

DISCOVERING, THE HARD WAY, HOW A HIGH PERFORMANCE COUPLING INFLUENCED THE CRITICAL SPEEDS AND BEARING LOADING OF AN OVERHUNG RADIAL COMPRESSOR—A CASE HISTORY

by

Joseph P. Corcoran

Turbo Group Leader

Kop-Flex, Incorporated

Baltimore, Maryland

Harvey Rea

Maintenance Engineer II

IMC Kalium

Regina, Saskatchewan, Canada

Guill A. Cornejo

Vibration Specialist

Solar Turbines, Incorporated

San Diego, California

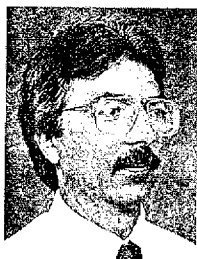
and

Martin L. Leonhard

Mechanics Manager

Atlas Copco Energas GmbH

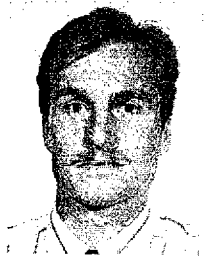
Köln, Germany



Joseph P. (Joe) Corcoran is the Turbo Group Team Leader for Kop-Flex, Incorporated, in Baltimore, Maryland. At Kop-Flex, he is responsible for the engineering and design group that selects, designs, and processes orders for high performance disc, gear, and diaphragm couplings, and monopole sensor torque meters used mainly in the turbomachinery market. He has 12 years experience in power transmission and custom coupling

design and field service at Kop-Flex, and has previous experience as an operations engineer responsible for two 80 ton per day oxygen plants for the City of Baltimore.

Mr. Corcoran has a B.S. degree (Mechanical Engineering) from the University of Maryland. He is a member of the Vibration Institute and ASME.



Harvey Rea has the position of Maintenance Engineer II with IMC Kalium, in Regina, Saskatchewan, Canada. At the Belle Plaine potash mine/refinery, he serves in the roles of maintenance supervisor, project manager, and consultant to the local departments and other IMC facilities in the field of materials selection and fabrication, corrosion control, predictive maintenance, and vibration analysis. He started his engineering career within the

R&D group of a steel manufacturer, joining IMC in 1990.

Mr. Rea received a B.S. degree (Mechanical Engineering) from the University of Saskatchewan (1989) and is a registered Professional Engineer in the Province of Saskatchewan.



Guill A. Cornejo is the Program Manager of the Power Systems Vibration Laboratory at Solar Turbines Incorporated, in San Diego, California. The Laboratory's charter is to analyze, troubleshoot, and verify the health of high speed turbomachinery packages like turbo-compressors, turbogenerators, and turbopumps. The healthy vibration response of a modern turbomachinery package system is the engineered outcome

of its rotordynamics, structural support, and electronic control. Analysis of a package system involves steadystate and transient evaluation of engine/power turbine rotor and gear shaft lateral and torsional vibration, gear/teeth vibration and mesh, and the relationship of this evaluation to process or load demand and package structural vibration response.

Mr. Cornejo received a B.S. degree (Mechanical Engineering) from the University of California at Davis (1975). He also earned an M.S. degree (Mechanical Design) (1977) followed by the Degree of Engineer (1981), specializing in vibration, from Stanford University.

ABSTRACT

A case is presented where significant unacceptable vibration levels were encountered during mechanical testing of a single stage overhung radial compressor used in a mechanical vapor recompression process. The contract high performance disc coupling (modified for use between a test gearbox and the overhung compressor) was one of the main focuses of attention in arriving at a solution to the problem.

Presented is the logic and courses of action taken to eliminate the coupling as the cause of the excessive vibration, and the analysis used to modify the coupling to make it the major part of

the solution. This resolution involved significantly increasing the coupling's weight to change the critical speeds of the compressor rotor/bearing/coupling system, while at the same time improving the damping and stability in the coupling end bearing; enough to allow the train to run successfully.

However, this extra weight coupling solution was not normal, and, except for time considerations, would not have been attempted. So, also discussed is how the actual problem (running at a critical speed without adequate damping) could have been discovered in the design stage or, once the problem occurred, been solved more easily if API procedures and guidelines had been referenced and used.

INTRODUCTION

The compressor package was the scheduled critical path element to the construction of a mechanical vapor recompression evaporator circuit that forms the "heart" of an expansion to IMC Global's Hersey, Michigan, plant (Figure 1), which produces potash and salt products. Delays in the shipment of the "Hersey" compressor unit to Solar Turbines for mechanical integration would directly postpone the commissioning of the entire facility. Therefore, when a vibration problem appeared on the test stand, the need for expediency prompted the direct involvement of the user in its resolution.

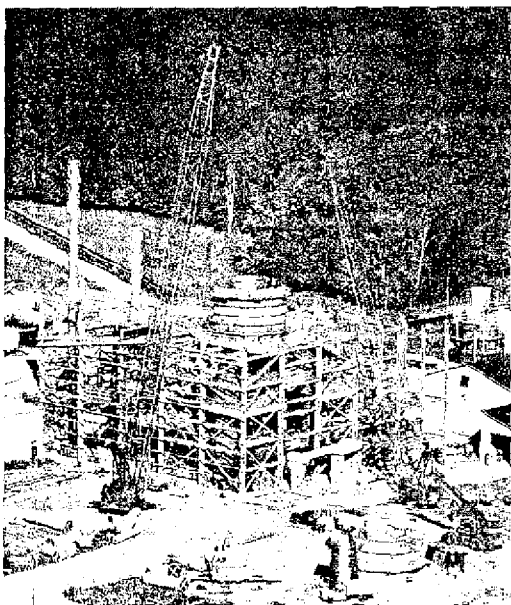


Figure 1. Hersey Plant Site Under Construction.

Although the technical challenge in defining the root cause of the excessive vibration was significant, an equally challenging task was the coordination of the efforts of the three primary manufacturers, the various subsuppliers, the owner, and the process consultants who initiated the equipment orders. The working group assembled to solve the problem included various combinations of seven different firms in 10 geographic locations spanning 12 time zones and included personnel of diverse background.

The process consultant was given responsibility for procurement of the complete gas turbine/coupling/compressor unit (Figure 2). The compressor unit (Figure 3) is a single stage, overhung centrifugal unit, vertically split case with adjustable inlet guide vanes, operating on water vapor at inlet conditions of 280°F and 28.8 psia (198,570 Pa) delivering 303,000 lb/hr (38.2 kg/sec) at 54.72 psia (377,300 Pa) and 423°F. Shaft sealing is via a three chamber floating carbon ring seal, utilizing steam as the sealing medium. The original bearing system was a pair of tilting pad units

consisting of five pads of 52 degrees arc length with no offset, "load-between-pads," and a dual-acting tilt pad thrust bearing integrated with the driven-end (outboard or coupling end) bearing. The inboard radial bearing is 160 mm diameter and the outboard radial/axial bearing is 120 mm. Nominal operating speed is 10,500 rpm steadystate. The drive unit is a Solar Turbines Taurus 70 two shaft gas turbine that delivers a rated 7982 hp (5952 kW) through a Kop-Flex #254 RZ/MS marine style high performance disc coupling (Figure 4).

10500 RPM DESIGN SPEED

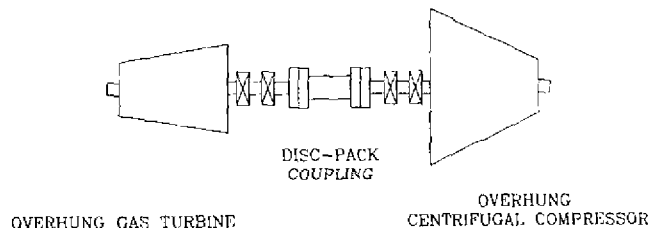
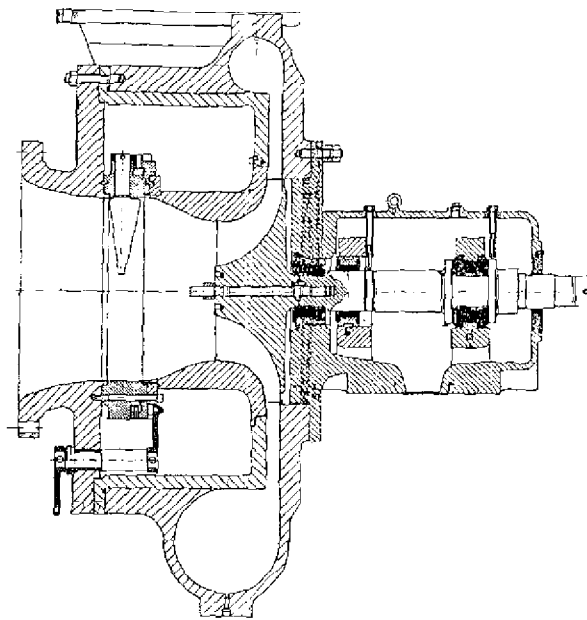


Figure 2. Contract Train.



SINGLE STAGE DIRECT-DRIVEN TURBOCOMPRESSOR

Figure 3. Overhung Steam Compressor.

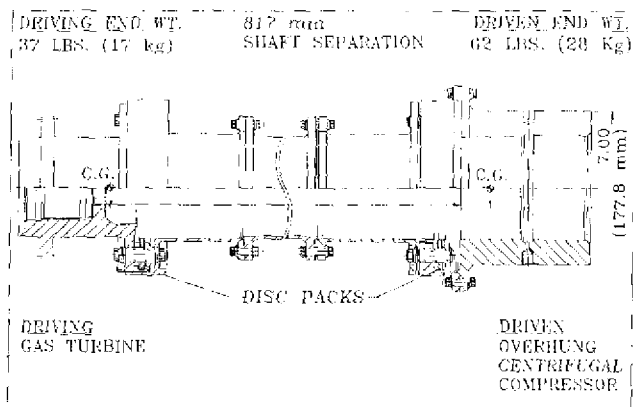


Figure 4. Original Contract Coupling.

The justification for the order was the result of a review of an existing facility that utilized a similar but smaller compressor in a comparable service. The physical scale of the similar unit was approximately 75 percent of the "Hersey" machine output. This newer first-article compressor featured a larger titanium impeller diameter (scaled up from 600 mm to 980 mm) and lower process speed, 10,500 rpm vs 15,000 rpm.

The order for the package was not to a single vendor. Rather, separate orders were placed for the compressor and for the turbine, and a subsequent order was placed for mechanical integration by the turbine manufacturer (who was to provide the job coupling complete with a slave adapter suitable for the final mechanical test at the compressor manufacturer's facility). API specifications were not designated for the main equipment train. Therefore, allocation of "unit" responsibility was not fully clarified and was defined by each manufacturer's scope of component supply. This differs from API 617 "Centrifugal Compressors for Petroleum, Chemical, and Gas Service Industries" [1], which assigns train responsibility to the compressor manufacturer. Also, in API 617, the compressor manufacturer is typically responsible for supplying the coupling.

In this case, however, the compressor manufacturer was only responsible for the compressor and its lateral analysis. This included a stiffness map of the undamped compressor rotor response, a plot of the mode shapes for undamped criticals, and a damped unbalance response of the compressor for minimum/maximum bearing clearances and lube oil temperatures. A speed stability plot showing log decrement vs operating speed, mass elastic data, shaft details, and thermal growth data were also requirements.

The turbine manufacturer, on the other hand, was to provide lateral and torsional analysis of the entire train, but only as a final deliverable. In fact, the turbine manufacturer's lateral analysis was not completed until after the compressor was built and the problems on the test stand were rectified.

The coupling manufacturer sold the coupling to the turbine manufacturer and supplied with it the standard drawing and mass elastic data, including half weights, total inertia, and center of gravity locations for each end of the coupling (Figure 4). Even though API procedures were not specified for the rest of the train, API 671 "Special Purpose Couplings for Refinery Service" [2] was specified for the coupling by the turbine manufacturer. Because of this, records of balancing and manufacturing of key locating dimensions were readily available and aided in the problem resolution.

However—and this is one of the keys to the vibration problem—because of time considerations, the compressor manufacturer used assumed preliminary coupling data (weight and center of gravity) in their original lateral analysis, based on previous compressor designs. They did not use the data that were supplied to the turbine manufacturer, until it was realized that the coupling as delivered was much lighter than that modelled in the lateral analysis.

This problem could have been avoided if API 617 had been specified. Even though the user thought that the basic specification for the machine should have been API 617, the compressor manufacturer used API 672 "Packaged, Integrally Geared Centrifugal Air Compressors for General Refinery Service" [3] as a guideline. However, again, neither one had actually been specified by the user contracted packager.

API 672 allows the use of previously done critical speed analysis (with purchaser approval). Because the compressor manufacturer had made the same frame machine before, they used similar data in the lateral analysis; i.e., coupling data that were not correct. The relevance of this will be discussed later.

MECHANICAL TEST PROCEDURE

The mechanical test procedure for the compressor was according to the vendor's internal standards that, again, were largely based on API 672. It was to be performed at 89 percent of

operational speed (9340 rpm) due to limitations of the test facility. The test setup consisted of a 400 kW DC variable frequency motor driving a Cardan shaft into a double helix, parallel-shaft increasing gearbox (1:3.11 ratio). The gearbox, in turn, mated to a slave adapter that drove the spacer and driven portion of the final site high performance disc pack coupling (Figure 5), and the overhung rotor compressor. A schematic of the test stand drive train with its various components is presented in Figure 6.

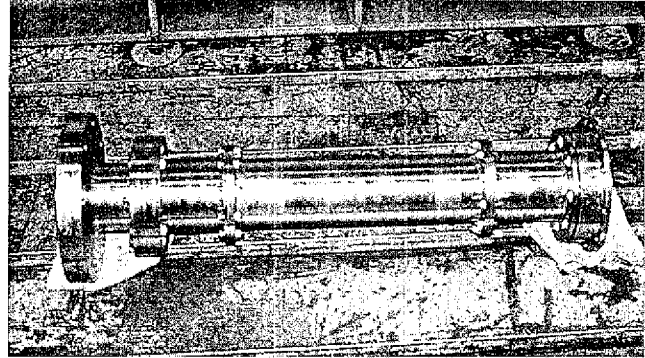


Figure 5. Modified Contract Coupling for Test. (Less compressor rigid hub (right side), with slave adapter (left side).)

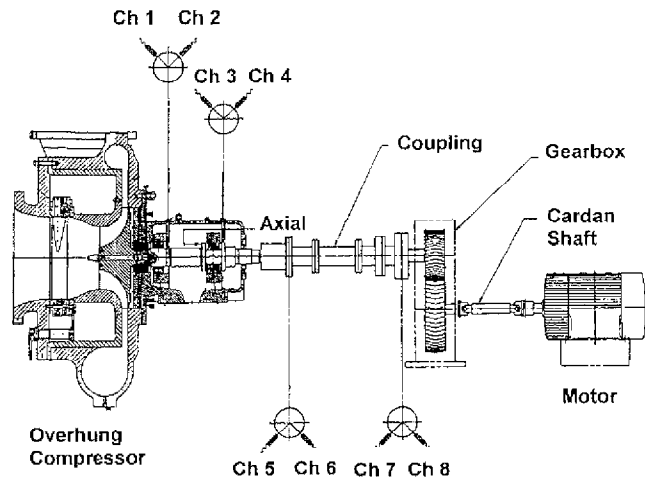


Figure 6. Original Test Arrangement.

The compressor volute was evacuated to a vacuum of approximately 100 mbar to reduce required power. Initial instrumentation was via two sets of proximity probes, one at each compressor bearing, which were specified for online monitoring to provide signals to industry standard hardware/software. Speed was given by a one/rev tachometer. Proposed deliverables included a reasonable array of FFT analysis, Bode plots, polar plots, and shaft centerline information. In addition, other data on the lubrication system, volute, and bearing temperatures were to be provided. The allowable overall direct vibration at the bearings was 35.2 μm that included an allowable electromechanical run out of 6.4 μm . This procedure emulated the testing of the existing similar unit that had been commissioned without incident.

PROBLEM DISCOVERY— UNBALANCE IS SUSPECTED

Initially, the vibration problem presented itself in what appeared to be a relatively straightforward manner. As the test speed of 9450 rpm was approached, the overall direct vibration (uncompensated) rose to values sometimes greater than 100 μm at around 8500 rpm and the tests were aborted. The machine was not operated above

this 8500 rpm limit. The 1× filtered vs direct amplitude was comparable with little or no phase shift. FFT analysis indicated a clear 1× response that was initially interpreted as simply an unbalance condition. A typical set of system responses for these initial observations is shown in Figures 7, 8, and 9.

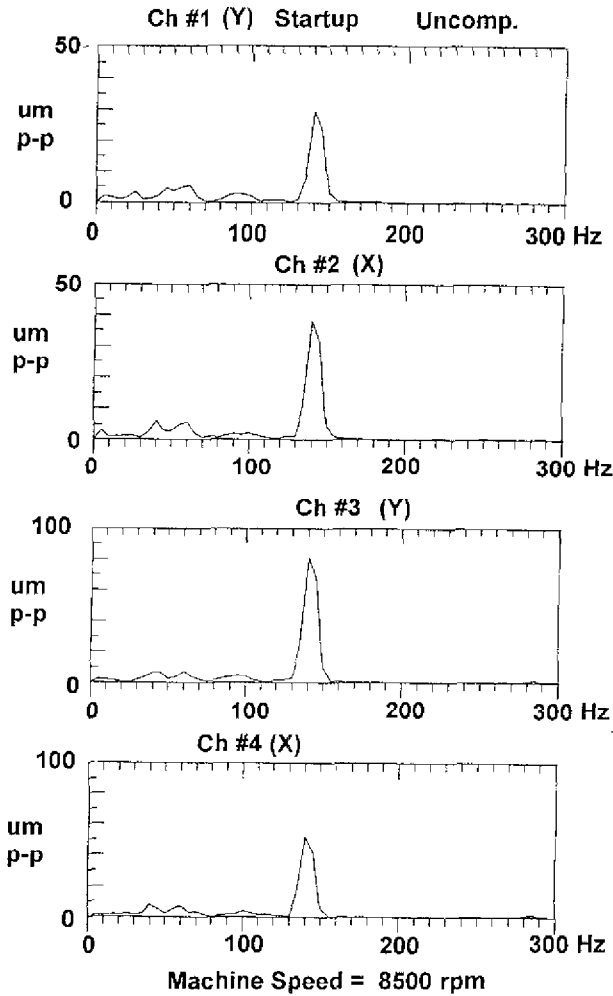


Figure 7. Original "Unbalance" Spectrum.

AN IMPORTANT DISCOVERY

Previous to this first test run, when an unexpectedly light coupling was delivered to the compressor manufacturer, a new lateral analysis was done with the actual contract coupling weight. The analysis had been done originally using a coupling mass of 75 kg at the outboard compressor shaft end (rotor A, Table 1). The model for rotor A is shown in Figure 10. Depicted in Figure 11 is the forced damped unbalance response mode shape at 10,500 rpm, with an unbalance of 1.06 kg-mm at the coupling end of the rotor, and a resultant deflection amplitude of 16 μm (0-p). Lateral critical speeds, based on the unbalance response for the compressor, were originally predicted at 2750, 6250 (Table 1, column A), and 26,000 rpm.

The contract coupling as delivered, however, was a standard design weighing 28 kg at the shaft end. The actual weight was somehow missed in the communication between the compressor and turbine manufacturer, as mentioned previously. The rotor model (rotor B) and forced damped unbalance (using the same unbalance of 1.06 kg-mm at the coupling end) response model and mode shape is shown in Figures 12 and 13, respectively. Calculated critical speeds and the amplification factor are shown in Table 1, column B. The first critical speed was at 3800 rpm and the second was at 9100 rpm, very near the problem speed on the test stand!

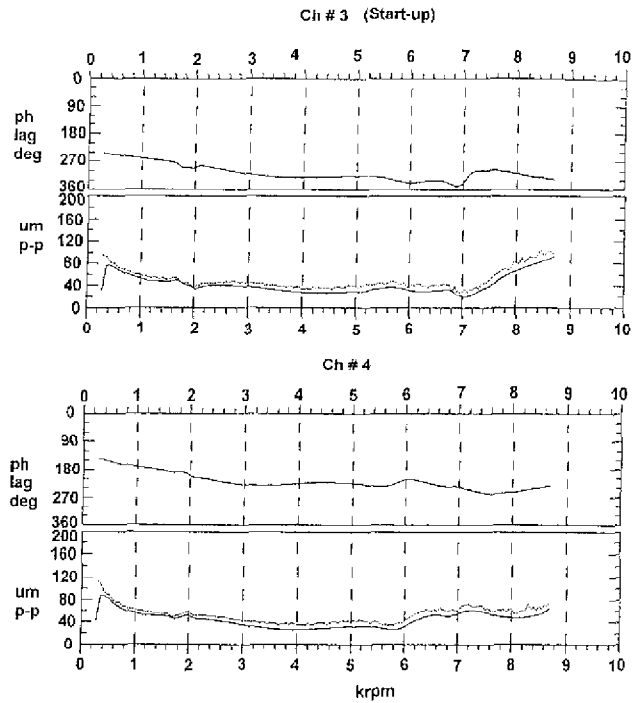


Figure 8. "Unbalance" Bode Plots.

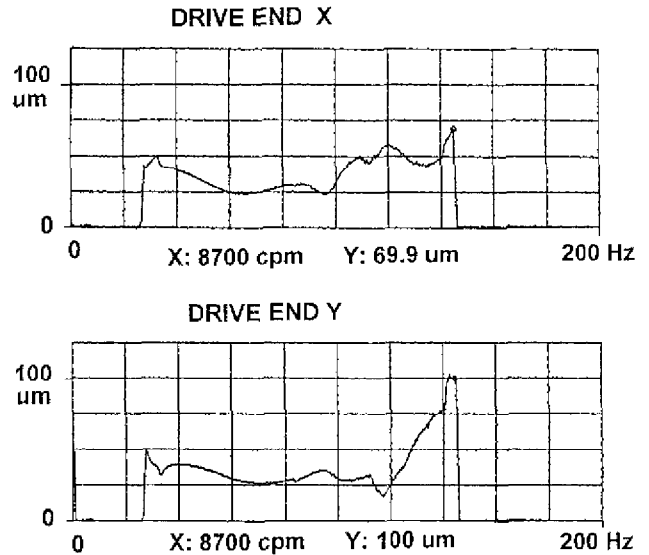


Figure 9. "Unbalance" Spectrum.

Table 1. Lateral Vibration Analysis for Different Coupling Configurations.

	ROTOR A	ROTOR B	ROTOR C	ROTOR D	ROTOR E
Coupling Design	Original analysis	Actually Delivered	Shop test coupling	Shop test coupling w/ weight ring	Final heavy contract coupling
Comp. End Coupling Mass (kg)	75	28	2.7	30.2	73.6
Total Comp. Rotor Mass (kg)	483	406	381	Note:	Note:
1st Resonance Speed (rpm)	2750	3800	3950	All data similar to	All data similar to
Separation Margin (%)	74	64	62	Rotor model B	Rotor model A
Amplification Factor	1.2	1	<1		
Mode Shape	Conical	Conical	Conical		
2nd Resonance Speed (rpm)	8250	9100	12,800		
Separation Margin (%)	40	13	22		
Amplification Factor	1.3	1.2	1.2		
Mode Shape	Cylindrical	Cylindrical	Cylindrical		

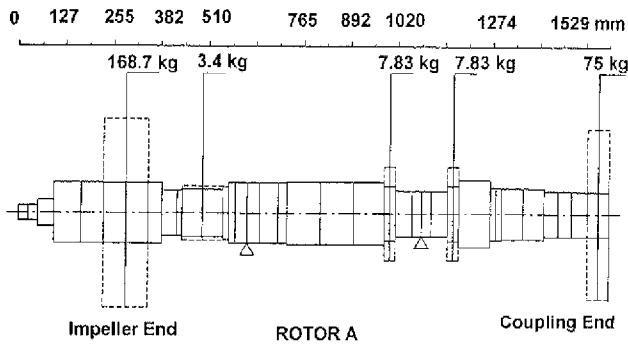


Figure 10. Rotor Model A—Design Model with Design Coupling Weight.

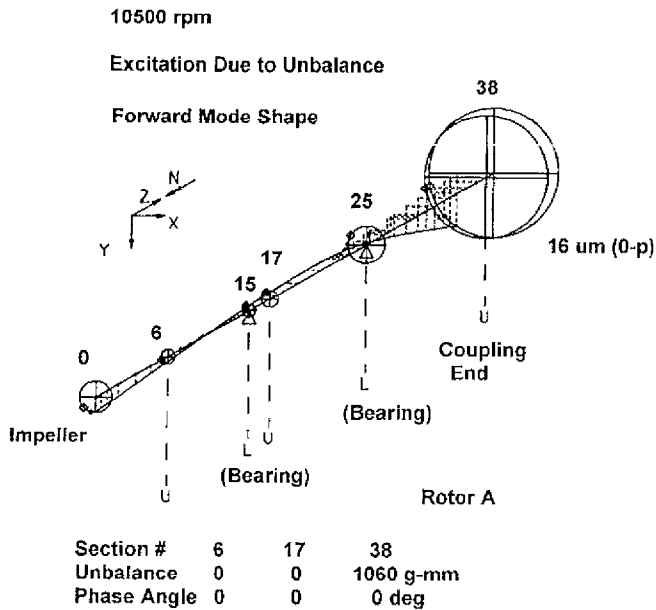


Figure 11. Model A Unbalance Response Mode Shape.

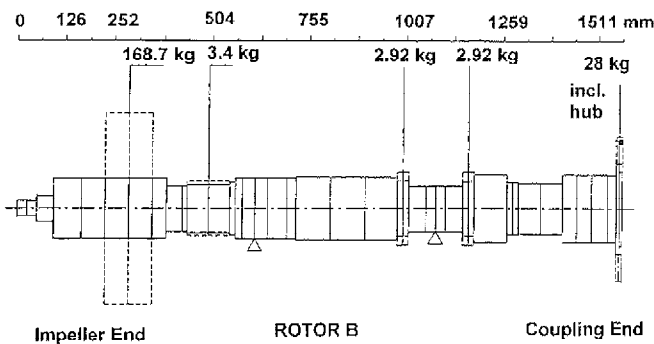


Figure 12. Rotor Model B—With Actual Contract Coupling Weight.

But, even though the observed test resonance was very likely related to the new rotor (rotor B) model second critical, the rotor had been calculated to be critically damped, with an amplification factor of 1.2 (Table 1). Damped Eigen value stability analysis showed no problems. Log decrement values were well in the stable range. The high amplification was completely unexpected and more investigation was necessary.

During the first test runs after the rotor rebalance, the magnitude of vibration (amplification factor $\gg 20$) prohibited passage

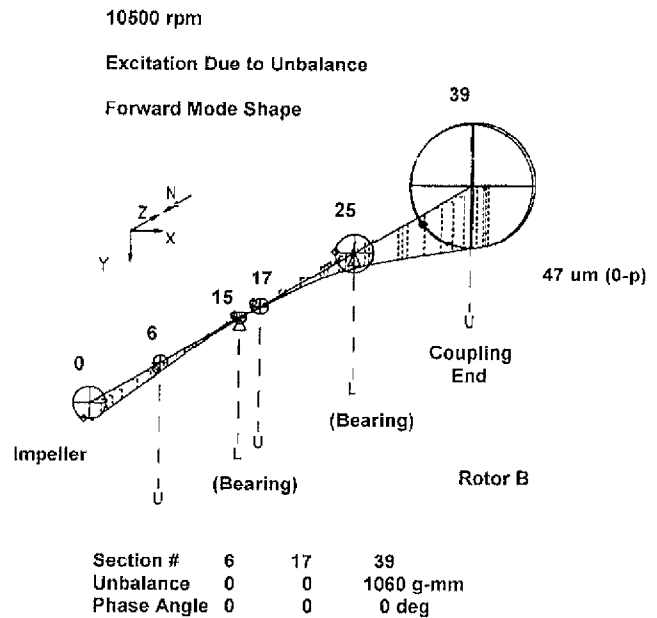


Figure 13. Unbalance Response Mode Shape for Model B.

through the resonance. The observed vibration magnitude was up to 250 μm with a corresponding phase shift. This result appeared to be only the initial onset of the resonance. The machine could be run up to 9300 rpm, with stable amplitudes at 40 μm to 60 μm with no phase shifting. But once the speed was increased above 9400 rpm, the amplitude would increase dramatically. Depicted in Figures 14, 15, and 16, are the nature and magnitude of the vibration.

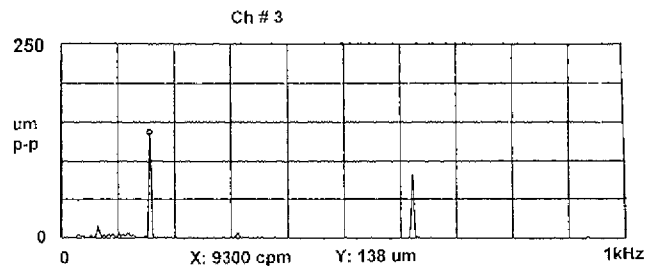


Figure 14. "Resonance" Spectrum.

A subsequent series of trials was attempted to probe the extent and magnitude of the resonance, but they were limited by the calculated bearing clearances. During this exercise, many parameters were altered focusing on the bearing system, coupling axial preloads, and alignment and a plethora of vibration, physical, and thermal measurements intended to isolate the source of the resonance. The inevitable excursions beyond calculated bearing clearance limits of 230 μm (original minimum design clearance) required many iterations of equipment disassembly to inspect for bearing damage, and cost precious time.

Since the calculations showed that there should not have been a problem, even close to a resonance, and since time was such a critical issue in this project, anything and everything were suspected as a possible source of the unexpected resonance amplification.

As part of this initial investigation, it was also theorized that a source originating within the test setup could be the source of the unbalance or was forcing a resonance within the system. The drive motor, Cardan shaft, test gearbox, test lubrication system, and test bed foundation were all considered, and eventually eliminated through the battery of tests conducted. Summarized chronologically

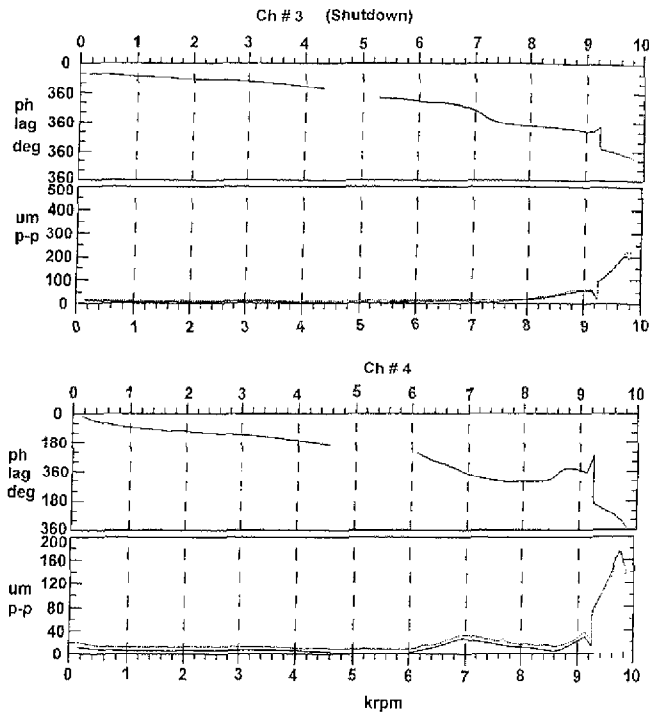


Figure 15. "Resonance" Bode Plots.

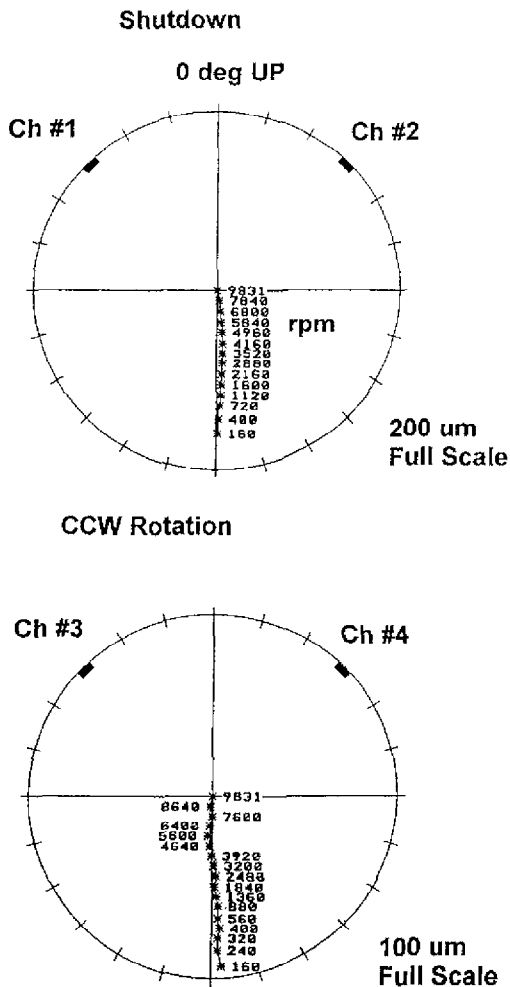


Figure 16. "Resonance" Shaft Centerline Plots.

in Table 2 is a small sample of many of the troubleshooting changes to the system, and the subsequent test runs and unacceptable results. The timeframe is from after an unbalance problem was suspected, to when a simulation of the original analysis heavy coupling was attempted. Note that the amount and type of testing shown give an indication of how unexpected the vibration problem was.

Table 2. Chronological History of Some of the Original Troubleshooting.

ACTION	DESCRIPTION
1	Change of maximum eccentricity of the rotor from 0.1 to 0.012 mm.
2	Single step low speed rotor rebalance.
3	Reduce the radial clearance of the radial tilting pad bearings from s/D = 0.21% to s/D = 0.14%.
4	Review assembling process of the core unit.
5	Do a free mode natural frequency check of the spare rotor unit.
6	Set up the complete machine in the testbed.
7	Recheck alignment.
8	Install an axial probe.
9	Install velocity probes to analyse case vibration.
10	MECHANICAL TEST RUN/Result: Max vibration @ coupling end bearing is 125 um at 9450 rpm with continuous phase changing.
11	Install dial indicators at the bearing housing to check thermal displacement.
12	Install vibration probes at the fixed parts of the contract coupling.
13	MECHANICAL TEST RUN/Result: Max vibration @ coupling end bearing is 125 um at 9450 rpm with continuous phase changing.
14	Change the axial displacement of the coupling to 0.0 mm cold.
15	MECHANICAL TEST RUN/Result: Max vibration @ coupling end bearing is 190 um at 9300 rpm with continuous phase changing.
16	Install different cardan shaft coupling between variable speed motor and gearbox.
17, 18	MECHANICAL TEST RUNS (2)/Results: Max vibration @ coupling end bearing is 100 um at 9310 rpm with continuous phase changing. FFT analysis inconclusive.
19	MECHANICAL TEST RUN without the compressor connected to the gearbox.
20	Increase radial bearing clearance at the coupling end from 0.14% to s/D = 0.21%.
21	MECHANICAL TEST RUN/Result: Max vibration @ coupling end bearing is 200 um at 9315 rpm with continuous phase changing.
22	Decrease radial bearing clearance at the coupling end from 0.21% to s/D = 0.14%.
23	Install different intermediate gearbox (ratio 1:5).
24	MECHANICAL TEST RUN/Result: After running for .5 hr @ 7000 rpm bearings overheated (100 C).
25	Check bearings visually; no problem noticed.
26	MECHANICAL TEST RUN/Result: Vibration reduced to 45 um maximum, but bearing temperature up to 122 C.
27	Second oil system hooked up.
28	MECHANICAL TEST RUN/Result: Vibration reduced to 45 um maximum, but bearing temperature >120 C.
29	Bearing damage noted and incorrect assembly of the bearing support detected.
30	Install new bearings.
31	Add a mass ring to the coupling to simulate original model.
32	MECHANICAL TEST RUN etc. etc. etc.

The only known fact was that the resonance was too large in amplitude to allow passage within 70 percent of calculated bearing clearance and certainly too close to operating speed to allow for an adequate separation margin. Following are summaries of the main theories behind many of these initial tests and actions, some of which are documented in Table 2, all done in an attempt to discover the source of vibration.

- Prior to the recognition of the severity of the resonant condition, i.e., when a simple unbalance was suspected as the source of the problem, the following was done:
 - Compressor rotor balance check—Each component of both the primary and spare rotating element was disassembled, inspected, measured, and rebalanced to a maximum residual

unbalance quality of ISO 1940 Gr. 1.0. The rotating element was step assembly balanced starting with the bare shaft, shaft with coupling contract rigid hub, and completed with the impeller. Typical values for separate components during each of the numerous rebalancing procedures were in a range of 0.4 mm/s (65.05 g-mm) to 0.9 mm/s (165.4 g-mm). The complete assembly had a mass of 375 kg plus 18 kg for the rigid coupling hub.

- *Rotor manufacturing tolerance check*—The manufacturing tolerances of each component of the rotating element were confirmed separately and as an assembly. As an example, one theory was that a measured eccentricity of 100 μm on one segment of the shaft was a source of the vibration. However, improvements to this had no significant effect. The final shaft mechanical TIR was typically 3.2 μm to 5.4 μm on bearing surfaces and 12 μm relative between bearing surfaces.

- Once the resonant condition was recognized, the focus turned to isolating the source of the resonance and/or resonance amplification via the following measures:

- *Campbell diagrams frequency check*—Campbell diagrams were developed for the complete rotor in the free-free mode and compared with calculated values. These were eventually rechecked with no relevant frequencies being noted.

- *Motor frequency check*—The variable speed DC motor was theorized as a driving force to the test train resonance. Numerous readings throughout the train did indicate low transmission levels of the motor fundamental frequency, but these were not considered significant.

- *Cardan shaft*—The Cardan shaft (Figure 17) was also theorized as providing a source of the vibration. Following checks of the original unit balance records that showed an acceptable residual unbalance, another unit of approximately double the mass and physical size was substituted, with no significant influence. Therefore, this component was eliminated from the discussion.

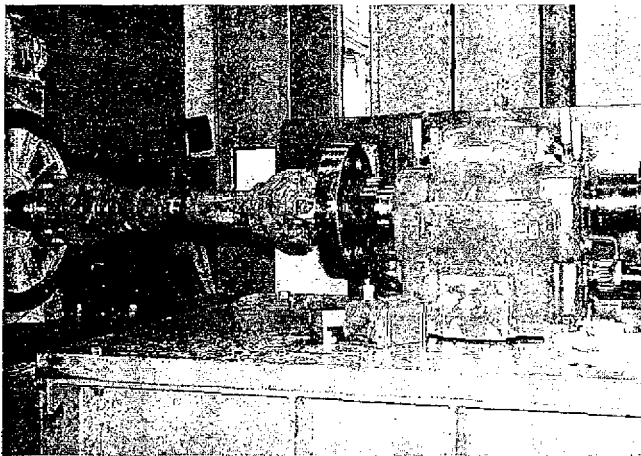


Figure 17. Original Gearbox and Cardan Shaft.

- *Test gearbox*—The test gearbox was suspected as inducing the vibration. The original unit, also depicted in Figure 17, was a double helical unit of 1:3.11 ratio used to increase the available output speed of the DC motor. The gearbox generated some relatively high vibration readings. However, attempts to isolate this source of the resonance were inconclusive following disassembly and inspection of gearing and bearings, mechanical runout measurements and rebalancing of both input and output shafts, and gearing. Analytical methods incorporating the characteristics of the output pinion into the lateral/torsional model also showed no significant influence. A high level subsynchronous vibration (0.45 \times to 0.49 \times) was eventually isolated within the output shaft

bearing system. This gearbox was replaced with a larger capacity unit of 1:5 ratio that would allow operation at the full operating speed of 10,500 rpm and is shown in Figure 18. There was no measurable change to the vibration characteristics. Measurements of radial and axial vibrations, thermal growth, and mechanical runouts on input shafts, output shafts, bearing systems, and casing provided no relevant information. Due to the lack of system response following the gearbox replacement, this component was eliminated from the investigation.

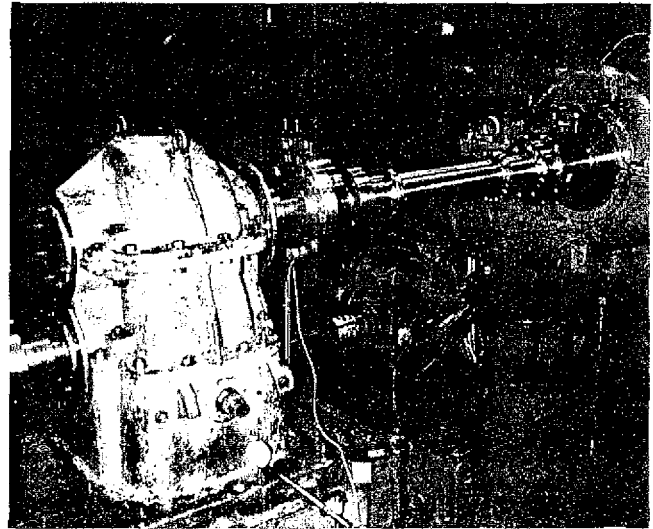


Figure 18. Replacement Gearbox.

- *Test bed subframe resonance*—The possible resonance of the test bed subframe was investigated, but showed no relevant resonance or transmission of vibration.

- *Overhung bearing pedestal design*—A possible structural resonance was theorized within the overhung bearing pedestal design. The housing is a horizontally split design that allowed for impact testing of the unit in various stages of assembly and disassembly. However, the webs carrying the bearing systems are quite massive; therefore, the resultant Campbell diagrams showed no significant resonant frequencies. The fasteners mounting the bearing caps were eventually modified to provide a stiffer bearing assembly, but the effect was not quantifiable. This assembly is shown in Figure 19.

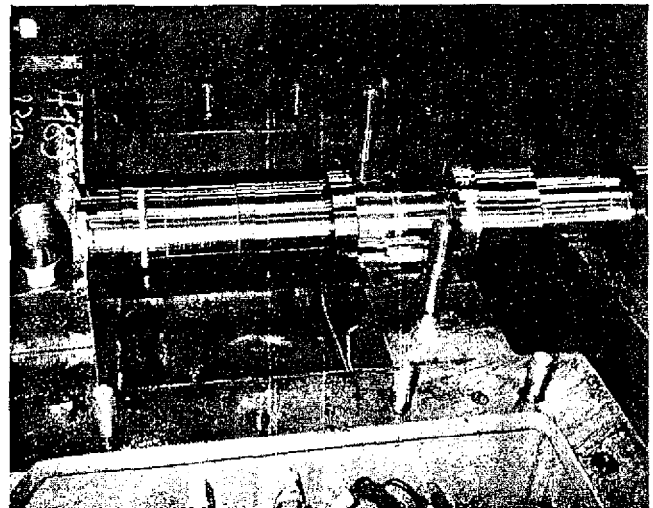


Figure 19. Bearing Pedestal.

- *Probe resonance*—The resonant frequencies of the relatively long proximity probes, mounted to the upper half of the bearing pedestal casing, measuring the bearing locations were questioned and checked both via calculation and impact tests with no apparent relevance.

- *Probe mounting location*—The mounting point of the proximity probes was moved from the upper half of the bearing pedestal cover to the bearing caps. During the final trials, it was noted that accelerometer readings on the bearing pedestal casing were out-of-phase with the revised proximity probe location, which explained many of the previous shaft orbit readings that did not make sense. The effect of this change was never quantified, but obviously influenced, the data.

- *Lubricant drainage*—The drainage of lubricant from a cavity outboard of the outboard bearing was questioned, in a theory that inadequate drainage would create a condition of excessive lubricant level in the outboard radial/axial bearing. This level was measured under various conditions with no evidence to support the theory.

- *Bearing system*—The bearing system damping and stability was in question from the beginning, but the rotor and bearing analysis showed that there should not have been a problem. In order to try to control the excessive vibration, the bearing parameters of lubricant viscosity and internal clearances were modified in an attempt to maximize available damping, given the constraints on minimum clearance, operating temperatures, and lubrication requirements. The influence of the bulk lubricant temperature and pressure were shown to have little effect.

Numerous trials altering the internal radial clearances were attempted with some success. For example, in the first trial run after the rotor was rebalanced, radial bearing clearances were reduced from $s/d = 0.21$ percent to $s/d = 0.14$ percent to get more stiffness than was originally required by calculation. Other checks were made. However, the clearances required to achieve a significant improvement created overheating conditions within the bearing.

OVERHUNG BEARING STATIC LOAD TEST

To check the original lateral model, a test was devised whereby an additional mass, a weight ring (Figure 20), was added to the contract coupling hub. Although the results (Figure 21) were promising, in that the resonant center frequency was decreased to 7000 rpm, the amplification factor was still too high (between 9 and 10). The ability to manufacture the ring to acceptable tolerances (for balance purposes) was cited as part of the reason for the high amplification factor. In other words, in order for a coupling/weight ring to run properly, tight tolerances on the locating diameters between the coupling hub and the trial weight were required, but were not achieved. It was at this point that thought was given to manufacturing a new half coupling that had the extra weight used in the original rotor lateral analysis.

It should also be noted here that there appears to be jump phenomena depicted on Figure 21, which indicates a highly nonlinear condition. However, it was assumed at the time, possibly incorrectly, that the odd vibration behavior was related to the unbalance in the weight ring assembly.

FOCUS ON THE COUPLING

Because of the lack of success in isolating the source of resonance elsewhere, and because calculations of the various bearing modifications and interpretations of some of the testing results showed that there should have been sufficient damping, focus was placed on the coupling as the cause. A mechanical, quality, or design flaw was suspected as the reason that the compressor rotor/coupling/bearing system did not behave as expected. Possible defects or problems included an axial natural

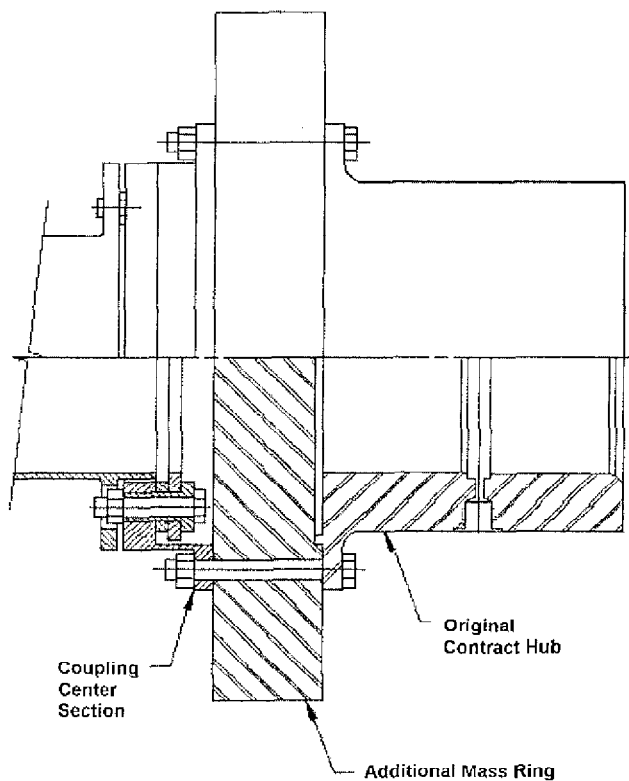


Figure 20. Original Test Coupling with First Trial Weight.

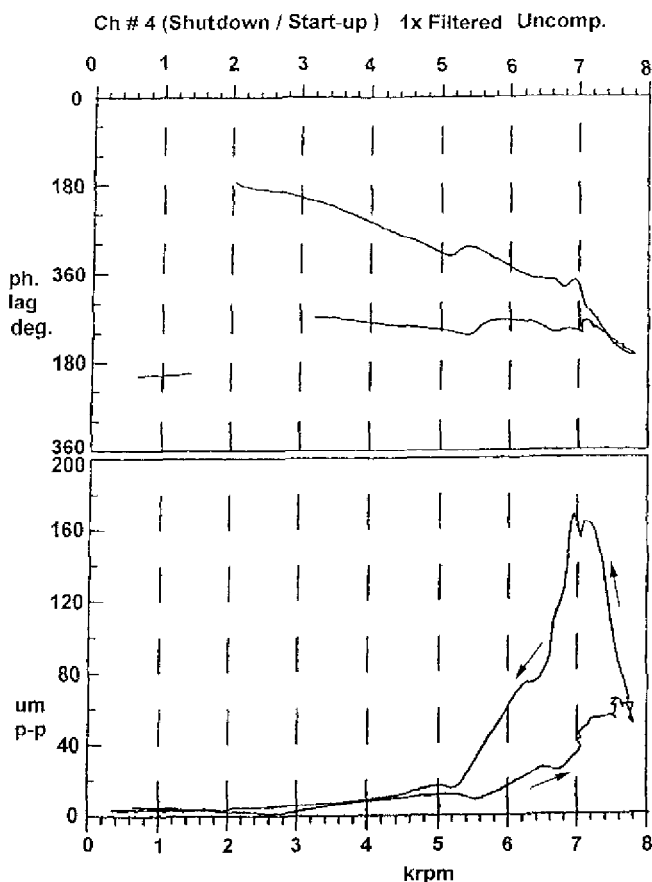


Figure 21. Bode Plot of Original Test Coupling with First Trial Weight.

frequency interference, an excessive amount of unbalance, and/or “shifting,” or changing of the balance quality or stiffness characteristics of the disc pack flexible elements at high speed.

Axial Resonant Frequency

A coupling axial frequency resonance had already been ruled out because the vibration occurred irrespectively of the axial position of the coupling. Trials that varied between the recommended extension to allow for expected thermal growth, the neutral position, and the compressed position showed no influence (Table 2). The thermal growth of the test setup, measured in a multitude of locations, provided no significant findings. Any influence of the coupling axial preload was also ruled out for the same reasons.

Because the disc coupling has a nonlinear stiffness, its axial stiffness and, therefore, its axial resonant frequency changes with deflection. If the coupling was vibrating axially at a frequency, the vibration characteristics would have changed when the coupling was deflected in a different position. They did not. Also, historically, these types of disc couplings do not have problems with axial frequencies. The spring rate is nonlinear and they are self-damping [4].

Alignment

Coupling alignment was also considered critical and various trials—one of which is shown in Table 2—had already been conducted to determine the sensitivity of the system to slight misalignments. Different combinations were tried, including preloading the compressor outboard bearing system in the vertical direction, while operating the lube system at operating pressure and temperature. Also, aligning the system with and without the volute evacuated was tried. The different alignments did not significantly affect the vibration pattern.

Coupling Lateral Natural Frequency

The lateral natural frequencies of the coupling system had been checked against the range of calculated values via impact testing. There was general agreement with those values and, hence, a problem here had previously been ruled out.

Coupling Lateral Shifting at Speed

In addition, it was theorized that the coupling was displacing in a lateral mode at speed, thereby creating an eccentric condition. Proximity probes had been already mounted to record the lateral vibrations of the coupling on either side of the two disk packs, with no conclusive evidence of a resonance condition caused by the coupling.

Cross Coupling

Normally, the rotor lateral analysis is modelled assuming the coupling can be divided into two distinct half sections, and that each rotor is isolated from the other by the coupling flexible elements. A more thorough test stand rotor analysis (with the compressor and the gearbox rotor) was done using the coupling flexible element bending moments and lateral stiffnesses (Figure 22), but no significant difference was calculated between the models.

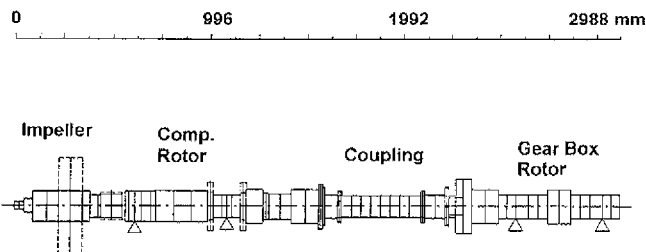


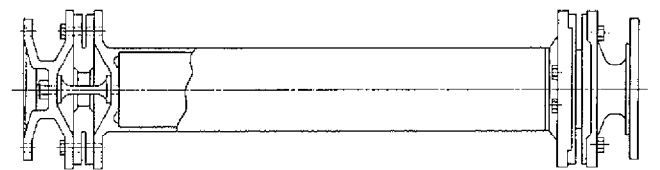
Figure 22. Test Train Model.

Again, because of the lack of success in isolating the problem elsewhere, some additional attempts were made to show that the coupling was the problem:

- *Test without impeller*—The first of these was one where the impeller was removed in an attempt to isolate the coupling as the source of the resonance. It was thought that by removing the impeller, the rotordynamic characteristics of the rotor would be changed to such an extent that, if there were still the same vibration behavior with the same coupling mounted, then the problem must be in the coupling. In other words, if the rotating element was altered drastically and the result was similar, then the constant, being the coupling, must be the source.

The vibration characteristics were, in fact, very similar to that depicted in Figures 14, 15, and 16, and led some to the conclusion that there was a coupling problem. However, others questioned if the conclusion derived from this result was valid. There were other possible explanations—such as an overhung weight critical speed interference without enough bearing damping. In addition, the removing of most of the static load from the bearing system could, in itself, promote instability by altering the design conditions at the tilting pad bearing. Further testing was requested by the user and the coupling manufacturer.

- *Shop coupling test*—A different, shop test coupling (Figure 23) was balanced, then connected to the compressor rotor and spun. Again, the reasoning was that if the newer coupling worked and the contract one did not, there must likely be something wrong with the contract coupling.



250 Nm Lightweight Test Coupling

Figure 23. Shop Test Coupling.

This arrangement was able to run successfully. The overall vibration was within API levels, there were no phase changing effects at speeds up to 10,900 rpm, and there was no significant amplification factor during shutdown (Figure 24). But, as before, there were questions about the validity of using these test results to deduce that the contract coupling was the problem. First of all, the couplings were completely different designs with different physical characteristics.

The biggest distinction was the weight. The new test coupling weighed 2.7 kg and the contract coupling weighed 28 kg on the compressor end. In addition, the differences in flexible element bending, axial, and radial stiffnesses were significant. In other words, since there were all these dissimilarities that could affect the forces on, and therefore the vibration of, the end of the rotor shaft, the fact that the new coupling ran successfully could not prove that contract coupling was defective.

The most likely interpretation of the actual result was that the reduced coupling mass shifted the resonant center frequency above the operating speed, as predicted by the analysis of rotor C. Calculated resonances, separation margins, and the amplification factor are shown in Table 1, column C.

It can be seen here that the overhung mass of a coupling can have a significant effect on rotor critical speeds, as noted by Wolford [5]. The effect of the reduced overhung test mass was not only to raise the second critical speed, but also to raise the ratio of the second to first critical, as predicted by Weaver [6].

These results prompted the question of whether a lightweight coupling system could be designed with an adequate torque rating, in order to shift the resonance above the operating speed. An

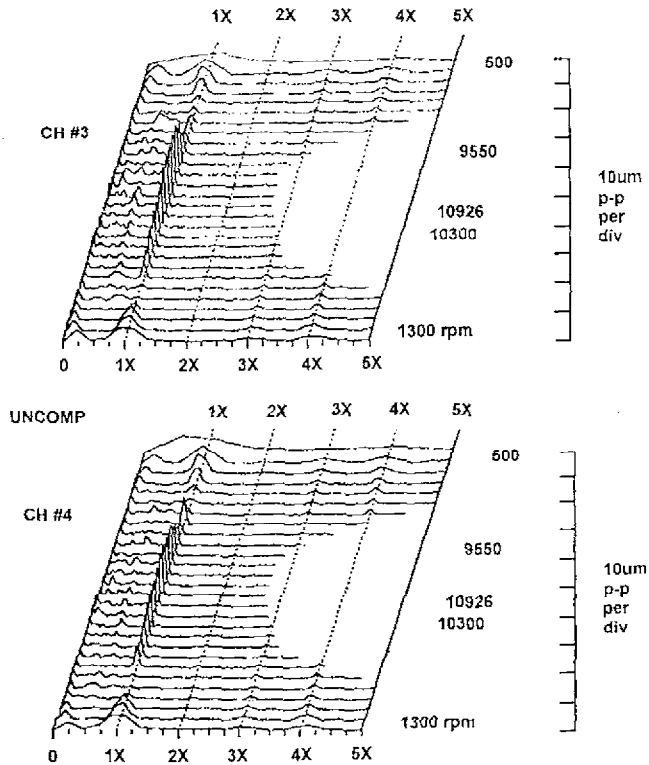


Figure 24. Shop Test Coupling Waterfall Diagram.

attempt was made to redesign the contract coupling to one that had similar weight and center of gravity characteristics (center of gravity closer to the compressor bearing) of the test coupling. This proved to be futile, because the difference in weight between the couplings was so great that a practical contract coupling design that could still handle the torque could not be achieved.

- *Shop coupling with weight ring*—It was decided then to retest the newer coupling arrangement with a weight ring attached to it, to match the effective overhung moment of the original contract coupling (column D of Table 1).

This was the *critical* test. Even though there were questions about how effectively the newer test coupling with weight ring could model the contract coupling—again considering, for example, the different flex element stiffnesses—it was pretty much decided by all parties that if this weight ring arrangement (rotor D) ran successfully, the original contract coupling design and manufacturing quality would have to be *seriously* reviewed. If it did not run successfully, the rotordynamics of the coupling/bearing/rotor design and, especially, bearing damping and stability would have to be questioned. This test would be key in determining the future direction of the problem solving efforts.

As it turned out, the test coupling with the simulated weight could *not* be run acceptably on the initial tests. The vibration levels were too high. In addition, the vibration “shifted” with abnormal phase changing around 9000 rpm to 9500 rpm, and the amplification factor was very high ($\gg 20$), all the same vibration characteristics as with the original contract coupling.

More modifications were done to the bearings and housing—described later—so that this arrangement was eventually able to run successfully.

However, since the initial rotor D test failed, and the attempt to come up with a *lightweight coupling design that could handle the torque loading failed*, and, most important, since the *user* did not want to put a “sensitive” unit into the field, it was decided to duplicate the original rotor model weight for the half coupling (75 kg). This would achieve a higher static bearing load (in case

stability was a problem), and lower the center frequency of the resonance to create a greater separation margin. Improved balance and manufacturing tolerances, compared with the very first trial weight test with the contract coupling, would also be required.

Coupling Rebuild and Rebalance

The entire coupling system was returned to the manufacturer where it was inspected, measured, and reassembled with a new half coupling with a mass of 75 kg to match the *original specification* shown in Table 1. The extra weight (47 kg) was incorporated in a new rigid hub design (Figures 25 and 26). This new hub was termed “the pregnant hub.”

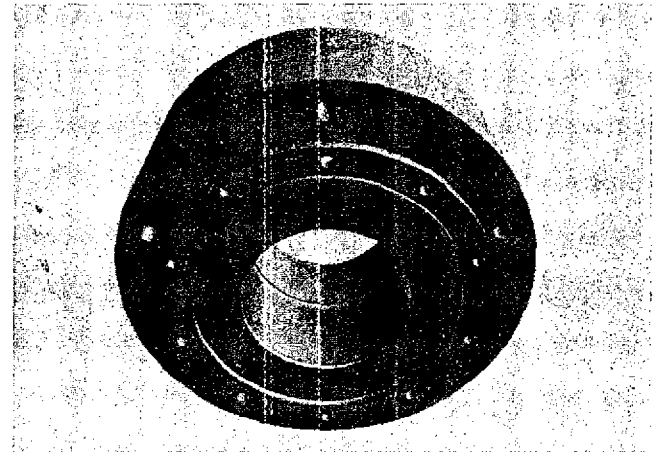


Figure 25. Heavy Hub. (Front view showing mating flange and threaded field balance holes.)

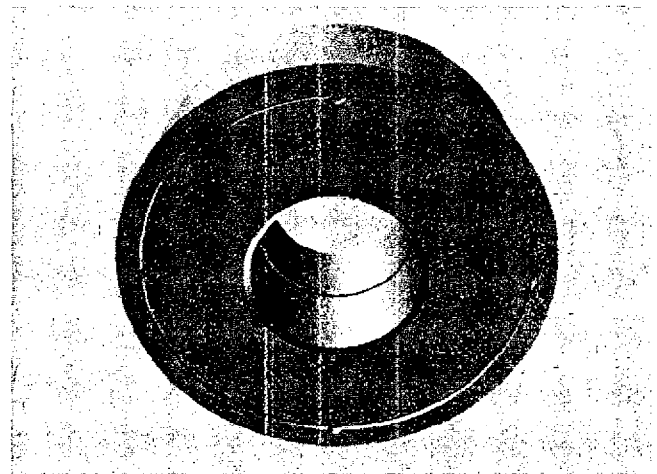


Figure 26. Heavy Hub. (Back view showing field balance groove.)

After manufacture, the heavy hub was balanced with the rest of the contract test coupling that had been returned to the coupling factory. Careful attention was paid to the coupling balance and a balancing procedure was performed with tighter than normal tolerances. Coupling locating pilot and bore runouts were also kept to a minimum.

This was to ensure that if the new heavier coupling were, somehow, to be operating still too close to a critical, and there was still *not enough bearing damping*, then the better the coupling balance and the better the chances would be of running successfully. Also, a better balanced coupling would pass through the calculated second critical at 7000 rpm with less of an amplification factor.

The contract coupling, which had been returned to the factory for inspection, had significant unbalance (230 g-mm) in the rigid hub; unbalance that was not reflected on the original balance report. Apparently, permanent field corrections (drilled holes in the hub body) had been made to the rigid, to account for the unbalance introduced in the hub when a threaded plug was inserted into a hydraulic port hole for dust/contaminant protection (Figure 27).

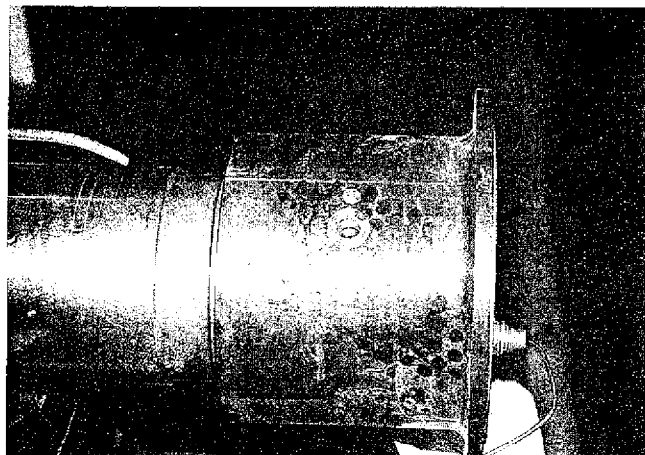


Figure 27. Unbalance in Original Contract Rigid Hub.

Rotor response to unbalance calculations were done to see if this rigid hub unbalance could have caused the severe vibration levels. The results were marginal; the unbalance in the rigid could have contributed to the problem, but it was probably not the only cause.

In any event, because balance corrections of a nonpermanent nature might be required, the new heavy hub was outfitted with a balance correction groove (Figure 26) and moveable balance weights.

Final Test Series

The final test series at full operating speed included the new “heavy” 75 kg half coupling mated to the rest of the test stand coupling (Figure 28), and significant bearing modifications.

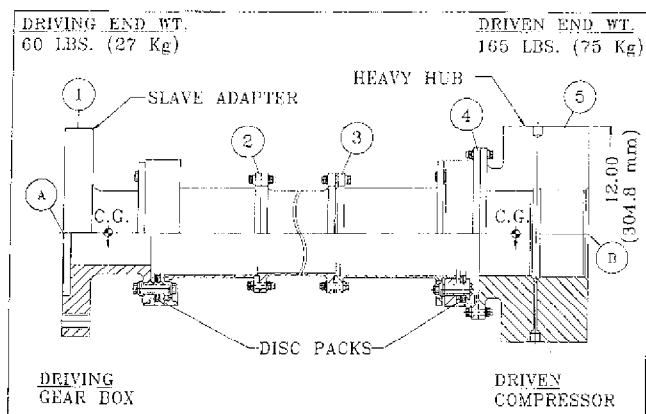


Figure 28. Final Test Coupling with Contract Heavy Hub with Indication Points.

Significant bearing changes that had already been done in the first series of tests and showed some limited improvements were:

- Diametral bearing clearances were reduced by about 33 percent (from the original design) to 3.98/4.80 mils (101 μm/122 μm) from 6.18/7.44 mils (157 μm/189 μm), in order to increase the stiffness and also improve damping.

- The bearing design had already been changed, in previous tests, from a center pivoted to a 60 percent offset design.

- Oil flow to the coupling end bearing had been increased 33 percent.

New bearing improvements in the final stages of testing were:

- The bearings were fixed in their housings with stiffener screws.
- The oil flow through the impeller bearing was increased by 20 percent from the original design, and the oil flow to the coupling end bearing was increased another 20 percent (in addition to the previous 33 percent).
- At the radial part of the combined coupling end thrust-journal bearing, special tip seal rings were installed to keep the oil in the bearing to a low pressure of about 0.2 bar and reduce side leakage.

In parallel, squeeze film damper bearings were being avidly investigated in case the new bearing modifications did not work.

Additionally, the shaft vibration probes were removed from the more flexible cover of the bearing carrier and fixed at special stiff probe supports close to the bearings.

There were a few things done to ensure that the coupling operated well balance-wise. The main focus of these other efforts was on installation. Specific instructions were given to the test floor to turnover the coupled rotors before the test and check the runouts of key diameters on the installed coupling (Figure 28) to ensure that they were within the values of Table 3.

Table 3. Coupling Installation Indication Values (TIR).

LOCATION	MICRONS	INCHES
A (Gear pilot)	12	0.0005
B (Comp. Shaft)	12	0.0005
1	25	0.001
2	50	0.002
3	50	0.002
4	38	0.0015
5	25	0.001

This was an important inspection, just as important as the coupling balancing operation. For example, if it were determined from indicator readings that the hub body (measurement 5 in Figure 28) ran out 0.002 in (51 μm) TIR to the center of shaft rotation, that would mean that the coupling’s axis of rotation was shifted 0.001 in (0.002/2) from the center of shaft rotation. This is equivalent to 1000 micro-in (25,400 μm), which would be 20 times the API 671 [3] assembly balance tolerance of 4W/N (50 μm). This topic is covered in the literature [4].

Along with this, it was suspected that there was possible clearance between the gearbox flange and the coupling slave adapter at the test gearbox locating pilot. If there was, the coupling could be installed off of its balance center, and again, a small amount of shift (0.0005 in (12.7 μm) or more) would cause significant unbalance. Instructions were given to check for pilot clearance, and if found, to center the coupling and tighten the bolts properly so that the coupling could not shift in the clearance during high speed operation.

Indexing instructions were also given in case the measured runouts were excessive. Mainly, instructions were supplied on how to match the indicator “high spot” of the shaft with the “high spot” of the coupling hub bore to reduce the overall runout (and possible unbalance) as much as possible.

“THE SUN IS SHINING TODAY”

After the mechanical test run with the heavy coupling arrangement (rotor E), it was reported “... the maximum direct (vibration amplitude) values (were) $< 25 \mu\text{m}$. Therefore, the sun is shining today!!!” The final run values are shown in Figure 29. The compressor was shipped shortly thereafter for mechanical linkup with the gas turbine.

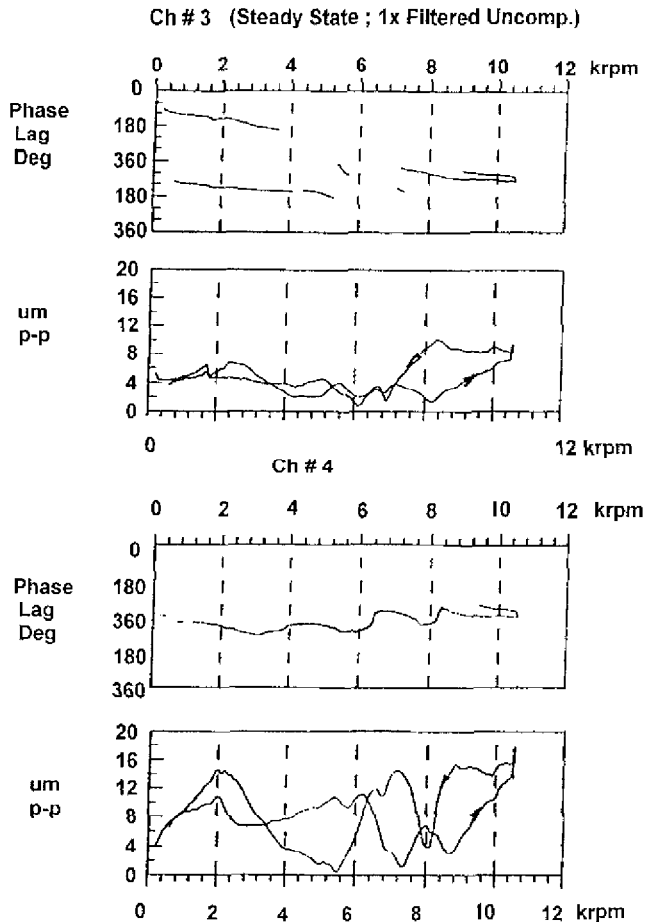


Figure 29. Final Successful Run Bodé Plot.

At the time of the writing of this study, the compressor is running successfully in the field and the plant is producing product. The compressor has operated at all of the expected operating conditions, with acceptable vibration ($< 25 \mu\text{m}$) at the bearings.

CONCLUSIONS

- The journal bearings had been designed for minimum oil flow. This would not be a problem normally, except that the reduced coupling weight of the contract coupling likely did not provide sufficient static load for a stable bearing design. This probably led to cavitation and lack of damping. In this case, the main countermeasure was to increase the oil flow to the level needed for an unloaded bearing, and to use end seals. More work is being done on this topic (calculations, etc.), to see if this theory was, in fact, the case and to see if the cavitation could have been predicted. The solution of the bearing problem, and its unexpected poor performance, will be the subject of another study.

- The implementation of relevant API Standards and their proper interpretation, from the onset of this project, could have significantly reduced the confusion in defining responsibilities and could have resulted in a more expedient solution to the problem. In fact if API 617 had been specified, the proper coupling weight would most likely have been used in the original analysis and at least this part of the problem would have been avoided.

- The coupling can have a significant influence on the rotordynamic performance of high speed turbomachinery. Its effects cannot be ignored or taken lightly.

- Though a heavier coupling helped solve the problem and worked well, it should not be a preferred solution to a rotordynamics problem. Altering the design of a coupling in the middle of the testing cycle should be avoided. The proper coupling should be designed early in the cycle.

- Machines should be designed to run away from critical speeds unless proven bearing designs, at actual job conditions, are used to provide adequate stiffness and damping.

- Axial resonant frequency interference of a nonlinear self damping disc coupling should be well down on the list of things to check when there is significant axial vibration.

- A check for a proper coupling installation (indicator checks after complete coupling installation) is extremely important to rule out the possible problems due to excessive unbalance from coupling runout.

- Test procedures that do not duplicate full operational conditions can result in confusion in the determination of the sources of vibration.

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