FIELD EXPERIENCES WITH RUB INDUCED INSTABILITIES IN TURBOMACHINERY*

David G. Goggin
Radian Corporation
P.O. Box 9948
Austin, Texas 78766

SUMMARY

Rotordynamic instability problems are not uncommon in high speed industrial turbomachinery. One type of the many destabilizing forces that can occur is caused by a rub between the stationary and rotating parts.

Descriptions are given of several cases of rub induced instabilities. Included in the descriptions are the conditions at onset, the whirl frequency and direction, and the steps taken to eliminate the problem.

INTRODUCTION

Rotordynamic instability problems are not uncommon in high speed industrial turbomachinery. Actually, a large percentage of industrial machines operating at high speed will sustain some low level subharmonic vibrations even during normal operation. Some of these units have been observed to develop instability problems. One mechanism that has been responsible for some of these instabilities is a rub between the rotating and stationary parts.

The mechanics involved during a rub have been described by several authors. Den Hartog (ref. 1) describes rubs from the standpoint of dry friction whip with Coulomb friction between the rotating and stationary parts providing the destabilizing force. Ehrich (ref. 2) added stator flexibility to this model in an attempt to define the conditions necessary for a rub to be unstable since not all rubs produce instabilities. Both analyses conclude that the instability should produce a backward whirl at the rotor natural frequency.

Bentley (ref. 3) proposed and experimentally demonstrated that several mechanisms, including a partial rotor-stator rub, could produce a subsynchronous vibration due to the periodic variation it causes in the rotor's support stiffness. The subsynchronous vibration was stated to occur most often at exactly 1/2 the running speed when operating at or slightly above twice the first critical speed. The whirl direction could be forward, backward, or even in a single plane. Although less common, the whirl frequency was noted to also have occurred at 1/3 and 1/4 running speed. It was also stated that it was possible, but not likely, for the subharmonic to occur at 1/5, 1/6, . . . and 2/3, 3/4, 2/5, 3/5, . . . of running speed. This parametric excitation mechanism was further analyzed by Childs (ref. 4 and 5). The rub model used included the effects of both Coulomb friction and the

* This work was funded by The Hartford Steam Boiler Inspection and Insurance Company.

periodic variation in support stiffness. The analysis indicated that zones of instability exist around integer multiples of the first critical speed within which a rub could cause an instability to occur. The frequency of the whirl would occur at slightly more than the first critical speed causing a vibration at exactly 1/2, 1/3, . . of running speed. The possibility that the rub would be unstable was found to be primarily dependent on the amount of damping of the first mode, the magnitude of the Coulomb friction factor, and a parameter q which represents several parameters that essentially indicate the severity of the rub. An unstable rub was found to be promoted by low damping, a high friction coefficient, or a high q factor, i.e., a severe rub.

In this paper some field experiences with rub induced instabilities are reviewed. As with most industrial problems, the pressure to resume production often prevented a more detailed investigation. But most of the essential characteristics involved are presented including the conditions at onset, the whirl frequency, the direction of whirl, and the steps taken to correct the problem. Some rubs, of course, have occurred that were not unstable, such as the one shown in figure 1 from a 5000kw turbine generator. These are usually characterized by increases in the higher order harmonics as well as a general increase in the low level broad band vibration. While these types of rubs can be serious, it is usually the unstable rubs that have the most potential for damage and so are of deep concern to the industry.

BACK PRESSURE TURBINE ON SYNTHESIS GAS COMPRESSORS

Figure 2 shows a schematic of a 20,000 H.P., 11,000 rpm Synthesis Gas Compressor train composed of a condensing turbine, a back pressure turbine, a low pressure and a high pressure compressor. The back pressure turbine, or topping turbine as it is referred to, is a two stage 12,000 H.P. drive through turbine using 10.34 MPa (1500 psi) inlet and 3.79 MPa (550 psi) exhaust steam. The first and second critical speeds are 6750 rpm and 13,000 rpm respectively as determined from proximity probe data during start-ups and overspeed trip tests. The 5800 kg (913 lb) rotor is supported in a 12.7 cm (5 in) diameter bearing at the inlet end and a 7.6 cm (3 in) diameter bearing at the discharge end. Both are 5 shoe tilting pad bearings with a load on pad orientation. This machine, like many others, tends to show some low level (2.54 μm or 0.1 mil) subharmonic vibration over a broad band of about 30-150 Hz during normal operation.

During the start-up of one of the many units like this, a vibration instability problem developed at the discharge end of the topping turbine. The vibration levels, as indicated by proximity probes, increased from 25 - 38 μm (1.0-1.5 mils) to 76 - 89 μm (3.0-3.5 mils) in a few seconds. Since a real time analyzer was present to monitor the start-up, it was observed that the vibration was dominated by a large fluctuating component at 5150 rpm (85.8 Hz) or exactly 1/2 running speed. The suddenness with which the instability developed, the rapid fluctuations in the subharmonic component, and the fact that the subharmonic was at exactly 50% of running speed, led to the suspicion that a rub was involved. No explanation could be made for the fact that the whirl frequency was 5150 rpm (85.8 Hz) when the first critical was 6750 rpm (112.5 Hz). The problem was almost completely isolated to the topping turbine discharge end bearing. Some 1/2 running speed vibration was observed at the inlet end bearing, but only low levels (<5 μm or 0.2 mil) of this component were observed on other rotors in the train.

A shutdown and inspection of the bearing revealed that everything was normal including proper orientation, clearance, crush fit, alignment, etc. The bearing's inboard oil guard, however, was found to be heavily rubbed and was replaced with one that was checked to insure it had the proper clearances. This was the only change made before the machine was restarted. Since no subharmonic vibrations occurred during start-up and subsequent operation, it was concluded that the rubbing oil guard had been the source of the instability.

Another identical unit exhibited the same instability characteristics. This unit had developed a history of vibration problems during its eleven years of operation. One problem was occasional bouts of fluctuating high vibration at the topping turbine discharge end bearing. It had been found that using water on the bearing pedestals to change the alignment would sometimes reduce or eliminate the vibrations. An analysis was made during one vibration excursion to determine what effect the water was having. Figure 3 shows a spectra taken at 3:29 p.m. with water being used on one pedestal. There was a running speed vibration of 64 μm (2.5 mils) at 10,200 rpm (170 Hz) and a 1/2 running speed peak of 38 μm (1.5 mils). The 1/2 running speed peak was superimposed over the normal band of low level subharmonic noise and there was some electrical interference at 60 Hz, 120 Hz, 180 Hz, etc. The water was taken off the pedestal and figure 4, taken at 3:35 p.m., shows the resulting increase in the subharmonic. The water was then put on the other pedestal and the results are shown in figure 5, taken at 3:47 p.m. The instability had ceased suddenly and the running speed vibration slowly decreased from 64 μm (2.5 mils) to 38 μm (1.5 mils). During a subsequent outage, the bearings and seals at this point were inspected and found to be severely rubbed over a bottom quarter of their diameter. It was concluded that this was another rub which had produced both a large subsynchronous component as well as an increase in the unbalance from non-uniform heat input to the shaft. Once the rub was removed by changing the alignment, the subharmonic component ceased suddenly and the running speed vibration returned to normal slowly.

On a later occasion, this same turbine experienced high vibrations during a start-up. The machine was warmed up and brought up to minimum governor speed (9000 rpm) one evening and the load was to be progressively increased over the next 24 hours. However, at 11:00 the next morning while operating at 10,583 rpm, the vibration levels at the topping discharge end suddenly increased from about 50 μm (2 mils) to in excess of 127 μm (5 mils), the limit of the vibration monitors. By 12:00, an analysis had revealed (fig. 6) vibration levels in excess of 152 μm (6 mils) with a running speed component of approximately 91 μm (3.6 mils) and a fluctuating 1/2 running speed component of 114 μm (4.3 mils). Table 1 details the vibration levels during the excursion. A check of the gap voltages showed an increase of 0.9-1.1 volts had taken place. While some of this may have been due to the increase in the rotor orbit shifting the rotor's mean position, it still indicated that the rotor was located significantly lower in the bearing than normal. A slight increase in speed to 10,672 rpm failed to reduce the vibration level as it had with other rub induced instabilities. During the increase, the subharmonic tracked the increase in running speed so as to remain at exactly 1/2 running speed. The direction of whirl during the entire episode was forward. Water was used on the turbine casing supports in an attempt to lower the bearing relative to the shaft and hopefully clear the rub. When water was placed on one support pedestal, the vibrations increased to about 229 μm (9 mils). But when water was quickly placed on the other pedestal as well, the vibrations began to decrease. Within an hour, the gap voltages had returned to normal and, as shown in figure 7, the vibration levels were almost normal. During a subsequent inspection of the turbine, the bottom part of the bearings and seals at the discharge end showed severe rub damage and some breakage. Again, the instability was thought to be caused by a rub brought about by a severe misalignment during a thermally transient condition. There was still no explanation for the whirl frequency to be at 5250 rpm (87.5 Hz) when the first critical speed was 6750 rpm (112.5 Hz).

TURBINE ON CO_2 COMPRESSOR

Figure 8 shows a schematic of a CO_2 compressor train driven by a 8200 H.P., 9300 rpm, single extraction, condensing turbine. The 19,240 kg (3030 lb) rotor was supported on 15.2 cm (6 in) journals with sleeve type bearings. The first and second critical speed were approximately 4900 rpm and 10,150 rpm respectively as determined from proximity probe data during start-ups and overspeed trip tests. This unit had been in operation over nine years with no reported vibration problems except for occasional temporary bows during start-ups. The machine's susceptibility to thermal bows was due to short warm-up and start-up periods that were necessitated by plant design. Due to the trouble free operation of the unit, it had only been overhauled once five years after installation. The train was located outdoors with a protective roof over it.

A problem was first noted one February when the turbine thrust position monitor showed a temporary alert condition after a sudden change in wind direction. Future changes in wind direction caused the problem to recur with enough regularity that a portable blower was used to keep a constant air flow directed at the thrust bearing housing. This worked until March, when a rainstorm occurred which resulted in all the turbine vibration alarms going off as well as a thrust alert condition. The condition lasted for about one minute and subsided as quickly as it started.

During a brief outage in the summer, the bearings were inspected and found to be acceptable. But the bearing's oil seals were replaced at the governor end due to excessive leakage. These seals were noted to be tighter than normal with 50-75 μm (2-3 mils) diametral clearance. During the start-up, another vibration excursion occurred. The turbine had reached minimum governor speed and the speed was being increased slowly as needed. At 9250 rpm, the vibration levels suddenly increased from about 25-38 μm (1.0-1.5 mils) to about 76 μm (3 mils) and fluctuated rapidly between 25-38 μm (3 and 4 mils). The thrust monitor showed an alert condition also. The condition lasted two to three minutes and it was found that a slight increase in speed caused the vibrations to return to normal.

During a rainstorm several days later, another vibration excursion occurred lasting less than a minute with symptoms very similar to the previous excursion. The problem subsided when the speed was increased from approximately 9300 rpm to 9500 rpm. Several more vibration excursions occurred during rapid changes in weather conditions over the next few weeks.

Due to the transient nature of the problem, the plant connected their tape recorder to the proximity probe monitors and the operators were instructed to turn the recorder on during any sudden changes in weather that might precipitate another vibration excursion. Local thunderstorms resulted in three vibration excursions in the next 24 hours.

Typical frequency spectra from just prior to and just after the start of a vibration excursion are shown in figures 9 and 10 respectively while operating at 9360 rpm (156 Hz). The transition time between figure 9 and figure 10 was less than 0.1 second. The overall vibration level had been 33 μm (1.3 mils) with 15.7 μm (0.62 mils) at running speed and approximately 5 μm (0.2 mils) at 62 Hz or 2/5 of running speed. During the excursion, the vibration level increased to approximately 76 μm (3 mils) with the majority of this increase due to the 62 Hz component. The amplitude of the subsynchronous component averaged about 53 μm (2.1 mils), but was fluctuating wildly over a range as wide as 13 to 89 μm (0.5 to 3.5 mils) but typically from 38 to 64 μm (1.5 to 2.5 mils). The orbit developed into a "double orbit" from a relatively circular one and the whirl direction was noted to be forward. Various other frequencies noted during the excursion were found to be sum and difference frequencies of the two dominant frequencies of 62 Hz (subsynchronous vibration) and 156 Hz (running speed). These are shown in more detail in figure 11. A very narrow band analysis verified that the subsynchronous component was 2/5 of running speed to within 0.1 Hz.

The problem was thought to be due to a transient rub condition since:

1) the condition was closely related to weather conditions,
2) the wide fluctuations in the subharmonic component,
3) that slight increases in speed caused the instability to cease.

The unit had a great deal of hot piping attached, all of which was routed through the roof and exposed to the weather. Significant piping strains associated with transients was thought to have contributed to the units sensitivity to changes in weather. The fact that the 62 Hz component was present at a very low level prior to the excursion was not thought to be very significant since a review of the vibration records showed that it was almost always present with levels between about 1 to 5 μm (0.05 to 0.2 mils). Also, as stated earlier, it is not uncommon for high speed units to display a low level subharmonic even during normal operation. The reason the whirl occurred at 62 Hz (3720 rpm) when the first critical was known to be at approximately 81.7 Hz (4900 rpm) remains unexplained.

The possibility that the instability was an oil whirl condition precipitated by a change in alignment, as has been known to occur, was considered but discarded for several reasons. First, changes in alignment would probably not cause the almost instantaneous change in subharmonic vibration level. Also, while subsynchronous components due to oil whirl instabilities can fluctuate, they have not previously been observed to fluctuate so rapidly and over so wide a range. And finally, the fact that slight increases in speed were sufficient to eliminate the instability was not at all a characteristic of oil whirl. The newly installed oil seals were obviously suspected as the source of the rub. However, due to the violence of the instability and the fact that these seals had probably rubbed at other times in the unit's nine year history, without causing an instability, it was decided that a full dismantle inspection was in order.

This dismantle inspection revealed that:

1) Standing at the turbine governor end, the governor end of the rotor was misaligned 510 μm (20 mils) low and 254 μm (10 mils) to the right and the coupling end was 101 μm (4 mils) to the right.

2) A sliding key at the turbine governor end meant to maintain axial alignment while sliding to accommodate thermal growth, had frozen preventing the case from moving axially.
3) The bearing oil seals were lightly rubbed.
4) The rotor was heavily rubbed at the center by the interstage labyrinths.

It was concluded that the frozen footing was primarily responsible for the rubs. The case was apparently bowing in the center quite severely during sharp thermal transients as it tried to accommodate the change in casing growth and pipe strain. This also was thought to be the reason for the rub during start-up since the case again could not shift to accommodate thermal growth. Although other rubs had probably occurred at some time in the machine's history, there had been none this severe as indicated by the depth of scoring on the rotor. This explained why this problem had not occurred before.

The rotor was realigned and the frozen footing was freed. The machine has since operated for a year and a half with no further problems except for the usual problem of thermal bows during startups.

CONCLUSION

Some cases of rub induced instabilities have been described. These have been noted to appear and cease very suddenly, can sometimes be controlled during operation by changing the alignment, and have sometimes been eliminated with minor increases in speed. On one occasion the whirl frequency was noted to track an increase in rotor speed so as to remain at 1/2 running speed. The whirl frequencies are usually at 50 percent of running speed and the whirl direction has often been forward. However, whirl frequencies have been observed at other fractions of running speed. Also, whirl frequencies have been observed at speeds significantly lower than their first critical speed. Some rubs have produced sum and difference frequencies based on the whirl frequency and the running speed frequency.

REFERENCES

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2. Ehrich, F. F.: The Dynamic Stability of Rotor/Stator Radial Rubs in Rotating Machinery. J. of Eng. Indus., Nov. 1969, pp. 1025-1028.

3. Bentley, D. E.: Forced Subrotative Speed Dynamic Action of Rotating Machinery, A.S.M.E. Paper No. 74-PET-16.

4. Childs, D. W.: Rub-Induced Parametric Excitation in Rotors. A.S.M.E. Paper No. 78-WA/DE-14.

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TABLE. I - SUMMARY OF VIBRATION LEVELS ($mils_{p-p}$)

TIME	Point #3						Point #4					
	VERTICAL			HORIZONTAL			VERTICAL			HORIZONTAL		
	½X	1X	Total	½X	1X	Total	½X	1X	Total	½X	1X	Total
10:45pm	—	0.5	0.6	—	0.35	0.4	—	0.8	0.95	—	0.6	0.75
11:43am	0.5	0.6	1.7	0.8	0.2	1.3	4.3	2.3	75	4.3	3.6	75
12:02pm							2.5	3.8	5.8	4.8	3.6	6.0
12:10pm							2.5	3.5	5.8	4.5	3.5	6.0
12:13pm							4.5		8.5	4.5		8.0
12:17pm							3.0		9.0	5.6		8.5
12:19pm									8.3			8.2
1:45pm	0.17	0.25	0.48	0.45	0.8	1.05	0.7	1.6	1.79	0.1	0.4	0.6

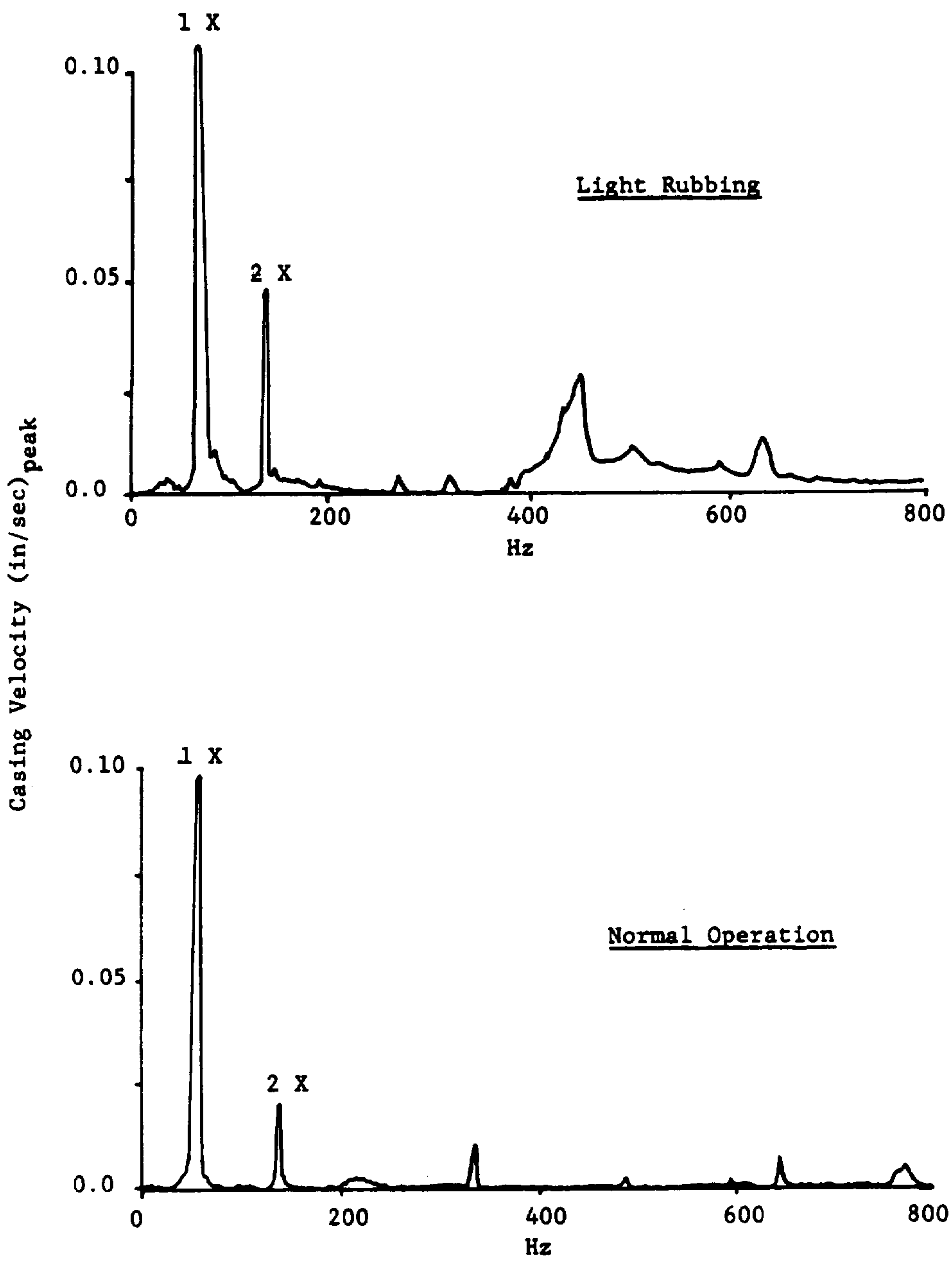

Figure 1. - Turbine case vibration on a 5000 kw turbine-generator.

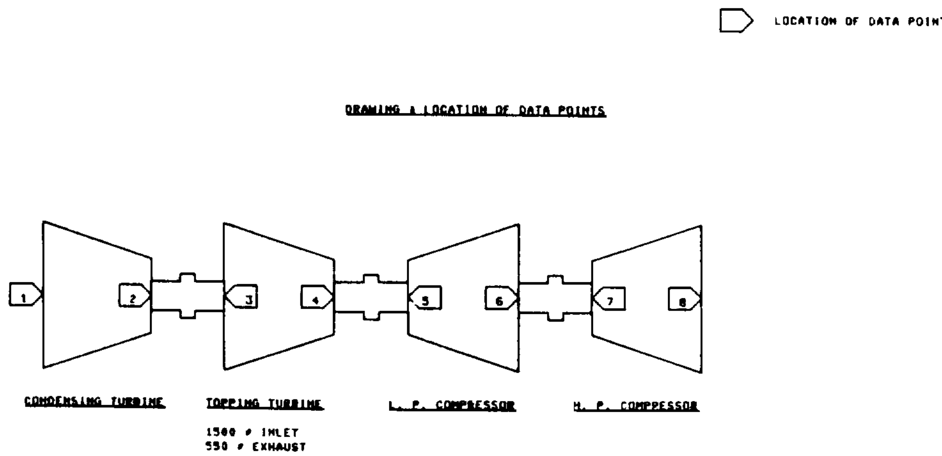

Figure 2. - Schematic of the Syn Gas Compressor.

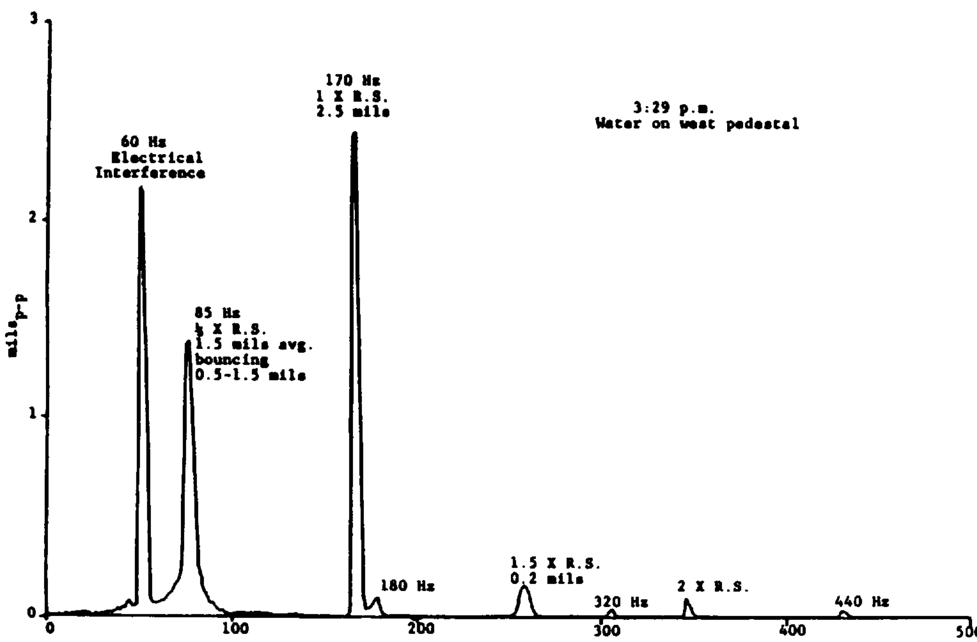

Figure 3. - Topping turbine discharge end vibration - with water on west pedestal.

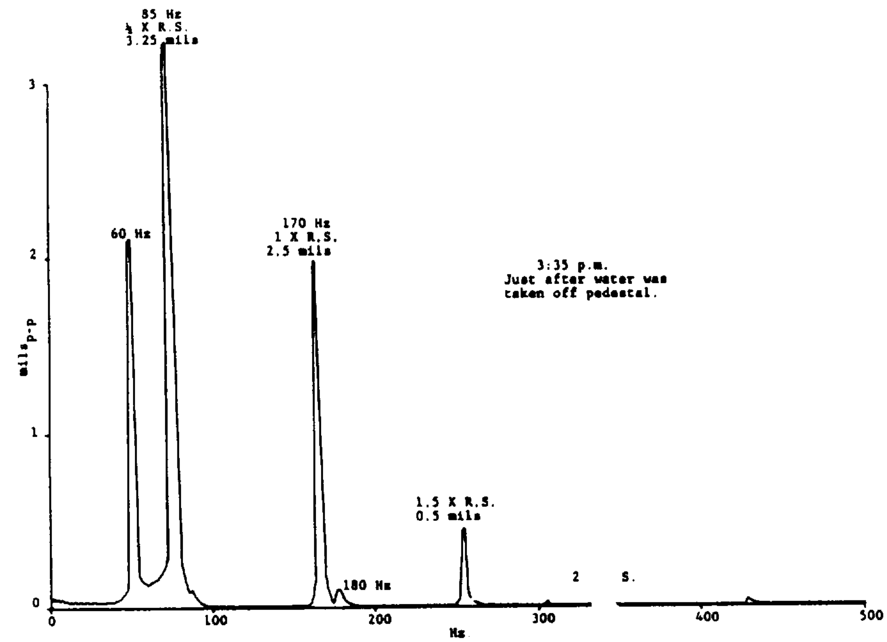

Figure 4. - Topping turbine discharge end vibration - water taken off pedestal.

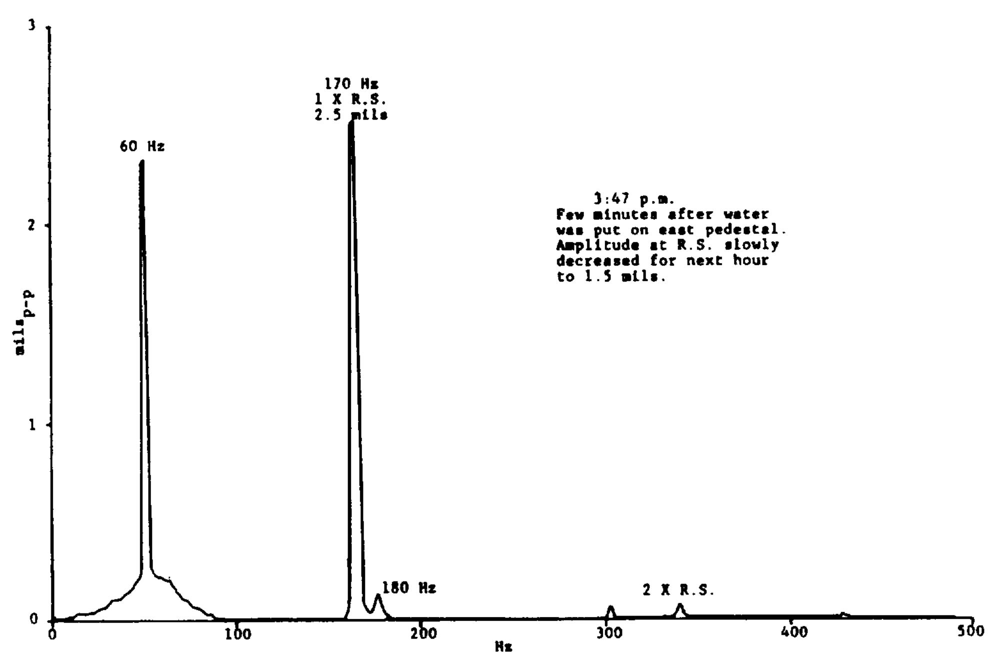

Figure 5. - Topping turbine discharge end vibration - with water on east pedestal.

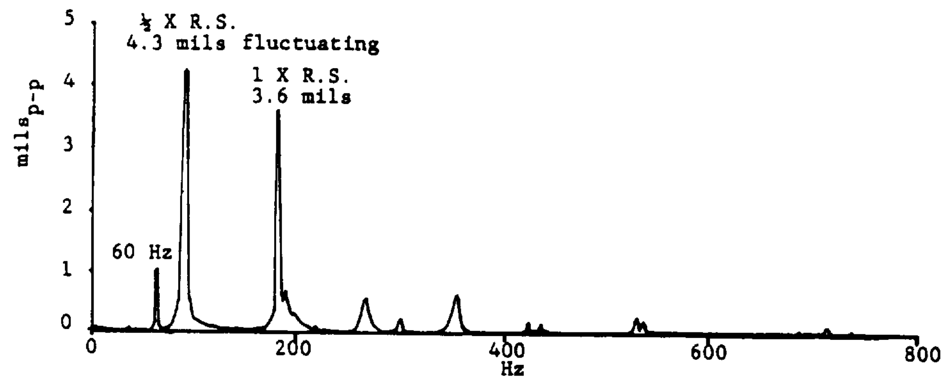

Figure 6. - During vibration excursion.

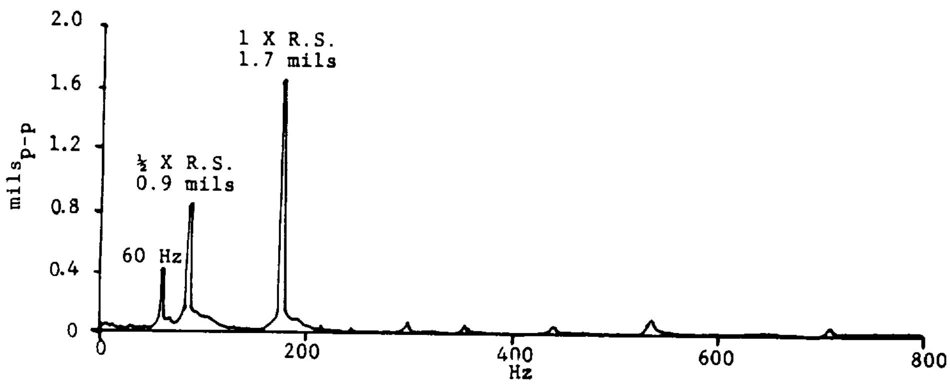

Figure 7. - At conclusion of vibration excursion.

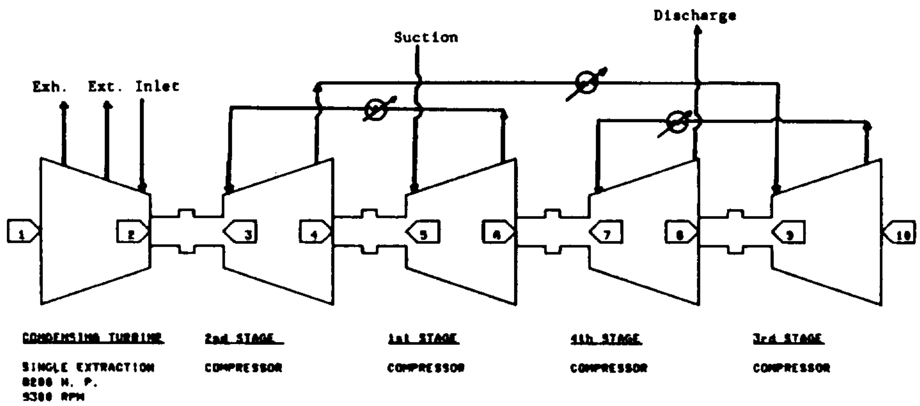

Figure 8. - Schematic of the CO_2 Compressor.

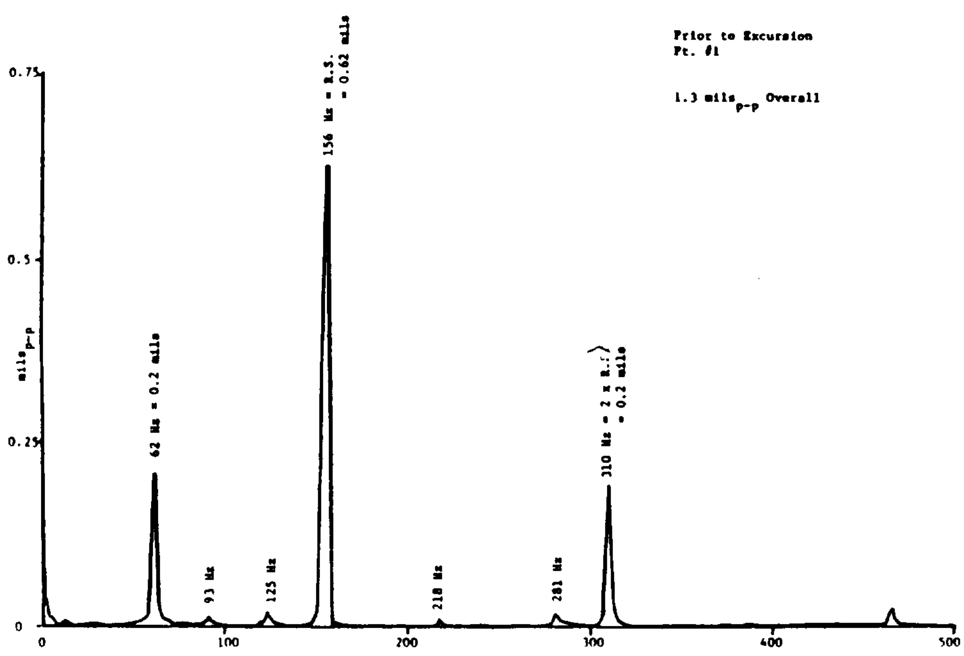

Figure 9. - Turbine outboard bearing - just prior to excursion.

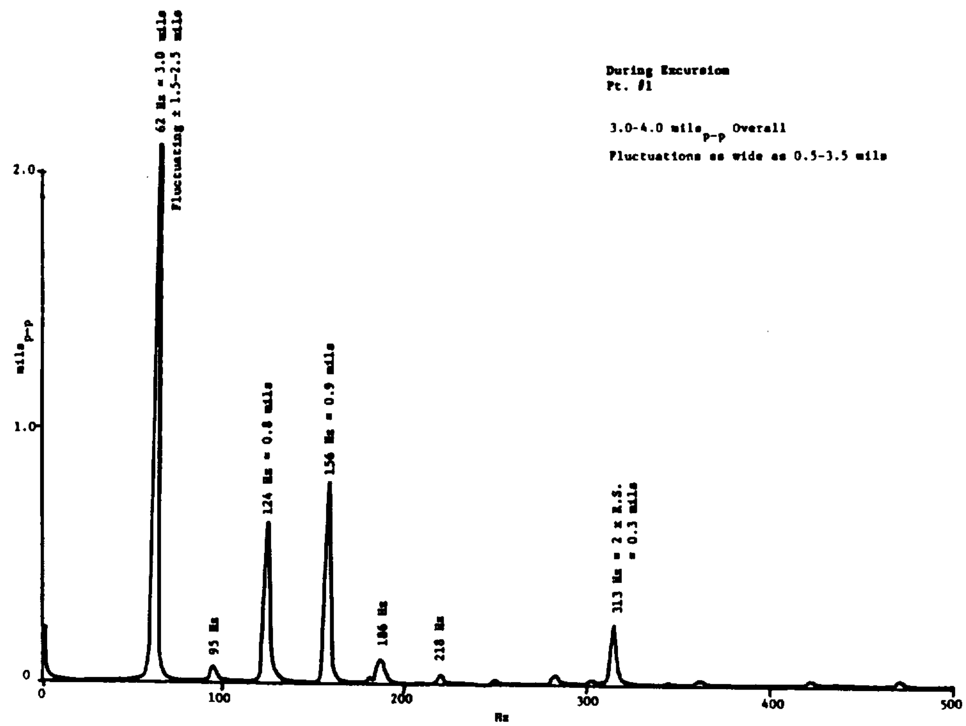

Figure 10. - Turbine outboard bearing - during vibration excursion.

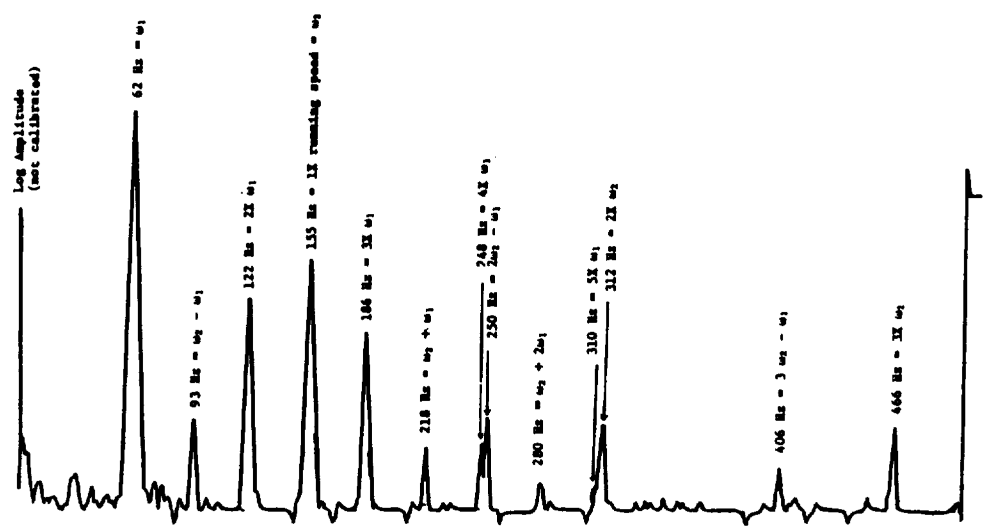

Figure 11. - Turbine vibration - during excursion: sum and difference frequencies.