

## PRACTICAL EXPERIENCE WITH UNSTABLE COMPRESSORS

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### SUMMARY

High and low pressure compressors which operate well above their fundamental rotor-bearing lateral natural frequencies can suffer from destructive subsynchronous vibration. Usually the elements in the system design which contribute to this vibration, other than the shafting and the bearings, are the seals (both gas labyrinth and oil breakdown bushings) and the aerodynamic components.

Using analytical mathematical modeling techniques for the system components, an attempt is made to gauge the destabilizing effects in a number of compressor designs.

Recommendations are made, based on experiences with stable and unstable compressors, which can be used as guides in future designs.

### INTRODUCTION

During the past ten years, the author had the experience of analyzing new designs or "trouble-shooting" existing machines suffering from high vibration. The practical experience gained is the main subject of this paper.

Over 200 single-stage, custom designed, overhung, high-speed compressors have been shipped by the author's firm. These machines compress such gases as air, nitrogen, oxygen, hydrogen, ammonia, natural gas, carbon dioxide, carbon monoxide, and freon. All have been analyzed before manufacture and have functioned well in the field. The methodology used in the rotor-bearing design analysis is referenced herein.

Over 50 compressors of predominately the "straddle-mount" design have been analyzed to determine the causes for high vibration and to propose subsequent corrective actions. A number of these compressors have been selected as representative examples of subsynchronous vibration and are discussed herein.

Before getting into the detailed discussion of these experiences, a review of the literature on the subject of subsynchronous vibration in turbomachinery is given.

It should be noted that a complete list of references with discussions on the subject of rotor-bearing dynamics emphasizing subsynchronous instabilities could be the subject of another paper since there have been numerous important contributions in this area during the past seven years. The reference list provided herein cites publications which are familiar to this author and are related

to the theme of the present paper. By no means is this list meant to be complete.

#### DISCUSSION OF REFERENCES

In 1973, Reference 1, which should be considered a classic in rotor dynamics, was presented at an ASME conference. This paper offered a method for calculating the damped critical speeds of a general flexible rotor in fluid-film bearings. The recommended procedures for preparing the mathematical model for the rotor, bearings, seals, and fluid dynamic forces and the gauging of the rotor-bearing system sensitivity to vibration in terms of a system logarithmic decrement have been followed by the author's firm. The extensive use of this methodology over this past time period has provided the author with the experience reported herein.

Procedures for rotor-bearing dynamic analysis have been recommended in References 2, 3, 4, and 5, and it would behoove the manufacturers and users of rotating mechanical equipment to be aware of the content of these papers.

Lack of confidence was previously stated in the use of these analytical methods in References 6 and 7 as a substitute for full-load testing. However, actual tests (Reference 8) and experiences have provided a better understanding such that confidence has been regained — at least in this author's viewpoint.

Practical experiences with high pressure compressor vibration instabilities, with which this author is intimately familiar, are reported in References 9, 10, 11, and 12.

Various important contributions that have been made which should aid in the preparation of the mathematical models of the bearings, seals, and fluid dynamic components are included in the reference section. In the bearing area, References 3, 13, 14, 15, and 16 are cited. In the squeeze-film, damper bearing area, References 17, 18, 19, and 20 are cited. In the oil breakdown bushing seal area, References 21 and 22 are useful. In the labyrinth seal area, including the balance piston, References 12, 23, 24, and 25 should be referred to. In the area of aerodynamics or fluid dynamic excitation, References 23, 26, 27, and 28 form an excellent foundation.

As mentioned above, confidence in available analysis techniques, such as Reference 1, has grown. Presently, manufacturers of compressors are analyzing, designing, and manufacturing compressors without troublesome subsynchronous vibrations (References 29 and 30).

An excellent discussion on mechanisms causing unstable whirl in rotating machinery is presented in Reference 31. One message obtained from this reference is that liquids can either provide significant benefits or problems depending on how they act with the rotor.

It is the author's opinion that considerable knowledge has been accumulated over the last ten years in the area of vibrations in rotating machinery in general and in subsynchronous whirl in particular.

A serious problem in compressor design and/or trouble-shooting with regard to rotor dynamics and vibration sensitivity is the inability to prepare the proper mathematical model of the contributing members and to police the design, manufacture, and assembly procedures such that the model agrees with the machine and vice versa.

In the following, a discussion is given on a number of case histories where subsynchronous instability was a predominant vibration problem. Where possible, a gauging of the damping or lack of damping is given in terms of system log decrement (References 1 and 2). Also, practical problems which reflect upon the analytical model versus the actual machine are discussed.

#### DISCUSSION OF EXPERIENCES

The following discussion is divided into two sections where first the overhung (or cantilevered) compressor design is addressed and, second, the straddle-mounted (or simply supported) compressor design is addressed.

##### Overhung Designs

In Reference 20, reference is made to two particular three-stage overhung design, low pressure compressors that were completely stabilized by employing squeeze-film damper bearings. The rotors are supported on tilt-pad bearings and only labyrinth seals are used.

In one of these designs which compresses air and operates at 52,000 rpm, well above its first rotor-bearing natural frequency (approximately 12,000 cpm), the base rotor-bearing log decrement varies from 0.05 to 0.25. This range is a function of the clearance tolerance in the tilt-pad bearings. The machine is marginally stable; however, it did run stably after modifications were made to the wheel-to-shaft fits and the oil distribution within the bearing pad clearances. This is a case where the machine did not represent the mathematical model. In overhung designs, which operate at high speed, gyroscopic stiffening is important and can be present only if the wheel-to-shaft fits are retained at operating speed. Furthermore, in gear-driven units, the gear loads on the bearings have a variable load direction with speed which must be accounted for. A good design for this application is to have each tilt-pad in the bearing serviced by an individual inlet restrictor.

In a second three-stage overhung design which operated at 33,000 rpm, and also well above its first rotor-bearing natural frequency of approximately 9000 cpm, the rotor-bearing log decrement was calculated to range from 0.08 to 0.15 because of clearance tolerances in the tilt-pad bearings. This machine had proper wheel-to-shaft fits and properly lubricated bearings and ran stably in air. However, when compressing CO<sub>2</sub>, the machine was unstable in a subsynchronous mode. Because rotor modifications were impractical, a squeeze-film damper bearing was employed to achieve a log decrement well above 0.5 and a stable running machine. This machine provides a gauge on the stability of machines with gases with molecular weights of 29 and 44. (Note that compressor dynamic stability is affected by gas density and mass flow levels. However, for machines operating with similar temperatures and pressures the gas molecular weight becomes a convenient gauging parameter.)

Lessons that can be learned from these experiences are as follows:

- Wheel-to-shaft fits must be retained at all speeds.
- Tilt-pad bearings must have the proper oil distribution in order to achieve the calculated stiffness and damping.
- Higher molecular weight gases do provide larger fluid dynamic forces, thus necessitating a higher base system log decrement.

In Reference 29, the state-of-the-art of single-stage, high-speed, overhung compressor design is discussed. This author had the opportunity to make rotor-bearing dynamic analyses on over 200 compressors of this design. None of these compressors is suffering from subsynchronous vibrations. The basic elements that comprise the dynamic mathematical modeling of this machine are a flexible, overhung rotor, tilt-pad bearings possessing stiffness and damping, and fluid dynamic forces at the wheel and/or gas labyrinth seals. These rotor-bearing systems are operating above the first natural frequency and below the second.

A gauge that has been followed for this machine design is the minimum allowable system base, log decrement versus gas molecular weight (MW). For example, the following is a list of system log decrement versus gas molecular weights for machines that have run stably above their first damped natural frequency and are free from subsynchronous vibrations:

<u>Gas</u>	<u>MW</u>	<u>Minimum Log Decrement</u>	<u>Ratio of Operating Speed to First Damped Natural Frequency</u>
Ammonia, Oxygen	17, 32	0.32 to 0.45	3 to 2
Carbon Dioxide	44	0.42 to 0.52	~2
Freon-Air Mixture	94	0.62 to 0.75	2 to 1.5

These values have been used to provide machines with stable operation and are not to be misconstrued as threshold values.

The key parameters for achieving high stability (high log decrement) for these compressors are

- Minimum overhang length
- Minimum wheel weight and inertia
- Maximum shaft diameter between overhang and inboard bearing centerline
- Bearings with a nominal machined clearance ratio of 3 mm/m, a nominal geometric preload of 0.15, and usually load orientation on a pad.\*

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\* It should be noted that changing the load orientation of the tilt-pad bearing is an effective means of gaining stability in marginal machines.

## "Straddle-Mount" Designs

An excellent discussion is given in Reference 30 on the design and full-load testing of a high pressure natural gas compressor with this mounting type. The particular 13,000 horsepower compressor, reported in this reference, operates at 6500 rpm and compresses natural gas (molecular weight of approximately 20) to pressures over  $2.76 \times 10^7$  Pa (4000 psig). The base rotor-bearing log decrement of this machine was reported at 0.44 with a damped natural frequency of 3900 cpm. The breakdown oil seals were designed to be ineffective as destabilizing elements. An aerodynamic cross-coupling stiffness was used at each impeller to represent the fluid dynamic excitation. The value of  $1.15 \times 10^6$  N/m (6570 lb/in.) per impeller brought the theoretical log decrement down to 0.19. The machine ran stably under full-load test.

This author has performed analyses on other similar compressors and some of the experiences are reported in the following paragraphs.

Reports on a very high pressure natural gas compressor are given in References 10 and 11. This particular eight-stage compressor is presently operating at 8500 rpm ( $\sim 20,000$  horsepower) with a discharge pressure approaching  $6.2 \times 10^7$  Pa (9000 psig). At various stages in this compressor development, calculations were made to assess the rotor dynamic stability of the machine. The first design had a base rotor-bearing log decrement of 0.21 with a damped natural frequency of 3500 cpm. However, it was determined that, if the oil breakdown seals were active dynamically, they could short circuit the effect of the bearing damping and reduce the log decrement to a low value of 0.030 with a damped natural frequency of 4300 cpm.

By tuning the mathematical model to the machine performance, it was further determined that a fluid dynamic excitation, in terms of radial cross-coupling stiffness, was present at approximately  $2.98 \times 10^5$  N/m (1700 lb/in.) per impeller. By deactivating the seals (pressure balancing) and incorporating a squeeze-film damper bearing, stable operation was realized. A log decrement value of  $1.0^+$  was achieved with the damper design. In the final machine design stage, a rotor with a larger shaft diameter was employed. This machine has a base rotor-bearing log decrement of approximately 0.6 with a damped natural frequency of approximately 4800 cpm and proved very insensitive to seal and aerodynamic excitations.

From the above, it would appear that, for natural gas injection compressors, an aerodynamic excitation cross-coupling stiffness of say  $3.5 \times 10^5$  N/m (2000 lb/in.) per wheel is a reasonable design value. A rotor-bearing system capable of retaining a positive log decrement with this excitation should be stable in the field. This statement is made based on the assumption that the oil breakdown seals are not locked or are deactivated dynamically.

A simple breakdown seal can have an axial force as high as  $\pi/2 D t \Delta p$ . For a seal with  $\Delta p = 1000$  psi, a diameter,  $D = 6.5$  in., a face of  $t = 1/4$  inch, the axial locking force is approximately 2500 pounds.\* For three seals and assuming

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\* In SI units, this would correspond to a pressure of  $6.89 \times 10^6$  Pa, a seal diameter and face width of 0.165 m and  $6.35 \times 10^{-3}$  m, respectively, and locking and radial forces of  $1.11 \times 10^4$  N and  $3.34 \times 10^3$  N, respectively.

a coefficient of friction of 0.1, the radial load capacity could be as high as 750 pounds, or a value equal to a typical bearing load. It is easy to see that, unless a special pressure balanced seal is employed in medium-weight rotor, high pressure machines, the bearings and seals will interact as rotor supports. In heavy rotor, lower pressure machines this interaction is less likely.

Thus, the high pressure can indirectly cause instability in otherwise stable machines by influencing the gas density and, therefore, the fluid dynamic excitation levels, and by locking oil breakdown seals axially such as to cause interaction with the stable rotor tilt-pad bearing system.

Many natural gas reinjection compressors have been analyzed because of instability problems experienced at the oil field sites. The primary problems associated with these machines have been tilt-pad bearings with too high a stiffness and locked, oil buffered seals.

The analytical results for a representative machine are discussed in the following paragraphs.

This particular compressor was operating above 13,000 rpm with a damped natural frequency of approximately 5500 cpm. The base rotor-bearing log decrement was calculated to be 0.29. With a representative aerodynamic cross-coupling stiffness, the base log decrement was reduced from 0.29 to 0.11. However, with locked oil buffered seals, the base log decrement was reduced from 0.29 to -0.01. A softer, low geometric preload bearing provided more damping and increased the basic log decrement from 0.29 to 0.46. With the new bearing design and typical aerodynamic and oil seal destabilizing forces acting together, the base log decrement was reduced from 0.46 to 0.08 (still greater than zero). With circumferential grooved oil seals, the rotor-bearing, aero, seal log decrement was increased to 0.25.

Success was achieved at the site by incorporating the new bearing and seal designs.

Other recurring practical problems are worth mentioning because they have caused considerable loss of production and much grief between the analyst and the operating personnel.

In the bearing area, improper clearances, too small an axial length pad, and insufficient oil flow or improper oil flow distribution have caused problems.

In the damper bearing area, damper bottoming out, or hang-up, and too low an oil supply pressure have prevented effective damper performance.

Shafts which are too small in diameter or have improper wheel fits (too loose, too tight, or too long) are poor in mechanical design integrity and can be the main causes of high vibrations.

Finally, oil-buffered seals should be designed to act as sealing elements only and should not be part of the dynamic system of rotor and bearings. Thus, a pressure balanced seal should be a design requirement, and means for causing seal lock-up should be eliminated from the compressor design.

## RECOMMENDATIONS

1. Prepare a complete and accurate mathematical model of the dynamic system and then perform a dynamic analysis as suggested in Reference 1.
2. Police the design, manufacture, and assembly procedures such that the mathematical model agrees with the actual machine and vice versa. See discussion for details.
3. Design for a base rotor-bearing log decrement of 0.5 if possible as suggested in Reference 2. This appears to be representative of practical, stable systems.
4. Higher molecular weight gases require higher base log decrements. See discussion herein and design accordingly.
5. Beware of high pressure compressors because they can have inherent destabilizing effects from not only fluid dynamic forces at the impellers and in the labyrinth seal areas (including the balance piston), but from locked, high pressure oil breakdown seals.
6. Tilt-pad bearings with looser clearances (lighter geometric preloads) in conjunction with stiffer shafts usually yield higher system log decrements. Design with these facts in mind.

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