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Nonsynchronous Instability of Centrifugal Compressors

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The Increased use of flexible shaft centrifugal compressors has resulted IN an Increase IN instability vibration problems, particularly IN high pressure applications where the aerodynamic effects are significant Nonsynchronous instability vibrations can occur in any machine unless careful consideration IS given to all the factors that cause this type of vibration problem This paper discusses these factors and presents actual field instability vibration data obtained IN three case histories Data analysis and presentation techniques are described which allow a better understanding of the characteristics of nonsynchronous instabilities Using the instrumentation and techniques described IN this paper, the onset of instability of a machine can sometimes be anticipated and remedial action taken before unbounded vibrations occur The documentation of instability characteristics for severai machine modifications, including bearing span, bearing type, seals, impellers, etc , shows that present analytical techniques are not adequate to predict the observed phenomena

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Nonsynchronous Instability of Centrifugal Compressors

J. C. WACHEL

INTRODUCTION

Non-synchronous instability vibrations have caused severe failures in an increasing number of high pressure compressors and turbines in recent years. The cost of plant down time normally involved in such failures, together with the substantial expense in redesigning, retrofitting, and testing field machines often totals several million dollars per failure.

Non-synchronous instabilities normally occur in high-speed, high horsepower units which operate above the rotor's first critical speed. The whirling instability frequency is usually near one of the shaft critical speeds and can be caused by many factors, including hydrodynamic bearings, seals, internal friction, and aerodynamic cross coupling, The whirling motion can be subsynchronous or supersynchronous, and may be forward or backward precession.

Vibration data has been collected on several different compressors that have experienced severe non-synchronous vibration. Each of the compressors differed in manufacture, shaft diameters, weight, bearing span, critical speeds, and running speed. Using modern instrumentation, it was possible to observe in real time the instantaneous spectral characteristics of shaft vibrations as the compressors approached the onset of instability, i.e., before the machine experienced the high level vibration normally associated with full-scale instability, Analysis of the vibration data has shown that on most units that have non-synchronous instability problems, a trace of vibration at some instability frequency normally exists at all times; however, it is not possible to verify system stability from vibration measurements at one operating condition. The limits of instability can be fully defined only from testing over the full performance range of the machine, and even this approach is not always completely adequate. One tested unit ran satisfactorily for over a year before serious instability tripouts occurred, and subsequently the compressor failed eight times in three years. Because the stability margin on some units is so delicately balanced, its characteristics can be drastically changed whenever small changes are made in factors such as bearing clearance, oil temperature, unbalance, alignments, and the like.

It follows, therefore, that the threshold of stability can likewise be improved by small changes in these same parameters, but the exact improvement required to make an unstable system stable is difficult to predict. Lund (1)[1] had presented the logarithmic decrement technique for predicting rotor stability which represents a significant improvement over previous calculation techniques. Field experience shows that while this technique provides proper direction in designing for stability, uncertainty still exists in quantitatively predicting the onset of instability and defining the contribution of individual influencing parameters.

While many excellent papers are available which treat the mathematics of selected portions of the overall instability problem (1-3), little field experimental data on large high pressure compressors has been published. When analytical techniques are not adequate to explain observed machine ph-nomena, then one must rely upon actual field experiences to determine needed safety factors during the design process. Also, if instability vibrations occur, then the available mathematical models must be normalized to measured field data in order to obtain estimates as to the possible effects of system changes. This paper will present measured field data on three compressors which exhibited non-synchronous instabilities, The purpose will be to show the type of vibrations that occur and the data presentation techniques which can be used to more fully illustrate the characteristics of non-synchronous instabilities. These analyses can define stability thresholds and can be used to

[1] Numbers in parentheses designate References at end of paper.

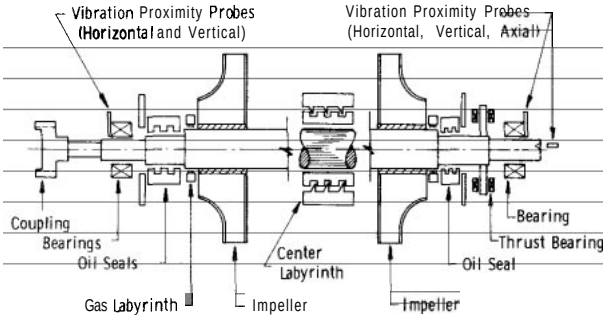


Fig. 1 Typical high pressure compressor design

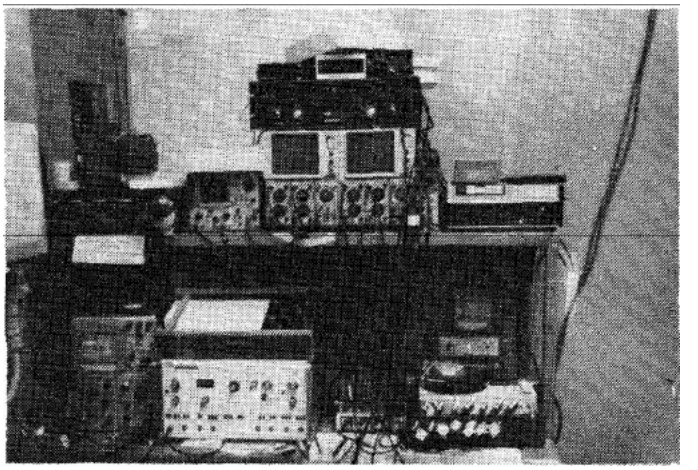


Fig. 2 Field instrumentation used to measure non-synchronous instability vibrations

evaluate the effects of seal, bearing, aerodynamic, and other modifications,

DESTABILIZING EFFECTS IN COMPRESSORS

The compressor indicated in Fig. 1 represents a typical design for high pressure centrifugal cnmpressors employing a back-to-back impeller design to minimize thrust balance and sealing problems. The vibratory response and stability characteristics of such rotors are normally checked in the design stage using digital computer programs employing the Myklestad-Prohl method. The stability margin expressed in terms of the logarithmic decrement can be determined by this method if the complex eigenvalues are used rather than just the real part.

To properly calculate the stability margin of a rotor, the mathematical model must be able to consider all possible destabilizing components. There are several mechanisms which have been observed to contribute to rotor instabilities (1, 5, 6):

1. Hysteretic or internal friction damping
2. Hydrodynamic fluid flim bearings and seals
3. Aerodynamic cross-coupling forces
4. Asymmetric shafting and/or bearing characteristics
5. Pulsating torque and axial loads
6. Fluid trapped in rotor
7. Stick-slip rubs and chatter
8. Dry friction whip.

For centrifugal compressors, the most likely contributors are the first five listed. To more clearly illustrate, let us refer to the compressor in Fig. 1 and systematically point out the prominent destabilizing forces for each of the rotor elements.

Coupling

- o Frictional damping at gear teeth
- o Frictional damping due to press fit on coupling hub
- o Pulsating torques
- o Misalignment causing pulsating torques and axial loads
- o Unbalance

Bearings

- o Hydrodynamic oil film corss coupling effects
- o Ratio of vertical and horizontal oil film stiffnesses
- o Improper support stiffness or asymmetry
- o Hydrodynamic effects from thrust bearing

Seals

- o Hydrodynamic fluid film effects
- o Frictional effects due to breakaway sealing force in axial direction
- o Aerodynamic effects

Labyrinth Seals

- o Hydrodynamic fluid film effects
- o Aerodynamic effects
- o Frictional damping due to shrink fits

Impellers

- o Aerodynamic diffuser-impeller interactions
- o Frictional damping due to shrink fits
- o Unbalance
- o Gyroscopic effects

System Related

- o Rotor too flexible
- o Bearing span too long

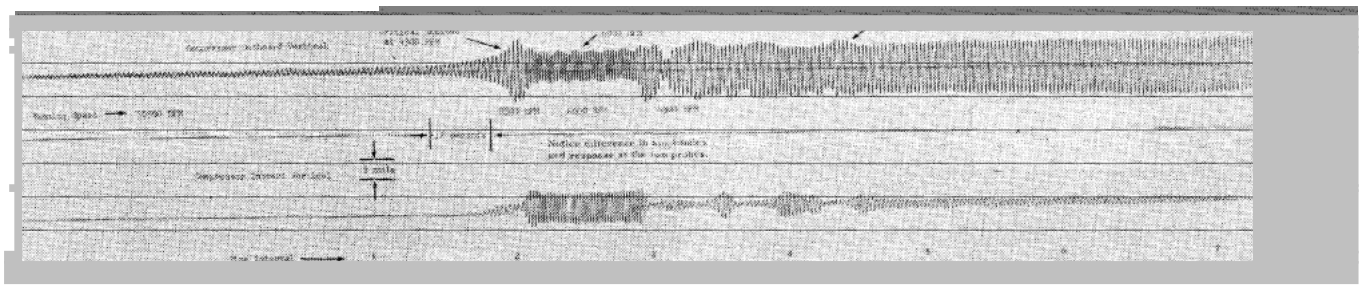


Fig. 3 Complex wave presentation of compressor instability

- Shaft diameter too small
- Impellers too heavy
- Ratio of running speed to first critical too large
- Damping in bearing ineffective due to bearing location
- Surge characteristics
- Rotating stall
- Pressure pulsations and acoustical resonances
- Inadequate pressure balance
- Inadequate foundation
- Shaft material damping.

Effective analysis of all combined effects of these destabilizing factors in the design stage is, of course, an impossible task, and research is currently underway at SwRI and elsewhere to improve available predictive techniques. Experimental facilities to study stability phenomena are expensive, and it is difficult, if not impossible, to systematically isolate all contributing factors. If the individual effects can be better defined through full-scale tests on field machines, then similitude and models can, perhaps, be used to improve the calculation procedures.

When instabilities are encountered on a machine, the procedure for the fixing the problem involves removal or reduction of the destabilizing factors. Because of the complexity of making major machine modifications, the first try in field modifications usually involves the bearings, seals, and outer labyrinths. If modifications in these areas are unsuccessful, then reducing shaft flexibility and modifying impeller aerodynamics usually follow. Damper bearings installed in series with the hydrodynamic bearings can significantly improve stability characteristics in some cases; however, the techniques for predicting their effect requires further work.

ANALYSIS OF NONSYNCHRONOUS INSTABILITY VIBRATION DATA

When a compressor becomes unstable, vibration levels increase suddenly, sometimes from less than a mil to 10 mils (1 mil = 25.4 μm) in less than a second. Special care and considerable instrumentation is required to analyze the rapid transient and to present a maximum of information in a form which can be easily understood. Fig. 2 shows the field instrumentation system used in the present studies to document compressor operation. The instrumentation includes a real time analyzer, oscilloscopes, X-Y recorder, FM tape recorders, proximity probe instrumentation, transducer amplifiers for pulsation and accelerometer measurements, trim balance analyzer, spectral time history generator, order tracking instrumentation, tachometers, switch boxes, and signal cables. Using this instrumentation, it was possible to document the thresholds of stability and to evaluate the modifications that were made to the machines. To help in the explanation of the case histories, some of the data presentation techniques will be discussed.

The complex waves (amplitude versus time) of two shaft vibration probes during a compressor instability trip-out are given in Fig. 3. This method of presentation is important in obtaining the total peak-to-peak vibration amplitude as a bas'is for identifying damage to bearings, seals, and labyrinths, etc., due to touch-off or high vibration. It is difficult to define the system running speed from strip chart records, since the initiation of instability will completely mask other vibration components. In this case, a much clearer understanding of the sequence of events can be obtained by the developnent of spectral time histories (4) which give a three-dimensional view of frequency-amplitude-time.

The spectral time histories, or rasters,

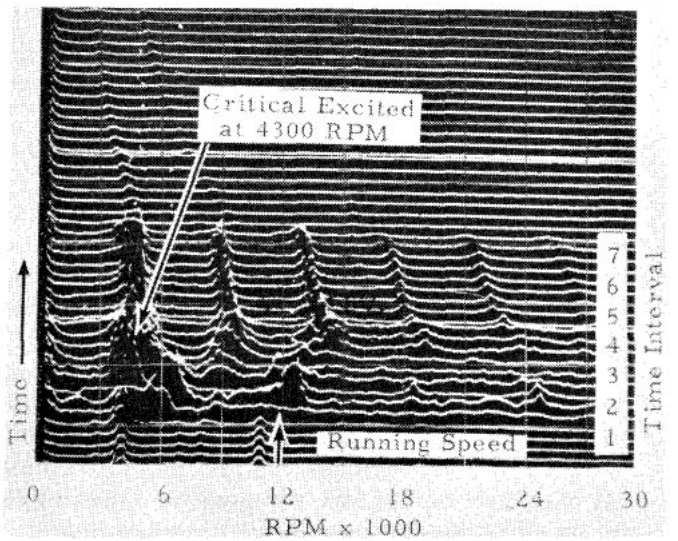


Fig. 4 Spectral time history of compressor tripout showing instabilities

of vibration data are generated using a real time analyzer, and can be taken either off the machine directly or from FM tape. By making sequential frequency analyses and incrementing the analysis up vertically on a storage oscilloscope face, a frequency analysis versus time record can be conveniently generated and effectively displayed. A photograph of such a signature gives the spectral time history of whatever event one wants to analyze. Fig. 4 is a spectral time history of the same compressor rotor instability shown in Pig. 3. The time intervals marked on the strip chart correspond to the numbers on the analysis.

Since the use of the oscilloscope limits the length of time that can be analyzed, another technique was developed so that the spectral analysis could be taken directly from the real time analyzer at its basic repetition rate and displayed on a fiber optics strip chart recorder. With this equipment, the complex wave can be displayed along the side of the frequency analyses, allowing a direct comparison of overall peak-to-peak amplitudes with amplitudes of each harmonic component. Fig. 5 gives a spectral time history of a compressor trip-out using the fiber optics strip chart recorder. There are 25 analyses performed per second, and the chart speed is 2 in. (50.8 mm) per second. The primary advantage of this presentation technique lies in its signal enhancement ability whereby the first onset of instability can be accurately determined. Analysis of considerable data shows that in most cases, the onset of instability can be anticipated by monitoring the amplitude of the non-synchronous instability from the instantaneous

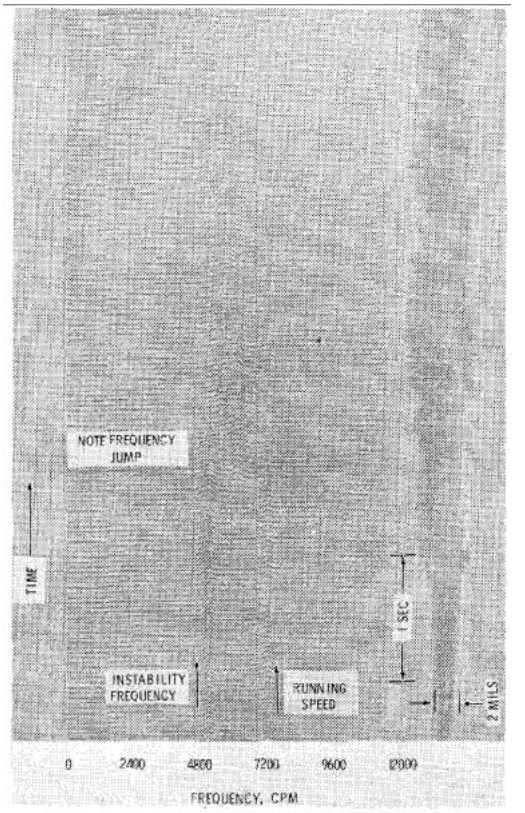


Fig. 5 Spectral time history generated by fiber optics recorder

real time analysis. Before the instability becomes boundless, there will be a noticeable increase in the instability component amplitude or at least an increase in the spectral content near the instability frequency range. This is vitally important to operating personnel if a problem is encountered, since it provides a technique whereby incipient instability may be predicted at its very early stages. This is extremely valuable if it can prevent the machine damage resulting from touch-off in the bearings, seals, and labyrinths.

To determine the maximum amplitudes of the instability frequency components that occur during tripouts, it has been found that a peak store plot from the real time analyzer memory gives the most accurate information,

The author has been involved in the solution of several instability problems in large compressor units and has acquired tape recordings of their vibration trip-puts and characteristics. The compressors, which were manufactured by different companies, had significant differences in horsepower, speed, suction and discharge pressures, and service. The characteristics of the instabilities were similar, however. Since instabilities can continue to be a problem as com-

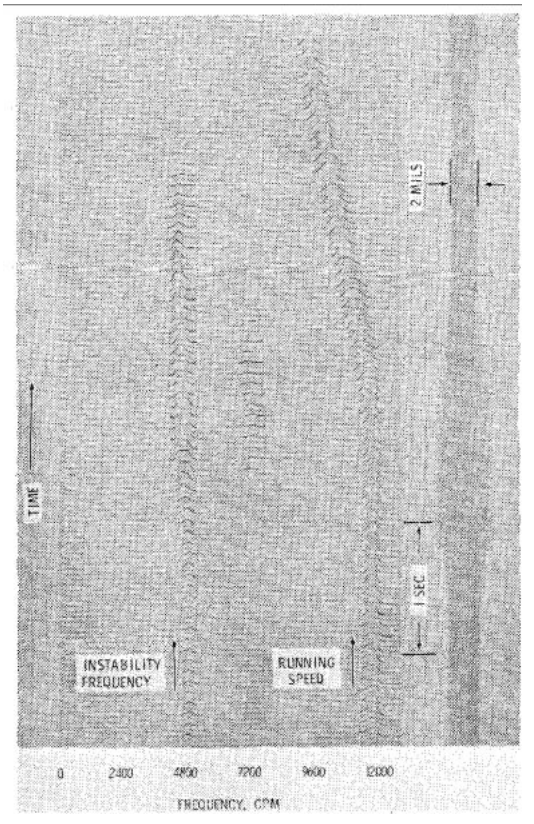


Fig. 6 Instability tripout of nine-stage compressor

pressor speeds, pressure, and horsepower are increased, three case histories will be discussed so that a better understanding of the vibration characteristics can be obtained. The data presented will illustrate the need for additional research in stability prediction.

Case No. 1

The spectral time history of the compressor instability presented in Fig. 4 was for a 13,000 hp, 10,600 rpm, eight-stage compressor with back-to-back impellers. The compressor had a 64-in. (163-cm) bearing span with a critical speed of 3800 cpm and a rigid bearing critical of 4300 cpm. The suction pressure was 150 psi (10.3 bars) and the discharge pressure 500 psi (34.5 bars), The complex wave (Fig. 3) shows that the instability component at 4300 cpm increased from 1 to 4 mils (25.4 to 101 μm) over about a 1-sec interval and then sharply increased to 16 mils (406 μm) in approximately 0.2 sec. The vibrations then shifted to 6000 cpm and then locked in on 4300 cpm (16 mils, or 406 μm) until the compressor speed was below 4000 rpm. The inboard vertical probe had slightly different characteristics, illustrating the need for full instrumentation. This compressor failed eight times due to these non-synchronous vibrations. The seals and labyrinths were wiped in an increasing bow pattern such that the inner labyrinths had approximately 0.10 in. (2.54 mm) of material removed. The mode shape was so pure that the pieces appeared to have been turned in a lathe,

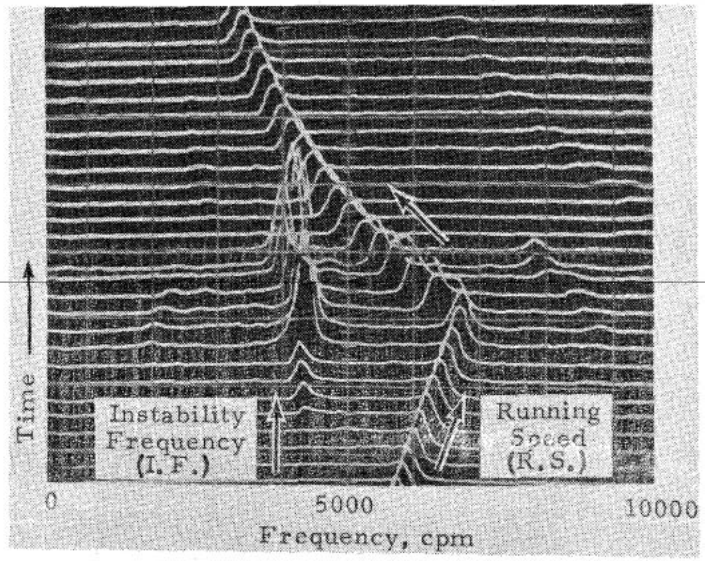
Several modifications were made in an attempt to solve this problem. The impeller hubs were undercut to reduce the hysteresis effects at the mating surfaces. The clearances in the seals and labyrinths were increased. The five-shoe tilted pad bearings were modified by reducing the pad areas on the side and by increasing the radial clearance to force the rotor to vibrate in a horizontal elliptical orbit. In this compressor, the recommended changes were sufficient, and the machine has run for several gears without further non-synchronous vibrations.

Case No. 2

A non-synchronous vibration also occurred in a 1300-hp, nine-stage injection compressor with back-to-back impellers. The suction pressure was 1100 psi (75.9 bars) and the discharge pressure was 3200 psi (221 bars). The rotor had a bearing span of 60 in. (152.4 cm) with oil seals located slightly inboard of the bearings, The calculated first critical speed was 3800 cpm. The characteristic of the non-synchronous instabilities is illustrated in the spectral time histroy of a shaft vibration signal recorded during a trip-out (Fig. 6). The unit was running at 11,000 rpm with a small amplitude of instability at 4500 cpm. When the suction pressure was increased, the vibration trip-out occurred. The large increase was at 4500 cpm with a small amplitude of 6800 cpm. Since the critical speed was at 3800 cpm, it can be seen that the oil seals were effectively increasing the response frequency.

Data taken from other runs showed that normally, a sub-synchronous compenent at one-half running speed was present in the vibration as the compressor speed increased from 8500 to 10,000 rpm. Prom 10,000 to 11,400 rpm, this frequency component remained at approximately 4500 to 5000 cpm. As the speed approached 11400 rpm, the amplitude would suddenly increase, causing the unit to shut down. At a constant speed, the sub-synchronous frequency was a funtion of the pressure ratio across the compressor. The design discharge pressure could not be reached.

Several field modifications were made to these units without success. These included changes in the seals, bearings, impeller undercuts, press fit, shaft diameter, bearing span, and diffuser volute. When these modifications were



Pig, 7 Tripout of compressor as speed is increased (1 mil/div, 1.28 sec/line)

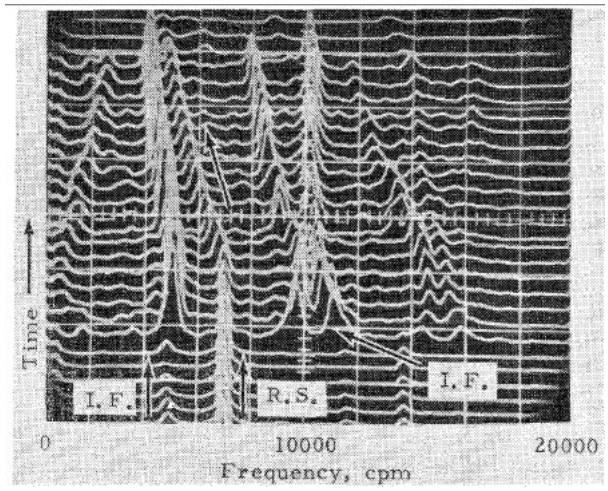


Fig. 8 compressor instability with improved seal design (1 mil/div, 0.64 sec/line)

unsuccessful, the bearings were changed to six pad titled pad bearings and squeeze film damper bearings were installed on each end of the rotor. Although the calculation procedures at that time predicted a significant improvement in the stability threshold, there was little, if any, improvement, even when combined with a slightly larger shaft diameter and a reduction of 6 in. (15.2 cm) in the bearing span. Only after the shaft diameter was further increased was the compressor able to reach its design pressures.

With the shortened bearing span and increased Shaft diameter, the first critical increased from 3800 to 4600 cpm. The frequency of' the original instability componet was 4500 cpm, whereas for the modified rotor the frequency was 6400 cpm. Although this was a significant change, the compressor still could not reach design pressures, again illustrating the difficulty in predicting the effectiveness of proposed modifications. The delay and cost of increasing shaft diameter (often requiring new forgings for the shaft), and for casting, forging, and machining new impellers again suggests a very real need for improved calculation techniques.

Case No. 3

A recently installed reinjection compressor experienced excessive non-synchronous vibrations on start-up. The 22,000-hp, eight-stage compressor with back-to-back impellers was rated at 8500 rpm, had a design suction pressure of 3500 psi (241 bars) and discharge pressure of 9200 psi (634 bars). The first critical speed of the rotor with a bearing span of 81 in. (206 cm) was 3800 cpm. The rigid bearing critical was 4200 cpm. Floating oil seals were located a Pew inches inboard of the bearings. The compressor could not be brought to design speed and pressure without tripping out on high vibrations (Fig. 7). The units were monitored by shaft vibration probes which automatically shut down the unit whenever the vibrations exceeded 2.5 mils (64 μm); however, due to the monitor's finite response time and suddenness of the instability trip-outs, vibration amplitudes equaling total bearing clearance were experienced.

The frequency of the non-synchronous instability was at 4400 cpm. As in most instability problems, the frequency was higher than the rigid bearing critical speed. This can occur if the floating oil seals are carrying some load and effectively reducing the bearing span. If this is true, then the seals also cause destabilizing cross-coupling terms. These can greatly affect the threshold of stability. For this shaft, the effective oil seal stiffness had to be nearly 500,000 lb/in. (8.9×10^6 kg/m) to get an instability frequency of 4400 cpm. With this stiffness for the oil seals, the calculated log decrement was 0.08 compared to 0.3 calculated for the original rotor, neglecting the effect of the seals.

To reduce the effects of oil seals, two circumferential grooves were cut into the sealing surface. The pressure balance of the rings was improved and the coefficient of friction of the sliding surfaces was reduced, The compressor was still unstable, as can be seen in Fig. 8, where the spectral time history of one instability trip-out is displayed for unit speeds up to minimm governor speed (7200 m). A large amplitude

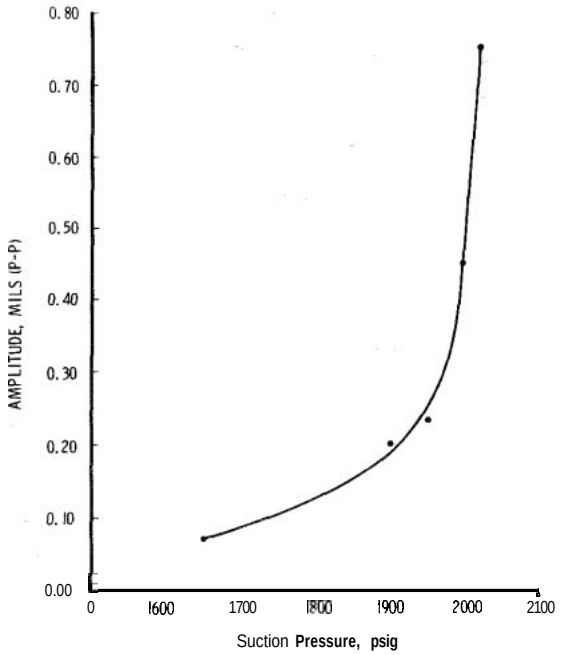


Fig. 9 Instability amplitude versus suction pressure

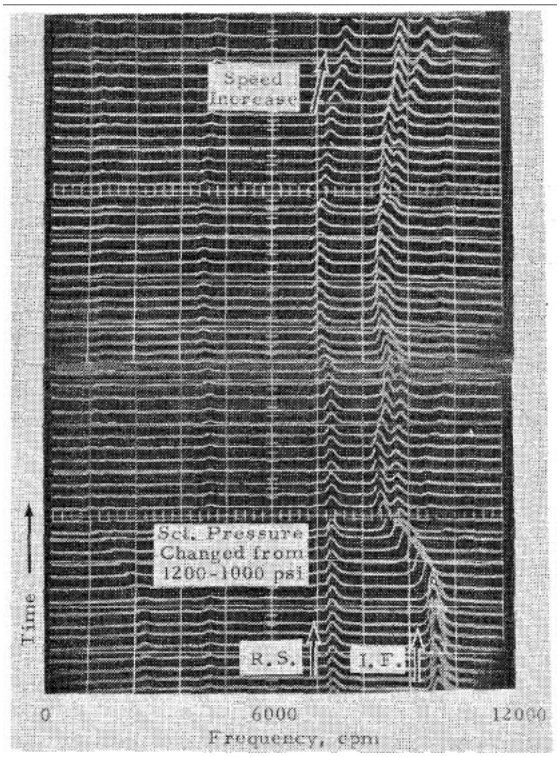


Fig. 10 Variation of instability frequency with suction pressure (1 mil/div, 10.24 sec/line)

non-synchronous instability occurred at 4700 cpm; however, frequencies above running speed at 9500 and 10,500 cpm were also excited. Notice that as the unit speed is reduced, the compenent at 10,000 cpm remains and does not follow speed. The rotor was found to be sensitive to the rate of acceleration; therefore, by slowing down the start-up procedure, it was possible to operate in the normal speed range. To more fully define the stability limits, data was obtained throughout the entire performance map. The system parameters were changed until the non-synchronous component became large. The procedure was then to reduce speed and/or pressure until the vibrations were sufficiently reduced. Fig. 9 shows that for constant speed of 7600 rpm, the amplitude of the instability component at 5160 cpm increased as the suction pressure was increased. A check of the mode shape of the instability component showed that it was a forward precessional, even mode. The frequency of the instability component which originally was at 4400 to 4800 cpm moved up to 5160 cpm as pressure was increased. When the suction pressure was held constant at 1500 psi (103 bars) and the speed increased, the instability amplitude also increased. The speed effects were quite noticeable when the instability amplitude was high; for example, at the highest suction pressure measured (2025 psi, or 140 bars), the amplitude of the instability component reduced from 0.75 to 0.07 mil (19 to 2μm) when the speed was reduced 200 rpm while everything else remained constant.

Several seal designs were tested; however, there was little improvement in the overall rotor stability. The type of seal design greatly affected the frequency of the non-synchronous instability and the threshold speed. One seal design had large radial clearances and only one land (less than 0.2 in.) or 5 mm, long). The test was primarily to study the effect on the instability frequency since the seal oil leakage was excessive. The results were unexpected. Theory would indicate shortening the seal length which would reduce its load-carrying effect, thus reducing instabilty frequency. However, these results showed that non-synchronous instabilities occurred at frequencies above running speed, viz: 10,000 and 11,000 cpm. As trip-out approached, a component near 4000 cpm appeared. The trip-out was caused by a 6-mil (152-μm) compenent at 4700 cpm when the running speed reached 6900 cpm.

Another test using a different seal design also showed instabilites above running speed. These instability frequencies were a function of suction pressure. Compressor speed was 7523 rpm and suction pressure 1200 psi (82.8 bars). Fig. 10 gives the frequency analysis showing 0.5 mil (13 μm) at running speed and 1.0 mil (25 μm) at 10500 cpm and a trace at 4500 cpm. As the suction

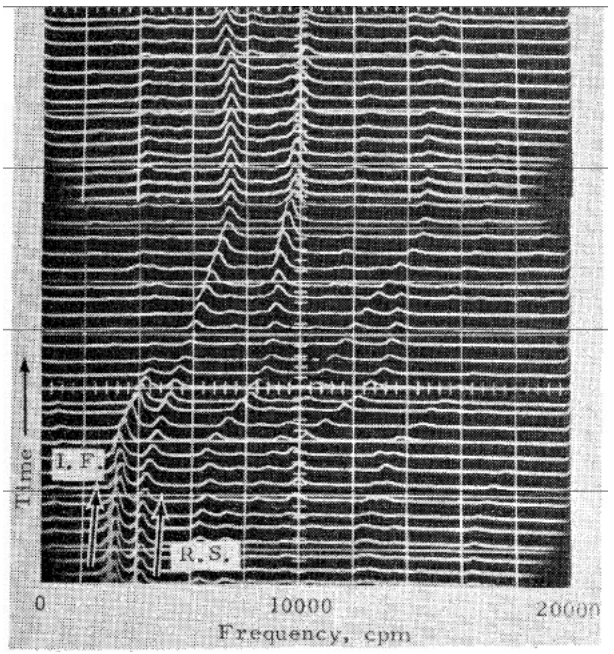


Fig. 11 Instability excited as compressor goes through first critical (2mil/div, 5.12 sec/line)

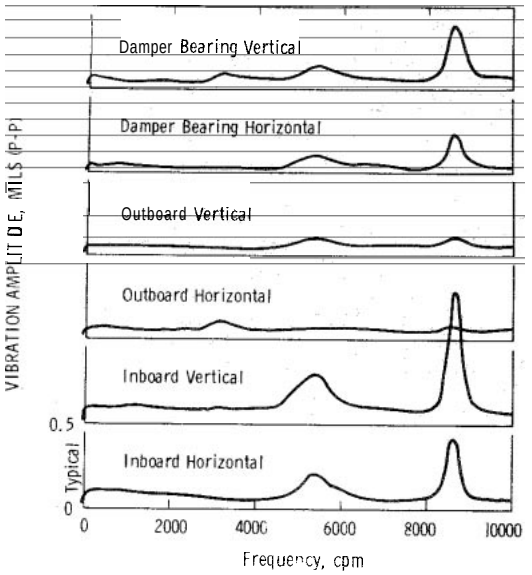


Fig. 12 Frequency analysis of shaft and damper bearing vibrations with discharge pressure of 8250 psi

pressure was reduced, the higher frequency component lowered to 8900 cpm and then separated into two components, 8900 and 9300 cpm.

Pig. 11 shows that an instability component at 0.8 times running speed occurred for this seal design when the running speed was 4000 rpm or slightly above the first critical speed. This instability followed running speed until the running speed reached 5500 rpm and then it diminished. In this data, instability occurred when the ratio of running speed to first critical was 1.25, showing that a ratio of running speed to first critical of less than 2:1 does not necessarily insure that a rotor will be stable. The majority of the instability trip-outs were at speeds less than the design speed. Thus, the effective ratio of running speed to first critical was normally less than 2:1.

The data presented in the foregoing shows that by changing nothing but the oil seals in a large compressor, the stability frequency characteristics were dramatically changed; however, no change in seal design could make this system stable. This indicates that the predominant destabilizing factor was not the seals.

Major efforts were then expended to reduce other destabilizing factors, including aerodynamic changes to the impellers and diffusers, modified seal design based upon field data, shortened bearing span and change in bearings to five-shoe non-preloaded titled pad bearings. Based upon log decrement calculations, these changes should have represented a significant improvement.

Vibration data taken with these modifications installed Fig. 5 showed a violent instability. A small trace of instability component (5200 cpm) was present with a fluctuating amplitude. The frequency then shifted up to 5800 cpm, and the amplitude jumped to greater than 6 mils (152 μm) in approximately one second. The instability component was particularly sensitive to pressure ratio across the machine, which confirms that the aerodynamic destabilizing effects were of major importance and overshadowed other improvements that were made.

The stability of the unit was markedly improved when a damper bearing was installed in series with the inboard bearing. Fig. 12 gives the frequency analysis of the shaft vibrations and two probes monitoring the damper bearing for 8450 rpm with a suction pressure of 3350 psi (231 bars) and a discharge pressure of 8250 psi (569 bars). The significant improvement in the stability characteristics of this rotor illustrates the potential that damper bearings have in high pressure applications.

Squeeze film damper bearings are essentially a film of oil between the outside of the bearing and the case, to which oil is contiuously supplied. Stiffness of the damper bearing is usually supplied by a mechanical support such as a squirrel cage cylinder with ribs, welded rod support, corrugated metal ring, or o-rings (7). There are difficulties involved in optimizing the damper bearing stiffness and damping values (1,

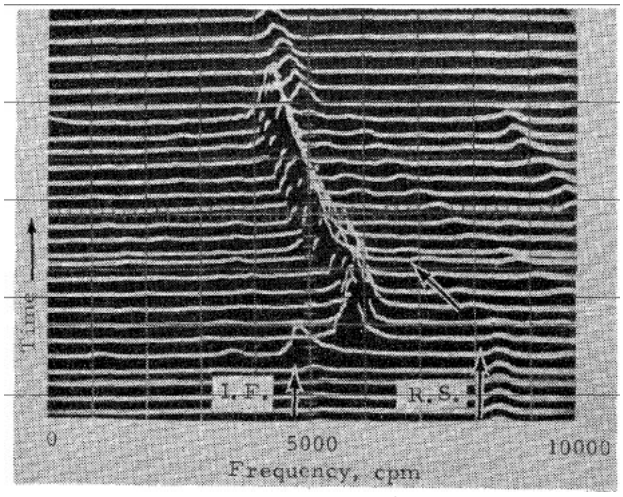


Fig. 13 Instability tripout of rotor with damper bearing installed (8.0 mil/div, 0.64 sec/line)

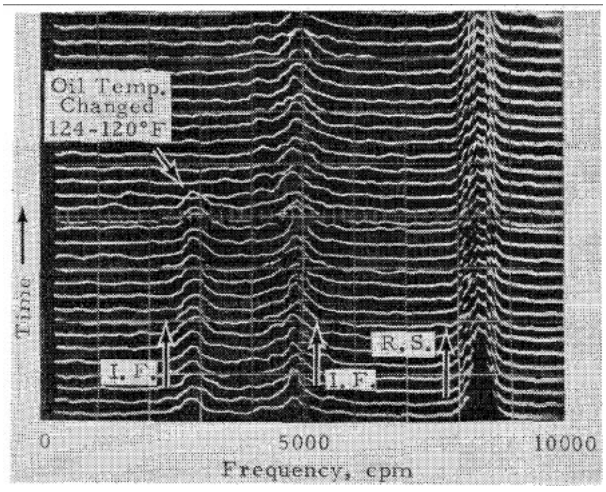


Fig. 14 Elimination of instability component as oil temperature reduced (0.5 mil/div, 5.12 sec/line)

5, 8). Due to the inadequacies of the mathematical techniques for predicting instabilites, the application of a damper bearing to an industrial compressor involves some tuning to the individual rotor. This can be amply illustrated by the fact that a damper bearing installed in a second identical unit was not successful in eliminating instability trip-outs (Fig. 13). After some tuning, the stability was significantly improved.

In the process of checking out performance of the damper bearing, the oil temperature was varied to see if it had a significant effect. At an oil temperature of 124 F, the frequency analysis showed instability components at 2800 and 4800 cpm along with the running speed component. Using the spectral time history instrumentation, some surprising results were noted as the temperature was lowered, At a temperature of approximately 121 F, the 0.25-mil (6-μm) component at 2800 cpm disappeared (Fig. 14).

This case history illustrates that considerable research effort is still needed to develop adequate mathematical models to simulate the non-synchronous instability phenomena measured in these machines. It is apparent, for example, that many factos influence the onset, frequency, and amplitude of instability vibrations, and that a systematic separation and analysis of variable will be required before the analytical techniques achieve the accuracy required for machine design.

SUMMARY AND CONCLUSIONS

A general discussion of the destabilizing elements in modern high-pressure compressors has been presented, Instrumentation and data presentation techniques which have been developed for analyzing field vibration data of machine instabilities have been discussed, and their use in developing solutions has been illustrated in three sample cases. Based upon the detailed data on these and other instability problems, the following conclusions are suggested:

1 The rotor instability problem is not associated with any one manufacturer but is a rotor dynamics problem that will have to be dealt with by every manufacturer who intends to market high pressure or high ratio machines.

2 The log decrement techniques for estimating the stability is an improvement over techniques used previously.. Consideralble research is still required to define the proper magnitude of the destabilizing aerodynamic and seal effects To model the rotor instability responses shown in the spectral time histories may require a completely new approach.

3 The aerodynamic effects appear to be much larger than theoretical considerations would indicate.

4 Floating oil seals greatly influence the instability frequency responses; based on field studies to date, more quantitative design techniques for design of these devices are still needed.

5 Although it is highly desirable to keep the ratio of the running speed to first critical less than 2:1, the data presented shows that this does not guarantee stable operation. In general, whenever non-synchronous instabilites are involved, the best approach is to stiffen the shaft

by increasing the shaft diameter or shortening the bearing span, or preferably both. A search of other literature also reveals that this has been the most successful approach.

6 Recognizing that changing shaft diameters and bearing span are impractical in many cases, efforts are often required to eliminate or reduce all the destabilizing components that are acting on the rotor.

7 Squeeze film damper bearings can cause significant improvement in the overall stability of rotors. It should be recognized that some tuning of the stiffness and dampong values may be necessary before any noticeable improvements are obtained. Until an accurate model exists for predicting rotor instability response, there is a possibility that retrofitting a damper bearing to an existing installation will not be sufficient.

8 It has been shown that by the use of special instrumentation and presentations from real-time analyzers and spectral time history rasters, the onset of non-synchronous instability can be anticipated. Using this warning, it is sometime possible to prevent damaging machine vibrations by changing system variables such as speed, pressure, ratio, or oil temperature.

9 Special analysis techniques and presentations are required to understand the sequence of events that occur when a field compressor experiences non-synchronous instabilities. Techniques to accomplish this have been discussed and used in three sample cases. When a field instability problem occurs, it is important to properly instrument and record the vibrations so that accurate assessment of changes can be made.

10 Full load-pressure shop tests should give a good indication as to the stability of a unit. While such tests in the shop may not adequately simulate foundation effects of a field unit, these should be of secondary importance to many of those which can be identified.

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