Rotor Dynamics

The primary factor to assure long-term reliability of any rotating machinery is a good understanding of the rotor dynamics of that equipment. While there are many facets to rotor dynamics, the first and primary concern is good balance.

ROTOR BALANCING

The purpose of balancing rotors is to improve the mass distribution of the rotor and its components (caused by machining tolerances and non-uniform structure) so the mass centerline of the rotating parts will be in line with the centerline of the journals. To accomplish this, it is necessary to reduce the unbalanced forces in the rotor by altering the mass distribution. The process of adding or subtracting weight to obtain proper distribution is called balancing.

Correction of unbalance in axial planes along a rotor, other than those in which the unbalance occurs, may induce vibration at speeds other than the speed at which the rotor was originally balanced. For this reason, balancing at the design operating speed is most desirable in high speed turbomachinery.

To help understand the effects of altering the mass distribution of a compressor rotor, it is important to categorize rotors into three basic groups:

1. Stiff shaft rotors-Rotors that operate at speeds well below the first lateral critical speed. These rotors can be balanced at low speeds in two (2) correction planes and will retain the quality of balance when operating at service speed.
2. Quasi-flexible rotors-Rotors that operate at speeds above the first lateral critical, speed, but below the higher lateral critical speeds. In these cases, modal shapes or modal components of unbalance must be taken into consideration, as well as the static and couple unbalance (Fig. 1). Low speed balancing is still possible due to balancing techniques which correct the static and couple unbalance and sufficiently reduce the residual modal unbalance to retain the quality of balance when run at service speeds. Most multi-stage compressor rotors fit in this category.
3. Very flexible rotors-Rotors that operate at speeds above two or more major lateral critical speeds. Due to their flexibility, several changes in modal shape occur as speed is increased to the operating range. These rotors require high speed balancing techniques utilizing numerous balance planes to make the necessary weight distribution correction.

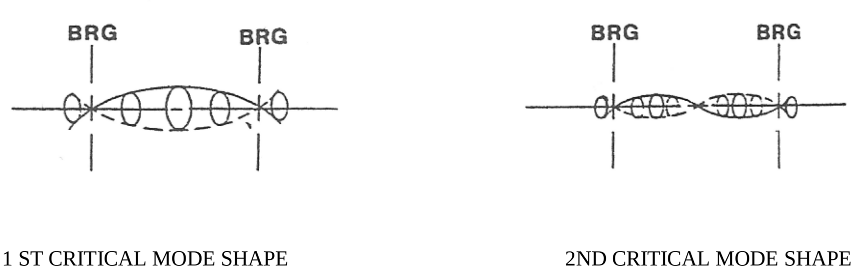


FIGURE 1. Rotor lateral critical speed mode shapes: left, 1st critical mode shape, right, 2nd critical mode shape.

Two-Plane Balancing

The completed rotor is placed in a balancing machine on bearing pedestals. The amount of unbalance is determined and corrections made by adding or removing weight from two predesignated balancing planes. These two planes are usually near the 1/4 span of the rotor.

Following the final corrections, a residual unbalance check is performed to verify that the residual unbalance is within the allowable tolerance.

Maximum allowable residual unbalance guidelines:

$$\text{oz.} - \text{in.} = \frac{4 \times (\text{rotor weight})}{\text{rotor speed}}$$

$$\text{oz.} - \text{in.} = \frac{56{,}347 \times (\text{journal static loading})}{N^2\ (\text{max. continuous rpm})}$$

Three-Plane Balancing

For low speed balancing of quasi-flexible rotors, major components (shafts, wheels, etc.) are individually balanced using two-plane balancing. For clarity, static (or force) and dynamic (or moment) balancing will be referred to as single-plane and two-plane balancing, respectively.

The rotor is completely assembled using pre-balanced components. Using the two-plane technique, the amount of unbalance in the two outer planes (wheels) is determined. This is resolved into a force component and a moment. The single-plane (force) correction is made as near the center of gravity of the rotor as possible. The residual unbalance moment is then corrected in two planes through the end wheels, which are usually situated near the one-quarter point of the rotor span between bearings. Following the final unbalance corrections, a residual unbalance check is performed to verify that the remaining unbalance is within tolerance.

The following items will help to ensure a satisfactory balance:

1. All components (wheels, impellers, balance piston) are individually balanced prior to assembly on the balanced shafts.
2. Mechanical ronout is checked and recorded prior to balancing.
3. Combined electrical and mechanical ronout checks are performed and recorded.

HIGH SPEED BALANCE

Axial unbalance distribution along any rotor is likely to be random in nature. Local unbalances can result from shrink fitting discs, impellers, sleeves, etc., along with residual unbalances present in all component parts. The vector sum of the unbalance distribution is compensated for in completely assembled rotors by balancing the assembly in a low speed balancing machine. The low speed machine measures either bearing displacement or bearing forces and provides information for correction in two or three transverse radial planes. The low speed machine can measure only the sum of the unbalances; therefore, the individual unbalances can excite the various flexural modes when the rotor is accelerated to operating speed. Additionally, when a rotor is operated at its maximum continuous speed or trip speed, forces acting on the various component parts, along with temperature changes, will alter the distributed unbalances. Rotors are processed through the high speed facility to measure vibration amplitudes and display mode shapes with the intention of minimizing these deflections.

Setup and Operation

Prior to operation, the rotor is placed in isotropic supports which are contained in a vacuum chamber (Fig. 2). The vacuum chamber is sealed and evacuated to 5 to 7 Torr. The rotor is accelerated slowly to maximum continuous speed while monitoring vibration levels and critical speeds.

Vibration tolerances are formulated to limit alternating forces at the bearings to 10% of static load (0.lg).

During the first run, if vibration levels do not exceed limits, the rotor is accelerated to trip speed or rated overspeed and held there for a minimum of 15 minutes. This process "seasons" the rotor by stabilizing the rotor temperature and at the same time seats component parts. After overspeed, the rotor is brought to rest and again accelerated to operating speed while vibration levels are recorded. The procedure is repeated until repetitive vibration levels are observed, and the balancing process is started. (If the rotor is within tolerance at this point, a plot of vibration levels through the entire test range is made and the test is terminated.)

When repetitive indications are available from the measuring instrumentation, they are stored in thy computer memory and considered to be the reference condition or initial unbalance. Test weights are now fabricated and one test weight is added in one correction plane. The rotor is accelerated to maximum continuous speed and unbalance response at various speeds is recorded. These test runs are repeated with one test weight in each correction plane.

When the test runs are completed, a correction weight set is determined. Calculations are conducted by the computer via the influence coefficient method. The calculated weight set is then applied to the rotor and additional measuring runs are made to check the results of the correction process. The sequence is repeated until the tolerance level is reached.

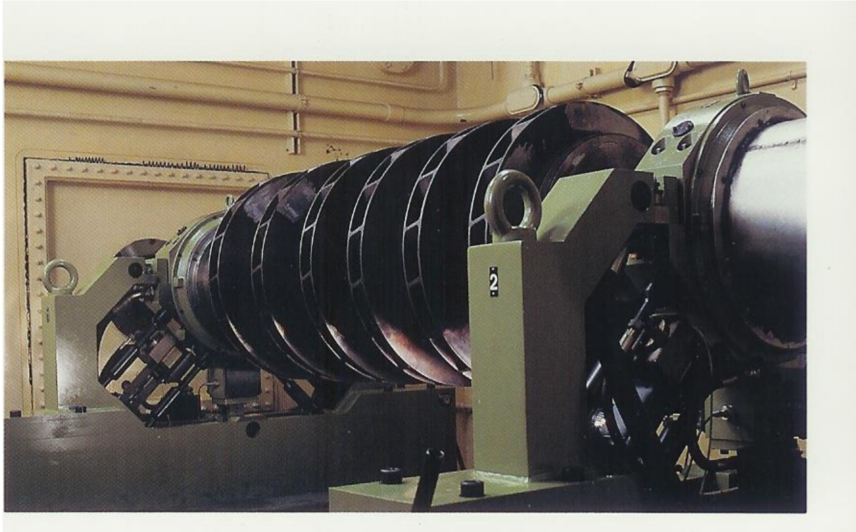
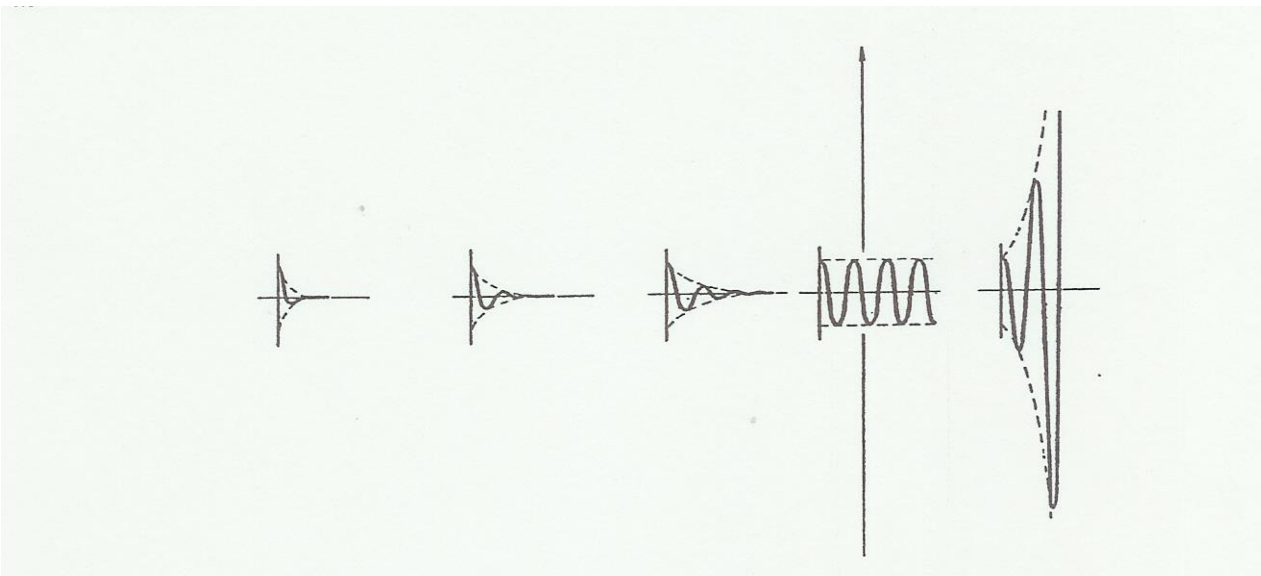


FIGURE 2. Compressor rotor in at-speed balancing machine. The rotor is balanced at operating speed in a vacuum chamber.



VERY STABLE STABLE NEUTRAL UNSTABLE

FIGURE 3 The natural response of stable and unstable systems.

ROTOR STABILITY

One of the most important problems affecting the operation of high speed turbomachinery is stability of rotor motion. The susceptibility of a rotor to self excite can mean the difference between a smooth running piece of equipment and virtual self destruction.

Stability

The stability of a vibratory system is determined by observing the motion of the system after giving it a small perturbation about an equilibrium position. If this motion dies out with time and the system returns to its original position, the system is said to be stable; on the other hand if this motion grows with time, it is said to be unstable (see Fig. 3).

Idealized Damped System

In order to understand rotor instability, it is first necessary to understand a simple idealized damped system. This is normally represented by a mass (M) supported by a spring *(K)*, with a dash pot *(b)* to damp the system motion (see Fig. 4).

In this example, the system is stable. Both the spring force and the damping force tend to return the system to the original position once the system is disturbed by an external force. The slope of the damping curve is positive. Increasing velocity generates an increasing force which opposes motion. Likewise, the spring force increases and opposes increased displacement from the equilibrium point.

In an unstable system, just the opposite occurs. Negative spring forces and/or negative damping tend to increase velocity and displacement with time.

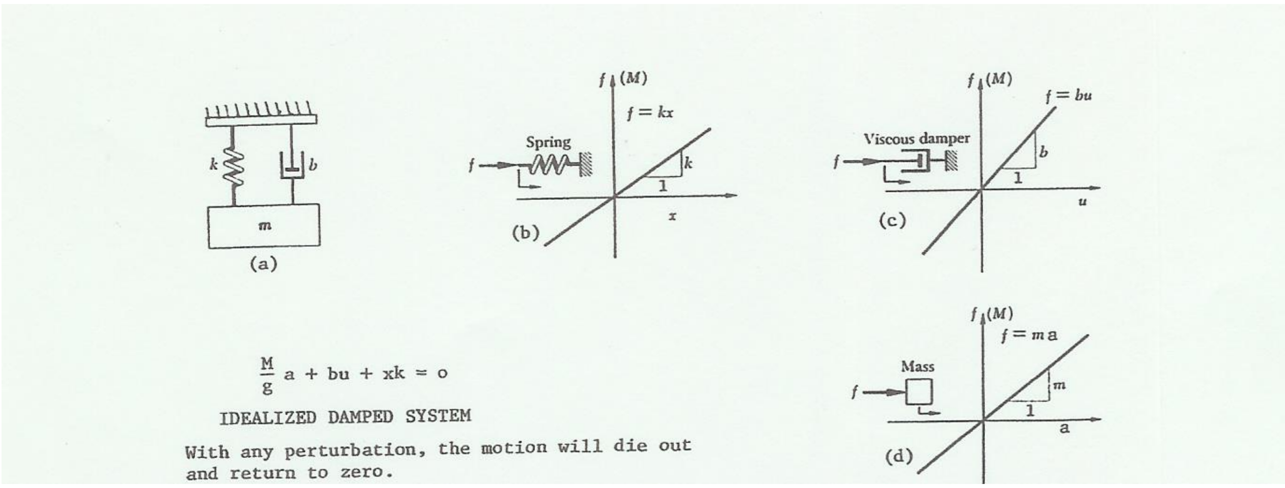


FIGURE 4 An idealized damped system. With any perturbation, the motion will die out and return to zero. [1]

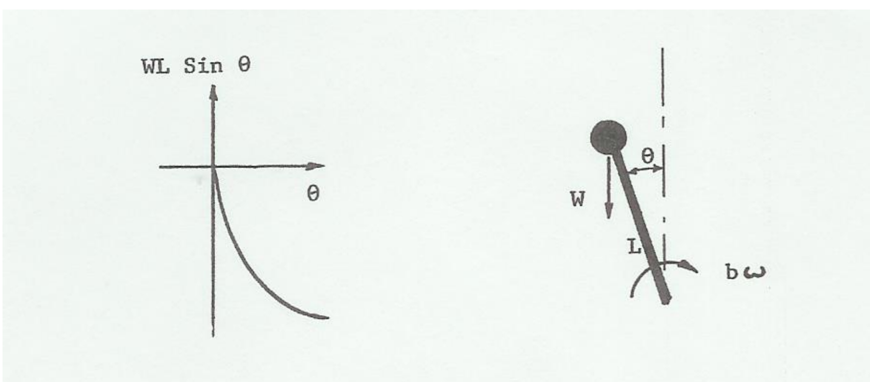


FIGURE 5 An inverted pendulum. The "negative spring" force increases with increased. angle *cp.* Time behavior is divergent and unstable. [1]

Negative Spring

A good example of a "negative spring" is an inverted pendulum or a valve in the near closed position (Figs. 5 and 6). It is easy to visualize what happens to the balanced inverted pendulum once disturbed. A small initial disturbance will "push it over the hill." Due to the configuration of the system, the force of gravity will overcome any damping forces in the system and the pendulum will go to and remain at a fully extended position.

Similar to this are the problems of valve chatter. The example in Fig. 6 shows a model of a fuel valve. When near the fully open position, the net spring constant is positive, and the system is stable. However, when near closed, the pressure forces exceed the spring forces and

the net spring force is negative, causing the valve to become unstable.

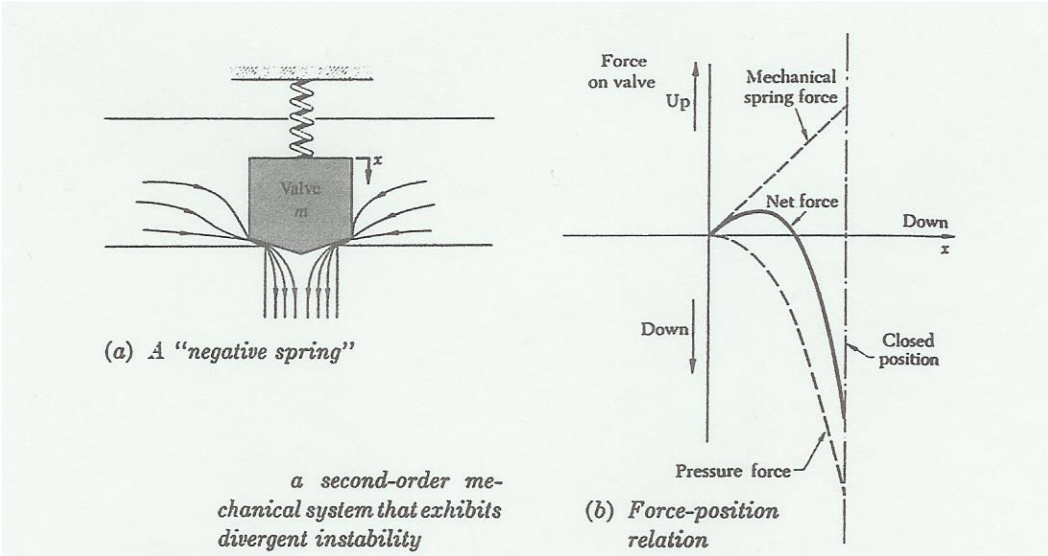


FIGURE 6 A valve in a near closed position is a second-order mechanical system that is unstable. [1]

Negative Damping

Negative damping is much more common in familiar physical systems than many may realize. A good way to understand what negative damping is, is to simply look at it as the opposite of positive damping. Damping, as usually thought of in a positive value, is represented as a dash pot or shock absorber in an automotive suspension system. The dash pot or shock absorber generate a force that is a function of velocity (see Fig. *4c).*

$$F = bV$$

Since the force generated is increasing with velocity and opposes the velocity, the dash pot tends to reduce oscillation of the system. This is "positive damping."

Now consider a system where the damping force generated decreases with an increase in velocity.

$$F = -bV$$

In this situation, the force will tend to increase the oscillation of the system. This is called "Negative Damping."

Examples of instability due to negative damping include the dry-friction vibration produced while playing a violin or the tool chatter produced in a machining operation (Fig. *7b).* In rotating machinery, a labyrinth seal rub, or friction between elements on a rotor, can produce similar results.

These dry-friction systems are inherently unstable because the effective system damping constant is negative. An increase in velocity leads to a decrease in the friction force opposing that velocity.

The "galloping transmission line" problem is also a good representation of negative damping (Fig. *7a).* Aerodynamic lift forces in the direction of velocity lead to unstable oscillations.

Vibrations which are sustained by the energy from a steady or non-oscillating force are known as "self-excited oscillations." The unstable, growing motion of each system described above can occur only because energy is being supplied to the system from an external source. The energy to sustain vibration of the violin string is supplied through the bow by the musician's arm.

Rotor Stability

During stable motion, a rotor assumes a deflected shape dependent upon the rotor elastic properties and the forces exerted upon it by the bearings, seals, aerodynamics, and unbalance. The deflection is primarily induced by a mass unbalance distribution along the rotor. The deflected rotor then whirls about the axis of zero deflection at a speed equal in direction and magnitude to the rotational or operating speed of the machine, and the whirl motion is therefore termed synchronous. Although the actual path described by each point in the rotor need not be circular, the size of the orbit remains constant in time (assuming the speed and unbalance distribution remain

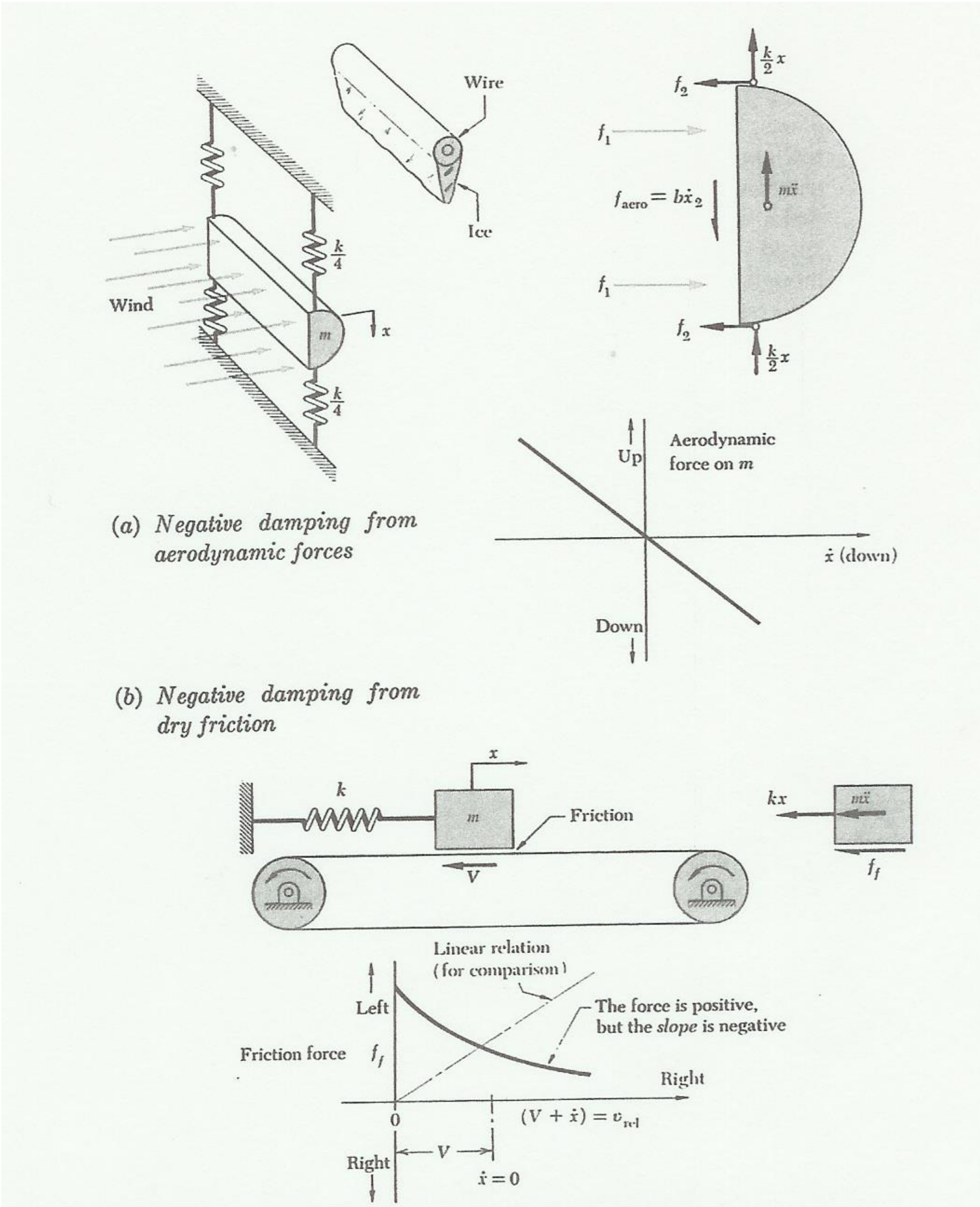


FIGURE 7 Two second-order mechanical systems that exhibit oscillatory instability. *a)* The "galloping transmission line" problem is a result of aerodynamic lift forces in the direction of velocity that lead to unstable conditions. *b)* Dry-friction vibration produced while playing a violin is an example of negative damping. [1]

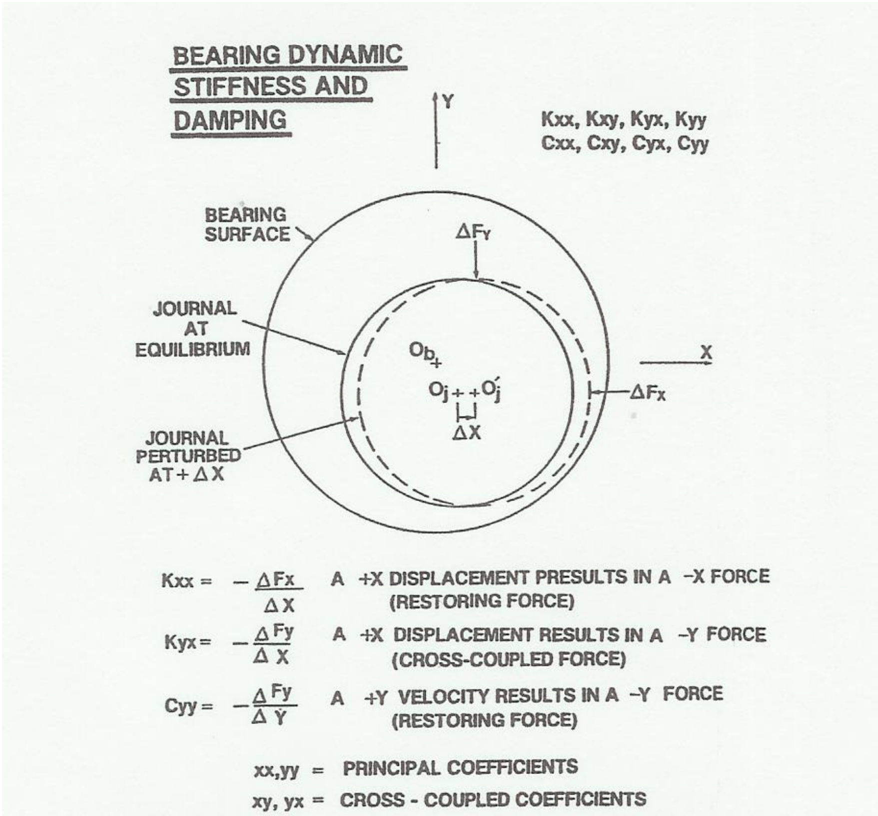


FIGURE 8 Bearing stiffness and damping. If the amount of cross coupling is large enough so that the cross coupled forces exceed the damping forces, the rotor will be unstable. [2]

constant). When the motion is unstable, however, the size of the orbits described by each rotor point increases with time and the whirl motion of the rotor, while generally in the direction of the operating rotation, is at a lower frequency. Such whirl motion is generally termed subsynchronous. With very few exceptions, the whirl frequency is half of the operating speed when the rotor speed is less than two times the first critical speed. At and above two times the first critical, the whirl frequency locks onto the first critical frequency.

The operating speed at which the motion becomes unstable is dependent on the type of instability mechanism and its magnitude, but is generally above the first critical speed of the rotor. This speed is generally termed the *instability onset* or *threshold speed.* Although the rotor whirl orbits generally increase only to a certain size because of non-linearities in the system, they can become quite large; and in many instances it is not possible to operate the rotor much faster than the instability threshold speed, and in some instances actual rotor failure may occur.

Instability Mechanisms

The significant mechanisms which cause rotor instability can generally be found in one or more of the following categories:

1. Hydrodynamic bearings
2. Seals
3. Aerodynamic effects
4. Rotor internal friction

The first three categories are usually those encountered in practice, although internal friction damping is generally present in all machines to some degree and reduces the overall stability of the machine. In certain instances, such as with unrelieved shrink fits or unlubricated splines, internal rotor friction may become a significant source of rotor instability.

Although the mechanisms causing instability are varied, they all have certain characteristics in common, the most important of which is a cross coupling effect of the forces (see Fig. 8). The cross coupling can exist in both velocity (damping) and displacement (stiffness), but displacement cross coupling is the more important of the two, and is characterized by a force due to a displacement of the rotor acting at a right angle to the displacement vector. The cross coupled forces act in a direction such that they oppose the damping forces; and if the amount of cross coupling is large enough so that the cross coupled forces exceed the damping forces, the rotor will become unstable. In effect, the cross coupling forces create negative spring and damping forces. [2]

Susceptible Designs

As discussed in the previous section, mechanisms for instability are: bearings, seals, aerodynamics, and rotor internal friction. Designs most susceptible to instability will have several or all of the above mechanisms at work.

The cross coupling forces in hydrodynamic liner bearings are relatively low on low speed, heavily loaded bearings. On high speed, lightly loaded bearings, the cross coupling forces are relatively high. Similar statements can be made concerning bushing type seals. However, the difference with bushing seals is that they are mounted so as to "float" with the shaft. Actual forces transmitted via the floating bushing seal if properly balanced should be negligible.

Cross coupling forces occur, not only in bearings and seals, but also anywhere along the rotor where there is relatively close proximity between rotating and stationary parts. Cross coupling forces occur in these areas similar to those in a hydrodynamic liner bearing: between impellers and diaphragms, shaft and labyrinth seals-balance piston seal, shaft seal, and impeller eye seal.

Aerodynamic cross coupling forces are most significant for high pressure, high speed (tip speed), close clearances, non concentric, mid span areas. The most pronounced example of these features is the center seal (balance drum) on a back-to-back design compressor, since this design has all of the above features. The center seal has a very close clearance to minimize leakage from one section to the next. The seal is of significant length (large L/D) to further minimize leakage, and concentricity is generally poor due to rotor sag. A regular balance piston seal has the same effect, but to a lesser degree due to it's location at one end of the rotor away from maximum deflection point for the first critical speed (see Fig. 1).

Most high speed turbomachinery rotors are made up of several parts that are somehow fitted together. These fits can be slip fits, bolted or riveted arrangements, shrink fitted, or even welded joints. Any of these joints that have any relative movement during operation can generate destabilizing forces or "negative damping."

Analytical Analysis

The analysis of rotor bearing systems for stability reveals a large amount of useful information on the behavior of the system. It answers the question as to whether or not the system is stable at the operating speed for which the analysis was performed.

The rate of decay of a damped vibrating system takes the form of the natural log of the ratio of successive peak amplitudes (Fig. 9) and is therefore called the logarithmic decrement *(a)* and is used as a measure of the rotor damping. If the log dec value for the first forward even mode at the first critical frequency of the rotor is positive, the rotor is stable. If the log dec value is negative, the rotor is unstable and whirling of the rotor at that frequency will occur.

As with any analytical method, some assumptions are necessary; and the model does not quite represent the actual situation. For this reason, a safety factor is applied.

It is very important to realize that there is a significant difference in rotor dynamics programs used.

Some stability programs use damping and stiffness parameters which are frequency depending. That is, the stiffness and damping parameters used are based upon individual pad data and varies with the particular frequency being evaluated and the synchronous speed. Other stability programs use stiffness and damping parameters at synchronous speed or are synchronous dependent.

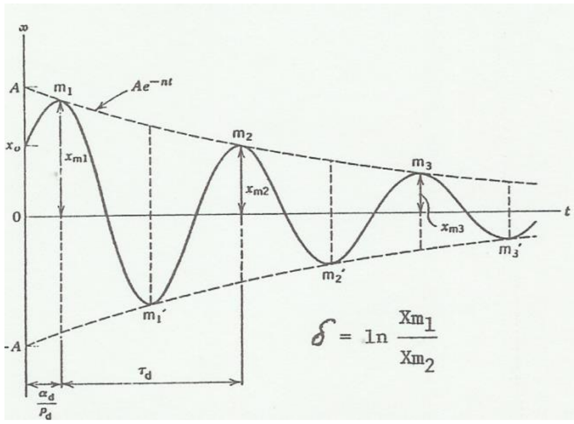


FIGURE 9 Logarithmic decrement (log dec.). The time decay of a viscous damped system is a natural log function. [3]

Because of the differences in the various rotor dynamics programs, one must be very careful in comparing the results of one program to another. Log decrements are widely being used as a measure of rotor damping. A particular log decrement value is dependent upon the stability program in use. For example, a log decrement of + 0.05 may be considered as having adequate stability margin utilizing one program, while another may require a log decrement of +0.4.

It is also most important when discussing log decrement values as to whether the value being referenced is a basic log decrement without aero excitation or is an adjusted log decrement value, which includes aero cross coupling effects.

Identifying Instability

As noted previously, rotor instability shows up as just under half running speed frequency when operating below 2x the first critical speed. When operating above 2x the first critical speed, the instability locks onto the first critical speed (see Fig. 10). Operating vibration levels during unstable operation can be relatively low or so high that the unit is inoperable. Energy levels can be so high during instability that the unit can self destruct in a few seconds of operation. The energy levels can be so low that the only problem is that bearing life is reduced.

The equipment needed to analyze the rotor is a vibration pickup and a frequency analyzer. Start-up and shutdown response can be recorded to obtain actual NC1 and the point at which instability first begins.

A non-contacting probe is preferred, since it records actual rotor movement. Casing velocity or acceleration measurements can introduce questionable data due to foundations or piping vibration.

API limits on subsynchronous vibration is 1/4 of running speed amplitude in the operating speed range. Accordingly, if levels are less than this, operation is satisfactory and corrective procedures are not necessary. If the subsynchronous level is greater than 1/4 of RSY amplitude, then corrective measures should be taken.

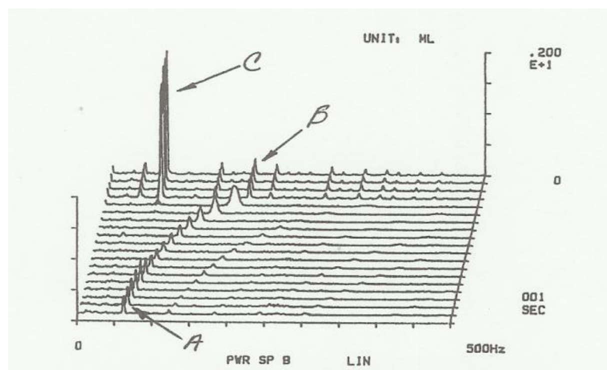


FIGURE 10 Waterfall spectra of compressor shutdown showing aerodynamic instability. *u)* High response at running speed indicated rotor first critical speed. *b)* Synchronous amplitude at normal operating speed. *c)* Subsynchronous amplitude at normal operating speed. Note how the subsynchronous frequency (c) aligns with the first critical speed frequency (a).

Corrective Measures

Whether instability of a rotor system is discovered by operation or by analysis, the best tool for correcting the situation is a rotor stability analysis program. Possible corrective measures can then be analyzed and the best possible solution can be selected. Some methods of improving stability are listed below. Please note these are general comments and are not always true for every case. One change may help one case while of little benefit to another. Each situation must be separately studied.

A. Labyrinth seals

1. Buffer

Supplying buffer to a labyrinth seal introduces to the seal a gas flow that is absent of any tangential velocity. This reduces the average tangential velocity of the gas through the seal, thereby reducing the cross coupling forces (which are function of the tangential velocity). See Figs. 11 and 12.

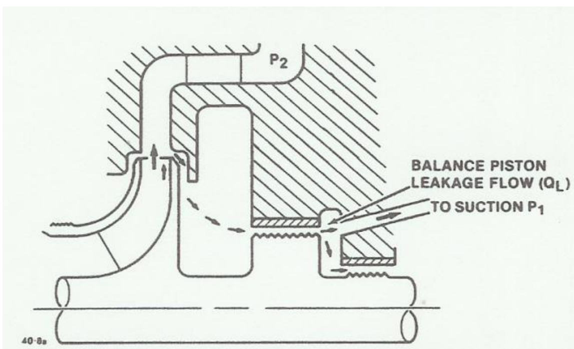


FIGURE 11 Conventional balance piston flow. The flow entering the balance piston seal has a circumferential component due to the rotation of the impeller. [4]

2. Dewhirl vanes

Dewhirl vanes (Fig. 13) or straightening vanes upstream of a labyrinth seal provide similar effects to buffer by reducing or eliminating tangential velocity before the gas enters the seal.

3. Honeycomb seals

Seals that have a honeycomb pattern (Fig. 14) on the stationary portion and smooth rotating drum tend to have a reduced average tangential velocity over standard labyrinth seals. This reduced tangential velocity reduces cross coupling forces and therefore delays the onset of instability. Additionally, the cavities of the honeycomb act as dampers contributing to the system stability.

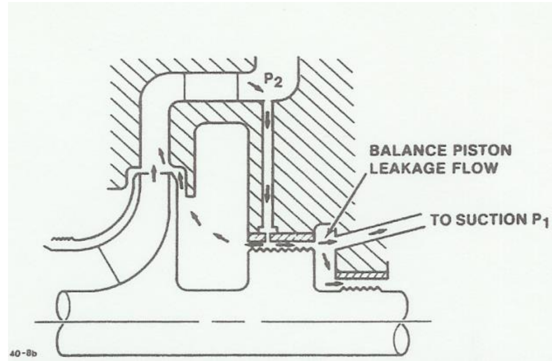


FIGURE 12 Balance piston flow with buffer. The buffer introduces the gas without a tangential velocity, thus reducing the average circumferential velocity in the seal and therefore reducing the crosscoupling forces. [4]

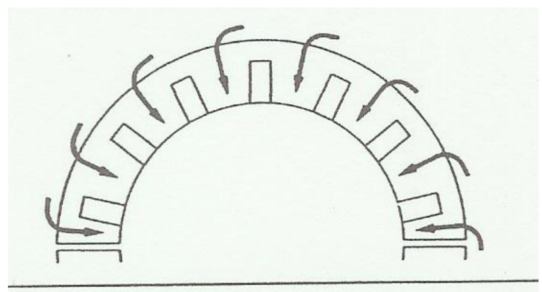


FIGURE 13 Swirl brakes eliminate tangential velocity of gas prior to entering labyrinth seals *(courtesy Sultzer).*

4. TAM seal

The TAM seal is a labyrinth seal with teeth on the stationary member and a smooth rotating drum. Specially designed barriers in the circumferential cavities between the labyrinth teeth break up the tangential swirl of the gas and generate pressure dams with damping properties.

5. Concentricity

Cross coupling forces are related to the non-concentricity of the labyrinth seal. Since cross coupling forces are only fully developed in a non-concentric seal, centering of the shaft in the seal will reduce these forces significantly. This is one reason why a compressor which has a balance piston near one journal bearing may be stable, while a similar compressor with a balance piston in the center of the rotor is unstable (eccentricity due to rotor sag).

6. Gas density
Cross coupling forces are a function of gas density. Higher gas density will give higher cross coupling forces and vice versa. Although it is usually not a practical solution, reducing gas density will improve stability.

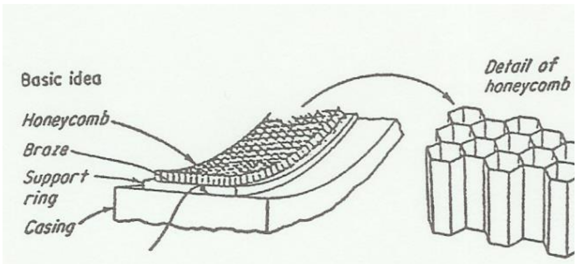


FIGURE 14 Honeycomb seals can add stability to the rotor system.

B. Journal bearings
1. Liner bearings
Liner bearings have inherently high cross coupling forces due to the high density of the oil lubricant, the close clearances, and the usual high offset of the bearing to journal. Several means of altering the liner bearing to reduce or counteract the cross coupling forces are available. These include: dam type bearings, lemon shape, offset bearings, and lobed bearings. Generally the most reliable correction is to change to tilt pad bearings.
2. Tilt pad bearings
Tilt pad bearings by nature have little or no cross coupling forces. Thus, changing from liner to tilt pads can eliminate the instability if the primary cause is the bearing. If, however, the primary source is elsewhere and tilt pads are already in use, the tilt pad bearing can be enhanced. This can be done by several methods: reduced positive preload, increased *LID*, offset pivots, load on pad orientation (for 5 pad bearing), increased clearance. A damped bearing support system can further stabilize the system.
3. Magnetic bearings
Stability or dampening characteristics of this type of rotor bearing system is dependent on the control system. The interaction of the power and feedback circuits will determine stability of the overall system.
4. Sleeve seals
Sleeve type seals or bushing seals have the same characteristics as liner bearings, except that the seals, if properly designed, are "free floating." This means that the seals are concentric and negligible forces are transmitted to the shaft. Wear on the seal face can restrict the seal radial movement and lead to subsynchronous vibration.
C. Rotor design
1. Stiffness
Increasing rotor stiffness improves rotor stability. Anything, therefore, that increases the critical speed will improve stability. This includes reduction of bearing span, increased shaft diameter, reduced overhang, or reduced rotor weight. A common parameter to consider is the ratio of the running speed to the first critical speed. Lower values are preferred.
2. Rotor fits
Any parts that can rub during rotor flexing can cause negative dampening or destabilizing forces. It is therefore prudent to eliminate any source of rubbing between parts such as is caused by loose or slip fit sleeves, marginal shrink fits, or sleeves that touch end to end. Bolted, riveted, or shrunk fits must remain tight at speed so as to eliminate relative motion.

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