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**SOME FIELD EXPERIENCE WITH SUBSYNCHRONOUS VIBRATION OF  
CENTRIFUGAL COMPRESSORS**

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A lot of large chemical fertilizer plants producing 1000 ton  $\text{NH}_3$ /day and 1700 ton urea/day were constructed in the 1970's in China. During operation, subsynchronous vibration takes place occasionally in some of the large turbine-compressor sets and has resulted in heavy economic losses. In this paper, two cases of subsynchronous vibration are described.

(1) Self-excited vibration of the low-pressure (LP) cylinder of one kind of  $\text{N}_2$ - $\text{H}_2$  multistage compressor

(2) Forced subsynchronous vibration of the high-pressure (HP) cylinder of the  $\text{CO}_2$  compressor

**THE SELF-EXCITATION OF THE  $\text{N}_2$ - $\text{H}_2$  GAS COMPRESSOR**

The general arrangement of the  $\text{N}_2$ - $\text{H}_2$  gas compressor is shown in figure 1, and the sketch of the rotor of the LP cylinder is shown in figure 2. The specifications of the LP compressor are given below:

Shaft power, kW		8300
Max. speed, rpm		11230
Weight of rotor, kg		500
Length of rotor, mm		2050
Span, mm		1635
Diameter of main portion of the shaft, mm		150
Diameter of journal, mm		
Driven end (DE)		114
Nondriven end (NE)		104
L/D of tilting pad bearing		
Driven end (DE)		0.4
Nondriven end (NE)		0.41
Length of oil ring seals, mm		
Outer		25.4
Inner		23.8
	1st SEC	2nd SEC
No. of impellers	5	4
Suction pressure, bar	26	
Discharge pressure, bar	54	94.5
Suction temperature, °C	40	
Discharge temperature, °C	139	121

Both the thrust and journal bearings are of the tilting pad type, and the journal bearings have 5 pads. The compressor is arranged with back-to-back impeller staging. The length of the interstage labyrinth seal with 38 teeth is about 120 mm and the clearance between the surface of the shaft and the tips of the teeth is about 0.3-0.4 mm.

In the early stage, the compressor set ran at about 10050 rpm for more than three years. Its normal operation at 90 percent of capacity lasted for about 100 days after the speed was raised to 10700-10750 rpm. But it was tripped off because of a sudden destructive vibration on one early morning in May 1982. All the bearings and labyrinth seals of the LP cylinder were destroyed. The machine could not be started up again without the replacement of these parts. But the operators could not find any evidence of anything unusual or of abrupt changes from the instrument panel, and there was no extraordinary signal on the recording charts before the violent vibration occurred. The same failure occurred once again on another morning one month later.

In order to make the machine more stable, several modifications made during overhaul were as follows:

- (1) Increasing the viscosity of the oil by using No. 30 turbine oil instead of No. 20 and choosing 45 °C as the oil inlet temperature
- (2) Decreasing the bearing L/D ratio to 0.39
- (3) Using the seal rings which have the largest clearance
- (4) Creating an equalizing pressure zone in the middle of the interstage labyrinth by turning off 6 teeth to a depth of 1/2 the tooth height
- (5) Making a full speed dynamic balancing for the rotor

After the overhaul, a violent vibration occurred again when the compressor was started and run to a speed of 10200 rpm, and later on start-up failed again and again for eight times. All the modification steps taken during the overhaul were eliminated, returning to the original condition. A new gear coupling was used to replace the old one, which had some wear on the surface of the teeth. The lubrication condition of the coupling was also improved. The geometric dimensions of the tilting pad bearings were carefully controlled according to the design requirements. All these steps, however, did not make any significant progress in stability.

According to the vibration spectrum analysis, it was found that there was an instability component with a frequency of 75 Hz, which is just over the 1st natural frequency of the rotor on rigid support. The frequency was unchangeable, staying at 75 Hz. In fact, the maximum wear of the labyrinth seals occurred in the middle of the casing, and the distribution of the abrasion wear along the axial line of the compressor answered to the 1st mode shape of vibration. This means that the vibration resulted from self-excitation.

In order to get a better understanding of the characteristics of the compressor, to search into the mechanism of the instability and to predict the

effectiveness of the proposed modifications, a stability analysis was performed.

A technique similar to that described by Lund (ref. 1) was used to solve the logarithmic decrement and the damped natural frequency of the system. Though the precision of the calculation may be limited because of the boundary conditions, such as the dynamic coefficients of the bearings and seals, the tendency indicated by the calculation is valuable.

The following system states were considered for stability analysis.

(1) Rotor-bearing system: In the calculation of this state, all the aerodynamic and hydrodynamic effects of seals were neglected. So there are no excitations from labyrinth or floating ring type oil film seals, etc. In the calculation, the internal friction and damping of the shaft material were also neglected.

The results are presented in figure 3-a. The abscissa of points at which the dotted line and the damped natural frequency lines intersect are critical speed values. The figures printed beside the points on the frequency lines are  $\delta$  in the corresponding states.

As shown in figure 3-a, all the values of  $\delta$  are greater than zero up to  $n = 15000$  rpm. This means that if only the effect of the bearings is considered, the system will not lose its stability within its operation range. This conclusion tallies with the characteristics of tilting pad bearings.

(2) Rotor-bearing-seal system: The action of an oil ring seal on the rotor is something like that of a bearing though its effect is not immense.

By considering the effect of seal elements, the sign and value of  $\delta$  obtained in the stability analysis was changed to  $-0.038$  at a running speed of about 11000 rpm. Though the calculated instability threshold is somewhat greater than the practical value, the calculation indicates that, at least, the stability safety factor of this machine is small. The small excitations produced by the seal elements are great enough to cause instability. In order to find a solution, the effectiveness of several proposed modifications was also evaluated by the stability calculation method.

It was found that the following measures would significantly improve the stability of the rotor system:

(1) To enlarge the shaft diameter and/or shorten the bearing span, i.e., to increase the 1st critical speed of the rotor system

(2) To modify the parameters of the oil ring seals, especially that of the construction of the long middle labyrinth

(3) To use a flexible support squeeze film damper

It will take much time to adopt any of these modifications and the factory cannot be permitted to stop its production that long. Therefore effective, easily implemented measures, though temporary, were urgently sought.

According to the experience with these failures, we found that whenever the magnitude of the 75 Hz component grew, the machine ran unstably. If the magnitude of this component becomes greater than that of the main component, i.e., the running frequency component, catastrophic vibration can occur at any moment. The spectrums of different conditions are shown in figure 4.

This phenomenon was used to monitor the operation of the LP cylinder when the compressor was brought into operation with a reduced load of 80-85 percent after a thorough fixing of the whole system. Whenever the amplitude of the 75 Hz component grew and approached that of the running frequency component, the speed and load had to be reduced for a while until the peak disturbance exciting the LP cylinder rotor decreased to a safe limit.

The 75 Hz component can be monitored only by a real-time spectrum analyzer, so monitoring is not convenient. Because the figures of the shaft center orbit are closely related to the amplitude of the subsynchronous component and can be easily observed on an oscilloscope screen, a cheap oscilloscope, as a temporary measure, was installed in the field to monitor the condition of the LP cylinder.

The scheme of this installation is shown in figure 5. When the machine worked normally, the boundary line of the shaft center orbit was clear and stable. Otherwise the boundary line became indistinct and oscillatory, and the orbit became large as shown in figure 6. With the help of this method, production was preserved for more than one year before the final modification of the LP cylinder was performed.

#### THE FORCED SUBSYNCHRONOUS VIBRATION OF CO<sub>2</sub> COMPRESSORS

In China, centrifugal compressors have been generally used in urea production to compress CO<sub>2</sub> gas to a final pressure of 145 kg/cm<sup>2</sup>. The main parameters of the most common machines are shown below:

Composition of gas	CO <sub>2</sub> -94.5%	Air-4%	Others-1.5%
Relative humidity			100%
Av. mol. wt.			41.2% (40 °C)
Suction temperature, °C			<40
Suction pressure, kg/cm <sup>2</sup>			0.96
Discharge temperature, °C			127
Discharge pressure, kg/cm <sup>2</sup>			144

Type of rotor	Turbine	LP	HP
Normal speed, rpm	7200	7200	13900
1st crit. speed, rpm	4500	4200	8000
2nd crit. speed, rpm	9500	9200	17520

The schematic diagram of this compressor is shown in figure 7, and the sketch of the HP rotor is shown in figure 8. Most of the HP cylinders of this type of CO<sub>2</sub> compressor suffered from high vibration. Up to now, this serious vibration problem has not been solved completely. Many factories can only raise the alarm threshold to continue working. Even so, the HP rotors often work over the raised alarm threshold. This was the reason why we surveyed the vibration failure of this type of machine and collected the information about it.

There are nearly 20 similar HP rotors in Chinese urea synthesis factories. These rotors were manufactured according to the same detailed drawings and can be used interchangeably. The difference between them is only the dimension tolerances. Their performances, however, seemed to be similar but not quite the same.

We put forward some typical cases for analysis:

(1) A few of these machines work quite well. The peak-to-peak amplitudes of HP rotors measured by eddy current probes near the bearings are less than 30  $\mu\text{m}$ . For example, in factory A, the vibration amplitudes of driven end (DE) and nondriven end (NE) HP rotors are less than 30  $\mu\text{m}$  and 25  $\mu\text{m}$ , respectively. Their vibration spectrums are shown in figure 9. The main vibration component is a synchronous component. The subsynchronous components are very small. Only the component at 180 Hz is slightly greater. The main component in the pressure pulsating spectrum of the discharge pipe is also at 180 Hz.

(2) In some factories, factory G for example, the vibration amplitudes of the HP rotors are at a high level ( $>100 \mu\text{m}$ ). The spectrum is shown in figure 10. The synchronous component is leading. After full-speed dynamic balancing, the amplitude can be greatly reduced. So the vibration results from the loss of balancing accuracy.

(3) The amplitudes of many HP rotors are not very high, but the vibrations of the pipes are a serious problem. The vent pipes, drainage pipes, and instrument tubings connected to the discharge pipe were often broken by strong vibration during the early stage after the factories were put into operation, and this breakage was the main reason the compressor shut down. Strong vibration of the pipes may result from resonance of the pipes with one of the subsynchronous frequencies in the discharge gas stream. In factory B, the spectrum obtained during the measurement of the natural frequency of the pipes is shown in figure 11-a. The main component is at 115 Hz. The spectrum of the discharge gas pressure pulsating under normal conditions is shown in figure 11-b. The main component is at 190 Hz. The vibration spectrum of one pipe section is shown in figure 11-c. The main component is at 110 Hz. The amplitude of this pipe at that time is 12-18  $\mu\text{m}$ . Under abnormal conditions, the spectrum of the discharge gas pressure pulsating is shown in figure 11-d. At that time, the main component is at 110 Hz, nearly equal to the natural frequency of that pipe section. So the amplitude increased suddenly to 330  $\mu\text{m}$  as shown in figure 11-e. In the spectrum of the rotor at DE, the 115-Hz component, as shown in figure 11-f, appeared simultaneously. In some factories, the pipe vibrations were eliminated by varying the turn radius or support condition of the discharge pipes.

(4) In the start-up period of many of these machines, the subsynchronous components (mainly at 53, 120, and 180 Hz) were small if the speed was less than 3000 rpm, but they grew gradually with the speed. During the period of rising pressure, the amplitude of the synchronous component grew a little. At the moment when the pressure rose to 110  $\text{kg}/\text{cm}^2$ , the machines vibrated violently. For example, in factory C, the amplitude of DE rose from 25 to 37  $\mu\text{m}$ , and not only the pipes, but also the foundation and floor vibrated so intensely that they generated a very loud noise. The vibration spectrums of the rotor and the pressure pulsating of the discharge pipe at that moment are

shown in figure 12-a and b, respectively. As compared with the spectrum at 145 kg/cm<sup>2</sup>, there is a noteworthy subsynchronous component at 58.3 Hz. Its amplitude exceeds that of the synchronous component. At 145 kg/cm<sup>2</sup>, the main component in figure 12-c and d is the synchronous one.

(5) During the operation of many rotors, instantaneous high amplitude vibration occasionally appeared. The total amplitude reached a maximum of 110 μm. At that moment, accompanied by a low, deep sound of gas flow pulsation, the vibration of the foundation and floor deck increased greatly. The flow rate of the compressor dropped 10 percent. In a case of violent vibration, in factory C, the recorded process parameters were an HP rotor speed of 13400 rpm, a flow rate of 28000 Nm<sup>3</sup>/hr and a discharge pressure of 145 kg/cm<sup>2</sup>.

The amplitude of the HP rotor was greater than 50 μm. The foundation and floor deck vibrated severely. The spectrums in the normal and alarm states are shown in figure 12-c and figure 13, respectively. After making a comparison between these two spectrums, one may come to the conclusion that the violent vibration was caused by the 58.3-Hz subsynchronous component which grew remarkably and exceeded the synchronous component. The same condition occurred in other factories. It is considered that the vibrations are caused by rotating stall. Therefore, two methods have been used to increase the volume flow rate of the fourth stage of the HP cylinder:

(1) To increase the temperature of the fourth stage inlet gas

(2) To increase the circulation flow in the fourth stage by using a 4-to-4 bypass valve

These methods can decrease the vibration in some factories, and a comparison between amplitudes (μm) gives the following:

	Factory E			Factory F		
With 4-to-4 bypass valve closed	96	94	81	68	53	31.7
With 4-to-4 bypass valve opened	43	31	20	40	34	31.7

The effectiveness of these methods shows that the above estimation is correct. Method (2) is rather simple but costs an extra 10 percent of energy. Method (1) has only limited effectiveness. A better solution was needed at that time.

After a thorough study, a newly designed rotor with adjusted impeller structure parameters was constructed by Shen-yang Blower Manufactory. In operation, its performance is satisfactory. Under design conditions, its total vibration amplitude can reach a very low level and the amplitudes of the subsynchronous components are very small. The normal spectrums of rotor and gas pressure are shown in figure 14-a and b. If the operation was not correct, the large subsynchronous component at 55 Hz certainly appeared and caused vibration. The abnormal spectrums are shown in figure 14-c and d. Subsynchronous components at 110 and 165 Hz can also be significant.

Some discussions and concluding remarks about the vibrations of CO<sub>2</sub> compressors are as follows:

(1) The high vibration of CO<sub>2</sub> HP rotors is due to the forced subsynchronous vibration caused by aerodynamic instability. Its characteristics are similar to those pointed out by reference 2. The amplitude is stable. The machine can run for a long time in a state of serious vibration.

(2) On some occasions, the amplitude of the HP rotor increases to 100 μm, but the compressor operation appears to be normal. Under other conditions, however, the amplitude of the HP rotor is only about 40 μm, yet the floor deck and the foundation vibrate severely. These phenomena depend upon whether the pipes are excited by the gas stream or not.

(3) The main subsynchronous components are about 60, 120, and 180 Hz. The component at 60 Hz (within the range of 57-64 Hz), which is 25 percent of the synchronous frequency, seems more powerful. According to reference 3, it may be considered that the vibration is caused by the rotating stall in the diffuser. As statistics (ref. 4) show, however, the vibration is closely related to the ratio of the relative inlet velocity  $W_1$  to the relative outlet velocity  $W_2$  of the impellers. When the HP rotor of a CO<sub>2</sub> compressor works with  $(W_1/W_2) < 1.4$ , it operates quite well, otherwise the vibration will be high. This means that the impeller parameters have a more important effect on the rotating stall than the static components even if the subsynchronous frequencies are as low as 25 percent of their own.

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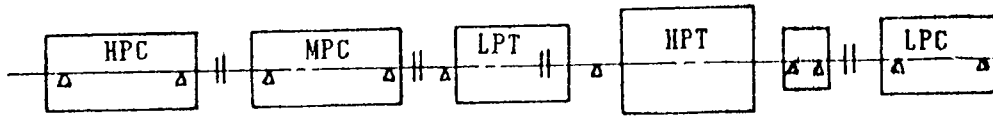


Fig. 1 General arrangement of the N<sub>2</sub>-H<sub>2</sub> compressor set

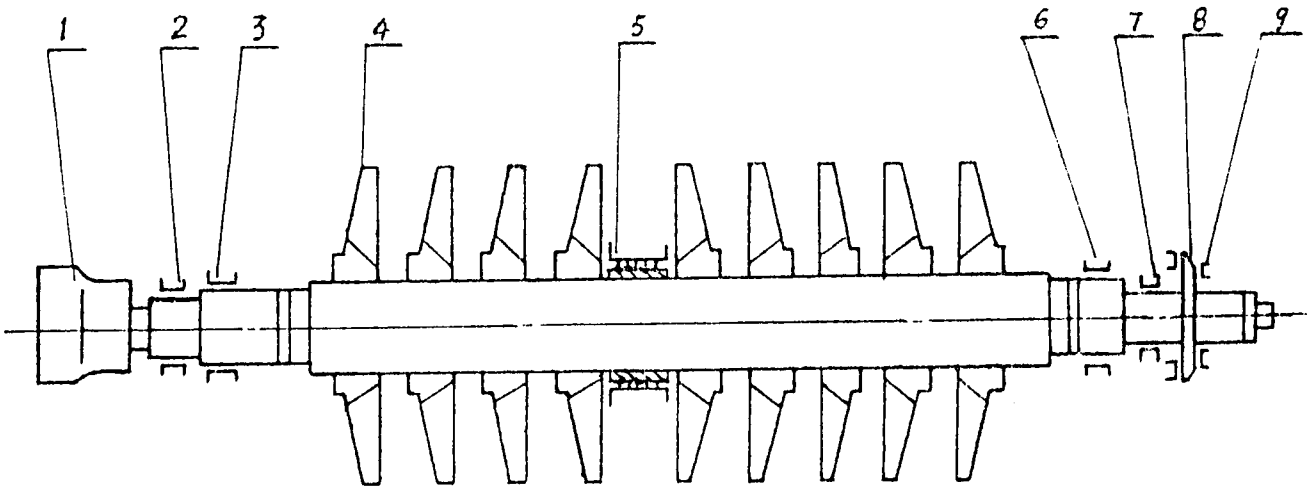


Fig. 2 Sketch of the low pressure rotor

- 1 gear coupling
- 2,7 bearing
- 3,6 oil ring seal
- 4 impeller
- 5 interstage seal
- 8 thrust disk
- 9 thrust pads



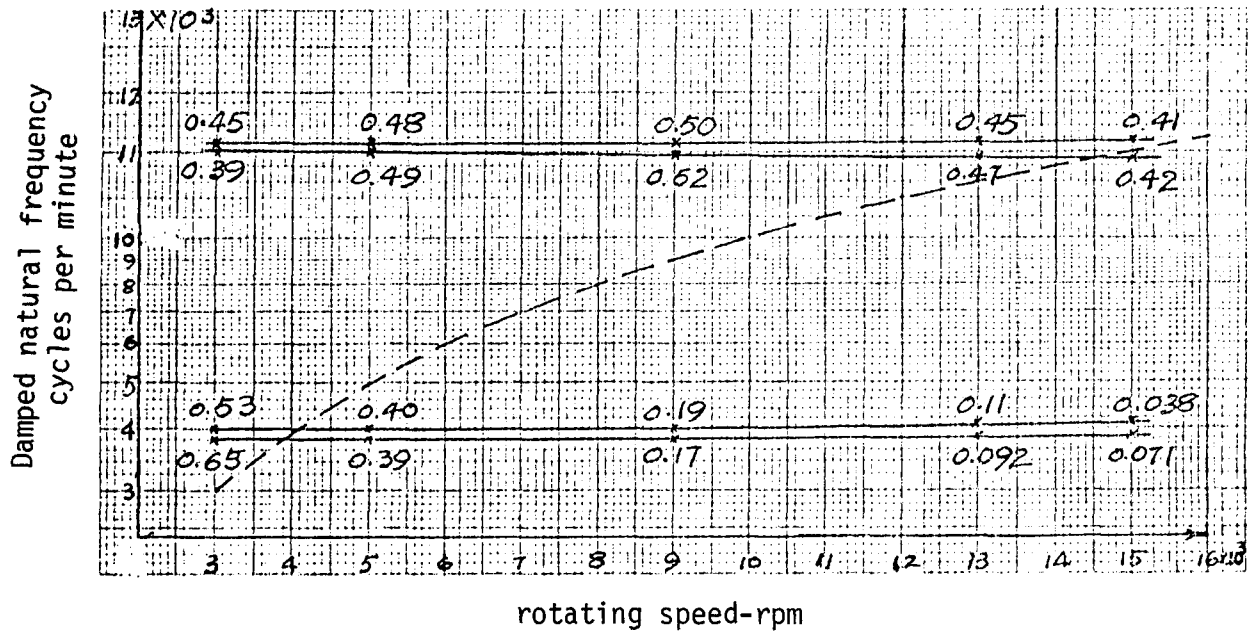


Fig. 3-a  $\delta, \omega$  versus rotating speed (rotor-bearing system)

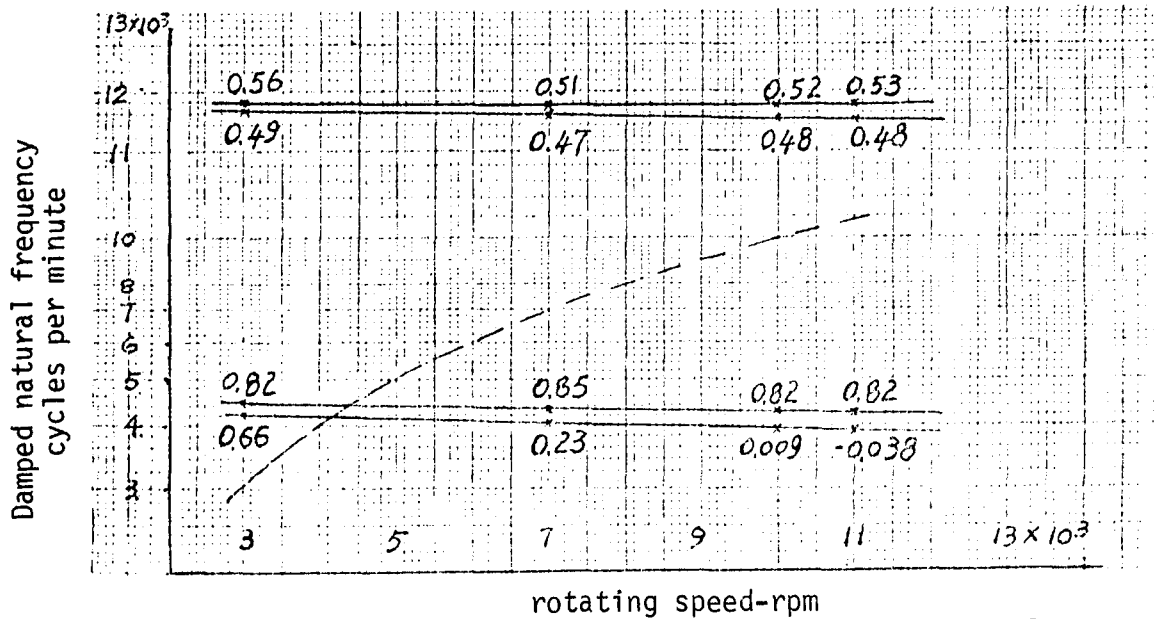


Fig. 3-b  $\delta, \omega$  versus rotating speed (rotor-bearing-seal system)

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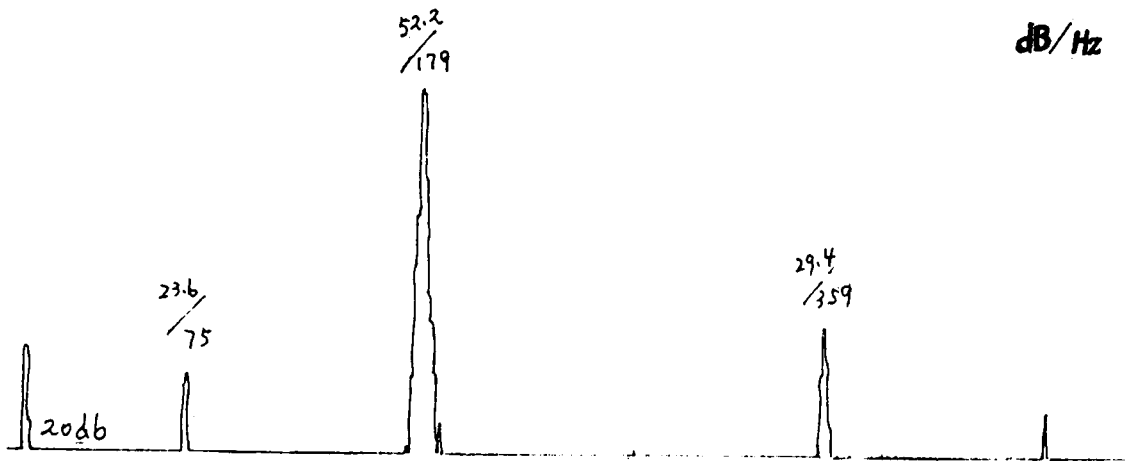


Fig. 4 - a

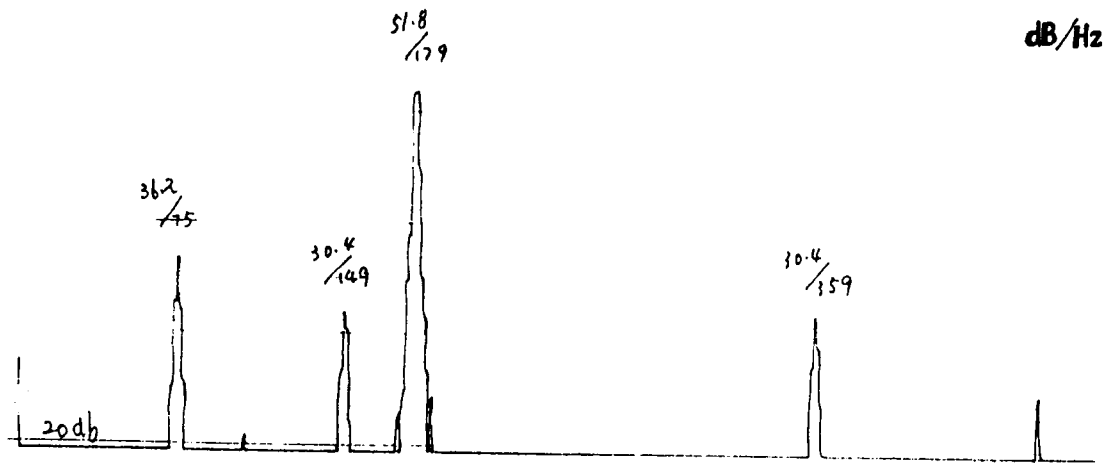


Fig. 4 - b

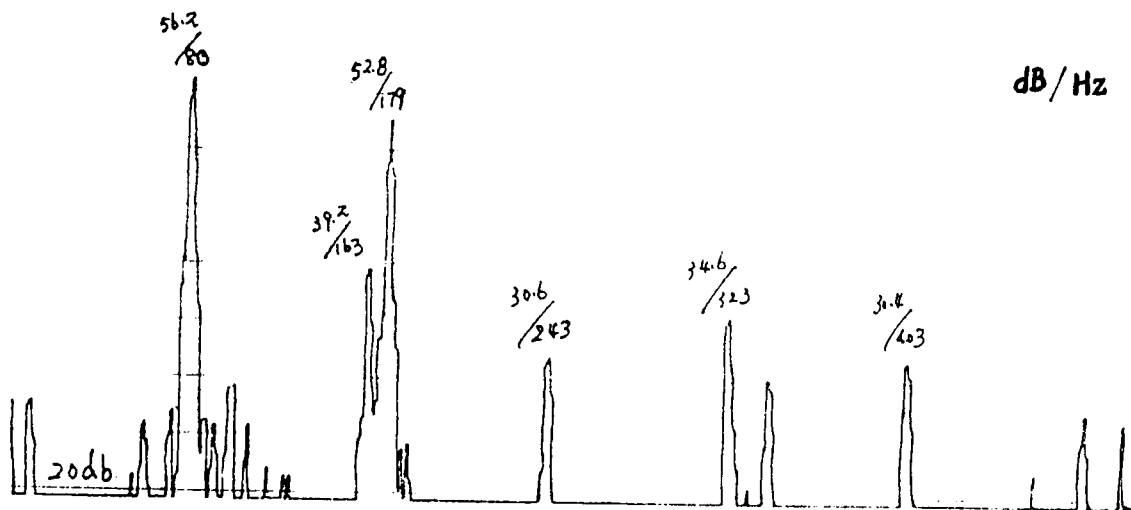


Fig. 4 - c

Fig. 4 Spectrum  
 a - stable state  
 b - transient state  
 c - unstable state

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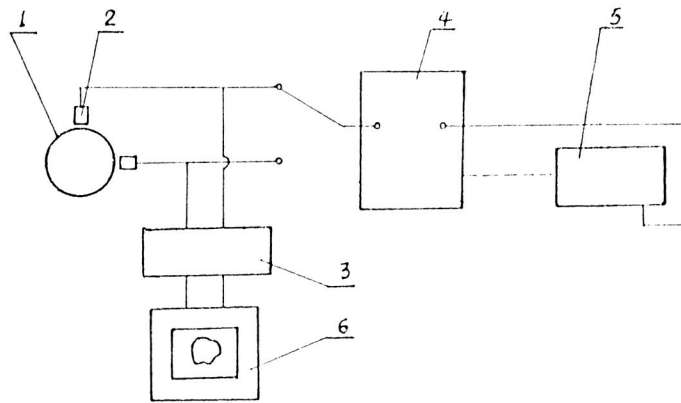


Fig. 5 Schematic diagram of orbit monitoring installation and spectrum

- |                       |              |                  |
|-----------------------|--------------|------------------|
| 1 - shaft             | 2 - probe    | 3 - amplifier    |
| 4 - spectrum analyzer | 5 - recorder | 6 - oscillograph |

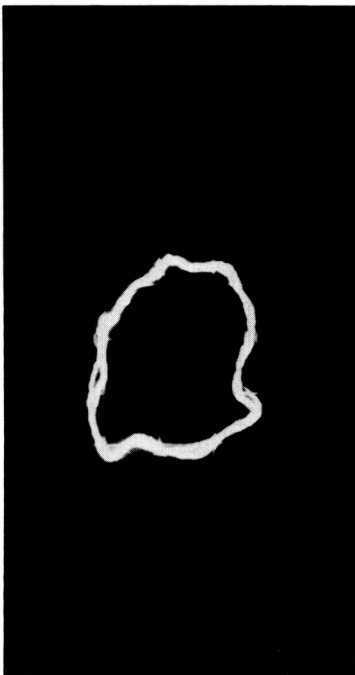


Fig. 6 - a



Fig. 6 - b  
Fig. 6 Orbit of shaft centre  
a - normal state  
b - transient state  
c - unstable state

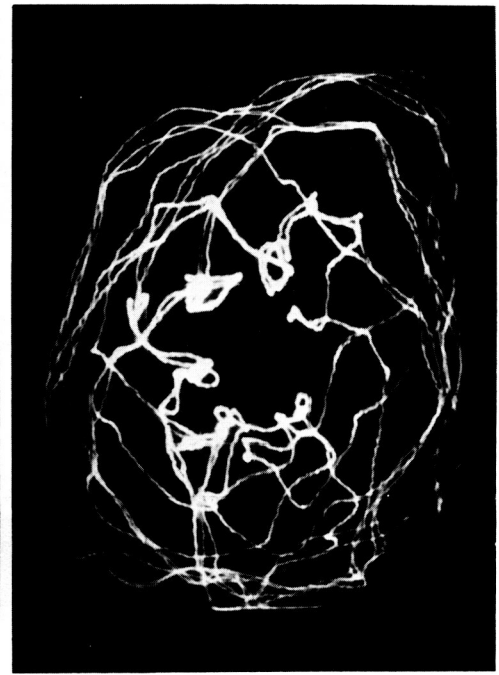


Fig. 6 - c

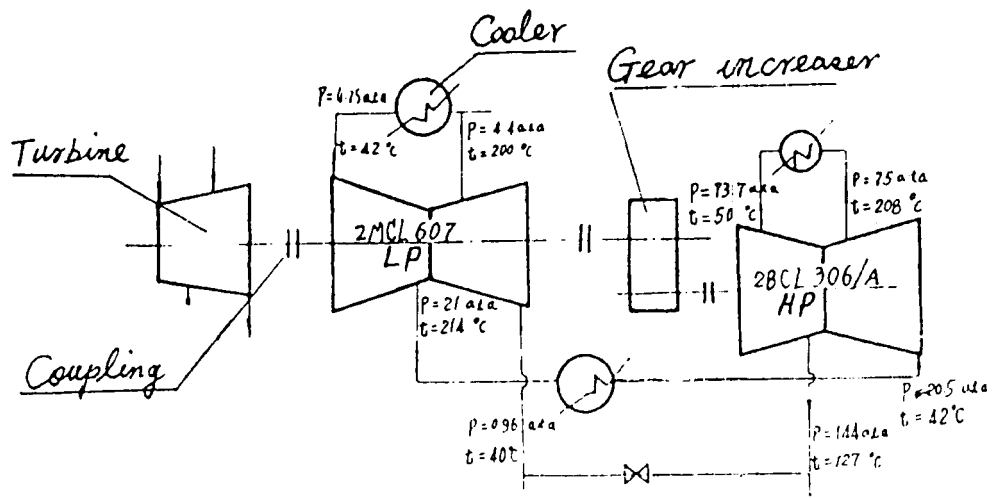


Fig. 7 Schematic diagram of CO<sub>2</sub> compressor

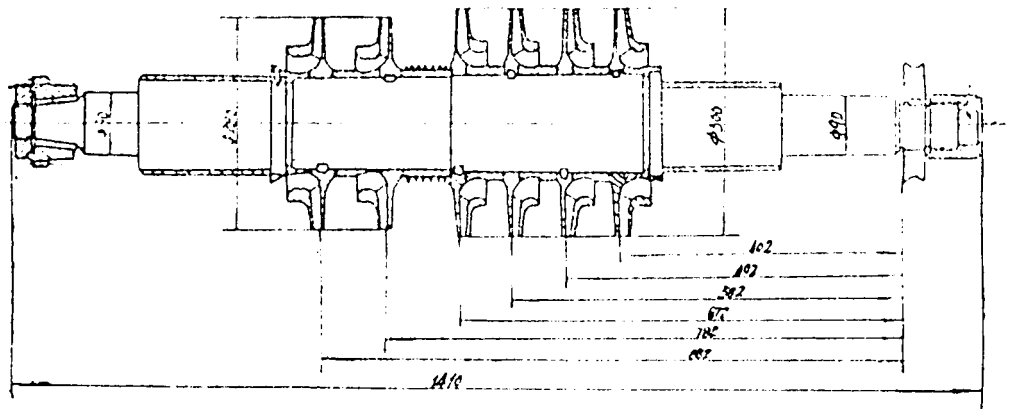


Fig. 8 Sketch of CO<sub>2</sub> high pressure rotor

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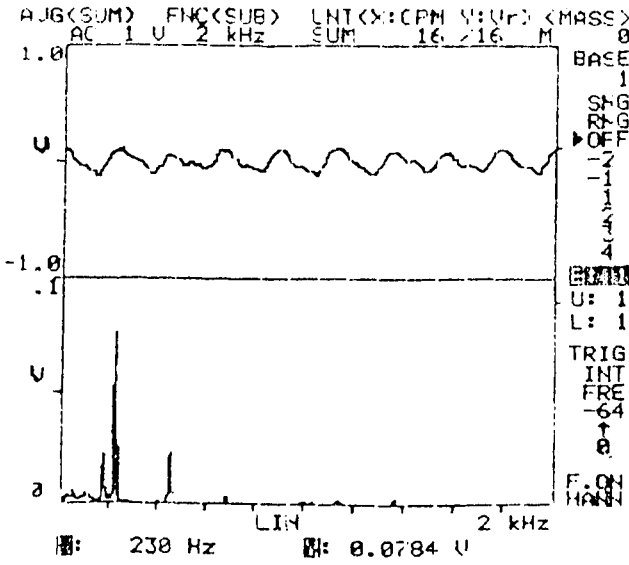


Fig. 9 - a

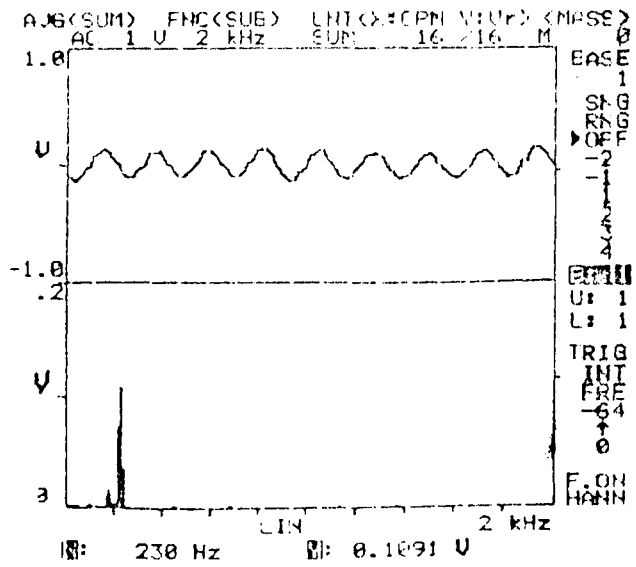


Fig. 9 - b

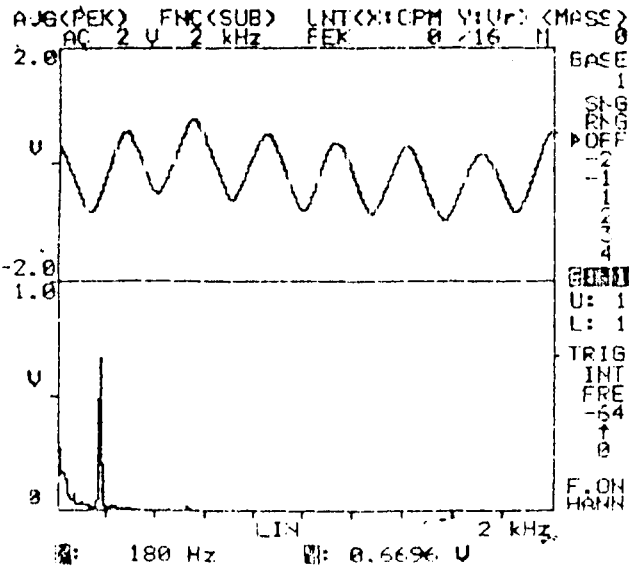


Fig. 9 - c

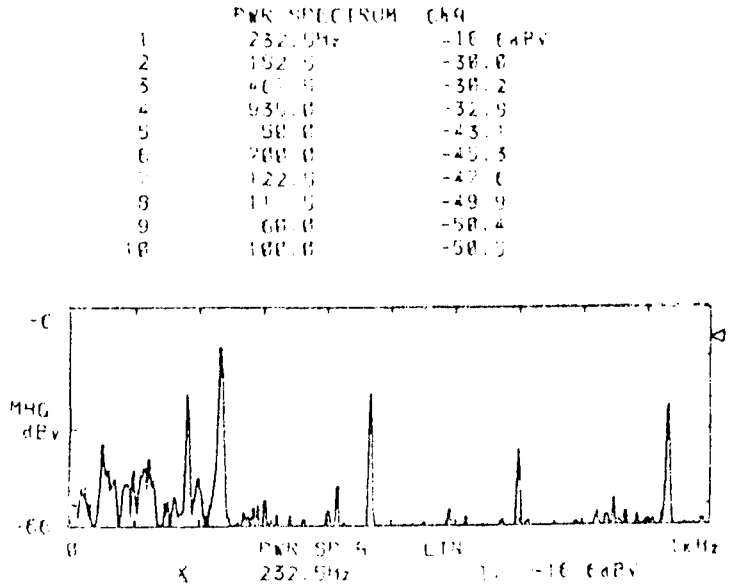


Fig. 10 Vibration spectrum - plant G

- Vib. 9 Vibration spectrum - plant A
- a - DE
  - b - NE
  - c - gas pressure pulsating in outlet pipe

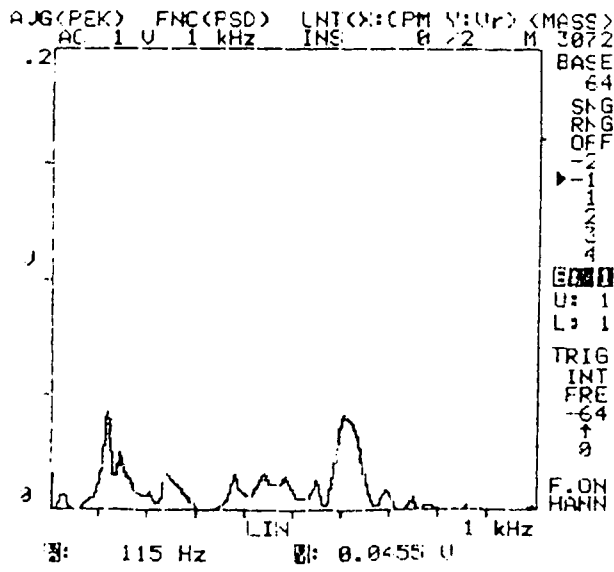


Fig. 11 - a

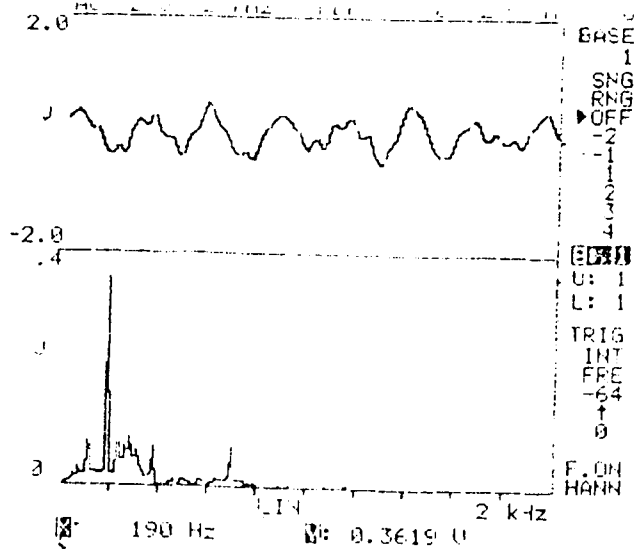


Fig. 11 - b

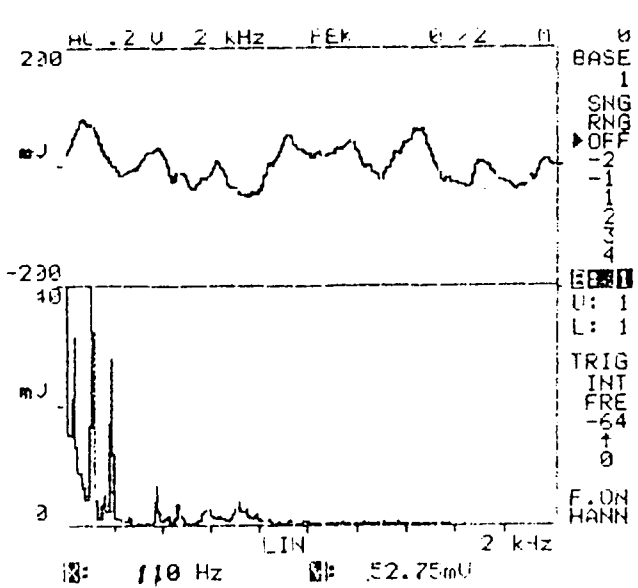


Fig 11 - c

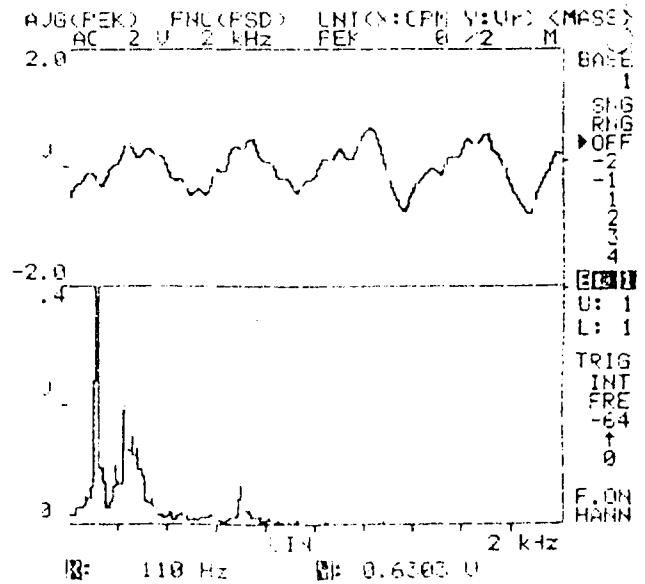


Fig 11 - d

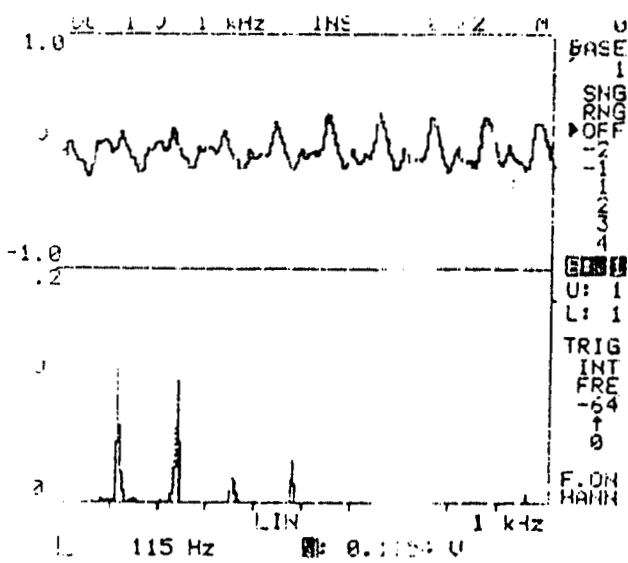


Fig. 11 - e

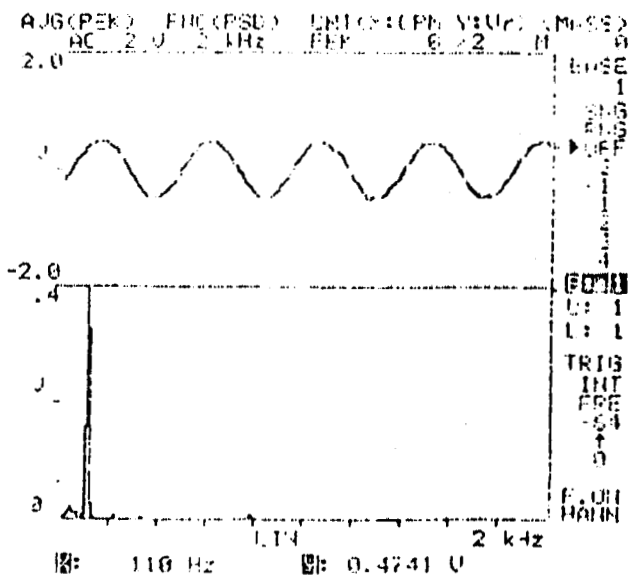


Fig. 11 - f

- Fig. 11 Vibration Spectrum - plant B
- a - measurement of nature frequency of discharge pipe
  - b - pressure signal under normal condition
  - c - vibration of pipe under normal condition
  - d - pressure signal under abnormal condition
  - e - vibration of pipe under abnormal condition
  - f - DE

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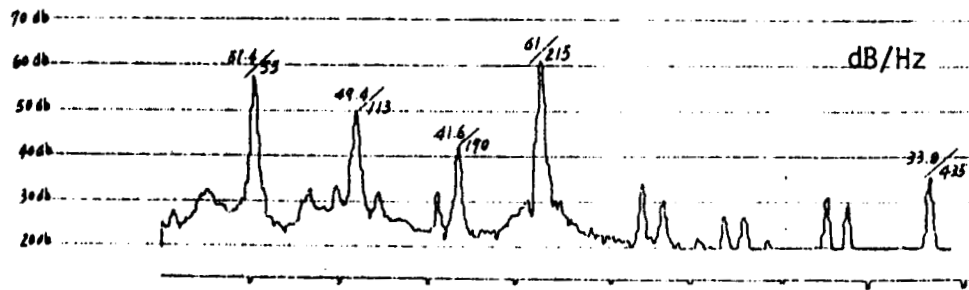


Fig. 12 - a

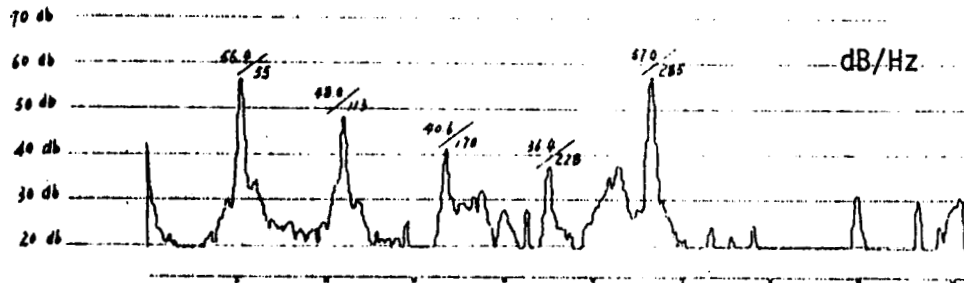


Fig. 12 - b

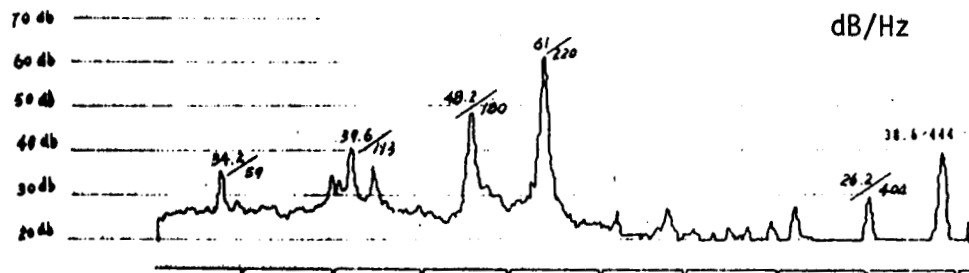


Fig. 12 - c

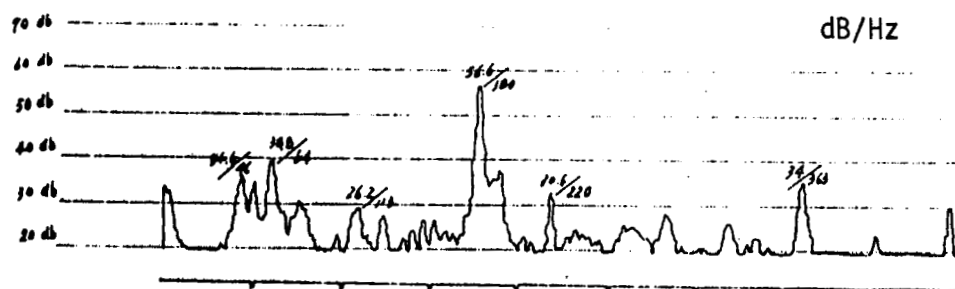


Fig. 12 - d

Fig. 12 Vibration Spectrum  $\frac{1}{2}$  plant C

- a - DE. at 110kg/cm<sup>2</sup>
- b - gas pressure signal at 110kg/cm<sup>2</sup>
- c - DE. at 145 kg/cm<sup>2</sup>
- d - gas pressure signal at 145kg/cm<sup>2</sup>



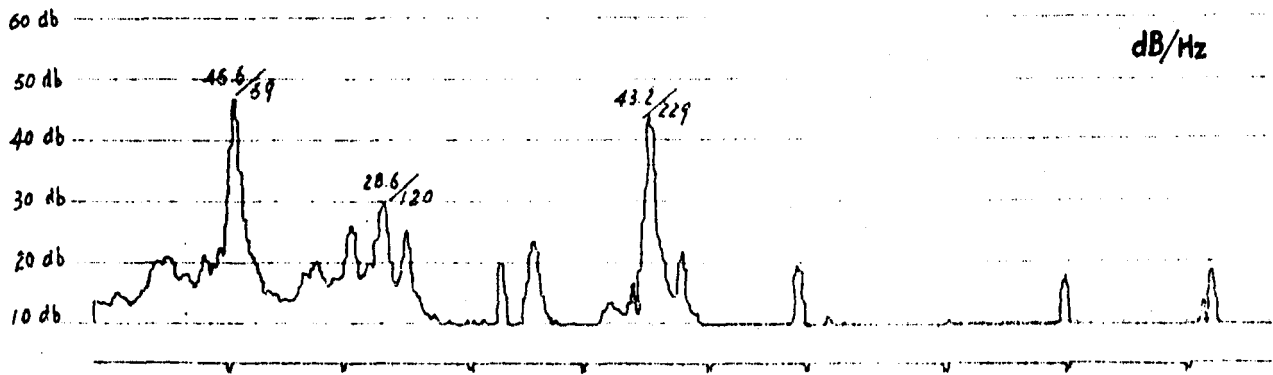


Fig. 13 Vibration Spectrum - plant C  
DE at 145kg/cm<sup>2</sup> under violent vibration

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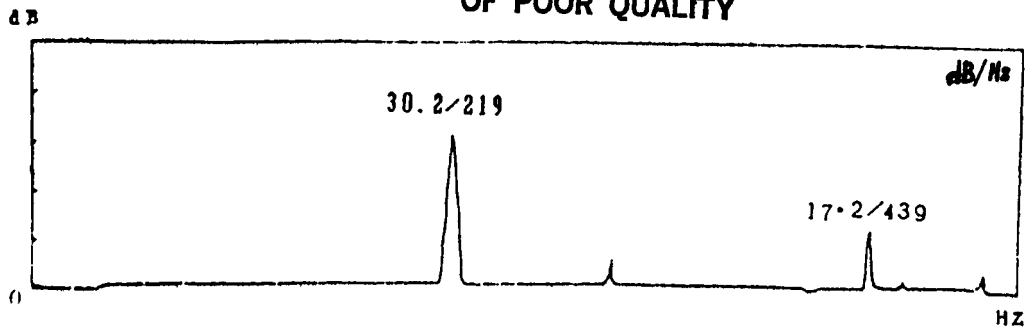


Fig. 14 - a

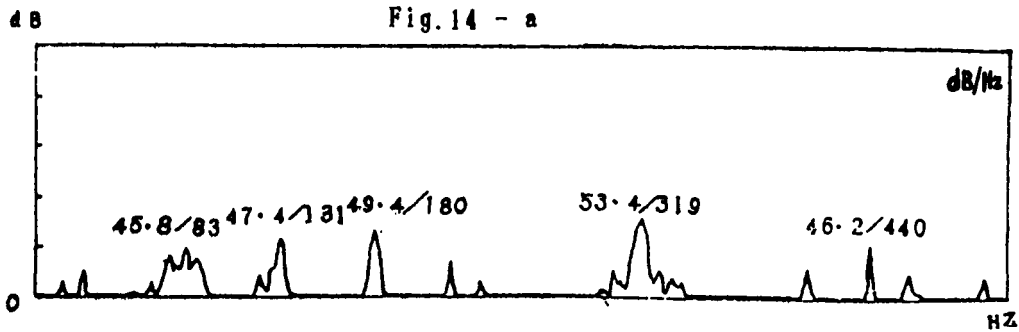


Fig. 14 - b

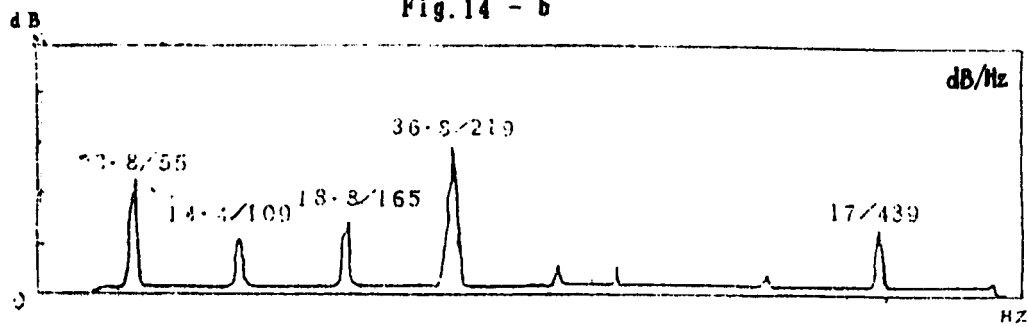


Fig. 14 - c

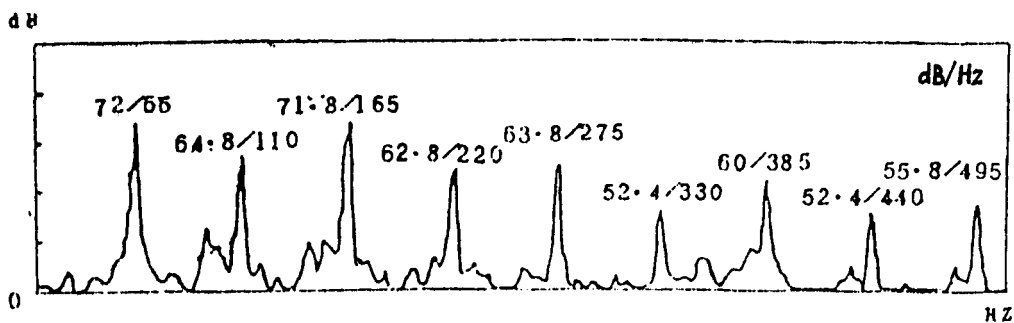


Fig. 14 - d

Fig. 14 Vibration Spectrum of newly designed CO<sub>2</sub> HP rotor  
a - DE, under normal operation condition  
b - gas pressure signal under normal condition  
c - DE, under incorrect operation condition  
d - gas pressure signal under incorrect operation condition