N$, $(2nZr + mZg)N$, etc.

Supplement

This study was done when Dr. Guo, first author, belonged Ebara Research Co., Ltd.

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