

AERODYNAMIC INSTABILITY: A CASE HISTORY

Robert C. Eisenmann
Machinery Diagnostics, Inc.
Minden, Nevada 89423

The identification, diagnosis, and final correction of complex machinery malfunctions typically require the correlation of many parameters such as mechanical construction, process influence, maintenance history, and vibration response characteristics. This paper reviews the progression of field testing, diagnosis, and final correction of a specific machinery instability problem. The case history presented in this paper addresses a unique low-frequency instability problem on a high-pressure barrel compressor. The malfunction was eventually diagnosed as a fluidic mechanism that manifested as an aerodynamic disturbance to the rotor assembly.

MECHANICAL CONFIGURATION

The machinery train consisted of an extraction steam turbine driving two barrel compressors. From a process standpoint the first two stages of compression were contained within the low-stage compressor. The third and fourth process stages were handled within the high-stage compressor. Each compression stage was equipped with interstage cooling and knockout drums. The initial design criteria called for a process stream rich in carbon dioxide. The actual composition of the gas consisted primarily of carbon dioxide (85.6 to 90.6 mol %) and water vapor (5 to 10 mol %).

The configuration of the high-stage rotor is presented in figure 1. The 10 wheels (5 per compression stage) were located back to back to minimize thrust load.

The compressor was equipped with 5-pad, tilt pad radial bearings and a traditional thrust arrangement. The unit exhibited an average first critical speed of 5100 rpm. Shop acceptance tests revealed normal behavior for each of the three machines. Hence up until the time of actual field commissioning there was no indication of a potential instability problem.

INITIAL FIELD STARTUP CHARACTERISTICS

During initial field startup with process gas the high-stage compressor wrecked. The general consensus of opinion was that the system was underdamped. To provide additional rotor damping, the original equipment manufacturer designed and retrofitted a set of squeeze-film damper bearings. This change proved successful, and the unit was easily brought up to normal operating speeds.

It appeared that most of the problems were over and the remainder of the plant startup could commence. However, when loading of the machine was attempted, shaft vibration levels increased dramatically. At discharge pressures of about 1500 psig a significant low-frequency excitation appeared. As discharge pressures were increased, the magnitude of the low-frequency vibration would likewise increase.

00108-0211
Typical shaft response for this unit is shown in figure 2. Vibration amplitudes at running speed were approximately 1 mil, and most of the motion occurred at a frequency of 552 cpm. This motion was strongest on the coupling end bearing and somewhat lower on the outboard bearing. The shaft orbit was nearly circular with predominant motion at 552 cpm and a series of inside loops at running frequency. Under this set of operating conditions the rotor made almost 17 revolutions for every 1 cycle of the low-frequency excitation. This low-frequency vibration was audible on the compressor deck and the associated piping. In fact, the machine sounded more like a reciprocating engine than a centrifugal compressor. One of the workers claimed that it sounded like there was an "armadillo" in the piping.

INITIAL FIELD TEST RESULTS

A test program was established to run the machine during various conditions and to characterize the vibration response. The results of these preliminary tests are summarized as follows:

(1) The low-frequency vibration was present at significant amplitudes when discharge pressure exceeded 1500 psig and did not exist at discharge pressures below 1100 psig.

(2) The low-frequency vibration initially appeared at frequencies of 780 to 660 cpm and then shifted to a frequency of approximately 540 cpm as discharge pressure and temperature increased.

(3) The low-frequency vibration was measurable on the entire structure and the associated piping system.

(4) The concrete support structure did attenuate this motion, and the foundation below the baseplate was quiet.

(5) During startup and shutdown conditions the rotor system was well behaved and passed through its first critical in a normal fashion.

(6) There were no reciprocating machines in the immediate area that could influence this unit.

The initial test program confirmed the presence of this low-frequency vibration and tied down the fact that it was strongly related to discharge pressure. The initial test program did not provide any substantial clues as to the origin of the low-frequency vibration.

One suspicion was that there might be an excitation originating from the fourth-stage aftercooler. During one of the shutdowns the exchanger head was pulled, and a bundle of corrosion coupons was found bolted in place. That was the proverbial "armadillo" in the piping system. The coupons were removed and the machine restarted. At 1500 psig the low-frequency excitation reappeared; thus scratch the "armadillo" theory. Somewhat later the aftercooler was removed completely and replaced with a spool piece. Again no influence was found on the low-frequency action.

PRESSURE PULSATION TEST RESULTS

A second phase of field testing was then implemented. During these tests the shaft vibration was measured concurrently with the dynamic pressure pulsation at the fourth-stage discharge flange. The significant results of this test are summarized in figure 3. From these data it is apparent that a strong relationship must exist between low-frequency shaft vibration, dynamic pressure pulsation, and discharge conditions.

After following several other blind alleys, we established a test plan to make a series of dynamic pressure pulsation measurements throughout the high-stage piping system. Figure 4 shows the approximate location of available test points A to J, corresponding to the casing drain at the third-stage suction all the way through the fourth-stage discharge piping. During this test the compressor exhibited a low-frequency vibration at 558 cpm (fig. 5). The measurements revealed that under this condition the third stage (points A, B, and C) did not exhibit the low-frequency activity, and neither did the suction of the fourth stage (point D). However, the fourth-stage casing drain (point E) and the fourth-stage discharge piping (points F to J) did reveal a definite pressure pulsation at 558 cpm.

These same data were observed in another manner (fig. 6) by plotting dynamic pressure pulsation amplitudes versus distance from the fourth-stage discharge flange. This showed that the magnitude of the pressure pulsation decayed rapidly as a function of discharge pipe length. Since the low-frequency excitation did not appear on the suction nozzle to this stage, it was concluded that the exciting force was internal to the compressor and probably located between the fourth-stage suction and discharge flange.

After many hours of discussion with the original equipment manufacturer it was generally agreed that the rotating assembly could not generate the type of frequency measured. Since everything on the rotor system appeared to be normal, the next step was to begin examining the entire machine drawing set for anything that was different from traditional design. During this review someone mentioned the balance port at both the third and fourth-stage discharge nozzles. This balance port consisted of a rectangular passage that connected one side of the discharge nozzle with the other side of the nozzle (fig. 7). The passage was an integral part of the casing and was intended as a means to pressure balance each discharge nozzle.

From this discussion it was decided (1) to plug both ends of each balance port, (2) to hand grind the surface in contour with the nozzle, and (3) to provide a drilled weep hole in one side. After return to the plant site it was confirmed that a balance port did exist. Running a piece of wire through the hole on one side confirmed that the balance port was a circumferential passage that connected both sides of each discharge nozzle.

BEHAVIOR AFTER MODIFICATIONS

The unit came up smoothly through the first critical and reached normal operating speed without any problem. This time the machine came up to 1500 psi, then 1600 psi, without the low-frequency excitation. Finally the compressor was fully loaded, and the previously encountered vibration at 540 or 660 cpm did not appear. Figure 8 is representative of the final condition of this machine at 9340 rpm and a discharge pressure of 2150 psig.

Note that running speed motion was decreased to amplitudes of 0.13 mil vertically and 0.09 mil horizontally. This was due to refinement of compressor rotor balance during the course of this project. There are two minor components at 900 and 2580 cpm that appeared at amplitudes of 0.04 and 0.06 mil, respectively. These components were considered to be nondestructive and were later found to be associated with the stiffness of the O-rings installed with the squeeze-film dampers.

As part of the overall testing program X-Y shaft proximity probes were installed on the floating squeeze-film bearing housing to observe shaft motion relative to the housing. In addition, the standard X-Y shaft probes were mounted on the case and provided a measurement of shaft motion with respect to the case. The 1X vectors from both sets of probes are presented in figure 9.

From this plot of shaft motion with respect to the case and the bearing, the vector difference must be the motion of the squeeze-film bearing housing with respect to the case. Hence the squeeze-film housing was moving and providing the necessary additional system damping.

CONCLUSIONS

This unusual problem was corrected by plugging the discharge nozzle balance ports. The low-frequency vibration and the pressure pulsation in the fourth-stage discharge were eliminated. However, it is difficult to verify the exact nature of the specific physical mechanism and the associated forcing function. This uncertainty is due to a lack of knowledge of the accurate fluid properties of the process gas stream plus insufficient documentation of flow characteristics during the execution of field tests.

Sometime after the conclusion of this project one opinion was that the balance port passage acted as an acoustic resonator (open at both ends). On the basis of the passage length and the estimated sonic velocity, the fundamental acoustic resonant frequency was approximately related to the measured low-frequency excitation. Another opinion was that the measured frequency was related to a vortex shedding phenomenon as the discharge flow passed across the rectangular balance port passages. In either case a more rigorous examination of the specific flow characteristics would be required to gain a proper understanding of this fluid-related excitation.

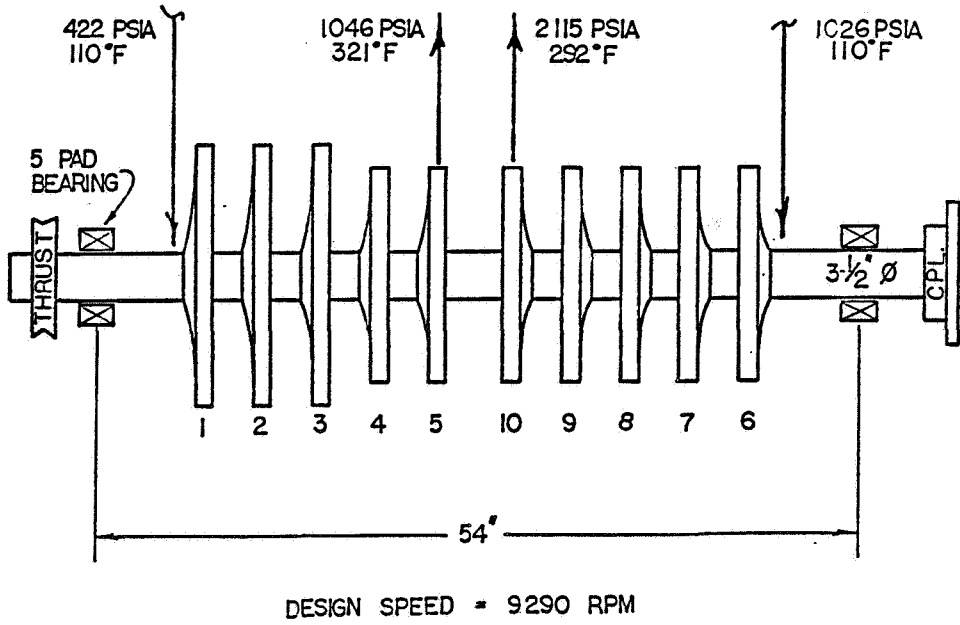


Figure 1. - Ten-stage rotor configuration.

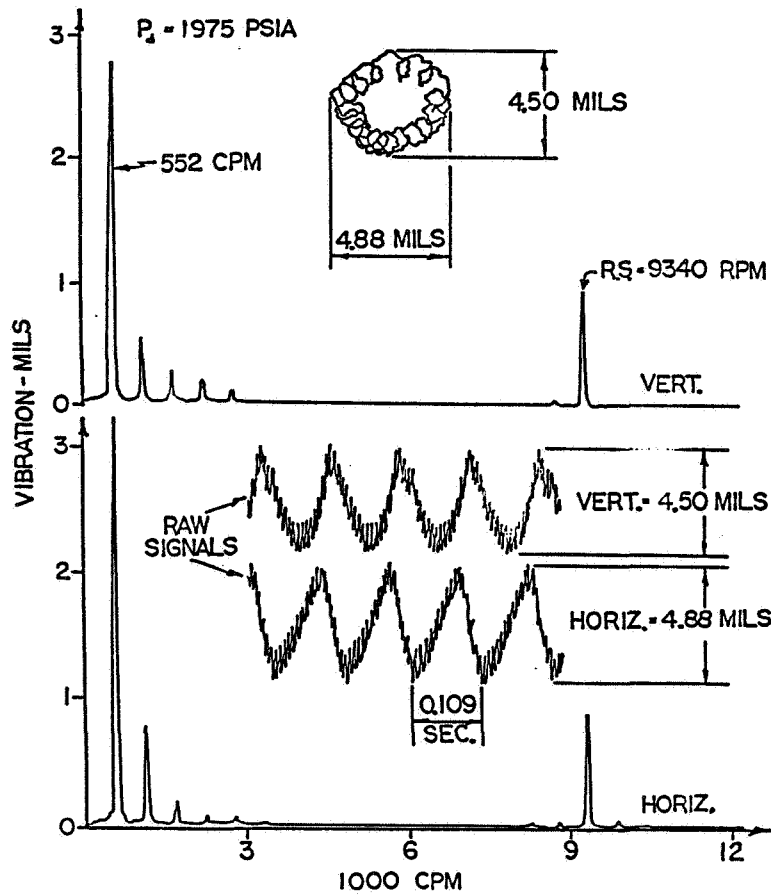


Figure 2. - Initial coupling-end shaft vibration.

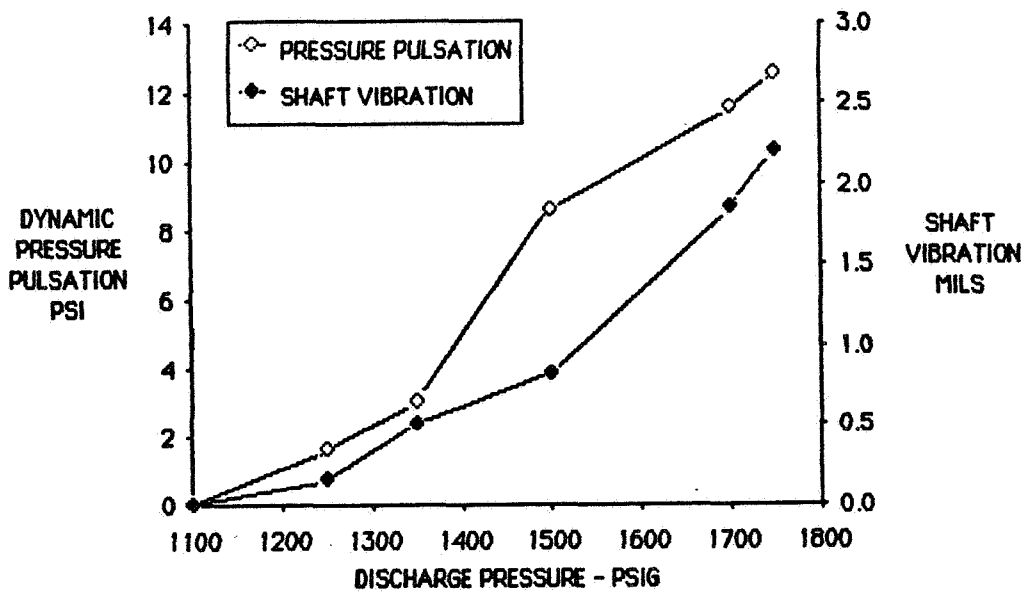


Figure 3. - Pressure pulsation and vibration versus discharge pressure.

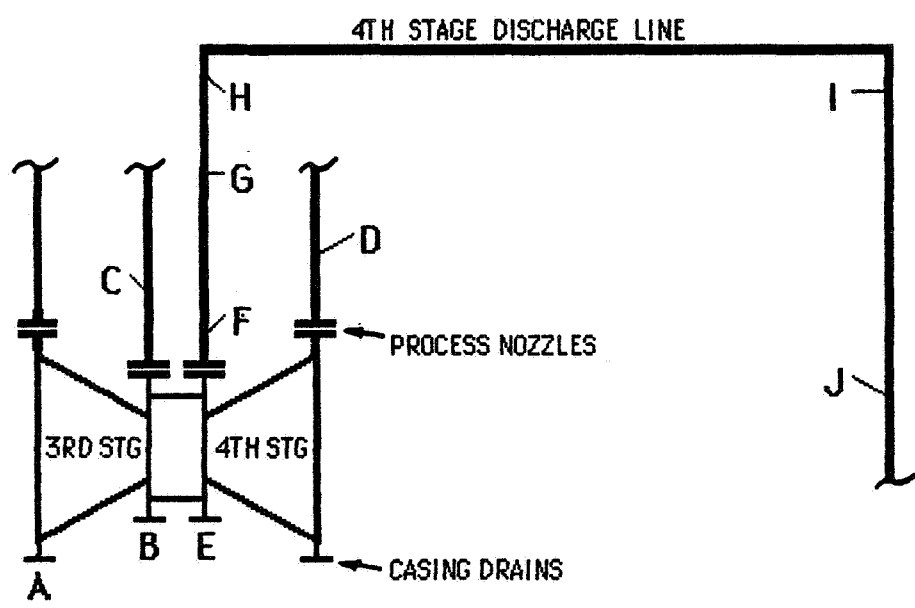


Figure 4. - Location of pressure pulsation transducers.

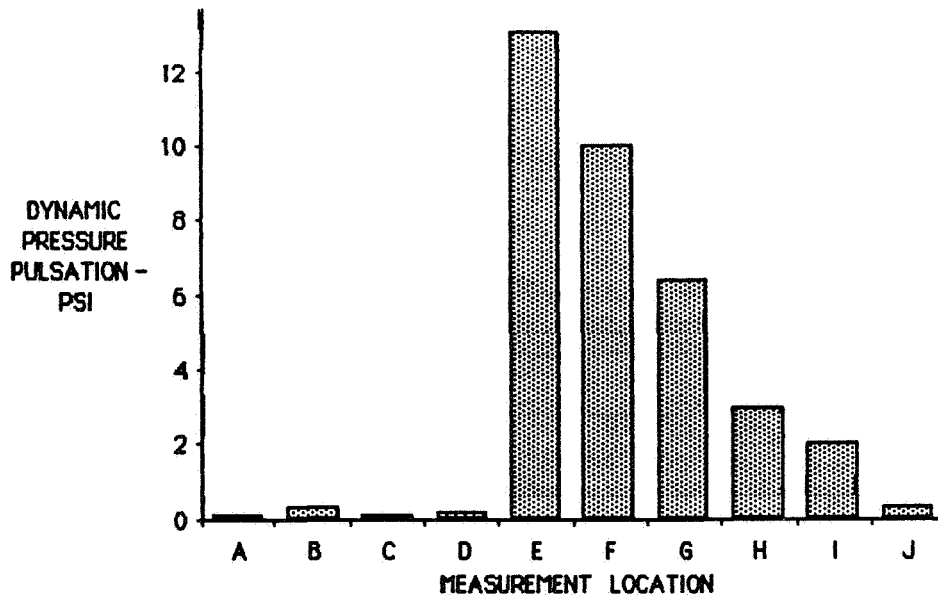


Figure 5. - Dynamic pressure pulsation at 558 cpm.

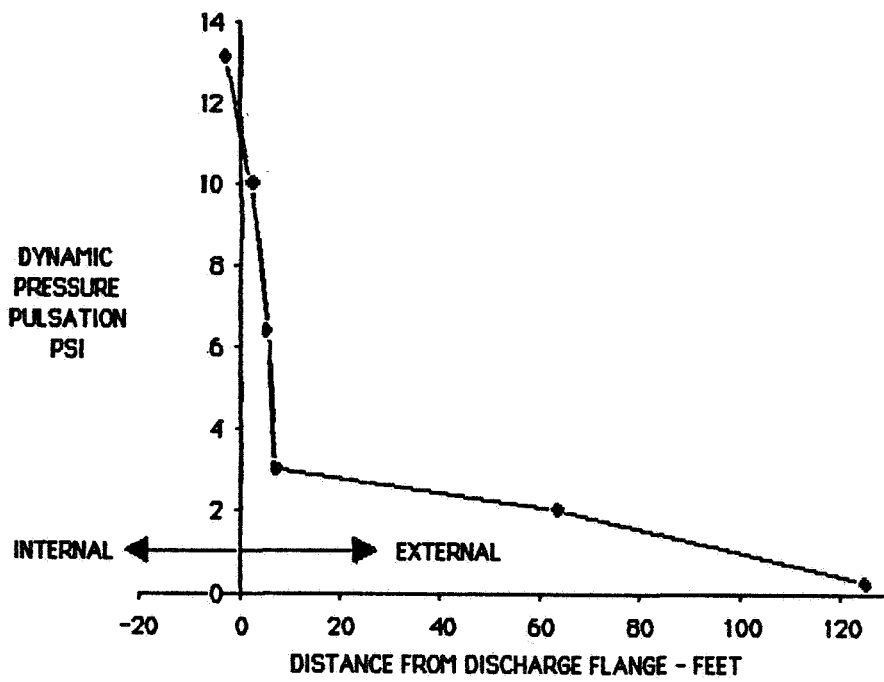


Figure 6. - Pressure pulsation with respect to discharge pipe length.

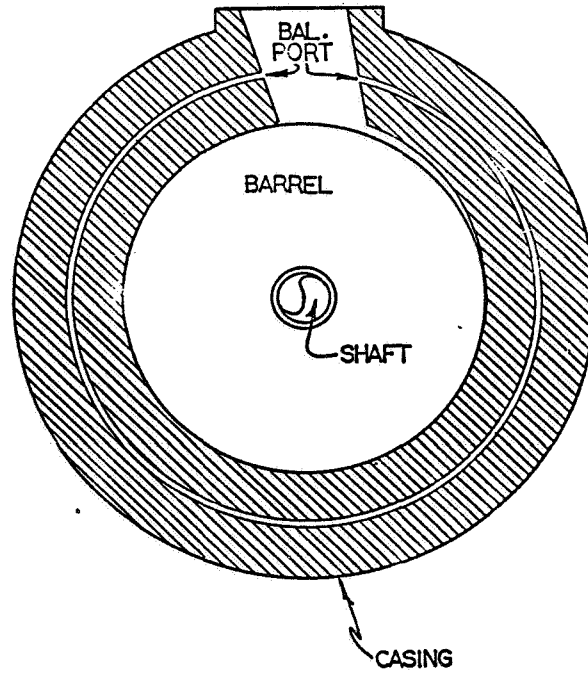


Figure 7. - Cross section through discharge nozzle.

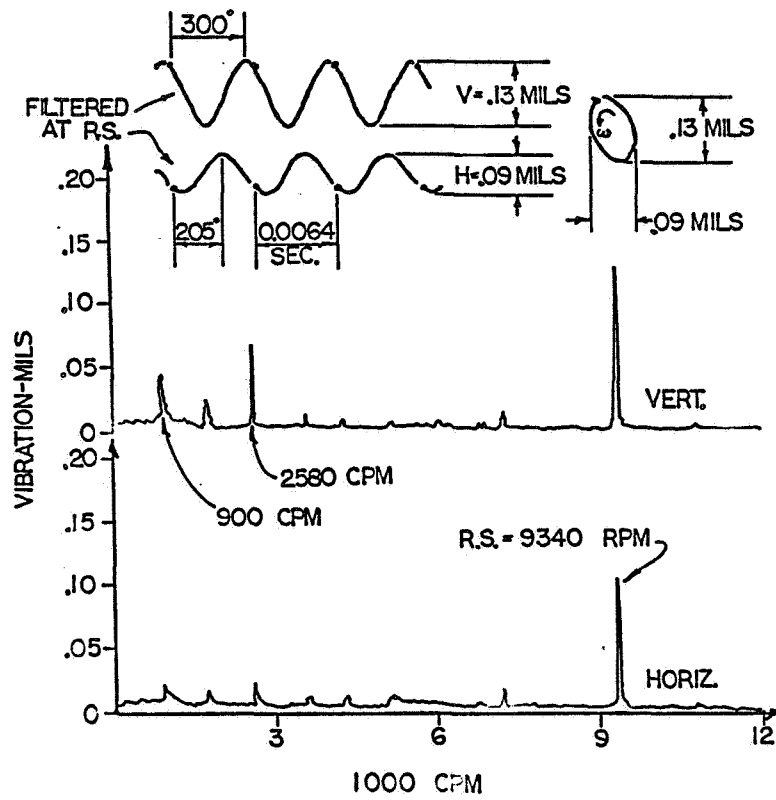


Figure 8. - Final coupling-end shaft vibration.

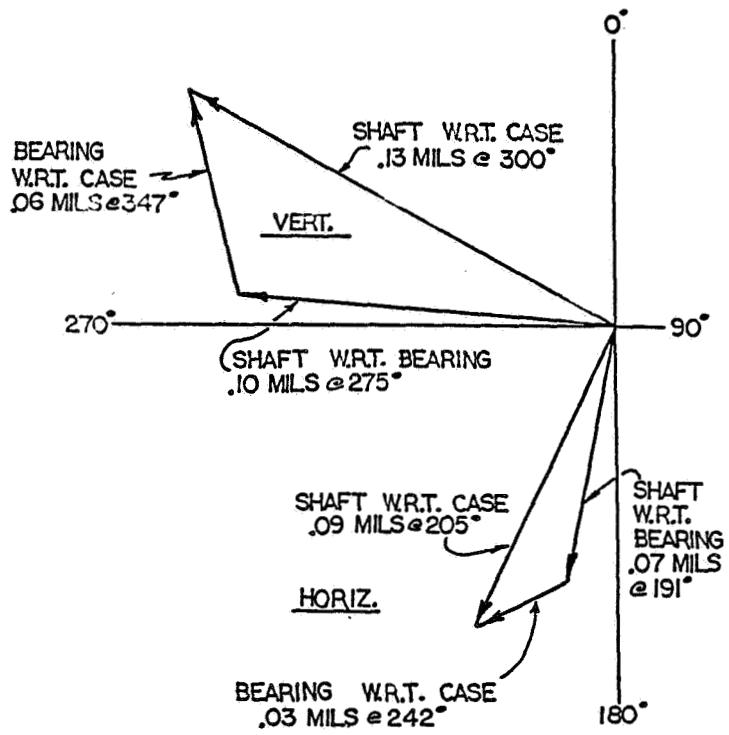


Figure 9. - Relative motion of coupling-end bearing.