THE PARAMETERS AND MEASUREMENTS OF THE DESTABILIZING ACTIONS OF ROTATING MACHINES, AND THE ASSUMPTIONS OF THE 1950'S

Donald E. Bently
Bently Nevada Corporation
Minden, Nevada 89423

SUMMARY

Employing a rotor built to produce one of the forward circular self-excited category malfunctions known as "gas whip", "steam whip", or "aerodynamic whip", it is possible to show the results of deliberate perturbation while the rotor is still in stable operation. This test indicates that the destabilizing actions are not mystical or unmeasurable and that the mathematical modeling done today can be more realistic than the models assumed in the 1950s and which still exist essentially unchanged more than 25 years later. The continued use of the original modeling is unfortunate in that it has led to the use of inappropriate words to express what is happening and a lack of full understanding of the category of forward circular whip instability mechanisms. It is hoped that this work, although incomplete, will be followed by better mathematical theory and better experimental tests to help clarify the mysteries of rotating machinery stability.

SYMBOLS

The measurements and calculations are expressed in both SI and English units.

Q Amplification factor (dimensionless), as specified for lateral and angular position, and for synchronous or balance resonance speed

ζ (Zeta) damping factor (dimensionless) = $1/2Q = D/2\sqrt{KM}$

ω (Omega) angular velocity (radians/sec)

K Spring coefficient (lb/in)

M Mass (lb $\sec^2$/in)

D Damping coefficient (lb sec/in)

SYSTEM DESCRIPTION

The test rotor kit for this study used a slightly tapered aluminum wheel 3.2 inches long and 4.25 inches in diameter mounted at the mid-point of the span on a 0.375-inch diameter rotor shaft. The tapered wheel was fitted into an equally tapered seal housing mounted to the base of the rotor kit. The seal housing was designed for adjustment in the axial direction thereby permitting selection of a desired radial clearance from the aluminum tapered wheel. A diametral clearance of 30 mils was employed for all of these tests. An antifriction bearing was employed on the driver end of the rotor system and a bronze OIL-LITE bearing was used on the free end of the system. The total bearing span was 23.25 inches.

Dynamic motion measurments were taken adjacent to the seal housing with Proximity transducers mounted in an X-Y orientation. The data from the transducers was conditioned via a Bently Nevada Digital Vector Filter (DVF 2) and an HP 3582A Spectrum Analyzer and displayed on a Tektronix oscilloscope. The digitized data from these instruments was then acquired through the Bently Nevada ADRE Computer System and reduced into polar, Bodé, and spectrum cascade graphical formats.

TEST PROCEDURE

The rotor system speed was increased from 0 to 10,000 rpm; however, the acceleration rate was very slow to provide accurate identification of the translational and pivotal balance resonance speeds. The first balance resonance was observed at 1550 rpm. The test procedure was as follows:

ROTATIVE SPEED TESTING

The rotor system was slowly accelerated and the location of the first balance resonance was plotted and defined.

The rotor system was then accelerated to the rotative speed at which the onset of instability occurred.

The nature and frequency of the instability were then described.

The system was then accelerated until the shaft deflection initiated a rub between the tapered wheel and the seal housing.

This data base was then presented in polar, Bodé, and spectrum cascade formats.

PERTURBATION TESTING

A free-spinner perturbation device was attached to the rotor system adjacent to the seal housing.

With the rotor system held stationary, the free spinner was accelerated to 8000 rpm. This procedure documents the location of the first balance resonance.

The rotor system was then operated at a constant speed (approximately 4000 rpm). The free spinner perturbation device was operated in the forward mode (rotating in the same direction as the rotor system) and also in the reverse mode (opposite rotation of the rotor system).

STEADY STATE ATTITUDE ANGLE TEST

A spring scale was attached to the perturbation spinner keeping it from rotating but allowing application of a known soft unidirectional preload (like gravity) in order to determine the steady-state attitude angle of this rotor system at various speeds.

RESPONSE OF THE TEST ROTOR SYSTEM

Using a gap setting of 30 mils diametral clearance, it was observed that with forward circular whip at the self-balance resonance speed ("first critical") of 1550 rpm the rotor became unstable at about 6000 rpm. This is shown in the "There it is; darned if it isn't " cascade plot of figure 1. The polar plots and Bodé plots of the rotor system response with deliberate unbalance shown in figures 2, 3, and 4 confirm the self-balance resonance rpm to be centered at 1550 rpm.

A further test was run with the stator removed to confirm that contributions from other forward circular whip categories, especially internal friction, were not making large contributions to the action. It is the nature of rotor systems that this category is distinctly mutually aiding and abetting (in fact it is only the lagging attitude angle mechanisms of dry, semi-dry, and lubricated rubs that provide a limit cycle of these instabilities). The rotor exhibited about a 5 degree synchronous leading attitude angle through 10,000 rpm; thus, it was concluded for this fairly basic test that the other contributions were negligible.

The synchronous amplification factor Q (from figure 3) is approximately 6.6, so the damping factor ζ is about 0.075.

Next, the free spinner was run in the forward direction with just sufficient unbalance to drive the system without rubbing at the resonant peaks shown in figure 8. Obviously, the translational self-balance resonance damping factor is reduced virtually to zero, due to the existence of the destabilizing aerodynamic whirl forces. Figures 5, 6, 7 and 8 show this action.

The next test was run in exactly the same manner except that the perturbation from the free spinner is the reverse of shaft rotation. The cascade, polar and Bodé plots are shown in figures 9, 10, 11, and 12. The evidence that the aerodynamic forces are providing a net force adding to the damping to the reverse perturbation may be observed. The reverse resonance amplification factor is approximately $Q = 2$, yielding a reverse resonance damping factor of $\zeta = 0.25$.

It may be observed from figure 1 that the resonance increases from 1580 rpm with rotative speed of 3800 rpm to 1980 rpm at a rotative speed of 6800 rpm. Since this system has very little gyroscopic action for the translational balance resonance, and is not yet rubbing at 6800 rpm, it must be concluded that an additional direct spring is being added by the gas bearing. A calculation by the point transfer matrix method shows that for the resonance to increase by that amount, the gas bearing has an added 30 lb/in stiffness at its lateral location at the rotative speed of 6800 rpm. This effect seems to increase linearly with rotative speed.

The seal in the cartoon (figure 13) demonstrates the delicate balancing act that any bearing or bearing-like area always has in either a liquid, gas, or mixed medium. With the vertical input force shown, the shaft is moved over with just enough positive attitude angle to allow the lubrication wedge to balance the load exactly. In the situation of oil whip, steam whip, pumping whip, internal friction whip and all the rest of the forward circular self-excited instability mechanisms except oil whirl, it is easy to see that if a force were set up which acted counter to the system damping such as to nullify the effect of the damping, then the shaft is restrained only by the $(K-\omega^2M)$ term. Below a self-balance resonance region the spring constant K predominates, above a resonance the ω^2M term predominates. However, exactly at resonance, shaft orbit energy is exchanged between kinetic and potential energy (evenly between spring and mass). The limit size of the shaft orbital whipping is determined by additional drag-type damping from dry, semi-dry, and lubricated rubbing at seals and bearings.

While the steady-state attitude angle apparently can be any value (for this particular forward circular instability mechanism), high positive attitude angle (in the direction of rotation) is an indicator of a tendency toward instability. This test rotor, however, exhibited only 20 degrees positive steady-state attitude angle at 5000 rpm.

The situation of the very commonplace oil whirl forward circular instability is a most unusual type of instability mechanism. Rather than determining its frequency from spring and mass, it is reliant on the average speed of lubricant around the bearing and sets up oscillation by a difference equation which acts like a second order differential equation but is frequency dependent on rotative speed. Further, pure oil whirl will occur only when the steady-state attitude angle reaches and attempts to exceed 90 degrees.

In figures 2, 3, and 4 a small structural resonance at approximately 1600 rpm, caused by local variations in amplitude and phase on the Bodé plots may be observed, and also a pivotal balance response at 7800 rpm, but both of these may be ignored in this study.

For 30 years many experimenters have noted that these forward circular self-excited mechanisms have a tendency to get locked into 1/2, 1/3, 4/9, etc., of rotative speed. This author spent many hours puzzling about that tendency until he observed that it was simply a separate, resultive Mathieu effect from the nonlinearities of the partial rub. The mechanism of the Mathieu, Hill, Meissner, Duffing equation as applied to rotating machinery was described and documented by this author six years ago in an ASME paper, and recently further studied by D. Childs and by M. Adams. Any integer fraction of rotative speed may be latched onto by way of the Mathieu action, but this effect has very little to do with the prime action. It does, however, cause upward frequency shift in the "There It Is Again" cascade plots shown in figure 1.

On the general subject of rotor instability, it should be noted that there is a Mathieu effect that is a full reverse circular self-excited action, but it requires such poor damping to reverse orbiting action that this malfunction category remains a laboratory curiosity, and does not seem to appear in operating machinery.

FREQUENCY OF OCCURRENCE OF THE FORWARD CIRCULAR WHIRL AND WHIP SELF-EXCITED MECHANISMS RELATIVE TO ROTATIVE SPEED AND TO SELF-BALANCE RESONANCE SPEEDS

Oil Whirl: This mechanism occurs at an average rate of lubricant travel around the bearing, from 15% of rotative speed on a lightly loaded starved bearing to 48% of rotative speed (usually in the 40 to 48% speed range), and is governed by the relative roughness of shaft and bearing. It can be pulled to 50% or a little higher with a smooth bearing and rough shaft, but this is rarely a consideration. It can also be locked to any integral fraction of rotative speed by a resultive Mathieu action and must have 360 degree lubrication, except for the lightly loaded starved case. The 360 degree lubricant may be pure liquid, pure gas, or a mixed flow. The bearing with stable void islands in the high clearance area exhibits a classical Half Sommerfeld Curve and never can oil whirl as long as that is maintained. Pure oil whirl, like internal friction, must exhibit very high positive attitude angle.

Oil Whip, Radial and Thrust: This mechanism occurs at the self-balance resonance nearest the 40 to 48% rotative speed. With poor system damping it may occur from 15 to 85% of rotative speed, but occurrences are usually at 37 to 47%. It can also be locked to integer fractions of rotative speed by Mathieu action. This mechanism frequently occurs with 360 degree liquid, gas, or mixed flow lubrication but may also occur on the Half Sommerfeld Curve, with a small attitude angle. Thrust oil whip is a laboratory curiosity as thrust bearings are always segmented. However thrust bearing-like surfaces of impellers are fully subject to this action, whether handling gas or liquid.

Internal Friction: This mechanism occurs at the self-balance resonance speed that allows the greatest shaft deflection. Rotative speed must be any speed above that resonance speed. This action is also subject to locking at an integral fraction of running speed by Mathieu action and must exhibit a high positive attitude angle as in the case of oil whip.

Steam Whip, Pumping Whip, Aerodynamic Cross-Coupling, Alford Whip, etc.: This mechanism occurs at any self-balance resonance which allows major deflection of the shaft by bowing or by eccentricity in the bearings. In several instances of poor damping after seals have been opened by a prior malfunction, this excitation has occurred above running speed, but it most often occurs below running speed and at the translational self-balance resonance. On these compressors, the synchronous amplification factor to translational self-balance resonance had increased to $Q > 6$, so not only has the natural damping degraded, but the shaft is easily deflectable due to the excessive seal clearance. It may also experience locking by Mathieu action.

Vortexing, Helmholz, and Near Surge: While these are oscillations which may be separate from the rotor action as reported by P. Ferrara in an ASME paper two years ago and observed by this author, they are often highly mobile in frequency, and if they succeed in getting close to the self-balance resonance rotor rpm, will tend to latch onto that resonance, strongly exciting the instability. Their occurrence frequently has been observed from 10% to over 200% of rotative speed.

Entrained Bubbles in Pumps: While this is a forced category action, its symptoms are the same as the forward circular self-excited category. The bubbles amount to a lack of fluid mass and therefore provide the imbalance. The frequency is just below rotative speed down to 80% of rotative speed. A self-balance resonance in this range may be strongly excited by this circulating imbalance.

Conclusions: Some Notes on the Studies of Rotating Machinery Instabilities.

Even though a rotor is a simple structure that goes around in a circle and the compressible or incompressible fluids in bearings and seals between rotor and stator also go around, the rotor system tends to do more tricks than a monkey on a 100 meter rope. In every rotating machine, velocity-to-displacement is sinusoidal in timing, and therefore if the shaft is orbiting, the velocity vector is 90 degrees ahead of the displacement vector. Every rotating machine therefore has cross-coupled tendencies, allowing forces to be set up which may act against the damping forces. The more easily deflectable the shaft, the easier it is for these forces which act against damping to occur.

Add to this the complexity of the Navier-Stokes equation and suddenly a simple object becomes difficult to describe mathematically. Furthermore some unfortunate assumptions and misused expressions have come into existence and continue to be used. The use of the word "criticals" to express the rotor self-balancing speed regions is one example. Another is the use of the expression "unbalance sensitivity." This should be properly stated as an "imbalance response."

Another pitfall expression is "influence coefficient." The expressions "kelley constant" or "finagle factor" are equally inappropriate. When a calibration weight is added to a machine to unbalance it, there is a direct response observed at the lateral plane where the weight is added, and a transfer response at each other plane. It is a vector quantity, not a scalar, and it has dimensions. Typically, an unbalance weight of 1 gram installed at 8 cm balance hole diameter at 0 degrees, yielding a direct response of 4 p/p (peak-to-peak) mils at 172 degrees and a transfer response to an adjacent lateral place of 2 p/p mils at 355 degrees, yields at a specific speed:

Direct response $$\frac{4 \text{ p/p mils } \angle 172^\circ}{8 \text{ gm cm}} = \frac{0.5 \text{ p/p mils } \angle 172^\circ}{\text{gm cm}}$$

Transfer response $$\frac{2 \text{ p/p mils } \angle 355^\circ}{8 \text{ gm cm}} = \frac{0.25 \text{ p/p mils } \angle 355^\circ}{\text{gm cm}}$$

The use of the expression "log decrement" is also unfortunate. Log decrement is applicable where there are responses to unit or step impulses, as in diving boards. Amplification factor Q, damping factor ζ and attitude angle are directly applicable to the study of rotor instability. Often the expression "negative log decrement" is used to express instability. Of course it does, but the expression does not pass the "so what" test. Once the log decrement and damping factor become negative the rotor system is unstable by definition.

Perhaps the most misleading expressions, however, are the references to "cross springs" and "cross dampers." When these originated in the 1950s they were probably adequate for the original experiments, plus they have the misfortune of fitting into point matrix equations very neatly; too neatly. If these cross-coefficents are to be used, they must first be proven. Once proven, they may be acceptable provided that they are <u>not</u> a function of the term that they are multiplied by. For example, in the force term KxyY, if Kxy is a function of Y itself, then no differentiation or integration can be done on the term treating Kxy as a constant and still retain much relationship to reality. The term, however, could be a function of anything else.

It is most interesting that the strength of aerodynamic forces are often referred to in terms of lbs/in or newtons/meter instead of pounds, or newtons, or, if treated as moments, should be ft-lbs or newton-meters. A typical statement is, "This machine has an aerodynamic cross-coupling of 40,000 lb/in."

It would seem logical to go back to the basics and re-examine the fluid mechanics by both careful experiment and by sophisticated computer studies of fluidics finite elements and also by observing the results of iterative solutions to bearing and seals where the computer is given very few assumptions and is working on the Navier-Stokes equation with inertial effects included.

It should be more widely recognized that there are clearly two different mechanisms of instability in fluid film bearings. They are (1) the widely accepted stability rules of Half Sommerfeld assumed lubrication and (2) the largely neglected 1956 works of Cole and Hughes. The transparent bearing clearly shows that when a bearing can mix flow it can change suddenly to a 360 degree bearing and become unstable.

In addition to further experiments and mathematics to clarify the forward circular instability mechanisms of rotors, the following general rules should be applied for better control of harmful actions:

1. Provide more passive damping to the rotor system (this has limits if damping must be at the bearing as pointed out by E. Gunter).

2. Provide active damping by way of force balance active bearings (noting the limitation above).

3. Control the machine design, such as limiting the soft unidirectional preloading by gasses and liquids as well as limiting the introduction of the aerodynamic forces which act against the damping force.

4. Deliberately introduce reverse circular whirl mechanisms to the rotor system such as the propeller whirl shown by Chen and by J. Vance to neutralize the forward instability tendency of virtually all rotating machinery.

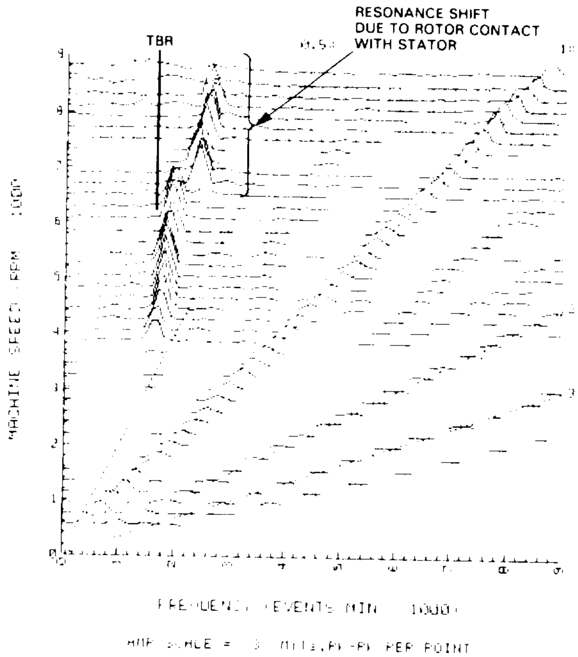


FIGURE 1

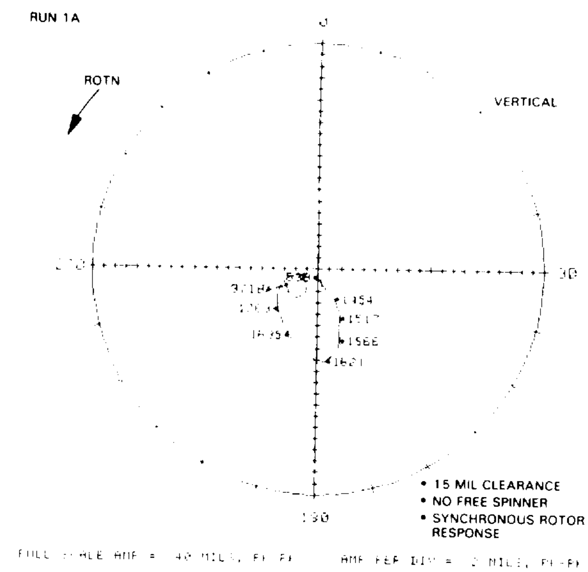


FIGURE 2

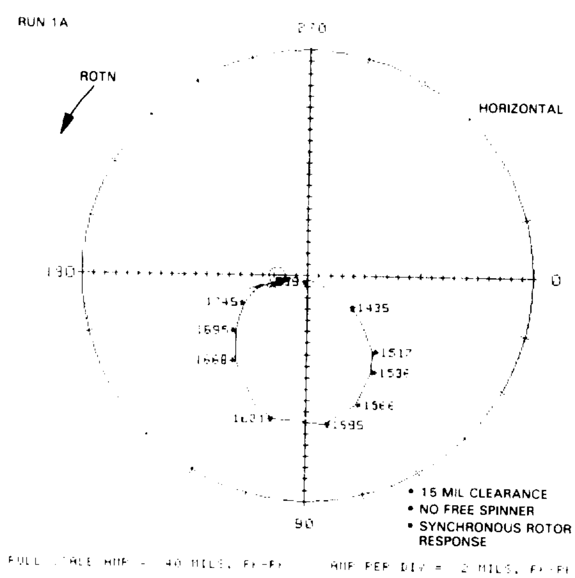
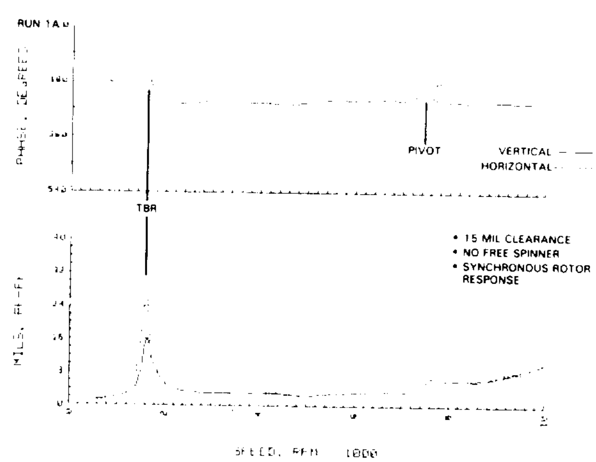


FIGURE 3



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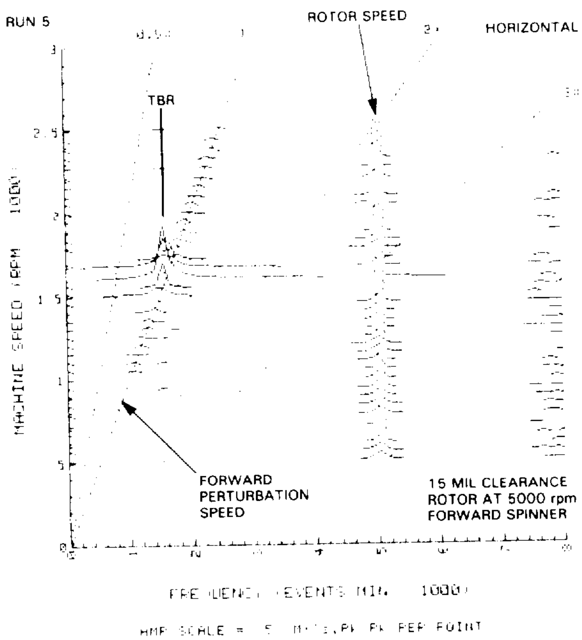
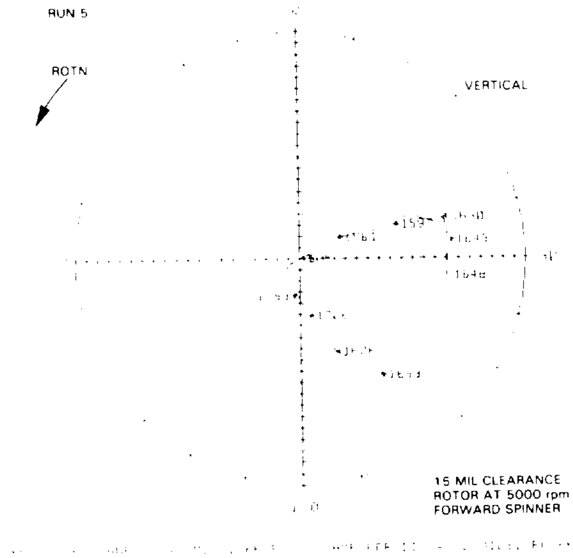


FIGURE 5



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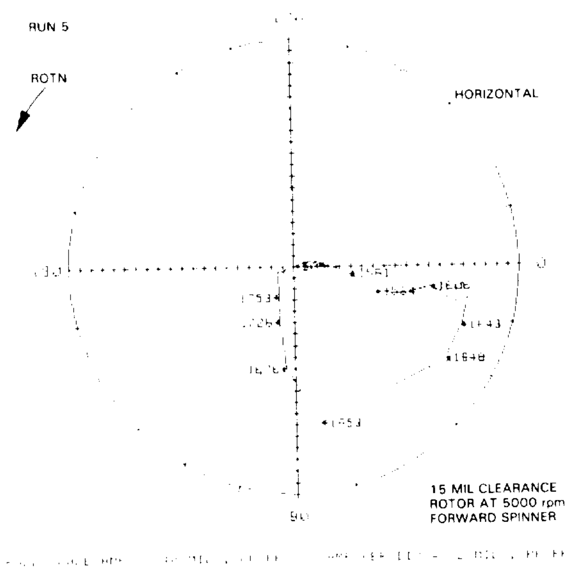


FIGURE 7

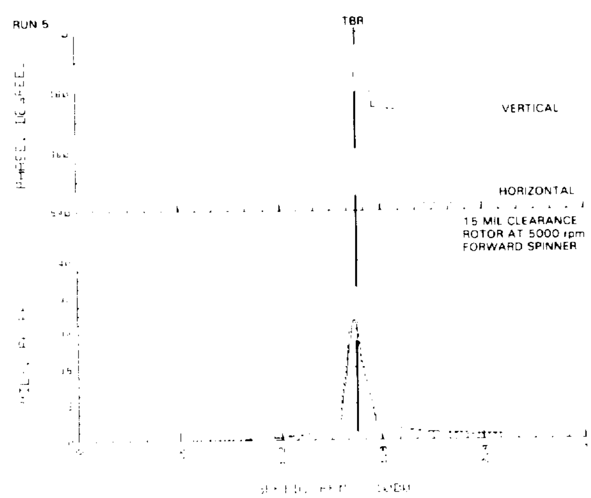
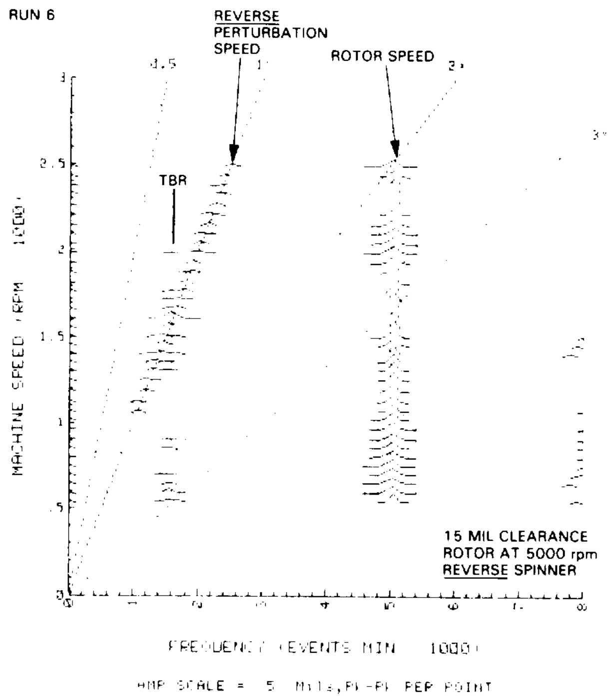


FIGURE 8



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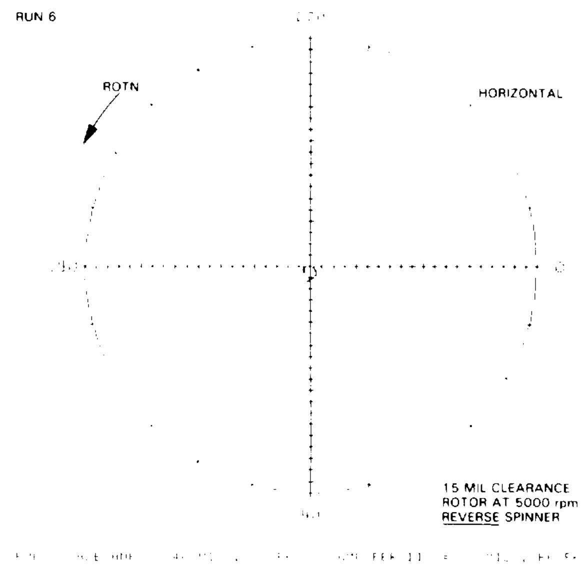
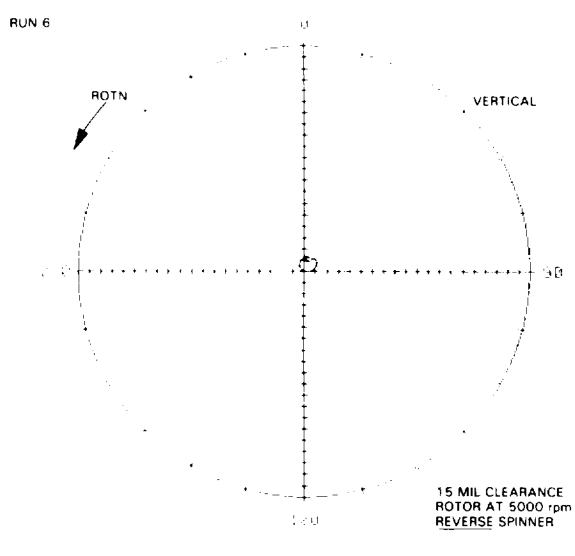


FIGURE 10



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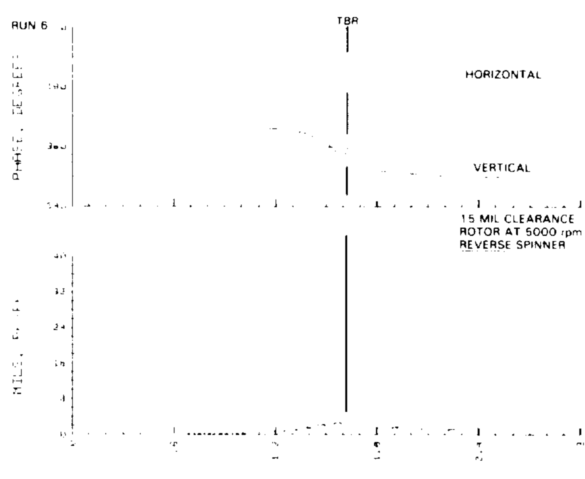

FIGURE 12

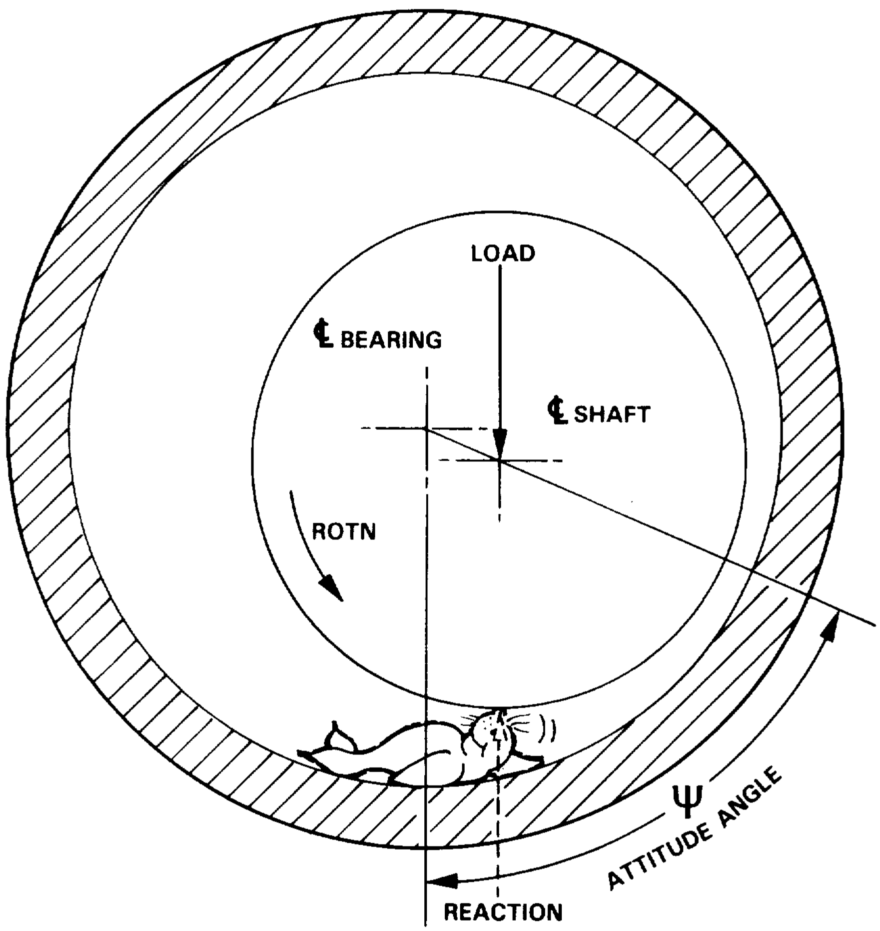

FIGURE 13