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Foundation-rearing-rotor interaction problem in controlling vibrations in a 120 mw turbo generator

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SYNOPSIS  A two-pole turbogenerator posed problem of resonance and excessive vibrations at the operating speed during test runs in the Test-bed. The entire system comprising the foundation, pedestals supporting the generator, journal bearing seats, bearing oil-film, and the rotor has been modeled and analyzed for computing the rotor's critical speeds as also the foundation's natural frequencies. Some of the system's parameters have been derived from actual tests. Sensitivity analysis for the rotor's critical speed that coincided with the operating speed of machine showed that this critical speed is relatively insensitive to the foundation mass as well as stiffness, and largely depends upon the stiffness characteristics of the journal bearings. The remedy to avoid excessive vibrations at operating speed has been thus found to lie in improving the journal bearings and their seating.

INTRODUCTION

A 120 MW capacity two pole turbogenerator set with spherical bearings showed excessive vibrations of the foundation during test runs on the Test-bed. The vibration levels at the Test-bed were observed to be much higher than in the Over-speed Test-cell. Problem of large vibrations had been encountered with some of the similar turbogenerators on actual foundations although the machines had been accurately balanced to well within the permissible eccentricities of their rotors. It was decided to analyse the system comprising the foundation, pedestals, journal bearings and rotor as one system considering the interaction between the machine and the foundation. The analysis presented in this paper brings out the cause of excessive vibrations and indicates the significance of various parameters of the system in controlling the natural frequencies of the system, hence the vibration amplitudes.

TEST DATA

Vibration amplitudes observed in the various test runs of the machine are as follows:

(a) Over-speed Test-Cell
Maximum displacement amplitude at the operating speed of 3000 rpm was 11 micron and at speeds of 1000 and 2250 rpm were 60 and 35 micron respectively. Resonance was observed at 1100 and 2250 rpm. The former coincided with the first critical rotor speed for equivalent rotor support stiffness $K_e = 64.1$ t/mm at each end of the rotor.

(b) Test-Bed
With slip-ring shaft: The amplitudes of vibrations at the journal bearings gradually built up from 40 to more than 100 micron in about two hours, which is possible only near resonance. The temperature of bearing oil stabilised in this duration with accompanying change in the oil-film stiffness and hence the critical speed of rotor.

Without slip-ring shaft: Resonant peaks were observed at 350, 700, 1200, 1400 and 3000 rpm. Whereas the maximum amplitudes of foundation vibrations did not exceed 150 micron at speeds of 350, 1200 and 1400 rpm, the amplitudes went beyond 150 micron at 700 and 3000 rpm speeds, which had to be quickly passed through to avoid further building up of the vibration amplitudes.

Simultaneous observations were made for (i) measurement of displacements using a BTH Rotabalan balance indicator system having least count of 0.01 mils (0.25 micron) and (ii) recording acceleration amplitudes of the various harmonic components present at any running speed, using an optical type vibration frequency analyser. Observations recorded at running speeds of 1000, 1500 and 3000 rpm are reported in Tables I, II and III respectively.

At 1000 rpm speed (Table I) only first two super-harmonics are seen to be significant in contributing to the acceleration response. The acceleration peaks increased to about double the values on raising the machine speed from 1000 to 1500 rpm (Tables I and II). The exact values at 1000 rpm could not be determined due to limitation of the measuring device which was not calibrated for frequencies lower than 1200 rpm.

Table IIIA shows that several superharmonics are excited at the operating speed of 3000 rpm. The second and third superharmonics are of significant acceleration values. Whereas the overall acceleration peak can be many order greater than the 50 Hz component (about double at the bearings, 3 times at the T.G. base, and 4 times at the foundation), the overall displacement peak...
is only 2 to 3 microns greater than the fundamental harmonic (Table IIIB).

Table I. Recorded Acceleration Amplitudes of various Harmonics at 1000 rpm

<table>
<thead>
<tr>
<th>Location</th>
<th>Single Acceleration Amplitude (xg) for frequency components (Hz)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Bearing-1 Horizontal</td>
</tr>
<tr>
<td></td>
<td>24</td>
</tr>
</tbody>
</table>

Table II. Recorded Overall Displacement and Acceleration Amplitudes at 1500 rpm

<table>
<thead>
<tr>
<th>Location</th>
<th>Overall Peak Acceleration (xg)</th>
<th>Overall Peak Displacement (micron)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bearing-1</td>
<td>Horizontal 0.070</td>
<td>Vertical 0.065</td>
</tr>
</tbody>
</table>

Table IIIA. Recorded Acceleration Amplitudes of Various Harmonics at Operating Speed

<table>
<thead>
<tr>
<th>Location</th>
<th>Single Acceleration Amplitude (xg) for Frequency Components (Hz) of 50 100 150 200 250</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bearing-1</td>
<td>Horizontal .550</td>
</tr>
<tr>
<td>T.G. Base</td>
<td>Horizontal .123</td>
</tr>
<tr>
<td>Foundation</td>
<td>Horizontal .061</td>
</tr>
</tbody>
</table>

Table IIIB. Recorded Overall Acceleration and Displacement Amplitudes at 50Hz

<table>
<thead>
<tr>
<th>Location</th>
<th>Overall Peak Acceleration (xg)</th>
<th>50 Hz comp.</th>
<th>Overall Displacement (micron)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bearing-1</td>
<td>Horizontal 1.160</td>
<td>Vertical 0.330</td>
<td>Horizontal 0.272</td>
</tr>
<tr>
<td></td>
<td>Bearing-1 Vertical</td>
<td>T.G. Base Horizontal</td>
<td>Vertical</td>
</tr>
<tr>
<td></td>
<td>55</td>
<td>58</td>
<td>12</td>
</tr>
<tr>
<td></td>
<td>55</td>
<td>57</td>
<td>11</td>
</tr>
</tbody>
</table>

one at pedestal top and one at foundation top. No slip was observed between T.G. base and pedestal top. The load-deflection curve is shown in Fig.1 where maximum deflection at the 7 t load is 79 microns, which is about half of the severest vibration recorded at resonance of 3000 rpm. From Fig.1 the average static stiffness is obtained as 90 t/mm. Accounting for the strain-rate at the operating speed, the dynamic lateral stiffness of the foundation-pedestal system was obtained as 93.6 t/mm or half this value at each end giving $k_{2h} = 46.8$ t/mm.

![Fig.1 - Static Load-deflection Characteristic of Turbogenerator Support.](image)

MATHEMATICAL MODEL AND ROTOR'S CRITICAL SPEEDS

For analytical study a mathematical model of the entire system comprising the foundation, pedestals, spherical seats, bearing oil, rotor and stator (including end-shields and base plate) has been obtained by lumping the stiffness and the masses. The rotor is directly supported by the oil film. A relatively precise model is shown in Fig.2 and a simplified model

TESTS FOR DETERMINATION OF STIFFNESSES

Combined Foundation-Pedestal Stiffness

The lateral stiffness of the foundation-pedestal system was determined experimentally by applying a maximum static load of 7 t at the pedestal top level and observing the deflections by means of four dial gages - two at T.G. base,
of the same system is shown in Fig. 3. In Fig. 2 the combined stiffness of the bearing oil-film and seating is marked by a 2x2 matrix, [K_1].

This matrix includes two cross-coupling terms, k_{12} and k_{21}, between the horizontal and vertical directions of bearing vibrations. These cross-coupling terms have not been considered in the simplified model shown in Fig. 3.

For computing the critical speeds of rotor, the damping in the system has been disregarded and the simplified model has been used.

To determine the influence of the system's stiffnesses on the critical speeds of the rotor, each rotor support has been idealised by a two degree of freedom system as shown in Fig. 4.

The two degree of freedom system was further substituted by an equivalent massless spring, the stiffness, k_e, of which can be shown to be given by the following expression:

$$\begin{align*}
k_e &= k_1 \frac{1 - \frac{m_1 p^2}{k_1}}{1 + \frac{m_2 p^2}{k_2}} \\
&= \frac{m_1 k_1 + m_2 k_2}{m_1 k_2 + m_2 k_1} \frac{1}{1 + \frac{m_2 p^2}{k_2}} \\
&= \frac{m_1 k_1 + m_2 k_2}{m_1 k_2 + m_2 k_1} \frac{1}{1 + \frac{m_2 p^2}{k_2}} \
\end{align*}$$

or 

$$k_e = k_1 \left[ \frac{N}{D} \right]$$

where 

- p = critical speed of rotor in rad/sec.
- m_1, m_2 = lumped masses in the two degree model of Fig. 4.
- k_1, k_2 = discrete stiffness in the two degree model.
- N = numerator within the brackets in eq(1).
- D = denominator within the brackets in eq (1).

The idealisation of equivalent massless spring enables considering the whole system in two parts. Thus the critical speeds of rotor were computed for arbitrary values of its support stiffness, k_e, the results of which are plotted in Fig. 5. For this computation the rotor mass and stiffness was discretised at 30 points and numerical analysis carried out on a computer (Den Hartog, 1956). To find the actual critical speed the value of k_e is obtained from eq (1) and Fig. 5 for various trial values of p. When the values of k_e obtained from eq (1) and Fig. 5 turn out to be equal, the corresponding trial value of p is the actual critical speed of the rotor.

A plot of k_e vs p obtained from eq (1) is shown in Fig. 6. The value of k_e is seen to first reduce from its static value, as given by eq (3), to zero and then reduces further to -\infty, indicating complete 'absorption' of vibration of mass m_1. At this point it has a discontinuity from -\infty to +\infty beyond which it continues to reduce as p increases. The zero values of k_e correspond to the natural frequencies of the two D.O.F system of Fig. 4.

It can be seen that for m_1=m_2=0 or if p=0, eq (1) reduces to the familiar form

$$k_e = k_1 k_2 / [k_1 + k_2]$$

The idealisation of equivalent massless spring enables considering the whole system in two parts. Thus the critical speeds of rotor were computed for arbitrary values of its support stiffness, k_e, the results of which are plotted in Fig. 5. For this computation the rotor mass and stiffness was discretised at 30 points and numerical analysis carried out on a computer (Den Hartog, 1956). To find the actual critical speed the value of k_e is obtained from eq (1) and Fig. 5 for various trial values of p. When the values of k_e obtained from eq (1) and Fig. 5 turn out to be equal, the corresponding trial value of p is the actual critical speed of the rotor.
MASSES IN THE DYNAMIC MODEL

The mass of the machine, excluding the rotor, bearings, and end-shields, is nearly a uniformly distributed load of very high rigidity in both the planes of vibration.

Weight of rotor  
Weight of stator (including base pl)  
Weight of 6 No. pedestals  
Weight of each end-shield + bearing, m1g  
m1  
Effective lumped weight at pedestal top level,  
m2g  
m2 = 164 + 18.6/3 -2x2.1  

The lumped weights in the model of Fig. 3 are obtained as follows:

(a) In the 1st and 3rd modes having equal displacements of the same sign at the two ends of mass m2

(b) In the second mode, if displacements at ends of m2 are of equal magnitude but opposite sign

The lumped mass m2 is thus a function of the mode shape and hence the rotor's critical speed, and has been obtained by equating the kinetic energies of the actual and idealised systems.

STIFFNESSES IN THE SPRING-MASS MODEL

Besides the rotor's own flexural stiffness, which governs its critical speeds, the stiffnesses of the end supports of the rotor are constituted by four elements in series, viz., oil-film, spherical seat of the bearing, pedestals and foundation. These have been denoted by k1', k2', k3 and k4 respectively. The combined bearing oil-film and seating stiffness has been denoted by k1 while the combined pedestal and foundation stiffness by k2. Further subscripts h and v have been added to k1 and k2 to denote the horizontal and vertical directions respectively.

Stiffness of Oil-film

The dynamic characteristics of the bearing, defined by the forces developed in the oil-film may be expressed by the following equations under the assumptions of the linear theory (Smith, 1969).

\[
\begin{align*}
\frac{P_x}{W} &= \frac{2a_{11}x}{c} + \frac{2a_{22}y}{c} + \frac{2b_{11}x}{c} + \frac{2b_{22}y}{c} \\
\frac{P_y}{W} &= \frac{2a_{22}y}{c} + \frac{2a_{22}y}{c} + \frac{2b_{22}y}{c} + \frac{2b_{22}y}{c}
\end{align*}
\]
The duty parameter, where \( N \) speed of machine =

Further, the eccentricity ratio, for medium grade turbine oil having temperatures between 22 to 36 centipoise, (Smith, 1969). The duty parameter, \( S \), is given by

\[
S = \frac{\mu N D L}{W c^2} \quad \ldots \quad (5)
\]

where \( N \) = speed of machine = 3000 rpm
\( D \) = diameter of bearing = 38.1 cm
\( c \) = diametrical clearance = 0.5 mm
\( W \) = reaction of rotor at each bearing = 14.5 t
\( L \) = Length of bearing = 25.4 cm

The duty parameter, \( S \), at \( N = 3000 \) rpm lies between 0.40 and 0.65 for the above range of \( \mu \). Further, the eccentricity ratio, \( n \), is obtained as 0.47 for \( S = 0.40 \) and 0.34 for \( S = 0.65 \) (Smith, 1969)

(i) For \( S = 0.40 \) (or \( n = 0.47 \))
\[
\begin{align*}
    a_{11} &= 2.1, a_{12} = 1.2, a_{21} = -3.4, a_{22} = 2.2 \\
    b_{11} &= 3.2, b_{12} = b_{21} = -2.1, b_{22} = 6.3
\end{align*}
\]

(ii) For \( S = 0.65 \) (or \( n=0.34 \))
\[
\begin{align*}
    a_{11} &= 2.2, a_{12} = 2.3, a_{21} = -3.9, a_{22} = 1.7 \\
    b_{11} &= 4.8, b_{12} = b_{21} = -2.2, b_{22} = 7.5
\end{align*}
\]

The values of the cross-coefficients \( a_{12}, a_{21} \) being of the same order as the direct coefficients \( a_{11}, a_{22} \) the cross-coupling effect of the bearing stiffness is significant. However, in order to examine the effect of bearing stiffness relative to the stiffness of other components (pedestals, foundations and seating), a simplification can be made by omitting the cross-stiffness coefficients, thus making the two vibration directions independent of each other. Also setting the \( b \) coefficients to zero while analyzing the undamped critical speeds, eqs (4) directly yield the bearing oil-film stiffnesses as follows:

\[
\begin{align*}
    k_{bh} &= \frac{P_x}{x} = 2a_{11} W/c = 121.8 \text{ t/mm} \\
    k_{bv} &= \frac{P_y}{y} = 2a_{22} W/c = 116.0 \text{ t/mm}
\end{align*}
\]

adopting average values of \( a_{11} = 2.1 \) and \( a_{22} = 2.0 \) within the range of \( S \) considered

**Stiffness of spherical seating**

(i) Horizontal stiffness:
\[
\text{Diametral clearance, } c' = 0.127 \text{ mm}
\]
\[
\text{Reaction from rotor, } W=29/2 = 14.5 \text{ t}
\]
\[
\text{Horizontal stiffness, } k_{sh} = 2W/c = 228.3 \text{ t/mm}
\]

(ii) Vertical stiffness:
\[
\text{Being very high, } k_{sv} = \infty \quad \text{(assumed)}
\]

**Combined Bearing Oil-film and Seating stiffness**

\[
\begin{align*}
    k_{bh} &= 121.8 \times 228.3 = 279.4 \text{ t/mm} \\
    k_{bv} &= k_{sv} = 116.0 \text{ t/mm}
\end{align*}
\]

**Pedestal stiffness**

Each of the six pedestals have a lateral stiffness of 100 t/mm and vertical stiffness of 2670 t/mm. Also it can be shown that, in the discrete model, three pedestals are effective at each end in the 1st and 3rd modes and only the two end ones in the 2nd mode of vibration, so that

\[
\begin{align*}
    k_{ph} (1) &= k_{ph} (2) = 300 \text{ t/mm} \\
    k_{ph} (2) &= 200 \text{ t/mm} \\
    k_{pv} (1) &= k_{pv} (3) = 8000 \text{ t/mm} \\
    k_{pv} (2) &= 5333 \text{ t/mm}
\end{align*}
\]

**Foundation stiffness**

(i) Horizontal stiffness: Knowing the pedestal stiffness \( k_{ph} = 300 \text{ t/mm} \), and the experimentally determined combined pedestal and foundation stiffness, \( k_{2h} = 46.8 \text{ t/mm} \), the lateral stiffness of the foundation can be determined from

\[
300 \times k_{fh} / (300 + k_{fh}) = 46.8
\]

or \( k_{fh} = 55.5 \text{ t/mm} \)

(ii) Vertical direction: \( k_{fv} = 17500 \text{ t/mm} \)

**Combined Pedestal and Foundation Stiffness**

(i) Horizontal direction: In the 1st and 3rd mode
\[
k_{2h} (1) = k_{2h} (3) = 46.8 \text{ t/mm (determined exply)}
\]

In the second mode
\[
k_{2h} (2) = 46.8 \times (2/3) = 31.2 \text{ t/mm}
\]

(ii) Vertical direction: In the 1st and 3rd modes
\[
k_{2v} (1) = k_{2v} (3) = 8000 \times 17500 = 5490 \text{ t/mm (8000+17500)}
\]

In the second mode
\[
k_{2v} (2) = 5490 \times (2/3) = 3660 \text{ t/mm}
\]

**Critical Speeds and Natural Frequencies**

Besides the critical speeds of the rotor (where in the distribution of stiffness and mass along the rotor is considered), the foundation has its own natural frequencies of vibration. Whereas the flexibility of the rotor is responsible for its critical speeds, it is treated as rigid together with the rest of the machine in determining the natural frequency of vibration of the foundation. The critical speeds of the rotor have been denoted by \( p_r \), \( p_v \), \( p_z \) etc. Only the first natural frequency of the foundation mode is of interest, since the second and higher modes of vibration do not contribute significantly to the response at the pedestal-top level. The foundation mode frequency has been denoted by \( p_f \).
Further subscripts h and v have been added to indicate the horizontal and vertical directions of vibrations.

\[ p_{h} = \frac{(30/\pi) \sqrt{k_{2h}/(m_{1}+m_{2})}}{\text{rpm}} \quad \ldots (6) \]

where \( m_{1} = 29/2 + 2.1 \) is 16.6 t, \( m_{1} = 1.69 \) t/s²/m and \( m_{2} = 8.46 \) t/s²/m. In the horizontal direction, \( k_{2h} = 46.8 \) t/mm so that eq (6) gives \( p_{h} = 648 \) rpm. The resonance observed at 700 rpm being close to 648 rpm, should be the actual value of \( p_{h} \). This indicates a 16.7% increase in the computed value of \( k_{2h} \).

**ANALYSIS OF RESULTS FROM OVER-SPEED TEST-CELL**

The rotor's first critical speed, \( p_{1h} \), was observed to be 1100 rpm. This was associated with the horizontal stiffness of the foundation. From Fig. 5, for \( p = 1100 \) rpm, \( k_{h} = 64.1 \) t/mm. Also \( k_{h}=k_{1h} = 79.4 \) t/mm, \( m_{1} = 0.21 \) and \( m_{2} = 8.46 \) t/s²/m. Substituting in eq (1) gives \( k_{e} = k_{2h} = 558 \) t/mm. Also as \( k_{2} \) tends to infinity, eq (1) reduces to

\[ k_{e} = k_{1} (1-m_{2}^{2}/k_{1}) = k_{1}m_{2}^{2} \quad \ldots (7) \]

Working out \( p \) by trials such that \( k_{e} \) obtained from eq (7) and Fig. 5 is the same gives \( p_{1h} = 1102 \) rpm. Thus a reduction in \( k_{2h} \) from infinity to 558 t/mm reduces \( p_{1h} \) by only 2%, showing very high stiffness of the OSTC foundation.

Similarly, the higher modes critical speeds of rotor for horizontal vibrations have been obtained as \( p_{2h} = 2516 \) and \( p_{3h} = 3513 \) rpm.

In the vertical direction, the foundation stiffness, \( k_{2v} \), is much higher than \( k_{2h} \) and may be taken as infinity. The rotor's first critical speed in vertical direction, \( p_{1v} \), is 1170 rpm. The rotor's critical speeds in the Over-speed Test-cell are summarised in Table IV.

<table>
<thead>
<tr>
<th>Table IV. ROTOR'S CRITICAL SPEEDS IN OSTC</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Masses (t-s²/m)</strong></td>
</tr>
<tr>
<td>---------------------</td>
</tr>
<tr>
<td>( m_{1} = 0.21 )</td>
</tr>
<tr>
<td>( m_{2}(1) = m_{2}(3) )</td>
</tr>
<tr>
<td>( m_{2}(2) = 3.2 )</td>
</tr>
<tr>
<td>( k_{1v} = 116.0 )</td>
</tr>
</tbody>
</table>

*Values of \( k_{e} \) are given within paranthesis.

**ANALYSIS OF RESULTS FROM TEST BED**

**First Critical Speed of Rotor for Vertical Vibrations**

Same value as in the OSTC, i.e., \( p_{1v} = 1170 \) rpm is obtained in the Test-bed. The foundation stiffness \( k_{2v} = 5490 \) t/mm. Whereas a 33% increase in \( k_{v} \) increases \( p_{1v} \) to 1200 rpm, variations in \( m_{1}, m_{2} \) and \( k_{2v} \) have a much smaller influence on \( p_{1v} \) for the actual values of these parameters in the present problem. However the resonance observed at 1200 rpm does not correspond to this mode of vibration.

**Critical Rotor Speeds for Horizontal Vibrations**

The first critical speed, \( p_{1h} \), is computed as 607 rpm. The resonance observed at 3000 rpm was identified to be associated with the rotor's second critical speed. Working backwards, the value of \( m_{2}(2) \) was obtained as 0.20 m² = 0.722 t-s²/m against the assumed value of 0.1667 m².

Since \( p_{2} \) happened to coincide with the machine's operating speed, it became necessary to somehow alter \( p_{2} \). A sensitivity analysis was therefore carried out to determine the extent to which the various parameters affecting \( p_{2} \) influence its value.

**Influence of \( m_{2} \):** Reduction in the effective mass of the generator support, \( m_{2} \), from 0.220 m² to 0.1667 m², i.e., by 32%, results in a small increase in \( p_{2} \) from 3000 to 3065 rpm, i.e., of 2.16%, with increase in \( k_{1} \) of 11.45%. Hence the foundation and pedestal does not significantly affect the second critical speed of rotor, which happens to coincide with the operating speed of the machine.

**Influence of \( k_{1} \):** Increasing the combined foundation and pedestal horizontal stiffness, \( k_{1} \), by 25% increases \( p_{2} \) marginally from 3000 to 3004 rpm, i.e., by 0.13% and the equivalent stiffness of rotor support, \( k_{1} \), by 0.68%. Hence \( k_{2} \) has insignificant influence on \( p_{2} \).

**Influence of \( k_{2} \):** Increasing the combined bearing seat plus oil-film stiffness, \( k_{2} \), by 10% produces an increase in \( p_{2} \) of 78 rpm, i.e., by 2.6% and in \( k_{2} \) of 12.9%. Hence the effect of increasing \( k_{2} \) on \( p_{2} \) is quite significant, and should be utilized in pushing \( p_{2} \) upwards to avoid resonance at the operating speed.

**Third speed of rotor for horizontal vibrations:**

It is difficult to estimate the third mode's effective mass \( m_{3}(3) \) precisely, but it would be within 0.33 m³ and 0.50 m³. Assuming \( m_{3}(3) = 0.40 m^{2} \), the third critical speed is determined as \( p_{3} = 3525 \) rpm. Since it is much above the operating speed of 3000 rpm, the third mode of vibration does not pose problem of excessive vibrations.

The rotor's critical speeds in the Test-bed are summarised in Table V.
**DISCUSSION**

Turbogenerators should be designed to have the critical speeds of their rotors removed away from the operating speeds to avoid high vibration amplitudes that occur near resonances. Since the machine analysed in this study was to be finally mounted on a high tuned foundation, it suited it to have a stiff rotor, which it had (Flint, 1967). Preliminary interaction studies between foundation and rotor have been reported by Geiger (1956) and Flint (1967) and summarised by Major (1980).

In the present study a more involved analysis of a generator which showed resonance right at the operating speed of 3000 rpm during test runs on the Test-bed, reveals inadequate stiffness and instability (half-speed whirl) of the journal bearings. The second critical speed of the shaft, which should have been about 4000 rpm, has been found to come down drastically to 3000 rpm. It is seen that the foundation parameters, both mass and stiffness, have an insignificant effect on this critical speed. Subsequently the bearing was checked and found to have a somewhat loose seating. Tightening of the bearing seats pushed the resonant speed to about 3200 rpm. Use of bearing oil of somewhat higher viscosity with improved cooling arrangement was also suggested to further increase the oil-film stiffness, as it significantly affects the rotor's critical speeds. Also half-speed oil-whirl, that produces disturbances of frequency nearly half the running speed, is present, as evident from the relatively sharp peaks at 1200 rpm in the Test-bed and 2250 rpm in the OSTC, which are nearly twice the rotor's first critical speed of 607 rpm and 1100 rpm in the two situations respectively.

**REFERENCES**


