Dynamic and Vibration Analysis of a Rotating Blade for the HANARO Cooling Fan

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1. Introduction

The HANARO, a multi-purpose research reactor of a 30 MWth, open-tank-in-pool type, has been under normal operation since its initial criticality in February, 1995. The HANARO cooling water system is composed of a heat exchanger, cooling water pumps and a cooling tower. The cooling tower in HANARO enables the reactor to operate normally due to the heat removal process generated by nuclear fission. The cooling tower is equipped with a cooling fan in order to move the air through which absorbed heat of the 2nd coolant. In this paper, dynamic analysis of the rotating blade in the fan is calculated chronologically by a computer simulation method. Also, variable natural frequencies of the blade associated with the angular speed of the hub are calculated to analyze the influence on the resonance.

2. System Modeling

Generally, a rotating blade is able to be assumed as an Euler - Bernoulli beam which can be disregarded for the shear and rotary inertia effect. Fig. 1 represents the blade before and after the deformation in the reference frame.

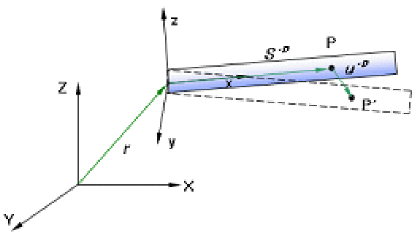

Figure 1. Configuration of an Euler-Bernoulli beam undergoing. Elastic deformation

In order to derive the equations of a motion, the E.O.M employing the principle of virtual work is calculated as follows

$$-\int_{\Omega}\mu\delta r^{P^T} r^P d\Omega+\int_{\Omega}\delta r^{P^T} f^P d\Omega+\int_{\sigma}\delta r^{P^T} T^P d\sigma = \int_{\Omega}\delta\varepsilon^{P^T} r^P d\Omega=\delta W \quad (1)$$

The first term is the inertia force, the second term is the body force, the third term is the surface force and the last term of the right side of the equation is the elastic force respectively.

After differentiating it twice and substituting this to equation (1) and by adding the acceleration equation, the final form is expressed as follows.

$$\begin{bmatrix} M & \Phi_q^T \\ \Phi_q & 0 \end{bmatrix}\begin{bmatrix} \ddot{q} \\ \lambda \end{bmatrix}=\begin{bmatrix} Q \\ \gamma \end{bmatrix} \quad (2)$$

Here, to reduce the number of variables, equation (2) can be transformed into a mode function.

For rotating blades, the deformation of the blade is not changed linearly and a geometric stiffness effect must be considered due to the overall motion (geometric nonlinear) and a geometric stiffness matrix must be added to this equation. Therefore, finally the modal equation employing a geometric stiffness term can be expressed as follows

$$M\ddot{q}_f+\hat{K}q_f+(-\Omega^2\hat{M})q_f+M_\alpha q_f+S\dot{q}_f=Q' \quad (3)$$

3. Numerical Analysis and Results

Given system consists of a hub and a rotating blade and the degrees of freedom (D.O.F) between the two bodies are all constrained. Fig. 2 shows the system modeling by combining the hub with the blade

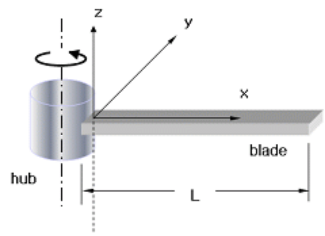

Figure 2. System modeling of the rotating blade

Until the dynamic equilibrium state is settled, imposed angular speed of the hub is assumed as follows

$$\omega=\frac{\Omega_s}{T_s}\left[t-\frac{T_s}{2\pi}\sin\left(\frac{2\pi t}{T_s}\right)\right] \quad (4)$$

Where, Ω_s is the angular speed of the hub at a dynamic equilibrium and T_s is the settling time by the dynamic equilibrium.

By using these relations, in the case that ω is 6 rad/s the deformation results of the blade versus time are shown in Fig. 3.

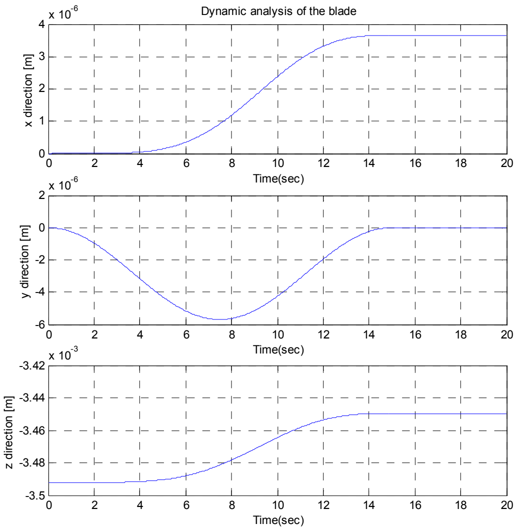

Figure 3. Deformation graph of the blade at 6 rad/s

The first graph in Fig.3 shows the deformation of the *x* direction. As the angular speed is increased, the length of the blade is increased in the *x* direction by the centrifugal effect and this trend is similar to that of the angular speed of the hub. The second result represents the deformation of the *y* direction. By increasing the angular speed and acceleration, the deflection of the blade moves in the opposite direction. This effect is well known for a membrane locking. This trend is similar to that of the opposite value for the angular acceleration. And finally, the last graph shows the deformation of the *z* direction. In the initial state, the blade is deflected due to the gravity force of the blade. When the angular speed increases, the deflection of the *z* direction becomes small. It is estimated that the centrifugal force of direction applies to the direction to maintain the blade horizontally.

This time, the vibration characteristics of the blade were simulated at a variable angular speed. These results are shown in Fig. 4.

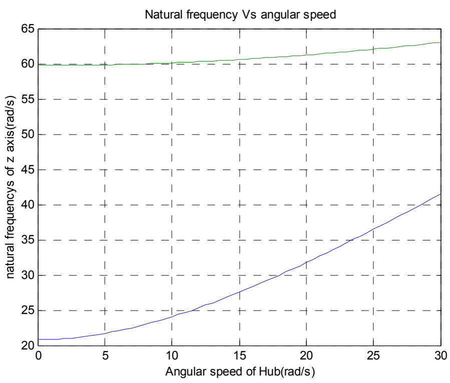

Figure 4. Natural frequency variation versus angular speed

Where, the lower curve (blue line) represents the 1st natural frequency and the upper curve (green line) represents the 2nd natural frequency. These results show that the natural frequencies of the system are increased in monotone. In this figure, the 2nd natural frequency is not related to the resonance effect because the maximum angular speed of the hub is about 30 rad/s and the 1st natural frequency has a deviation of 30 % for the angular speed of the hub. So, it is estimated that this system is stable for the resonance effect.

4. Conclusions

In this paper, the dynamic analysis of the rotating blade as a part of the HANARO cooling tower is simulated by using the numerical integration method. The multibody dynamic equation which contains the *x* direction has been derived without ignoring this term in spite of the large natural frequency. As a result, the deformation of the blade for each direction is not much higher than expected. Finally, natural frequency variation due to the MISV(Motion Induced Stiffness Variation) has no problem for operating the cooling fan. And it shows that resonance effect does not occur for the system because the difference between excited and natural frequency is much higher.

Acknowledgements

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