N92-14347

INVESTIGATION AND CONTROL OF ROTORDYNAMIC INSTABILITY IN TYPICAL LARGE TURBOGENERATORS

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This paper describes the investigation and results of recent studies to solve oil whip in typical large turbogenerators. It includes the following:

(1) Calculations are given of the instability speeds and system damping of rotor-bearing systems. The polynomial-transfer matrix method which was developed by the author during the last few years is used in the calculations.

(2) Vibration measurements and data indicate the stability of these units.

(3) Research reveals that the cause of the instability lies in the three-bole offset bearings.

(4) Work was done to solve these problems, and industry tests were performed on one of these abnormal systems.

INTRODUCTION

Since 1970, many 200 MW-130/355/535 turbogenerator systems have been in operation in China, and these units have gradually become the main fossil fuel power generating units. Unfortunately, oil whip takes place occasionally in some of these systems (about 13 percent of the total number).

This paper describes the investigation and results of recent studies to solve this rotordynamic instability problem.

TYPE OF UNITS AND BEARINGS

Figure 1 shows the rotor-bearing system investigated. It consists primarily of HP, IP, and LP sections and a generator. The turbomachinery has a total weight of 50,400 kgf and an overall length of 18,823 mm. The generator rotor weight is about 42,000 kgf and its length is 8000 mm.

The rotors are supported by seven three-pocket bearings (TPB) which possess three axial grooves. However, each land contains a spiral pocket, creating the convergent film shape shown in Fig. 2. The principal parameters of these TPB are given in Table 1.

CALCULATION ANALYSIS OF THE INSTABILITY

Calculation of the Stiffness and Damping Coefficients of the Bearing

Four stiffness coefficients $K_{xx}, K_{xy}, K_{yx}, K_{yy}$ and four damping coefficients $C_{xx}, C_{xy}, C_{yx}, C_{yy}$ representing the dynamic character of the bearing were obtained by solving the Reynold's equation (1) with the finite difference and finite element methods.

$$\frac{\partial}{\partial x}\left(\frac{1}{Kx} \cdot \frac{h^3}{12\mu} \cdot \frac{\partial p}{\partial x}\right) + \frac{\partial}{\partial z}\left(\frac{1}{Kz} \cdot \frac{h^3}{12\mu} \cdot \frac{\partial p}{\partial z}\right) = \frac{u}{2} - \frac{\partial h}{\partial x} + \frac{\partial h}{\partial t} \tag{1}$$

where

x circumferential coordinate in the direction of rotation

z axial coordinate

μ dynamic viscosity of oil

p pressure

u tangential velocity of the journal surface

K_x, K_z turbulence coefficients

Considering that the actual flow state is between laminar and turbulent flow, and to compare the results, the calculation was carried out on the basis of three flow states:

(1) Laminar flow
For this condition, $K_x = K_z = 1$

(2) Turbulent flow
According to Ng-Pan's law, $K_x = 1 + 0.00113R_h^{0.9}$
and $K_z = 1 + 0.00036R_h^{0.96}$

(3) Laminar-turbulent mixed state flow

In the calculation of this state, first the Reynold's numbers R_h at each point on the net of the bearing surface were evaluated to determine the parameters of each point, then correct turbulence coefficients were selected.

Calculation of the Instability of the Rotor-Bearing System

In the calculation process, the following two methods are employed:

$\underline{K_{eq} - \gamma_{st}}$ method. - Usually, the equivalent stiffness K_{eq} of the oil film and the whirl rate γ_{st} (where $\gamma_{st} = \omega/\Omega_{st}$, ω, whirl frequency; Ω_{st}, the threshold speed) can be used to analyze instability. K_{eq} and γ_{st} are, respectively,

$$K_{eq} = \frac{K_{xx}C_{xx} + K_{yy}C_{yy} - K_{xy}C_{yx} - K_{yx}C_{xy}}{C_{xx} + C_{yy}} \tag{2}$$

$$\gamma_{st}^2 = \frac{(K_{eq} - K_{xx})(K_{eq} - K_{yy}) - K_{xy}K_{yx}}{C_{xx}C_{yy} - C_{xy}C_{yx}} \tag{3}$$

They are the function of the bearing load coefficient ζ

$$\zeta = \frac{W\psi^2}{\mu u L} \tag{4}$$

where

W load acting on bearing

ψ clearance ratio

L length of bearing

The threshold speed of the elastic rotor with a single disk is therefore

$$\Omega_{st} = \frac{\omega_o^2 m}{2K_{eq}\frac{\mu L}{\phi^3}} + \omega_o \sqrt{\left(\frac{m\omega_o}{2K_{eq}\frac{\mu L}{\phi^3}}\right)^2 + \frac{1}{\gamma_{st}^2}} \tag{5}$$

where ω_o is the natural frequency of the shaft supported by two stiff pedestals.

Transfer Matrix-Polynomial Method (TMPM) (Ref. 1)

This method uses the block matrices for transferring the sequence of calculations along the rotor. The characteristic polynomial can be directly obtained, then all complex eigenvalues are solved with the Bairstow-Newton approach. In the eigenroot $S = \lambda + i\omega$, λ is the system damping. Meanwhile, according to λ, the logarithmic decrement δ corresponding to various speeds can be obtained. It is clear that the stability state of the systems is known immediately on the basis of the value δ or λ.

A program based on the previously mentioned transfer matrix polynomial algorithm was developed by the author during the last few years. The first half of this program employs new recurrence expressions, equations (6) and (7):

$$L_i = U_{i-1}L_{i-1} + W_iM_i \tag{6}$$

$$M_i = U_{i-1}M_{i-1} + R_{i-1}L_{i-1} \tag{7}$$

The second half of the program applies the polynomial algorithm. The size of the new transfer matrices is one-fourth of those used in the Murphy and Vance's method (Ref. 2); therefore, the CPU time and the storage required are greatly reduced.

NUMERICAL RESULTS (Ref. 3)

The results of calculating the threshold speeds Ω_{st} of 200 MW units by the $K_{eq} - \gamma_{st}$ method for the laminar state are presented in Table 2. As shown in this table, the Ω_{st} of bearings 6 and 7 is the lowest, only 3557.6 r/min, and it is merely 1.19 times the working speed 3000 r/min. The next lowest value is the Ω_{st} of bearing 3, 3911.9 r/min.

The Ω_{st} of bearings 6 and 7, as determined by TMPM for the same state, are also listed in Table 2. It should be noted that the results from these two algorithms are almost identical.

A diagram of the damped natural frequencies for the laminar state of the oil film resulting from TMPM is shown in Fig. 3. With the rotor speed as the abscissa and the damped natural frequency as the ordinate, a number of the modes are shown in this figure. The parameter values on the curves give the logarithmic decrement δ.

In Fig. 3, it is noted that the δ of the first forward whirl changed from a positive value to a negative value when the rotor speed reached 3601.5 r/min, then the generator rotor began to become unstable. This frequency is rather lower than the first natural frequency of the generator supported by the stiff pedestal and is consistent with the measured value of the first critical speed.

Again, it should be noted in Fig. 3 that the value of δ for the generator rotor is only 0.08 at the working speed of 3000 r/min and is much less than the one recommended by current standards (Ref. 4).

Table 3 shows the variation of δ with the bearing dimensionless load $\bar{P}$ and the oil inlet temperature T_{in} for the laminar-turbulent mixed state. At nominal working parameters (i.e., at $\bar{P} = 1$ and $T_{in} = 42$ °C), Ω_{st} is 3004.3 r/min. When T_{in} increases to 48 °C, Ω_{st} can also increase to 4593.8 r/min. The reduction of the bearing load can also sharply decrease Ω_{st}. A more common cause of bearing load variation is the differential thermal growth of the bearing pedestals. Besides showing that the stability of the 200 MW generator units is inferior at the nominal speed, an analysis of the calculated results of Table 3 shows that the stability can easily deteriorate with $\bar{P}$ and T_{in}.

The damped natural frequencies of the HP, IP, and LP sections of the turbomachinery rotor-bearing system are shown in Fig. 4, whereas Table 4 presents the δ values of the second forward whirl of these three rotors at 2700, 3000, and 3300 r/min, respectively. From Fig. 4, it can be seen that the stability of all of the rotors, except the LP rotor, are poor.

To thoroughly investigate this kind of turbogenerator and to better examine the calculation program from an engineering viewpoint, a comparative calculation was made between these 200 MW units and a 250 MW unit known for its good stability. The numerical results for this 250 MW unit are listed in

Table 4. It further demonstrates that the 200 MW units are much less stable than the 250 MW units (Ref. 5).

MEASUREMENT AND FIELD DIAGNOSIS TEST ON THE INSTABILITY VIBRATION OF 200 MW TURBOGENERATORS

Because of the instability of the 200 MW units, a series of measurements and tests in the field were performed to diagnose and determine the cause of violent vibrations that occurred in some of these units, severely affecting normal work as well as safety.

The results of the measurements and tests follow:

(1) The large vibrations that sometimes occurred in some 200 MW units was oil whip. In all of these cases, it is clear that the whip originated in bearings 6 and 7, and then expanded to the turbomachinery causing the whole rotor-bearing system to vibrate violently.

(2) Meanwhile, it was found that the dominant frequency of the generator vibration was 18-19 Hz, which is approximately the first natural frequency of this rotor under actual operating conditions. The maximum vibration amplitude of these two bearings could reach 0.6-0.8 mm (p-p) during whip.

(3) The oil temperature in the bearings is of vital importance to stability during the operation. Oil whip can be temporarily avoided by increasing the oil inlet temperature. Consequently, before whip was completely eliminated in these defective units, some defective units could be operated reluctantly by raising the oil inlet temperature to 48-50 °C for several months, even a year.

(4) Spectrum analysis on some of the 200 MW units with higher oil inlet temperatures indicated that, except for units with whip, a subsynchronous component whose frequency ranged from 18-19 Hz at bearings 6 and 7 or half running speed at bearing 3 exist in most units, though these subsynchronous vibration values are small and there was not typical instability, i.e., whip or obvious oil whirl. Figure 5 shows the spectrum of shaft vibration of bearing 6 at 3000 r/min in this unit (Ref. 6).

IMPROVEMENT OF THE UNITS ON SITE

So far, all of the calculations and measurements have fully demonstrated the poor stability of the 200 MW turbogenerators. Consequently, three main measures were taken to improve these systems:

(1) Modify the geometric parameters of the TPB, e.g., decrease the effective length of the bearing.

(2) Re-adjust the vertical height of the bearings and make the center of bearing 6 higher than bearing 5 by 0.05-0.1 mm in the stopped condition to compensate for the rise of the center of bearing 5 during operation.

(3) Use bearings with good stability instead of the TPB.

For some defective units, actions (1) and (2) have been effective, but for a few units, the situation did not take a favorable turn.

It is obvious that the former bearings at least bearings 6 and 7, should be replaced by high-stability ones.

The new design adopted the elliptic bearings shown in Fig. 5. A detailed calculation of the logarithmic decrement δ shows that this improved rotor-bearing system will be much better than for previous systems of stable TPB units. From Table 3, it can be seen that at working speed, the new value is about three times the previous value of 0.08. It is also noted that δ is still 0.213 at the maximum calculation speed 4000 r/min, and this means that the threshold speed of the new rotor-bearing system has gone beyond 4000 r/min.

TEST AND RESULTS OF IMPROVING A UNIT BY REPLACING THE BEARINGS

Bearings were replaced in a unit with severe whip at Xu Zhou Power Plant to solve instability and to test the elliptic bearing on site.

Since they were put into operation at the end of 1985, this unmodified system has experienced whip 39 times. Lastly, the oil inlet temperature has to be maintained above 48 °C to avoid the emergence of whip. Figure 6 gives a typical whip spectrum of this unit at an operation speed of 3000 r/min and an oil inlet temperature of 48 °C (Ref. 7). After failures in modifying the design parameters of the TPB and in readjusting the centers of the bearings, it was finally decided to replace bearings 3 and 7 with the elliptic ones.

The test of the unit with the new bearings was to verify the stability and to determine general vibration characteristics. The test was carried out under the following two special severe working conditions:

(1) The oil inlet temperature was decreased slowly from 48 to 35 °C, which is below the normal value by 5 °C.

(2) The speed of the unit rose to 3300 r/min, then the trip occurred.

During the whole test period, the unit operated smoothly under several working conditions from startup to full power 200 MW, and no whip was encountered.

From Figs. 7 and 8 (Ref. 8), it can be seen that the predominant frequency coincided with the operating speed of the turbine. Between the speeds of 2700 and 3300 r/min, the maximum amplitude of the 18.75 Hz component was only 0.7 μm and is considerably smaller than the amplitude shown in Fig. 5. It is, therefore, of interest to note that whip has no longer occurred and that instability is considered to be completely solved. In addition, the test proves the following properties of this new bearing rotor system:

- From the Bode figure, it is seen that there is no resonance peak at bearings 6 and 7 in connection with the second critical speed of the generator rotor from 2600 to 3250 r/min.

- The unbalance response of the rotors mounted in the new bearings is normal and its influence coefficients are approximately the same as those in the previous TPB units. The maximum amplitude of each bearing is less than 0.05 mm when the rotor passes various critical speeds.

- Other operating parameters - for example power loss, temperature of bearing surface, and amount of oil flow - are normal, too.

CONCLUSIONS

The TMPM used to calculate the 200 MW units is a valuable analytical tool. The resulting logarithmic decrement can be effectively employed to evaluate the stability of large turbogenerators. The CPU time of the corresponding program is faster than other ones.

The detailed calculation and measurement revealed that the stability of the 200 MW units is inferior and is not satisfactory for operation. In addition, investigation indicated that the main reason for poor performance is the TPB which has poor stability properties with middle to heavy bearing loads.

Adoption of elliptic bearings for 200 MW units is a valid measure for eliminating whip in these units. Meanwhile, elliptic bearings do not produce side effects on other general properties.

A thorough investigation is needed to choose the optimum parameters for elliptic bearings under various working conditions.

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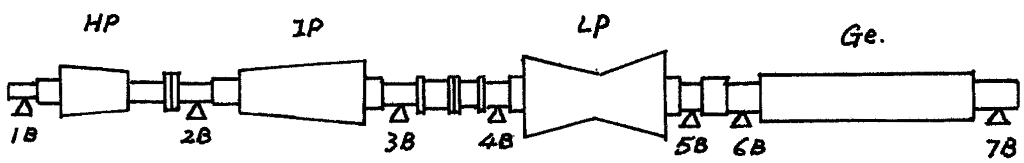


Fig.1 Schematic Diagram of Rotor-Bearing System of 200 MW Turbogenerator

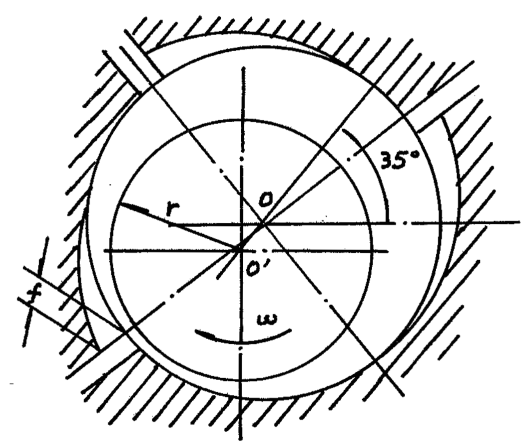


Fig.2 Schematic of the Three-Pocket Bearing

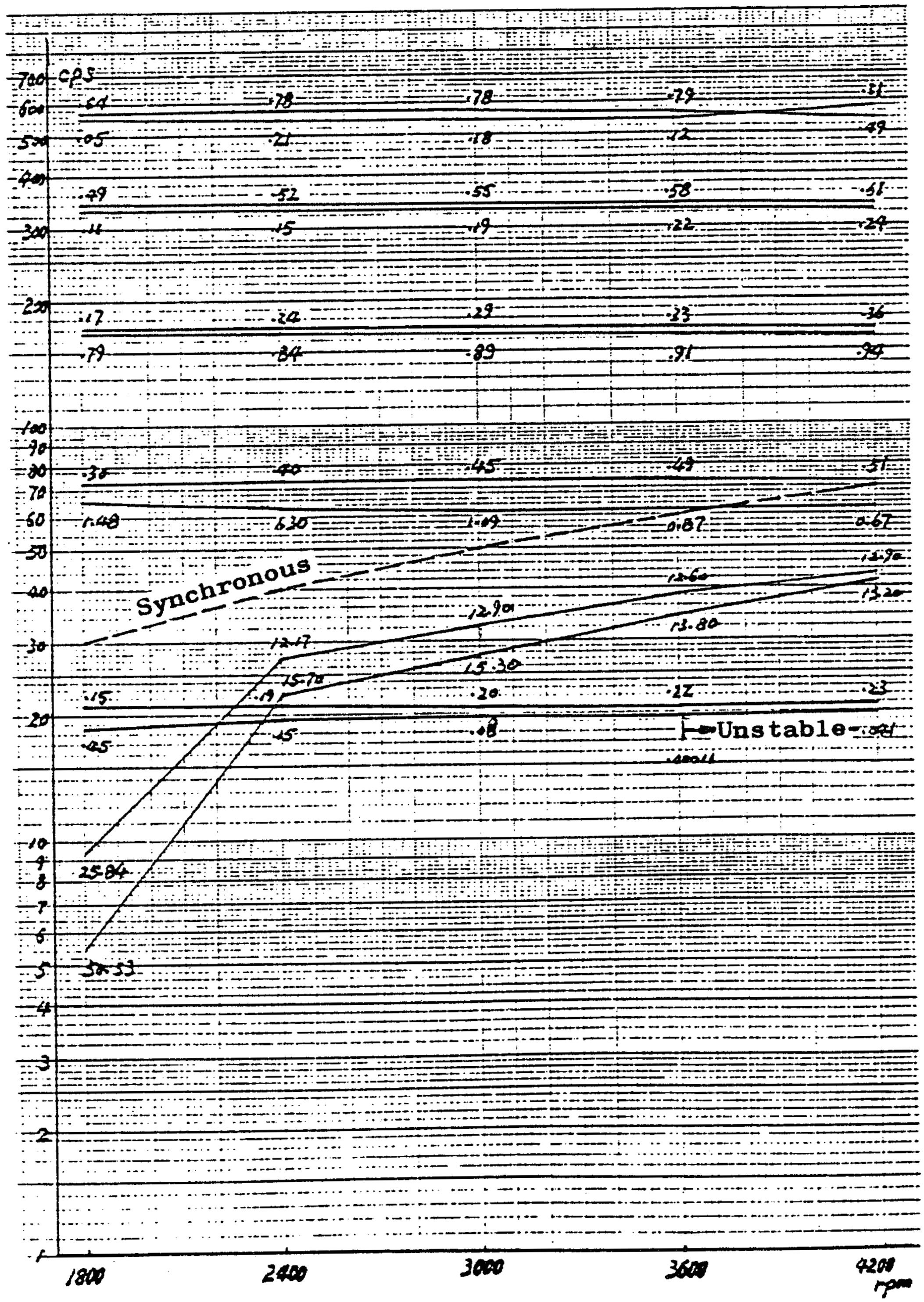
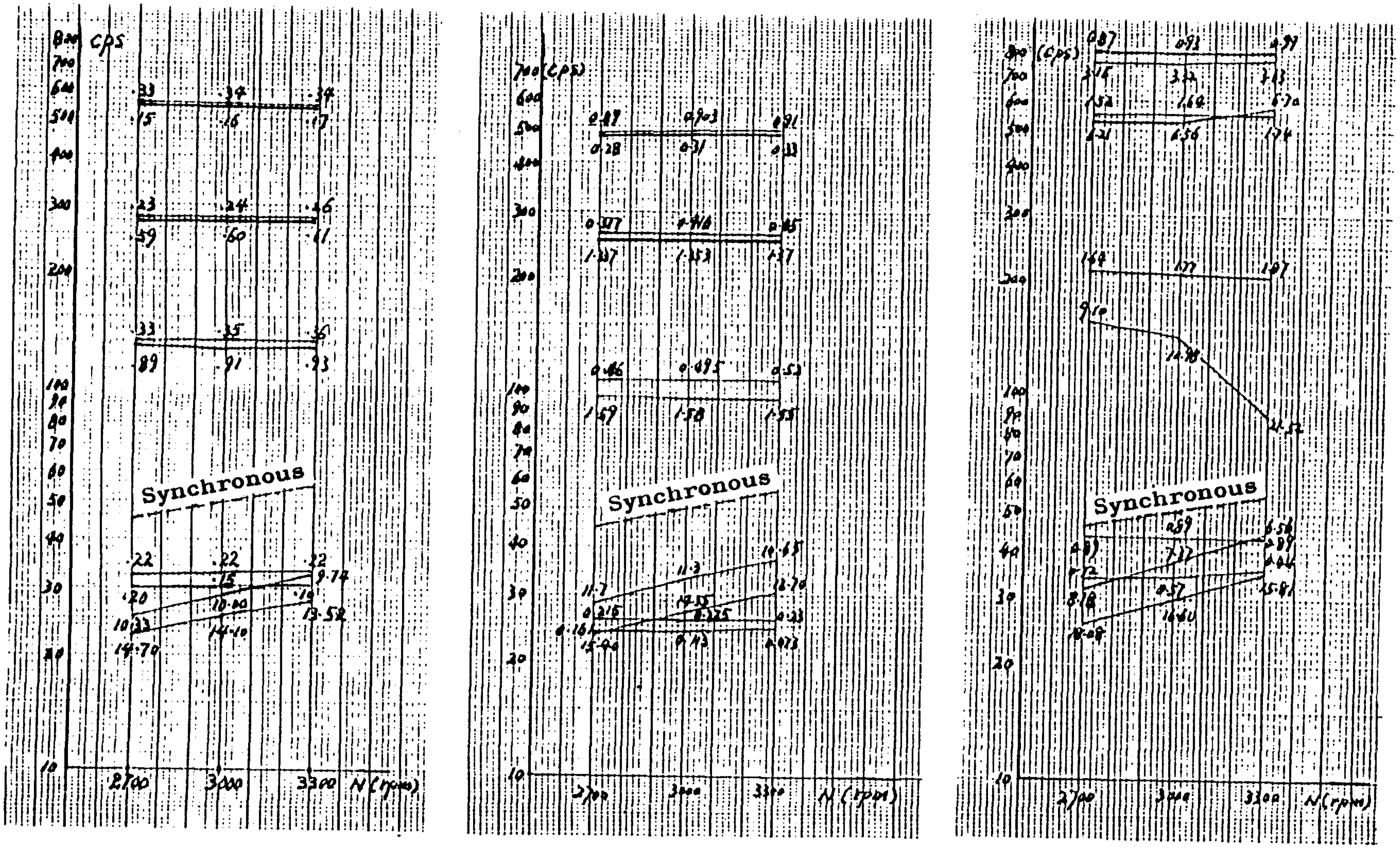


Fig.3 Damped natural frequencies of 200MW generator in three-pocket bearings



(a) HP rotor (b) IP rotor (c) LP rotor

Fig.4 Damped natural frequencies of 200MW turbomachinery

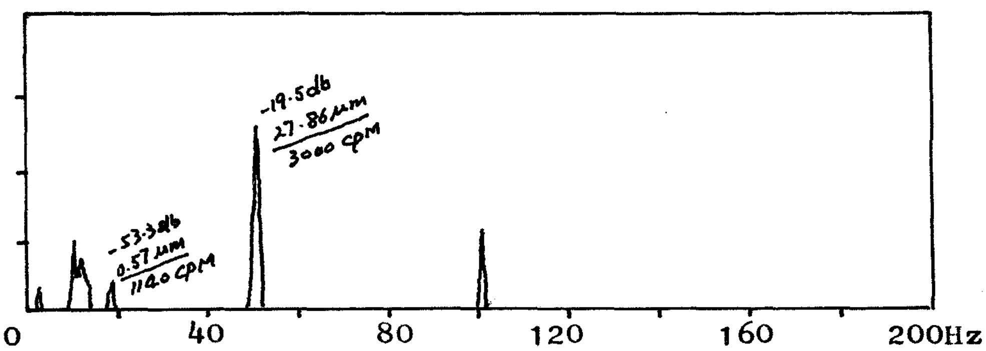


Fig.5 Frequency Spectrum of horizontal vibration of No.6 bearing of a 200MW unit at full load, 3000r/min, Dec. 12,1985

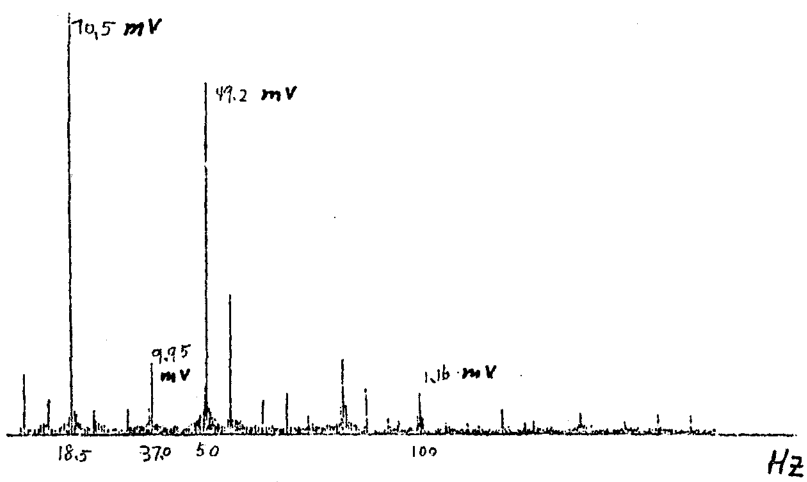


Fig.6 Typical spectrum during the whip at No.6 bearing of No.6 unit of Xu Zhou Power Plant before the replacing TPB (7)

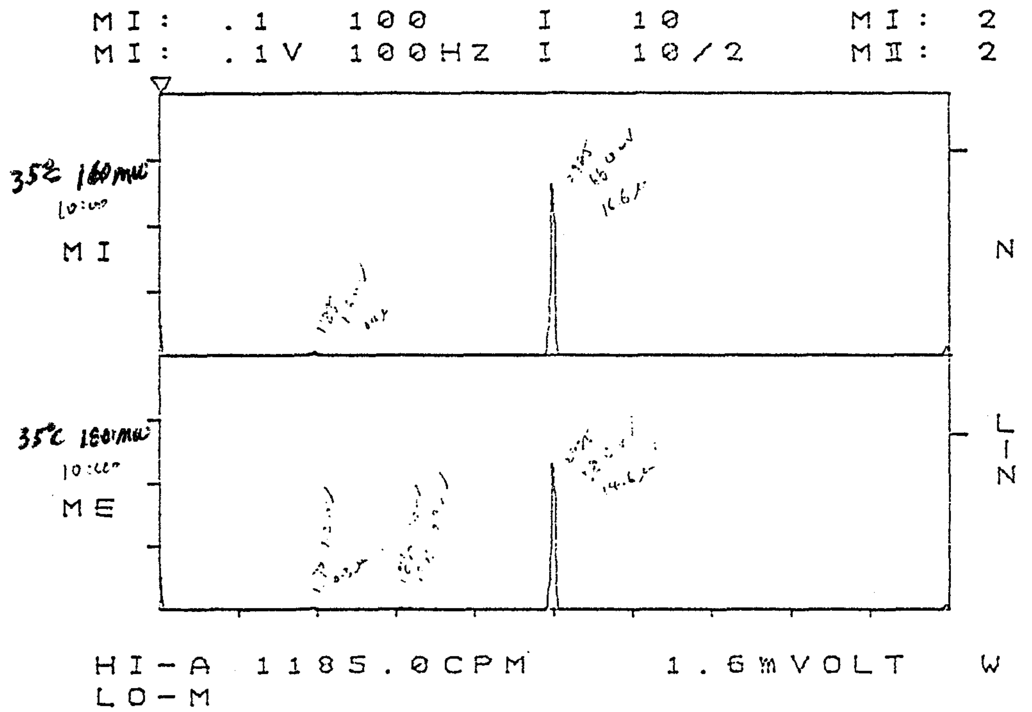


Fig.7 Frequency spectrum of vertical vibration of No.6 bearing of No.6 unit in new elliptic bearings of Xu Zhou Power Plant, power output: 160 and 180MW, oil inlet temperature: 35°C

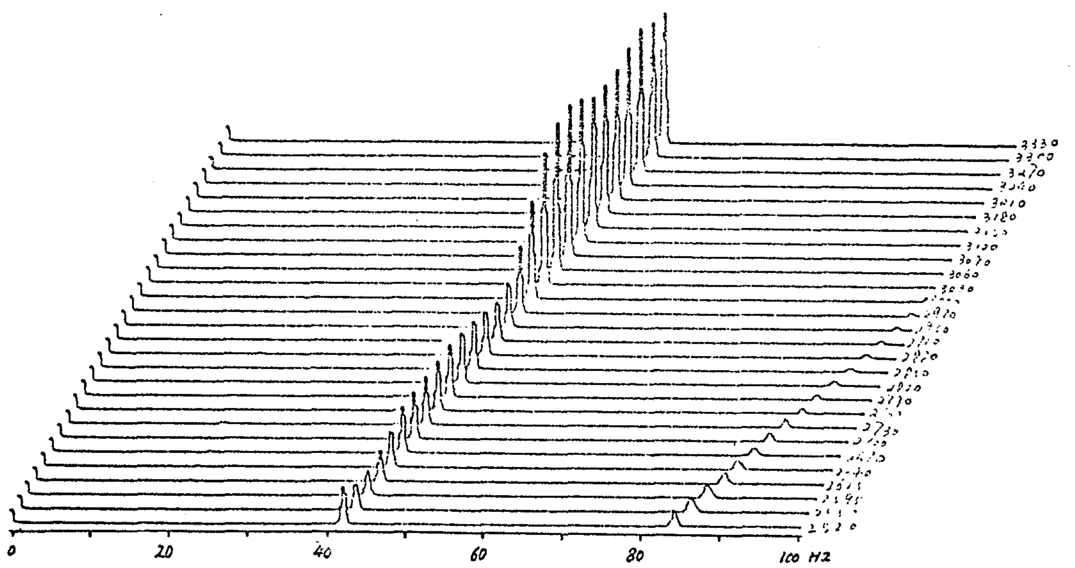


Fig.8 Spectrum cascade of vertical vibration of No.6 bearing of No.6 unit in new elliptic bearing of Xu Zhou Power Plant, speed from 2520 r/min to 3330 r/min, Lin..

Tab. 1 TP Bearing data of 200MW turbogenerators

No. of bearing		1	2	3	4	5	6 and 7
Diameter D(mm)		250	300	360	360	360	420
Ratio L/D		0.8	0.8	0.75	0.75	0.75	0.7619
Load W (Kgf)		3900	13900	12237	10040	8923	21400
Specific pressure P_m	P_a	7.652E5	1.893E6	1.235E6	1.013E6	9.006E5	1.565E6
	Kgf/Cm	7.8	19.3	12.59	10.33	9.18	15.95

Table 2 Calculation results of threshold speeds Ω_{st} of units (r/min)

Calculation condition			No. of bearings					
			1	2	3	4	5	6,7
200MW unit	$K_{eq}-\gamma_{st}$, Single disc Laminary		4220.8	4487.8	3911.9	4353.2	4206.0	3557.6
	TMP, Multidisc, Lami.							3601.5
	TMP, Multidisc Laminary-turbulence	T_{in}=42°C						3004.3
		T_{in}=48°C						4593.8
250MW unit	TMP, Multidisc Turbulence							5527.0

Table 3 The calculation results of the threshold speeds of TPB units versus dimemsionless load of bearing and oil inlet temperature

Dimensionless load of bearing		100%		90%		80%		70%	
T_{in}	T_m*	Ω_{st}	ε_o**	Ω_{st}	ε_o	Ω_{st}	ε_o	Ω_{st}	ε_o
48°C	60°C	4593.8	0.571	3620.7	0.543	3018.3	0.515	2651.4	0.478
42°C	55°C	3004.3	0.536	2458.9	0.510	2228.3	0.479	2171.3	0.443

* T_m: the average temperature in oil film of bearing

** ε_o: eccentricity ratio

Table 4 The logarithmic decrement δ of 200MW unit with TPB and 250MW unit with elliptic bearing

	Speed (r/min)	HP	IP	LP	Speed (r/min)	Ge.
200MW unit	2700	0.2001	0.1613	0.7217	2400	0.15
250MW unit		0.708		0.465		0.47
200MW unit	3000	0.1477	0.1131	0.5717	3000	0.08
250MW unit		0.679		0.354		0.25
200MW unit	3300	0.1041	0.0729	0.4411	3600	0.00011
250MW unit		0.654		0.264		0.15

Table 5 The elliptic bearing parameters

No. of bearing	3,4 and 5	6 and 7
Diameter D(mm)	360	420
Assembled clearance	0.0015	0.0015
Elliptic ratio	0.55	0.55
Length/D (L/D)	0.75	0.7619

Table 6 The logarithmic decrement δ of 200MW unit with new elliptic bearing of generator rotor

Speed (r/min)	δ	whirl frequency (Hz)
3000	0.217	16.00
3200	0.202	16.11
3400	0.218	16.34
3600	0.128	16.44
3800	0.287	15.62
4000	0.213	16.01