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## INSTABILITY OF MULTISTAGE COMPRESSOR K1501

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The K1501 compressor, driven by a steam turbine, is used to transport synthetic gas in fertilizer plants of 1000 tons daily production. The turbo-compressor set, which had been in operation since 1982, vibrated rather intensely, and its maximum load was only about 95 percent of the normal value. Damaging vibration to pads and gas-sealing labyrinths occurred three times from 1982 to 1983 and resulted in considerable economic loss. From the characteristics of the vibration, we suspected its cause to be rotor instability due to labyrinth-seal excitation. But, for lack of experience, the problem was not addressed for two years. Finally, we determined that the instability was indeed produced by labyrinth-seal excitation and corrected this problem by injecting gas into the middle-diaphragm labyrinths.

This paper primarily discusses the failure and the remedy described above.

## INTRODUCTION

In recent years, China imported many large chemical plants that have various kinds of compressors. Unacceptable vibration has occurred in some running compressors. The most serious occurrence was with the K1501-K1501 set that transported synthetic gas for a fertilizer-producing process (at 1000 tons a day). Figure 1 shows the K1501-1501 set, which consists of one steam turbine (including two cylinders) and three compressors. Its maximum power is 19 786 kW and its maximum continuous speed is 11 230 rpm. It has a traditional concrete construction pad.

Intensive vibration originated from the low-pressure compressor, K1501LP. Figure 2 shows the longitudinal section of its rotor. Table I shows the parameters of the tilting pad of both bearings. Figure 3 shows the gas labyrinth construction of the middle diaphragm. Figure 4 provides a critical speed map and mode shapes for the rotor.

Vibration occurred suddenly each time. The first accident took place on May 7, 1982, when the speed was 10 800 rpm. The following characteristics were observed:

- (1) The vibration of the shaft exceeded 80  $\mu\text{m}$  suddenly.
- (2) The vibration accompanied intensive sound radiation (over 95 dB).
- (3) Sealing oil oozed out of the gas exit tube.
- (4) Intense floor vibration followed.

When the cylinders were opened and examined, the following damages were observed:

- (1) The pads of radial bearing 074 were obviously deformed because of intensive journal vibration.

(2) The gas labyrinth of the middle diaphragm and the ends of the cylinder were worn out. Some wear scars remained on the surface of the shaft.

When only the damaged parts were replaced, the vibration and damage reoccurred when the speed reached 10 820 rpm.

There are three identical sets of this kind in our nation - in Nanking, Anqing, and Guangzhou. This type of failure occurred in all three compressors. The most serious failure was in Nanking. Because the failures were clearly not accidental, the Chemical Engineering Ministry began and directly guided experimental research in Nanking. (All the phenomena described above occurred in Nanking.) The following is a description of the whole test, which was divided into three steps.

#### FIRST OF STEP OF TEST (May 1982 - January 1983)

At first we did what we could to minimize downtime and production losses. The origin of the vibration was rotor instability as will be detailed. Although the labyrinth-seal excitation was considered to be the cause of the instability, the possibility of oil whip had not been dismissed at that time. Therefore the following measures were adopted:

- (1) Six teeth in the middle of the diaphragm labyrinth were removed.
- (2) The ratio of bearing width to diameter was decreased from 0.40 to 0.39 mm.
- (3) Additional displacement transducers were installed on bearings 074 and 075 (fig. 1) to improve vibration monitoring.

Because none of these measures eliminated the rotor instability, a large quantity of measurements and analyses were then made. This work was performed mainly by Zhejiang University. Because vibration was monitored closely during the experiments, no serious damage occurred.

Figure 5 shows frequency spectra of the shaft vibration in direction  $x$  on bearings 074 and 075 at a speed of 10 220 rpm as measured by a transducer on November 19, 1982. From figure 5, the rotating speed was 170.3 Hz, and the frequency of fractional frequency whirl (FFW) was 80 Hz. The frequency ratio  $n_f$  equalled 0.47. The vibration amplitudes of the components in this figure are almost equal. Figure 6 shows a frequency spectra of the shaft vibration in direction  $y$  on bearing 074 and in direction  $x$  on bearing 075 at 9460 rpm on November 19, 1982. Again, the vibration amplitudes were almost equal. The frequency of FFW was 77 Hz (measured), the rotating frequency was 157.7 Hz, and the frequency ratio  $n_f$  equalled 0.488. So obviously the FFW appeared at 9460 rpm.

Figure 7, which gives a speed-spectrum map from an experimental recording, indicates that the FFW appeared just at 8000 rpm and that  $n_f$  equalled 0.5. Amplitude increased rapidly with increasing speed, but frequency dropped slightly.

Figure 8 shows an evolution of the shaft center orbit measured on bearing 074 at 9200 to 9480 rpm on December 18, 1982. It was photographed from the cathode oscilloscope.

All of the test results fully demonstrated that the vibration came from the instability of the rotor, but the nature of the instability was not determined. So a power spectrum referring to the vibration of bearing 074 and to the pulse pressure of the gas outlet was made. The transfer function and the coherent function were also calculated. However, the nature of the instability was still not evident.

The following additional changes were performed in this step:

(1) The inlet temperature of the lubricant oil was changed from 40 to 45 °C.

(2) The opening level of the compressor-protecting valve was changed from 40 to 100 percent.

(3) The inlet gas temperatures and pressures of the compressor were changed.

Yet no consistent, useful results were obtained.

At the same time, some temporary measures were taken during the tests:

(1) A middle vertical slipping pin was added on the body of the compressor to prevent motion caused by expansion of the cylinder.

(2) The supports were packed with insulation to prevent the cold current heat losses from influencing them.

Attention was also paid to the pads and oil-sealing slipping of the cylinder, but no effects were obtained.

#### SECOND STEP OF TEST (February - October 1983)

To clarify the nature of the instability, we decided to observe the dynamic properties of the rotor K1501LP on a balancing machine. This test was performed by Hangzhou Steam Turbine Works (ref. 1). The test included the following:

- (1) High-speed balancing of the rotor
- (2) Observation of the critical speed and its insensitivity to other parameters
- (3) Increasing the number of the oil-inlet holes in the bearings
- (4) Increasing the clearance in the bearings

Figure 9 shows the Nyquist tracks before and after the balancing correction. Before the correction the severity of the bearing vibration increased quickly when the speed exceeded 9000 rpm. For the sake of safety the pedestal stiffness was increased from 820 to 2250 N/ $\mu$ m. When the speed reached 11 230 rpm, the two bearings were vibrating at 6.5 and 2.2 mm/sec, respectively. This result indicated poor balancing. We found that the mixed vibration of the second and third modes appeared before the second critical speed was reached. Using the method of modal separation, we corrected the unbalance of the rotor. The bearings were vibrating at only 0.3 and 0.35 mm/sec, respectively, when the pedestal stiffness was restored to 820 N/ $\mu$ m. The first

critical speed was 4000 rpm and the second was 10 700 rpm; these speeds are in agreement with the results of our calculation. The results of our experiment on insensitivity indicate that the rotor is sensitive to unbalance at 11 230 rpm. (This does not conform to American Petroleum Institute (API) standards.)

The Nyquist tracks of the rotor with one and with five oil-inlet holes (one hole for each pad) in the bearing are shown in figure 10. The dotted lines in the figure show a change of rotor vibration for one oil-inlet hole. However, oil whip did not take place. The oil clearance was increased from 0.15 to 0.20 mm, but the state of the shaft vibration did not change. It was impossible to test the labyrinth-seal excitation, for the rotor was driven in a vacuum chamber.

In brief, high-speed balancing showed clearly that the instability of the rotor did not come from the oil whip.

### THIRD STEP OF TEST (November 1983)

To eliminate the vibration failure rapidly, we cooperated with Mitsubishi in completing a series of tests on November 5-8, 1983. The test speeds were to 10 222 rpm on the first day, and to 10 550 rpm on the third day. All the tests were within specifications. The test procedure for the fourth day is shown in figure 11. After reaching 10 750 rpm and running for an hour, the shaft vibrated intensely.

Figure 12 shows a real-time analysis of the shaft vibration during stable operation. Figure 13 shows the same analysis during a period of instability. Figure 14 gives a three-dimensional spectrum analysis of the shaft vibration in direction x on bearing 075. Figure 15 shows an orbit of the shaft center during instability.

The tests just described make clear the inevitability of rotor instability. But comparisons between figures 7 and 14 show that the latter tests did not produce considerable FFW before instability appeared, and the amount was less than that of the operational speed-frequency map by 25 dB (fig. 12). Just before the rotor became unstable (fig. 14), the frequency of the FFW was 74 Hz/sec (which is the same as in fig. 12). This frequency suddenly rose to 80 Hz after 5 sec (which is in agreement with fig. 13). The frequency ratio  $n_f$  equalled 0.4465.

### MEASURES TAKEN

Although much important data on the dynamic properties of the rotor were acquired through the three steps of the tests, the nature of this instability has not been directly determined. According to the information and inference, however, the origin of the instability is labyrinth-seal excitation. The compressor manufacturer also provided some suggestions for improvements.

Finally, a decision was made that gas should be injected into the labyrinths of the middle diaphragm. The new construction is shown in figure 16. After this measure was taken, the instability was eliminated. Figure 17 shows a frequency spectrum of the shaft vibration on bearing 074 at 10 690 rpm before corrective measures were taken, and figure 18 shows this spectrum after corrective measures were taken. It can be seen that the FFW was eliminated.

The same results were obtained in Anqing immediately after the Nanking tests.

### CONCLUDING REMARKS

The following are some of the author's viewpoints included only for reference:

1. Instability of compressor K1501 was excited by the labyrinth seal without regard to bearings, oil seal, and other factors.
2. The fractional frequency whirl (FFW) resulting from the labyrinth-seal excitation was a positive precession rather than negative precession as usually observed (ref. 2).
3. Comparisons between figure 7 and 14 show that the evidences of instability appearing after high-speed balancing of the rotor differed from those before balancing. Evidently the rotating speed at which the visual FFW occurred increased.
4. The middle-diaphragm labyrinth with gas injection as shown in figure 16 effectively eliminated the excitation existing in it, so that the method presented by Kirk (ref. 3) was again proved to be reliable.

### REFERENCES

1. Zhou, Ren-mu: High-Speed Dynamic Balancing of Multistage Rotor of Compressor. Jour. of Applied Mechanics, vol. 1, no. 1, 1984 (Chinese).
2. Traupel, W.: Thermische Turbomaschinen, zweiter Band, 3. Auflage, S. 475, 1982.
3. Kirk, R.G.; et al.: Analysis and Identification of Subsynchronous Vibration for a High Pressure Parallel Flow Centrifugal Compressor. Rotordynamic Instability Problems in High-Performance Turbomachinery, NASA CP-2133, 1980, pp. 45-63.

TABLE I. - PARAMETERS OF TILTING PAD BEARING

	Bearing	
	074	075
Diameter of journal, mm	101.6	114.3
Width of bearing, mm	41	47
Specific load, N/cm <sup>2</sup>	59.3	48.7
Clearance, mm	0.11 to 0.14	0.14 to 0.17
Number of oil-inlet holes	1	1

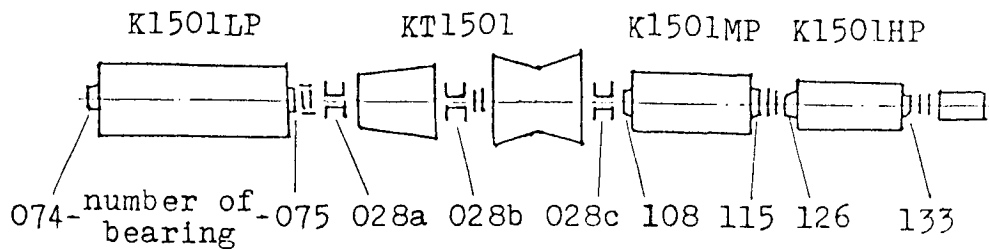


Figure 1. - Steam turbine compressor set of KT150-K1501.

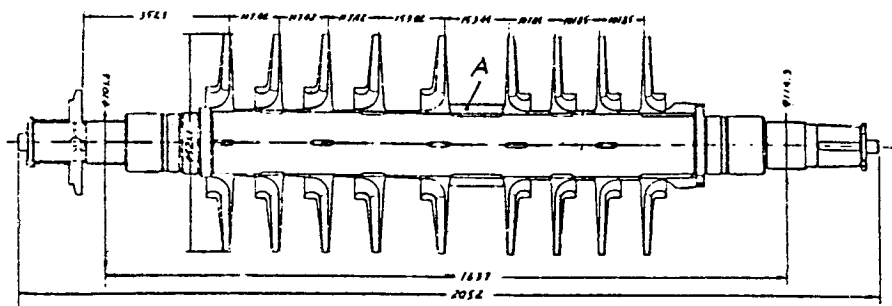


Figure 2. - Rotor of K1501LP.

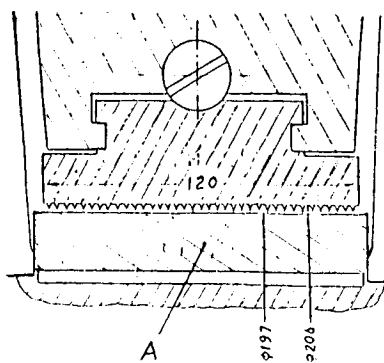


Figure 3. - Labyrinth construction.

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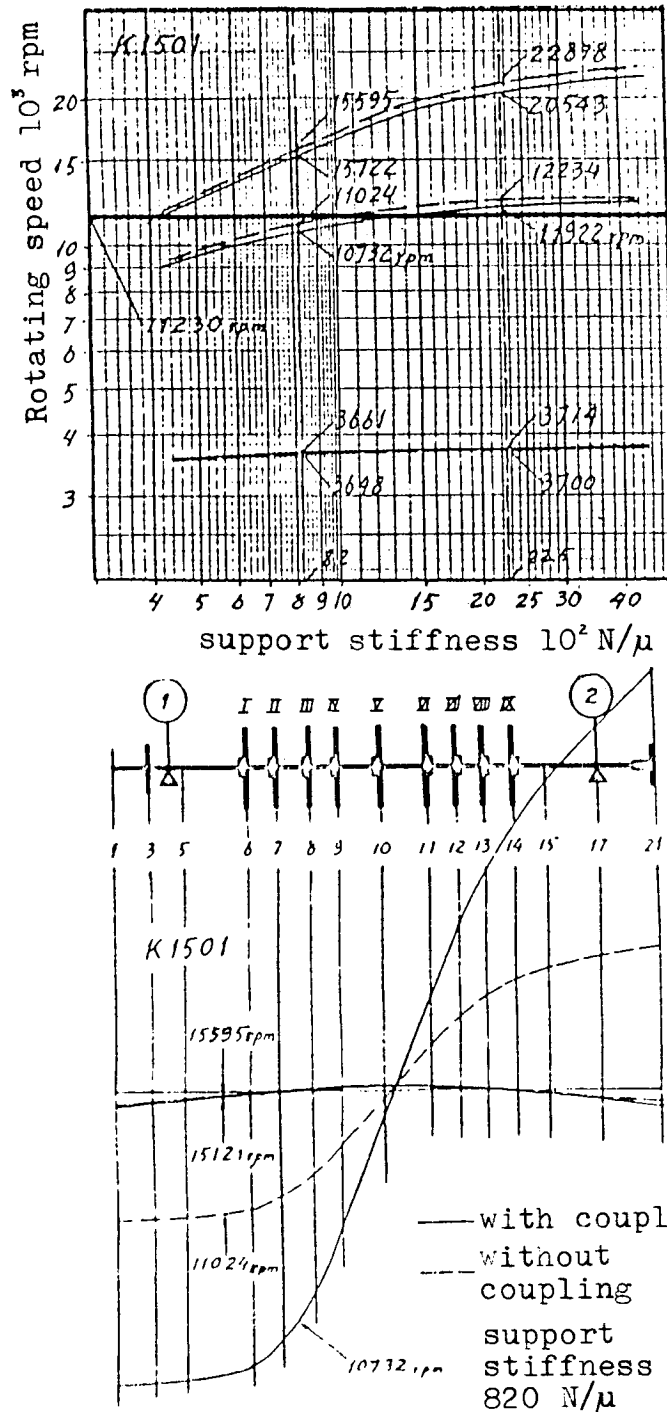


Figure 4. - Critical speed and mode shape of rotor K1501LP.

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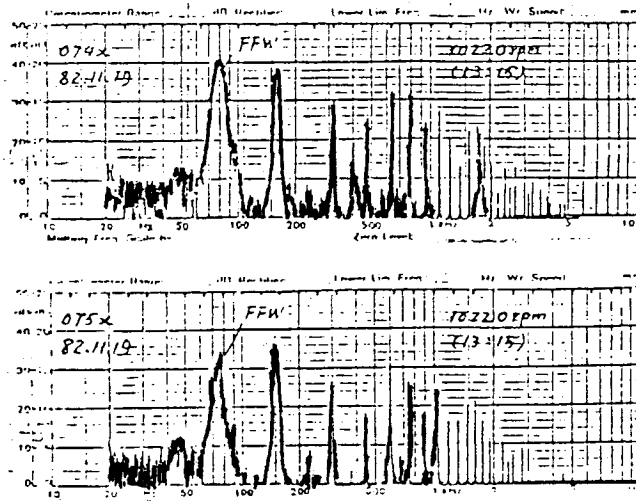


Figure 5. - Frequency spectrum of shaft vibration in bearings 074x and 075x at 10 220 rpm (Nov. 19, 1982).

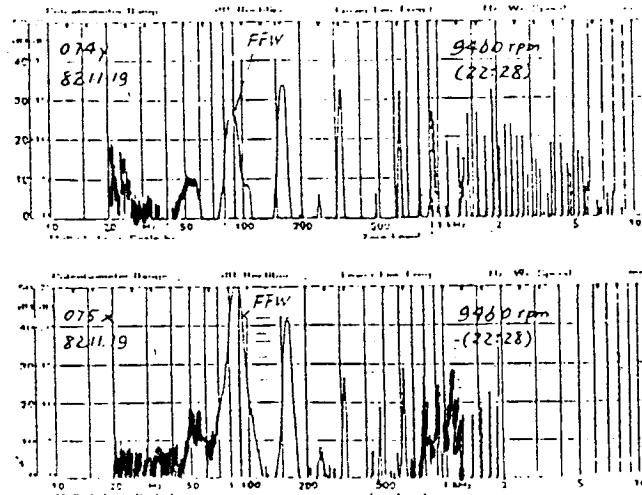


Figure 6. - Frequency spectrum of shaft vibration in bearings 074y and 075y at 9460 rpm (Nov. 19, 1982).



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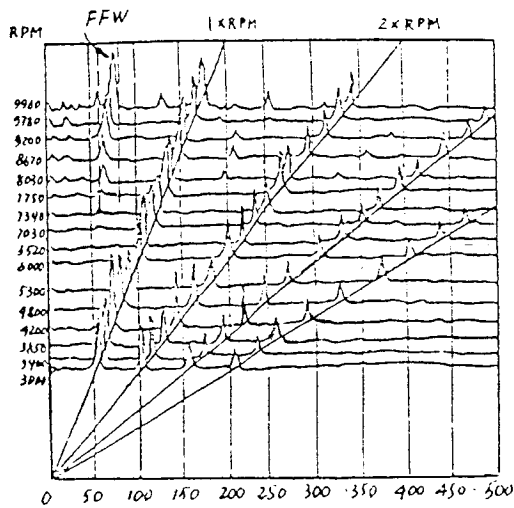


Figure 7. - Vibration speed-spectrum map.

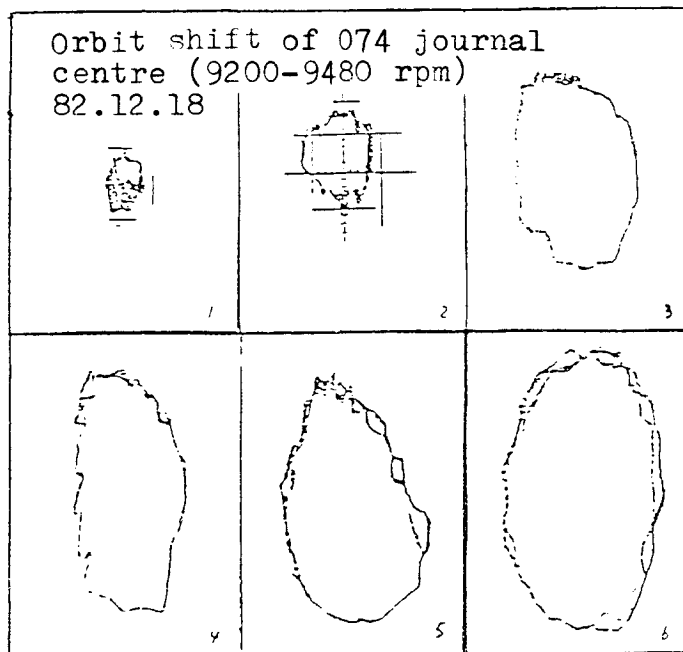


Figure 8. - Orbit of shaft center on bearing  
074 from 9200 to 9600 rpm.

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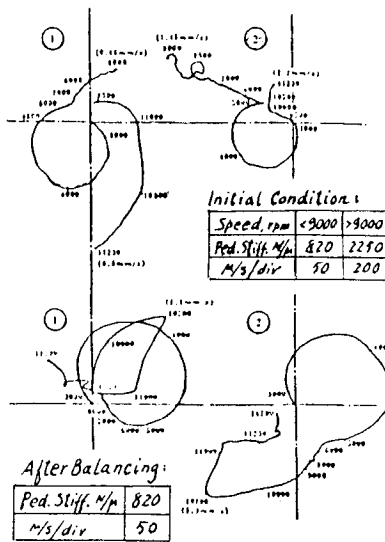


Figure 9. - Nyquist tracks before and after rotor balancing correction.

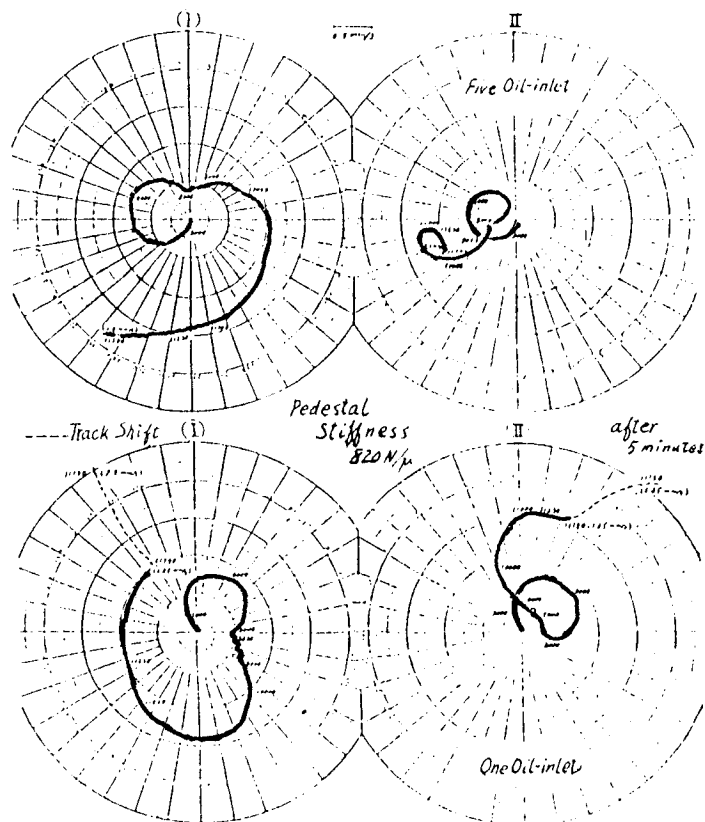


Figure 10. - Influence of number of oil-inlet holes on shaft vibration.

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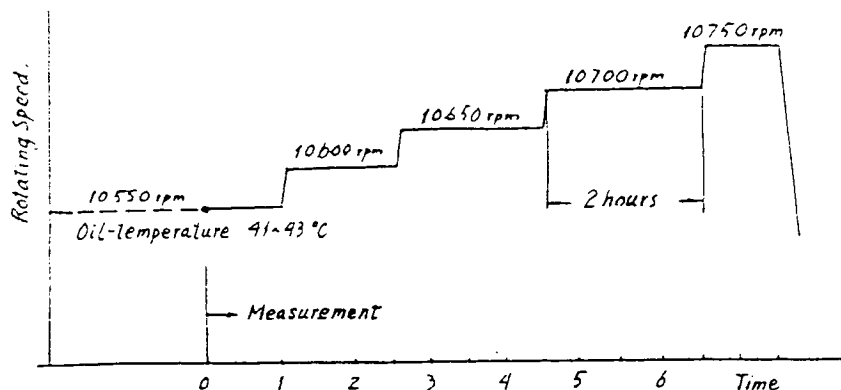


Figure 11. - Test procedure for November 8, 1983.

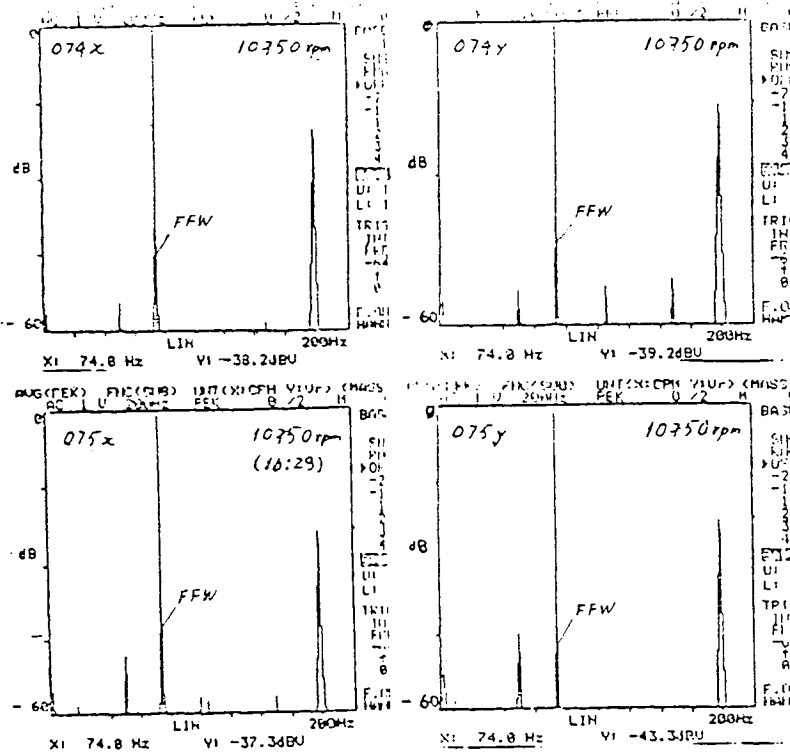


Figure 12. - Real-time analysis of shaft vibration at 10750 rpm before rotor instability.

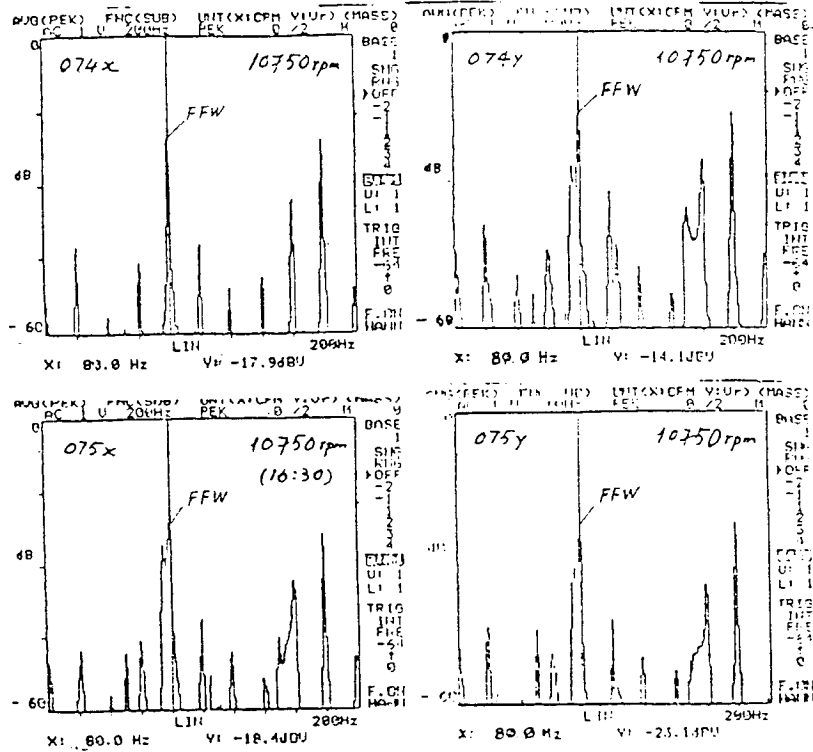


Figure 13. - Real-time analysis of shaft vibration at 10 750 rpm during rotor instability.

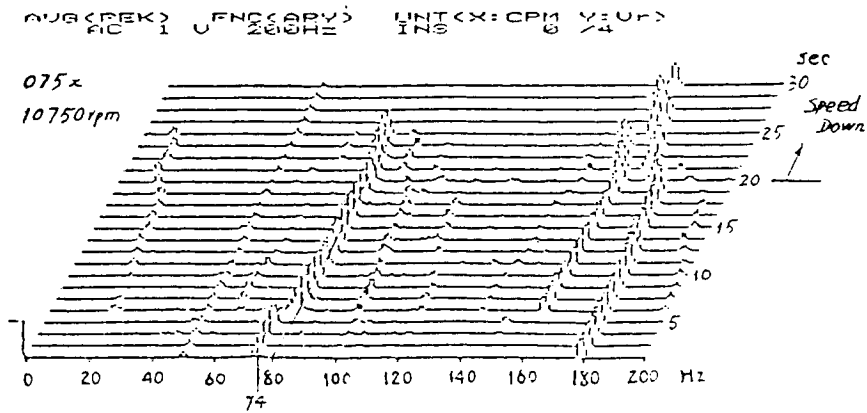


Figure 14. - Three-dimensional spectrum of shaft vibration in bearing 075 at 10 750 rpm during instability.

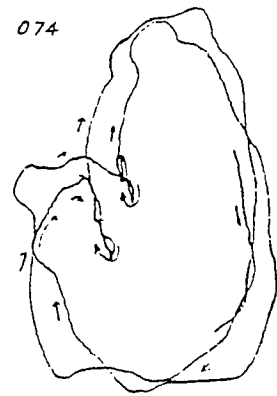


Figure 15. - Orbit of shaft center on bearing 074 at 10 750 rpm during instability.

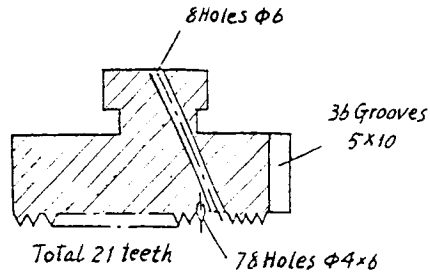


Figure 16. - Construction of  
new labyrinth.

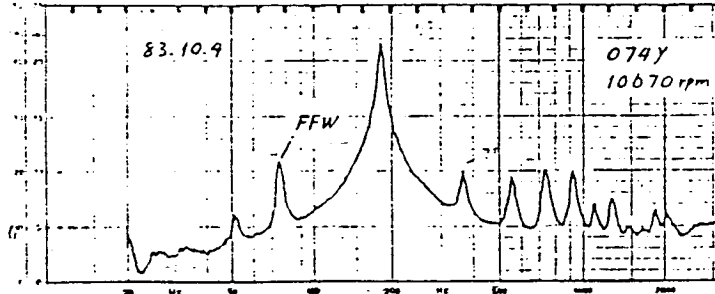


Figure 17. - Frequency spectrum of shaft vibration on bearing 074y taken with old labyrinth.

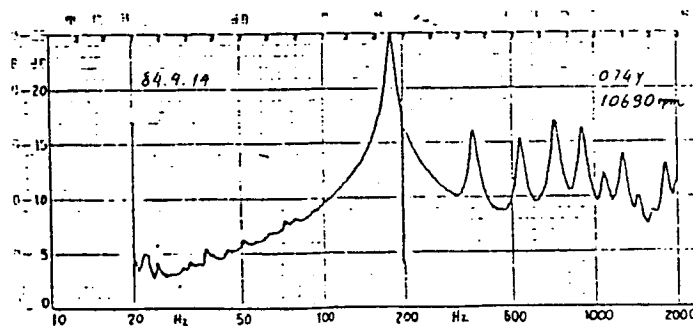


Figure 18. - Frequency spectrum of shaft vibration on bearing 074y taken with new labyrinth.