A CENTRIFUGAL COMPRESSOR SYNCHRONOUS MOTOR CASE HISTORY

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ABSTRACT

This case history involves a large synchronous motor-centrifugal compressor with gear and and elastomeric coupling. The vibrations have been excessive for years. Short operation life has been the result. Shaft criticals and structural resonances have been identified, along with coupling wear and resulting balance problems. An unusual problem with the shim packs under the motor has been observed. Which apparently results in non-linear vibrations. Skid design problems are discussed.

1. INTRODUCTION

The unit described in this case history has a synchronous motor driving a centrifugal compressor through a coupling made up of a gear coupling and an elastomeric coupling. The centrifugal compressor consists of three separate cantilevered wheels driven through pinions from a large bull gear. The input to the compressor goes to the bull gear. The elastomeric coupling is used to control the transient torsional vibrations caused by the synchronous motor during start-up. Both the gear and the elastomeric couplings had shorter than expected lives.

The gear coupling had been replaced with a new coupling just before starting the testing described here. The old coupling had excessive radial clearance. Also, the old gear had straight teeth on both sides. The new coupling had the male teeth crowned.

Picture 1 to 7 show the machine from different angles.

2. OBSERVATIONS

Vibrations measurements of the shaft relative to the bull gear case and vibrations of the motor housing showed that there was a shaft critical speed very close to run speed on the compressor and there was a structural resonance of the motor near run speed. Refer to Figures 1 to 3 for bode Plots for vibrations during rundowns of the unit. Bump test results seemed to confirm the motor resonance observation from the Bode plots in Figure 1 and 2.

3. DEALING WITH THE COMPRESSOR

The compressor shaft vibration turned out to be strongly influenced by the unbalance at the elastomeric coupling. This is counter-intuitive, since the elastomeric coupling is on the other side of the spool piece from the compressor.

The elastomeric coupling has a centring bushing to keep the spool piece rotating on the shaft centre-line. This bushing was not functioning correctly. It allowed the spacer half to move to different radial positions despite efforts to shim the bushing. We were able to improve the balance of the coupling, but could not fix the problem. Installation of a new bushing was recommended.

It is also apparent that the long-term runability of this unit is not likely to be good since the compressor shaft exhibits a shaft critical near run-speed [Figure 3]. As long as the alignment and coupling balance are kept to very good standards, the vibrations will be acceptable. The sensitivity to small changes in balance and alignment remains of concern.

4. DEALING WITH THE MOTOR

The motor horizontal vibrations showed that a structural resonance or mechanical natural frequency [MNF] was present very close to run speed. This showed another reason why the unit had had so many maintenance problems over its life. Bump tests were performed that confirmed the proximity of an MNF to run speed. Note that there is a 180 degree difference in phase between the ends of the motor in Figure 1 and 2.

Inspection of the skid [see Pictures 4 and 5] suggested that removing some of the horizontal stiffness in the pony skid [the top part of the skid] should lower the offending MNF and reduce vibrations to acceptable levels. In a spirit of bonhomie, we embarked on the uncharted sea. Cutting progressively more and more of the gussets reduced the apparent MNF from bump testing, but was unsuccessful in reducing the vibrations at the top of the motor even though the operating deflected shape [ODS] changed [see Figure 4].

A different course of action was chosen. Since under-tuning was not working, how about over-tuning? The pieces of gusset that had been removed [which we had carefully numbered in case of this very eventuality] were welded back in place, and additional gussets were welded to the outside of the pony skid [see Picture 6]. The result of over-tuning was not entirely clear as we were unable to achieve enough separation between the bump test MNF and run speed. However, the ODS plot shown in Figure 4 indicates that the problem is likely due to the movement of the motor relative to the skid across the shim packs. This movement appears to be non-linear. Hence the apparent improvement in MNF without significant improvement in running vibrations.

The shim packs were not optimum either. There were too many shims in each pack and the shims were probably not entirely free of oil.

Refer to Picture 7. It shows a location provided by the motor manufacturer for a dowel. Each anchor bolt location has this provision. Installation of dowels was recommended.

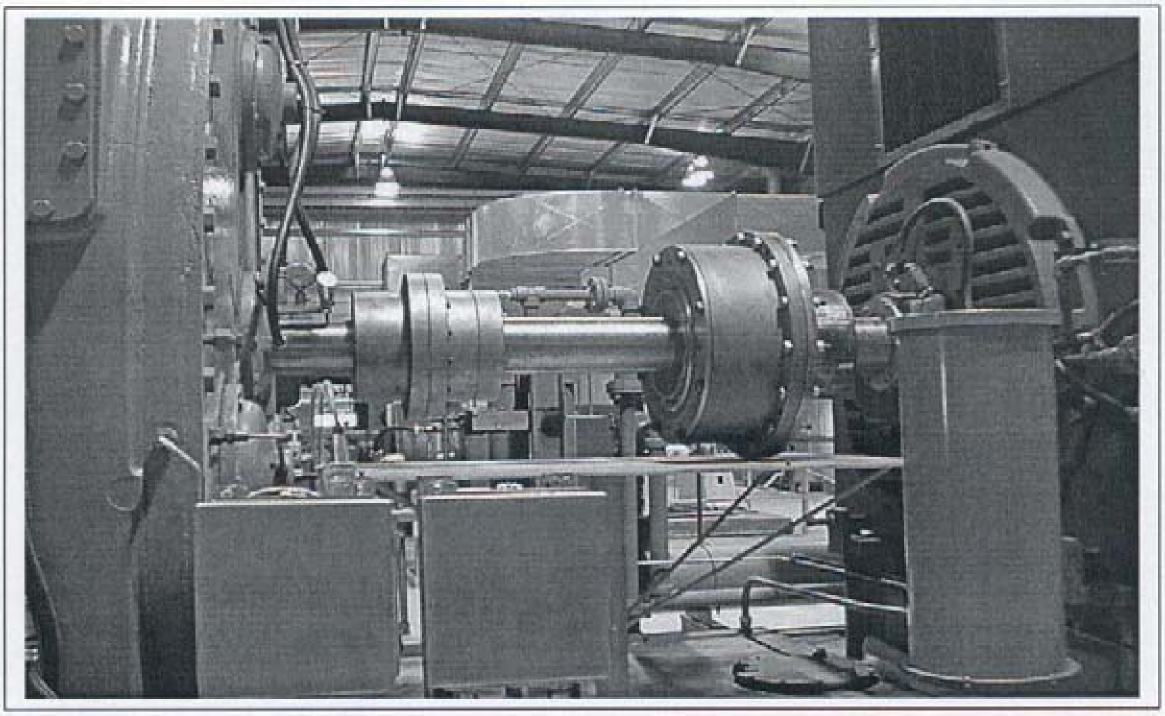
5. CONCLUSIONS

Design of machinery should include efforts to avoid shaft critical speeds and structural resonances close to run speed.

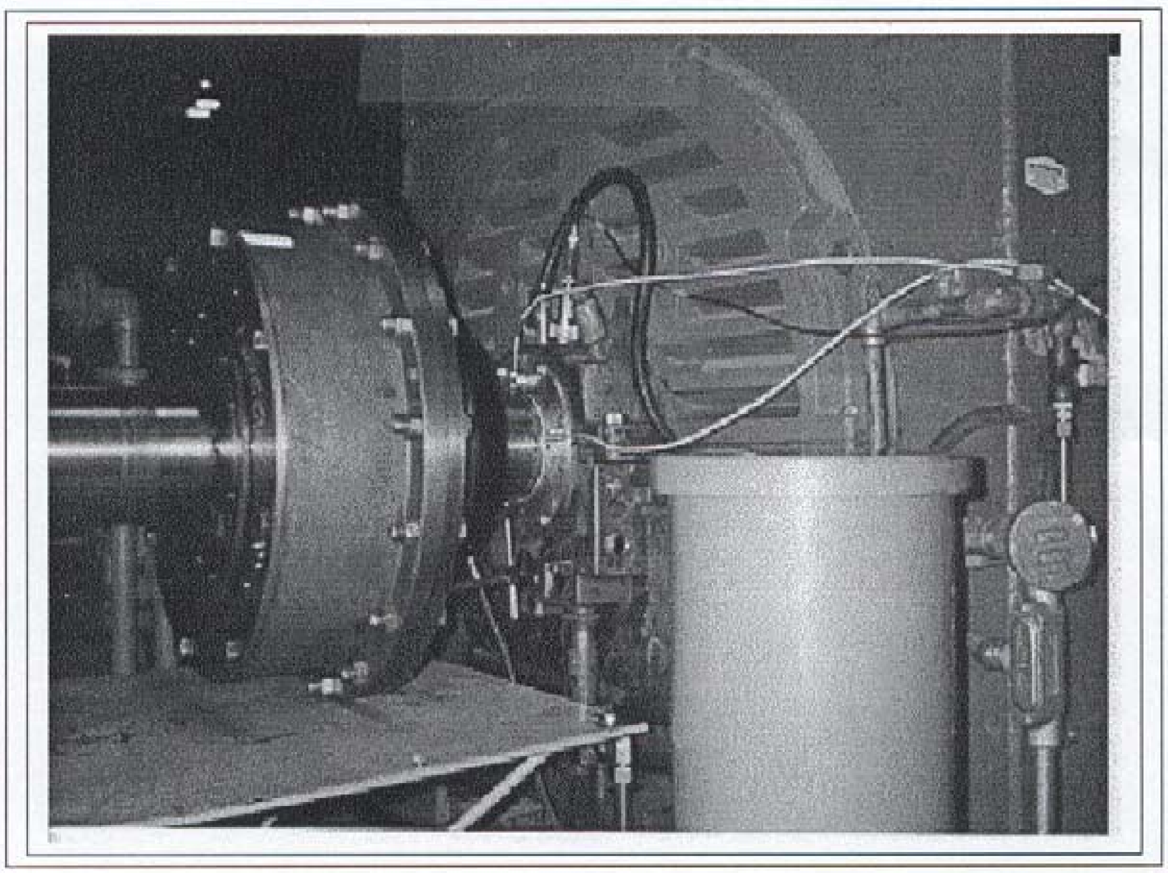
Skid designs should be developed that are sufficiently stiff to control the dynamic response of the structure on it.

Motors should be doweled to the skid or sole plate after installation and alignment testing is complete.

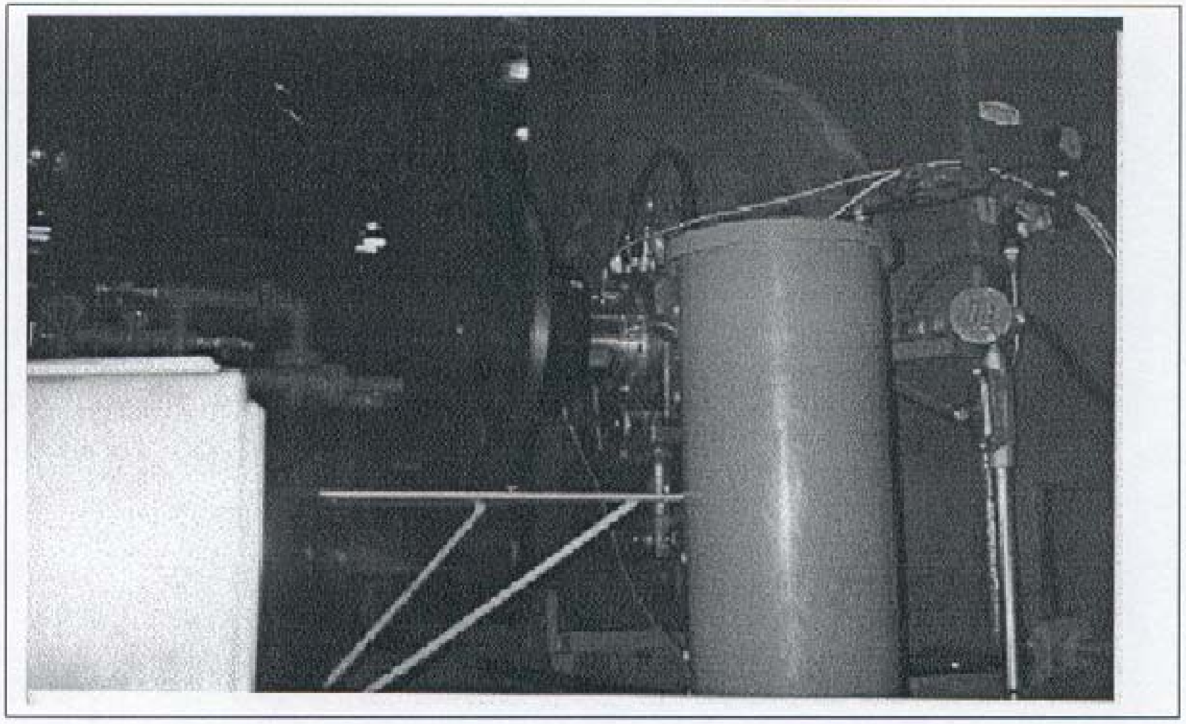
Shim packs should be limited to 3 shims made of stainless steel. The surfaces of the sole-plate, motor base, and shims should be clean and dry.



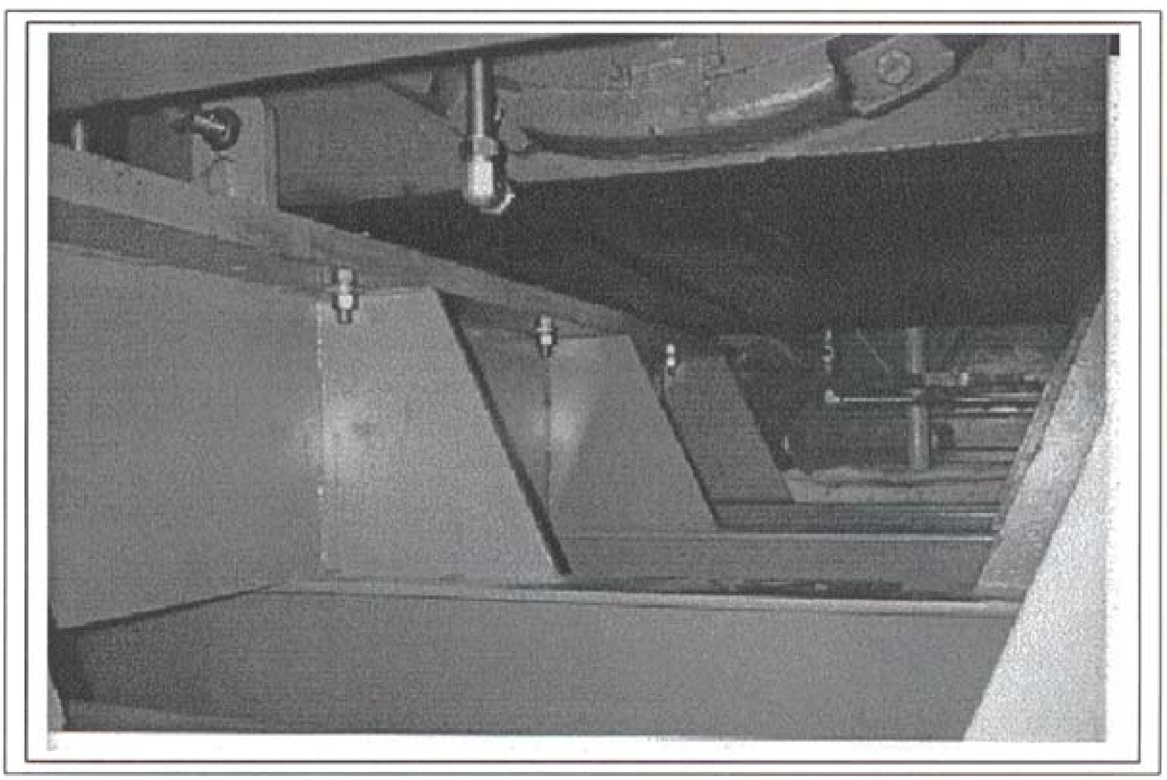
Picture 1: Gear coupling on the left, and Elastomeric coupling on the right [102847]



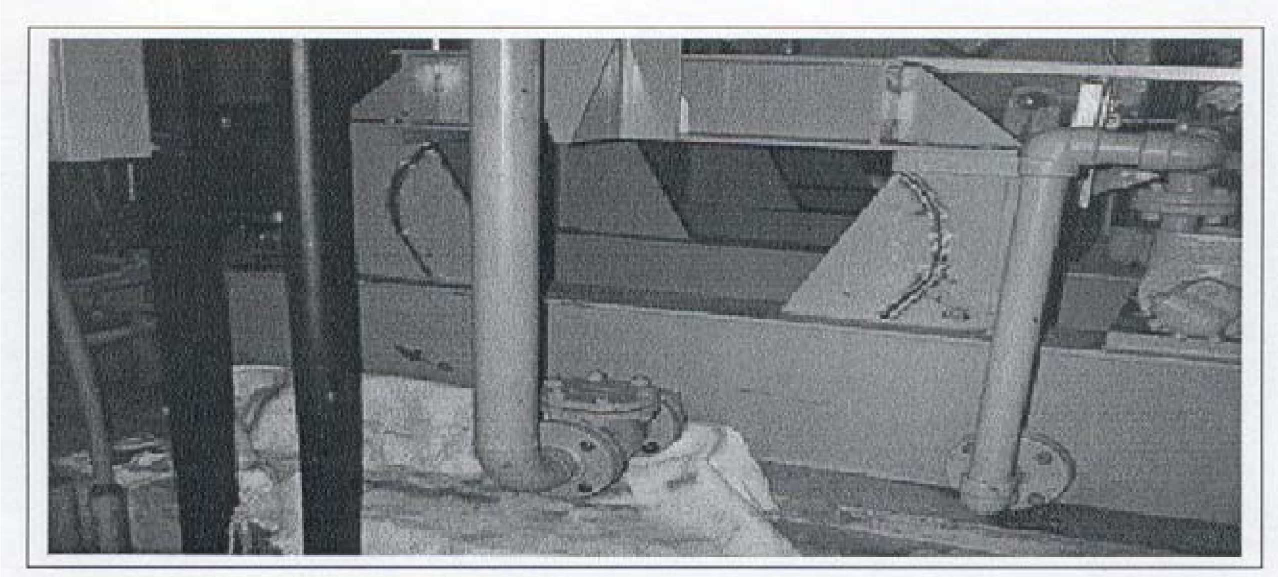
Picture 2: Motor with Elastomeric Coupling, Note: Proximity Probes



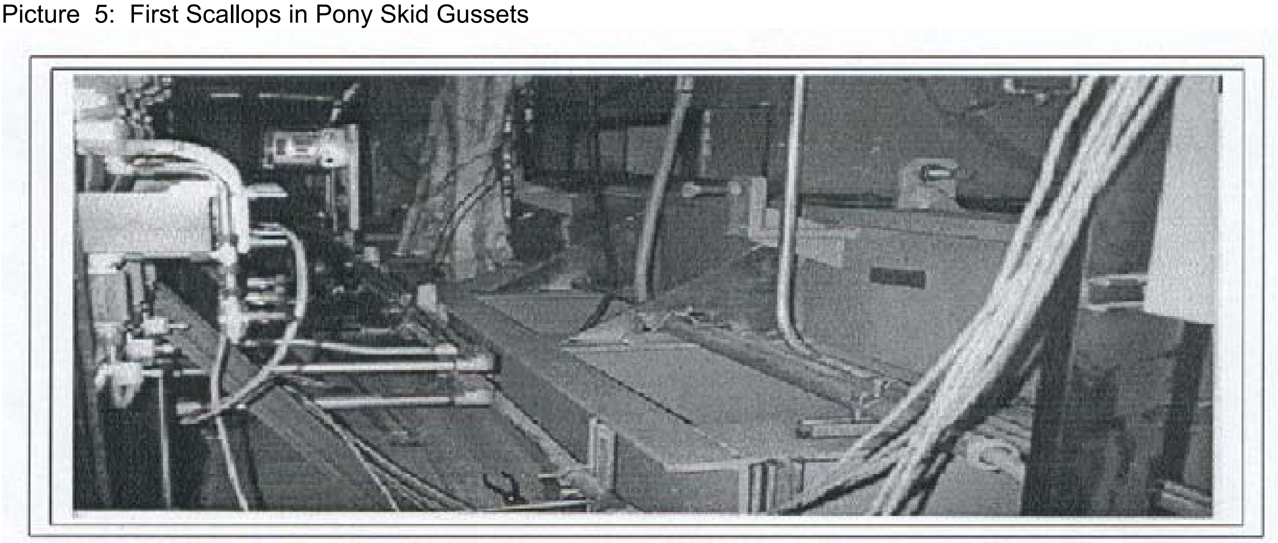
Picture 3: Motor Solo, Coupling Removed



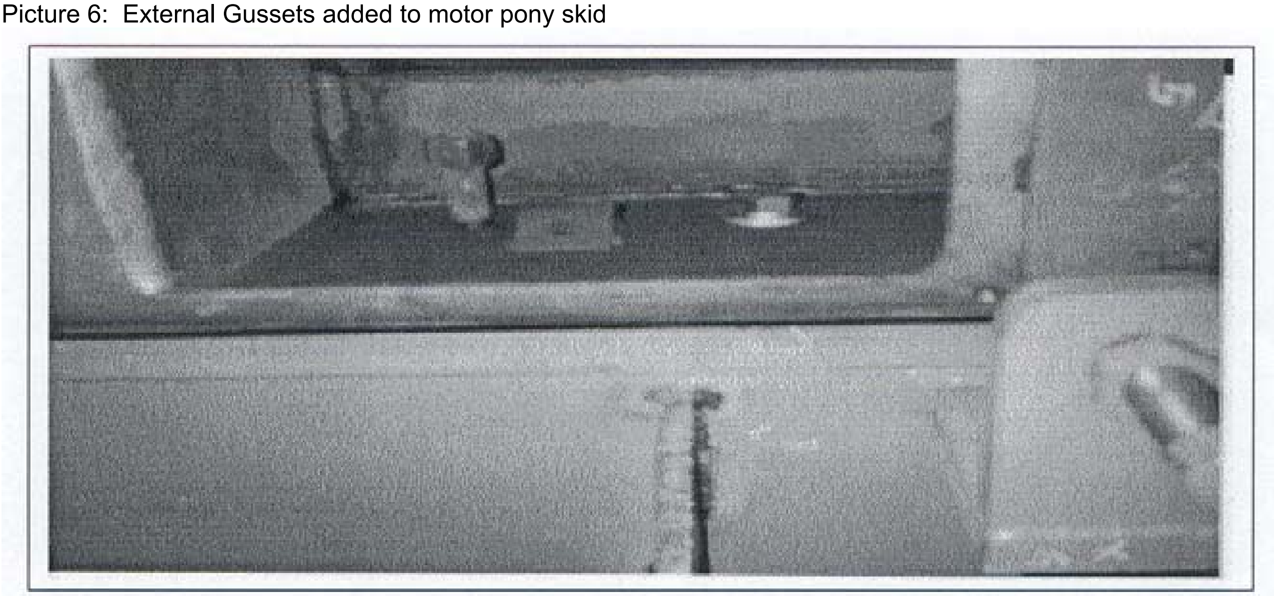
Picture 4: As Found Gussets, Pony Skid under Motor



Picture 5: First Scallops in Pony Skid Gussets



Picture 6: External Gussets added to motor pony skid



Picture 7: Motor anchor bolt, jack bolt, and dowel location

APPENDICES

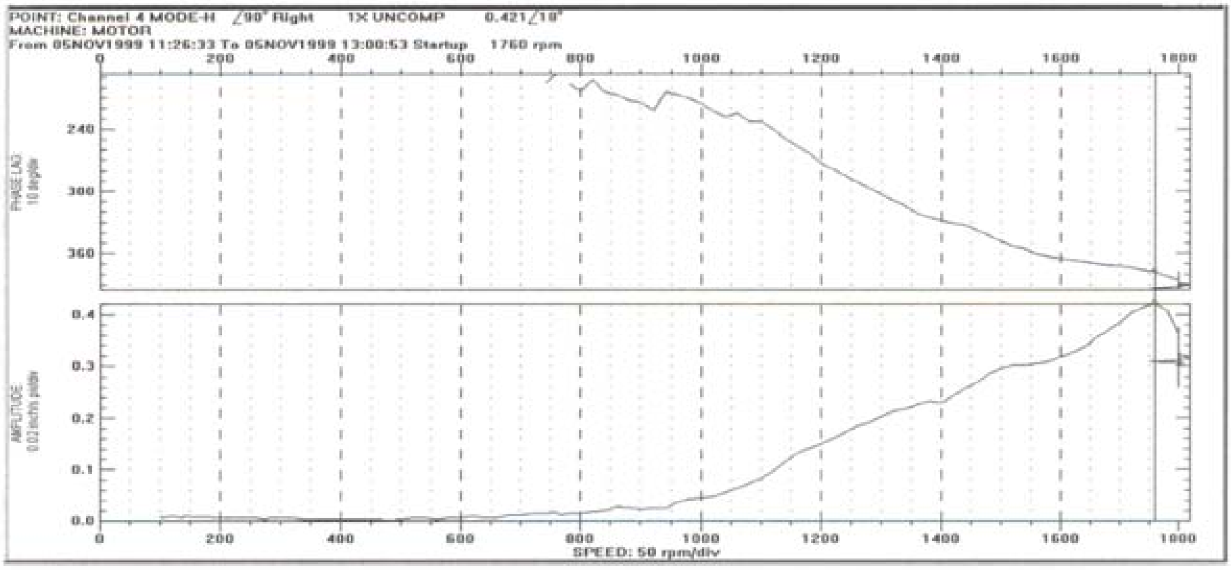


Figure 1: Rundown vibration plot for the motor outboard end, horizontal [6 gussets cut]

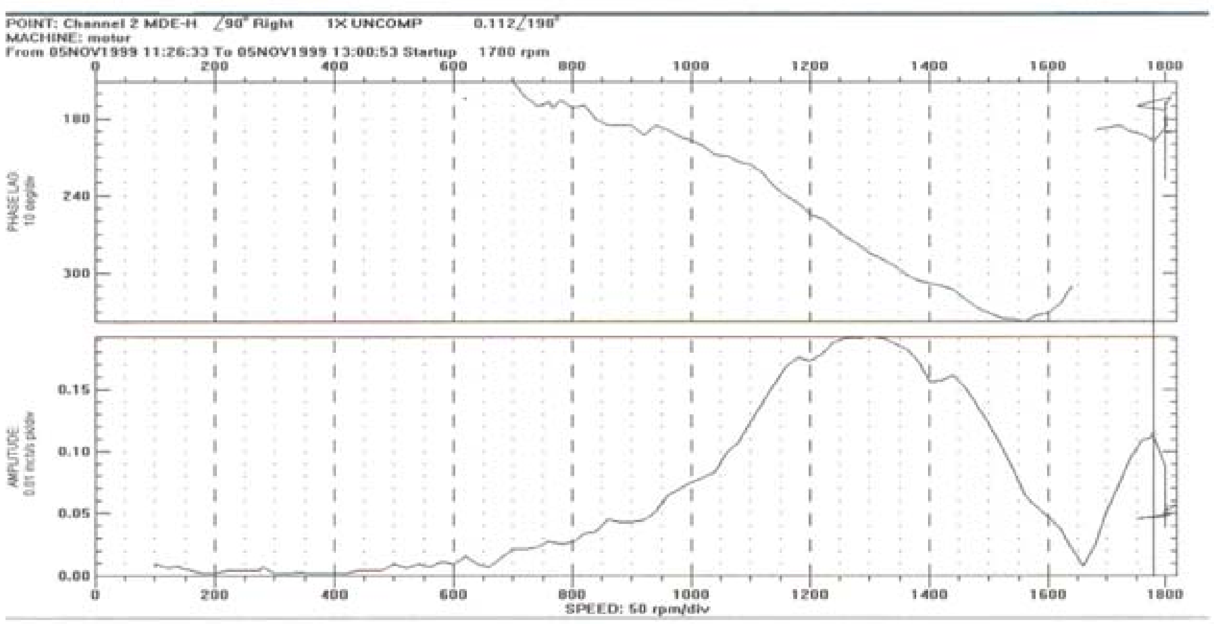


Figure 2: Rundown vibration plot for the motor inboard end, horizontal [6 gussets cut]

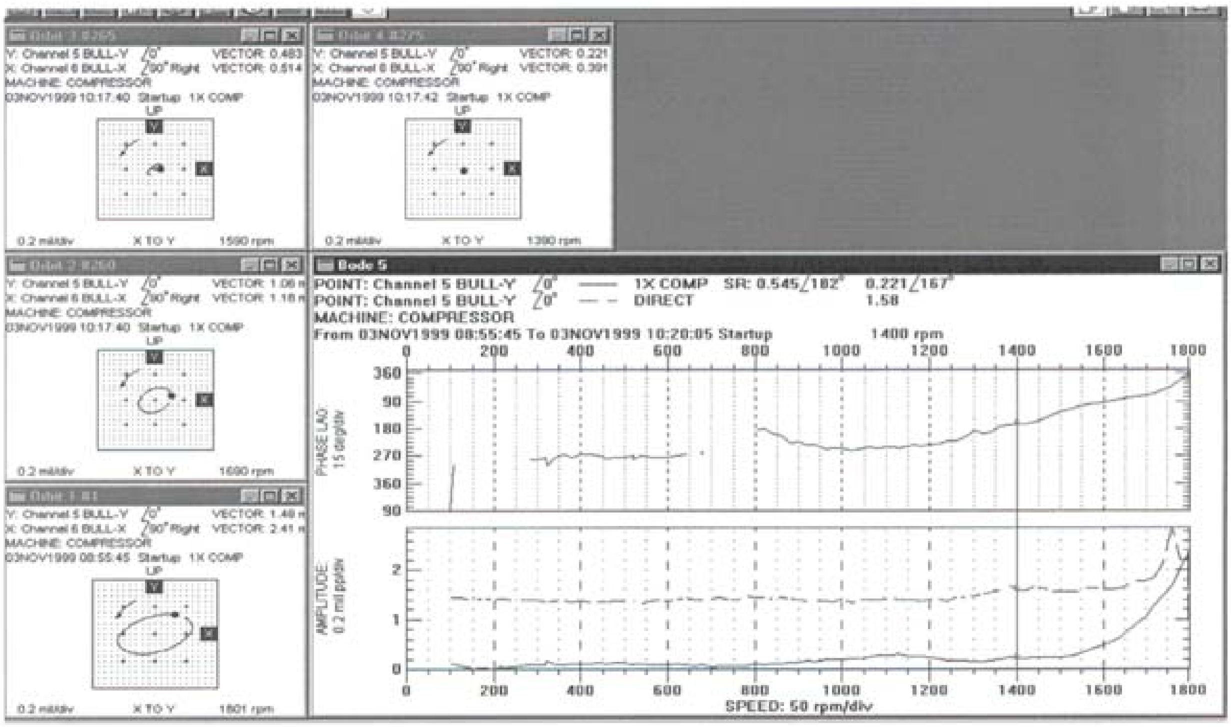


Figure 3: Rundown vibration plot for the gearbox drive end, shaft relative to housing
[The vibration increases from 1600 to 1800 rpm by 4.7 times while the ratio of the speed squared is 1.26. Resonance is suspected.]

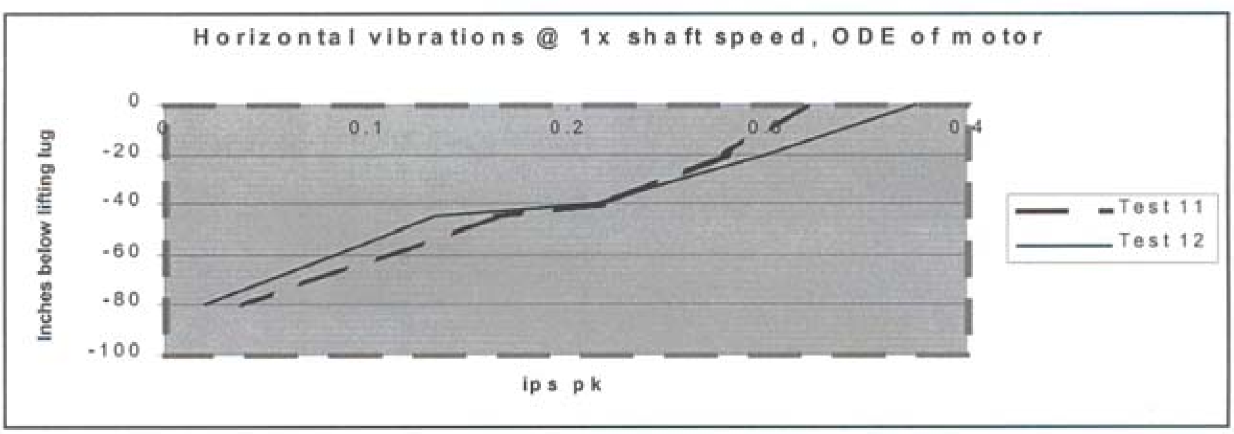


Figure 4: Operating Deflected Shape [ODS] of motor outboard end with gussets cut out [Test 11, see Photograph 4] and with gussets replaced plus external gussets added [Test 12, see Photograph 5] Note the discontinuity at the –45 inch level.
The shim pack between the motor feet and the top of the pony skid is at this level. Bump testing showed the horizontal out-of-phase mechanical natural frequency of the motor was about 23 Hz for Test 11 and 32.5 Hz for Test 12.]

Biography:

Brian Howes is Chief Engineer for Beta Machinery Analysis Ltd., Calgary. His previous experience includes: research and development in the area of pulsations and vibrations of reciprocating compressor piping systems, 28 years of troubleshooting problems in many countries for a wide range of equipment including turbines, centrifugal and plunger pumps, centrifugal, screw and reciprocating compressors, pulp refiners, paper machines, ball mills, furnaces and piping systems. He has a Master of Science in Solid Mechanics from the University of Calgary, and is a member of the Board of Directors of the Canadian Machinery Vibration Association (CMVA).