# Air-Hammer Instability of Externally Pressurized Compressible-Fluid Bearings

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## ABSTRACT

The pneumatic stability of an externally pressurized gas bearing was investigated using perturbation testing. Bode plots were used to determine the onset of instability, and full-spectrum cascade plots were used to determine the air hammer vibration frequency. The data demonstrate that air hammer is a nonsynchronous excitation of the system resonant frequency and derives its driving forces and energy from the compressible fluid supply. A discussion of air hammer is given.

**Keywords**: Air Hammer, Gas Bearings, Pneumatic Instability.

## INTRODUCTION

Widespread use of externally pressurized compressible-fluid bearings could have an enormous impact on industries that rely on rotating machinery. Some potential advantages are the elimination of oil from machines (with the accompanying elimination of filters, separators, heat exchangers, and some seals), and the virtual elimination of bearing mechanical losses and bearing heating. To realize these benefits, the issue of stability must be successfully dealt with. Gas bearings are susceptible to two types of instability that are clearly distinguishable by their underlying physical nature. The first is the well-known fluid-induced instability called oil whirl. The second is a pneumatic instability commonly known as air hammer.

Air hammer has been variously described in the literature as a self-excited pressure variation in the gas film [1], an instability indistinguishable from oil whirl [2], as an annoying “howling” that inexplicably comes and goes when the researchers did various things [3], and as a “troublesome phenomenon” the researchers could not identify [4]. Opinion on the cause of air hammer ranges from a “hysteresis effect” due to traversing the laminar/turbulent flow regime boundary [3] to “discrepancies between the discharge and supply of gas” [5]. It is the authors’ purpose here to clarify experimentally the nature of air hammer as it relates to gas bearing applications for rotating machinery.

## A DESCRIPTION OF AIR HAMMER

Although some researchers have declined to distinguish between air hammer and oil whirl, the two are distinct phenomena that have separate causes and must be managed separately. A comparison between the two phenomena will serve to illustrate the nature of air hammer.

Oil whirl is a nonsynchronous rotor precession. The term “nonsynchronous” is used in reference to vibrations at a frequency other than the rotor rotative speed. The forces driving the motion are fluid forces within the bearing, and the energy needed to sustain whirl derives from the rotation of the rotor. The oil whirl frequency is proportional to the rotative speed, usually tracking at about 48% of the running speed.

If a rotor-bearing system resonance exists near the whirl frequency, the destructive phenomenon known as whip can occur. In the case of whip, the precessional frequency excites the rotor resonant frequency, often a bending mode with high midspan amplitudes. The potential for whip effectively limits the speed of many fluid-film supported rotors to under about twice the speed of the first bending resonance.

Air hammer is likewise a nonsynchronous precession with many similarities to whirl and whip, but also with some important differences. Like whirl, air hammer is typically observed as a nonsynchronous precession. Like whirl, the forces driving the vibration act on a rotor at the bearing. The energy, however, does not derive from rotation, but from the fluid delivery system that supplies the externally pressurized bearing. This is underscored in the experiments to be described by the observation of a rotor undergoing a 7000-cpm precession with no rotation at all.

Whip can be described as the nonsynchronous excitation of a rotor-bearing system natural frequency. Similarly, air hammer is also the nonsynchronous excitation of a system natural frequency. The essential difference is the mechanism for exciting the resonance, and that the resonance can be excited whether the running speed is above resonance, below resonance, or zero.

## CREATING AIR HAMMER IN BEARINGS

Air hammer is dominated by the specific frequency of a rotor-bearing system resonance. Therefore, rotor dynamics must be considered an important factor in creating air hammer. Rotor mass, mass distribution, rotor stiffness, bearing stiffness, and support stiffness all affect pneumatic stability by determining the system resonant frequencies and mode shapes.

The precessional motion is created by the gas film and is self-sustaining. This implies that significant motion must be present in the bearing for air hammer to occur. For that reason, only modes that do not have a nodal point in or near the bearing can exhibit the instability.

The energy required to sustain air hammer vibration comes from the pressurized gas supply. A volume of gas can do sustained work on a system only when cyclical displacement and pressure are slightly out of phase, as in reciprocating heat engines. However, since there are no timing valves controlling the gas film in a bearing, work is not usually done except at particular frequencies of boundary motion. This leads back to the first condition mentioned, i.e. the rotor dynamic resonant frequencies. Air hammer does not occur in every gas bearing -- only in those for which a system resonant frequency is coincident with extracting work from the pressurized gas film, and for which there is sufficient energy in the supply to drive the resonance.

Air hammer can be brought on or assisted by rotation or external perturbation. An air bearing system that is stable at rest can become unstable as the rotative speed increases. Assisted air hammer occurs because of the combined motion of the rotor's synchronous response and the rotor natural frequency. Assisted air hammer excites a system resonance, which occurs in conjunction with the synchronous response. As the rotor reaches the threshold speed for assisted air hammer, the amplitude of the synchronous response changes very little while the overall amplitude sharply increases as the resonant frequency component becomes excited.

## THE AIR HAMMER MECHANISM

Consider two parallel plates in relative harmonic motion. When the volume between the plates decreases, an incompressible fluid filling the volume is displaced unconditionally. Fluid is forced out from between the plates at a flow rate proportional to the relative velocity of the plates. Resistance to this displacement flow, which to an approximation is proportional to the flow rate, creates a reaction force on the plates. Excepting cavitation, the same occurs when the plates move apart. Aside from inertial reaction, the force required to move the plates when a fluid is present is roughly proportional to velocity and in phase with velocity. It is, therefore, identified as damping.

If the fluid is compressible, a fluid displacement does not necessarily accompany every change in film volume. With very slow plate motion, a compressible fluid is still displaced and damping occurs. However, for extremely fast motions a compressible fluid may not be displaced at all. If that is the case, no damping occurs. What occurs instead is a reaction force proportional to the displacement due to the compression of the gas. In other words, the film behaves as a spring element rather than a damper. The frequency dependence of stiffness and damping on boundary oscillation frequency has been established by analysis and simulations [6-7]. It is noted that when a system has little or no damping, resonances are easily excited.

An externally pressurized bearing is created by more or less regularly replenishing the gas film with gas from an external supply. If the feeding is approximately a constant mass flow rate, the average pressure of the film during expansion will always be a little larger than the average compression pressure. In other words, thermodynamic work is being done on the plates using the energy supplied by the external gas source. The amount of work is strongly frequency-dependent. If the frequency of a resonance extracts enough energy from the pressure-fed gas film to sustain that frequency, the resonance will be sustained as an air hammer instability.

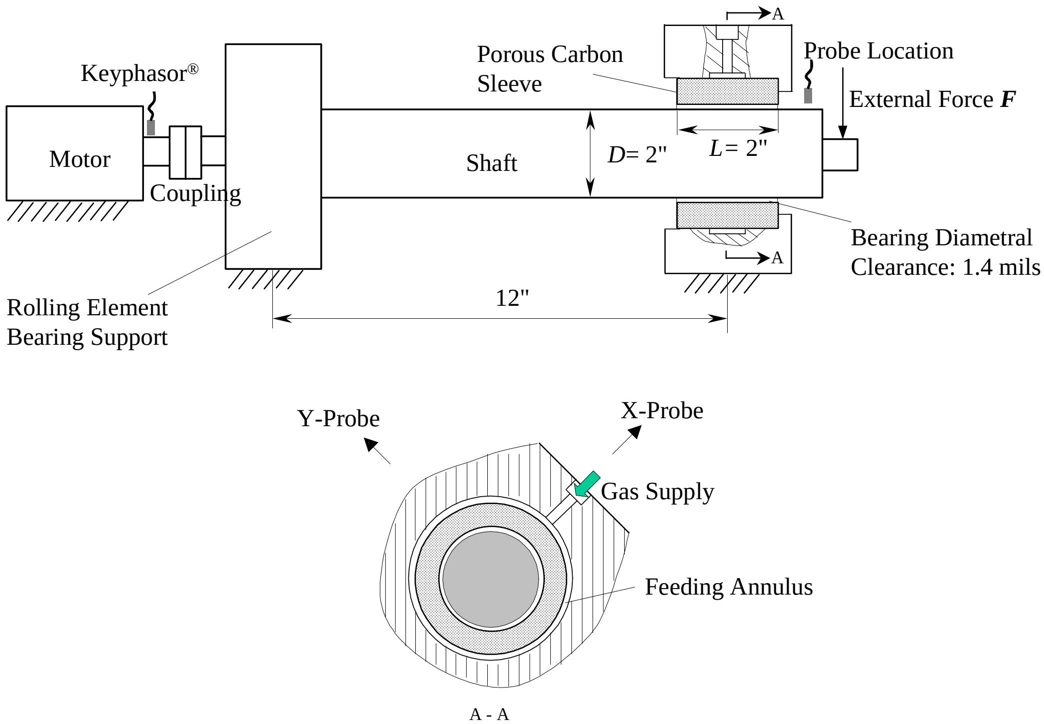


**Figure 1. Gas bearing test stand.**

## EXPERIMENTAL RESULTS

A test stand was constructed to allow determination of air bearing stiffness and damping characteristics. The rotor was designed to operate in a rigid-body conical mode, with the test bearing located at the anti-node.

The rotor had a total length of 460 mm (18 in.) and a diameter of 50.8 mm (2 in.). The drive end was supported in a precision rolling-element bearing, while the non-drive end was supported in the test bearing. The midplanes of the bearings were 305 mm (12 in.) apart. A balance wheel was attached to the outboard non-drive end of the rotor, to which calibrated unbalance masses could be attached. Eddy-current displacement transducers were mounted in X-Y configuration on the inboard and outboard ends of the test bearing. The test bearing consisted of a porous carbon-graphite cylindrical sleeve. Two grades of porous carbon were used, with 10% and 20% available porosity.

Air hammer was detected in experiments through a sharp increase in overall vibration amplitude. The frequency spectrum confirmed that the additional vibration did not track the running speed but was constant. The results of testing the 10%-porosity bearing at 100 psig are shown in the following figures.

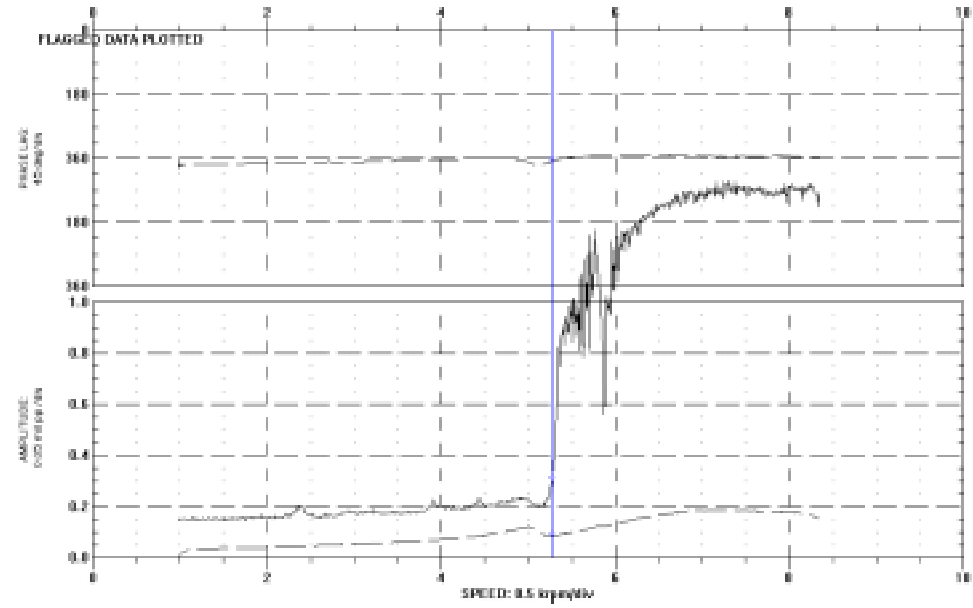


**Figure 2. Bode plot depicting typical air hammer onset.**

The Bode plot in Figure 2 shows a sharp increase in vibration levels as the rotor speed passes 5280 rpm. The vibration is visible in the direct, unfiltered data, but is absent in the 1X-filtered data (data filtered to the running speed) indicating that the air hammer vibration has a different frequency component. The cascade spectrum plot in Figure 3 confirms this and shows that the vibration is a mainly forward precession at 11.5 kcpm.

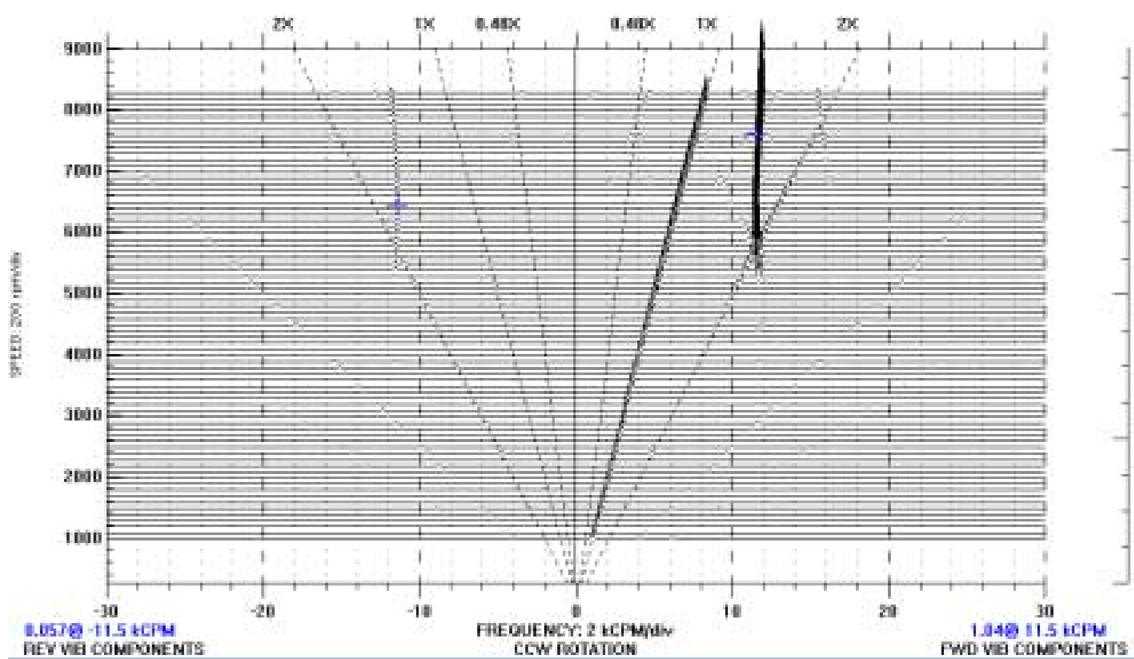


**Figure 3. Cascade plot showing changes in frequency spectrum as running speed increases.**

The test was repeated at various supply pressures. The bearing stiffness is directly dependent on supply pressure, so the effect of increasing pressure is primarily that of increasing the resonant frequency of the rigid-body conical mode. Table 1 summarizes these tests. Onset speed refers to the rotor speed at which air hammer was first observed as the rotor speed increased. Air hammer frequency refers to the frequency at which the instability presented itself on the cascade plot.

**Table 1. Air hammer observations on 10%-porosity bearing.**

| Supply Pressure | Onset Speed | Air Hammer Frequency |
|---|---|---|
| 80 psi | > 10,000 rpm | not observed |
| 85 psi | 7770 rpm | 10,400 cpm |
| 90 psi | 6960 rpm | 10,900 cpm |
| 95 psi | 6300 rpm | 11,200 cpm |
| 100 psi | 5280 rpm | 11,600 cpm |

It is noted that the air hammer frequency corresponded exactly to the conical rigid body resonance of the rotor based on rotor mass and observed bearing stiffness. As the resonant frequency was increased through higher supply pressure, air hammer was seen to occur at lower rotor speeds. One hypothesizes that air hammer would become spontaneous if the stiffness could be increased without practical limit.

In tests using the 20%-porosity porous carbon bearing, spontaneous air hammer was observed (see Table 2). The higher porosity effectively provided a greater trapped volume of gas within the bearing clearance, and hence greater opportunity for the fluid to be compressed rather than displaced. Thus, air hammer was far more readily observable. As stiffness was increased through increasing the supply pressure, the air hammer onset speed became generally lower until it occurred in the absence of any rotor rotation. The unexpectedly high onset speed at 19 psi was repeatable, but the reason for it is unknown.

**Table 2. Air hammer observations on 20%-porosity bearing.**

| Supply Pressure | Onset Speed | Air Hammer Frequency |
|---|---|---|
| 18 psi | 3600 rpm | 6120 cpm |
| 19 psi | 4030 rpm | 6510 cpm |
| 20 psi | 2100 rpm | 6900 cpm |
| 21 psi | 300 rpm | 6990 cpm |
| 22 psi | 0 rpm (spontaneous) | 7080 cpm |

## CONCLUSION

Based on these experimental observations, it is reasonable to project that for a given geometry (recess volume, exit restriction, etc.) of an externally pressurized air bearing, there is a frequency at which vibration becomes self-sustaining through the action of the pressurized fluid film. If the rotor-bearing natural frequency exceeds this self-sustaining frequency, or if the resonance combined with synchronous precession creates sufficient film boundary motion, and if the fluid supply can deliver enough energy to the rotor to sustain the resonance, then air hammer instability results.

The known traditional remedies for air hammer include adding mass to the rotor, soft mounting the bearing, reducing the supply pressure, and eliminating bearing recesses [8]. The experiments presented here explain why these remedies work. Adding mass, soft-mounting the bearings and reducing pressure all effectively reduce the system resonant frequency. At lower frequencies, the gas film is more likely to produce damping and less likely to exhibit spring behavior. Reducing the size of bearing recesses provides less opportunity for gas to be compressed and more chance to be displaced during boundary motion, as well as limiting the amount of energy available for transfer to the rotor.

With additional data on the effect of bearing geometry and recess size, it will be possible to propose a comprehensive air hammer stability criterion and assess the stability margin of new applications in the design stage. The ability to document stability margin against air hammer will permit many new, critical applications of externally pressurized compressible fluid bearings.

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