Diagnosis of Excessive Vibration Signals of Two-Pole Generator Rotors in Balancing

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Cause of excessive vibration with twice the rotational speed of a two-pole generator rotor for the fossil power plants was investigated. The two-pole generator rotor, treated as a typically asymmetric rotor in vibration analysis, produces asynchronous vibration with twice the rotational speed, sub-harmonic critical speeds, and potentially unstable operating zones due to its own inertia and/or stiffness asymmetry. This paper introduces a practical balancing procedure, and presents the results of the investigation on sources of the excessive vibration based on the experimental vibration data of the asymmetric two-pole rotor in balancing.

Key Words : Balancing, Run-Out, Asymmetric Rotor, Sub-Harmonic Critical Speed, Asynchronous Vibration, Two-Pole Rotor

1. Introduction

Generator rotors for large-scale fossil power plants are generally designed in the shape of two poles, and have the complicated cross-sectional structure, as shown in Fig. 1. In spite of the care taken in design, manufacture, and assembly of the rotor for maintaining symmetric structure, weak asymmetry of stiffness and/or inertia in the radial direction of the rotor cross-section is inevitable due to its complicated structure and assemblies, i. e. rotor slots, vent slots, copper coils, insulation materials, ventilation holes, wedges, etc. The two-pole generator rotor, treated as a typically asymmetric rotor in vibration analysis, produces asynchronous vibration with twice the rotational speed, sub-harmonic critical speeds, and potentially unstable operating zones for its own inertia and/or stiffness asymmetry. (Lee, 1993; Lee and Joh, 1994; Dimarogonas and Haddad, 1992; Ehrich, 1992) This dynamic characteristics is similar to a simple rotor with asymmetric stiffness, as stated in the appendix of this paper. For these reasons, the two-pole generator rotors for large-scale fossil power plants are designed and manufactured to have symmetric structure as much as possible. Allowable limit of the asynchronous vibration is much stricter than that of the synchronous vibration in the balancing process.

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This paper introduces a practical balancing procedure, including runout compensation and thermal sensitivity test, and identifies the sources of the excessive vibration reported in the balance process of the asymmetric two-pole rotor.

2. Balancing of Generator Rotors

2.1 Balancing

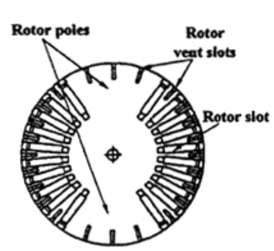
The objective of balancing is to reduce the residual unbalance of rotors to a required level, consistent with the high quality goals. The method used for dynamic balancing of generator rotors in the factory is the modal balancing (Darlow, 1989). Balancing starts at the first mode, called the first critical speed, and proceeds through each mode in turn to the highest mode within the operating speed range. Balancing is completed at the rated speed.

A balance weight distribution, based on the calculation or previous experience, is applied to the mode being balanced. The distribution for each mode is different, and is required to accomplish two things: To be effective in changing the vibration for the mode being balanced, and not to disturb the balance of any lower modes already balanced. The weight distributions are calculated from the mode shapes of the undamped natural frequencies, or critical speeds.

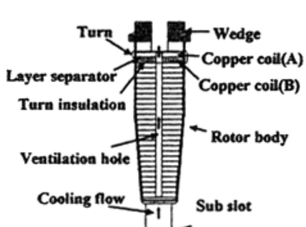
The major balance corrections made on the rotor are in the body section. This is accomplished by changing the length of the threaded steel body plugs in the balance planes of the poles, as shown in Fig. 1 (a): The planes 3~5 and 9~11 are provided for the first critical, 1~2 and 12~13 for the second critical, and 6~8 for the third critical. Other balance planes are located in the main fan rings, collector fan ring, turbine coupling, and either collector-end coupling or collector-end balance ring.

Table 1 represents specifications of balancing facilities, and Fig. 2 shows a generator rotor being balanced in the balance bunker. Vibrations of journals, couplings and collectors in the y- and z-directions are measured using each pairs of proximity probes mounted perpendicular to each other in the bearings and fixtures, respectively. Vibrations of the bearing pedestals in the y- and z-directions are measured using velocity transducers.

(a) Overall view



(b) Transverse-cross section



(c) Field coil and slot

Fig. 1 Structure of a two-pole generator rotor

A low speed (rigid-body) balance is generally not performed on generator rotors prior to the high speed balance. An initial low speed balance does not provide any productivity advantage as high speed balancing is the best way to remove the temporary gravity sag which affects the low speed balance results. The procedure of the high speed modal balance for the generator rotor in the balance bunker is outlined as follows (Hanjung, 1994):

(a) Take an initial speed-vibration curve to determine critical speeds and balancing speeds, and reach over-speed (105% of rated speed).

-Starting at 500 rpm, take once-per-revolution (1x) vibration data every 100 rpm at all probes. Take the generator rotor to as high a speed as

Table 1 Specifications of balancing facilities

Rotor weight range	1.6~320 ton
Max. Rotor length	22.0 m
Max. Rotor diameter	6.0 m
Max. Rotating speed	8,000 rpm
Vacuum	1.0 mbar

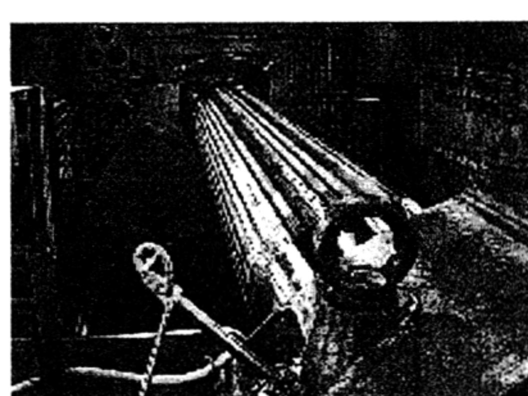


Fig. 2 Generator rotor in balance bunker

possible toward over-speed without exceeding vibration limits.

-Balance, as necessary, to reach over-speed.

(b) Refine the balance at each critical speed and at rated speed to the levels required to start the thermal test

(c) Perform the thermal test and check the rotor for shorted turns.

(d) Refine the balance to within the limits specified for passout. Passout vibration limits for generator rotor balance, 1x, peak-to-peak, are 50 μm for journals, 76 μm for collectors and couplings (overhangs).

(e) Start 105% rated speed and record 1x and twice-per-revolution (2x) vibration data on all probes every 100 rpm down to 500 rpm. At rated speed, record response spectra from 0-200 Hz from all probes. The vibration at the criticals must also be within the passout limits. The 2x vibration limits are 12.7 μm for journals and overhangs at rated speed, and 12.7 μm for journals and 25.4 μm for overhangs at remainder of speed range. In addition, the response at any probe location, at any frequency other than 1x or 2x, should not exceed 7.6 μm peak-to-peak per the response spectra at rated speed.

A great deal of effort and care are taken during manufacture and assembly directed toward producing vibration-free rotors. Mechanical unbalance is not necessarily the only cause of vibration either. Vibration can also be caused by thermal or frictional asymmetry, stiffness asymmetry of the rotor cross-section, or by poorly machined journals. All of these factors must be considered during the design and manufacturing stages.

2.2 Runout subtraction

When vibration measurements are taken in location where runout may exist (such as a coupling periphery, a collector, etc.) the measuring probe will respond to the runout and the vibration. Hence vibration probes at these locations on a perfectly balanced rotor will indicate the runout magnitude rather than zero. (Wowk, 1991; Collacott, 1979)

In order to determine the vibration of the shaft due to unbalance the runout vector must be subtracted from the probe reading. The runout vector is obtained by taking data from the vibration probes at a speed at which the rotor is not bending (less than half the first critical speed) so that the runout vector can be measured. (Hanjung, 1994) Once the runout vector is known it can be vectorially subtracted from any vibration reading to obtain the actual vibration due to unbalance.

The proximity probes, in addition to responding to the mechanical runout, are also sensitive to changes in the material properties of rotor shaft periphery. This effect has been termed electrical runout. This type of runout is also subtracted by the procedure described above.

The engineering process instruction (Hanjung, 1994) allows two steps to determine if the final balance is acceptable or not: If the measured readings are within the acceptable targets without subtracting runout the balance is acceptable. If any measurements are beyond the acceptable targets then the runout readings taken at 500 rpm can be subtracted. If the resulting values are within the targets the balance is acceptable.

The purest way to handle runout is to subtract it from all readings as is required for other rotors. The runout for generator rotor scan be ignoreed because of the high sensitivity to unbalance and the complex mode shapes of the higher criticals.

2.3 Thermal sensitivity test

Thermal sensitivity is defined as a change in vibration due to application of field current to the rotor. Rotors in both the factory and field exhibit thermal sensitivity. Objectionable thermal sensitivity can be caused by uneven circumferential temperature or axial force distributions around the rotor due to several mechanisms-shorted field turns, blocked ventilation passage, asymmetries in the friction of the slot contents, or a combination of these items. (Hanjung, 1994; Watanabe, 1996) The primary driver of this second cause is the large difference in thermal expansion coefficients between copper coils and the steel alloy rotor and components. If the rotor winding is not balanced both electrically and mechanically in the circumferential direction, the generator rotor will be unevenly loaded which can cause the rotor to bow and the vibration to change.

A measurement of this change is made during the factory balance cycle to insure that large vibration changes do not occur due to heating the rotor. In general, this test is performed at either 100 rpm above the third critical speed, or at rated speed, whichever is lower. Also, as the final step of the thermal test procedure, a specific test for shorted turns, called a flux probe test, is performed. (Hanjung, 1994)

Compromise balancing (Hanjung, 1994; Watanabe, 1996) has been successfully used to minimize the vibration on thermally sensitive rotors. If the thermal vector is larger than a prespecified value, then it will become necessary to bring the unit down and to correct the cause of the thermal sensitivity.

3. Vibration Signals and Analysis

Balancing engineers reported that in spite of a great deal of balance work, the twice-per-revolution (2x) vibration level of the coupling in the z-direction at rated speed 3600rpm, could not be reduced to the target value -12.7 μm for journals and overhangs at rated speed, 12.7 μm for journals and 25.4 μm for overhangs at remainder of speed range, though synchronous (1x) vibration level of the generator rotor, was reduced to the target values -50 μm peak-to-peak for journals and 76 μm for overhangs, as shown in Figs. 3 and 4. Figures 5 and 6 show Bode plots of the corresponding 1x and 2x vibration signals measured on bearing pedestals using velocity transducers.

Figure 5 clearly shows that the first, second, and third critical speeds of the generator rotor exist at 980, 2500, and 3550 rpm, respectively. Vibration peaks at 500, 1250, 1770, 2230 rpm in Fig. 6 are caused by each sub-harmonic resonance of the critical speeds due to the asymmetry of the generator rotor, like the asymmetric-simple rotor as described in the appendix. Modal analysis of the generator rotor in the balance cell using the laboratory-developed finite element S/W resulted in the critical speeds of 994, 2430, 3533,

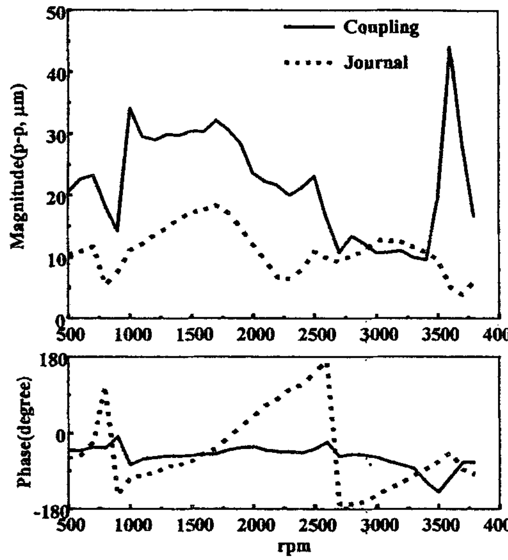


Fig. 3 1x vibration in z-direction: uncompensated

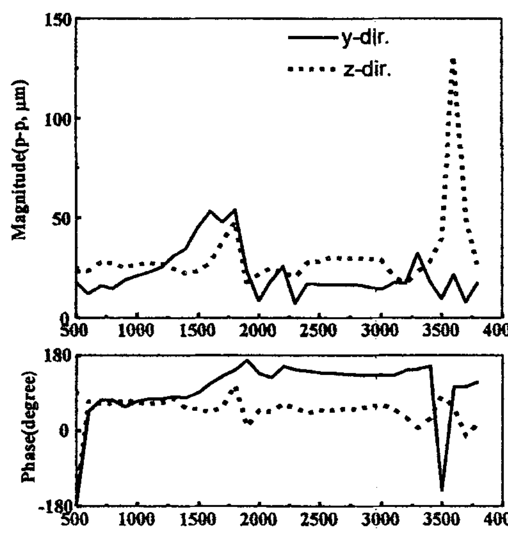


Fig. 4 2x vibration of coupling: compensated at 500 rpm

and 4400 rpm, which supports the above observations. In the analysis, the generator rotor was modeled as symmetric, and the employed S/W takes the gyroscopic and shear effects of rotors into consideration.

The 2x vibration peak in the y-direction at 3600 rpm in Fig. 4 is not observed. The excessive 2x vibration level of 120 Hz in the z-direction at rated speed is irrelevant to any critical speeds and

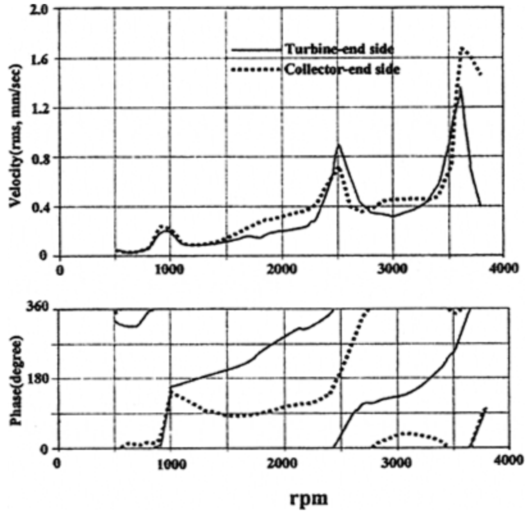


Fig. 5 1x vibration of bearing pedestals in z-direction

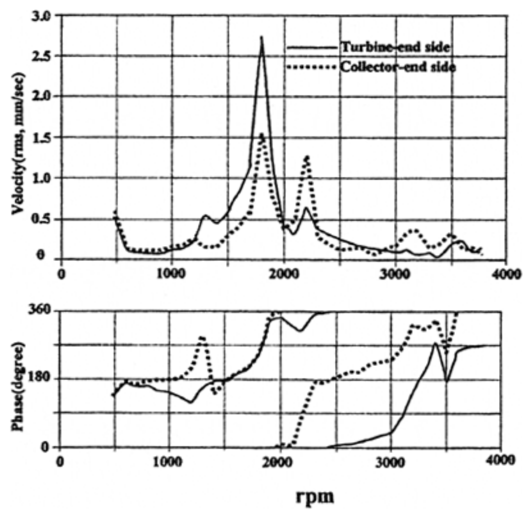


Fig. 6 2x vibration of bearing pedestals in z-direction

sub-harmonic resonances of the generator rotor, as shown in Figs. 5 and 6. These observations suggest that a cause of the excessive vibration level is relevant to possible local resonance at 120 Hz of a fixture supporting the z-direction proximity probe. The excitation source of 120 Hz can come from 2x vibration of the asymmetric generator rotor in rotation at the speed of 3600 rpm.

An impact test on the fixture was performed to investigate the possible local resonance of the fixture. Figure 7 shows the frequency response

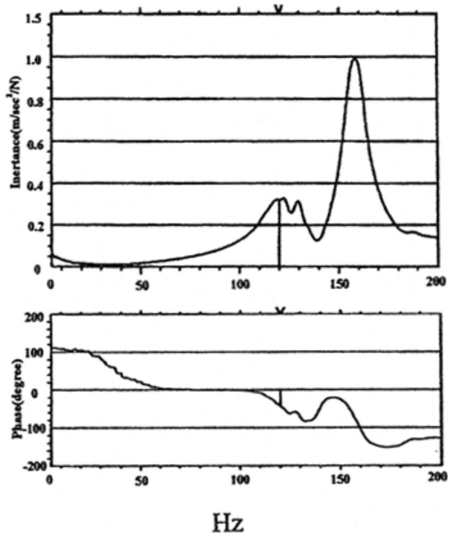


Fig. 7 Frequency response function of fixture

function of the fixture, where the first natural frequency of 120 Hz is clearly observed. The problem of the excessive 2x vibration level of 120 Hz in the z-direction at the rated speed was solved after reinforcing the fixture to have its first natural frequency beyond 150 Hz.

Balancing engineers also have reported that in spite of many times of balance work and regardless of compensating vibration signals with the runout vector of the generator rotor, the 2x vibration level of the journal exceeded the target value, 12.7 μm, as shown in Figs. 8 and 9. The runout vector was taken at 500 rpm according to our engineering process instruction for balance work.

As described earlier, the first critical speed of the generator rotor in balance cell exists at 980 rpm, as shown in Fig. 5, and thus the first sub-harmonic critical speed due to asymmetry of the generator rotor also exists around 500 rpm, as shown in Figs. 6 and A-2 in Appendix. The runout vector taken at 500 rpm includes actual vibration due to the sub-harmonic resonance as well as pure runout of the journal. Therefore, the runout vector taken at 500 rpm is no longer correct to be used for the runout compensation. Compensating the vibration signal with a new runout vector taken at 350 rpm far from the sub-harmonic critical speed have brought the vibration to an acceptable level as shown in Fig. 10.

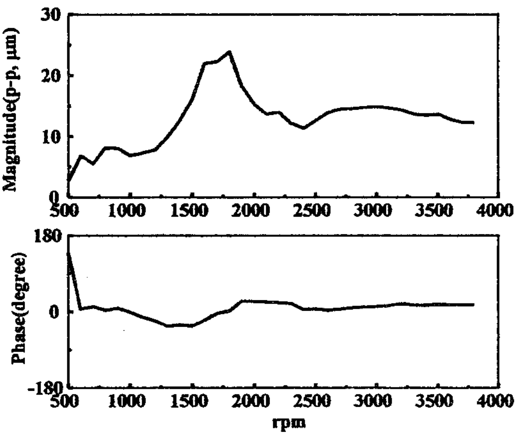


Fig. 8 2x vibration of journal in z-direction: compensated at 500 rpm

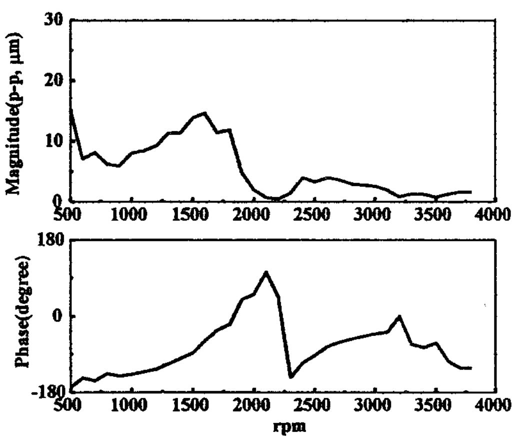


Fig. 9 2x vibration of journal in z-direction: uncompensated

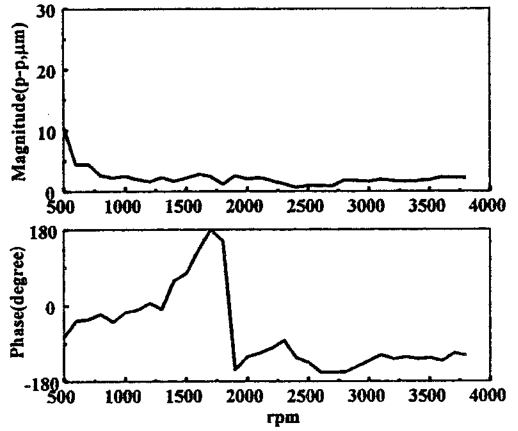


Fig. 10 2x vibration of journal in z-direction: compensated at 350 rpm

4. Summary and Conclusions

This paper introduces a practical balance procedure, including runout compensation and thermal sensitivity test, and presents the results of investigation on sources of the excessive vibration signals based on the experimental vibration data of the asymmetric two-pole generator rotor in balancing. The main source of the excessive vibration signals was turned out to be twice-per-revolution vibration and sub-harmonic critical speeds coming from the asymmetric structure of the rotor. In balancing and operating weakly asymmetric rotors like two-pole generator rotors, their dynamic behaviors should be taken into account for smooth and efficient balance work, in particular determination of the rotor speed for runout measurement, and smooth operation.

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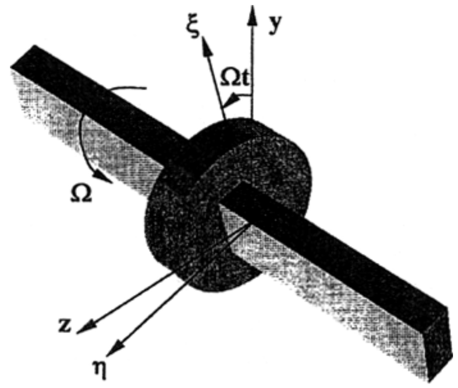

Fig. A-1 Simple rotor with stiffness asymmetry

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Appendix

Vibratory Characteristics of a Simple Rotor with Asymmetric Stiffness

The equations of motion for a simple, horizontal undamped rotor consisting of a single central disk on a massless shaft with the asymmetric stiffness k_ξ and k_η in the rotating coordinate ξ and η directions, as shown in Fig. A-1, can be written as (Lee, 1993)

$$m(\ddot{\zeta}+2j\Omega\dot{\zeta}-\Omega^2\zeta)+k\zeta+\Delta k\bar{\zeta} = me\Omega^2-mg\ exp(-j\Omega t) \qquad \text{(A-1)}$$

where $r=y+jz$, $\zeta=\xi+j\eta$, $r=\zeta\ exp(j\Omega t)$, $k=\frac{k_\xi+k_\eta}{2}$, $\Delta k=\frac{k_\xi-k_\eta}{2}$, the complex mass eccentricity $e=e_\xi+je_\eta$, and j is the unit imaginary number.

The unbalance response of the asymmetric rotor has the form

$$r_u=\left[\frac{e_\xi\Omega^2}{\omega_\xi^2-\Omega^2}+j\frac{e_\eta\Omega^2}{\omega_\eta^2-\Omega^2}\right]exp(j\Omega t) \qquad \text{(A-2)}$$

where $\frac{\Delta k}{k}=\frac{\omega_\xi^2-\omega_\eta^2}{2\omega_o^2}$ and $\omega_o^2=\frac{k}{m}=\frac{\omega_\xi^2+\omega_\eta^2}{2}$.

The response for the effect of gravity on disk has the form

$$r_g=\frac{-g(\omega_\xi^2-\omega_\eta^2)}{4\Omega^2(\omega_\xi^2+\omega_\eta^2)-2\omega_\xi^2\omega_\eta^2}exp(j2\Omega t)+C \qquad \text{(A-3)}$$

where C is a constant. The response of the rotor for the combined effect of gravity and unbalance is obtained by direct summation of the values obtained for each of these factors separately

$$r=r_u+r_g=\zeta_u\ exp(j\Omega t)+\zeta_f\ exp(j2\Omega t)+C \qquad \text{(A-4)}$$

If two stiffness k_ξ and k_η differ only slightly, i. e., $\omega_\xi=\omega_o$, $\omega_\eta=(1+\varepsilon)\omega_o$, , the sub-harmonic critical speed (Ω_c) can be written as

$$\Omega_c\approx\frac{\omega_o}{2}\left[1+\frac{\varepsilon}{2}\right]=\frac{\omega_\xi+\omega_\eta}{4} \qquad \text{(A-5)}$$

where ε is a small number. Figure A-2 shows a typical vibratory characteristics of the simple rotor with stiffness asymmetry.

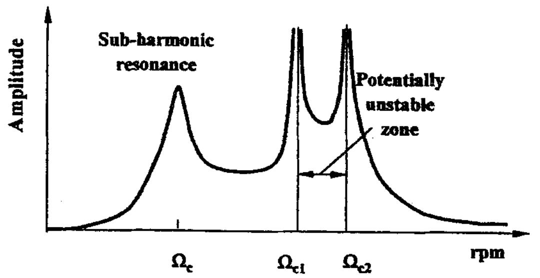

Fig. A-2 Vibratory characteristics of a simple rotor with stiffness asymmetry