Evaluation of cumulative damage failure of a reactor feed vertical pump under seismic excitation

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1 Introduction

In a nuclear power plant a vertical pump is used as an emergency pump against earthquake excitation. This type of pump has a merit that the settled area of the pump is small, on the other hand shaft length of the rotor is long and the structure is soft. Thus natural frequency of the shaft becomes low and deflection of the shaft becomes large. When the pump is subjected to a strong seismic excitation, impeller may be touched to a wearing. If the excitation is large, the water film between the impeller shroud and the wearing breaks. In this case, torsional vibration of the shaft is excited by the friction. If the national frequency of the shaft coincides with the predominant frequency of the seismic excitation, the torsional vibration becomes large by resonance. Thus cumulative damage due to shear stress increases and finally the shaft may be broken.

This paper deals with the evaluation of cumulative damage of the shaft of the vertical pump subjected to earthquake excitation. First, the friction force between the wearing and the impeller due to the break of water film is measured by experimental apparatus. Second, the fluid force between the impeller and the wearing is derived from Bernoulli's equation and is represented by nonlinear equation containing added mass, added damping and added spring terms. Third, the equation of motion is derived and is integrated by Newmark's β method. Fourth, the torsional moment due to impact is calculated then the maximum shear stress of the shaft is counted by both rain flow method and hysteresis loop method in order to estimate the cumulative damage, and the probability of the failure of the shaft is evaluated.

2 Modeling of the Vertical Pump

2.1 Equation of lateral motion

Vertical pump for nuclear power is shown in Fig.1 and it is considered as a simple shaft, rotor and casing system. Fluid force and nonlinear characteristics due to collision between reinering and casing are considered in this analysis. Neglecting a internal damping and gyromoment, equation of motion is represented as

$$
\left.
\begin{aligned}
&\ddot{x}+2\zeta\omega_n\dot{x}+\omega_n^2\, x = e_g\omega^2\cos\omega t+F_x/m+F'_x/m \\
&\ddot{y}+2\zeta\omega_n\dot{y}+\omega_n^2\, y = e_g\omega^2\cos\omega t+F_y/m+F'_y/m \\
&\qquad 2\zeta\omega_n=c/m\ ,\quad \omega_n^2=k/m
\end{aligned}
\right]
\qquad (1)
$$

where m: rotor mass, c: damping coefficient,
k: shaft stiffness
e_g: eccentricity of the rotor, ω: angular velocity
F_x,F_y: external force in x and y directions

When the rotor is rotating without contacting with the casing, the external force F in the equation is represented by a flow induced force which is approximately obtained by the Reynold's equation of the journal bearing. In this analysis infinitely short bearing theory is used and forces of radial direction F_ε and circumferential direction F_θ are obtained as follows ;

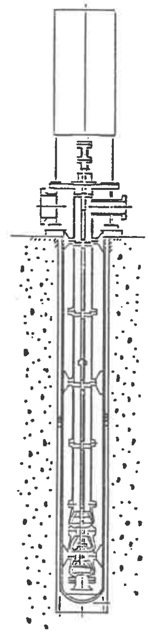


Fig.1 Vertical pump

$$
\left.
\begin{aligned}
&F_\varepsilon=\mu L^3\ \pi\ \dot{\varepsilon}\,(1+2\varepsilon^2)/c_r(1-\varepsilon^2)^{5/2} \\
&F_\theta=\mu L^3\ \pi\ \varepsilon\,(\omega-2\dot{\theta})/2c_r(1-\varepsilon^2)^{5/2} \\
&\qquad \varepsilon=\sqrt{x^2+y^2}/c_r
\end{aligned}
\right]
\qquad (2)
$$

where μ: coefficient of viscocity, c_r: clearance, L: wideth of the rotor
ε: eccentricity ratio, θ: direction angle of the eccentricity

Then the forces of x and y direction, F_x and F_y, are

$$
\left.
\begin{aligned}
&F_x=-F_\varepsilon\cos\theta-F_\theta\sin\theta \\
&F_y=-F_\varepsilon\sin\theta+F_\theta\cos\theta
\end{aligned}
\right]
\qquad (3)
$$

When the rotor is rotating under collision to the casing, external force F is a reaction force and it is represented as the radial force F_r and circumferential force (friction force) F_t as follows,

$$
\left.
\begin{aligned}
&F_r=K(\sqrt{x^2+y^2}-c_r) \\
&F_t=\lambda K(\sqrt{x^2+y^2}-c_r)
\end{aligned}
\right]
\qquad (4)
$$

where K: stiffness of the casing and rotor, λ: friction coefficient

Then the forces of x and y direction, F_x and F_y, are

$$
\left.
\begin{aligned}
&F_x=-F_r\cos\theta+F_t\sin\theta \\
&F_y=-F_r\sin\theta-F_t\cos\theta \\
&\qquad \theta=\tan^{-1}(y/x)
\end{aligned}
\right]
\qquad (5)
$$

When the rotor subject to a seismic excitation, external force of Eq.(1) is represented by Eq.(3) and Eq.(5) for without contacting and with contacting to the casing, respectively.

2.2 Equation of torsional motion

When the rotor contact to the casing, the rotor is excited the torsional vibration by the friction force. This torsional vibration may induce a torsional failure by low cycle fatigue of the shaft. So in this section equation of motion for torsional vibration is represented to analyze the low cycle fatigue of the shaft under the seismic excitation. The equation is

$$\left.\begin{aligned} &\ddot{\gamma} + 2\zeta'\omega_n'\dot{\gamma} + \omega_n'^2 = T/I \\ &2\zeta'\omega_n' = c'/I,\ \omega_n' = k'/I \end{aligned}\right] \qquad (6)$$

where γ: torsional angle, I: inertia moment of rotor-shaft system
C': torsional damping coefficient, k': torsional spring coefficient
T: exciting torque ($=F_t \cdot R$)

2.3 Friction Coefficient

Friction coefficient at the collision is obtained by experimental method. When the rotor is rotating, impulse excitation is added to the rotor and friction force is measured. Fig.2 shows, as an example, the relation between radial force vs tangential force at the rotating speed 30rps. From this figure, friction coefficient λ is obtained by the relation $F_t=\lambda F_r$ for each measured point and its mean value and variance are obtained, which are $\lambda=0.118$, $\sigma_\lambda=0.0012$.

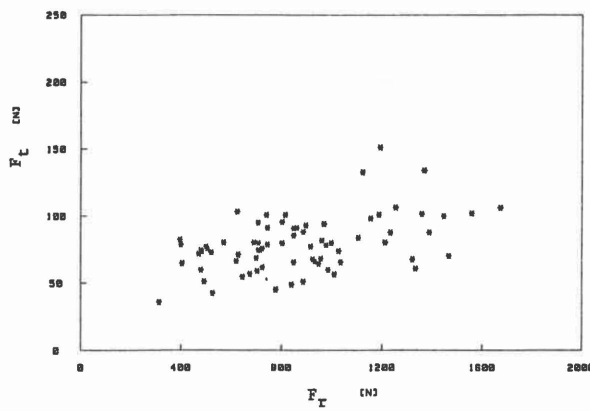


Fig.2 Friction force by experiment

3 Evaluation of Cumulative Damage

The rotor is excited torsional vibration by collision between rotor and casing. The torsional shear stress due to this vibration is represented by torsional angle and torsional moment as

$$\tau = (3T + \theta_L\, dT/d\theta_L)/2\pi r^3 \qquad (7)$$

where τ: maximum torsional shear stress, r: radius of the shaft
θ_L: torsional angle for unit length

When the cumulative damage of the shaft is considered, the shear stress in the shaft surface is almost equivalent to the principal component of stress. Then the cumulative damage D is represented by Meiner's low as,

$$D = \sum_i n_i / N_i \qquad (8)$$

where n_i is number that $\Delta\varepsilon_i$ is occured and N_i the iterative number that the test piece is broken by iterative strain $\Delta\varepsilon_i$. For the cycle count of the iterative force, Rain flow method is used.

4 Numerical Example

4.1 Numerical Integration of the Equation of Motion

The equation of motion derived in section 2 is nonlinear equation, so it is directly integrated by Newmark's β method to obtain the behavior of the rotor.

Time step of the integration is set to $\Delta t=0.002$sec in the region $\varepsilon=0\sim0.9$, $\Delta t=0.0002$sec in the region $\varepsilon=0.9\sim1.0$, $\Delta t=0.0001$sec in the region $\varepsilon=1.0\sim1.1$ and $\Delta t=0.005$sec in the region $\varepsilon\geq1.1$. For the torsional vibration, time step of the integration is set to $\Delta t=0.0001$sec. For the numerical calculation data of a vertical pump is shown in Table 1.

Table 1

Rotor mass (kg)	100	Shaft dia. (mm)	110
Shaft length (mm)	9800	Eccentricity (mm)	0.003
Eigen freq. in lateral (Hz)	20.0	Eigen freq. in torsional (Hz)	92.0
Rotating speed (rad/s)	62.8		

4.2 Motion of shaft center

The motion of the shaft center under seismic excitation is calculated by direct integration by the method described in section 4.

Figs.3 show Lissajous' figure of shaft center after 0.5∿0.75sec of sinusoidal excitation. In this figure, circle shows the clearance circle of which radius is 0.645mm. Figs.3(a) and (b) show the case that the excitation amplitude is 735gal and frequencies are 20Hz and 30Hz, respectively. The collision occurs in each case and the resonance phenomena is not so big. It is known from this result, if the exciting force becomes big enough, the resonance effect is elliminated by the nonlinearity.

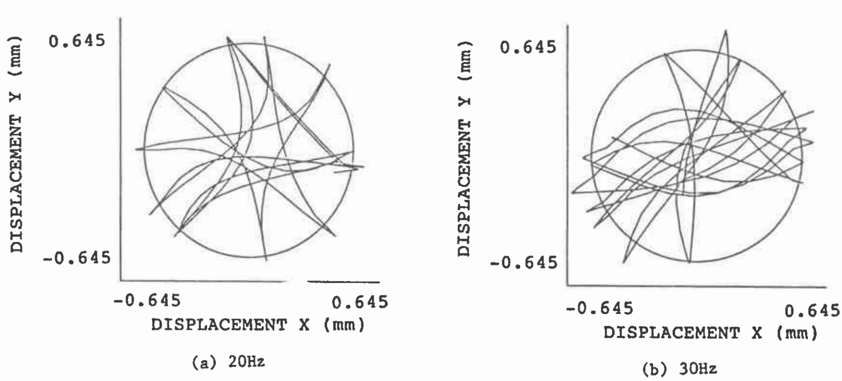
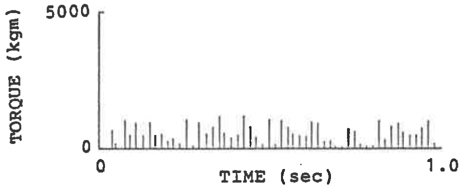


Fig.3 Simulation result of rotor motion (excited acc.=735gal)

Fig.4 shows the exciting torsional torque and responee of torsional torque by collision, in which the excitation is done in two directions. The figure is the results under the condition of 735gal and 20Hz. As the exciting frequency and exciting force increase, torsional torque occures in frequency and force. Maximum torsional angle in each collision is not so much changed by the exciting force and exciting frequency.



735gal-20Hz

Fig.4 Torque occured by collision

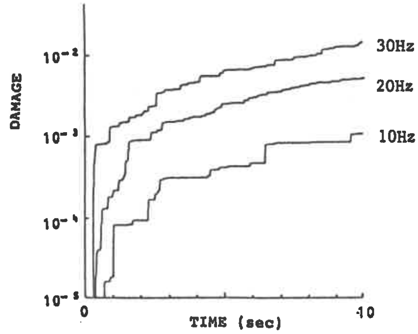
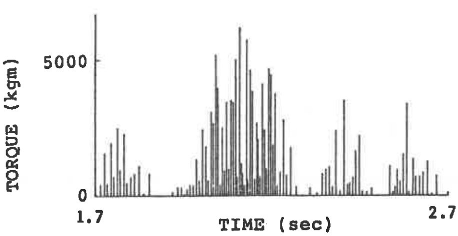


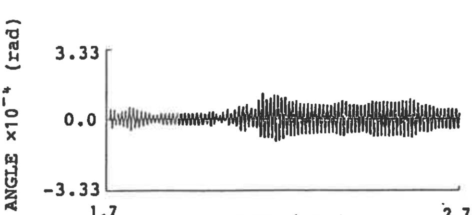
Fig.5 Cumulative damage (excited acc.=1960gal)

4.3 Cumulative damage

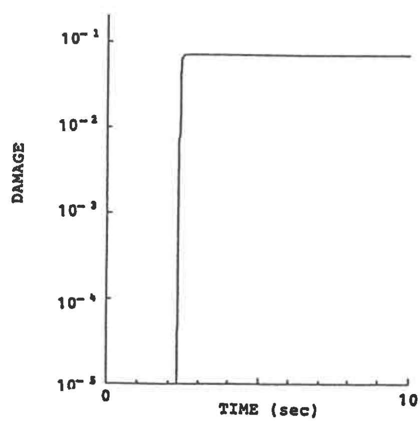
Fig.5 shows the time history of cumulative damage for the case that the rotor is excited sinusoidally in two directions and exciting force and frequency are changed. When the exciting frequency is 10Hz, the cumulative damage is small and as the frequency increases, the cumulative damage increases. Figs.6(a)∿(c) show the exciting torque, torsional angle of the shaft and the cumulative damage for the seismic force of times of Elcentro earthquake. Big acceleration in Elcentro earthquake is occured after 2sec to 2.3sec, so cumulative damage increased suddenly near this time period. Figs.6(a)(b) show the exciting torque and the torsional angle between these period.



(a) Torque excited by collision



(b) Torsional angle



(c) Cumulative damage

Fig.6 Calculation result for 14 times magnified Elcentro seismic

Fig.7 shows the cumulative damage after 10sec of excitation for change of the shaft diameter, where the change of shaft diameter is limited only for the case of consideration of cumulative damage. That is, it means that the natural frequency of the rotor is not changed and only shaft weakness is changed. The excitation wave is sinusoidal and Elcentro earthquake and these amplitude is changed. It is known from this figure that as the shaft diameter increases, the cumulative damage decreases exponentially.

From above results it is concluded that shaft of vertical pump is strong enough for the earthquake excitation.

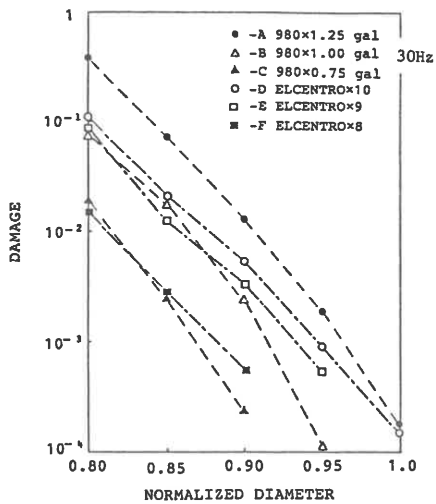


Fig.7 Cumulative damage for shaft diameter

5 Conclusion

From the above analysis, the followings are concluded ;

1. As the rotating speed increases, the friction force decreases, because as the rotating speed increases, the fluid force of lifting direction increases.

2. When the impact force is small, the shaft deflection becomes large in the case that the natural frequency is equal to rotating speed. But when the impact force is large, the effect of natural frequency diminishes and the rotor vibrates like a chatter phenomena, then the shear stress in the shaft increases.

3. The cumulative damage of the shaft due to the torsional vibration is obtained under sinusoidal excitation and Elcentro earthquake excitation. Generally, the damage increases as the vibrational amplitude becomes large. Finally it is concluded that there is little possibilities of failure by earthquake excitation in real pump shaft.

REFERENCES

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