CRITICAL AREAS IN CENTRIFUGAL COMPRESSORS

by

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INTRODUCTION

In recent years mechanical equipment has seen extensive uprating in the form of higher speeds, pressures, temperatures and power levels. The advance in the equipment has reflected itself in making economically feasible processes, which in the past could not be justified. At the same time difficulties with various pieces of equipment have resulted in large losses and have created an awareness of the necessity for highly reliable mechanical equipment.

Many organizations including users, manufacturers, and service organizations have participated in programs to minimize the risk and to implement procedures for evaluation.

Over a period of time numerous problem areas have been addressed and considerable case histories accumulated. Frequently solutions have been rather simple ones; whereas in other instances, the problems have required extensive re-design and in some cases, additional development.

DISCUSSION

The key factors which influence reliability can be conveniently placed in 5 categories as follows:

1. Design and material selection
2. Manufacture, quality control and assembly
3. Performance
4. Dynamic behavior
5. Key components and systems

Design

The area of design is a far too extensive one to do other than comment on certain aspects. One of the most important factors in achieving a successful design would appear to be all too obvious, that is assuring that a complete understanding of the operating specification has been transmitted and understood and subsequent to this determining whether this represents a significant departure from the standard line of equipment. This is extremely important if the resulting requirement means extrapolation well beyond demonstrated capability which requires significant departures from the normal design. A second keypoint with respect to a discussion of design is that the design philosophy must be one of a systems type approach. This approach must be extended to each major assembly and subsequently to the entire machine configuration.

This is perhaps best demonstrated by reference to one of the major components in a compressor, that is the rotor assembly itself. Aside from the selection and matching of the aero components, the rotor system must be adequately supported to operate smoothly and stably over the required range. Figure 1 shows a possible procedure wherein the selection of the rotor might be undertaken. An attempt is made to generalize on it. It should be recognized that many more iterations may well be required, however, the important point is that attention is drawn to the various characteristics which must be compatible in order to achieve a successful design.

In reviewing a design, it is well to pay attention to key components, such as bearings and seals, as well as to the impellers themselves. Some guidance may be based on past experience to determine to what degree the design deviates from standard practice. For example, peripheral speed may provide such a barometer with respect to the impellers. A shrouded, riveted impeller with a keyway in the bore normally would have a maximum velocity limit of the order of 800 feet per second. Although not meant to be an absolute barometer, values of this order and above should be looked at with caution. Welded construction or completely cast wheels would be expected to permit higher values. Care should be exercised in this regard, however, since the maximum allowable peripheral velocity will also depend on the wheel configuration and size.

Materials have several important aspects which in their own right cannot be overlooked. A material heat treated in the wrong range may well prove to be far more notch sensitive and prone to failure than the same material heat treated so as to achieve significant toughness.

Manufacturing

The second category of manufacturing, quality control and assembly must be considered partly in terms of design wherein sizes are selected, tolerances set, materials selected, and certain assembly procedures anticipated. One example where design and assembly interact might be in the way in which impellers are shrunk on the basic shaft. Specifically, an impeller shrunk on the shaft with a long, unrelieved shrink fit may provide sufficient excitation to cause instability. It follows that quality, workmanship and care taken in assembly are necessary to

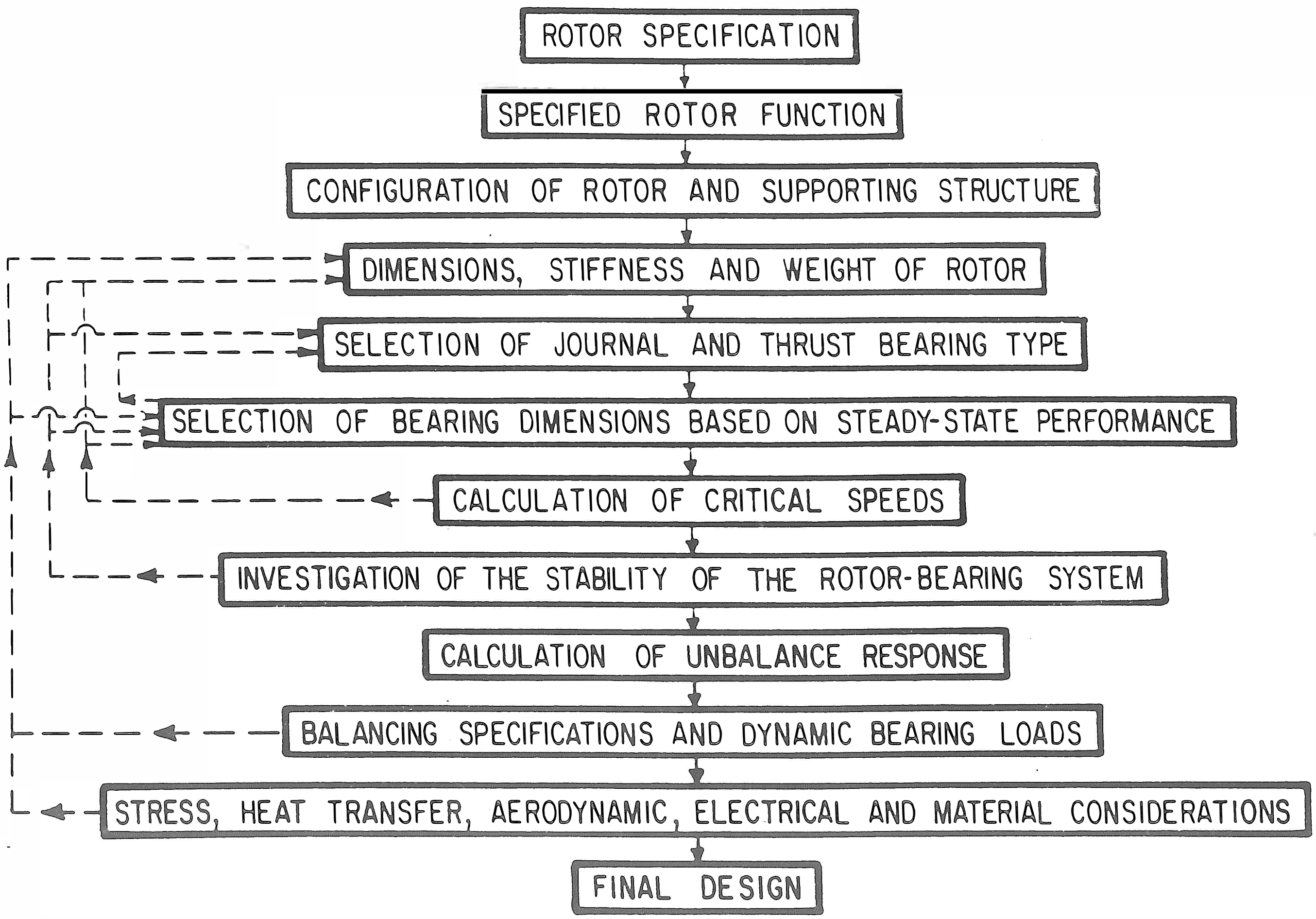

Figure 1. Logic Diagram of Design Procedure.

achieve the desired result. Assembly procedures may come into play here if uncontrolled heat which would alter the material properties is used in removal of impellers from the compressor rotor.

The above were intended to briefly demonstrate the interaction between design, manufacturing, materials and assembly.

Performance

Two of the more important parameters with respect to performance are often power level and surge margin. To a degree difficulties in these two areas stem from insufficient information concerning the selected geometry for use with other than standard gases. Often encountered is power dissipation somewhat greater than anticipated which can be of particular concern with marginal drive power and when uprating is anticipated. Surge data is generally based on air testing and it is not uncommon that the design operating point can be missed by a significant amount, (20%), when used with a process gas with thermodynamic properties much different than air. It is desirable to require at least a 25% margin for each wheel with respect to surge.

Dynamic Behavior

Dynamic behavior can take many forms, however, only two will be considered for purposes of this discussion: There are:

1. Dynamic behavior, generally synchronous, due to the interaction of the rotor system with bearing supports.

2. A self excited instability with a frequency generally a fraction of the running speed.

Any discussion of dynamic behavior must necessarily address itself to proximity to critical speeds and the general sensitivity of the rotor to large amplitude response most usually associated with deteriorating balance. This latter is of extreme significance since under the best of circumstances mass movement or deposits resulting in unbalance will take place. As a consequence, the rotor system must be able to tolerate reasonable amounts of balance change during its operating lifetime. One might say that a rotor capable of withstanding a certain amount of unbalance will also be less prone to rapid catastrophic type dynamic difficulties.

Although many techniques and presentations are made in analyzing and displaying dynamic behavior, we have found it convenient to discuss the critical speed behavior in the form of the map shown as Figure 2. This is a log log presentation with rotor speed on the ordinate and support stiffness on the abscissa. With this type of presentation the natural frequencies are calculated by assuming various values of support stiffness and generating the critical speed map. Because it is a log type presentation the region where the rotor is stiff or acts

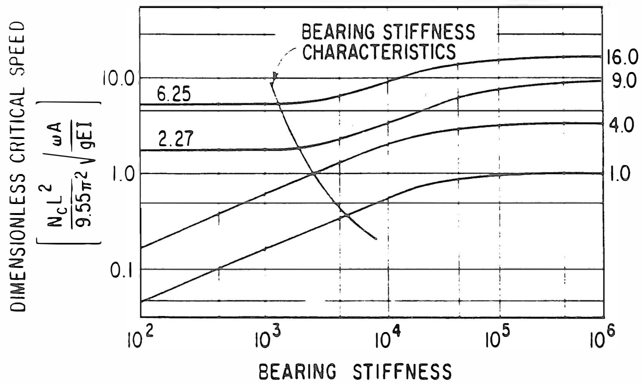

Figure 2. Typical Critical Speed Map

as a rigid body is shown as a straight line with a slope of $\frac{1}{2}$. As the rotor stiffness approaches the support stiffness in value. the curves tend to inflect. Finally, where the rotor is far softer than the supports the curves tend to be asymptotic to the abscissa.

Superposition of the bearing stiffness versus speed characteristic on this map gives the approximate location of the rotor critical speeds. The word approximate is used in this case since to this point the damping characteristics of the support system, and the bearings in particular, have not been considered.

The calculation procedure also gives the mode shapes at the critical which in their own right can present considerable guidance. If the mode shapes indicate relative amplitude at the bearing location, then it is reasonable to expect that the damping available on the bearings can be effectively utilized. Conversely, where a bearing location coincides with a nodal point, very little relative motion would be expected and the bearing would be virtually ineffective in controlling the vibrational amplitude of the rotor.

The next consideration is that of the response of the rotor to the unbalance. Figure 3 shows an amplitude versus speed characteristics for a typical rotor. Generally, some minor deviations will be noted from the predictions of the critical speed map since both support spring and damping characteristics are included. This

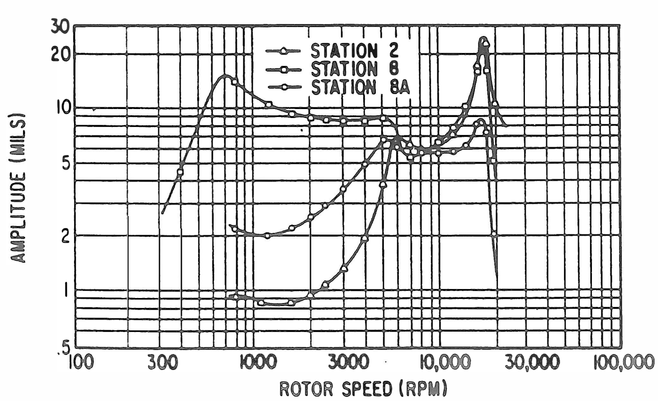

Figure 3A. Dynamic Response of Flexible Rotor Amplitude vs Speed.

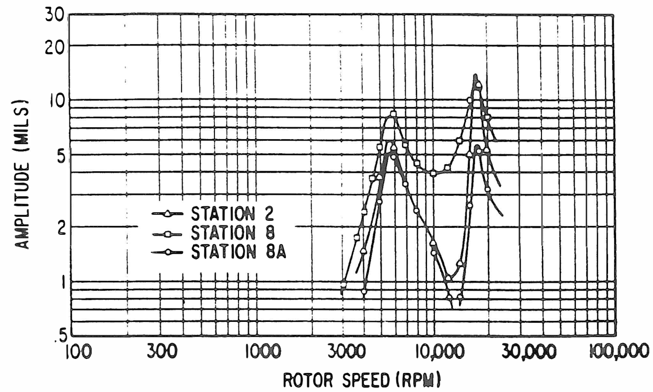

Figure 3B. Dynamic Response of Flexible Rotor Amplitude vs Speed.

form of calculation can be a significant design tool in that bearing variations, configurational changes and component changes could be readily evaluated. It was previously noted that damping is ineffective when the bearing location is on a natural node. This can be dramatically demonstrated using a response calculation. In several actual test cases the resulting amplitudes were found to be completely unacceptable. In a few of these, the minor change of swapping the thrust bearing and journal bearing location, convenient with some configurations was enough to transform the unacceptable configuration to one which was well controlled even with a nominal unbalance situation.

The question of stability is one which has become increasingly important as rotor systems operate at higher speeds and tend to become more flexible. This latter probably results from a desire to combine more stages into a single casing. In the past the solution for potential instability problems was to specify an inherently stable bearing. One such bearing type is a tilting pad bearing which is widely used. However, it should be noted that this bearing type alone does not assure stability, in fact, very frequently in combination with high pressure oil buffered floating seals the overall rotor assembly can be unstable. Recent techniques have utilized calculations of damped natural frequency as a mechanism for assessing stability. The specific calculations result in log decrements for the rotor system, which combined with theory and experience, has proven to be a powerful guide. Using a damped natural frequency calculation, it has been determined that log decrements above .5 will generally assure a stable system.

Assuming that a stable bearing is used, experience has shown that the critical speed map for figure 2 can provide some insights concerning how vulnerable the system can be. Two parameters have been used.

1. the ratio of operating speed to the first critical speed, and

2. the ratio of the critical speeds at support stiffnesses of 10^7 and 10^5 lbs/in taken from the critical speed calculation.

It is important to note that instability can also be excited by aerodynamic forces, hysteretic type damping, magnetic forces, etc.

Key Components and Systems

In a sense several of the key components have been touched upon in the preceding discussion. There are, however, numerous areas where past experience indicates that certain components can lead to serious operating problems. Because of this, it is probably in order to delineate these and comment briefly upon them.

The lubrication system is probably one of the key systems which should be evaluated. In doing so one must be assured that the lubricant is compatible with the process gas and that sufficient flow and heat removal capability is incorporated. Where there are auxiliary functions required such as moisture separation this must be properly designed and integrated with the system. Filtration, reservoir design, cover gas protection and other functions must be carefully considered. Other considerations with regard to support systems apply both to lube systems and seal systems. These require standby capability as delineated by API and other specifications. These standby or backup systems must be designed with as much reliability and capability as a primary system itself. It is a good policy to exercise these systems periodically to assure that they will be available when required.

Other key elements are:

1. Bearings;
2. Seals;
3. Balance pistons;
4. Couplings.

Although the above list can be expanded, these major items have demonstrated problems frequently enough that the current discussion is limited to them.

The area of bearings is a broad and complex one, however, as discussed previously aside from its basic requirements of supporting rotor loads, the bearing is called upon to provide the major damping utilized in restricting the vibrational amplitudes; as such it should be selected carefully and incorporated with the other parts of the system. Frequently, in a troublesome rotor, it is well to review carefully what the trade off possibilities are. One example is in the area of damping. Although a tilting pad bearing should be inherently stable in its own right, there are other bearing types which will provide greater damping to the system. This may be on an overall basis rule against the tilt pad. Another important consideration which combines bearings and dynamics is the way in which the bearing or bearing pedestal is mounted with respect to the remainder of the casing. To illustrate this, it is important to discuss the concept of effective damping. As indicated previously, when a bearing is coincident with a nodal location no relative amplitude is expected and consequently the damping will not be effective. Similarly when the bearing or bearing film has a soft support in series with it, the effective damping is significantly diminished. As a guide, when the pedestal support stiffness is equal to the film stiffness, the effective damping is reduced to approximately 40% of its original value.

Thrust bearings provide axial location and support the uncompensated aerodynamic loads. Tilting pad thrust bearings have been used quite effectively for this service. Problems arise when loadings are extended to high levels. Although these bearings often have considerable margin, the load sharing between pads will limit this quite significantly. It is only an equalizing or self adjusting model which will give the ability to support full theoretical loading.

Seals must be considered in two basic categories. The oil buffered variety and the commonly labyrinth gas seals. The former have special considerations including sealing the process gas with minimal contaminated oil loss, thermal characteristics and possible interaction with the bearing system which woud effect stability. Labyrinth seals and the selected clearances must be compatible with the dynamics of the system. The selection of materials is one of the more important factors.

Coupling problems, for the most part, can be contributed to two factors, lubrication and an overly optimistic design. A conservative selection procedure is indicated which sizes the coupling for tooth strength and wear characteristics. An article by Dudley* is a reasonable way to accomplish this end.

Finally, balance piston design must consider materials strength at operating temperatures and pressure drops and differential thermal expansion. Both thermal expansion and inadequate tooth strength have been causes of balance piston failures.

SUMMARY

The key message to be established by the preceding is:

1. the design philosophy must take on a systems approach

2. Careful attention to construction and quality control are mandatory.

3. Materials have been the cause of numerous field difficulties and should require certification on critical components.

4. Careful attention to several key components and systems will bear significant fruit in the form of reliability.

*Dudley, D. W., When Splines Need Stress Control, Product Engineering, December 23, 1957.