

FORWARD SUBROTATIVE SPEED RESONANCE ACTION OF ROTATING MACHINERY

by

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His interest in the rotating machinery arises from his interest in measurements of the behavior of these machines. He acts mainly as a collector and correlator of information on rotating machinery behavior, occasionally extrapolating historical information to improve the concepts on machinery malfunction mechanisms.

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ABSTRACT

A wide variety of mechanisms can create net effective forces on those rotor systems which follow the rotor around in its rotation direction. While whatever damping the rotor system has always exists, it is frequently the nature of these forward acting forces to act effectively opposite to the system damping. Once any one or a combination of these forces matches or exceeds the existing rotor damping the rotor system responds by excitation of resonances which lie below rotative speed.

To show this general category of machinery malfunction mechanisms, first the oil whirl and oil whip behaviors are shown in graphical non-mathematical forms. To show that its relationship to oil whip is more than symptom-deep, the next presentation is the internal friction malfunction. Following this, a series of other known energy transformers are shown to exhibit the behavior and relationship of this class of rotor malfunction mechanisms.

The relationships between the forward acting mechanisms, the non-directional mechanisms, and reverse mechanisms are discussed, and demonstrations of some of the forward mechanisms are performed.

Recent classical cases of some of the destructive mechanisms are presented, and the general category of malfunction is summarized.

INTRODUCTION

This presentation concerns the forward whirl tendency of rotating machinery, and studies the various energy transformer mechanisms that cause this action.

W. J. Macquorn Rankine¹ had the correct general idea in 1869 when he stated, "For a shaft of a given length, diameter, and material, there is a limit of speed, and for a shaft of a given diameter and material, turning at a given speed, there is a limit of length, below which centrifugal whirling is impossible."

"Impossible" is too strong a word for the statement, because if the unbalance is severe enough, "centrifugal whirling" may occur at very low speeds. Also, various other "whirls" may occur at low speeds. Rankine was, of course, referring to the first occurring balance resonance speed, which sometime after 1869 was misnamed "first critical."

In the more than a century that has passed since Rankine's contribution, our knowledge of machinery behavior (especially on the self-balancing action of high speed rotors as they are increased in speed through their balance resonance rpm regions) has markedly improved.

For instance, Bert Newkirk,² the brilliant astrophysicist, deciphered the oil whirl energy transformer fifty-one years ago. In the same research laboratory, Ed Kimball,³ a mechanical engineer, deciphered the paradoxical internal friction energy transformer. Unfortunately, these two malfunction mechanisms have always been treated as isolated topics. This paper will show that the findings of these observers have more in common than only their date and place of discovery.

It is the purpose herein to present the entire subject of subrotative resonance speed action of rotating machinery as a general category of machine malfunction action. Old and well-known mechanisms are presented here for foundation material. New ones are demonstrated and, finally, an integrated concept of the causes and effects of these energy transformers is presented.

TEXT

In order to obtain excitation of a resonance, it is either necessary to directly or indirectly force the action, or to provide a mechanism which, in effect, destroys the natural dissipative action of the machine. When the action is self-generated; that is, not externally forced on the machine, it is difficult to sort out the causative mechanism. For years this has caused difficulty in understanding what is happening to rotating machinery and, therefore, difficulty in controlling these destructive mechanisms on the machines. It is first necessary to look into the oil whirl and oil whip mechanisms, even though the pure oil whirl mechanism is not dependent on, or related to, a system resonance.

Oil Whirl and Oil Whip

Two of the most widely known energy transformers on rotating machinery are the oil whirl and oil whip phenomena, which as malfunction mechanisms, have a common cause. This energy transformer mechanism may express itself in any non-compressible fluid bearing or quasi-bearing wherein the shaft surface drags the lubricating fluid around in the direction of rotation.

It is necessary to maintain oil film for prevention of direct contact between the rotating surface and non-rotating surface by this oil drag mechanism, and to assist with removing dissipated energy from the bearing area. However, it is exactly this fluid drag mechanism which can act as the energy transformer which may, under certain conditions, convert useful torque energy into energy devoted to creating radial dynamic orbiting, forward whirl or whip of the rotor system.

Remedies such as unidirectional steady state preloads (created by devices such as oil pressure dams), have been known since 1924 when Bert Newkirk first described the oil whirl mechanism. Other fixes, such as lobe bearings and tilting pad bearings, are also well known. Unidirectional preload, such as the gravity load of the rotors of horizontal machines, is also well known. Unfortunately, despite the hundreds of publications that have appeared on oil whirl and oil whip since 1924, clear functional descriptions of what actually happens are lacking, and the papers which have carried this vital information have been virtually forgotten. The best description appears in a paper by F. F. Ehrlich⁴ written three years ago, which will be referred to repeatedly herein.

The first portion of this presentation attempts to clarify how the oil whirl and oil whip mechanisms operate so that better information will be available concerning how to prevent and control them. An attempt is also made to clarify the entire picture of how a rotating system behaves in both normal and malfunction modes.

Oil whirl is discussed first because it is the simplest mechanism. In situations where the oil whirl mechanism succeeds in exciting a resonance of the rotor system, the action is called oil whip. Both mechanisms are based on the basic fact that the rotating shaft surface in the bearing will drag the lubrication medium around with it, thus setting up a situation wherein useful torque energy is converted into harmful shaft orbital vibration energy.

The whirl action initiates because as soon as the wedge (plus other) forces exceed the viscous damping force, the oil wedge becomes the actual driving force. In other words, not only does the oil wedge overcome the rotor system damping, it becomes the actual forcing function.

The speed at which the action takes place could be determined by many things, singularly or in combination. For example, it could be determined by some dynamic disturbance force either on the rotating or the non-rotating system. It could be determined by the rotor system exhibiting a natural resonance, as in the case of oil whip excitation. However, it is well known that oil whirl may and does occur with none of these factors present. Therefore, the speed of the action must be determined by the energy transforming mechanism itself.

This occurs in the following manner: The velocity of the rotor film at the non-rotating surface of the bearing is zero. The velocity of the oil film at the rotating surface of the shaft

in the bearing is the velocity of that surface. Assuming laminar flow, the oil between must be traveling at the velocity as shown in Figure 1.

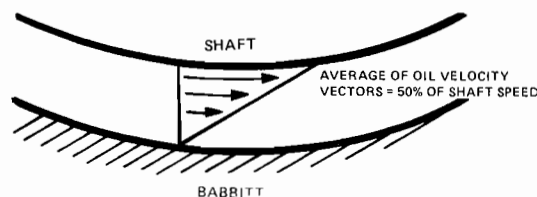


Figure 1. Oil Velocities in the Bearing Clearance.

On the assumption of triangular velocity distribution, early investigators reached the mathematical conclusion that the forcing function traveled at 50% of rotative speed, in the forward direction of rotation, of course. Obviously, the direction of action of this energy transformer is forward and cannot be a reverse (of rotation) action. This forward action is agreed upon by all investigators, as well as in evidence on operating machines.

Statistical evidence, however, overwhelmingly demonstrates that the pure oil whirl mechanism is strongly preferential at between 42% to 44% of rotative speed.

The main reason for the speed rate being lower than the theoretical 50% is obvious when it is recognized that the rotating surface is usually the shaft and the shaft surface is usually smooth. The usual bearing non-rotating surface is, however, relatively rough. This leads to the oil velocity across the clearance area, as typically shown in Figure 2a. The average oil velocity with this concave distribution favoring the non-rotating surface is less than 50%. This is the main mechanism which determines the speed of the forward circumferential travel of the wedge. Since it is the oil wedge that nudges the shaft around, it is also the speed of the forward (in the direction of rotative speed) motion that creates the oil whirl action.

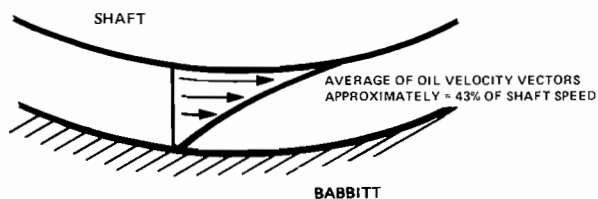


Figure 2a. Oil Velocities in the Clearance Areas (Normal Sleeve Bearing).

A secondary factor that slows the action below the 50% theoretical speed is that the oil will travel not purely around with the shaft; it may travel endwise, coming out of the edges, yielding a net average oil velocity lower than 50% of rotative speed.

It is obvious that if the shaft had a rougher surface than the sleeve bearing, the oil velocities would be as shown in Figure 2b, creating something like 55% of rotative speed action, instead of 43%.

As has been described, the direction of the oil whirl mechanism is forward because the wedge travels in the direction of shaft surface velocity. It is reasonably circular because the wedge must go around the bearing clearance circle in order to repeat itself and thereby survive.

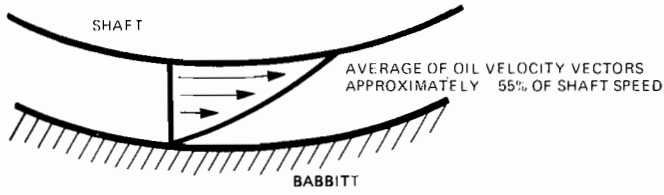


Figure 2b. Oil Velocities in the Clearance Areas (Normal Sleeve Bearing).

Figure 3b shows the singular stability point of oil whirl, where the shaft has zero eccentricity to the bearing*. No wedge can develop; therefore, no whirl is incurred.

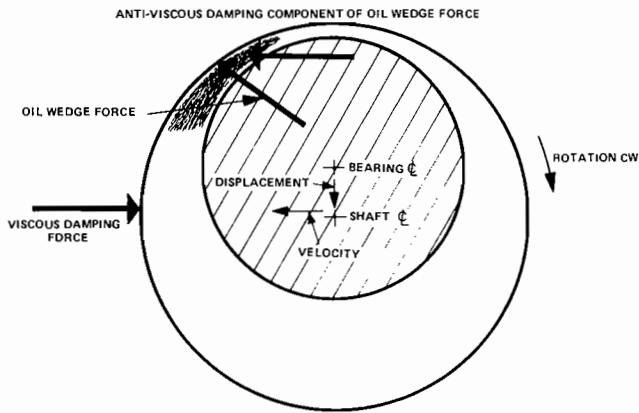


Figure 3a. Inverted Bearing Oil Whirl.

If the bearing were built inside out; that is, with the center stationary and the outside turning, as shown in Figure 3a, the same effects are obtained.

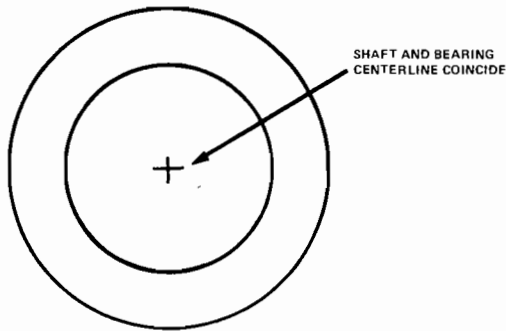


Figure 3b. Stable Singular Point; Oil Whirl

Figures 4a and 4b show the position, velocity, and the forces acting in an oil whirl action. Figures 5a and 5b show the action a few milliseconds and 1/4 of an orbit later.

It may be observed from Figures 4a, 4b and 5a, 5b that as soon as the oil properties, speed, preload, and clearance are appropriate, the oil wedge force component, acting direct-

ly against the viscous friction component, overcomes (but does not eliminate) the viscous damping force.

Once this happens, the rotor proceeds to vibrate at whatever average speed the oil film circulates, usually near 43%, and in a forward (with the direction of shaft rotation) circular orbit.

This mechanism is presented in theoretical form in many fine papers, including an excellent modern presentation by Kirk and Gunter.⁵

It is interesting to note that the oil whirl mechanism depends on two different viscous properties of the fluid, the viscous drag and the viscous "squeeze" coefficients. The drag determines the strength of the oil wedge (the lowest shear coefficient oil being the weakest wedge), and the squeeze determines the viscous damping. Unfortunately, in over fifty years there are no known studies on the oils for the shear viscosity vs. the squeeze viscosity effects on the oil whirl and whip stability. Perhaps there will be within another fifty years.

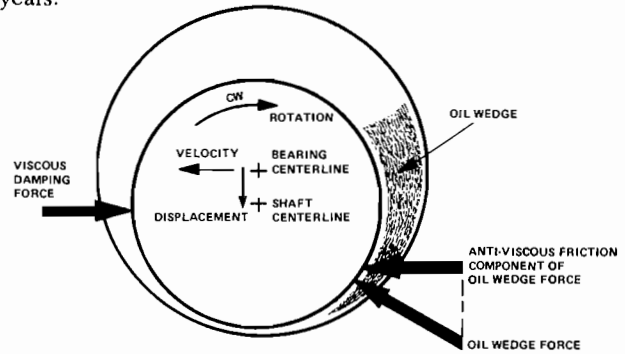


Figure 4a. Oil Whirl; Shaft at 6 o'clock.

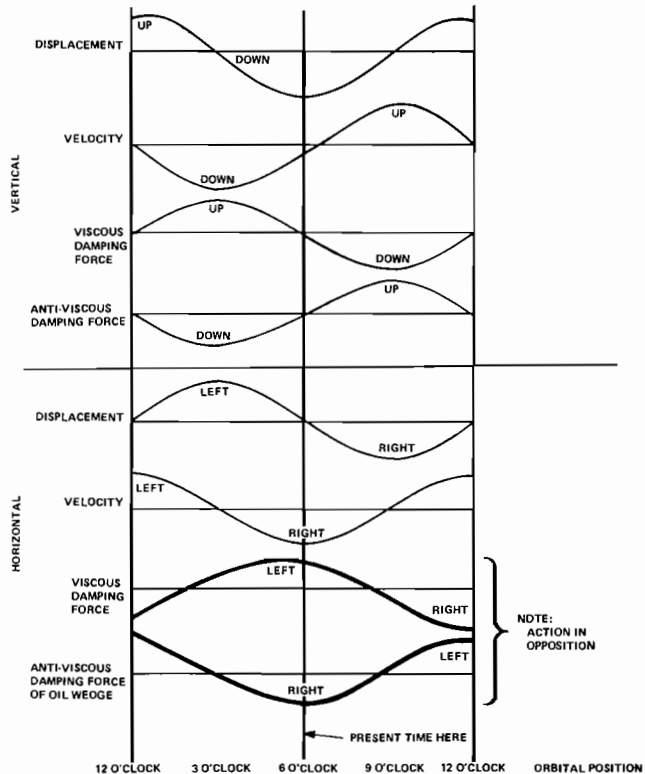


Figure 4b. Oil Whirl; Shaft at 6 o'clock (Horizontal/Vertical).

*See Appendix

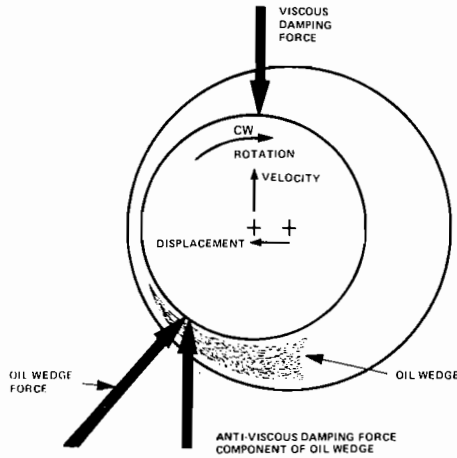


Figure 5a. Oil Whirl; Shaft at 9 o'clock.

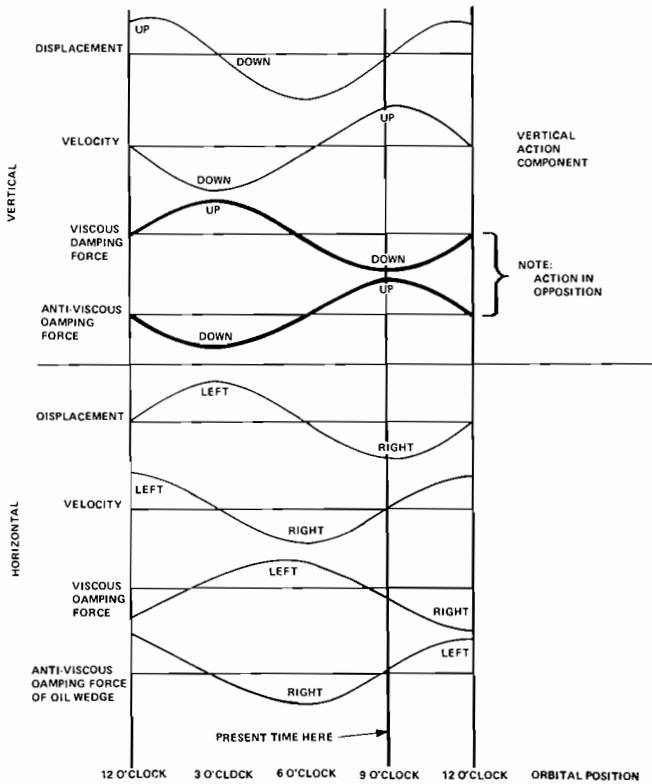


Figure 5b. Oil Whirl; Shaft at 9 o'clock (Horizontal/Vertical).

Oil Whip

Lacking any influence other than average oil velocity, the oil whirl tracks a fairly constant ratio of rotative speed. However, if the rotor system has a resonance below rotative speed, such as a balance resonance, oil whirl and resonance get involved with each other. When this occurs, it is called oil whip. As with oil whirl, there are many fine papers on the subject. One of particular interest was written by Hull⁶ in 1957, who, 31 years earlier, had been a research assistant to Newkirk and Kimball. Hull's major contribution was the introduction of perturbation techniques to rotating machinery. While perturbation is not yet popular in design studies and in acceptance tests on machinery, it surely will be employed if the field is to progress.

Figure 6 shows a typical balance resonance response of a rotating machine. For purposes of simplifying the discussion, the translational balance resonance speed range (bounce) is assigned a center of 4,300 rpm, and the pivotal resonance speed range is at 13,000 rpm. Further, the system has an apparent "Q" (amplification of unbalance at bounce resonance of 4) because the bounce peak is 100 micrometers p/p compared to 25 micrometers after the rotor balances itself translationally.

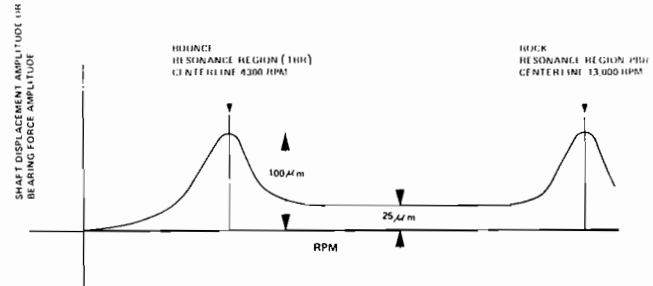


Figure 6. Balance Resonance Response of a Rotating Machine.

It is obvious that in this situation the oil whirl action and the resonance action will aid and abet each other. In fact, the only reasons why the machine is not immediately destroyed are the non-linear characteristics of the various friction and spring restraints, whereby the spring is usually stiffer with large amplitudes (except at times on electric motors and generators, where the spring may weaken), which move the resonance higher in rpm, and the viscous and cross drag damping increase to limit excursion amplitudes sufficiently to avoid machine destruction.

As the machine speed is increased or decreased to move the oil whirl rpm above and below the bounce rpm, the two interact, pulling on each other to arrive at a compromise forward whip orbit frequency. The amplification factor (Q) has gone from 4 to infinity.

With the typical damping of many good running machines, the two interact only over a fairly narrow speed range around the balance resonance.

The better the machine is damped, and the more mismatched the vertical and horizontal spring stiffnesses are, the lower the capability of the whirl to "pull" the resonance frequency.

However, a model machine deliberately built with poor damping, or a machine that has been injured by major rotor displacement action which thoroughly clears all the seals, can allow oil whirl to pull the resonance considerably. The actual rate shift of an oil whip is mainly determined by the oil whirl frequency.

A model rotor system designed with an antifriction bearing at the inboard end and an oil whirl at the outboard end, readily exhibits the ability of the balance resonance region to shift the 43% of rotative whirl mechanism to the wide rpm range of action of oil whip. Figure 7 shows the kit with a bounce resonance at 4,000 rpm, rotative speed at 4,385, and oil whipping at 3,700 rpm. This is 85% of rotative speed whip, with the resonance pulled down by 7.5%.

Figure 8 shows the same machine with the mass increased by 8 times to produce bounce resonance at 1,000 rpm. With the machine's rotative speed at 10,000 rpm, the whirl occurs at 1,500 rpm. In this situation the bounce resonance is effectively pulled to 50% higher and the oil whip is at 15% of rotative speed.

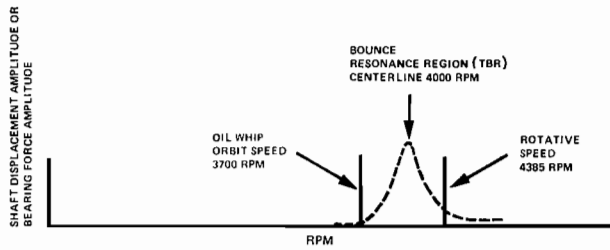


Figure 7. An 85% of Rotative Speed Oil Whip.

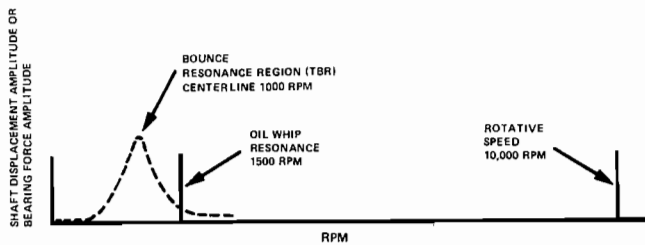


Figure 8. A 15% of Rotative Speed Oil Whip.

This range of 15% to 85% of rotative speed is to be expected on high speed rotating machinery.

While oil whirl and whip are usually energy transformers at a bearing, it should be kept in mind that their conditions may occur at any lubricated shaft restraint, such as at the seal rings of tilting pad bearings, differential lube gas seal bearings, etc.

The Kimball Excitation of Resonance by Internal Friction

This energy transformer occurs when flexure stresses the rotor itself; oil whip occurs at bearing and other shaft restraint surfaces; that is, however, the only major difference.

In addition to date and place of discovery, internal friction and oil whip have these symptoms in common:

1. Both generate essentially forward circular orbits;
2. Both have shaft orbital motion forward, in the direction of shaft rotation;
3. Both are essentially at balance resonance speed below the rotative speed;
4. Both are insensitive to refinements in mass balance, but may be eliminated by rotor unbalance.
5. Both have singular centerline stability, as shown in Figure 3b.
6. Both may be initiated by an initial "bump" to deflect the rotor or any soft (long range) deflection force, such as gravity.

In response to simple additional tests, these two distinctly different energy transformer mechanisms do have different characteristics, as well as a more than symptom-deep common source:

Both have an energy-converting (transforming) mechanism which acts opposite of the "squeegee" damping friction, effectively overcoming it. The rotor system then, as a natural result thereof, vibrates at its effective translational balance resonance rate because there is, insofar as the rotor observes its situation, no damping, except as is incurred by major radial rotor excursion orbits.

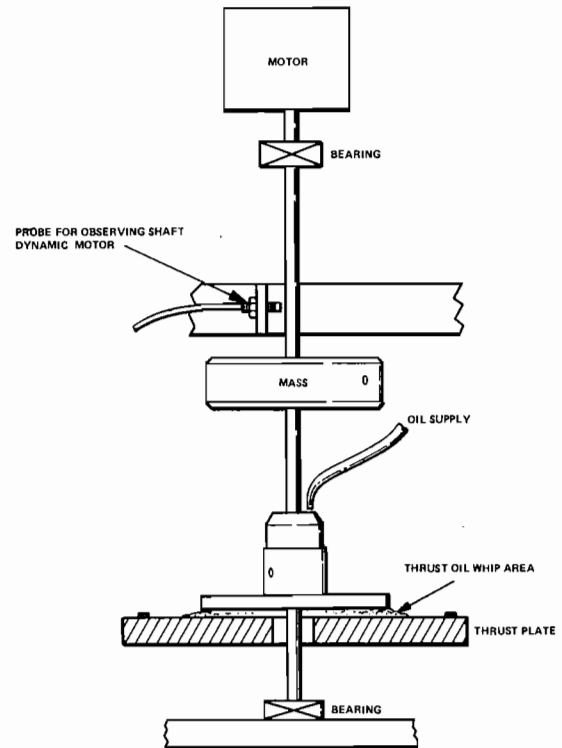


Figure 9. Thrust Oil Whip.

The tests for distinguishing these two malfunctions are sometimes not absolutely conclusive because of their "aiding and abetting" nature, but they are shown in Figure 9.

TEST	OIL WHIP RESPONSE	RESPONSE OF INTERNAL FRICTION AND OTHERS
Steady State Rotor Loading or Unbalance Loading	Stops	No response, or switches to Mathieu-Hill-Meissner-Duffing single hit action
Lube or seal oil* change of temperature or viscosity	May Stop	No change or slight change

Having established the common relationship of two well known forward circular actions at resonant speed, the following section presents another mechanism to consider.

Thrust Oil Whip

Since the idea of oil whip did not appear to be locked to the action of a cylindrical body, this author built a thrust oil whip rotor kit, as shown in Figure 10.

This whips just as well as a sleeve bearing. Furthermore, it exhibits the same mechanism when the thrust whip is placed on the other side of the plate. The viscous damping and the oil wedge vectors are in the thrust direction, converted to radial shaft deflection by moment action but again the action is forward circular excitation of a resonance below rotative speed. The one distinction between thrust oil whip and usual oil whip is that radial relative, oil whirl, is not possible, in thrust because thrust oil whip relies on shaft flexure.

Thrust Gas Whip

By removing the non-compressible fluid in the thrust whip concept and replacing it with gas, it is simple to extrapolate to the situation of the typical radial compressor.

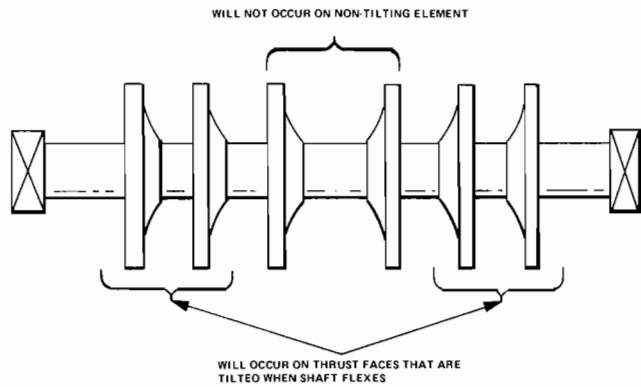


Figure 10. Thrust Gas Whirl.

Figure 11 shows a compressor and the areas where the energy transformer occurs.

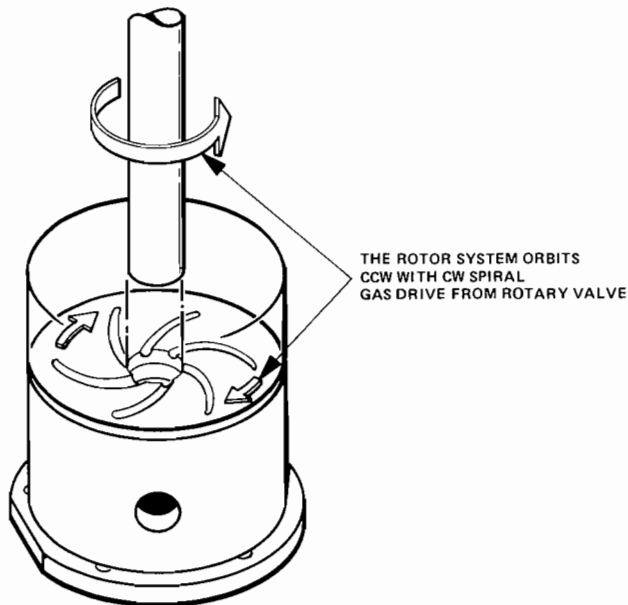


Figure 11. The Leaky Seal As a Rotary Valve Aerodynamic Oscillator.

Since there is a very little tilt at the centermost wheels, there can be very little of the torque energy converted to resonance excitation energy in that area. The action must occur at the outermost wheels where, due to flexure, the tilt is the greatest for the fundamental shaft bow shape of the bounce resonance.

It is probable that this thrust gas whip is a major reason, known or unknown, why some manufacturers refuse to quote more than four impellers on high pressure compressors.

The back-to-back arrangement of wheels, which helps balance the thrust load, could also contribute to the problem by a steady deflection of the rotor if the volutes do not balance gas pressures exactly, as there must be intake or discharge pipes near the center of the machine, where the rotor is much more easily deflectable than near the ends.

With inlet or discharge pipes near the easily deflectable area, any flow or pressure fluctuation can kick back at the rotor much more effectively than at the ends, which cannot help the stability situation.

Steam Whip

Steam whip is described by several authors, most recently by Ehrich, loc. cit. It is simply a radial gas whip which creates the anti-viscous friction force component by increased and decreased pumping efficiency at maximum and minimum clearance areas, respectively. This is shown in Figure 7. Once again the action is forward circular at a resonance below rotative speed.

Radial Gas Whip in Compressors

The argument for steam whip is logically extendable to radial and axial flow compressors, again creating the same action.

Scalloping edges of compressor impellers, for whatever other motive (such as to improve efficiency or to prevent a disc resonance), could be very harmful insofar as creating another contributing factor to the forward circular resonance malfunction group, because of the increased radial pumping which must exist due to the non-smooth edges. This improved pumping may improve efficiency, but it sets the rotor up for a malfunction mechanism very similar to steam whip.

The location of the steam and gas whirled and whips is on the outer periphery of any wheel, as distinct from the thrust gap whip which is confined to the wheels which must tilt when the shaft is deflected.

The other difference between the radial acting steam and gas differential pumping energy transformer and the thrust gas whip is that the radial transformer can generate whirl or whip, whereas the thrust gas whip transformer is confined solely to whip.

Differential Gas Seal Whip Mechanisms

In addition to the obvious direct oil whirl and oil whip possibilities inherent in various differential gas seals, there are several potential energy transformers inherent in various gas seal designs.

One possible contribution is the tendency of the thrust face of the seal to temporarily freeze in an off center position. This simply creates a hard, steady-state, unidirectional preload on the shaft with the resultant doubling of whatever forcing frequency of shaft vibration, which is not the topic under study. This contribution occurs when the seal unfreezes, allowing the shaft to jump, and as is the case with all of the forward circular excitations of resonance below rotative speed, the initial soft deflection may be provided by this jump to get the action initiated.

Another contributor is the radial sleeve bearing as an oil whip contributor. However, the big potential contributor of various differential lubed gas seals to the forward whip action is the timing relationship between the thrust face shimmy action and the radial sleeve bearing action of the seal. Figures 12a, b, and d, illustrate this action, while Figure 12c shows the non-action situation.

As developed here, the action relies on a series of circumstances, as follows: First, the shaft must be flexed so that the long profile seal ring loads its thrust face unevenly, as shown

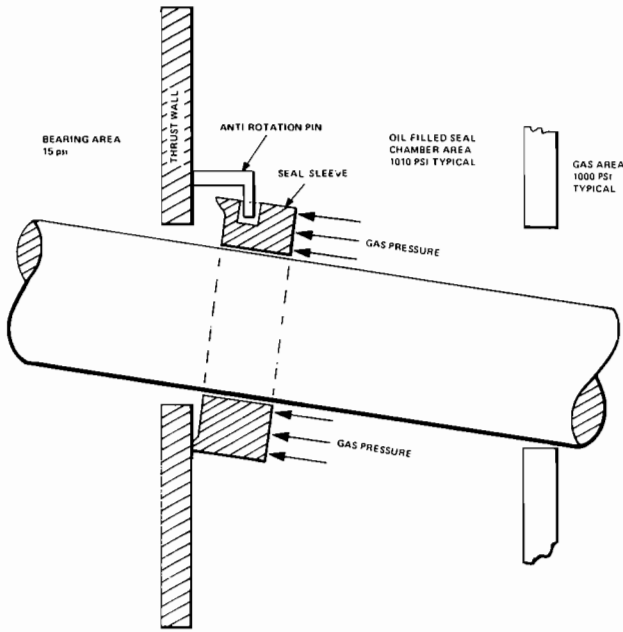


Figure 12a. Differential Lubed Gas Seal — Side View (Shaft Deflected to 6 o'clock).

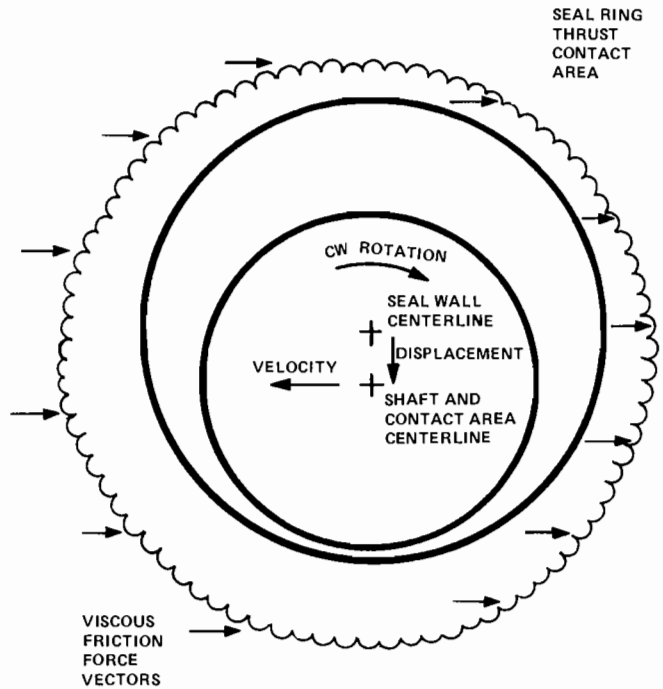


Figure 12c. Differential Lubed Seal — Non Action Situation (Shaft at 6 o'clock but Short Profile Seal with Symmetrical Thrust Loads).

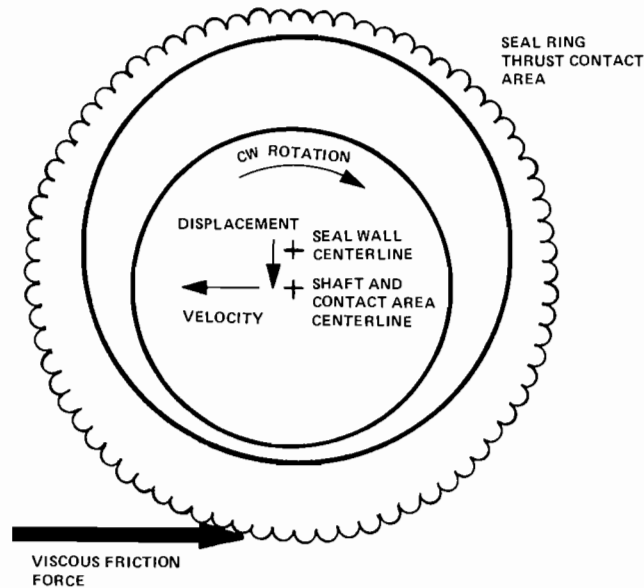


Figure 12b. Differential Lubed Gas Seal — Action Situation (Shaft at 6 o'clock Long Profile Sleeve Loading Thrust Unevenly).

in Figures 12a and 12b. This in turn allows the creation of a moment acting through the seal ring onto the shaft which, in turn, creates the anti-viscous friction vector force, as shown in Figure 12d.

From this sequence of events onward, the action is exactly like the Kimball, loc. cit., internal friction action. The essential difference is that the energy transformers occur at the non-rotating, shimmying seals at each seal bundle instead of at the shrink fits on the rotating shaft.

Further inspection of this mechanism shows that:

1. If the thrust seal bushing is at the zero motion nodal point of shaft flexure action, no energy can be transformed because the shaft centerline is tilting but not orbiting.

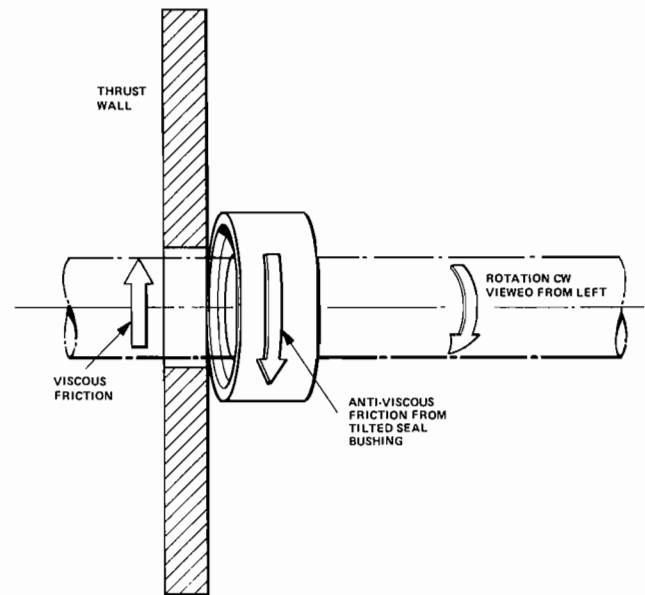


Figure 12d. Differential Lubed Gas Seal — Top View (Long Profile Seal, Shaft at 6 o'clock).

2. If the thrust seal bushing is at the maximum motion nodal point of shaft flexure action, no energy can be transformed, because the shaft centerline has no tilt, even though it has a potentially large orbit.

Murphy takes care of these non-action situations, of course, by placing the gas seals at the intermediate region where the shaft has both the requisite tilt and available orbit.

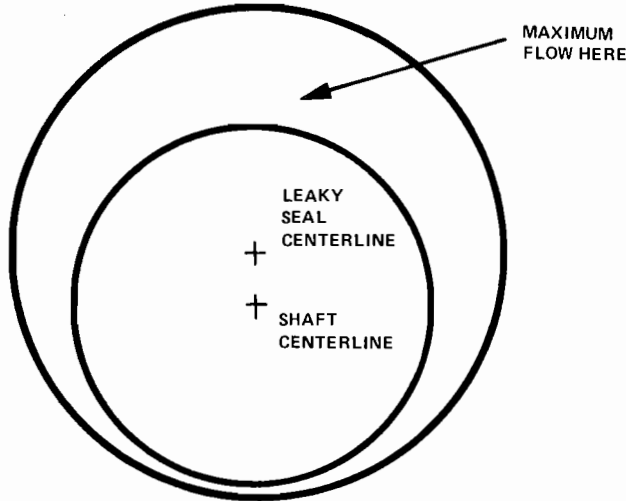


Figure 13. Rotary Valve Shaft (Leaky Seal, Shaft at 6 o'clock).

It must be pointed out that, at the time of this writing, a pure and clear demonstration of this energy transformer has not been accomplished, so that the presented material on differential lubed gas seals must be accepted with caution until clearly exhibiting models can be developed.

Rotary Valve Gas or Liquid Whip

The final energy transformer which drives the rotor in forward circular excitation of resonance below rotative speed presented in this paper was as subtle as a feather in concept, but can be as subtle as an earthquake in its effect on the machine.

It is a rotary valving mechanism whereby the driving energy is supplied from pumped gases or liquid instead of more directly from the shaft torque as do the previously described mechanisms.

This transformer is typically an opened seal, with the deflection of the shaft acting as a rotary valve by the leakage through the open clearance opposite the shaft deflection. A pressure difference on each side of the leaky seal is presumed, or else the seal would not have been required.

The gas or liquid circularly gated by this valve has no adverse effect by itself. What is important is what happens to the gas before or after the circular gate. This rotary valve is shown in Figure 13.

The typical situation in a rotating machine is shown in Figure 14. As shown, the gated flow appears to stabilize the machine. In fact, the first several experiments with this energy transformer indeed did add stability to the system, rather than de-stabilizing it.

The first two working models of rotary valve oscillators were the thrust unit shown in Figure 15, and its matching equal, the radial drive unit shown in Figure 16. As may be readily deduced from inspection, these drive the rotor in circular orbit in opposite directions of the spiralled gas feed.

The means by which this mechanism switches from adding stability on a rotating system to providing the energy to sustain rotor vibrations is a matter of the delay time between the opening of the port and the consequent action of the shaft. By the time the maximum pressure on the thrust face of the rotor

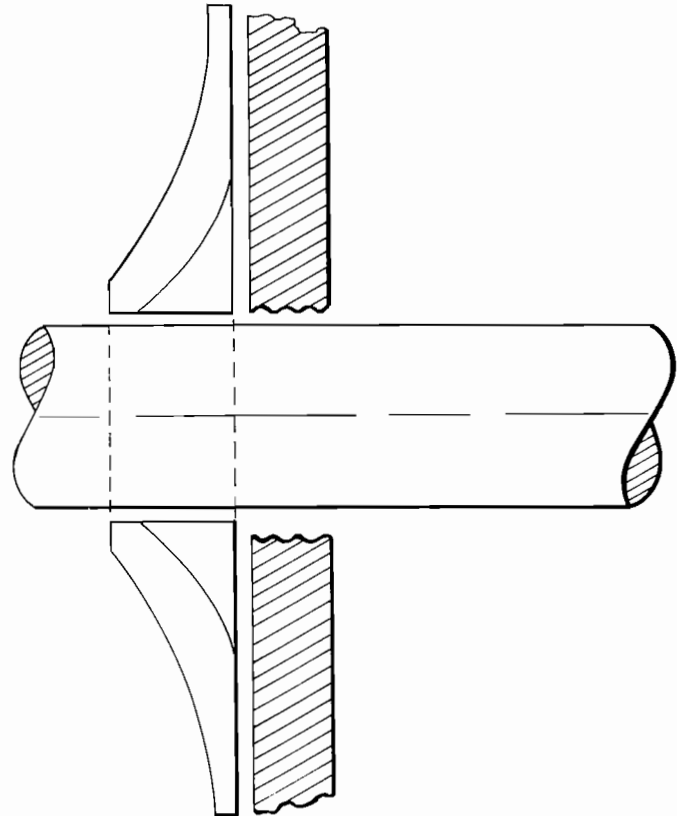


Figure 14. Rotary Valve in Typical Rotating Machines.

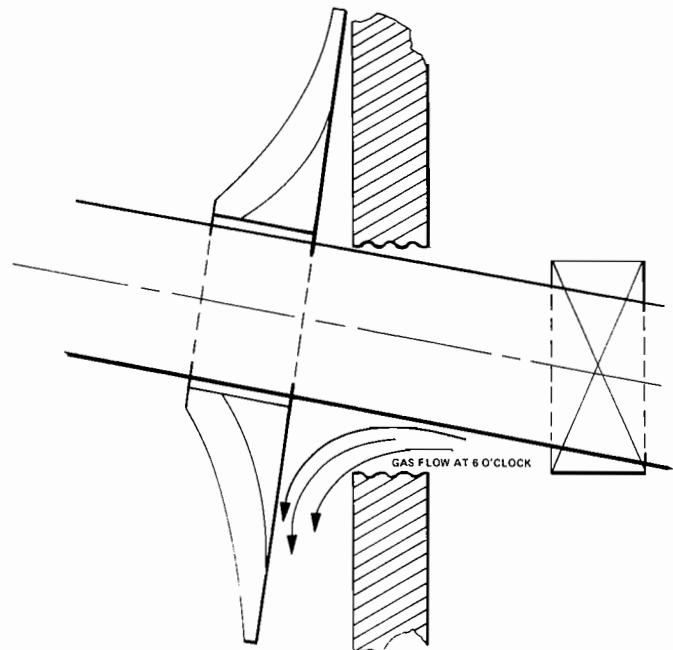


Figure 15. Rotary Valve in Typical Machine. (Shaft at 6 o'clock).

occurs by the opened leaky seal sector, the orbit has moved the narrow axial sector ahead by several degrees. Therefore, the gated gases succeed in chasing the deflection around, once again creating the forward whip.

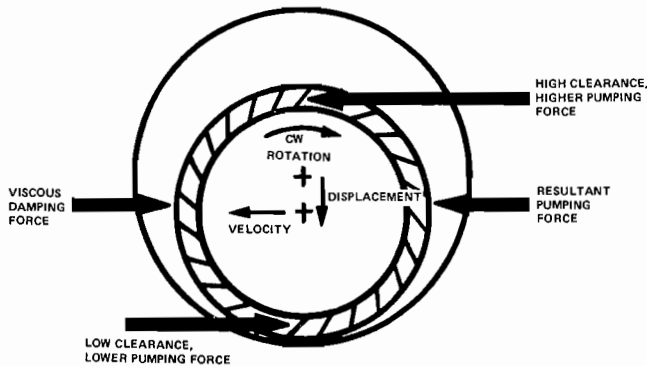


Figure 16. Steam Whirl.

The Unknown "Others"

Having described the actions of both old and new forward whip mechanisms, it is reasonable to extrapolate: "There are probably more of these which have occurred and will occur on rotary machinery." One need not expect to match observations on his machine precisely to one or more of the examples cited in this paper.

CONCLUSION I

All of the energy transformers shown, except oil whirl which was presented for reference in the introduction, can singularly or in consort convert useful torque energy to various degrees of harmful shaft orbital energy. These are:

1. Oil Whip
2. Internal Friction
3. Thrust Oil Whip
4. Thrust Gas Whip
5. Steam Whip
6. Radial Gas Whip
7. Differential Gas Seal Mechanisms Whip
8. Rotating Valve Gas Whip

All exhibit symptoms in common, as first described by Kimball and Newkirk, loc. cit., for the internal friction action:

1. Forward (absolutely),
2. circular (or nearly so), and
3. at or near a natural system resonance below rotative speed.

All of these mechanisms exhibit their initiation and operating systems in common by requiring the following sets of conditions:

1. The rotor system must have limited viscous damping.
2. The rotative speed must be above a natural rotor system resonance speed.‡
3. The shaft must be impulsed or softly deflected in order to initiate the whip action.

‡Numerous papers state that specific ones of these occur only when the rotative speed is between 2 and 2.5 or 2.7 times the "critical" speed. This statement must be discounted as shown by the operational facts and recently by papers, especially a book by Ales Tondl.⁷

All have in common a mechanism whereby a force is generated with a vector direction and strength to overcome the ever-present viscous friction vector force.

The Mathieu-Hill-Meissner-Duffing⁸ (MHMD) action on machinery is not considered in the forward whip category because:

1. It is usually not circular, and is never forward.
2. Except rarely, it is the result of another forcing function.

The Mathieu-Hill-Meissner-Duffing action, however, has in common these features:

1. It does excite the system resonance, even though it may drastically alter its rate, and
2. It occurs predominantly on poorly damped machines.

There is yet another kind of "common" feature between the MHMD and the forward whip mechanisms. You will often observe a forward whip mechanism locking onto precise ratios of rotative speed, usually at 50%, but occasionally also at 4/9, 2/3, 1/3, etc., when their action is close to exact speed ratios. This is MHMD action occurring, but it is also simply the result of the forcing functions. It is not likely to be the prime causative energy transformer when forward circular whip is being encountered.

CONCLUSION II

To check the rotating machinery for this popular category of malfunction, it is not only necessary to run at load and pressure; it is also a requisite that the machine be running with:

1. Seals opened to maximum expected operable clearance*;
2. The machine being brought to operating speed range, observing amplitude and phase of rotative speed to establish its initial resonances, and the "Q" amplification factor thereof; then,
3. Utilizing a Hull perturbation generator** on the machine while at all available operating speed and load conditions, including the opened seals, running the actual "as it operates" amplitude and phase vs. RPM of the perturbation. This will yield the true running amplification factor, not shown by (2) above.

CONCLUSION III

At this time and in the past few years, there are many major malfunctions which are directly attributable to this category of energy transformer.

It is reasonable to expect that, as machines are being pushed to design limits for efficiency, horsepower, speed, or whatever performance or cost parameter, unless prompt action is taken by all people concerned with rotating machinery, steps must be taken to reduce the severity of the problems in this category.

*Seals that are tight may well act as sufficient temporary support to the rotor to prevent the deflection. If the deflection is prevented, all the forward whip mechanisms are prevented. It must be presumed, therefore, that, by whatever means, the operating machine will have opened its seals. Any other assumption will lead to a non-indicative test for either actual forward whip, or sensitivity to forward whip.

**See Appendix.

It is obvious that if the original rotor system viscous damping is increased, as noted in Conclusion I, more stability will be obtained. That is, there is less tendency for a well damped machine to get into this mischief.

Successful field fixes have been achieved recently by the addition of the squeeze film damper bearing. In addition to this stabilizing fix, many machines have been brought on-line or back on-line by various direct or shotgun fixes which advertently or inadvertently reduced or eliminated one or more of the harmful energy transformers. Typical of these are:

1. Reducing the profile of the seal bushing, i. e., low profile.
2. Grooving the seal bushing to prevent oil whip.
3. Increasing shrink fits to get away from the Kimball internal friction.
4. Increasing shaft diameter to reduce the available flexure, the intent being to raise the "critical" to the old magical belief in a ratio of "2 to 2.5 rotative over 'critical' " speed.
5. Shifting the thrust balance piston pressures, which might have the effect of reducing the thrust gas whirl.
6. Switching from sleeve to tilting pad to escape oil whip bearings.
7. Resurfacing (to reduce its stickiness) the seal chamber thrust wall and replacing the seal elements.
8. Removing a wheel from the compressor, which might clear the problem at the expense of loss of head.
9. Providing more anisotropic bearings so that the net stiffness in two radial planes, typically vertical and horizontal, of the rotor system is sufficiently different to split the resonance action apart. This can strongly inhibit all forward circular whip action because these actions rely on finding a circularly symmetrical restraint. This is described by Gunter⁹ for the Kimball, loc. cit., internal friction mechanism, but is fully applicable to this entire category of malfunction action. Many machine designs have such bearing often with wider clearances horizontally than vertically. Many machines are also designed with steady state unidirectional preloading, such as gravity or gear forces to make the bearing appear to be mismatched, which splits the resonances by way of the inherent compliance nonlinearities of bearings.

All of these fixes are fine, especially if done from actual knowledge or pre-established good guesses, and not blind stabs for a quick fix.

One other possible fix, at this time accomplished on the author's experimental kits only, is to employ the circular valve, or its various equivalents, electrical or hydraulic, as a perturbation mechanism deliberately reversed to provide a force vector action to assist viscous friction.

Finally, there is one other "fix" available. That is to limit the speed of rotating machinery to below any balance resonance region. To do this, however, would be to reverse over 100 years of progress since Rankine. It is this author's opinion that machines can be designed, installed and operated efficiently and successfully above both translational and pivotal self-balancing resonance speeds.

APPENDIX

Hull Perturbation Methods

Eighteen years ago E. H. Hull, loc. cit., showed a rotor perturbation mechanism. He employed a set of vertical and

horizontal springs driven by an eccentric. The springs attached to the rotor by way of a floating bearing. See Figure 1 of Appendix.

By this means he applied sinusoidal input forces along the two radial planes. Timing the eccentric action by 90 degrees, he obtained a forcing function on the rotor system which was totally independent of rotating speed and either:

1. Forward circular,
2. reverse circular,
3. one plane only, or
4. any forward or reverse ellipse.

Hull employed both forward and reverse circular perturbation for his study, both below and above absolute shaft speeds.

To repeat Hull's experiment, this author employed a mass unbalance mounted on a low mass aluminum disc, which was in turn attached to an anti-friction bearing. This free spinner is driven as a tiny turbine by air nozzles; the free spinner is shown in Figure 2 of this Appendix.

This free spinner produces excellent results on single rotor kits. Such studies as:

1. Forward and reverse resonance speed,
2. actual damping of resonances located below running speed,
3. location of balance resonances and other resonances above running speed, and
4. demonstrations of the characteristics of the "resultive forced" rotor system category of malfunctions,

could yield extremely valuable information.

While this free spinner is marvelous for experimental studies, it is not necessarily an appropriate means to provide the forcing excitation on a machine on the test stand.

For actual acceptance tests, other devices may be employed, as is appropriate to the situation.

One device which can be attached to a machine under test is the caged free spinning unbalance. This is a version of the free spinner shown in Appendix Figure 2, and is sketched in Appendix Figure 3.

Since it may be much easier to attach this cage to a large rotating machine for acceptance or field analytic tests, it is likely to see wider use than the free spinner.

Another means to perturb the shaft for full acceptance tests is to substitute electro-magnetic pull for Hull's two radial axis springs.

It should be observed that these devices should stay off of the zero motion nodal points for the shaft mode to be perturbed. Should it be desirable to perturb about a zero motion nodal point, a torque moment should be applied in place of the force from the single free mass.

The stable point of oil whirl shown in Figure 3 is noted by Mel Giberson in his discussion on the paper, "Stability Contours for a Whirling Finite Journal Bearing," by Akers, Michaelson, and Cameron.

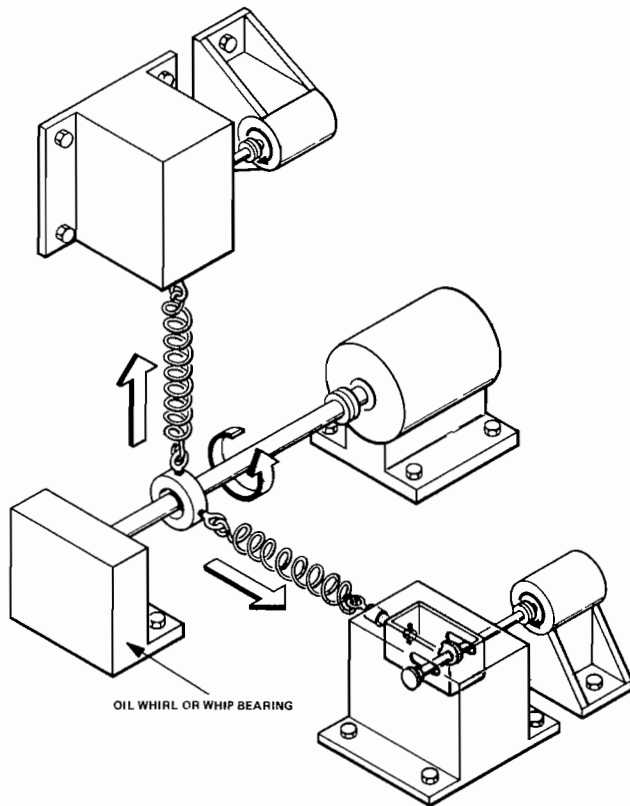


Figure 1. Hull Perturbation.

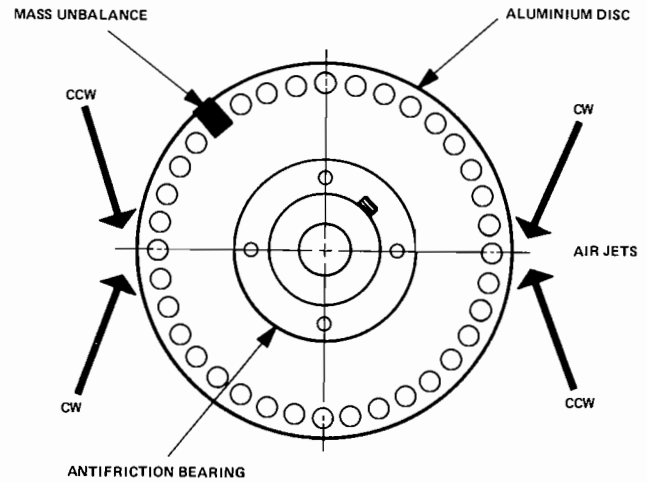


Figure 2. Free Spinner Perturbator.

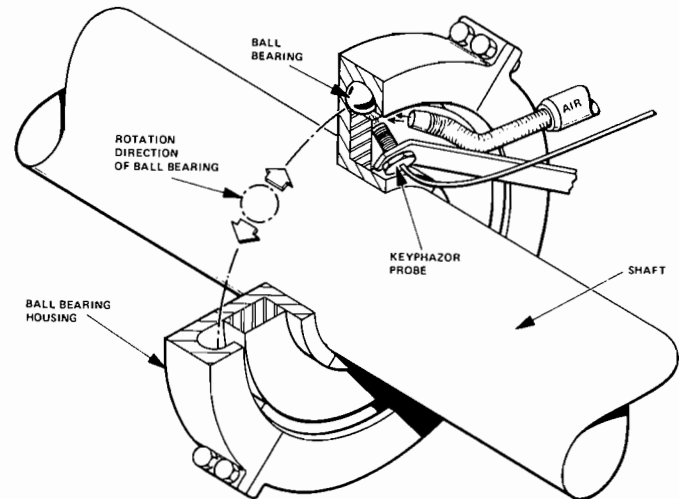


Figure 3. Caged Free Spinner.

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