

**EXPERIMENTAL ON-STREAM ELIMINATION OF RESONANT
WHIRL IN A LARGE CENTRIFUGAL COMPRESSOR**

**G. I. Bhat and R. G. Eierman
Exxon Chemical Company
Baytown, Texas 77520**

In October 1982, a severe resonant whirl condition was experienced when a multi-stage centrifugal compressor was first operated at higher than originally anticipated speeds and loads. Diagnosis of this condition was made easy by a large-scale computerized Machinery Condition Monitoring System ("MACMOS"). This computerized system was immediately able to verify that the predominant subsynchronous whirl frequency locked in on the first resonant frequency of the compressor rotor and did not vary with compressor speed.

Compressor stability calculations showed the rotor system had excessive bearing stiffness as well as inadequate effective damping. An optimum bearing design was developed to minimize the unbalance response and to maximize the stability threshold.

The above experience is not unusual and parallels that of many process plants using large centrifugal gas compression machinery. Of interest, however, is the approach taken by the plant to find a temporary remedy. The effective compressor bearing loading and effective clearance characteristics were modified with the machine continuing its process operation at normal load and speed. This approach involved the controlled application of heat to the compressor support legs while closely monitoring machine behavior. The experiment established the feasibility of extending the onset of rotor instability in the event that plant operations would call for higher speeds before the optimized bearings became available.

DISCUSSION OF PROBLEM UNIT

Description of Unit

The compressor is a two stage eight impeller horizontally split machine driven by a steam turbine. Oil seals are used at both ends and labyrinth seals are used at the center. The compressor runs at 175 psig inlet and 575 psig discharge. The rotor is supported by five shoe tilting pad load between pad bearings with 0.5 preload. The rotor has an unusually large ratio of bearing span to shaft diameter (99 in/5 in). The compressor train is safeguarded against high vibration with dual voting logic vibration trip circuits which shut the compressor down if vibration amplitude exceeds 4.5 mls at any time (Figure 1).

A Kingsbury thrust bearing is used to absorb the thrust. The compressor is driven through a Bendix type diaphragm coupling incorporating a torque meter and

hot alignment tubes. The compressor and the driver are supported on a solid concrete foundation.

Problem

Prior to the first half of 1982 the compressor was running with satisfactory performance at base capacity operating conditions of 6000 rpm as the maximum speed (Figure 2). The rotor was subsequently modified for a different mole weight gas with a revised maximum speed of 6500 rpm. The manufacturers' lateral critical speed and rotor sensitivity studies did not reveal any potential problems (Figure 3). The rotor was high speed balanced at 6700 rpm with an exceptional balance quality of 2 oz-in. residual unbalance (Figure 4).

However, when an attempt was made to increase the speed above 6000 rpm with an increase in load, a severe change in shaft vibration was noticed (amplitude increase from less than 1 mil to 3.6 mils). Below 6000 rpm the compressor was running at less than 1 mil vibration. Analysis of vibration spectra obtained from a computerized Machinery Condition Monitoring System ("MACMOS") revealed that the high vibration amplitudes occurred at a predominant frequency of 46.5 Hz (2760 cpm), which coincides with the first resonant frequency (critical speed) of the compressor rotor (Figure 5). Varying the rate of buffer gas injection to the seals produced significant changes in the vibration amplitude suggesting that this so called resonant whirl condition was aerodynamically induced. Aerodynamic impulses created by gas exiting impeller vanes (load dependent) provide excitation force which, for systems with inadequate damping, can amplify the vibration behavior at the first critical speed of the rotor. It is also believed that the low residual unbalance in this machine allowed a forcing impulse other than unbalance to predominate and thus excite the first resonant mode of the rotor.

A detailed analysis of the rotor-bearing system indicated excessive bearing stiffness as well as inadequate effective damping. A computerized study revealed that a four pad load between pad bearing is an optimum design for the machine which would maximize the stability threshold speed while yielding acceptable unbalance response (Table 1).

However, since the new four pad bearings were not readily available, the approach taken by the plant to find a temporary remedy with the machine continuing its process operation at normal load and speed, is innovative and worth sharing.

EXPERIMENTAL PROCEDURE

The goal was to establish the feasibility of increasing the rotor stability threshold, as plant operations would require increasing the rotor speed before the optimized bearings became available.

It was anticipated that the goal could be achieved by modifying effective compressor bearing loading and effective clearance characteristics without shutting the compressor down.

For the purpose of this experiment, the four compressor support legs were provided with induction heating elements which were wrapped with insulating material. In addition, dial indicators were set up to monitor vertical and horizontal movement of the compressor casing. Power supply and temperature control

responsibility were assigned to the truck-mounted laboratory, which was positioned near the compressor platform.

During the test relevant process parameters were held constant and were continuously monitored by MACMOS on a second-by-second basis and then converted to six-minute averages. The vibration behavior of the machine was, of course, also logged by an automated computer and a FM tape recorder during the entire test.

The actual test was conducted by step-wise heating of compressor legs in the following sequence: (1) inboard legs only, (2) outboard legs only, (3) all four legs, and (4) two diagonally opposite legs at a time. Temperatures started at 150°F and were raised at 50°F increments until final support leg temperatures of approximately 400°F were reached. The vibration amplitude behavior was continually monitored on platform-mounted vibration monitors and readings confirmed by simultaneously observing the computer-generated log. The taped vibration signals were subsequently analyzed and their spectra plotted as shown in Figure 6.

The onset of high vibration was noted on MACMOS printouts as typically shown in Figure 7. Whenever a preset limit is exceeded for a given parameter, an alarm occurs and data logging is initiated. Relevant data can then be retrieved and examined.

Figure 7 shows excessive average vibration of the compressor outboard bearing as monitored by an eddy current probe. Alarm-initiated data logging was actually triggered before 10:27:39 by vibration spikes which must have exceeded the preset limit of 3.2 mils. Note that outboard eddy current probe readings are represented by the numeral "8" whose second-by-second values are hand-connected for easier viewing in Figure 8.

The experiment demonstrated that low-vibration operation at train speeds of approximately 6,360 RPM was feasible with compressor outboard support legs heated to approximately 375°F. At these conditions, the compressor outboard end had grown 32 mils in the vertical, and 4 mils in the horizontal direction. The subsynchronous (2,760 cpm) vibration component was smaller than the once-per-revolution component at this speed. This is graphically illustrated in Figure 9, which shows compressor outboard spectra obtained under similar loads and speeds, with dissimilar support leg temperatures. At ambient temperatures, the subsynchronous frequency registers an uncomfortable 2.8 mils on the vertical eddy current probe. When the compressor outboard legs were heated to approximately 375°F, the once-per-revolution and subsynchronous vibration components dropped below 0.35 mils.

CONCLUSION

Severe aerodynamically induced subsynchronous vibration problems developed when the normal operating speed of a large centrifugal compressor was increased. An experiment was carried out to extend the onset of rotor instability to higher speeds on-line, without changing the basic rotor bearing system characteristics. The test results indicated that the effective compressor bearing support characteristics could be modified and the stability threshold be increased to an acceptable level.

This experiment has demonstrated that with proper instrumentation and monitoring, it is feasible to extend the stability threshold speed of centrifugal compressor, without requiring a shutdown of the equipment.

ACKNOWLEDGEMENT

The authors acknowledge Mr. H. P. Bloch, Mr. H. G. Elliott, Mr. D. G. Stroud, and Mr. R. H. Schmaus for their support during this test and subsequent data reduction.

REFERENCES

- (1) "Four Pad Tilting Pad Bearing Design and Application for Multistage Axial Compressors", J. C. Nicholas and R. G. Kirk, ASME Paper, 81-LUB-12, 1981.
- (2) "Selection and Design of Tilting Pad and Fixed Lobe Journal Bearings for Optimum Turborotor Dynamics", J. C. Nicholas and R. G. Kirk, Proceedings of the Eighth Turbomachinery Symposium, Texas A&M University, ed, P. E. Jenkins, November, 1979.
- (3) "The Influence of Tilting Pad Bearing Characteristics on the Stability of High Speed Rotor Bearing Systems:", J. C. Nicholas, E. J. Gunter, and L. E. Barnett, Topics in Fluid Film Bearing and Rotor Bearing System Design and Optimization, an ASME special publication, 1978, pp. 55-78.
- (4) "Optimum Bearing and Support Damping for Unbalance Response and Stability of Rotating Machinery", L. E. Barrett, E. J. Gunter, P. E. Allaire, ASME Paper No. 77-GT-27.

TABLE I. - SUMMARY OF POTENTIAL ALTERNATE BEARING PERFORMANCE

Bearing Case	\bar{K} (1)	$\frac{\epsilon}{C_{OPT}}$ (4)	AMP (2)	δ (2)	ω (2) CPM
5 SLBP L/D = .3/PLF = .5	11.4	0.24	(-93.6)	-0.034	2,451
5 SLBP L/D = .43/PLF = .0	8.10	0.45	22.0	0.143	2,399
5 SLBP (55 percent offset)	8.97	0.40	53.7	0.058	2,431
3 SLOP	11.8	0.69	15.1	0.208	2,403
5 SLBP	10.96	0.70	25.7	0.122	2,415
5 SLOP	15.5	0.63	24.6	0.128	2,439
4 SLBP	6.11	1.12	8.7	0.362	2,421
7 SLOP TRESS-32	10.96	0.86	12.6	0.251	2,415
7 SLOP TRESS-68	17.24	0.59	31.5	0.100	2,446

Ref: (1) "Optimum Bearing and Support Damping for Unbalance Response and Stability of Rotating Machinery", L. E. Barrett, E. J. Gunter, P. E. Allaire, ASME Paper No. 77-GT-27.

(2) "Stability and Damped Critical Speeds of a Flexible Rotor in Fluid-Film Bearings", J. W. Lund, ASME Paper No. 73-DET-103.

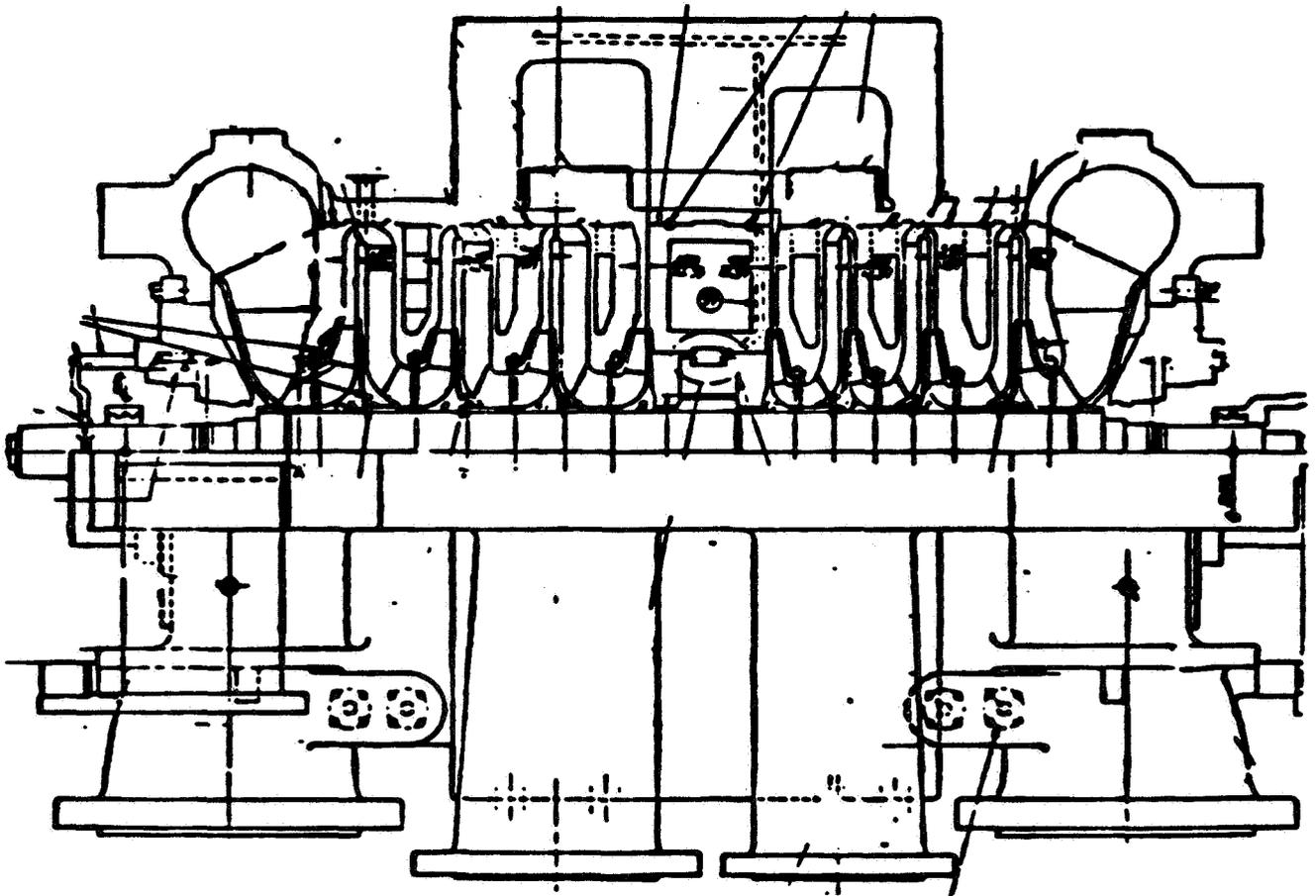
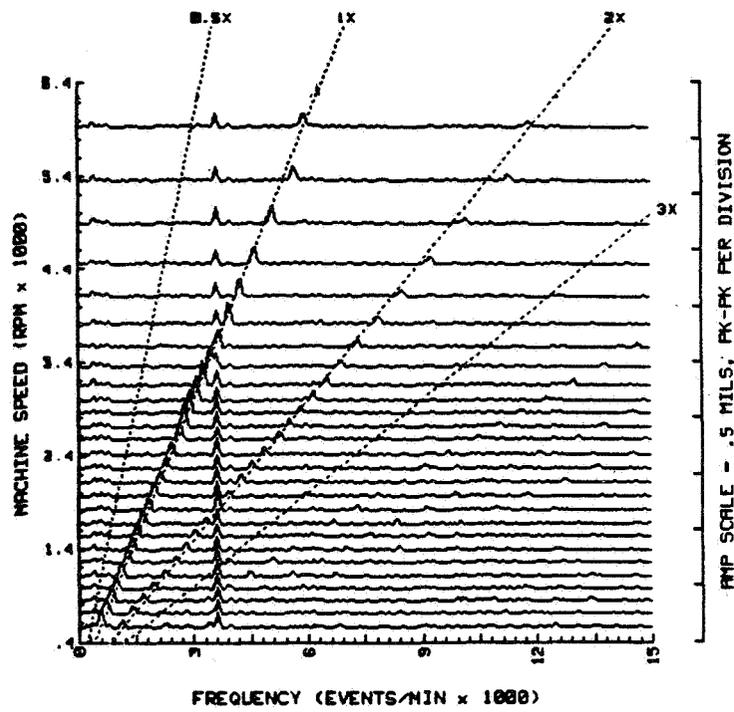


Figure 1. - Compressor X-section.



(a) Cascade plot.

Figure 2. - Base operating conditions.

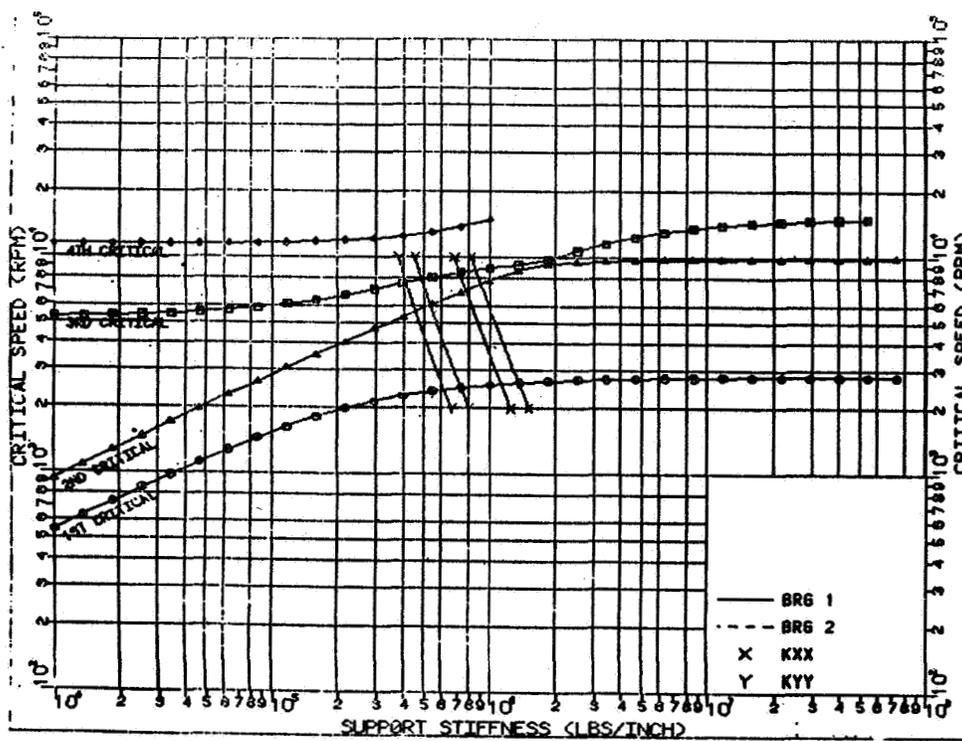


Figure 3. - Critical speed map.

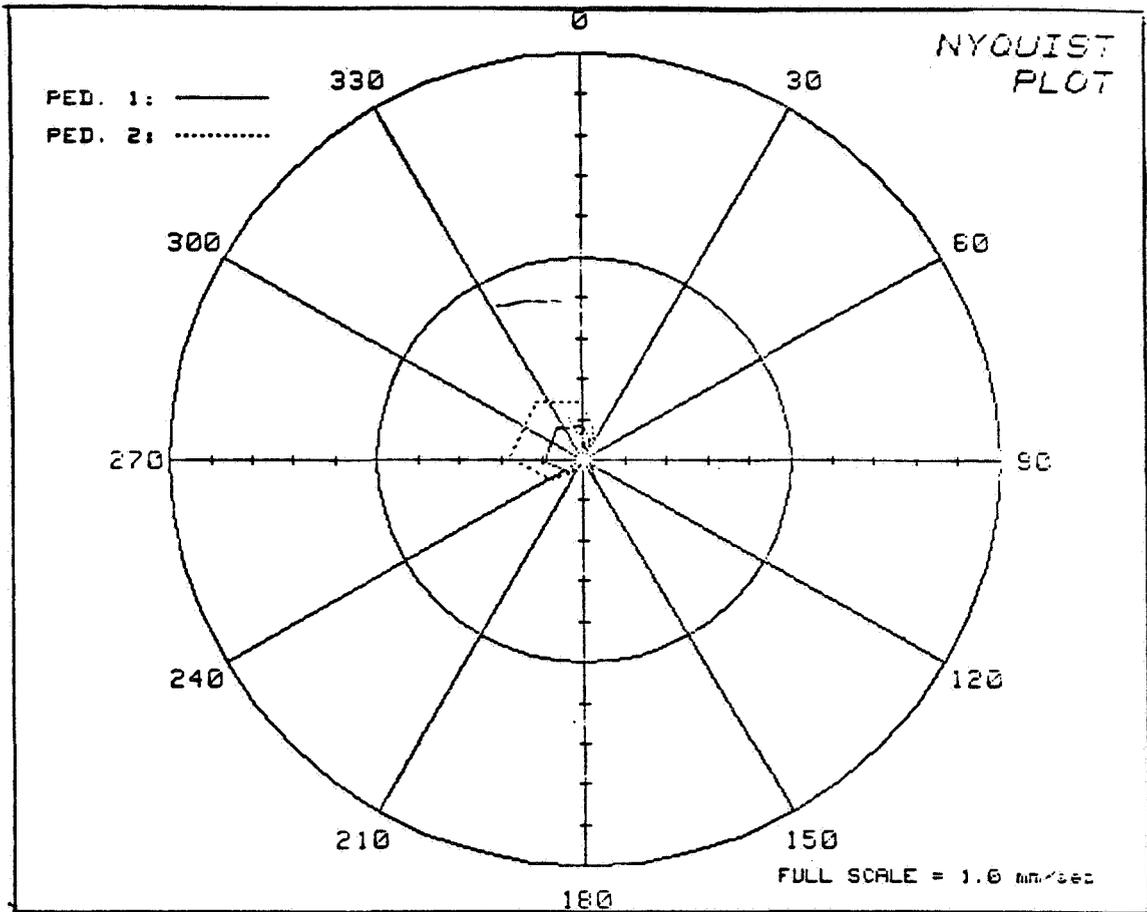
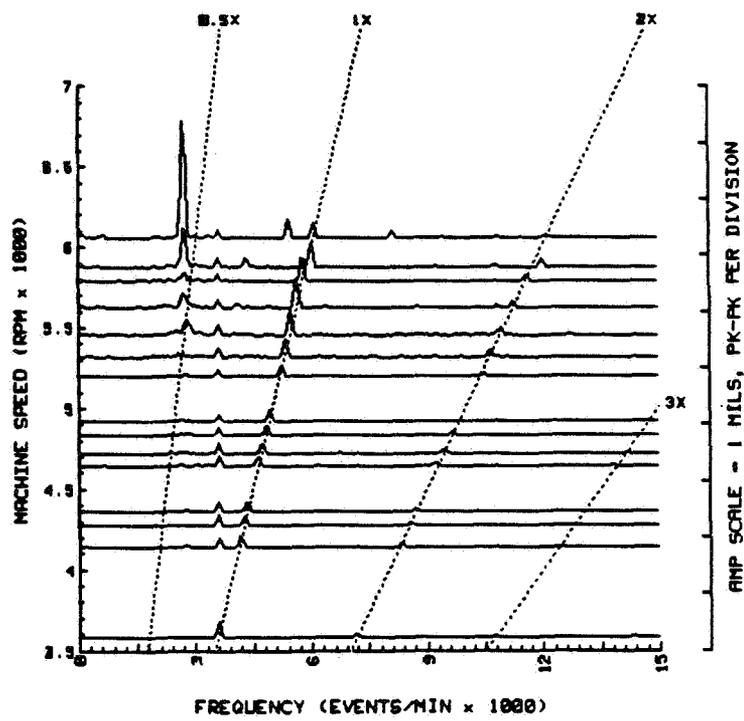


Figure 4. - High speed balance plot.



(a) Cascade plot.

Figure 5. - Uprated operating conditions.

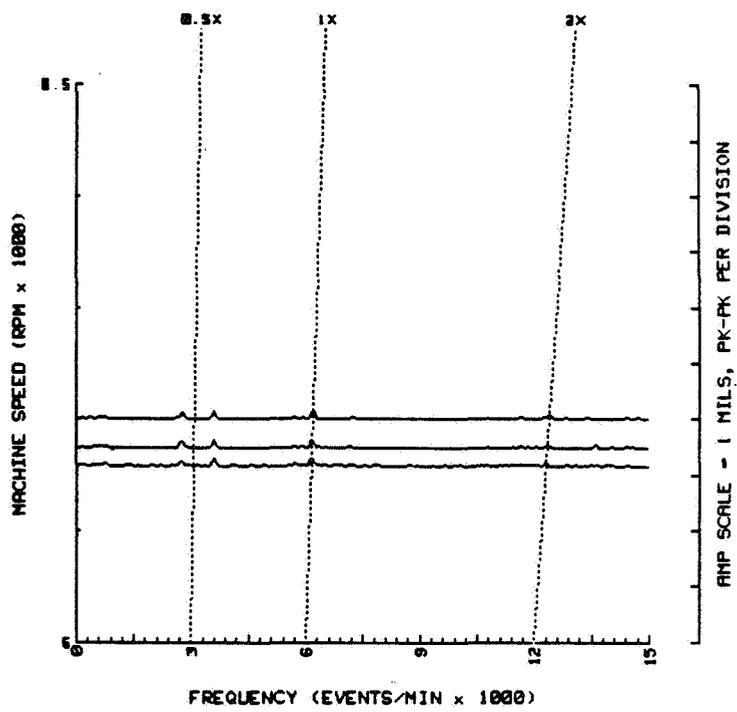


Figure 6. - Uprated operating conditions with heated legs: Cascade plot.

Time	Machine Speed (RPM)		Inboard Bearing Vib., Mils	Inboard Bearing (Radial) T, °F	Accelerometer (unbrated) Mils	Inboard Bearing (Radial) T, °F	Inboard Bearing (Radial) T, °F	Outboard Bearing		Rotor Thrust Position	Rotor Bearing Temp., °F
	1	2						Vib., Mils	G		
102829	6213.9	3767.8	1.3740	188.04	1.1424	188.54	188.54	1.8928	0	12.378	174.08
102834	6217.8	3773.7	1.4336	188.43	1.2288	188.43	188.43	1.9424	0.0960	12.324	174.43
102839	6213.9	3767.8	1.3440	188.26	1.0944	188.26	188.26	2.1424	0	12.301	174.24
102844	6209.9	3763.9	1.4360	188.35	1.0752	188.35	188.35	1.8116	0	12.278	174.14
102849	6213.9	3771.8	1.7312	188.24	1.1136	188.24	188.24	2.4748	0.0096	12.326	174.09
102854	6217.8	3767.8	1.8112	188.33	1.1328	188.33	188.33	2.3280	0	12.352	174.24
102859	6217.8	3771.8	1.5280	188.43	1.1424	188.43	188.43	1.5088	0	12.382	174.14
102864	6209.9	3769.8	1.4560	188.26	1.1136	188.26	188.26	2.0144	0.0288	12.326	174.14
102869	6213.9	3773.7	1.4096	188.35	1.1712	188.35	188.35	1.9120	0	12.352	174.24
102874	6217.8	3771.8	1.3936	188.26	1.1136	188.26	188.26	1.7984	0.0192	12.382	174.24
102879	6202.0	3769.8	1.3088	188.26	1.1040	188.26	188.26	1.7152	0.0288	12.301	174.08
102884	6204.0	3771.8	1.3104	188.33	1.20	188.33	188.33	1.7148	0.0096	12.326	174.08
102889	6209.9	3767.8	1.3136	188.43	1.2384	188.43	188.43	1.7136	0.0096	12.352	174.08
102894	6213.9	3771.8	1.3664	188.54	1.2384	188.54	188.54	1.8272	0	12.382	174.14
102899	6217.8	3769.8	1.4320	188.43	1.0944	188.43	188.43	1.960	0	12.301	174.14
102904	6204.0	3773.7	1.5248	188.26	1.0752	188.26	188.26	2.1376	0.0192	12.278	174.14
102909	6213.9	3773.7	1.60	188.43	1.0176	188.43	188.43	2.2880	0	12.326	174.24
102914	6209.9	3767.8	1.60	188.54	1.0272	188.54	188.54	2.1328	0.0096	12.352	174.24
102919	6204.0	3767.8	1.5332	188.43	1.0848	188.43	188.43	1.8864	0	12.382	174.24
102924	6209.9	3771.8	1.3784	188.74	1.1136	188.74	188.74	1.5948	0	12.301	174.24
102929	6204.0	3771.8	1.2896	188.43	1.1136	188.43	188.43	1.6312	0	12.326	174.24
102934	6209.9	3773.7	1.2472	188.54	1.0848	188.54	188.54	1.8016	0	12.352	174.24
102939	6213.9	3773.7	1.3008	188.64	1.1712	188.64	188.64	1.8144	0.0388	12.382	174.24
102944	6213.9	3771.8	1.3496	188.43	1.2672	188.43	188.43	1.6656	0	12.301	174.24
102949	6208.0	3767.8	1.2912	188.64	1.1520	188.64	188.64	1.5392	0.0192	12.326	174.24
102954	6213.9	3773.7	1.240	188.43	1.2096	188.43	188.43	1.5536	0.0288	12.352	174.24
102959	6209.9	3773.7	1.2464	188.54	1.2576	188.54	188.54	1.5872	0	12.382	174.43
102964	6213.9	3769.8	1.2976	188.64	1.1424	188.64	188.64	1.4224	0.0096	12.326	174.34
102969	6213.9	3767.8	1.2752	188.64	1.1136	188.64	188.64	1.7056	0	12.301	174.34
102974	6209.9	3767.8	1.3152	188.54	1.1328	188.54	188.54	1.6432	0.0096	12.326	174.43
102979	6217.8	3771.8	1.2832	188.43	1.1232	188.43	188.43	1.5472	0	12.301	174.34
102984	6213.9	3773.7	1.2384	188.83	1.1808	188.83	188.83	1.5248	0.0096	12.352	174.43
102989	6221.7	3767.8	1.2336	188.54	1.2096	188.54	188.54	1.5480	0	12.301	174.24
102994	6209.9	3769.8	1.2376	188.64	1.2288	188.64	188.64	1.7184	0.0192	12.326	174.43
102999	6217.8	3769.8	1.3248	188.43	1.2384	188.43	188.43	1.6864	0.0096	12.352	174.43
103004	6213.9	3771.8	1.3040	188.54	1.1904	188.54	188.54	1.6496	0.0096	12.378	174.43
103009	6217.8	3765.8	1.240	188.64	1.1424	188.64	188.64	1.5296	0	12.352	174.53
103014	6213.9	3771.8	1.1984	188.43	1.1232	188.43	188.43	1.4416	0	12.301	174.43
103019	6209.9	3765.8	1.1920	188.54	1.20	188.54	188.54	1.4224	0	12.326	174.34
103024	6213.9	3773.7	1.1888	188.64	1.2096	188.64	188.64	1.4112	0.0192	12.352	174.34
103029	6213.9	3773.7	1.1888	188.43	1.2096	188.43	188.43	1.4624	0	12.301	174.43
103034	6209.9	3769.8	1.2064	188.54	1.2096	188.54	188.54	1.5984	0.0192	12.352	174.43
103039	6204.0	3769.8	1.2640	188.64	1.20	188.64	188.64	1.5984	0	12.301	174.43
103044	6213.9	3773.7	1.2720	188.43	1.1424	188.43	188.43	1.5472	0	12.352	174.43
103049	6217.8	3771.8	1.240	188.54	1.1520	188.54	188.54	1.4624	0.0096	12.301	174.43

Figure 7. - "Alarm-initiated data log" in 10:28:08 to 10:28:10, and 10:28:18 to 10:28:23 time frame. Vibration exceeded 2 mils.

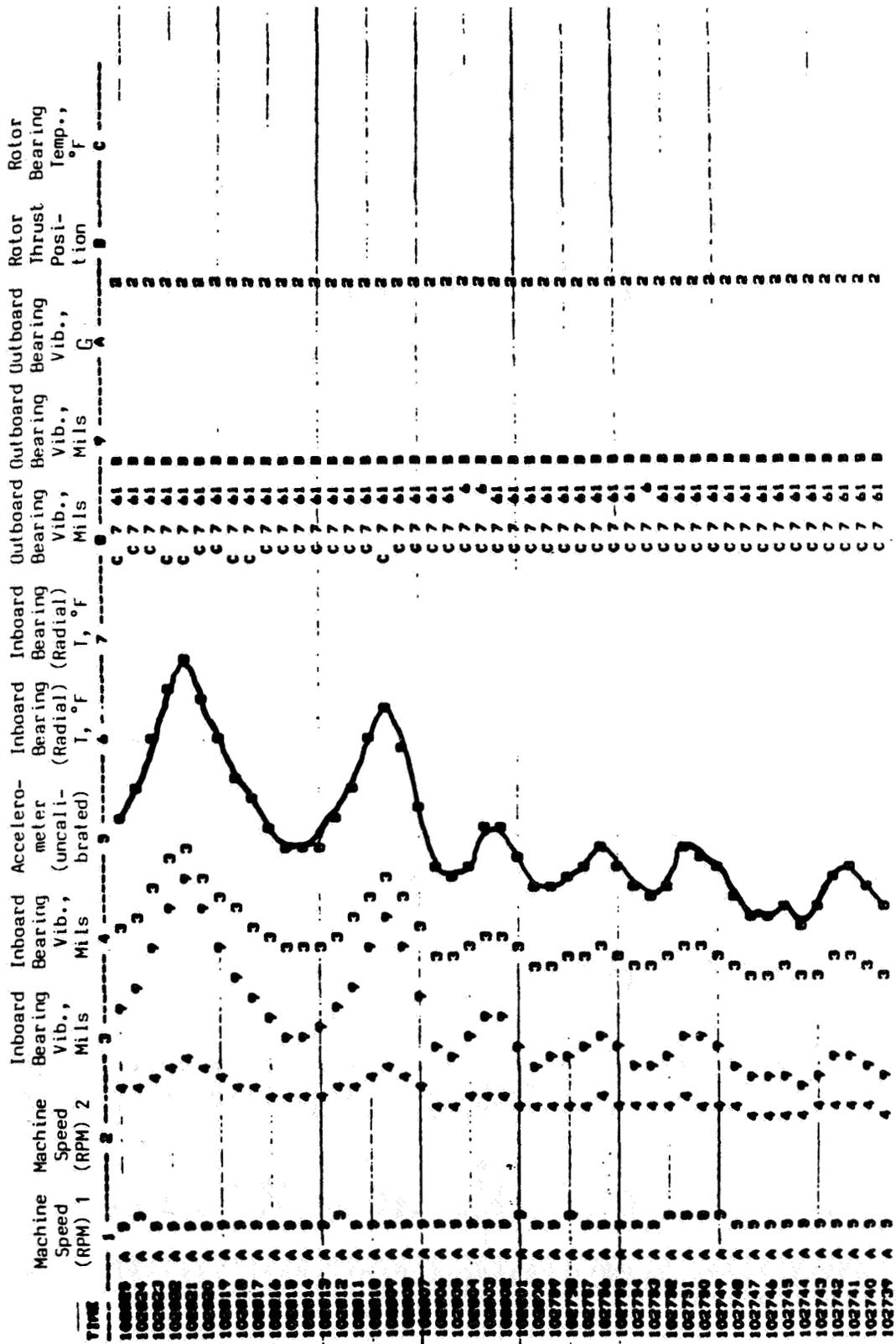


Figure 8. - Outboard bearing vibration in X-direction for 10:27:39 to 10:28:25 time frame.

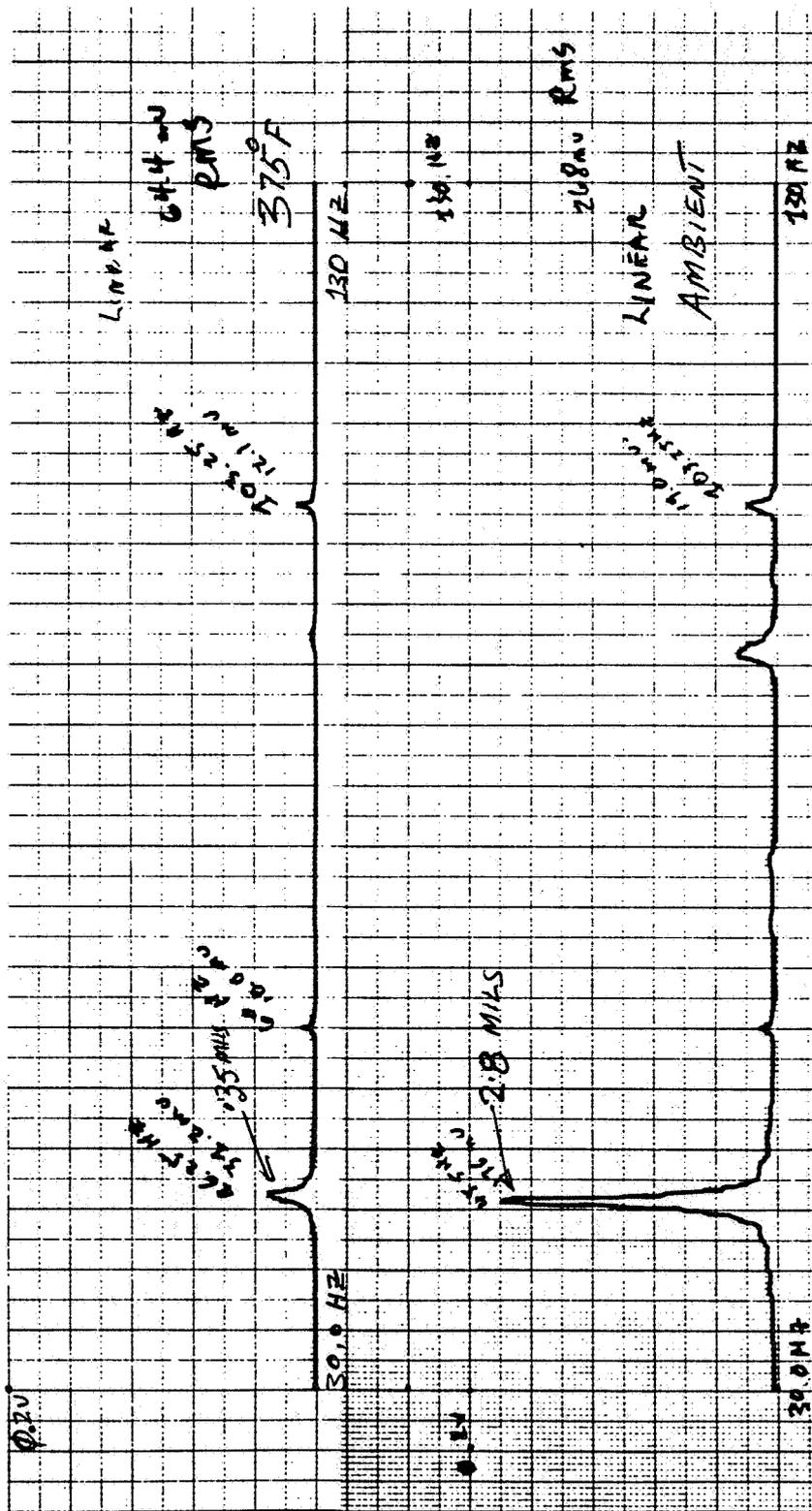


Figure 9. - Compressor outboard vertical vibration at near identical load with outboard support legs at ambient versus approx. 375° temperature.