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DIAGNOSIS AND PROGNOSIS OF TURBOMACHINERY VIBRATIONS

*(Al Libertador Simón Bolívar,
en el bicentenario de su nacimiento)*

ABSTRACT

Vibration analysis is rapidly becoming a requirement rather than a luxury in turbomachinery development and implementation. Presented herein is a discussion on rotating equipment vibration problems - their occurrence, their diagnosis by analytical and experimental methods, and a look to the future in this area.

RESUMEN

El análisis de vibraciones está convirtiéndose rápidamente en una necesidad más que en un lujo en el desarrollo e implementación de turbomaquinarias. En este trabajo se presenta una discusión acerca de los problemas de vibración en equipos rotacionales, su frecuencia, su diagnóstico por métodos analíticos y experimentales, y una visión sobre el futuro en este campo.

1.0 INTRODUCTION

The trend of turbomachinery, such as turbines, motors, compressors, pumps, fans and etc., is higher speeds, in order to achieve better efficiency and higher power density. As a result, vibration problems with these machines are more frequently occurring in oil fields, oil refineries and electric utility and petrochemical plants, due to lack of precaution at the design stage. Typically, a manufacturer will "size up" the existing machines to meet the demands of the user. This design procedure very often ignores consideration of stress and vibration analyses. The new machine may be eventually unable to reach its design speed and performance due to a critical speed problem, or a rotor-bearing instability problem. Vibration problems are not just a nuisance which may be tolerated, or cause compromise, but rather, they are detrimental to the machinery reliability and to the economics of the user and the manufactures.

During the past two decades, vibration analyses and field trouble shooting have been developed into a specialized engineering discipline, attributed largely to computer and instrumentation technology development.

Based on past "hand-on" experiences, the authors present herein their viewpoints of :

- Types of vibration problems frequently occurring,
- diagnosis of vibration problems by experimental means,
- computerized analysis of a vibration- related design audit, and
- the future trends in this area of the engineering.

2.0 CATEGORIES OF TURBOMACHINARY VIBRATIONS

There are in general, two types of vibrations, namely; structural type and rotating machine type. Interest in the structure-related vibrations includes those associated with machine casings, pedestals, piping and steel frames or concrete foundations which support the rotating machines, or can be excited by the machine vibrations. The main interest is in the second type which includes lateral and torsional vibrations and rotor-structure interaction.

2.1 LATERAL VIBRATIONS

This is a condition in which the rotor vibrates in planes perpendicular to its rotating axis. These are the most commonly occurring vibration problems, and are also separable into two modes by their causes.

a. Synchronous Vibrations: In this mode the frequency of the vibration is identical to the rotor speed. Its amplitude is associated with the amount of imbalance and the critical speeds. Uneven mass distribution of the rotor about its rotating axis, that is called imbalance, causes distributed unbalanced centrifugal forces along the axis during rotation. These forces, reacted upon by the bearings, are transmitted to the casing and foundation, and vibrations are thus created. The rotor can be balanced by adding, or cutting away some material at one, two or even several locations on the rotor. But the rotors state of balance in operation can be worsen due to corrosion, errosion or uneven deposits of material in the flow stream. Worse yet, if the rotor is operating close to a critical speed or passing through a very lightly damped critical speed, a normal amount of residual imbalance can cause magnified centrifugal forces to be imposed on the rotor. Imbalance can be caused by a loosened wheel at high speed; or a high spot rub on seals. An improper shrink fit design between wheels and shaft may cause shaft bending when passing through a critical speed and change the state of imbalance.

b. Subsynchronous Vibrations: Many different names have been given to this type of vibration, such as oil whirl and oil whip for rotors with oil bearings, steam whirl in steam turbines, half frequency whirl (HFW), or fractional frequency whirl (FFW), since the frequency mostly is less than or close to 50% of rotor speed. But generally, these vibrations can be classified as rotor-bearing system instabilities. A typical description is, as a rotor has succesfully passed through its first natural frequency or, critical speed, and continues on to higher speeds, to say, approximately, twice the first critical speed or during a load change, the rotor vibration will suddenly start to grow in a short period of time at a frequency close to the first critical (sometimes called first critical reoccurance). By changing the load and/or reducing the speed is the only way of reducing this type of vibration.

2.2 TORSIONAL VIBRATIONS

Angular, or twisting resonances about the rotor axis are possible, especially in systems with large inertias, such as, geared rotors, reciprocting machines, and synchronous motor driven compressor trains. The rotor systems with torsional problems often result in coupling failures, crushed keys and keyways, gear teeth pitting, or even cracked shafts. The 45 degree spiral crack of a shaft is a typical high-cycle torsional fatigue problem indication.

Industry's emphasis on energy efficiency has recently increased the use of synchronous motor driven systems. These systems with fixed electrical frequency drives, especially those with solid poles, are vulnerable to a torsional transient vibration problem. During every start-up the machinery train passes through a critical speed where the motor fluctuating torque with a " two - times slip" frequency coincides with the first system torsional frequency. If the acceleration is low, the vibratory torque on the rotor may exceed the yield strength and cause permanent strain and eventually a low-cycle fatigue failure.

2.3 ROTOR-STRUCTURE INTERACTION

An increasing awareness has been in the interaction of the rotating machines and their support structures. There are case histories suggesting(1)*

*Numbers in parenthesis refer to identically numbered references in Reference Section.

"a seemingly perfect (vibration free) machine, supported by a well designed structure, becomes a vibration troublesome dynamic system".

The fact the machine and the support structure, such as a concrete foundation, with or without pilings, are dynamic systems themselves, and their natural frequencies are designed to be away from the operating speed of the machine. However, when they are tied together, the total dynamic system may have resonance frequencies close to the operating speed. Therefore, it is the "dynamic interaction" which may have been overlooked.

Earthquakes vibrate foundations and machine casings. Large transient dynamic loads due to earthquake excitation can ruin the machine bearings if natural frecuencies of a machine/foundation system are inside the earthquake vibration spectrum.

3.0 VIBRATION DIAGNOSIS WITH EXPERIMENTAL MEANS

The cause(s) of an existing vibration problem can be difficult to pinpoint when time, available tools, and expertise are limited. The persons involves, even the "old pros" tend to be subjective putting emphasis on the causes known to their own fields of competency. In addition to an objective mind full of logic, these engineers should possess the following capability, or training:

- An understanding of the vibration phenomena obtained by acquiring experimental data of cause-related frequencies and amplitudes by proper instrumentation.

- An ability to relate the data to possible cause(s), or theories, and to use simple mathematical models of the machine/rotor structure to explain/verify the cause(s).

The first capability involves a knowledge of the experimental subject matter which follows.

3.1 VIBRATION INSTRUMENTATION(2)

Vibration instruments may be categorized into three kinds according to their functions:

- Transducers and conditioning devices,
- analyzers,
- recorders.

The transducers (or sensors) are electro - mechanical in nature and are the basic instruments used to monitor the vibration signals from a rotating shaft, machine casing or bearing block, etc.

For lateral vibrations, the Piezoelectric accelerometers can measure acceleration signals from one Hz to forty KHz. They need signal conditioning devices such as charge amplifiers to boost the signal power for displaying on an oscilloscope, or for determining the frequency content and amplitudes by use of analyzers. The measured amplitudes are proportional to the square of the vibration frequencies. Therefore, accelerometer are suitable for measuring high-frequency, low-amplitude type of vibrations.

The velocity probes as the name implies, measure vibration velocity. They are the most popular sensors used in the field because of their ruggedness and self-generating capability. (No need for a conditioning device). But they are made with a seismic principle (large mass on soft spring) and have a cut-off frequency at 10 Hz. Below this frequency the measurement is not accurate. Therefore, they are not suitable for low frequency measurements.

The displacement probes of the eddy - current type are common in high speed turbomachinary for monitoring shaft motions in bearings. Oil film bearings, as will be elaborated upon later, are the most sensitive part of these machines. High speed usually means high pressure and heavy casings. The heavy casing may not vibrate much, or be sensitive enough to relay the signal of an impending catastrophe or severe rotor vibration problem. Therefore, monitoring the journal motions with displacement probes is more direct and assuring. But the erroneous signal due to journal run-out (out of roundness or residual magnetism) is often on the same order as the vibration signal (measured by units of 0.001 inch), and must be compensated by accurate machining and electrically.

Preference of one type to another may be dictated by the circumstance and availability. The user should always bear in mind the proper meaning of the signals according to the sensors' function. In Figure 1, the same vibration is represented by three types of signals, i.e., acceleration, velocity and displacement. On the acceleration spectrum, the high frequency content seems predominant. But it may be normal such as from a gear box measurement, and the real problem signal may be hidden in the low frequency level such as the first small peak representing the synchronous vibration amplitude.

For torsional vibration measurements, the Torsiograph is the commonly used sensor for sensing angular vibration velocity. It is made with a similar seismic principle as the velocity probe and thus also has a low end cut-off frequency. It is usually used at one end of a shaft with an extension stud. For measurements at middle shaft locations, special rigs belt drives must be used. This is not a reliable means due to possible belt slip. A more recent popular method is shown in Figure 2. Here a non-contacting FM modulation principle is applied. Similar measurements may be achieved using precision gears and magnetic pickups. The number of teeth or blackwhite spacings, multipled by shaft speed, must be ten times larger than the torsional frequency to be measured. Strain gaging with a radio telemetry system is the ultimate way for measuring static and dynamic torques on a shaft. However, this method is expensive to implement.

After a vibration signal is measured, one wants to know what is the frequency content and what is the amplitude associated with each frequency (Figure 3). The former is related to the causes and the latter is an indication of the vibration seriousness. A portable analyzer with tunable frequency filters and an amplitude meter is adequate but relatively slow when analyzing the signal. Real time analyzers or the FFT analyzers (by digitizing and fast fourier transform computation) are ideal in that they display the whole spectrum of the signal and provide a complete picture of the problem.

. The enviroment in the fields is normaly harsh - - it may be too noisy, cold, hot or wet. Also, when many signals are measured at one time, their analysis on site is difficult. Therefore, using a magnetic tape recorder to store the signals simultaneously on tape for laboratory analysis is a standard practice today. Two types of recording i.e., direct and FM are available. Direct recording is suitable to high frequency and FM recording must be used for low frequency and static DC signals such as shaft torque or shaft displacements.

3.2 SIGNATURE ANALYSIS

The patterns of vibration signals measured at the machine casing, shaft, etc., are different even for identical machines and are called signatures. The purpose of analyzing these signatures is to determine the vibration cause(s) and seriousness as they are related to the conditions of the mechanical parts with vibration amplitude, frequency, phase angle, wave form, etc. Ideal situations for signature analysis would be to have the following:

- "Baseline data" measured when the machine was new, or with an identical machine without vibration problems, and

- analytical information such as critical speed maps, imbalance response plots and stability analysis results.

This is seldom the case for most vibrating troublesome machines, especially when they are old. Normally the vibration considerations were not commonly implemented in design or maintenance. In these situations, some published information such as,

Vibration frequency vs. causes (Table 1), and
Vibration measurement vs. seriousness (Figure 4),

are helpful in analyzing the signatures.

Additional to examinig the signal frequency spectrum (Figure 5) by using the analyzers, one can observe the wave form of the vibrations or the filtered signals. For example, in Figure 6, large amplitudes resulted by the beating wave form was obvious on the platform of the two-shaker tower. This was caused partially by the speeds of the shakers being too close to one another. Orbit analyses of journal(3) (shaft in bearing) motion as shown in

Figure 7 is also common practice for analyzing displacement signals. These wave form analyses are performed on the oscilloscope with or without using filters. When synchronous vibration is the only concern and the measured signal is cloudy with some unwanted signals, tracking filter plots such as in Figure 8 can be made for transient vibration to show the effect of a critical speed.

About a decade ago, a high frequency diagnostic scheme was developed for detecting impending ball bearing failure,(4) Basically, this problem mechanism (such as ball impacting a pitting spot) generates high frequency resonance of parts, such as a ball inner or outer race. The resonance has very small displacements in the unaudible frequency range. The ball impact frequency is relatively low and not detectable in the low frequency range. But the impacts modulate the resonance and become detectable in high frequency in a manner like an AM radio wave (Figure 9). The technique since has become a useful means for detecting impacts of other kinds, such as gear meshes in a transmission box for production or maintenance purposes.

To this point discussions have been on experimental topics to the first capability needed by a vibration, trouble-shooting engineer. The second capability is the ability to explain the observed vibration with possible theory.

In rality, there is no distinct simple mass-spring system as vibration textbooks often define (5), but the problem should always boil down to one or two major natural vibration modes as the problem.

For example, the problem with the shaker tower in Figure 6 was not a pure beating problem, because even with only one shaker operating, the vibration of the platform was still unacceptably high. At that time a question was asked whether the shakers' force could be isolated by resiliant pads. The answer was no, based on considering a dynamic mode with two degrees of freedom. The bottom (frame) mass is too small compared with the top mass (shaker bed) for isolation to work. The resiliant pads may, however, dampen part of the high frequency audible noise and make the enviroment quieter. To identify the natural mode of vibration, the main actor in the problem, one may consider the following:

- The whole tower with heavy machinery on the top half is a soft dynamic system and has several low frequency lateral swaying modes. They were indeed identified by touch during the end of a coast down of the shakers.

- The mode causing the problem should have a frequency close to 880 CPM, the normal shaker speed. To be so high in frequency, it should be a local vertical deflection mode of the platform.

By further study, one eventually could identify that beams were too soft such that they contributed to a vertical mode lower in frequency but close to the operating speed. The authors have no intention to generalize here but rather to use this for illustrating the mental process of applying simple math models in vibration diagnoses.

Shown in Figure 10 is another example of using simple models for explaining vibration phenomenon. The compressor train was in a troublesome axial resonance. Two screw jacks were used as temporary measures for bracing the casing to ground, and they had to be tuned to achieve minimun axial vibration. It was not difficult to explain the function or the effect of the screw jacks by using a model of two degrees of freedom. The difficulty was how to judge the cracks on the concrete piers. Were they result of high axial vibration, or the cause of lowering the axial frequencies? This is a typical "chicken or egg" (which comes first) story one may be confronted with in the field.

3.3 ROTOR BALANCING

Many vibration problems are of the synchronous type and can be corrected by balancing the unevenly distributed masses if the rotor is not operating near a critical speed. In the field, one can perform single plane or two-plane balancing. For single plane balancing(2), a trial weigth is put at a sensitive location of the rotor and the change of vibration is measured and calculated to determine the imbalance location. The procedure is called an influence coefficient method. The same used in two-plane balancing(6) needs two trial weights and two balancing planes when rotor imbalance is spread along the axial direction. Note that the calculation involves vector algebra because every measurement, or trial weight is defined by magnitude and a phase angle. Phase angle measurement is in some situations difficult to make. For example, at the top of a cooling tower with several identical fans running, the beating vibrations have fluctuating amplitudes. In this case one may choose to perform the single plane balancing without using phase angles(2).

The field balancing method above assumes the rotor is a rigid body which basically is true for a low speed rotor not passing any critical speed. The rotor axis bends into the critical mode shape when it passes through a critical speed. How much it bends depends on local imbalance mass distribution. Proper choice of balancing planes, or more than two planes are needed to balance a rotor at several speeds. A standard practice used is to balance the components of the rotor separately, assemble the components in two or more stages and trim balance the sub-assembly at each stage again by using one or two planes.

During the seventies, a laser balancing scheme was developed at Mechanical Technology Incorporated, New York, U.S.A., for automated mass production, (Figure 11). A computer controlled laser beam is directed to cut away the rotor imbalances at different balancing planes while it is rotating. The influence coefficients associated with different balancing planes and different measurements locations are pre-stored in the micro-computer for identical rotors. This scheme eliminates the time spent in stopping the rotor and cutting away mass (or adding mass) manually.

4.0 ANALYTICAL METHODS FOR VIBRATION PREDICTION

Machinery vibration dynamics involve the following elements:

1. Rotor (shaft and attachments such as wheels) mass and flexibility,
2. bearing (and/or seal) stiffness and damping,
3. foundation mass, stiffness and damping.

Assuming the pedestal and foundation are rigid and massive enough, the rotor and bearing are the first order of importance in vibration analysis.

During the past two decades, computer programs in rotor/dynamics have been developed for access by most large computer center librarys is the U.S.A. Mathematically, the rotor is treated as distributed masses linked with distributed lateral or torsional springs (Figure 12). Gyroscopic inertia effects of large wheels are included as a major difference from a traditional beam approach. The bearings are treated as concentrated springs and dashpots including cross-couplings effects between the two perpendicular directions.

4.1 BEARING STIFFNESS AND DAMPING

Until more recently, critical speed calculations frequently ignored the bearing effect. It is is now recognized that bearings are a critical dynamic element because of their stiffness contribution and the fact that they are primary damping sources of the dynamic system.

Rolling element bearings posses very little damping. A simple way to estimate their stiffness is given by available force-deflection relations (7). However there are no simple formulae to cover the different geometries for fluid film bearings, such as:

Plain cylindrical

Axial grooved

Elliptical

Off-set

Pressure dam

Partial arc

Tilting pad

Basic lubrication computer programs usually generate bearing data in non-demensional from, such as shown in Table 2. This table can be used for bearings of the same type and L/D ratio. An important parameter in fluid film, self-acting, bearing analysis is the inverse Sommerfeld number defined as :

$$\frac{1}{S} = \frac{W}{LD}\ \frac{1}{\mu N}\ \left(\frac{C}{R}\right)^2$$

It is a unique set of variables arranged to represent the "normalized load". The load is a function of the bearing dimensions (L,D,C,), oil viscosity (μ) and journal speed (N). Knowledge of this number provides the key to stiffness and damping by interpolation from the tables.

Hydrostatic bearings (externally pressurized) are required in certain enviroments such as sodium pumps for nuclear plants and non-oil bearing applications, such as food processing plants and steel mills. Simple formulae(8) for estimating this bearing stiffness were developed. Large dedicated computer programs are also available in the software market.

The wear rings in centrifugal liquid pumps(9) can be considered as another type of hydrostatic bearing. Of course the original function of the wear rings was meant to limit the leakage between high and low pressure zones. But the high axial leakage flow (usually turbulent) produces a kind of Bernoulli effect called Lomakin mass effects (10) and tend to force the journal to a concentric position. The equivalent fluid-film stiffness and damping of these rings are proportional to the pressure drop across the rings and the basic elements that a pump is called upon for proper dynamic behavior. If the pump is running dry, it's first critical speed could be very low compared to its normal speed. When it is pumping liquid and the rings are functioning as bearings, its critical speed may always be higher than running speed until the rings wear out, (clearance opens up to larger values).

4.2 ROTOR-BEARING LATERAL DYNAMIC ANALYSES

A) CRITICAL SPEED: Lateral vibration analysis starts with the critical speed or natural frequency calculation. The bearing damping may, or may not be included. A popular way is not to include the bearing damping and assume the bearing stiffness to vary over a large range. The purpose of this treatment is to evaluate the effect of bearing stiffness on critical speeds, usually up to three or five lowest criticals. Computerized results can be plotted onto a graph such as Figure 13; called a critical speed map. The map can be divided into three regions; (a) rigid rotor, (b) soft rotor and (c) rigid-support rotor. With the bearing stiffness versus speed superimposed on the map, one can locate approximately the system critical speeds. The critical speed calculation also provides rotor mode shapes (Figure 14) at different bearing stiffnesses. In the rigid region, the rotor generally has large mode shape displacements at the bearings which is indicative of effective use of bearing fluid damping. The rotor vibration problem is more controllable although the critical speed is lower. In the rigid-support region the opposite is true. A practically designed rotor-bearing system usually falls into the middle region. One cannot afford too soft bearings because this would mean large sizes and power losses. One should not design too stiff a bearing because this would thus dictate thin oil films, low system damping, and poor vibration control.

Critical speed calculations with the computer is inexpensive but reveals most of the rotor/bearing system characteristics qualitatively. An experienced analyst can forecast the imbalance or stability problem by examining only the critical map and mode shapes, or he can decide what further analyses (unbalance reponse or stability) are needed.

B) UNBALANCE RESPONSE: This analysis is concerned with how large the synchronous vibration amplitudes are at tight clearance locations such as the bearings, the labyrinth seals and etc., as the rotor passes through the critical speed zones and operates in its normal speed range. Rotors are normally balanced during the manufacturing stage with some residual imbalance remaining. The acceptable amount of residual imbalance in industry is about:

for low speed machine, $\frac{4W}{N}$ oz-in / balancing plane

for high speed machine, $56000\ \frac{W}{N^2}$ oz-in / balancing plane.

In order to excite a critical mode of vibration, imbalance must exist at locations where the the mode shape indicates large displacements. Since in practice, the exact location of imbalance is generally unknown, one must judge where it is likely distributed. A conservative approach is to put the acceptable imbalance at probable locations, such as at center wheel and end couplings, to simulate the worst possible practical response. A computerized response calculation uses the same mathematical rotor model as used for the critical speed calculation and in addition the bearing stiffness and damping is varied as a function of speed change. Figure 15 is a typical response plot indicating the vibration amplitude of a centrifugal compressor and emphasizing the first damped critical speed.

C) STABILITY: A rotor/bearing system operating much above its first critical speed may be unstable in the sense that slight speed increases or load changes would disturb the rotor into a large orbit whirl with a frequency very near the first critical frecuency(11,12 and 13).

By examinig the relative position between the operating speed, the rigid support first critical and the flexible support first critical, one may predict (14) the possibility of instability as shown by the example in Figure 13.

The cause of the problem can be explained generally by the existence of tangential (or cross-coupling) driving forces on the rotor as depicted in Figure 16. The tangential forces, usually in the forward direction, (rotational direction) are contributed by oil film bearings, locked oil ring seals, labyrinth seals, and submerged rotor with a small clearance to the statinary wall or impellers. The forces cancel the damping force in the bearing and tend to drive the rotor into a large orbit.

Many decades ago, engineers observed "oil whirl" in lightly-loaded or vertical journal bearings. This is a kind of rotor-bearing instability contributed mainly by the cross-coupling forces in the bearing. It was explained that as the journal speed (Ω) reached twice the critical speed, the oil pumped into the wedge equals the oil swept out of the wedge. Thus the oil film could not be maintained and the journal orbit grew. One of the cures was to load the bearings artificially by preload or a pressure dam pocket, or to cut grooves in an attempt to decrease the cross-coupling forces. Although oil bearings are generally loaded in turbomachinary, they still remain as the main source of cross-coupling forces. For high speed machinery, the tilting pad bearing became popular, because theoretically there exists no cross-coupling force in this type of moving pad bearing. Locked, floating ring seals function as plain journal bearings (15) and have caused many instability problems in high pressure centrifugal compressors. Other sources of instability were observed in the impeller narrow clearance area including the interstage labyrinth seals(16 and 17). The magnitude of these forces are still not well defined and generally use of load balanced seals and anti-swirl flow devices are used to offset the forces. A current design approach is to calculate the "stability margin" of a rotor system including bearing cross-coupling forces only. Another name used for stability margin is the log decrement of the first system natural mode when the rotor is running at operating speed. Note that the frequency of the first mode is close to the first critical speed, not necessarily, exactly the same. In mathematical terms, the mode is associated with a set of Eigenvalues/Eigenvectors of the dynamic system with multi-degree of freedom. The real part of the Eigenvalue, i.e., the growth factor (a) is a term representing the modal damping. The imaginary part is the natural frequency ($2\pi f$). The log decrement of this mode is defined as

$$\delta = \frac{-a}{f}$$

A design guideline from numerous experiences is to require the log decrement be greater than 0.3. Physically the log decrement is a measure of how fast the system can dampen the vibration in this mode due to any disturbance.

4.3 TORSIONAL VIBRATION ANALYSES

Torsional vibration modes are usual lightly damped -- less than 5% of critical damping. The bearings do not provide damping to the torsional system, unless the design is a geared rotor with a large gear motion due to torsional vibration. For these high "Q" natural modes, engineers are usually only concerned with their frequencies. If the frequencies are not close to the operating speed or speed range, and the system does not have a chance to dwell at any of these modes of vibration it should be safe. Therefore, only undamped torsional frequencies are calculated.

The results are commonly presented in a Campbell diagram format (Figure 17). This diagram indicates possible interfences with shaft speed and its harmonics. Large motor driven fans with a variable operating speed frequently suffer from this type of problem. The results are often failed couplings. To detune a natural torsional frequency away from a fixed operating speed can often be easily accomplished by changing the coupling torsional stiffness. If the operating speed is variable in a large range, the detuning may involve two or three natural vibrations modes.

Computerized torsional frequency calculations use similar mathematical rotor models as in the lateral analyses, but they are simpler and less expensive to perform.

For large geared rotor systems driven by synchronous motors, extra analysis of the torsional transient response is required to predict the maximum fluctuating torque during start-up. The motor oscillating torque at twice slip frequency,

$$f_s = 2\ (1 - \frac{\text{rotor speed}}{\text{synch. speed}})\quad \text{(line frequency)}$$

will excite the first torsional frequency, f_1 shortly before the motor goes into its synchronous speed. If f_s coincides with f_1 at a time when the motor average acceleration torque is low, and the oscillating torque is high, the lightly damped first mode can build up its energy and become a violent twisting vibration. To calculate this transient response involves :

- Condensation of the distributed model into a lumped inertia model with much less degrees of freedom.

- Tuning of the condensed model to yield approximately the same torsional natural frequencies as the distributed model.

- Integration of the condensed model with small time steps (by the Runge-Kutta method or Newmark-method) and prescribed initial conditions, motor torques, and load-speed functions.

Typical results of the shaft torque when passing through the critical speed area is shown in Figure 18.

Computerized calculations of this kind can be very expensive. The authors have developed a simplified and less expensive approach using modal data from the frequency analysis and a pre-integrated, non-dimensional plot. The new method is to be published in the near future.

5.0 FUTURE TRENDS

Figure 19 shows a high speed pump rotor with four stages which has the same output as a large low speed pump rotor with nine stages, it typifies the turbomachinery design trend of the future. Since high speed and small size goes with high pressure and small clearances, the design analysis consideration of rotor-bearing-seal dynamics will be inevitably among the top priorities in early stage of new designs. In other words, rotor dynamic analyses and the basic machine function analyses such as thermal cycle and efficiency become equally important. Dynamic analysis was once a luxury; it is now becoming a necessity. At the time of a new design concept plan on the drawing board and when the approximate rotor dimensions and bearing type/size are chosen, a rotor/bearing dynamic audit must be performed. A typical audit process(18) is shown in Figure 20. If a vibration problem is identified, the rotor design must be modified, even if a sacrifice of performance is required.

Large rotor/bearing dynamics computer programs containing all the basic analyses capability, emphasizing machine / foundation interaction, (19) streamlined input/output with graphics are being developed for large computer users. Software packages for these basic analyses will be available for micro-computer users and designers.

The machinery user, who is buying a new variety of turbomachinery, should request that the design comply with published guidelines, such as API(20) specifications or the equivalent in the rotor dynamics area, or, assure this is accomplished by having an independent third-party audit performed.

On-line computerized vibration monitoring signature analysis, vibration trend analysis, shut-down vibration data aquisition and analysis for important equipment are all on the horizon. A coined name "Down Time Management" (DTM) has appeared recently. The goal of DTM is to plan shut-down time by prediction of impeding vibration problems.

The machinery users will need more specially trained technicians and engineers dedicted to vibration related maintenance work. The personnel with proper machinery, computer and instrumentation background will be trained in vibration short courses and seminars which are becoming more available and popular.

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7.0 LIST OF FIGURES AND TABLES

LIST OF TABLES

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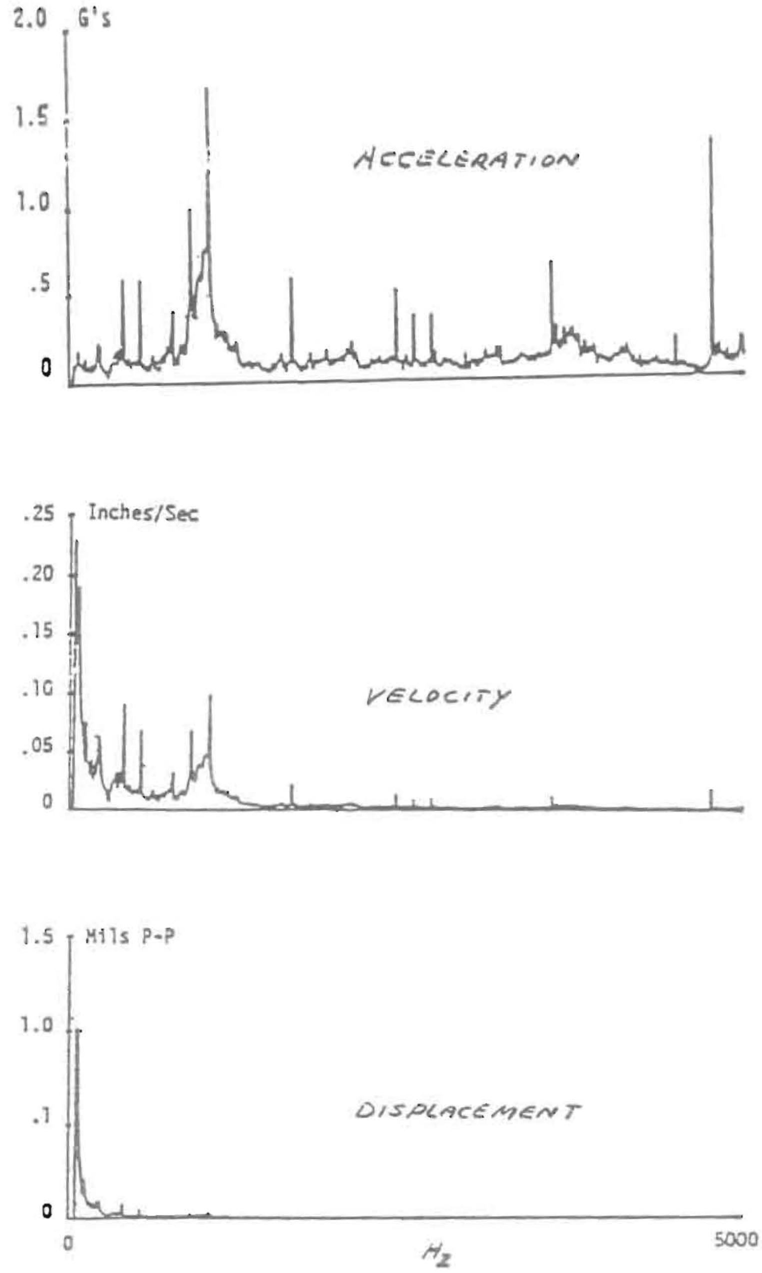
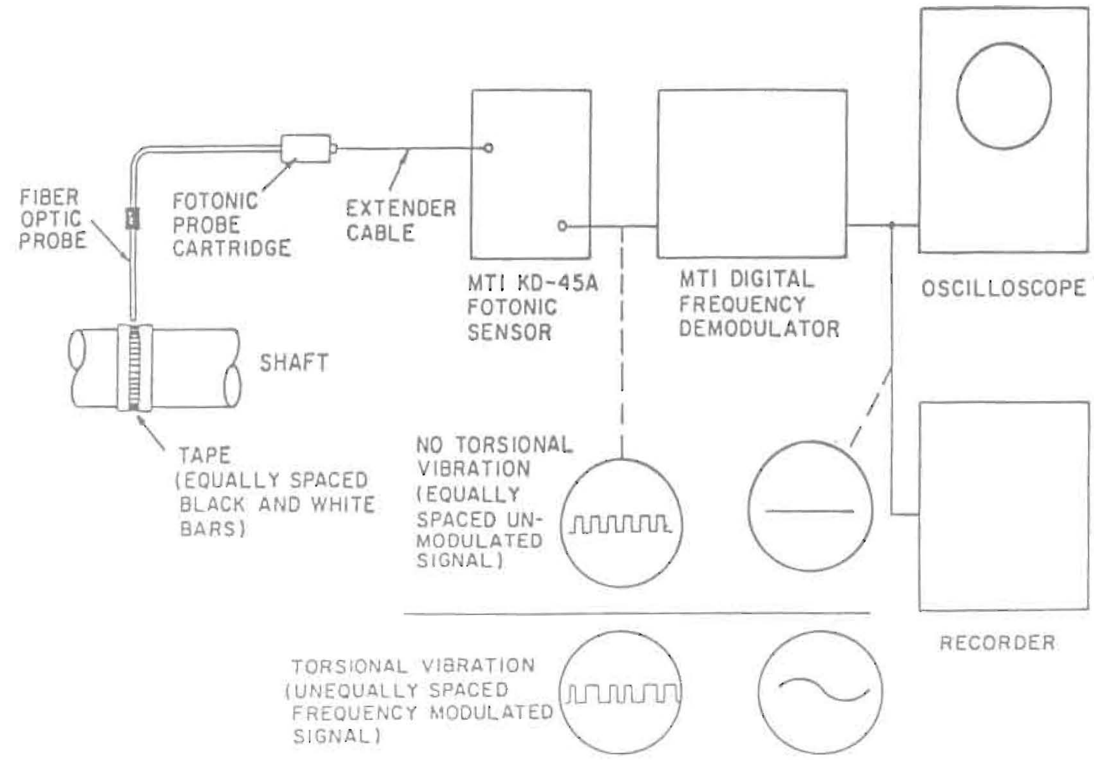


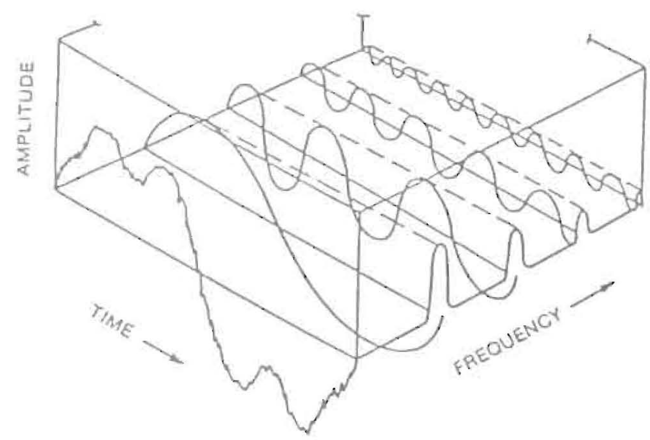
Comparison of Vibration Amplitude versus
Frequency Plots Measured with Displacement.
Velocity and Acceleration Devices .

Figure 1



Non-Contact Torsional Vibration Measurement System

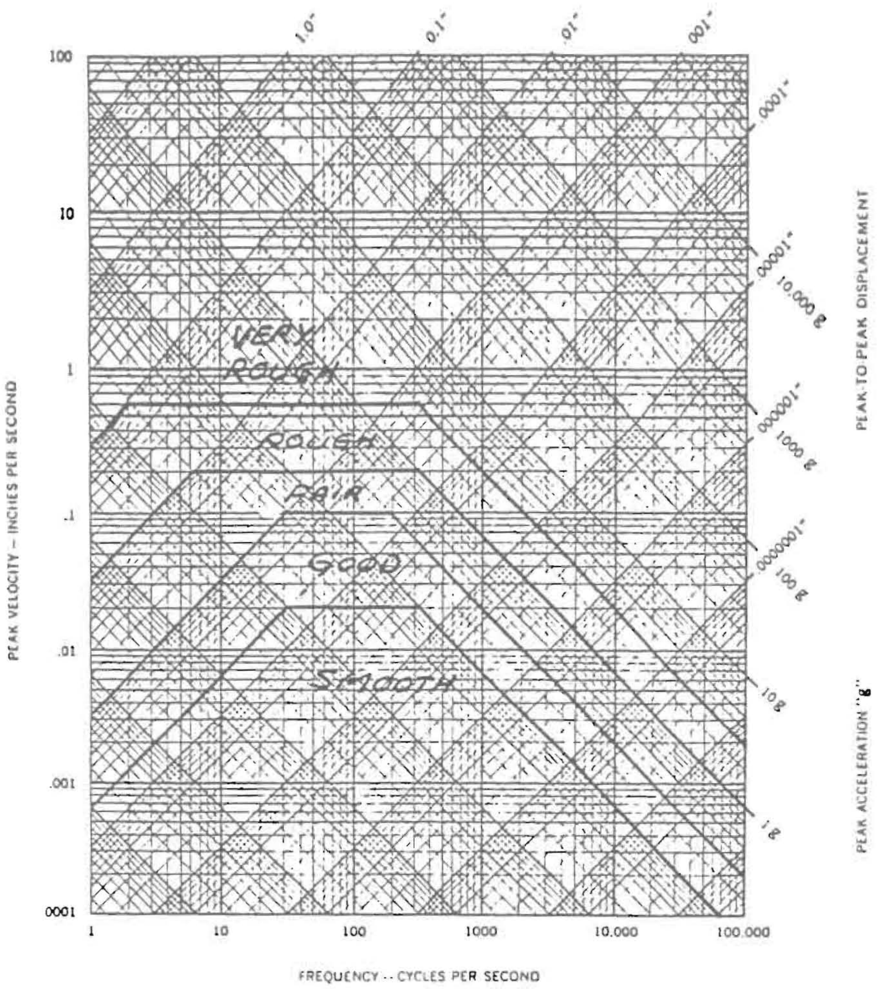
Figure 2



3-D Plot of Amplitude versus Time
at Different Frequencies

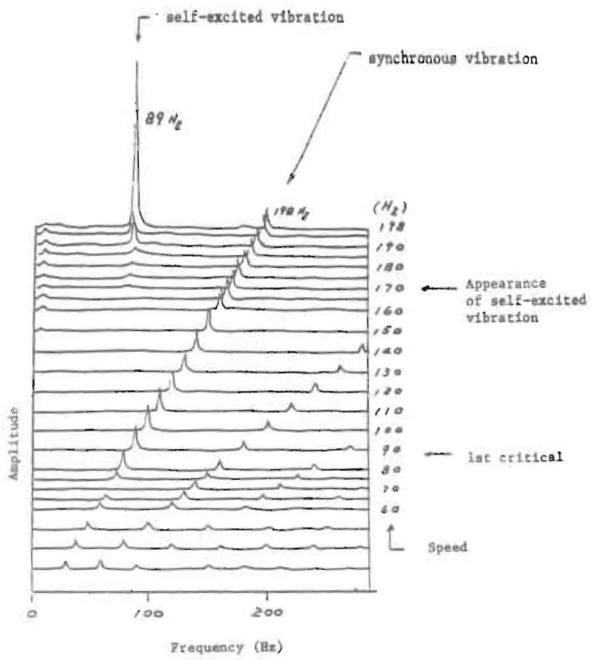
Figure 3

This chart allows one to correlate peak-to-peak displacement (inches) vs peak-to-peak velocity-ipsp vs peak acceleration in "g" units vs a known frequency in hertz (cps).



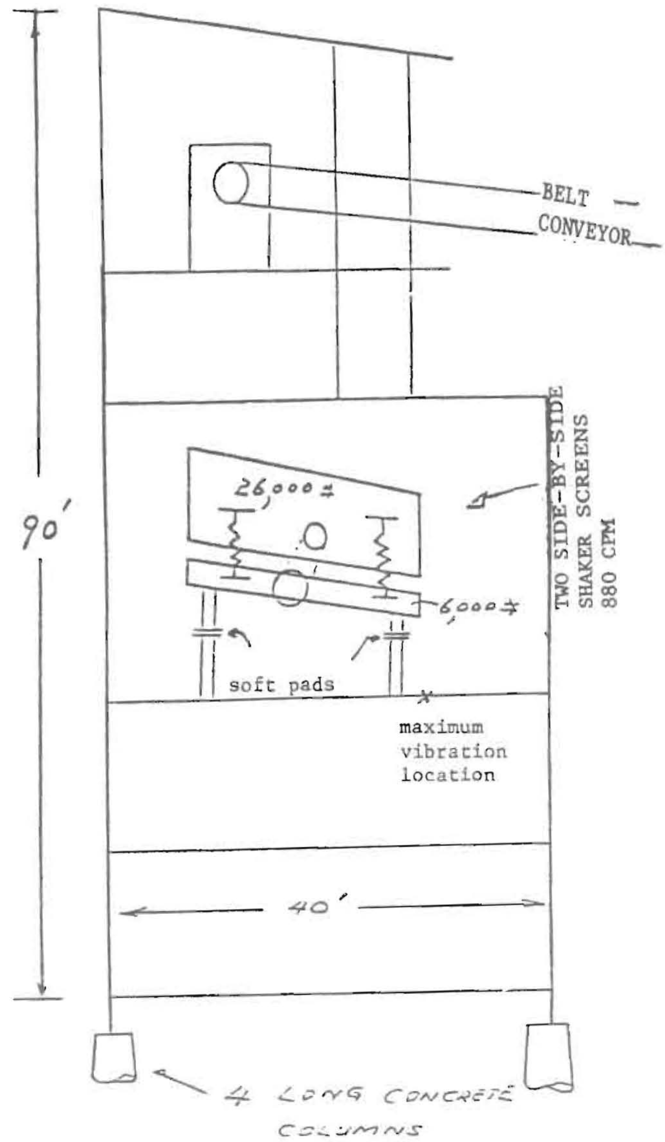
Vibration Unit Conversion Nomograph With Suggested Machine Operating Conditions

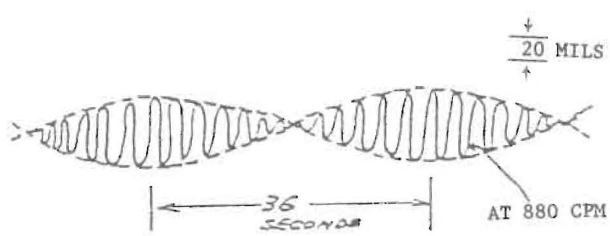
Figure 4



Vibration Signal Frequency Spectra at Different Speeds Showing Instability of a Multi-Stage Compressor (Ref. 15)

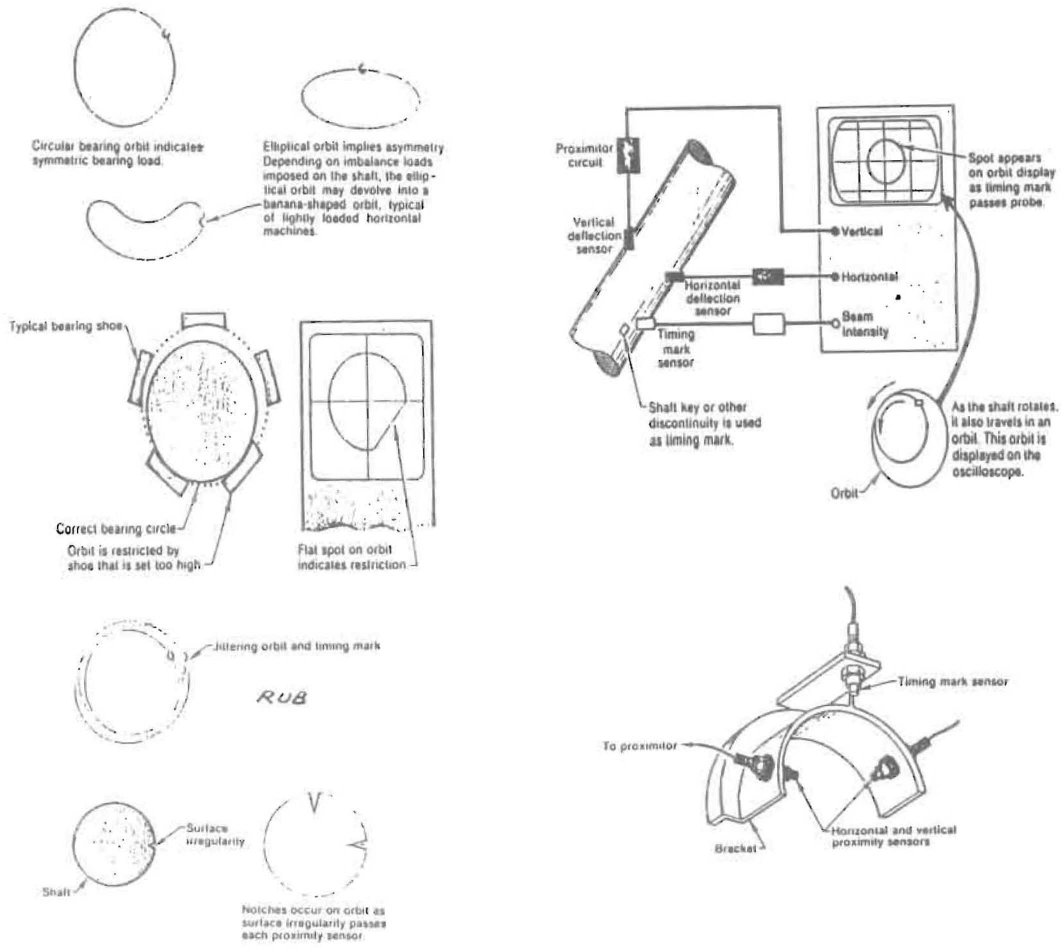
Figure 5





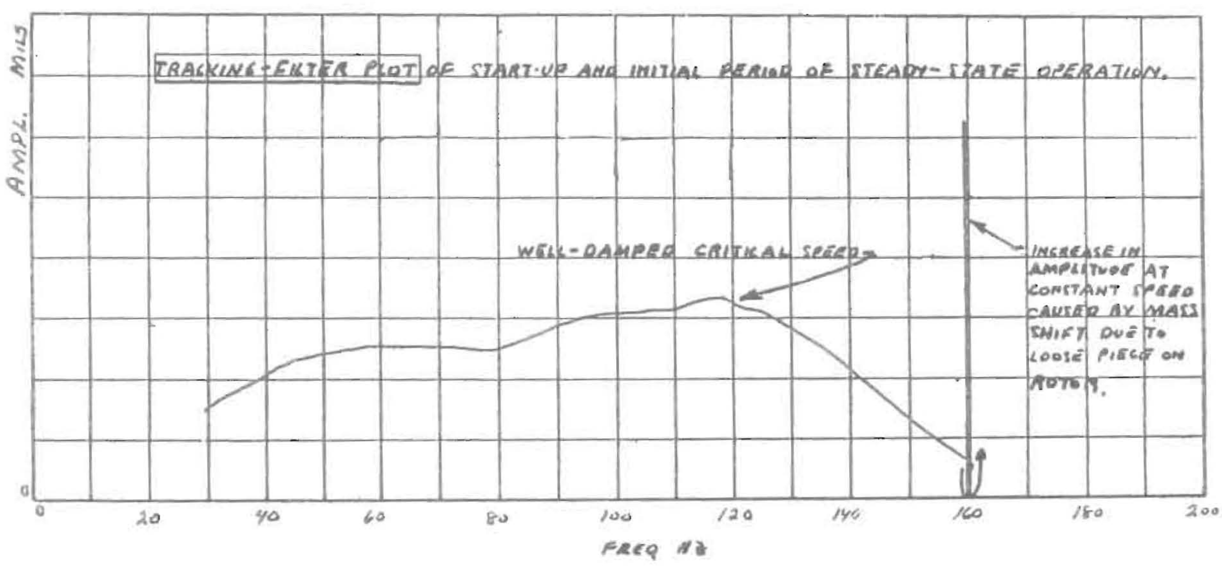
Example of Beating Wave Form-Two Shaker Tower

Figure 6



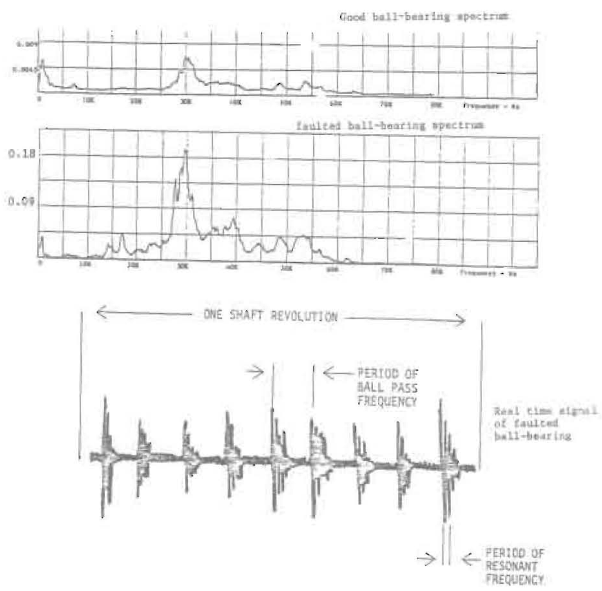
Journal Motion Orbit Examples (Ref. 3)

Figure 7



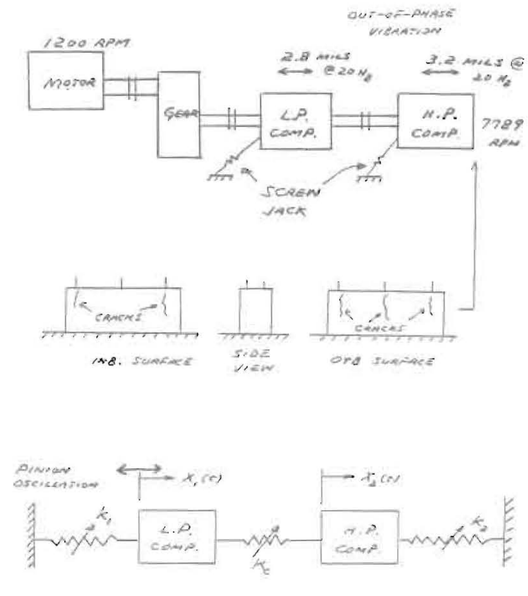
Tracking-Filter Plot (Amplitude of Rotor versus Rotor Speed)
Illustrating Critical Speed Response and loose wheel at full speed.

Figure 8



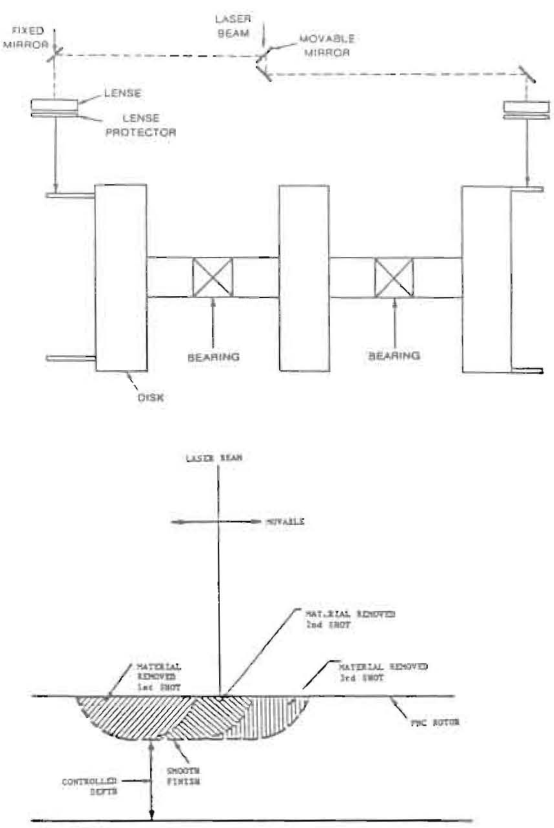
Modulated Resonance Appearing as High Frequency (AM Radio) Wave

Figure 9



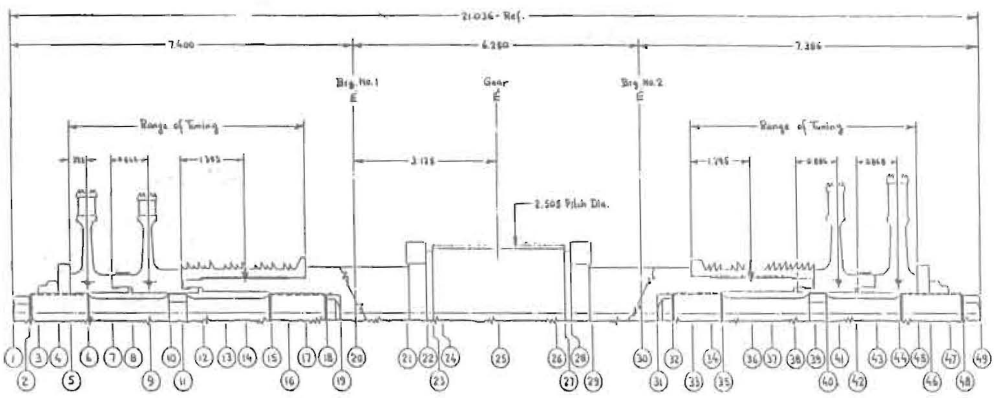
Two Degrees of Freedom Model of Compressor Train With Axial Vibration Problem.

Figure 10



Laser.Balancing Scheme with Micro-Computer for Automated Mass Production Balancing

Figure 11

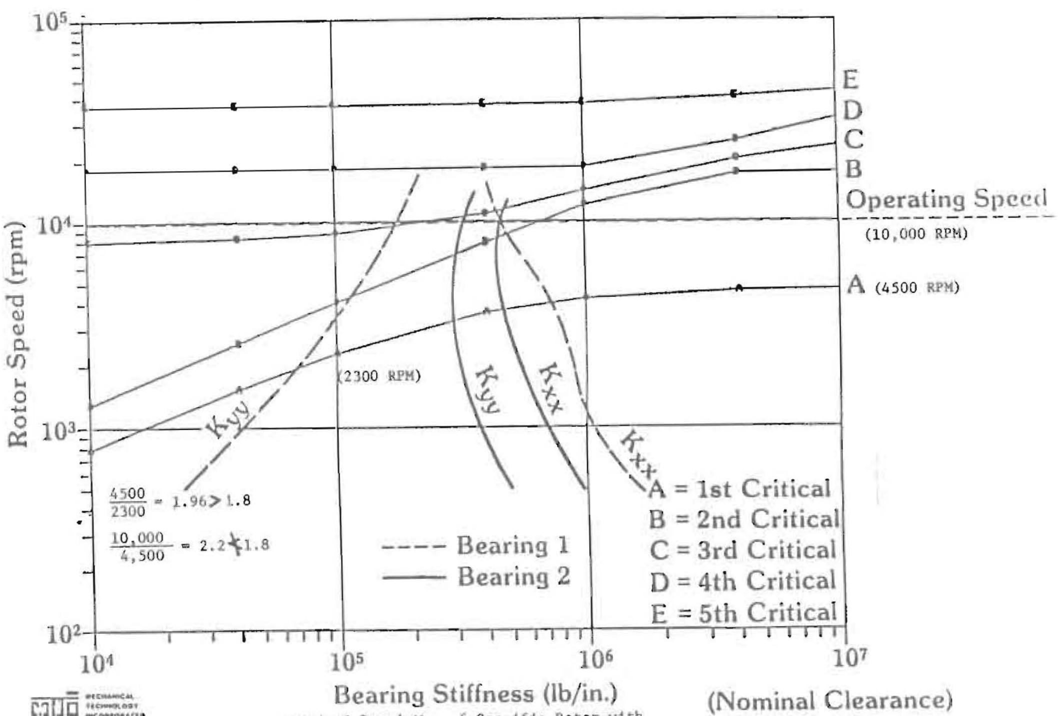


1	2	3	4	5	6	7	42	43	44	45	46	47	48	49	
—	—	—	0.2027	—	1.1719	—	—	—	1.8193	—	0.2027	—	—	—	Concentrated W - lbs.
—	—	—	0.0891	—	2.6406	—	—	—	4.7619	—	0.0091	—	—	—	Concentrated I_p - lb.in.^{2}
—	—	—	0.0499	—	1.3541	—	—	—	2.4393	—	0.0499	—	—	—	Concentrated I_t - lb.in.^{2}
0.375	0.199	0.438	0.250	0.395	0.469	0.430	0.430	0.488	0.474	0.250	0.438	0.277	0.375	—	Distance between Stations - in.
0.350	0.750	0.750	1.125	1.550	1.550	1.490	1.500	1.580	1.580	1.125	0.750	0.750	0.590	—	O.D. for Stiffness - in
—	—	—	0.375	0.750	0.750	0.938	0.937	0.777	0.777	0.375	—	—	—	—	O.D. for Weight - in.
—	—	—	0.375	0.750	0.750	0.938	0.937	0.777	0.777	0.375	—	—	—	—	I.D. for both Stiffness and Weight - in.
1	1	1	1	1	2	2	10	10	11	11	11	11	11	11	Material No. (Ref. to Computer Input)
430					520		280		240						Temperature of the Shaft - °F
17-4PH							17-4PH								Material of the Shaft
26.0					25.4		27.1		27.4						E - $(10)^6$ psi
0.282							0.282								ρ - lbs./in^3
7.46					7.34		7.64		7.67						G_{xy} - $(10)^6$ psi

Cut-away of Mathematical Model of Turbine Rotor
Illustrating Engineering Data and Station Numbering

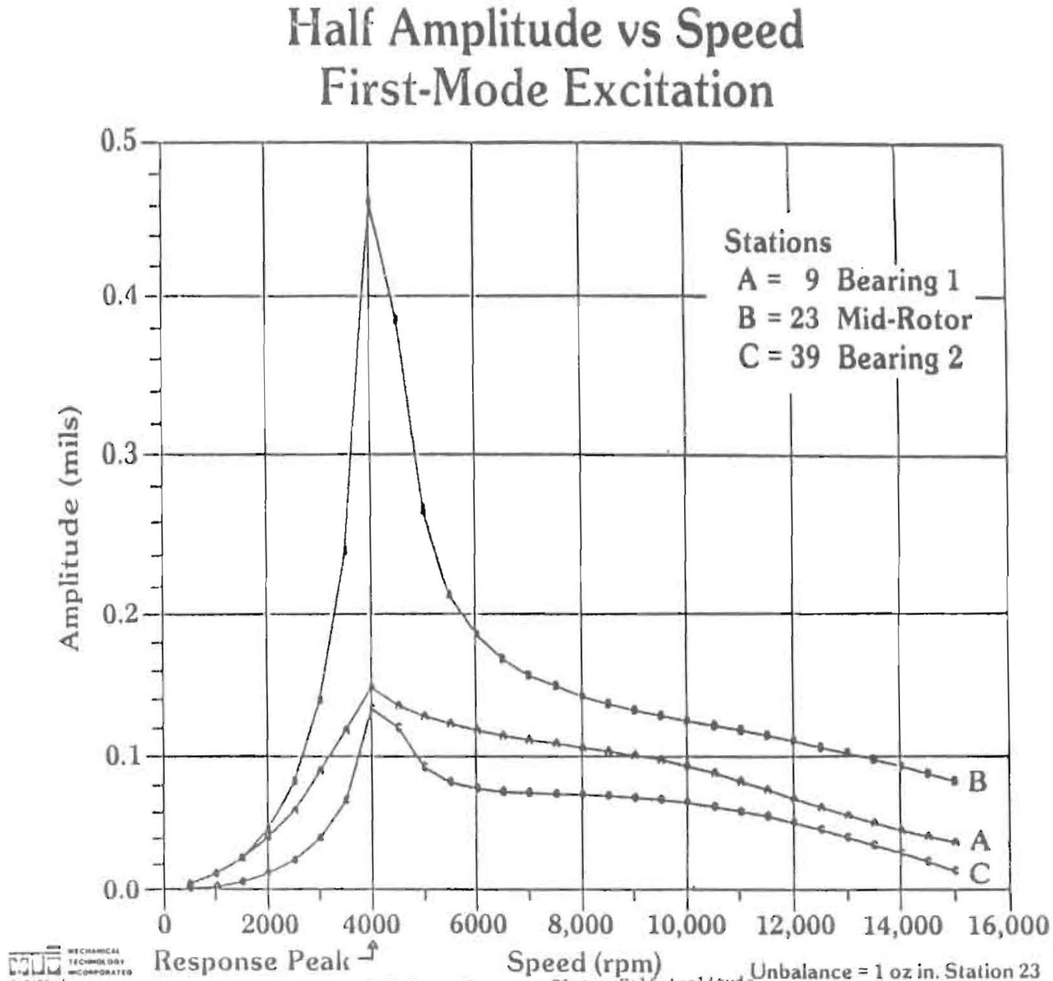
Figure 12

Critical Speed Map with Superimposed Bearing Stiffness



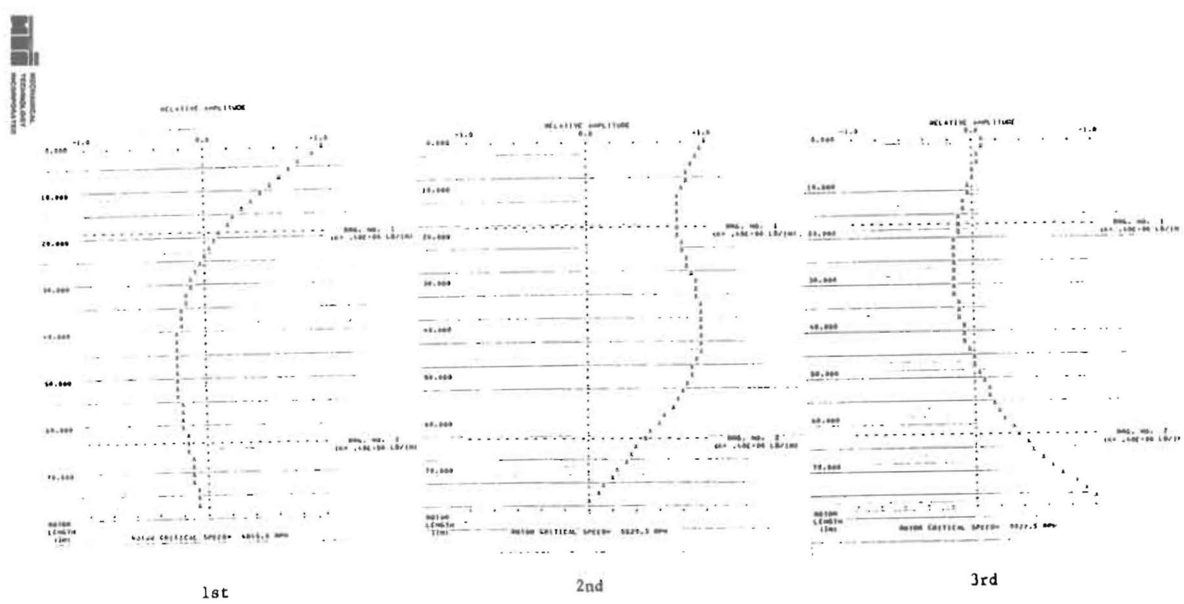
Critical Speed Map of Specific Rotor with
with Superimposed Bearing Stiffness

Figure 13



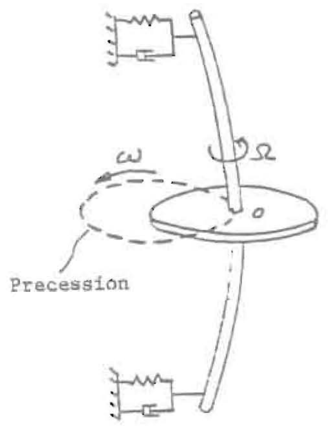
Unbalance Response Plot - Half Amplitude versus Speed with First Model Imbalance Excitation

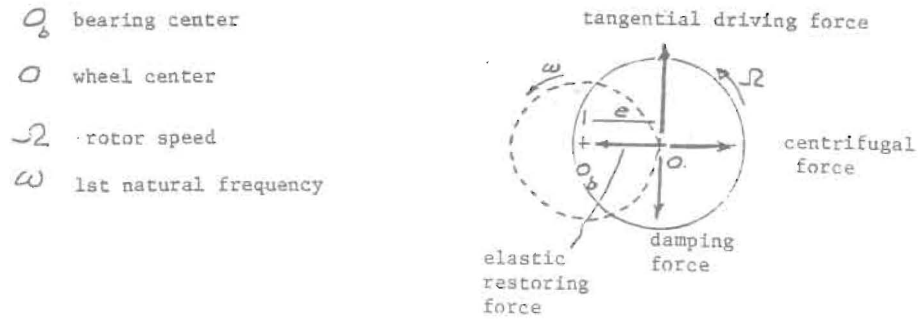
Figure 15



Typical Mode Shapes ($K = 4 \times 10^5$ lb/in.) of a Compressor Rotor

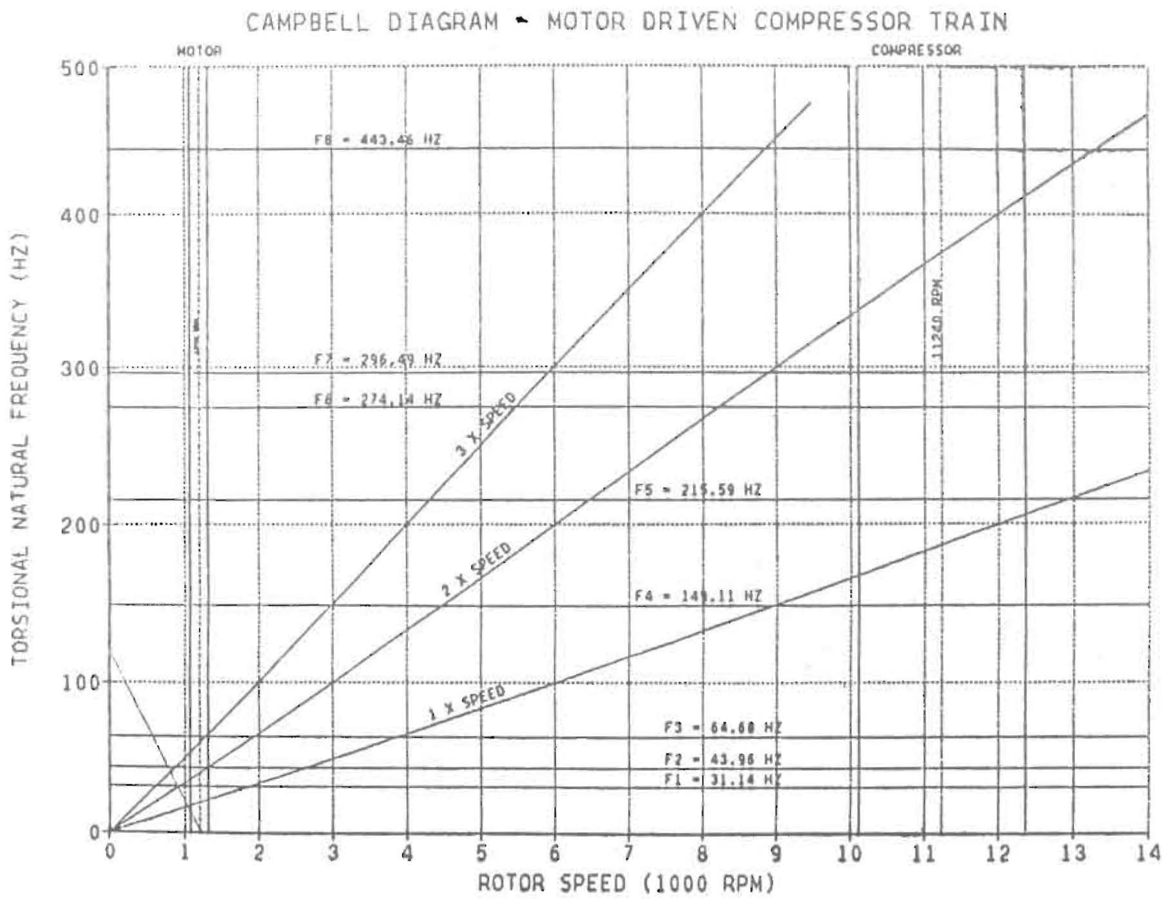
Figure 14





Tangential Driving Forces on Rotor-Cause Large Whirling Orbit with Subsynchronous Frequency

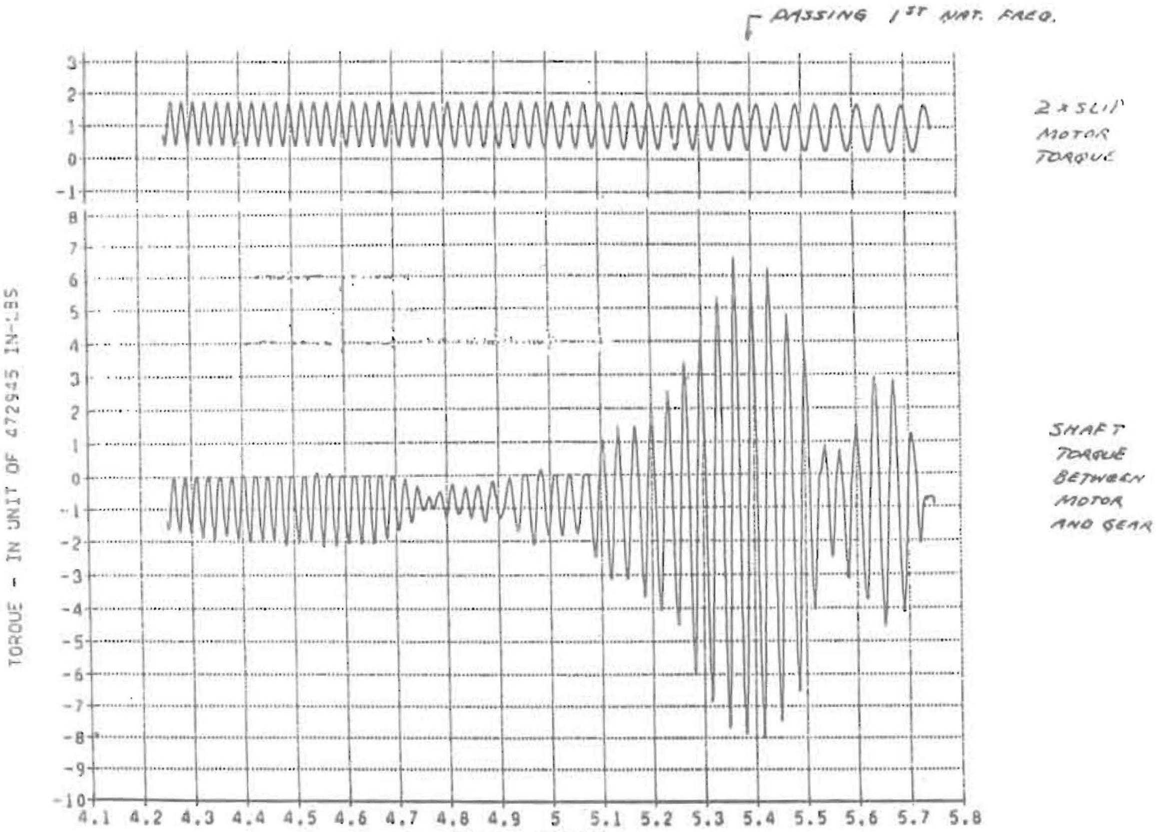
Figure 16



Campbell Diagram: Motor-Compressor Train

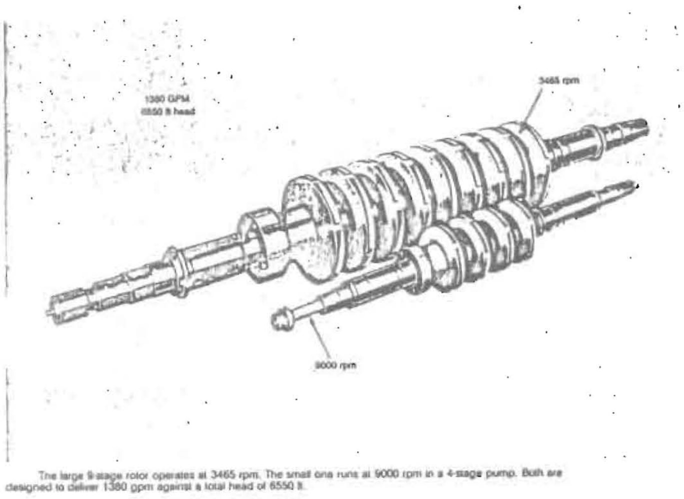
Torsional Analysis

Figura 17



Shaft Dynamic Torque versus Time During Start-up
Cycle Emphasizing Passage through Critical Speed

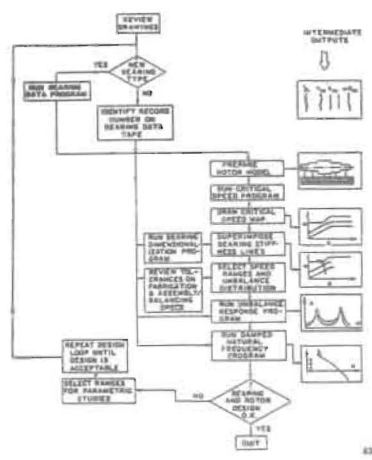
Figure 18



The large 9-stage rotor operates at 3465 rpm. The small one runs at 9000 rpm in a 4-stage pump. Both are designed to deliver 1380 gpm against a total head of 6550 ft.

Pump Designs: The Past and the Future (Ref. 21)

Figure 19



Illustrative Flowchart for Use in
Rotor-Bearing Design (Ref. 18) Audit

Figure 20

VIBRATION IDENTIFICATION TABLE				
Cause	Amplitude	Frequency	Phase	Remarks
Unbalance	Proportional to unbalance. Largest in radial direction	1 x rpm	Single reference mark	Most common cause of vibration
Misalignment couplings or bearings and bent shaft	Large in axial direction 50% or more of radial vibration	1 x rpm usual 2 & 3 x rpm sometimes	Single double or triple	Best found by appearance of large axial vibration. Use dial indicators or other method for positive diagnosis. If sleeve bearing machine and no coupling misalignment balance the rotor
Bad bearings anti-friction type	Unsteady - use velocity measurement if possible	Very high several times rpm	Erratic	Bearing responsible most likely the one nearest point of largest high-frequency vibration
Eccentric journals	Usually not large	1 x rpm	Single mark	If on gears largest vibration in line with gear centers. If on motor or generator vibration disappears when power is turned off. If on pump or blower attempt to balance
Bad gears or gear noise	Low - use velocity measure if possible	Very high gear teeth x rpm	Erratic	
Mechanical looseness		2 x rpm	Two reference marks Slightly erratic	Usually accompanied by unbalance and/or misalignment
Bad drive belts	Erratic or pulsing	1, 2, 3 & 4 x rpm of belts	One or two depending on frequency. Usually unsteady	Strob light best tool to freeze faulty belt
Electrical	Disappears when power is turned off	1 x rpm or 1 or 2 x synchronous frequency	Single or rotating double mark	If vibration amplitude drops off instantly when power is turned off cause is electrical
Aerodynamic hydraulic forces		1 x rpm or number of blades on fan or impeller x rpm		Rare as a cause of trouble except in cases of resonance
Reciprocating forces		1, 2 & higher orders x rpm		Inherent in reciprocating machines can only be reduced by design changes or isolation

Vibration Identification Table (Ref.22)

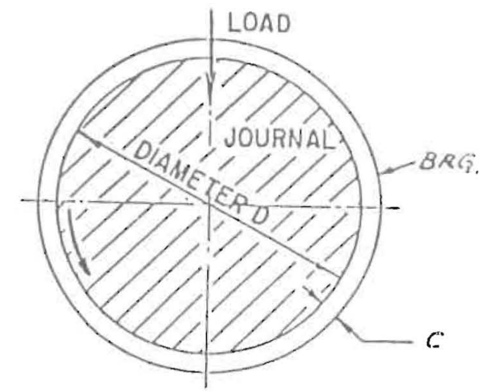
Table 1

STEADY STATE DATA, NORMALIZATION INDICATED

INVERSE SOMM NO. (1/S)	EB* COS(PHIB)	EB- BEARING ECC. RATIO	PHIB-ATTIT. ANGLE(DEG.)	FRICT. COEF. R*F/(S*C*W)	CIRC FLOW Q-X/(NDLC)	SIDE FLOW Q-Z/(NDLC)	TOTAL FLOW Q-T/(NDLC)
.210	.0100	.0909	83.68	19.83	1.710	.273	1.706
.301	.0200	.1282	81.03	19.92	1.765	.385	1.761
.503	.0500	.2036	75.78	20.21	1.878	.612	1.873
.780	.1000	.2896	69.80	20.73	2.015	.870	2.000
1.359	.2000	.4148	61.17	21.94	2.195	1.251	2.187
2.080	.3000	.5127	54.18	23.42	2.354	1.541	2.328
3.123	.4000	.5989	48.09	25.34	2.475	1.804	2.452
4.745	.5000	.6760	42.30	27.87	2.587	2.038	2.569
7.499	.6000	.7502	36.89	31.54	2.694	2.270	2.688
12.857	.7000	.8189	31.26	37.12	2.796	2.482	2.787
25.619	.8000	.8832	25.07	46.88	2.840	2.677	2.879
75.536	.9000	.9447	17.70	71.03	2.963	2.876	2.974
208.900	.9500	.9755	13.13	112.83	3.063	2.970	6.033

EIGHT STIFFNESS AND DAMPING COEFFICIENTS, NORMALIZED WITH RESPECT TO C/(S*W)

(1/S)	KXX	KXY	KYX	KYY	WBXX	WBXY	WBYX	WBYY
.210	.269	2.369	-2.228	.503	4.700	.492	.499	4.509
.301	.343	2.480	-2.343	.740	4.882	.808	.755	4.782
.503	.733	2.804	-2.143	1.174	5.401	1.195	1.153	4.550
.780	1.265	3.423	-2.157	1.804	6.405	1.799	1.806	4.908
1.359	2.928	5.068	-1.760	2.865	9.030	2.817	2.737	4.974
2.080	5.483	7.590	-1.730	4.430	13.234	4.474	4.513	6.254
3.123	10.601	11.453	-1.013	6.262	19.497	6.108	6.232	6.944
4.745	20.735	18.246	.488	8.836	30.566	8.826	8.699	7.914
7.499	41.870	31.378	2.341	14.185	53.248	14.976	15.132	11.357
12.857	99.495	59.315	8.975	23.234	100.570	24.322	24.720	15.009
25.619	301.330	141.530	30.635	46.005	242.080	49.189	49.983	23.384
75.536	1803.500	552.130	159.470	122.090	960.180	138.480	139.030	44.342
209.900	10366.585	2132.453	828.638	343.304	3220.696	311.309	281.106	65.550



PLAIN CYLINDRICAL BEARING

RECORD NO. 2

L/D	.500
PAD ARC LENGTH, DEG.	360.000
REYNOLDS NO.	0.000
LEADING EDGE ANGLE, DEG.	0.000

Typical Tabulation of Dimensionless
Data for a Fluid Film Bearing

Table 2