AN OPERATION HISTORY OF FRACTIONAL FREQUENCY WHIRL

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Kenneth J. Smith is a native of Michigan and graduated from the University of Michigan with B.S. degrees in Mechanical and Aeronautical Engineering in 1954.

Ken's career with turbomachinery began about 20 years ago at the NACA Lewis Laboratory where he participated in origination of centrifugal impeller design method developments using stream filament techniques and blade loading calculation procedures. Subsequently, he joined TRW and became acquainted with the practical aspects of designing and producing turbomachinery hardward including APU's, gas turbine driven electric-generator sets, pneumatically driven flap-drive mechanisms, and torpedo propulsion systems. Jet engine blade and vane component materials selection and producing methods, and composite material investigations formed an essential part of his work scope at TRW, also.

Since joining Cooper Energy Services in 1969, his involvement has been about equally divided between centrifugal compressor and power turbine design and development. As Manager of Rotating Product Engineering since 1973, he supervises design, shop production interfaces, development testing, and customer field acceptance of C-B rotating products.

ABSTRACT

The problem of a whirling rotor in a high pressure centrifugal compressor and the steps to arrive at a solution are described. Orbit shapes and vibratory motion are illustrated. Vibratory frequency ratios compared to undamped lateral critical speeds and rotor geometry are included. A comparison of different bearing, seal and shaft geometries is presented in terms of the relative aid gained from each. Analytical approaches are briefly reviewed in terms of the support added to selection of component geometries. The significance of operational parameters of speed and pressure on the vibratory behavior are mentioned, and some experience factors generalized for whirl suppression.

INTRODUCTION

Fractional frequency vibration in rotating machinery constitutes one of the most difficult but interesting problems encountered in rotor dynamics. As used in this paper, the term fractional frequency vibration is vibration in any mode form whose frequency is a fraction of the synchronous (running speed) frequency. The classical case of this type of vibration is "oil whip" (or whirl) which has a dominant frequency of approximately half the running speed and usually builds amplitude as the running speed is increased. A distinctly different form of fractional frequency vibration (which is often mistaken for oil whip) has been under intense investigation over the past few years as a result of the violently destructive behavior and analytically illusive nature of this phenomenon. It is often referred to as shaft whirl, resonant whirl, non-synchronous precession (NSP), or as I prefer to call it, fractional frequency whirl (FFW), and will constitute the main topic of this paper.

Cooper-Bessemer's major involvement with this problem began in November 1971, upon attempting to commission three natural gas re-injection compressor trains at the Kaybob South Beaverhill Lake plant near Fox Creek, Alberta operated by Chevron-Standard. The five month effort to solve the vibration problem constituted many tests and configuration alterations with interesting and sometimes frustrating results. A summary of these change and effect relationships is presented herein along with a brief review of supporting analytical studies.

Although the data presented is almost four years old now, it is relevant to current industry problems and recent analytical technique advancements [1, 2] complement the empirical results obtained. The experiences shared by Cooper-Bessemer, Chevron-Standard and Standard Oil Company of California (SOCAL) personnel in pursuing this problem were presented in reference 3 earlier this year. A paper covering the entire plant is given in reference 4.

GENERAL CONFIGURATION AND CHARACTERISTICS

A schematic of a single re-injection train at Kaybob is shown by Figure 1. There are three duplicate trains operating in parallel. The maximum flow requirement is 230 MMSCFD of 18.0 M.W. gas to be handled by two (2) trains at a maximum continuous speed of 11,400 rpm. Rated condition is 95 MMSCFD per train at approximately 10,200 rpm.

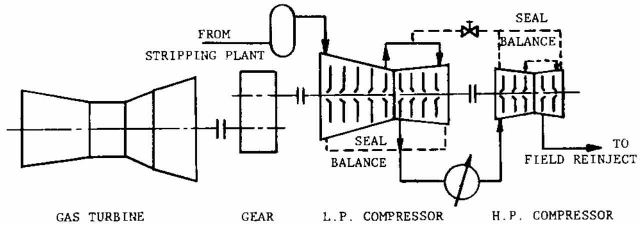
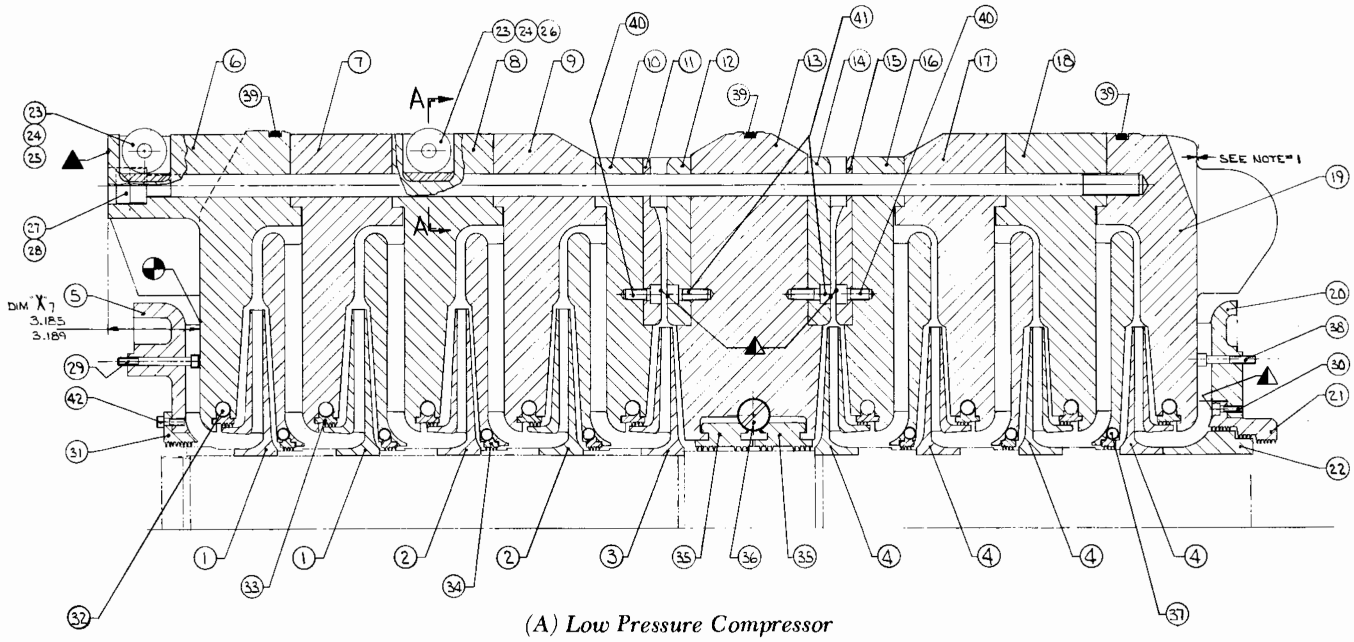
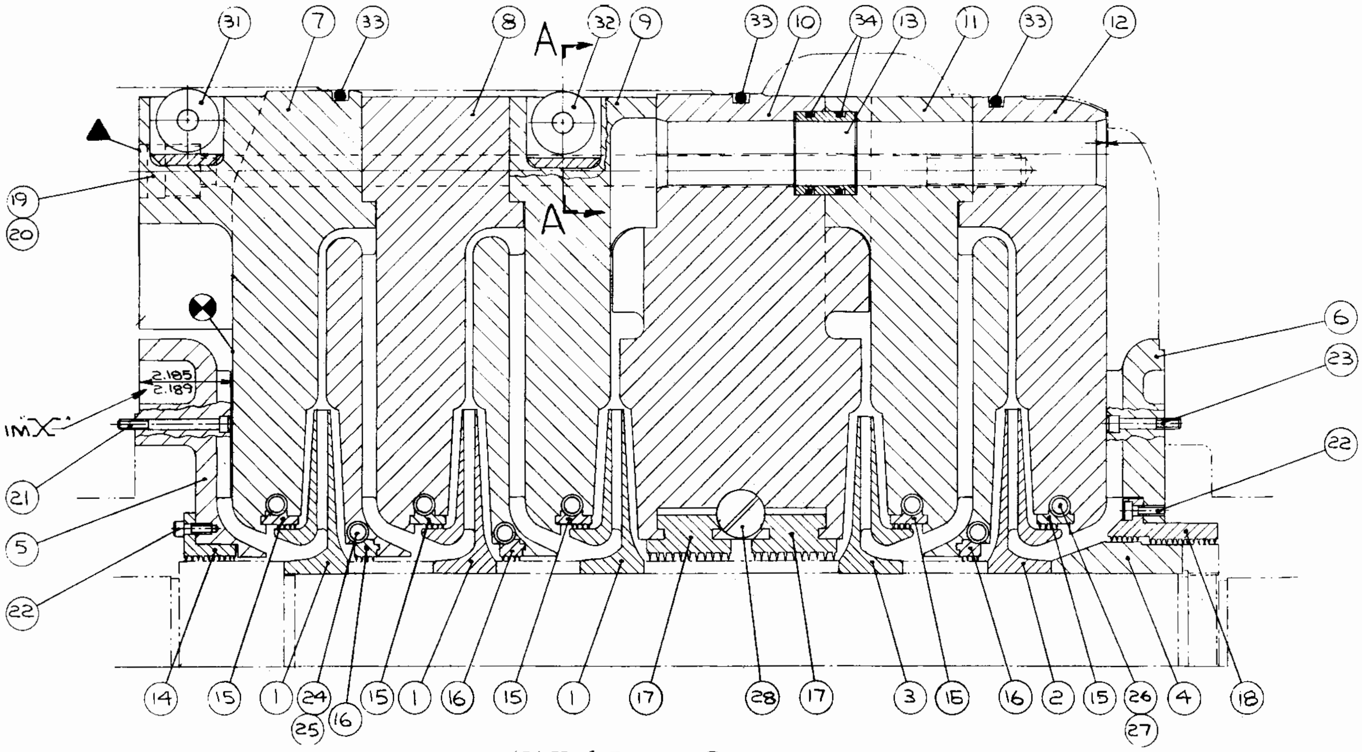


Figure 1. Schematic of Compressor Train

Figure 2 shows the aerodynamic assembly of the 9 stage low pressure (L.P.) compressor (a) and 5 stage high pressure (H.P.) compressor (b). Both compressors are arranged with back-to-back impeller staging. The gas is cooled between compressor cases. A balance piston, to control the magnitude and direction of rotor thrust, is located at the second section inlet of each casing. By design, the rotor thrust is in a direction to load the in-board shoes of the thrust bearing.



(A) Low Pressure Compressor



(B) High Pressure Compressor

Figure 2. Compressor Aero Cross Section

Both the thrust and journal bearings for each casing are tilting-pad type. The journal bearings have five (5) pads and the thrust bearing six (6) pads on the outboard side and twelve (12) pads on the inboard. The design thrust bearing load corresponds to 25-35% of the manufacturer's rating for the bearing at operating speeds.

Oil film seals are used for each casing. Single ring pressure-breakdown seals are used on the H.P. casing. Both inboard and outboard ends are balanced back to the L.P. first section discharge pressure condition. Seal rings for both L.P. and H.P. compressors are cylindrical with a heavily chamfered edge at the oil entering face. No "O" ring seal is used on the housing face of either the inner or outer seal ring; sealing against oil leakage past the interface is accomplished through lapping of the housing lip with the mating face of the seal ring. The seal ring material is babbitt-lined steel.

The initial configuration of the compressors contained a separate lube and seal oil system. This was to guard against a possible plant upset which could introduce H_2S into the reinjection gas stream in appreciable quantities and contaminate lubrication oil. This arrangement was later revised as subsequently discussed herein.

DEVELOPMENT HISTORY

During the design phase of this project, at the request of SOCAL an independent audit of the shaft train dynamics was performed by a consultant. Verification of the C-B calculated lateral critical speed values was obtained, but concern was expressed by the consultant over the ratio of running speed to calculated rigid bearing criticals. At maximum continuous speed, for the L.P. compressor, this ratio was near 3.0, and for the H.P. compressor, about 2.0. Based upon experience, it was felt shaft whirl might be a problem at these high ratios. However, analysis methods were not sufficiently advanced to provide conclusive arguments. C-B operating experience with methanol and ammonia synthesis compressors operating at or above these values was satisfactory, indicating that this singular parameter by itself is not controlling. It was agreed to instrument the compressor for shop testing so that whirl could be identified and to carefully monitor vibration during the tests. An ASME PTC-10 performance test, four (4) hour C-B standard mechanical test and API seal leakage test was performed on every compressor. Each turbine driver package was also given a C-B standard one (1) hour hot run mechanical test.

Shop Test Results

During the compressor mechanical test, particular attention was paid to the vibration characteristic of the rotors. A vibration frequency scan was made to obtain the amplitude of components at synchronous frequency, fractional frequency, and multiples, as well as the unfiltered (total) amplitude. All the compressors showed normal vibration characteristics with no determinable instabilities. Data from the compressor showing the greatest amplitude of non-synchronous vibration is shown on Figure 3. This was one of the L.P. compressors. The half-speed vibration amplitude climbed to about 40% of the total, but then tailed off as the speed increased. At the time, this was thought to be attributable to the test stand oil film seals being used. These rings contained "O" ring face seals which can be a source of ring lock-up. (Seal ring lock-up is a known source of fractional frequency vibration.) With remarkably easy hindsight, this was a stone which shouldn't have been left unturned.

Initial Site Testing

Start-up of the first compressor train (15-1801) was initiated on November 9, 1971. The normal sequence is to start seal oil pumps and pressurize casings to 700-750 psig, followed by gas turbine start-up and hold at warm-up speed equivalent to about 2500 rpm on the compressors. Upon completion of the warm-up time restriction, the turbine driver is accelerated quickly thru its critical speed to minimum governed speed corresponding to about 8500-8600 rpm compressor speed. At

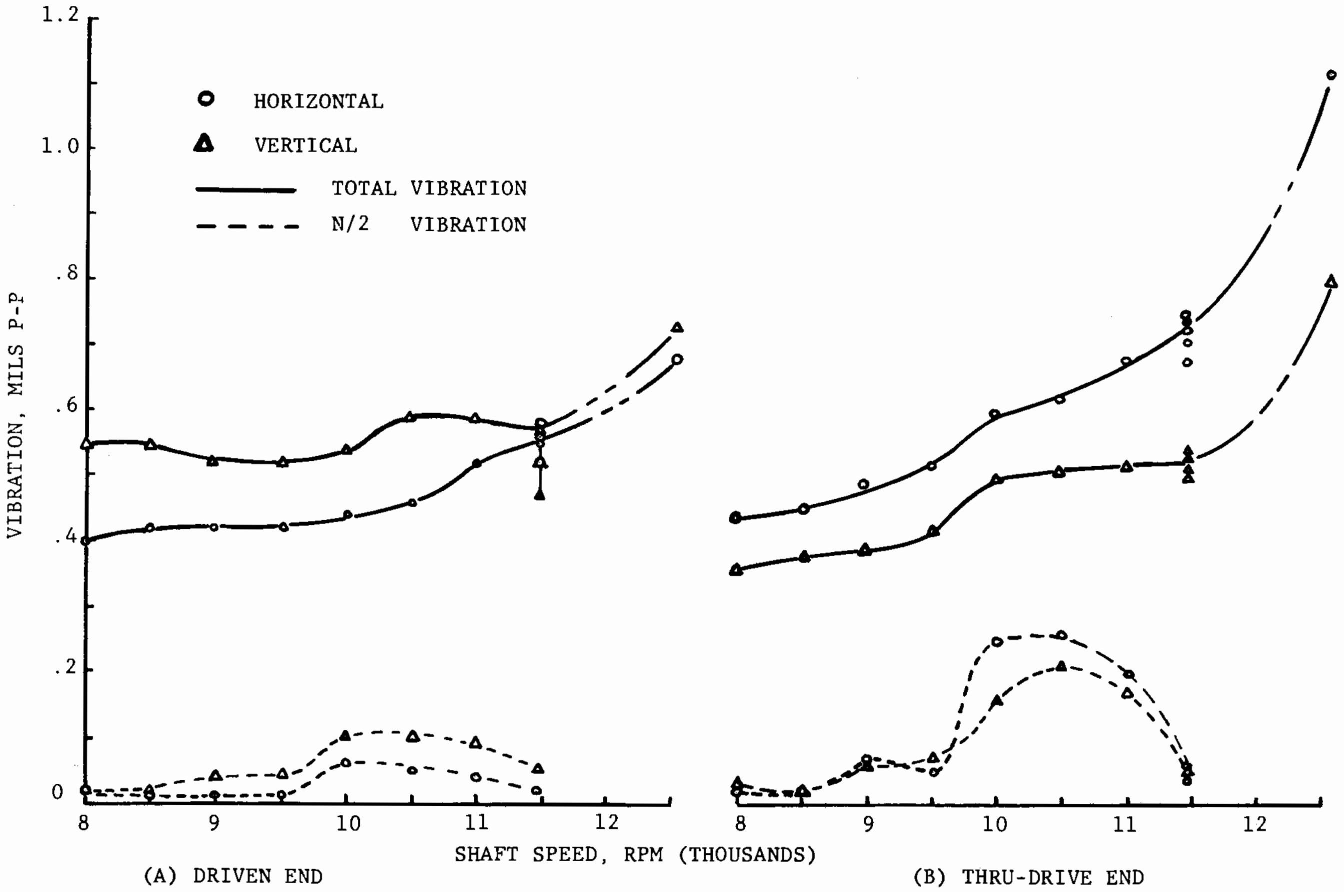


Figure 3. L.P. Compressor Shop Test Vibration

this point, the recycle valve is fully open. The speed is then increased, the recycle valve slowly closed, and the suction pressure raised until the discharge check valve opens and reinjection flow commences.

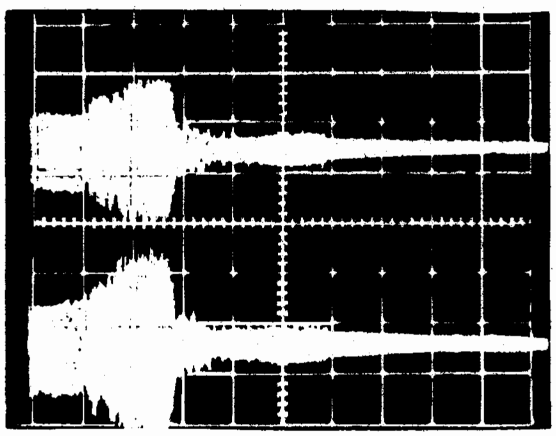
It became immediately apparent that a serious vibration problem existed in the L.P. compressor. At minimum governed speed (8500-8600 compressor speed), inlet pressures of 700-750 psi and discharge pressure of 1300-1500 psi, L.P. vibration amplitudes were .7-1.0 mils peak-to-peak at a frequency of about 4100-4350 cpm. This fractional vibration contributed about 60-70% of the total synchronous vibration amplitude. As the speed was raised, instability became worse. By the time 10,000 rpm shaft speed was reached, the L.P. rotor was experiencing frequent jumps to 3 and 4 mil p-p vibration at 4250-4300 cpm. This constituted 90% of the total synchronous amplitude. Orbits were mainly circular with diameters that would pulsate erratically. Both backward and forward precession was visible. Usually, the precession would be initially backward, then stop, then forward at a rapid speed to high vibration shutdown.

Changes in oil temperature and inlet pressure were introduced to determine their influence. Oil temperature ranging from 110-150 deg. F produced no substantial difference in orbit shape, size, or erratic behavior.

A large effect from compressor gas pressure was noted. With suction pressure below about 250 psi, the compressor could be run to 11,400 rpm with reasonable vibration amplitudes (~ 1.0 mil p-p) with only a trace of fractional frequency (4300 cpm) vibration. Allowable speed of operation decreased in relation to suction pressure increase with 700-750 psi suction limiting safe operation speed to about 9100 rpm.

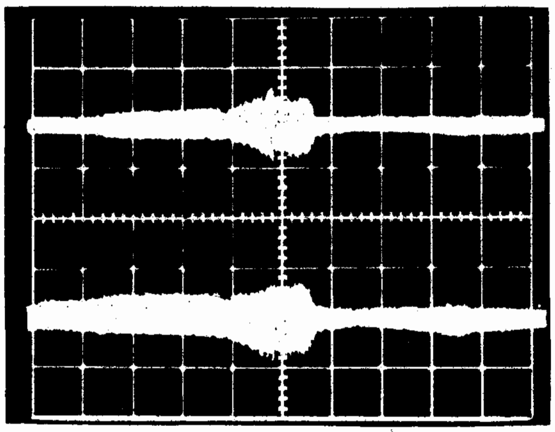
It was then decided to test a second train to determine if there could possibly be an unexplained acoustic resonance condition or other peculiarity on the 1801 compressor. Accordingly, unit 1802 was highly instrumented and cautiously started. Instability was again noted from minimum governed speed (8500 rpm) to about 10,500 rpm, at which time stability returned with a large, but manageable, 2 mil p-p vibration. This held through to the maximum continuous speed of 11,440 rpm. The L.P. discharge pressure was 1600 psi at this time. As the discharge pressure was raised, whirl initiated - first at 4700 cpm, then increasing to 5100 cpm as the L.P. discharge reached 3100 psi with an 1120 psi suction pressure. At 3,300 psi L.P. discharge pressure, the whirl rate suddenly jumped and the rotor went into a violent 9 + mil p-p vibration.

Figure 4 illustrates the severity and suddenness of the vibration onset. The orbit trace (about 6 x 9 + mils p-p in a 6 mil bearing clearance) is from the thru drive end of the L.P. compressor. Note the clear indication of the five pad configuration of the journal bearing. The rotor assembly and internal labyrinths were severely damaged as a result of this violent, high-energy motion. This vibration could be heard and felt in the control room some 50 feet away from the unit.



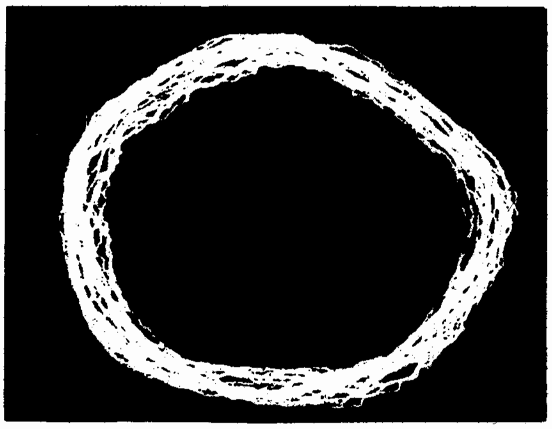
VERTICAL

HORIZONTAL



(A)

(B)



(C)

Figure 4. L.P. Compressor Field Vibration Oscilloscope Traces
(A) Thru-Drive End; 20 sec elapsed time
(B) Driven End; 20 sec. elapsed time
(C) Thru-Drive End; Orbit, 6 x 9 mils p-p

Throughout all of the vibration problems experienced on the L.P. compressors no sign of any distress occurred on the H.P. compressors despite their higher pressure level (4000-4500 psi).

Tests and Solution Development

The severity of the fractional frequency whirl problem was vividly demonstrated by the initial site tests of the two trains. For plant operation to continue, reinjection capability was essential hence an immediate solution to the problem was needed. To achieve this goal, a three-path parallel attack was undertaken:

1. Conduct continuous field test programs to develop modifications permitting a train to run at partial loads and supply answers for a more permanent fix.
2. Analyze and review previous C-B compressors installed in similar service and note specific differences. Enlist the aid of consultants to study the existing configuration, results of tests, and provide recommended modifications.
3. Select alternate machinery capable of meeting the requirement and determine delivery time and installation problems.

Fortunately, item 3 never progressed beyond the paper stage since progress on 1 and 2 proceeded fairly quickly. In the subsequent paragraphs, compressor mechanical feature modifications and their associated effects will be individually discussed.

Compressor Survey

A tabulated design-feature listing of seven C-B operational compressors having characteristics similar to Chevron was made and thoroughly reviewed. No singular outstanding characteristic could be differentiated. However, a link between impeller number and diameter, average pressure, and speed was hypothesized as perhaps relating to aerodynamic drag instabilities with subsequent destabilizing influences.

Compressor Shaft Seals

Much has been written and proven concerning the action of oil-film shaft seals on destabilization of high speed rotors. C-B studied these types of seals several years ago under an internal R&D project. The conclusion was that properly designed seals would "float" and not act as bearings. Nevertheless, this was one of the first components evaluated at the site.

Four different cylindrical seal configurations were examined:

(1) a ring with three different diameter lands in the bore separated by grooves with clearance converging in the direction of oil flow,

(2) a ring with clearance equal to the bearing clearance,

(3) a low-mass ring (ring weight reduced thru milling away non-critical areas), and

(4) a ring with a waffle-like internal bore grid.

None of these alternate designs significantly altered the vibration characteristics of the rotor.

Some interesting data was obtained in conjunction with the seal ring configuration evaluation. The seal ring housing was machined to provide for mounting of two proximity pick-ups, 90 deg. apart, sensing the outer ring (atmospheric side) position. A false reference was connected to the seal oil system to enable seal oil pressure to be regulated independently of gas pressure. Holding suction pressure constant, the shaft orbit and vibration levels and seal ring motion were monitored as discharge pressure and seal oil pressure were varied. It was found that holding a ratio of seal oil pressure to compressor discharge pressure of two (2) or greater, exhibited a large damping force on the rotor. However, the damping was insufficient to permit design pressure levels to be used, although full speed (11,440 rpm) could be reached up to about 200 psi L.P. suction pressure (500 psi discharge) by this ploy. Except during the extreme rotor vibration levels, the seal rings moved smoothly under pressure and rotor speed changes and maintained their position with little oscillation. (During high vibration periods, the rings showed low-amplitude erratic bouncing.) Figure 5 is a reproduction from a chart trace of the seal ring motion.

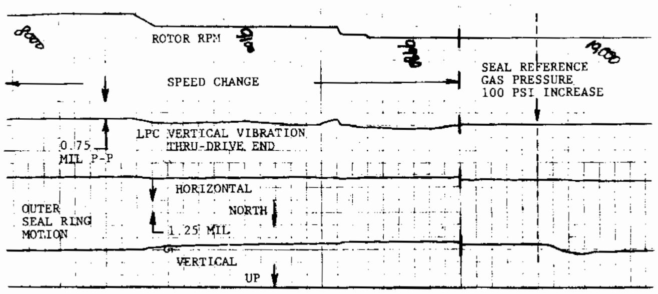


Figure 5. Chart Trace of Outer Seal Ring Motion, Thru-Drive End

These tests demonstrated that, indeed, seal rings do "float" providing damping, but do not necessarily act as bearings when design mechanics of the seals and housings are proper. Two different types of seal rings were measured — the three-stepped bore and a short-land cylindrical ring — with no behavior difference discernible.

Bearings

More was done with bearings experimentally and analytically than any other design feature. Reference 5 was extensively used in conjunction with C-B design modification selections and analyses during early investigations; later developments employed two consulting firms for design advice and data review. The highlights of each configuration and test results follow.

a. *Bearing Load*

The projected area of an original bearing pad was 2.65 square inches. The L.P. rotor weight was almost equally divided between the two bearings with light bearing pad loads imposed. A test was made wherein the width of the pads was substantially reduced thereby increasing the unit load by about 62%. No significant gain in stability or vibration level reduction was obtained from this change.

b. *Bearing and Foundation Support Symmetry*

For all rotor stability analyses, knowing the effective bearing stiffness is essential to the dynamic characteristic calculation. The foundation support stiffness for the L.P. compressor was measured in the field by a consultant firm using a "knocker" device and real time analyzer. The results of the measurement are shown on Figures 6a and 6b. Included are C-B calculated values from earlier data on an R&D test compressor casing of almost identical design. Correlation of measured data with the calculation was reasonably good and showed almost equal stiffness in the

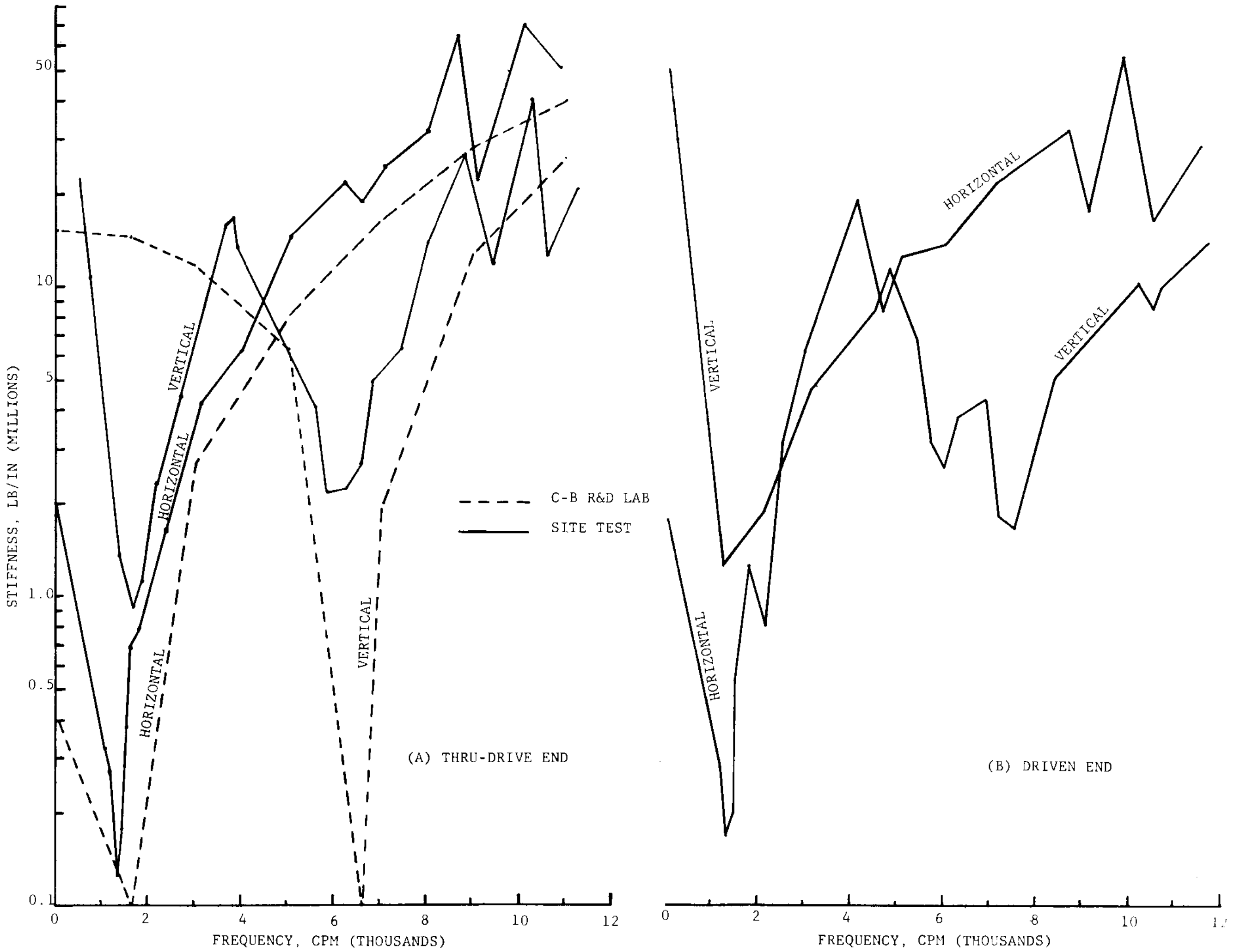


Figure 6. Foundation Support Stiffness Measurement Plot.

horizontal and vertical directions. Also, the two ends, thru-drive and driven, showed about the same characteristic.

From the early tests, the orbits of the L.P. rotor were generally circular (Figure 7) indicating about equal stiffness in both directions, corroborating the support stiffness measurement and oil film support uniformity of tilt pad bearings. As suggested by literature [5,6], rotor stability may be improved by introducing bearing stiffness asymmetry. This was accomplished by thinning the cross section thickness of two of the reduced width bearing pads by approximately 2 mils, and placing these two pads in the closest to horizontal position. Also, with reference to bearing cross coupling terms, some gains were felt to be realized by "clocking" the pad arrangement to locate a pad at the 12 o'clock position instead of the 6 o'clock location. The resulting bearing configuration is shown on Figure 8 and the orbit and vibration trace on Figure 9. The orbit ellipticity is easily noticed. A definite gain in rotor stability was achieved by this step. The threshold limit of the speed/pressure/vibration relationship permitted operation at 11,440 rpm up to suction pressures of about 400-450 psi. The FFW frequency remained at 4300 cpm, but was not discernible before 10,000 rpm/250 psi. This was still a long way from the required goal, however.

To further define the improvements noted, two variations of the asymmetric bearing were tested: first, the pads were clocked back to the original position with one pad at 6 o'clock; and second, the pads were remachined to provide a circumferentially offset pivot to improve oil wedge formation. Both of these steps produced retrogression, the second

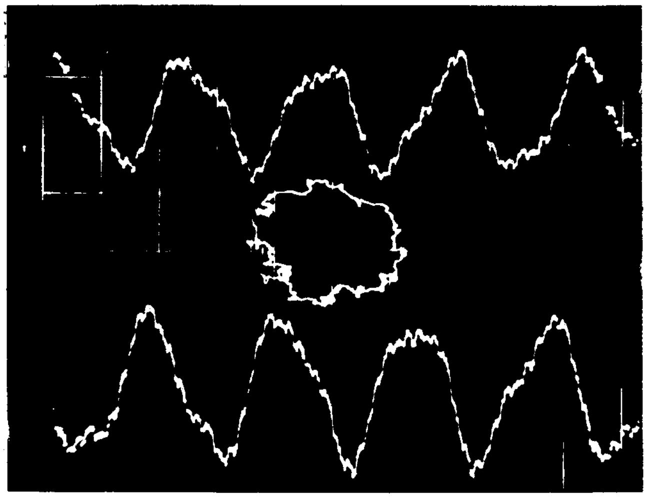


Figure 7. Orbit and Vibration Trace, L.P. Thru-Drive End (1 mil p-p/div; 11, 400 rpm)

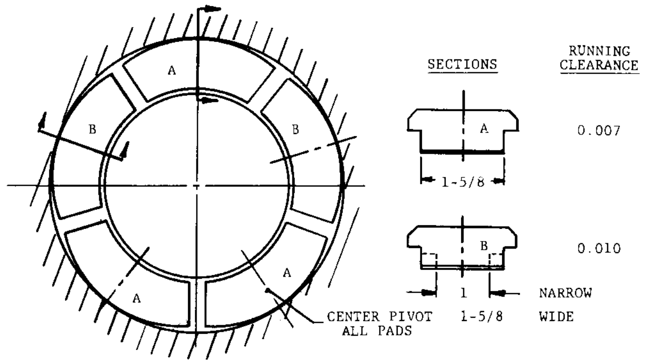


Figure 8. Clocked, Asymmetric Bearing Configuration.

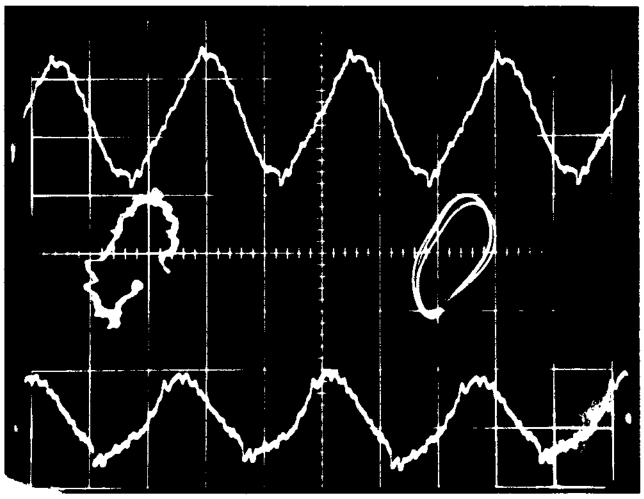


Figure 9. Asymmetric Bearing Trace, Orbit, and Orbit with Filter (1 mil p-p/div; 11,400 rpm; filter at 20,000 cpm).

more severe than the first. The minor increase in instability thru clocking back to the original pad at 6 o'clock was expected. However, the significantly worsened behavior with offset pivots was not. It was concluded that the offset pivot in combination with the asymmetry and different pad preload between the three original pads and two reduced width and thinned pads was one too many variables for the rotor to handle in a predictable manner.

c. *Squeeze Film Damper Bearing*

It was obvious from the data generated on the rotor behavior that a fairly major step needed to be taken to achieve acceptable dynamics. One such step involved design and test of a squeeze-film (oil damper) bearing. The generalized schematic of the bearing derived by the consultant is shown by Figure 10. Supporting the rather unconventional bearing pad configuration were analyses of the previous tilt pad bearing configurations and that of the squeeze film bearing. Figure 11 illustrates the analysis data derived from the consultant's computer program. The (A) graph indicates the level of damping achieved for an arbitrary initial displacement, the (B) graph the individual pad rotation about their pin, and the (C) graph the oil film pressure force. In the upper right, the path of the shaft center is described. — (D).

As mentioned, this program was used in conjunction with the previous tilt pad variations of bearings and with alternate shaft configurations (paragraph d, following) and indicated marginal stability for these configurations. As shown in Figure 11, the squeeze film projected excellent damping (top graph, A) although pads 3 and 4 "nose-dived" during rotation (graph B). This was remedied by pad redesign and resulted in the final general configuration shown by Figure 10.

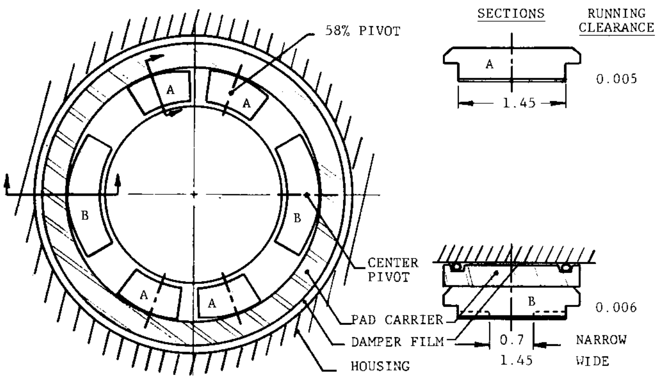


Figure 10. Squeeze Film Bearing Configuration.

The squeeze film bearing was installed and tested first with the "standard" 1801 rotor. The asymmetry of the design was pronounced in characteristic as shown by Figure 12. The horizontal pads had much reduced stiffness as is evident. If any gain was achieved with this combination, it was quite marginal. The major vibration frequency remained at 4350-4600 cpm with amplitudes equivalent to the 5 tilt-pad,

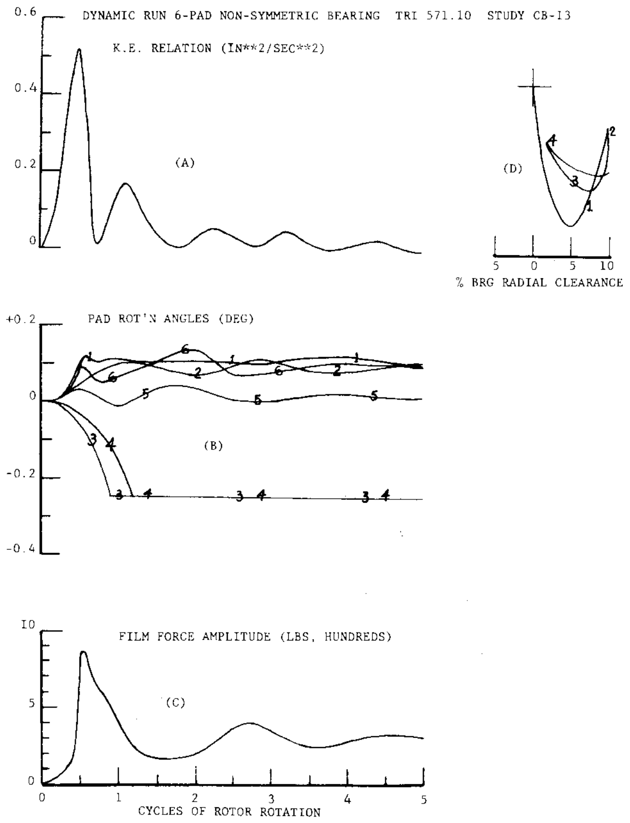


Figure 11. Squeeze Film Bearing Analysis Derivation.

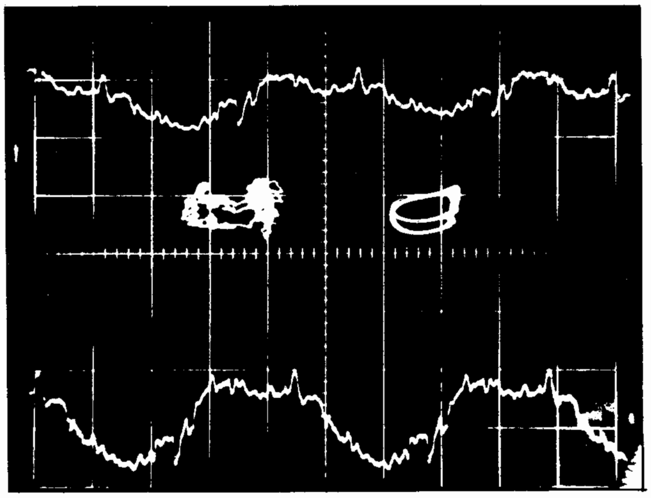


Figure 12. Vibration Trace and Orbits of Squeeze Film Bearing (0.5 mil p-p/div; 6650 rpm; filter at 20,000 cpm)

asymmetric bearing. Neither speed nor pressure level could be appreciably raised before exceeding safe operational vibration levels.

At this point, a reduced length rotor was introduced (see paragraph d, following) and the squeeze film bearing installed on it. Improvement in dynamic performance was substantial. Vibration was acceptable up to about 660 psi L.P. suction pressure (with discharge from the L.P. about 2100 psi) at speeds to 10,700 rpm. The vertical vibration was approximately 1.6 mils p-p overall and horizontal 3.5 mils. Coupling the L.P. rotor to the H.P. provided an even better stability. Almost full L.P. suction and discharge pressure (1120/2600 psia) was achieved at 9600 rpm (4000-4100 HP discharge). FFW was evident at a frequency of about 5100 cpm with a stable amplitude of about 40% of the overall 0.6 vertical, 1.0 horizontal. (The large horizontal component was decreased by making all the pads equal in face width.) This configuration was conditionally acceptable for plant operation.

One of the operational characteristics of this squeeze film bearing design was the improvement gained by increasing the damper film pressure. The damping capacity was significantly improved when the film pressure was raised to 275-325 psi. Above this pressure, some "tail-off" in damping was noted and below it, a substantial reduction occurred. This was hypothesized as being related to contact of the bearing pad cage with the outer shell (Figure 10), through deformation or vibrational forces, which required a hydrostatic force to overcome. It is obviously the wrong direction from the standpoint of the oil bulk modulus stiffness and related damping.

Another interesting facet which was subsequently explored related to the machined clearance (oil film thickness) between the pad cage and outer shell. The initial configuration had a 6 mil machined clearance. Since the damping force is proportional to the cube of the radius-to-clearance ratio, decreasing the clearance should improve the damping, and hence rotor stability. Accordingly, a revised damper film thickness of 4 mils was tried. This produced less stability at any film pressure than the 6 mil clearance. It was again hypothesized that metal-to-metal contact could not be eliminated, although subsequent analyses performed by another consultant questioned both the "O" ring stiffness effect and the film pressure distribution (damping) produced by the "O" rings acting as dams. In any event, the sensitivity of the damper film bearing to many design factors was amply demonstrated and supports the "tuning" necessary to optimize its performance as discussed in reference 7.

d. *Shaft Configuration*

The original design of the L.P. shaft followed C-B historical practice with regard to bearing span, fits, seal interfaces, and impeller assembly arrangement. With the rotor dynamics difficulties being experienced, all of our old ways were scrutinized in relation to published data of whirling shafts [5,6]. The succession of shaft configurations arising from analyses and tests is summarized on Figure 13.

The initial revision to the shaft/rotor was to decrease bearing span and modify shrink fits of all assembled components. This was accomplished thru elimination of the separate lube and seal oil system and by relieving all impellers and the balance piston in the center of the fit to provide a tight squeeze only at the end extremities. In addition, the center spacer O.D. was increased. These changes, in combination with the squeeze film bearing, provided marginally acceptable operation as reported in the previous subheading.

The next change took the previous configuration one step further. In a back-to-back impeller arrangement, a seal, usually near the center of the shaft, is required to separate the two sections. Historically, this was accomplished by using a heavy sleeve shrunk-fit on the shaft matching the labyrinth-seal bore. In shaft bowing flexure, the maximum deflection will normally be somewhere close to shaft center creating the maximum opportunity for friction hysteresis under the center sleeve. The second shaft revision made an integral center section shouldered on each end to provide impeller stops, and moderately increased diameter beneath impellers to increase shaft rigidity. Component interference fits were slightly increased from revision 1. This produced a moderate but still further gain in the speed/pressure, vibration relationship for the L.P. rotor.

The final L.P. shaft configuration used the integral center seal section and component interference fits of revision 2, but increased shaft diameter under impellers still further. This required some rather difficult impeller salvage steps, but these were accomplished satisfactorily.

The resulting rotor, in combination with either a damper film bearing, or a five tilt-pad bearing with zero preload, provided full operational range of the compressor with quite acceptable vibration behavior. FFW with a frequency of 7200 cpm constituted about 8% of the total vibration amplitude at 11,440 rpm rotor speed.

e. *Stationary Components*

During the evolution of the rotor and bearing configurations, C-B was constantly worried about the possibility of aerodynamic stimulation from stationary pieces. One basic change was made reflecting this concern. The original L.P. configuration used two channel-scrolls at the discharge of each compressor section. These dumped freely into a plenum machined into the casing wall and thence into the discharge nozzles. The channels were clocked in each sector to place them roughly at 90 deg. spacing. By analysis, the resulting pressure distribution and perturbations within the plenum should not be sufficient to produce stimulation.

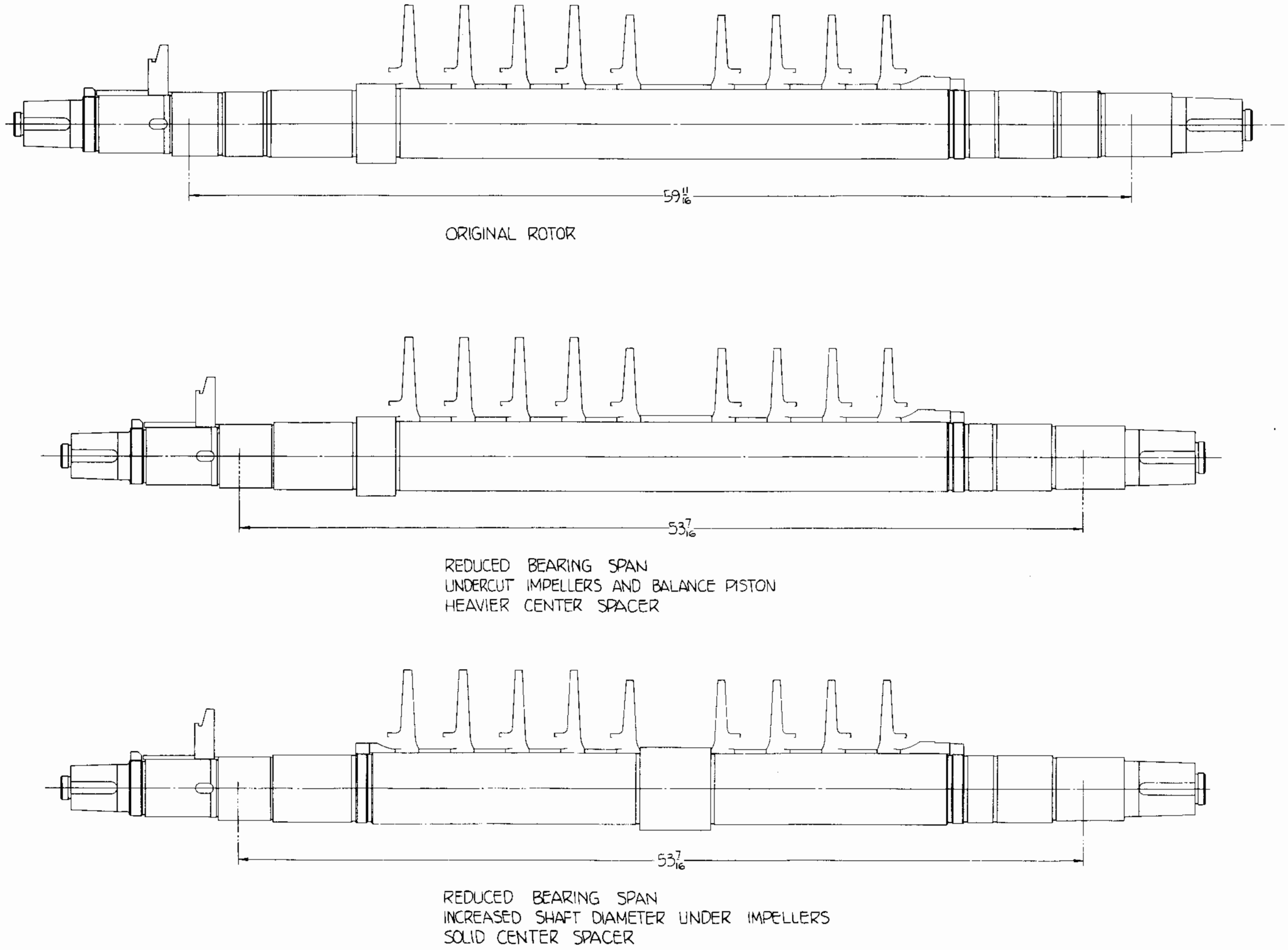


Figure 13. Shaft Configuration Modifications.

However, to play it perfectly safe, each channel-scroll was removed and replaced by a vaneless diffuser with multi-guide vanes at the O.D. In addition, the center-section labyrinth seal was divided into two sections by introducing a pocket at the center of the labyrinth.

These changes were done in combination with shaft and bearing changes so a true measure of their effect was not gained. The best which can be said is intuitively they couldn't hurt.

f. *Bearing Preload*

Earlier, the results of changing the journal pad pivot point and the results of bearing pad preload were briefly mentioned. Normally, tilt pad bearings are designed to operate with a positive preload — somewhere in the 15-40% range. (Preload is defined as the ratio of pad bore radius minus shaft radius minus radial running clearance to pad bore radius minus shaft radius, Figure 14.) The preload was known for the early configurations, but aside from trying to avoid negative preloads, it was not a test variable per se.

As part of a design review by a second consultant group, calculations indicated that a five tilt-pad bearing with zero preload was as effective in rotor damping as the squeeze film bearing. It was tried and proven to be successful with stabil-

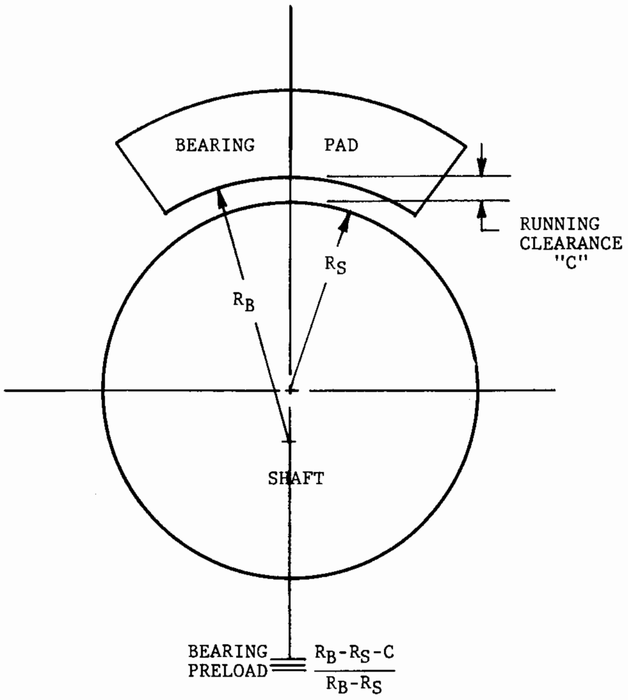


Figure 14. Bearing Preload Definition.

STABILITY VALUES FOR CASE 3 MTI 72-TR-17

N = 10,400 rpm

Rotor No. 4
5-pad non-preloaded bearing
Rigid Pedestals

Load Between Pads			Load on Pad		
Resonant Frequency (cps)	whirl Direction	log decrement δ	Resonant Frequency (cps)	whirl Direction	log decrement δ
47.2	F	2.62	70.2	B	1.28
64.1	B	1.57	38.7	B	6.18
55.1 rpm	B	4.97	121.3 (7290)	-	0.86
112.0 (6720)	B	1.94	132.2 (7940)	F	1.75
130.0 (7800)	F	1.40	93.2	F	3.8
82.9	F	3.81	277.0	B	0.066

Figure 15. Zero Pre-Load Bearing Rotor Dynamic Analysis.

ity and vibration levels matching those of the damper-film bearing. A chart of calculated dynamic characteristics is shown on Figure 15.

Final Configuration

Before the satisfactory L.P. compressor configuration was achieved, about 4-5 months of intense effort was consumed in testing and analysis. The final rotor configuration (ref. Figure 13) does not look radically different in overall appearance than the original. But, the changes introduced have permitted the compressor to operate successfully for about 3-½ years now with no requirement for rotor replacement due to vibration. Figure 16 provides the highlights of the rotor evolution and final vibratory characteristics. FFW is still present, but controlled to approximately 8% of the total vibration amplitude. It is worthy to again note and emphasize that at no time has the H.P. compressor exhibited this problem. Currently, reinjection pressures are in the 4400-4700 psi category. During final operational acceptance running, pressure levels to 5200 psi were generated.

ROTOR CONFIG.	1	2	3	4
Bearing Type	5 Pad Tilting; Asymmetric, Clocked	6 Pad Squeeze, Asymmetric	6 Pad Squeeze, Symmetric	6 Pad Squeeze, Symmetric; 5 Pad Tilting, Zero-Preload, Clocked
Bearing Span	59-11/16	53-7/16	53-7/16	53-7/16
Geometry	Standard	Relieved Impellers Heavy Center Sleeve	Integral Center Section Increased Shaft Dia.	Integral Center Section Larger Dia. Shaft
Calc. Undamped Critical Speed				
Nc1	3915	4642	4950	5451
Nc2	14,125	13,595	13,575	14,523
Observed Whirl Freq. cpm	4100/4350	5100/5400	6350/6490	7200
Max				
Ps	450 725	660 1120	1120 820	1140 1140
Pd	900 1500	2100 2500	3150 2475	3000 3150
RPM	9400 8550	10,700 9600	10,000 11,440	10,780 11,440
Vibration Mils p-p	0.8-1.5 0.7-1.5	1.6-3.5* 0.6-1.0	0.6-0.8 0.5	0.4 0.6-0.8

*Highly asymmetric in horizontal direction

Figure 16. L.P. Compressor Vibratory Characteristic Summary.

SUMMARY

In the narrative description presented herein, it has been the intent to provide general clues as to how a problem of FFW can be attacked. FFW is a significant problem which stems from the industry desire to minimize the number of casings yet maximize delivery pressure. High speed, high pressure, and long bearing spans are the recognized major contributors to FFW problems. However, the combinations of these parameters which define absolute guaranteed safe operating limits are not known. Analytical techniques are improving, but much more needs to be done in both analyses and testing.

From the testing accomplished in support of finding a solution to this particular problem, a few observations are noteworthy:

1. In FFW, the whirl frequency will be greater than the calculated undamped rigid bearing critical speed. The amount is probably related to the degree of damping on the rotor, including the effect of oil film seals.
2. Raising the rigid bearing critical speed of the rotor is the most direct means to reduce FFW vibration amplitudes. It may not be necessary to reduce the ratio of running speed to calculated rigid-bearing first critical below a value of 2 to obtain acceptable dynamic operation.
3. Asymmetry in rotor bearing stiffness, whether in the oil film or supports, helped stabilize this rotor.
4. Bearing damping is highly important in suppressing whirl; the nodes of the flexed shaft should not be at the bearings for that reason.
5. Properly designed seal rings provide a moderate damping force and need not be a major factor in FFW creation or suppression. Without careful design, they can be a major contributor.
6. Rotors subject to FFW can be triggered by subtle changes — i.e., discharge check valve cracking, compressor shutdown, or oil temperature variation. When triggered, the motion may be violent and self-destructive in seconds.
7. Aerodynamic drag forces should be considered an important FFW exciter. High pressure, high mole weight gases, large surface area (diameter) impellers and poor surface finishes are probable significant accentuators of whirl problems.

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