# FERMILAB COLD COMPRESSOR BEARING LIFETIME IMPROVEMENTS

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## ABSTRACT

The Fermilab Tevatron liquid helium cryogenic system has high speed, cold turbocompressors, made by IHI Co. Ltd., that allow for subatmospheric, lower temperature operation (3.5K). The present cold compressor design utilizes spiral wound dynamic gas foil bearings for the radial journal bearings. Fermilab has experienced a variation in journal bearing lifetimes leading to more frequent maintenance. Bearing lifetime appears to be satisfactory when the cold compressor is operated in a stable environment, such as in a stand-alone cryostat. However, the operating configuration calls for the cold compressor units to be mounted on other equipment. The degradation in bearing lifetime is attributed to the cold compressor being exposed to an externally induced vibratory environment. The vibratory environment will be characterized. Possible solutions to improve the bearing life will be presented, including provisions for additional support structure to minimize vibrations, and redesign of the foil bearing to improve tolerance to vibrations.

## COMPRESSOR AND BEARING DESCRIPTION

The IHI compressor, which consists of a single stage centrifugal impeller powered by a variable frequency induction motor, has been previously described.[1] The radial journal and thrust bearings are foil-type self acting (dynamic) gas bearings. The foils do not establish a supporting film until the shaft reaches an appropriate minimum lift off speed.

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There is contact and therefore rubbing occurring during startup and shutdown. This contact is handled by a Teflon coating on the inner surface of the stainless steel foil. The maximum shaft speed is 90 krpm, with typical operating conditions of 50 krpm. Each machine has two spiral wrapped journal bearings on a shaft of approximately 1.6 cm.

## OPERATION AND MAINTENANCE EXPERIENCE WITH IHI BEARINGS

The initial operating experience with the production units has been described in previous papers.[2,3] Over 130,000 total hours of operation have been accrued on these machines, at an average of over 4800 hours per machine. The IHI bearings have proven themselves operationally, even showing the capacity to survive off-design conditions of liquid inhalation onto the impeller. Yet, long-term operation has shown that journal bearings require more frequent replacement than expected. No problems have been experience with thrust bearings, so these will not be further addressed in this paper.

For reasons of saving space, the cold compressor cryostat mounts directly on top of the satellite refrigerator's valve box. The valve box houses a phase separator and piping to join the refrigerator to the magnets below ground. The valve box is fixed at the bottom and braced at the top, with internal piping runs which are not stiffly supported to limit heat leak. These mounting and flow restrictions place the turbocompressor in an environment where it is susceptible to external vibrations. The original prototype IHI cold compressor was tested in a free standing cryostat mounted directly to the floor. This prototype unit was run more than 10,000 hours at a variety of conditions, with numerous starts and stops, with no sign of bearing degradation. These tests, and prior vendor experience, led to expectations of 100,000 hours of journal bearing life. The reduction in production unit bearing life is attributed to their being mounted in a way that exposes them to an externally induced vibratory environment.

The initial indication of bearing wear occurred during planned bearing replacement. An upgraded version of the bearing, one that is designed to achieve operating stiffness at

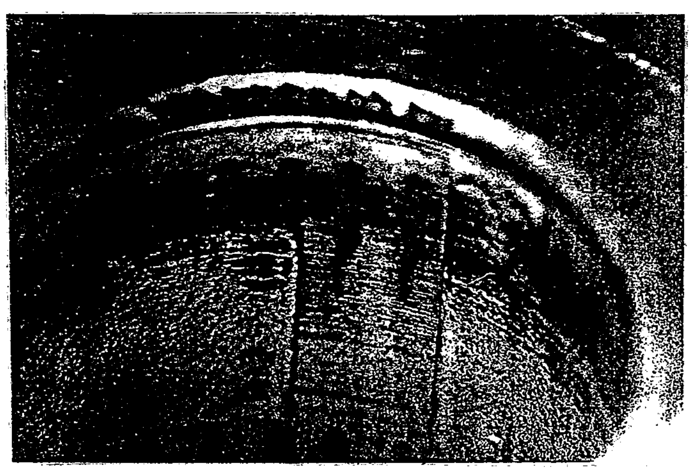

**Figure 1.** Representative journal bearing showing signs of severe wear.

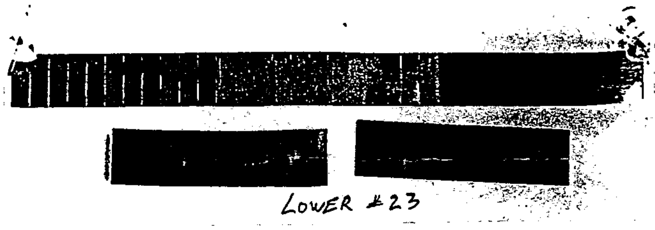


**Figure 2.** Failed journal bearing, unwrapped for display.

lower shaft speed (20 krpm vs. the original bearing's 40 krpm), was purchased and installed in August 1994. The original bearings had been operated for an average of 1060 hours on each of the 27 turbocompressors. Upon bearing disassembly, it was noted that the original bearings showed signs of more wear than was expected. Representative wear on a foil as it sits in its cartridge is shown in Figure 1.

During the next six months of upgraded bearing operations, three bearing failures occurred. The earliest one was after 2566 hours. The unwrapped foil from this high speed failure is shown in Figure 2. Two more failures occurred after approximately 3000 hours in units that only ran at minimum speed. Failures were characterized by the inverter reaching the maximum current when trying to start the motor drive. At least one of these units was operating in an environment where noticeable vibrations were present. In response, the minimum speed was elevated from 20 to 30 krpm to ensure bearing lift off, and no bearing failures have subsequently occurred.

This second set of bearings was removed and replaced in July 1998. This replacement was motivated by the desire to begin an impending Tevatron operation period with brand new bearings. Each machine had run for an average of at least 3800 hours. (The precise hour information is unavailable due to sporadic hour meter maintenance after this time.) Inspection revealed similar patterns and frequency of wear as seen during the 1994 bearing removal. Out of 24 units in service, 5 or 6 were deemed to have badly worn journal bearings.

In summary, the journal bearings showed wear that degraded their lifetime and in some cases led to premature failures of the rotating system. The wear was thought to be caused by excessive vibration, exceeding what this particular bearing can tolerate. With insufficient dampening, the vibration induced rubbing leads to wear and ultimately failure.

## SOLUTIONS CONSIDERED

Ideally, the cause of the vibrations should be eliminated. Unfortunately in this case, this cannot be done because the cause is not well understood. It is speculated that turbulent, or possibly two-phase, liquid helium flow in the loosely supported valve box piping is the cause. Pulsation from the reciprocating wet expander could also contribute to excitation of the bearing system. Given the difficulty caused by modifying the valve box, it is not practical to alter flow paths, reduce noise source, or stiffen up piping. Therefore, we are limited to modifying the compressor system itself, as discussed below.

## Enhanced support

Stiffening the cold compressor support with additional structure can reduce the amplitude of the vibration. A 3.35 m long, 10.2 cm square by 0.48 cm wall structural steel tubing column fixed at the floor and at the roof has been designed to provide additional stiffness to the IHI cold compressor structure. A photograph of this structure in a satellite refrigerator building is shown in Figure 3.

## Isolation

One could further reduce vibration by decoupling the compressor from the valve box. This could be achieved with additional bellows on the intake and exhaust helium and vacuum circuits while the turbocompressor is held by some other support system. This would involve modifying the flow passages to the impeller, which could have an effect on performance.

A stand alone cryostat for the IHI cold compressor would be an extreme way to isolate it from the valve box. This was not pursued due to the previously mentioned space limitations. Furthermore, there would be additional heat leak and pressure drop.

In some cases, a dynamic absorber can reduce vibrations for frequency specific resonance. However, an operating deflection shape analysis, done by an outside consultant, showed that the apparently flow-induced vibrations cannot be matched to a specific frequency, thereby reducing the viability of this option.[4]

## Improved bearing design

There are several styles of foil bearings. The style used on our machine, the spiral wound foil bearing, is an older concept. More modern designs often use either "tension" or "bending" style leaves.[5] Of particular interest is a style known as a "bump" foil, which has a spiral wound foil backed by a corrugated foil. For examples, see references.[6-9]

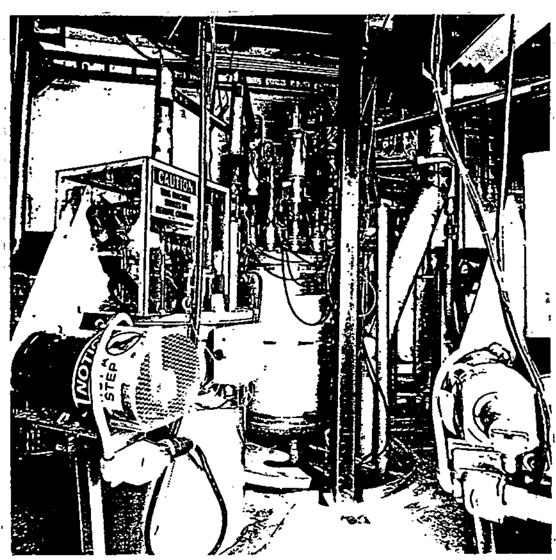


**Figure 3**. Support column installed for additional stiffening of cold compressor on valve box.

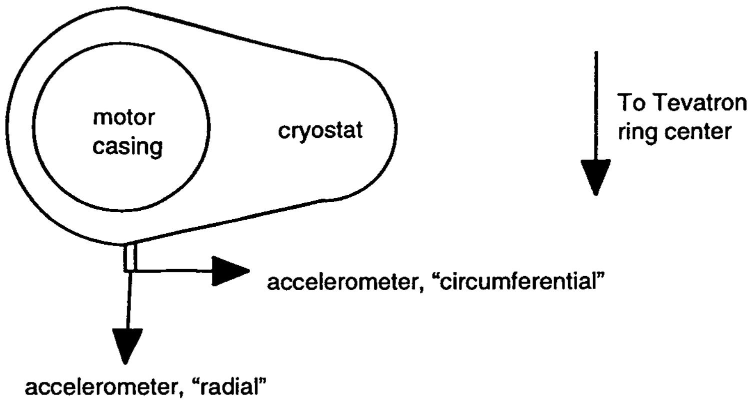


**Figure 4**. Schematic top view of IHI cold compressor, showing external accelerometer positions.

For our application, improved life could be gained by redesign of the bearings to a more state-of-the-art configuration. Bump foil bearings are an option, but a bearing expert would be required to predict the frequency dependent rotordynamic stiffness and dampening characteristics needed to develop a long life bearing system.

## INSTRUMENTATION

External vibration measurements were taken on cold compressor units installed in operating refrigerators. The standard method used vibration transmitters to integrate accelerometer inputs to produce a DC signal for peak velocity. Peak velocity is a good measure for assessing the severity of mechanical roughness in our frequency range.[10] The transmitters used have a peak velocity range of 0-2.54 cm/s. Measurements were made in the horizontal plane circumferential to and radial to the accelerator ring. The mounting positions of the accelerometers is shown in Figure 4.

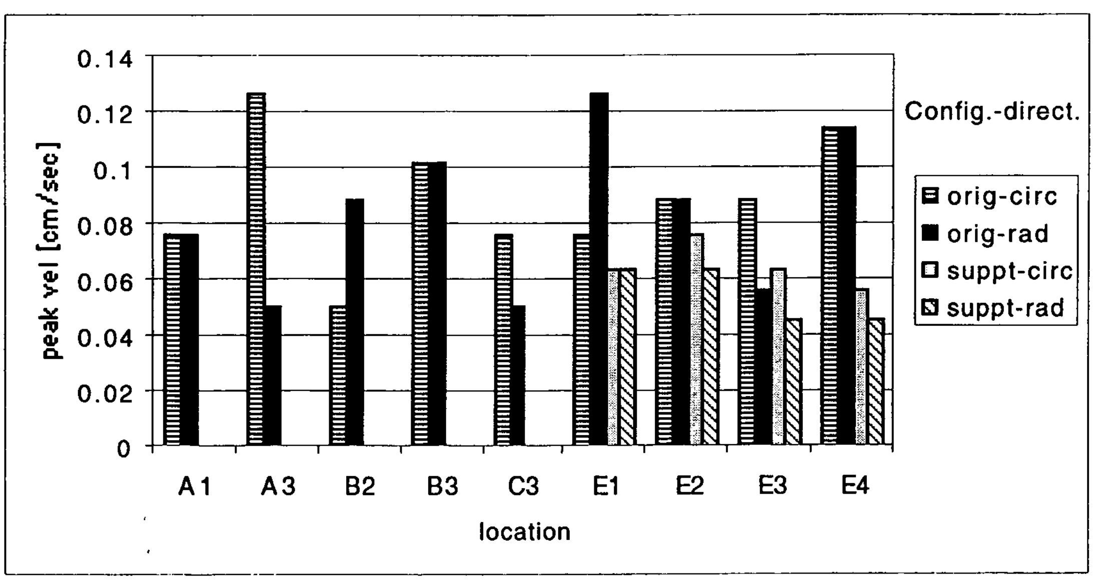


**Figure 5**. Cold compressor vibrations at various locations around Tevatron ring.

**Table 1.** High frequency accelerometer data at E3

| Add'l Suppt? | direct | freq[hz] | pk accel[g] | pk vel[cm/s] | p-p $\delta$ [$\mu$m] |
|---|---|---|---|---|---|
| None | "radial" | 27 | 0.0081 | 0.048 | 5.6 |
| Yes | "radial" | 29 | 0.0061 | 0.033 | 3.6 |
| None | "circum." | 31 | 0.013 | 0.066 | 6.6 |
| Yes | "circum." | 32.5 | 0.010 | 0.048 | 4.8 |

At one location, additional vibration measurements were taken at a high frequency. Raw accelerometer readings at 1000 hz were plotted to determine peak accelerations and frequencies. Assuming a sine wave profile, peak velocity and displacement can then be calculated.

## RESULTS

Peak velocity measurements were taken at nine different satellite refrigerator buildings. The average readings are shown in Figure 5. The cryogenic system was functioning nominally, producing 4.5 K refrigeration without the cold compressors operating. The unsupported configuration characterizes the current vibration environment. The supported configuration shows the effect of the box beam column, which was installed at four locations. In general, we see about a 30% decrease in vibration peak velocity with the support.

Comparative information was gained from the high frequency accelerometer readings taken at one location. These results, given in Table 1, agreed well with simultaneous readings from the velocity transmitters .

Some additional observations were made to judge off-nominal conditions. First, a range of potential maintenance activities on the valve box was simulated. A gentle case was modeled by applying pressure on the valve box with finger tips while a severe case was simulated by a man climbing on the valve box. For the unsupported cold compressor, the gentle case increased the vibration peak velocity from 0.076 to 0.15 cm/s, while the severe case gave 0.38 cm/s. With the support installed, these readings were reduced to 0.076 cm/s for the gentle case and 0.10 cm/s for the severe case. Next, vibration measurements were taken while the cold compressor was started, sped up, and stopped. No change in exterior vibration level was seen with or without the additional support.

## CONCLUSIONS

The lifetime of the cold compressor foil journal bearings needs to be increased for greater system reliability. The bearings suffer from exposure to the external environment that may generate about 0.09 cm/s peak velocity at the compressor. Adding a support column to some existing installations reduced these normal vibrations by about 30%. While this method offers some improvement, there may not be a significant increase in bearing life, and the additional structure would be a nuisance for certain maintenance activities.

The recommended action is to redesign the foil journal bearings. State-of-the-art foil bearings may provide longer life. A design study contract has been competitively awarded to a consulting firm, Turbotechnology Services Corporation of Scotia, New York, with expertise in foil bearing design.

## REFERENCES

1. J. D. Fuerst, Selection of cold compressors for the Fermilab Tevatron, in: "Advances in Cryogenic Engineering," Vol. 39a, Plenum Press, New York (1994), p. 863.

2. J. C. Theilacker, Tevatron cold compressor operating experience, *Cryogenics* 34:107 (1994), ICEC Supplement.

3. B. L. Norris, Initial performance of upgraded Tevatron cryogenic system, in: "Advances in Cryogenic Engineering," Vol. 41a, Plenum Press, New York (1996), p. 693.

4. M.R. Drake, private correspondence "Valve box vibrations" (Nov. 3, 1997), Mohler Technology, Boonville, IN.

5. N. K. Arakere and H. D. Nelson, An analysis of gas-lubricated foil-journal bearings, *Tribology Transactions* 35:1 (1992).

6. G. L. Agarawal, Foil air bearings cleared to land, *Mechanical Engineering* 120(no. 7):78 (July 1998)

7. L. San Andres, Turbulent flow foil bearings for cryogenic applications, *Journal of Tribology* 117:185 (January 1995)

8. I. A. Davydenkov et. al., Development of cryogenic turboexpanders with gas dynamic foil bearings, , *Cryogenics* 32:80 (1992), ICEC Supplement.

9. H. Heshmet and C.P. R. Ku, Structural damping of self-acting compliant foil journal bearings, , *Journal of Tribology* 116:76 (January 1994)

10. M. P. Blake, Vibration-part I: its effects, classification, and measurement, *National Safety News* (November 1975).