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# A HISTORY OF DEVELOPMENT IN ROTORDYNAMICS - A MANUFACTURER'S PERSPECTIVE

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The subject of rotordynamics and instability problems in high performance turbomachinery has been a topic of considerable industry discussion and debate over the last 15 or so years.

This paper reviews an original equipment manufacturer's history of development of concepts and equipment as applicable to multistage centrifugal compressors.

The variety of industry user compression requirements and resultant problematical situations tends to confound many of the theories and analytical techniques set forth. The experiences and examples described herein support the conclusion that the successful addressing of potential rotordynamics problems is best served by a fundamental knowledge of the specific equipment. This in addition to having the appropriate analytical tools. Also, that the final proof is in the doing.

## INTRODUCTION

While the subject of "rotordynamics" encompasses a broad range of lateral and torsional considerations, this presentation briefly reviews the manufacturer's efforts to correlate analytical procedures with machine operation including:

- verification of indicated critical speed with analytical results
- the influence of various components on the rotating system's behavior
- aerodynamic influences.

The review continues with the development of rotating system components and their successful application in a variety of services, and compares the results of this development and experience with a previously published graphical representation. (ref. 1, 2).

## NOMENCLATURE

Values are given in both SI and U.S. Customary Units. The measurements and calculations were made in U.S. Customary Units.

BHP = power, brakehorsepower
cfm = flow, cubic feet per minute

| | | |
|---|---|---|
| Hz | = | frequency, hertz |
| in. | = | length, inches |
| KPa | = | pressure, kilopascal |
| $kg/m^3$ | = | density, kilogram per cubic meter |
| kW | = | power, kilowatt |
| $lbm/ft^3$ | = | density, pound-mass per cubic foot |
| $m^3/hr$ | = | flow, cubic meter per hour |
| mil | = | vibration, 0.001 inch |
| mm | = | length, millimeter |
| N | = | running speed |
| NC | = | critical speed |
| $NC_1$ | = | first bending critical speed |
| psi | = | pressure, pound-force per square inch |
| r/min | = | speed, revolutions per minute |
| $\mu m$ | = | vibration amplitude, micrometer |

# ROTORDYNAMICS

## Phase One Testing

In the late 1960's a test rig (fig. 1) was established to monitor seal and rotordynamic behavior.

Reviewing the test rig (fig. 2), the casing was a standard multistage centrifugal compressor frame with pressure capability of 34,500 KPa (5000 psi), and speed capability to 14,000 r/min.

The casing was equipped with a rotor consisting of dummy weights installed on a shaft to simulate impellers. The test vehicle configuration allowed installation of a variety of bearing and seal combinations, variable rotor geometry and application of unbalance weights.

In this test rig, rotordynamic influences were monitored through a range of pressures without the influence of aerodynamic effects which normally result from gas compression.

Initial testing evaluated a rotating system configuration representative of components in use at the time. The configuration consisted of a rotor with a 1600 mm (63 in.) bearing span having ten weights installed simulating impellers. The bearings were tilting pad type having five shoes. The seals were standard ring type oil film seals (fig. 7) of low profile geometry (fig. 8).

Bearing vibration results for a speed range through 14,000 r/min are shown on figure 3. Rotor midspan vibration is shown on figure 4. The data definitions for figures 3 and 4 are made in Table 1.

Testing was done using two case (and therefore seal) pressures [1030 KPa (150 psi) and 6900 KPa (1000 psi)] and with a "tight", 0.127 mm (0.005 in.) and "loose", 0.241 mm (0.0095 in.) bearing clearance.

It was evident from reviewing the characteristics of peak locations (fig. 3, curves A vs. B, and C vs. D), and the vibration discontinuity evidenced in figure 3, curve C that both bearing and seal characteristics influence rotordynamic behavior.

Figures 3 and 4 also presented difficulties in data interpretation. For example, in figure 3, note the low [less than 12.7 $\mu$m (0.5 mil)] vibration level in the first critical speed range and the difficulty in pinpointing these critical speeds. Also, in figure 4, note the low speed amplitude being an appreciable portion of the full speed amplitude as well as the initial decrease in amplitude as speed increased. Phase data recorded was erratic and inconclusive.

These characteristics suggested a form of runout and the runout to be out of phase with the unbalance. Since rotor instability, as experienced, was associated with vibration at the first bending mode, correct modeling of the parameters influencing the first critical speed was important.

## Phase Two Testing

From the foregoing review of data, it was determined that more detailed testing was necessary to overcome the difficulties of data interpretation. Testing during this phase was set up to:

- operate without seals thereby eliminating the apparent seal effects on critical speed;
- intentionally unbalance the rotor at midspan to give a clearer indication of critical speed;
- run with "tight" and "loose" clearance bearings.

This phase of testing would be used to verify the analytical capability to predict rotor response using available bearing and unbalance response programs.

Figures 5A and 5B compare analytical and test results for bearing and midspan vibration vs. speed data for a "tight," 0.102 mm (0.004 in.) clearance bearing.

There is good correlation between the test (solid line) and calculated (dotted line) first critical frequency.

Figures 6A and 6B compare analytical and test results for bearing and midspan vibration respectively for a "loose," 0.241 mm (0.0095 in.) clearance bearing. Note in this data, there are some test coupling unbalance effects and second bending critical speed effects at higher speeds. The calculated values were analyzed with unbalance modeled to compare to only the first critical speed.

The amplitude ratio data of figures 5A and 5B, and 6A and 6B (summarized in Table 2) emphasize the ever present requirement for compromise in compressor component design/application.

From Table 2, while tight bearing provides a lower indicated vibration at running speed, which may be considered advantageous to the user, the ratios at critical speed indicate a more sensitive situation than the loose bearing.

The impact of subtle differences on the design of critical components and the impact on compressor operation must be a prime consideration in revamping or replacing parts.

## Phase Three Testing

Having recognized from the first phase of testing that oil film seals had a demonstrated effect on rotordynamics, the third phase of testing was established to evaluate various seal configurations and establish a basis for analytical predictability.

Figure 7 shows a typical ring type oil film seal in cross-section. Due to the axial load associated with the high pressure drop across its unbalance area, the outer ring is the component which influences rotordynamics.

Figure 8 shows several variations in outer ring geometry that result in different seal effects at a given pressure differential.

Testing was conducted with these different, albeit somewhat conventional, seal designs at varying seal pressures and varying levels of rotor unbalance. Test results indicated an unsettling effect of unbalance at low axial loads and indicated highly loaded seals of this geometry to be unpredictable.

Data from tests of these various geometries also provided a plausible explanation for the vibration discontinuity observed in figures 3 and 4. Since predictability is a requirement for reliability, it was determined that a new approach to seal geometry must be taken.

Several seal designs were conceived and tested. The tilt pad seal (fig. 9) evolved as the solution to the problem of predictability of seal effects while eliminating the propensity for oil whirl which had emerged as a problem during testing of other seal geometries.

Verification of the tilt pad seal geometry included testing various oil film clearances enabling this parameter, as well as axial load and unbalance to be included in the analysis as an accurate representation of the rotordynamic system. Upon verification, the tilt pad seal was released to production units and has been providing reliable service for over 12 years.

With the foregoing, the first three phases of the test program were complete. During these phases, over 200 tests were run to evaluate rotordynamic parameters.

# ROTOR INSTABILITY

With this progress in analytical capability and machinery experience, the inevitable result was to extend the equipment to higher heads, higher case lift, and higher pressures by design innovations such as back-to-back construction, variable stage spacing, inboard thrust bearings, and high pressure seals. Along the way in this evolution, a vibration problem defined as rotor instability was encountered. The rotor instability was evidenced by a pulsating vibration at subsynchronous frequency, the amplitude of which would increase, resulting in rotor interference with static parts. It was determined that the vibration was aerodynamically excited and the frequency coincided with the rotor's first bending critical speed.

This manufacturer's exposure to the subsynchronous vibration problem first occurred in the early 1970's. The problem surfaced in widespread geographical locations and encompassed both synthesis gas and natural gas compressors in applications at moderate and higher pressures. The approach taken was not to abandon the extended capability and advantages inherent in the design philosophy, but to improve understanding and solve the problem through incorporation of design advances. It was recognized that to accomplish this would require a test vehicle which had experienced a demonstrated, rotor instability in order to verify or evaluate design modifications.

## Instability Testing

A compressor that had experienced serious subsynchronous vibration was set up in the test facility (fig. 10) for full pressure, full power operation. A comparison of design and test capability conditions are shown in Table 3.

A series of 30 tests were run on a helium/nitrogen mixture through a range of flows, pressures and speeds.

The first series of tests (one through six) were baseline runs using the compressor as originally built, including five shoe, tilt pad bearings and standard ring type seals. These tests verified the field experience could be duplicated by the shop test. An example of the data which shows the impact of the subsynchronous component on midspan vibration is shown in figure 11. Note the Y-axis of this oscilloscope picture is vibration at 2.54 $\mu$m/Division (0.1 mil/Division) and the X-axis represents time, in this case, ten seconds. This data represents the maximum speed (9000 r/min) that could be achieved prior to completely unstable operation.

The remainder of the testing applied many of the bearing and seal component configurations to the "real condition" operating environment in combinations which had been shown to be successful in prior testing without aerodynamic influence, and which had already shown promising results when installed in field problem units. In addition, this test program examined a variety of modifications to internal hardware believed to influence the aerodynamic forces on the rotor system, as well as investigation of friction effects of couplings and shrunk on parts.

Figure 12 is indicative of the results of this testing, again showing midspan vibration and data at 2.54 $\mu$m/Division. The configuration included in this data included:

- damper, tilt pad bearings, five shoe;
- tilt pad seals;
- aerodynamic adjustments to stationary parts.

The success of these programs is represented by the application of these concepts, analytical techniques and components to a quantity and variety of user compressor requirements. It is important to note these components and concepts have been applied successfully to multistage compression equipment of both back-to-back, as well as straight-through rotor arrangements.

## Recent History

With the addition of hydrocarbon gas capability to the existing inert gas full load, full pressure test facilities in LeHavre, France, and Olean, New York, the manufacturer has expanded capability for evaluating rotordynamics/stability behavior in "real-life" conditions.

In 1985, the opportunity was presented to evaluate a compressor's performance under ASME Power Test Code 10, Class 1 conditions. The hydrocarbon gas blend was to match a unique "natural gas" injection application and at the same time, match and verify other aspects of the gas properties and gas behavior.

The compressor configuration was back-to-back, through-flow, without intercooling, having a total of six impellers and fitted with squeeze-film, tilt pad bearings, tilt pad seals, and the special aerodynamic division wall design; all of which had been proven successful on prior development tests and in long-term field operation. Additionally, all internal labyrinths were of conventional design. Bearing span, as well as other geometric parameters, were well within prior experience. Unique to this application was the hydrocarbon 35 mol weight gas being compressed to the 31,030 KPa (4500 psi) design, 33,100 KPa (4800 psi) maximum pressure.

During the full pressure, hydrocarbon performance test, a subsynchronous vibration component appeared and increased in intensity as pressure was increased going back toward surge on the 100 percent speed line. The subsynchronous vibration data during this test, although at very low levels [peaks to 7.6 $\mu$m, (0.3 mil)], had a pulsation characteristic that would be of potential concern to the operator. It should be noted that later inspection of compressor internals showed absolutely no distress to labyrinth seals or any other internal (or external) components that, if existed, would be indicative of high midspan excursions. Figure 13, Test A, shows representative data at the highest pressure tested during performance testing. The data is bearing vibration shown on a time-sprectrum plot. Table 4, Column A shows basic test conditions at that point. Note for reference, the highest vibration amplitude shown on figure 13 is 12.7 $\mu$m (0.5 mil) (Test C, 147 Hz).

On this project, two duplicate compressors (units I and II) were being supplied. While the compressor being discussed here (unit I) was being tested on hydrocarbon gas, the sister unit (unit II) was being tested at another facility on an inert gas and had not exhibited the subsynchronous component during its preliminary testing. Based on this, it was decided to run unit I on nitrogen at the highest pressures that could be achieved on that gas (similar to the conditions already experienced by unit II). This test is designated test B on figure 13 and Table 4. The results confirmed units I and II, under these conditions, had very similar operating characteristics free of subsynchronous vibration. This tended to lead the investigation to a detailed review of the compressor design as opposed to suspecting a random type problem such as might have occurred during assembly. This test also confirmed the need to run final verification tests on a gas blend closely duplicating field gas conditions.

Close inspection during disassembly confirmed correct parts assembly. However, scrutiny of the parts and manufacturing drawings revealed the stationary, flow-path components had not received some of the detailed design features that had been applied to other compressors in operation. These stationary components were remachined to conform to prior experience.

Test C (fig. 13 and Table 4) was the final verification test at maximum required pressure and speed. This test verified the adjustments made to the aerodynamic flow path stationary components brought the subsynchronous component to a low amplitude [peaks to 1.9 $\mu$m, (0.075 mil)], stable condition.

## RESULTS

The results of these, and similar test programs, are reflected by the experience in a wide variety of application circumstances. This wide variety of applications, it should be added, reinforces the analytic approach through incorporation of experience data into the evolution of the analytical process.

Based on knowledge gained from these research efforts, and application experiences, the manufacturer is hesitant to embrace as absolute many of the analytical techniques and empirical criteria presently published or available.

One such empirical criterion is represented by figure 14 (ref. 2). This representation plots points based on a compressor's flexibility ratio and average gas density in operation, with flexibility ratio defined as compressor maximum continuous speed divided by the first critical speed on stiff supports. These points are then compared to the "worst case" threshold line with the area above the line indicated as "unstable region" and the area below the line indicated as "safe region". According to the author (ref. 2), "The 'worst case' line given should be a useful rule-of-thumb for indicating a threshold-of-concern for subsynchronous instability in similar industrial centrifugal compressors."

To assist in putting such a chart into perspective, the parameters of figure 14, including the "worst case" line, have been used as a base for plotting a portion of the manufacturer's experience without showing duplicate units (fig. 15).

On figure 15, units numbered 1 through 46, represent a wide range of applications including natural gas (21 units), synthesis gas (12 units), as well as $CO_2$, air injection and mixed hydrocarbon service. Also represented is a wide range of service pressures from approximately 6900 KPa (1000 psi) through 72,415 KPa (10,500 psi). The period covered is 1969 through 1983. (Since this representation is considered to be only an illustrative tool, later experience has not been added.) As noted, the unit numbers enclosed in squares or boxes represent those units which were full load/full pressure tested prior to shipment. All other units were shipped having received, from a mechanical standpoint, only API-617 testing. For further reference, locations on this plot of test points A, B, and C from fig. 13 are shown.

## CONCLUSION

The subject of rotordynamics and stability is a complex, technical issue made more complex by the constantly changing users' compression requirements typical of the multistage compressor industry. This history of development and testing serves to demonstrate that causes of subsynchronous excitations are not particular to any one area of compressor design; i.e., bearings, main seals, internal labyrinths, stationary components, impeller inlets, exits, etc. Nor is the phenomenon unique to a given configuration. Therefore, the solutions to these problems cannot be addressed by close examination of a singular element or component of the compressor assembly.

The described development testing and operating experience has allowed this manufacturer to establish the analytical processes by continual data feedback, as well as to conceive and develop bearing and seal components, and aerodynamic concepts as necessary to address solutions to rotordynamics/stability problems.

Some industry publications (ref.'s 1, 2, 3, 4, 5, 6, 7, 8) covering a span from 1976 through to as recently as 1984, would imply to the reader that when problems arise, solutions requiring major geometry changes to compressor shafting, cases and stationary components are to be considered cost-effective solutions based on current state-of-the-art of theory and application. "Cost-effective" has been described, at least as associated with one reported incident, as between 3 million (ref. 2) to 4 million (ref. 3) Pound Sterling.

To date, problems addressed by this manufacturer have not required the radical solutions implied as necessary by the aforementioned references. More typical of the manufacturer's experience is the Arun, 49,060 KPa (7115 psi) injection experience (ref. 9). Despite such successes, it has been recognized that additional development is necessary. To this end, the test vehicle (fig. 10) has been re-established in the test facility to enable identification and quantification of the mechanisms leading to rotor instability.

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TABLE 1

Curve Definitions for Figures 3 and 4

| Line | Case (Seal) Pressure | | Bearing Clearance | |
|---|---|---|---|---|
| | KPa | psi | mm | in. |
| A | 6900 | 1000 | 0.127 | 0.005 |
| B | 1030 | 150 | 0.127 | 0.005 |
| C | 6900 | 1000 | 0.241 | 0.0095 |
| D | 1030 | 150 | 0.241 | 0.0095 |

TABLE 2

Vibration Ratio Comparisons
(Based on Test Data Figures 5 and 6)

1. Midspan to bearing vibration ratio at first critical speed ($NC_1$):

   tight clearance - 5.20
   loose clearance - 1.70

2. Tight to loose bearing clearance vibration ratio:

   a. midspan vibration
      at $NC_1$ - 5.20
      at 12000 r/min - 0.56

   b. bearing vibration
      at $NC_1$ - 1.70
      at 12000 r/min - 0.14

TABLE 3

Operating Conditions

| | Field | Test |
|---|---|---|
| mol wgt. | 11.6 | 11.0 |
| Flow, $m^3/hr$ | 6,540 | 6,610 |
| ft3/min | 3,850 | 3,890 |
| Inlet pressure, KPa | 1,720 | 1,585 |
| psi | 250 | 230 |
| Discharge pressure, KPa | 10,342 | 10,690 |
| psi | 1,500 | 1,550 |
| Power, KW | 10,146 | 10,205 |
| bhp | 13,600 | 13,680 |
| Speed, r/min | 10,436 | 11,000 |
| Bearing span, mm | 1,753 | 1,753 |
| in. | 69 | 69 |
| Number of impellers | 11 | 11 |

TABLE 4

Hydrocarbon Test Conditions

| Test | A | B | C |
|---|---|---|---|
| Gas | H.C. | $N_2$ | H.C. |
| Mol weight | 34.6 | 28 | 35.6 |
| Inlet pressure, KPa | 14,135 | 19,300 | 14,135 |
| psi | 2,050 | 2,800 | 2,050 |
| Discharge pressure, KPa | 32,910 | 30,340 | 33,100 |
| psi | 4,773 | 4,400 | 4,800 |
| Speed, r/min | 8,305 | 8,800 | 8,800 |
| Power, KW | 5,425 | 3,540 | 10,986 |
| BHP | 7,273 | 4,746 | 14,726 |

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1969 - SEAL / ROTOR RESPONSE FACILITY TEST RIG

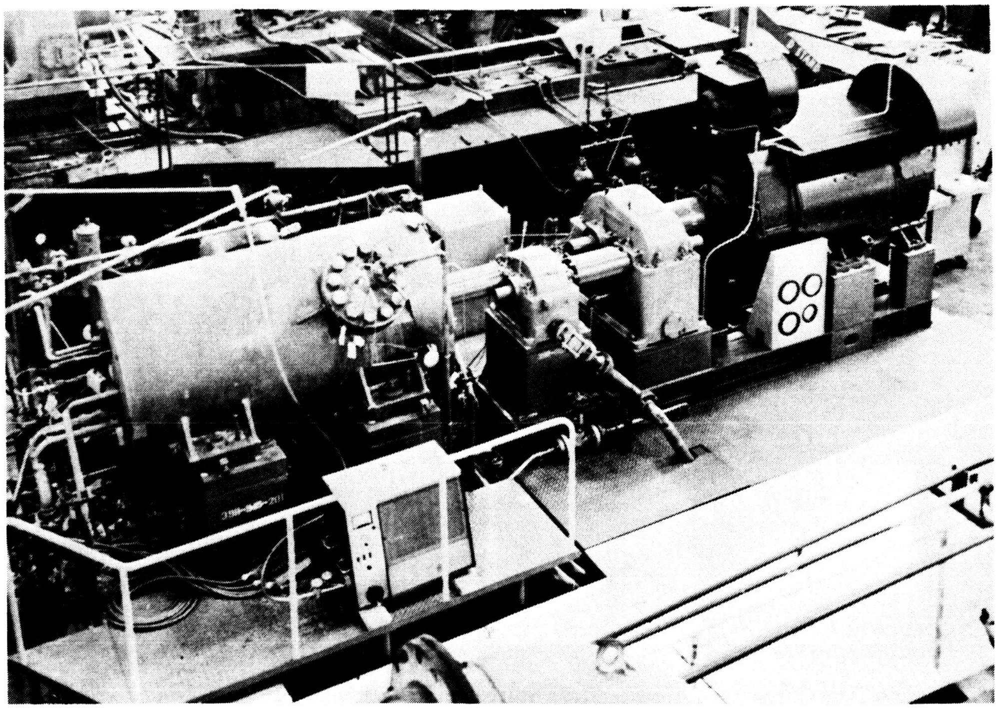
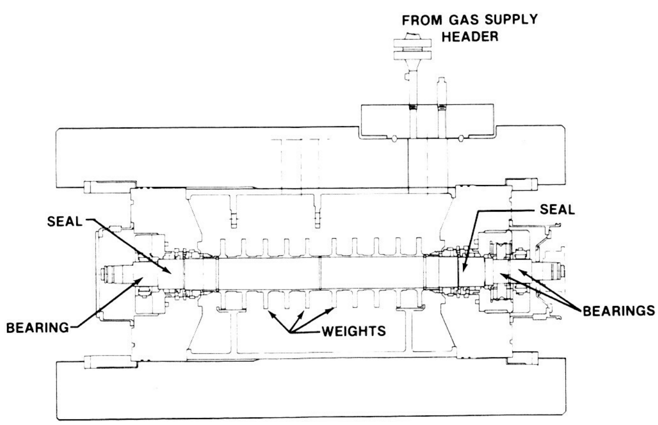


**Figure 1**




FACILITY - TEST RIG

**Figure 2**

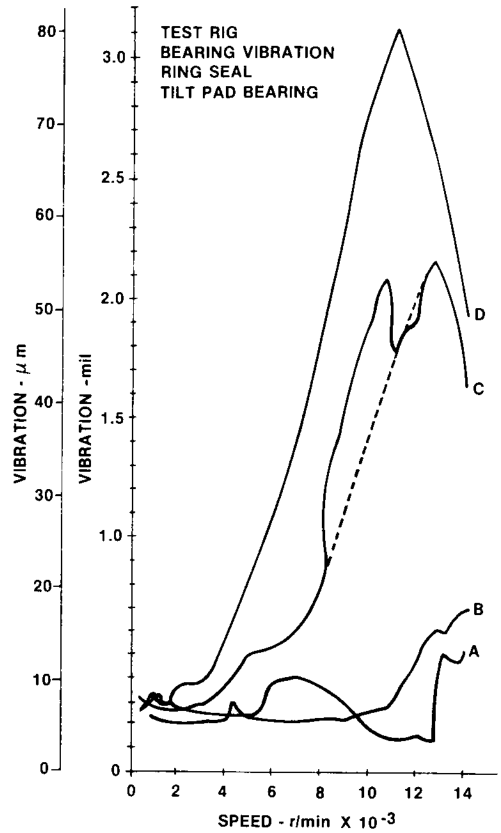



**Figure 3**

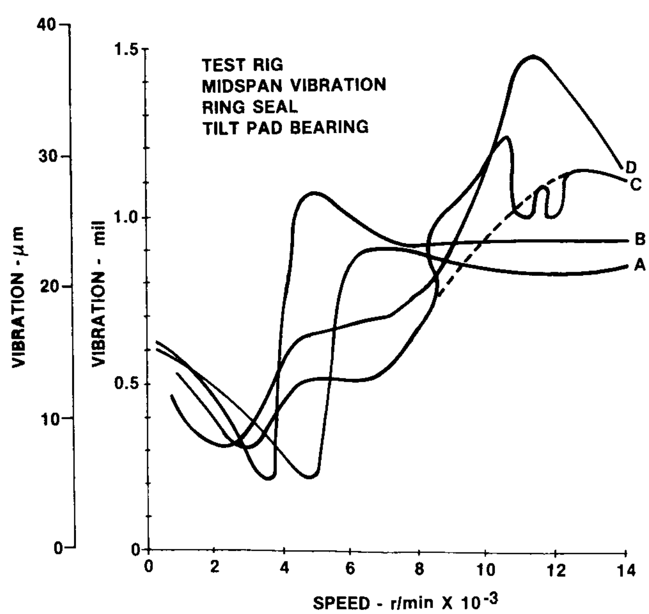



**Figure 4**

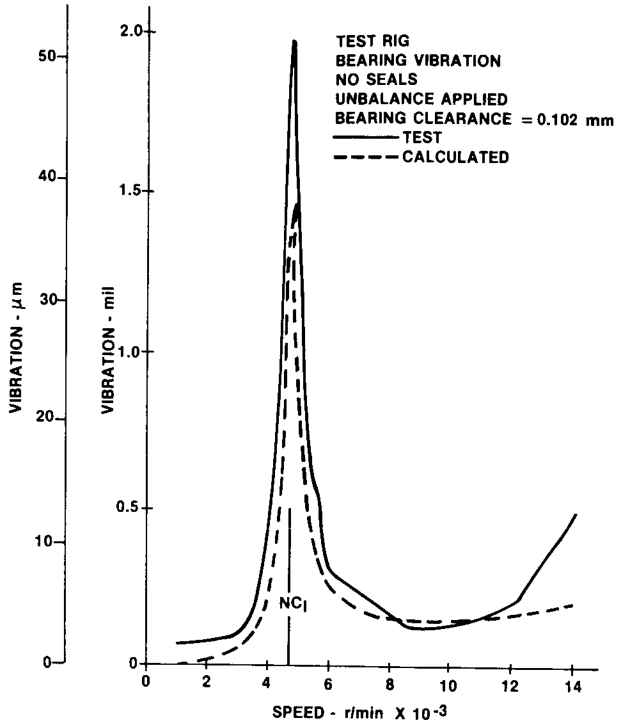



Figure 5A

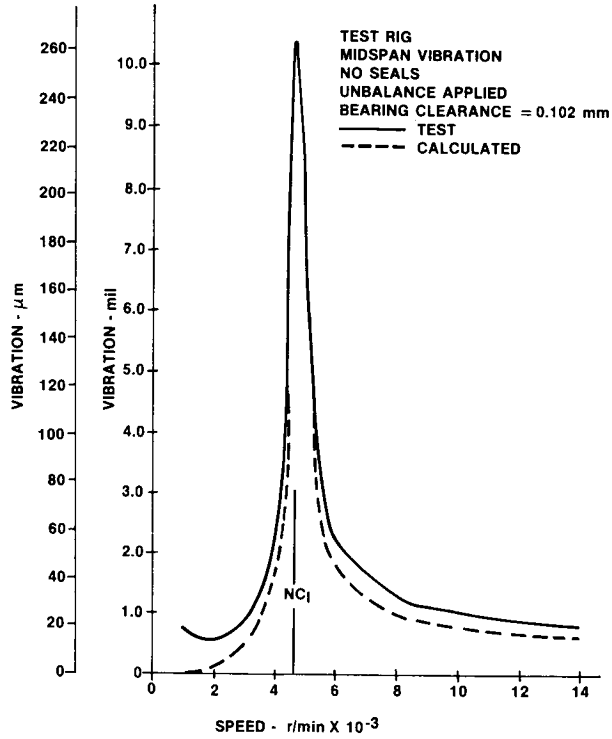



Figure 5B

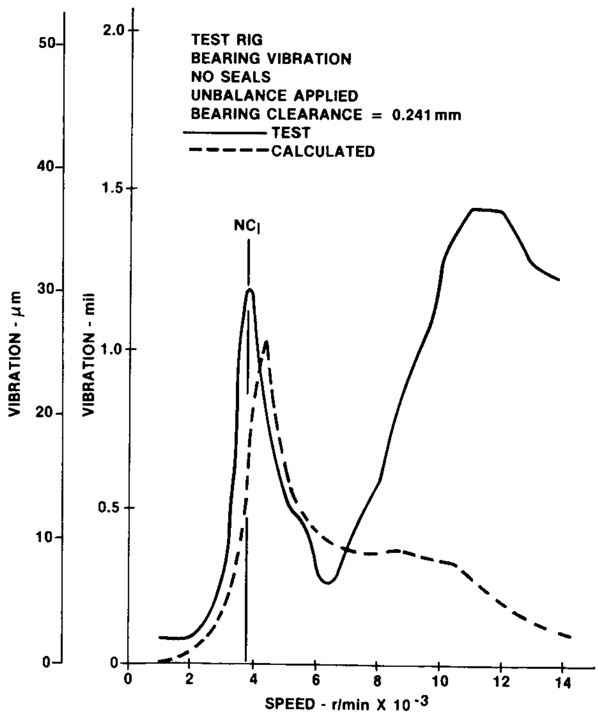



Figure 6A

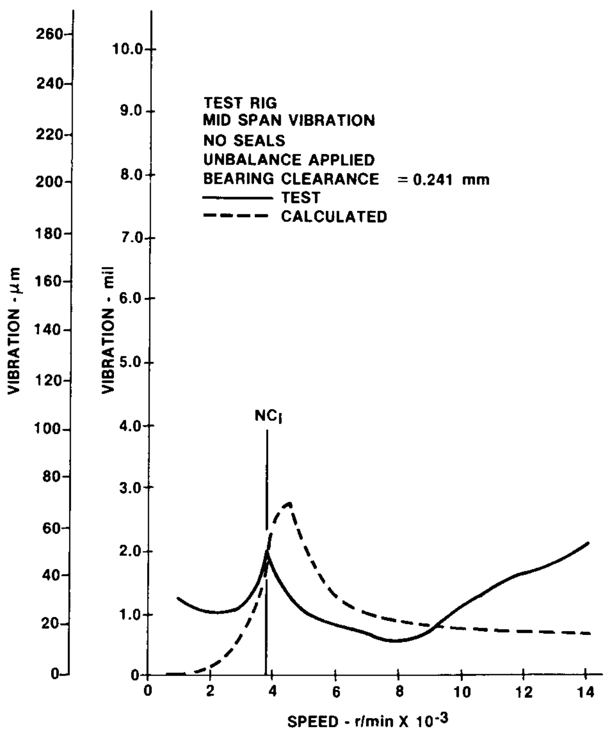
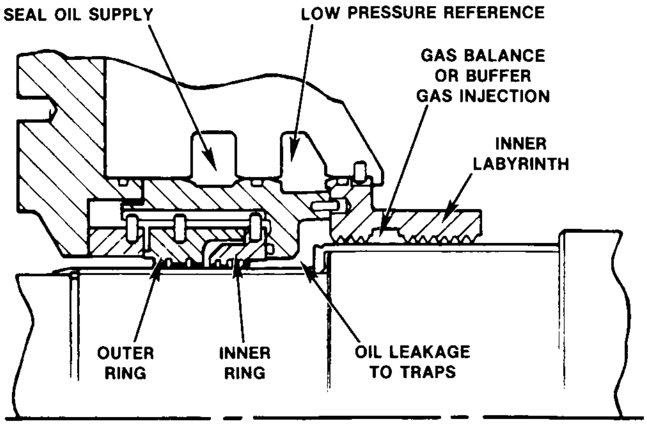
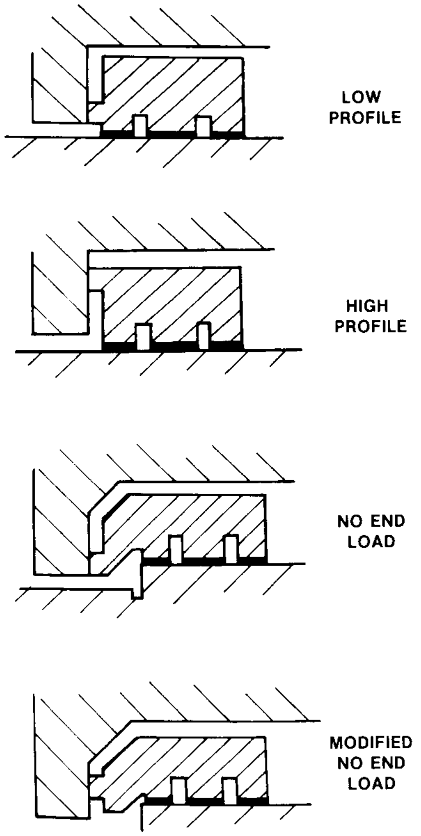



Figure 6B




RING TYPE OIL FILM SEAL

**Figure 7**




OUTER RING GEOMETRY

**Figure 8**

# TILT PAD SEAL



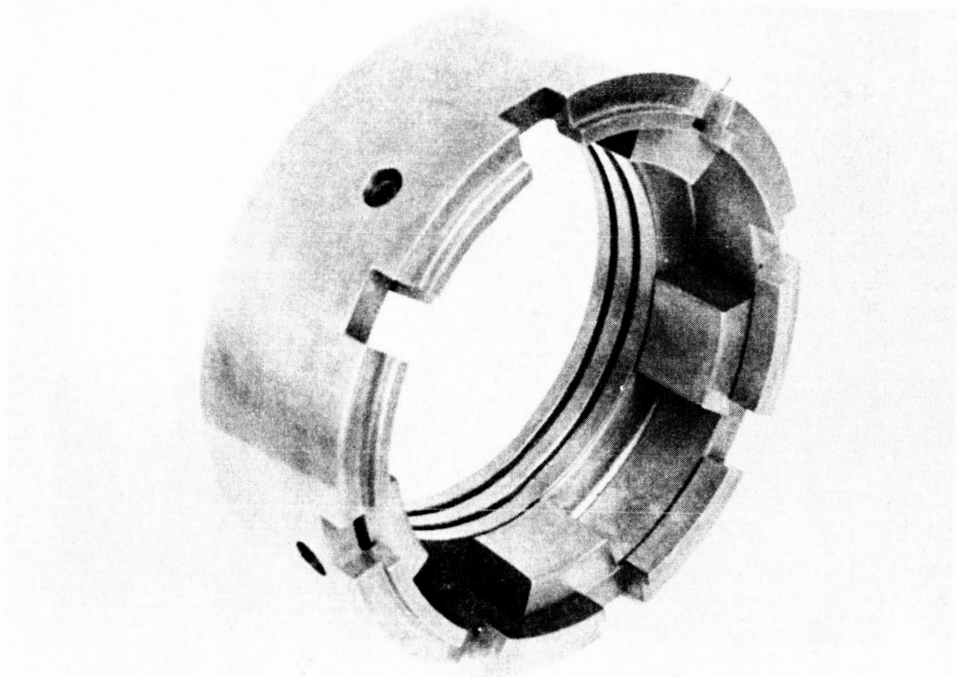


Figure 9

1973 - FULL LOAD / FULL PRESSURE TEST VEHICLE

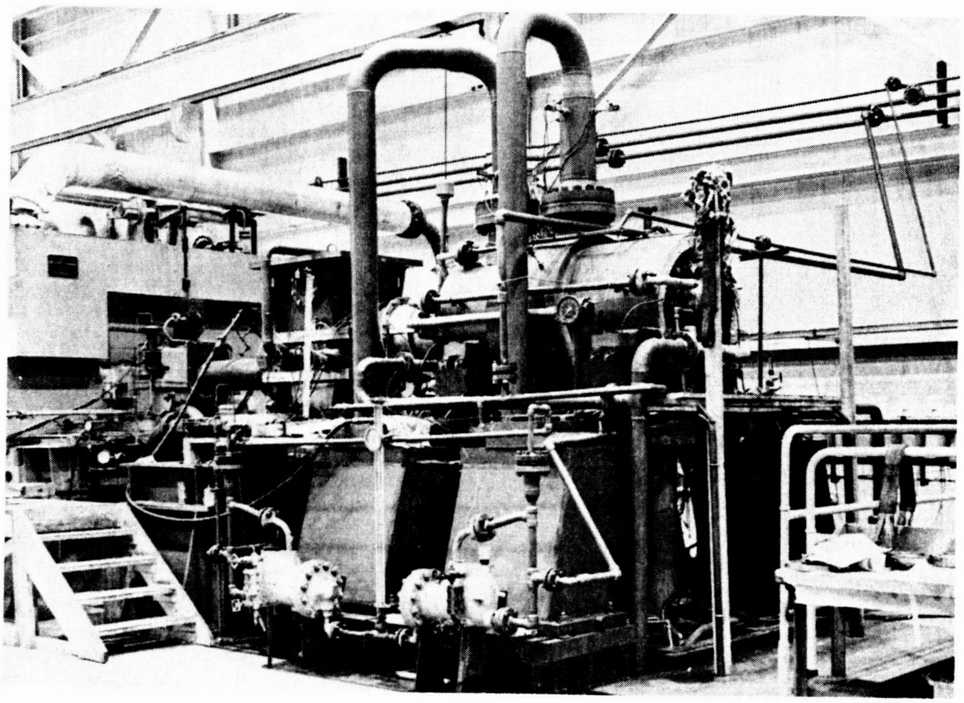
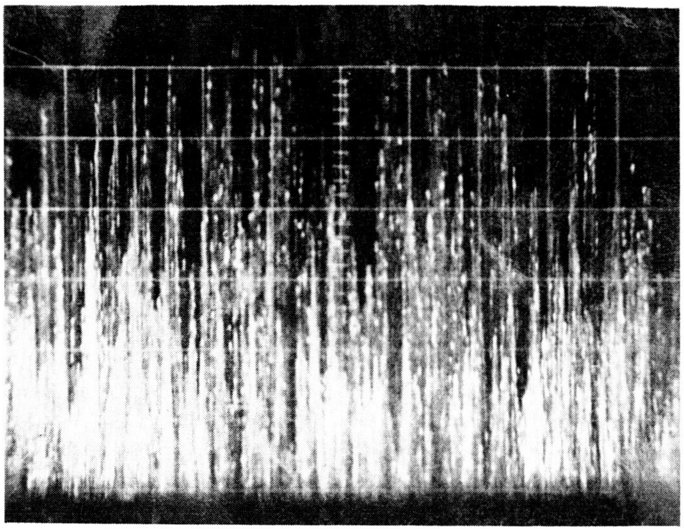


Figure 10

## VIBRATION AMPLITUDE VS TIME

TEST #6 V3H RUN 3



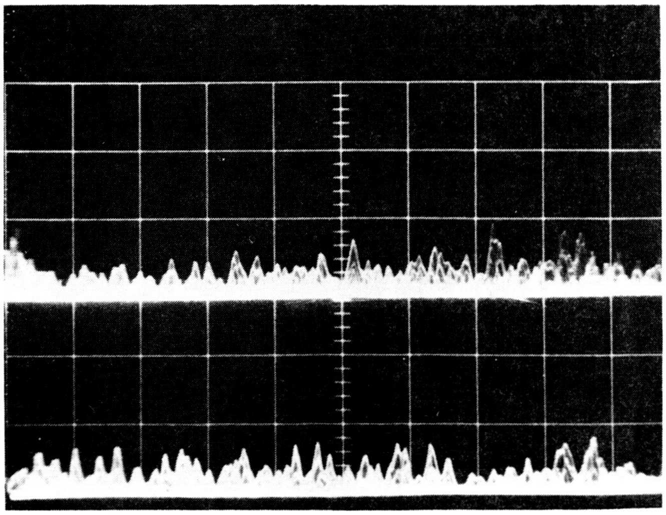
0.1 mil / division →

N = 9000 r/min — 10 sec —

Figure 11

## VIBRATION AMPLITUDE VS TIME

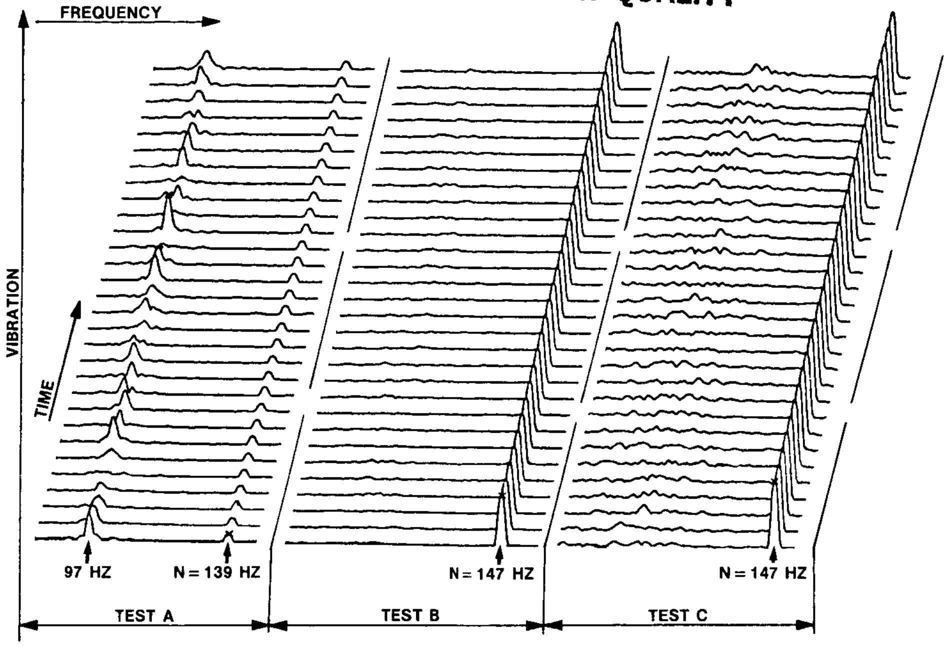
TEST 22 V3H RUN 8



0.1 mil / division →

N = 10,900 r/min —10 sec —

Figure 12




TIME SPECTRUM PLOT

**Figure 13**

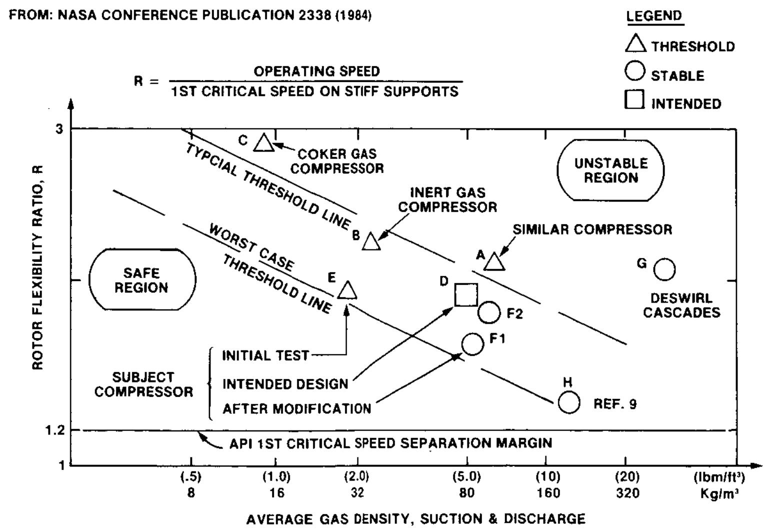



**Figure 14**

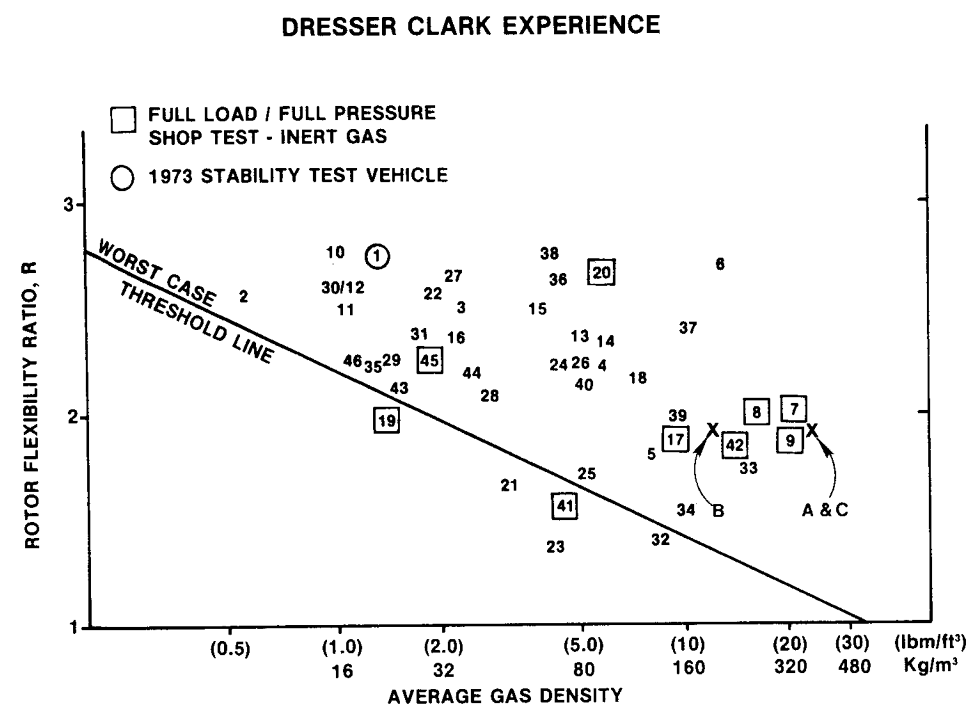

DRESSER CLARK EXPERIENCE
FULL LOAD / FULL PRESSURE
SHOP TEST - INERT GAS
1973 STABILITY TEST VEHICLE
WORST CASE
THRESHOLD LINE
ROTOR FLEXIBILITY RATIO, R
AVERAGE GAS DENSITY
(lbm/ft³)
Kg/m³
A & C
B

**Figure 15**