FIELD VERIFICATION OF LATERAL-TORSIONAL COUPLING EFFECTS ON ROTOR INSTABILITIES IN CENTRIFUGAL COMPRESSORS

J. C. Wachel and F. R. Szenasi
Southwest Research Institute
San Antonio, Texas 78284

SUMMARY

Lateral and torsional vibration data were obtained on a centrifugal compressor train which had shaft instabilities and gear failures. The field data verifies that the stability of centrifugal compressors can be adversely affected by coincidence of torsional natural frequencies with lateral instability frequencies. The data also indicates that excitation energy from gearboxes can reduce stability margins if energy is transmitted either laterally or torsionally to the compressors. The lateral and torsional coupling mechanisms of shaft systems have been investigated both in theory and in laboratory models by other investigators. This paper documents these coupling mechanisms in a large industrial compressor train and demonstrates the potential effect on rotor stability. Guidelines are set forth to eliminate these potential problems by minimizing the interaction of torsional and lateral responses and their effect on rotor stability.

INTRODUCTION

Rotor instability vibrations in compressors and turbines have occurred more frequently in recent years and have caused severe failures and costly downtime for several large projects. Rotor instabilities can occur in flexible shaft units which operate above their 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, aerodynamic cross coupling, and torsional coupling. The whirling motion can be subsynchronous or supersynchronous, and may be forward or backward precession; however, most serious problems are subsynchronous and have forward whirl (refs. 1, 2, 3, 4).

In the past few years, vibration data has been collected on several compressors that have experienced severe instabilities. These compressors differed in manufacture, shaft diameter, weight, bearing span, critical speeds, and running speed. Using modern instrumentation, the instantaneous spectral characteristics of shaft vibrations were observed in real time as the compressors approached the onset of instability, i.e., before the machine experienced the high level vibrations normally associated with full scale instability. Real time analysis of the vibration data has shown that on most units that have instability problems, a trace of vibration at some instability frequency normally exists at all times; however, it is not possible to verify the severity of the instability from vibration measurements at one operating condition. The threshold 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. Units have run satisfactorily for several years before serious instability trip-outs occurred. After one year of satisfactory operation, one compressor failed eight times in the next three years from instabilities caused by unexpected transients. 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 pressure ratio, flow, bearing clearance, oil temperature, unbalance, alignment, etc., or upsets in the process such as liquid slugs, surge transients, or electrical trip-outs.

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 sometimes difficult to predict.

There are several mechanisms which have been observed to contribute to rotor instabilities. The most sensitive elements which influence rotor stability include the following:

(1) hydrodynamic cross coupling in fluid film bearings
(2) seals and labyrinths
(3) aerodynamic cross coupling forces
(4) hysteretic or internal friction damping
(5) pulsations
(6) pulsating torque and axial loads
(7) asymmetric shafting
(8) fluid trapped in rotor
(9) stick-slip rubs and chatter
(10) dry friction whip.

To properly calculate the stability margin of a rotor, the mathematical model must be able to simulate all possible destabilizing components. The logarithmic decrement evaluation of rotor system damping as presented by Lund (ref. 4) is useful for predicting rotor stability. 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.

When instability vibrations occur in installed machinery, better estimates of the possible effects of system changes can be made if measured field data is available for normalization of the mathematical model. The normalization procedures compensate for unknown dimensional variations which affect bearing and seal properties and adjust for actual aerodynamic loading. This paper will present measured field data on several compressors which exhibited instabilities. Data analysis techniques presented can define rotor stability thresholds and the effects of modifications to seals, bearings, shafts, and process variations.

Comprehensive experimental data can be invaluable in defining critical elements in the computer simulation of rotor instability, and can aid in improving their modeling.

ANALYSIS OF ROTOR INSTABILITY VIBRATION DATA

Capturing rapid instability transients and presenting a maximum of readily understandable information requires specialized instrumentation to develop Campbell diagrams, spectral time histories, and order tracking plots. Field instrumentation used to document compressor instabilities, including a real time analyzer, oscilloscope, 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, are shown in figure 1.

A clearer understanding of the sequence of events during instabilities can be obtained by the development of spectral time histories than by viewing events on a strip chart recorder or oscilloscope. The complex waves (amplitude versus time) of two shaft vibration probes during a compressor instability trip-out are given in figure 2. Although this method of presentation is important in obtaining the total peak-to-peak vibration amplitude as a basis 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.

The spectral time histories, or rasters, of vibration data are generated using a real time analyzer, and can be taken either off the machine directly or from FM tape or using digital FFT computer techniques (waterfall diagrams). By making sequential frequency analyses and incrementing the analysis vertically on a storage oscilloscope, a frequency analysis versus time record can be conveniently generated and effectively displayed and photographed. Compare the spectral time history (figure 3) of the same compressor rotor instability shown in figure 2. The time intervals marked on the strip chart correspond to the numbers on the analysis.

By using a fiber optics strip chart recorder, the complex wave can be displayed alongside the frequency analyses, allowing a direct comparison of overall peak-to-peak amplitudes with amplitudes of each spectral component (figure 4).

The authors have found these data acquisition techniques to be particularly useful in the solution of instability problems in large industrial compressor units.

INSTABILITY OF SYN GAS COMPRESSOR

The spectral time history of the compressor instability presented in figure 3 was for a 13000 horsepower, 10600 rpm, 8 stage compressor with back-to-back impellers. The compressor had a 163 cm (64 inch) bearing span with a critical speed of 3800 cpm and a rigid bearing critical of 4300 cpm. The suction pressure was 10.3 bars (150 psi) and the discharge pressure 34.5 bars (500 psi). The complex wave (figure 2) shows that the instability component at 4300 cpm increased from 25.4 μm to 101 μm (1 mil to 4 mils) over about a 1 second interval, and then sharply increased to 406 μm (16 mils) in approximately 0.2 seconds. The vibrations then shifted to 6000 cpm and then locked in on 4300 cpm (406 μm, or 16 mils) until the compressor speed was below 4000 rpm. The inboard vertical probe had slightly different characteristics, emphasizing the need for full instrumentation. This compressor failed eight times due to these nonsynchronous vibrations. The seals and labyrinths were wiped in an increasing bow pattern such that the inner labyrinths had approximately 2.5 mm (0.10 inches) of material removed. The vibration orbit was so circular that the pieces appeared to have been turned in a lathe.

Several modifications were required to improve the machine's stability characteristics. 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 installed recommended changes were sufficient; and the machine has run for several years without further nonsynchronous vibrations.

TORSIONAL-LATERAL COUPLING EFFECTS ON STABILITY

A recent study involving a complex centrifugal compressor train reveals considerable evidence that the torsional natural frequency of the system coincided with an unstable vibration mode of the fourth stage compressor and contributed to the failures encountered. Others have discussed this problem, but little experimental data is available on large industrial units (refs. 5, 6, and 7).

The high speed compressor (17000 rpm) could not be operated above 70 percent load because the lateral vibrations would suddenly increase to destructive levels whenever the load increased. Vibrations as high as 101-127 μm (4-5 mils) occurred when the instability was excited, resulting in seal wipes, bearing failures, and shaft scoring.

In addition to the high lateral shaft vibrations on the fourth stage compressor, several gear failures were experienced in the intermediate and pinion gears. A complete investigation of the system, requiring extensive testing, was made to determine the cause of the problem and to evaluate the modifications. The data obtained illustrates the influence of the torsional natural frequency upon compressor instabilities and lateral vibrations in the gearbox. In addition, the influence of the lateral vibrations of the shafts in the gearbox upon compressor vibrations will be documented.

The system consists of a gas turbine, steam turbine, two gearboxes, and five centrifugal compressors (figure 5). The gas turbine gearbox has a bull gear, two intermediate (idler) gears, and two pinions. One pinion drives the first and second stage compressors in a back-to-back arrangement on opposite sides of the pinion. The other pinion drives the third, fourth, and fifth stage compressors; the third stage compressor is on one side and the fourth and fifth stage on the other side. Directly in line with the bull gear is another gearbox with an auxiliary steam turbine. This system is rigidly coupled between the pinions and compressors as opposed to having gear couplings, which caused lateral vibrations to be transmitted throughout the entire system.

The compressors were instrumented with proximity probes; however, to completely evaluate the instabilities in the fourth stage compressor and gear tooth failures, additional proximity probes were installed in the gearbox to measure lateral vibrations. In addition, three FM torsiographs were installed in the gearbox (figure 6), one monitoring the bull gear and one on each of the pinions. The FM torsiograph measures torsional vibrations by monitoring the gear tooth passing frequency signal from a magnetic pickup or proximity probe and demodulates this signal using a frequency-to-voltage converter. The resulting demodulated signal can be frequency analyzed to obtain the torsional velocity vibrations in the system.

The unstable lateral vibration characteristics of the compressor can be seen in figure 7 which gives the outboard horizontal vibrations in a raster plot or Campbell diagram presentation. This Campbell diagram was made directly in the field with a real time analyzer and appropriate electronic instrumentation. The instability vibrations at 4800 cpm are only about 13 μm (0.5 mil); however, the gas turbine speed had been lowered to 3460 rpm (compressor speed of 16400 rpm) to keep the instability amplitude from tripping the unit. It can be seen that the amplitude of the instability near 4800 cpm reduces as the speed is lowered. Both reduction in speed and reduction in load caused the instability amplitude to decrease. Vibrations at the same frequency (4800 cpm) as measured in the fourth stage compressor also occurred in other compressors in the train even though no apparent excitation source exists in the train at that frequency. The horizontal vibration of the third stage compressor (figure 8) shows vibrations at 4800 rpm which was the instability frequency that was tripping out the fourth stage compressor.

The torsional natural frequencies of the train can be seen from the Campbell diagram generated from the bull gear torsiograph signal (figure 9). The first three torsional critical speeds occurred at 1155 cpm, 1590 cpm, and 4760 cpm. One interesting phenomenon to note is that the first and second torsional critical speeds are always present, and modulate with fairly high amplitudes. Note that the third torsional natural frequency at 4800 cpm corresponds with the lateral instability frequency of the fourth stage compressor.

The maximum amplitude measured over the entire running speed range is recorded by the peak store envelope (figure 10). Figure 11 illustrates the correlation of the natural frequencies measured from the pinion gear torsiograph. The torsiograph on the pinion gives the same frequency information and also shows modulation of the first and second torsional critical speeds.

The fourth stage compressor originally had pressure pad bearings. At this high speed of 17000 rpm, it was suspected that a change to tilted pad bearings would be sufficient to solve the instability problem; however, to ensure that the system would still operate satisfactorily under design loading conditions, stability calculations were made for the original system and the system with tilted pad bearings. Also the effect of the change to tilted pad bearings upon the lateral critical speed response was investigated. The critical speed map was generated and forced vibration analyses were performed to verify that the location of the new critical speeds would be acceptable. The critical speed map in figure 12 has the presure pad and tilted pad bearing curves superimposed. The new bearings were four-shoe, load-between-pad bearings. The new system critical speeds are near the horizontal critical speeds for the original system. The compressor running speed is above the fourth critical speed; therefore, the change to tilted pad bearings would not definitely solve the problem since there are many systems which have tilted pad bearings and still have instabilities.

It has been found that to determine if proposed bearing modifications will be satisfactory from a stability standpoint, it is necessary to analyze the system for the fluid or aerodynamic loading that the compressor will be experiencing. To ensure that an adequate range of loading is covered, normally the log decrement versus effective aerodynamic loading is evaluated as plotted in figure 13. The lowest calculated instability forward mode near 4600 cpm agreed with the 4800 cpm which was actually measured (figure 7). The compressor stability is lowered as the effective aerodynamic cross coupling loading increases. With the new bearings, significant improvement in the log decrement is apparent and the tendency toward instability as a function of load is not evident. This calculated stability data indicates that the unit should be stable. The field data supports this analytical prediction, as can be seen from figure 14 which shows that as the gas turbine speed increases above 3000 rpm, no instabilities were found. Under maximum operating conditions, maximum pressure, and rated flow, vibrations in the low frequency range were carefully examined for all types of incipient instabilities and none was found. Therefore, from a stability standpoint, the changes made in the unit were satisfactory.

One interesting phenomenon that occurred on startup was the excitation of lateral vibrations in the fourth stage compressor due to the idler gear frequency excitation as it passed through the compressor shaft critical speeds. In figure 15 (the peak-store plot of the compressor shaft vibrations, as compressor speed went from 5700 rpm to 13300 rpm), the excitation of the shaft criticals at 3000-4450 cpm can be seen. These measured critical speeds match those determined from the critical speed map. On this initial startup, the high amplitude, low frequency vibrations measured on the fourth stage compressor were unexpected due to the predicted improvement in the stability characteristics for the rotor with the tilted pad bearings. When the experimental Campbell diagram of the vibrations was displayed, it was found that the instability was not the classical type of instability. The low frequency vibrations were a result of the idler gear frequency exciting the lateral critical speeds of the fourth stage compressor. The cause of the excitation was the lack of steady-state loading on the gears in the gearbox. Subsequent runs were made with higher suction pressures which significantly reduced the idler gear excitation. These characteristics were shown in figure 14 for increased suction pressure. These data show that in closely coupled systems the lateral energy can be transmitted throughout the train.

COUPLING OF TORSIONAL VIBRATIONS INTO LATERAL VIBRATIONS

The following data illustrates the coupling of torsional vibrations into lateral vibrations throughout the compressor train. While it is felt that the rigid couplings greatly influence the magnitude of the coupled torsional to lateral amplitudes, the mechanism which causes the increased lateral vibration at torsional natural frequencies is the same for all rotating equipment. The important factor to be stressed is that the coupling of the torsional vibrations into the entire system can also serve as an instigator of instability. Therefore, it is vitally important to design systems which are free from gear excitation which might coincide with the torsional natural frequency of the system or the instability frequencies of individual shafts in the system. The interaction that occurred between the torsional and lateral vibrations for this system is discussed below.

For the pinion driving the fourth stage compressor, the Campbell diagram in figure 16 shows that whenever one of the excitation sources matches a torsional natural frequency, the lateral vibrations drastically increase. The energy from the idlers can couple directly into compressor shaft lateral vibrations; therefore, it can be surmised that any time the idler gear, bull gear, or one of the excitation sources is coincident with the torsional natural frequency, increased lateral vibration would occur and this could be transmitted to the compressors.

In addition to presenting the data in the form of the Campbell diagram, order tracking was performed to separate the vibration components in the pinion vibration. Figure 17 gives the lateral vibrations at the bull gear, the pinion, and the idler gear frequencies present on the pinion over the speed range. Notice that whenever the idler gear frequency matches the torsional natural frequency at 4800 cpm, there is a sharp increase in the amplitude. The two idler gears had slightly different frequencies; therefore, there was a modulation of the amplitude of the lateral vibrations on the high speed pinion when tracking this frequency. This amplitude increase can be directly transmitted to the fourth stage compressor and serve as an instigator for the instability frequency.

The Campbell diagram for the other pinion lateral vibrations (figure 18) shows that the torsional energy is transferred laterally throughout the train. Again there are similar characteristics on the pinion with large responses at the first and the third torsional critical speeds, indicating that this excitation can cause large lateral vibrations.

The effect on the idler gear horizontal vibration is shown in figure 19. This is a time raster rather than a Campbell diagram. The increase in lateral vibration can be seen when the idler gear frequency approaches 4800 cpm. This data is pertinent relative to the two problems experienced: the instability and the gear tooth failures. Notice that every time one of the excitation frequencies is coincident with the torsional natural frequency, the torsional vibrations and the lateral vibrations increase. This energy is transmitted through the shafts to the first and second stage compressors, as can be seen in figure 20 which gives the second stage compressor inboard vibrations. There were large vibrations excited even on the second stage compressor; however, this compressor also had a lateral critical speed near 4800 rpm which may have caused some amplification. This data shows that if a compressor had a potential instability, it would be easy for the torsional resonances to act as perpetrators for instabilities.

For this system with the new tilted pad bearings, the stability was significantly improved. The new instability mode frequency was out of the range of the idler gear and bull gear excitations; therefore, this system was completely stable over the range of operations.

CONCLUSIONS

These field investigations have served to provide insight into potential destabilizing instigators and sensitive operating conditions for high speed rotor systems. Several useful guidelines have been set forth for evaluating new designs of high speed rotor systems as well as existing systems with chronic stability problems.

(1) Torsional vibrations in a system can serve as an instigator for an instability; therefore, it would be important in designing systems not to have the torsional natural frequency of the system coincide with potential unstable vibrating modes of a centrifugal compressor.

(2) In rigidly coupled systems, excitation sources in the gearbox should not match potential instability natural frequencies since this could serve as an exciting mechanism for the instability.

(3) In systems where an instability mode could occur near a torsional natural frequency of the system, a potential exists for coupling of the vibrations into the gearbox to increase the dynamic loads on the gears and in some cases can cause gear tooth failures.

(4) If gear failures are experienced in a gearbox, it is standard procedure to check for torsional natural frequencies in the system which can amplify the dynamic loads. The system should also be checked for unstable vibrations on one or more of the compressors which may be causing increased dynamic loads through the torsional-lateral coupling mechanisms as demonstrated in this example.

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FIELD INSTRUMENTATION

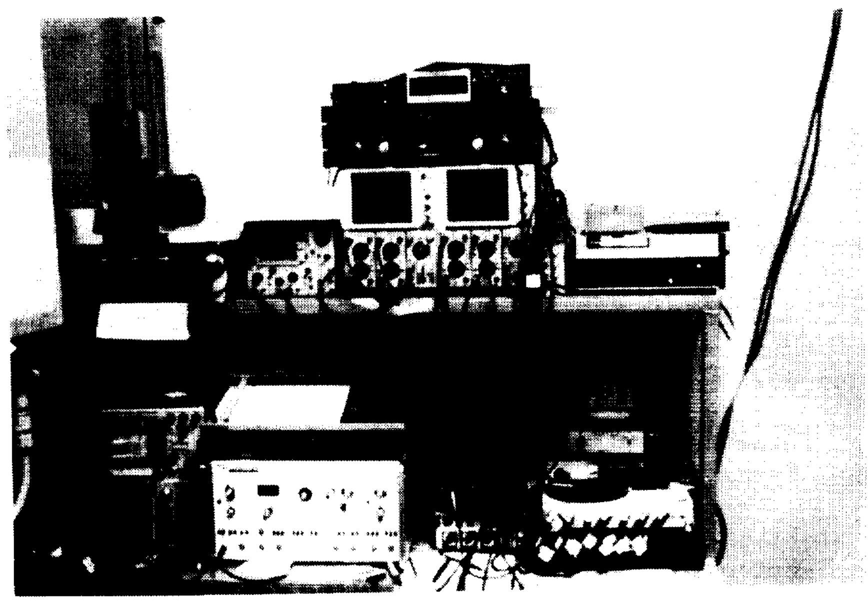


Figure 1

COMPLEX WAVE PRESENTATION OF COMPRESSOR INSTABILITY

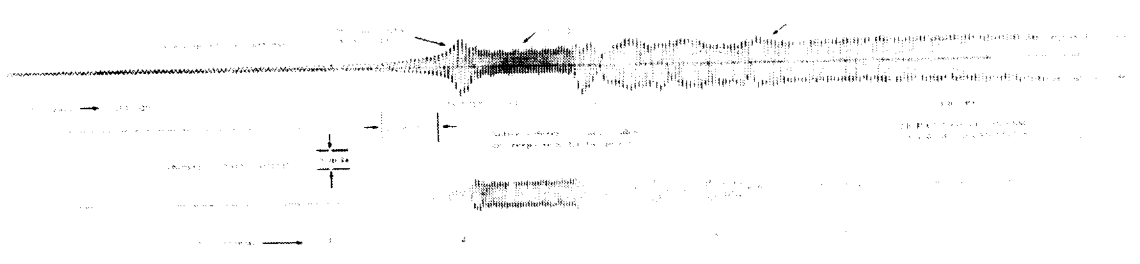


Figure 2

SPECTRAL TIME HISTORY OF COMPRESSOR
TRIPOUT SHOWING INSTABILITIES

254 μm/div (10 mils/div), 0.16 sec/line

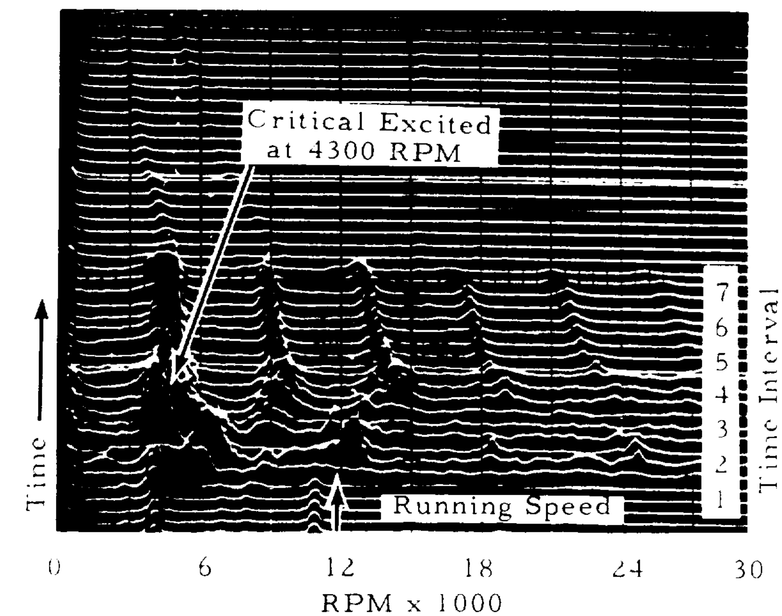


Figure 3

SPECTRAL TIME HISTORY GENERATED
BY FIBER OPTICS RECORDER

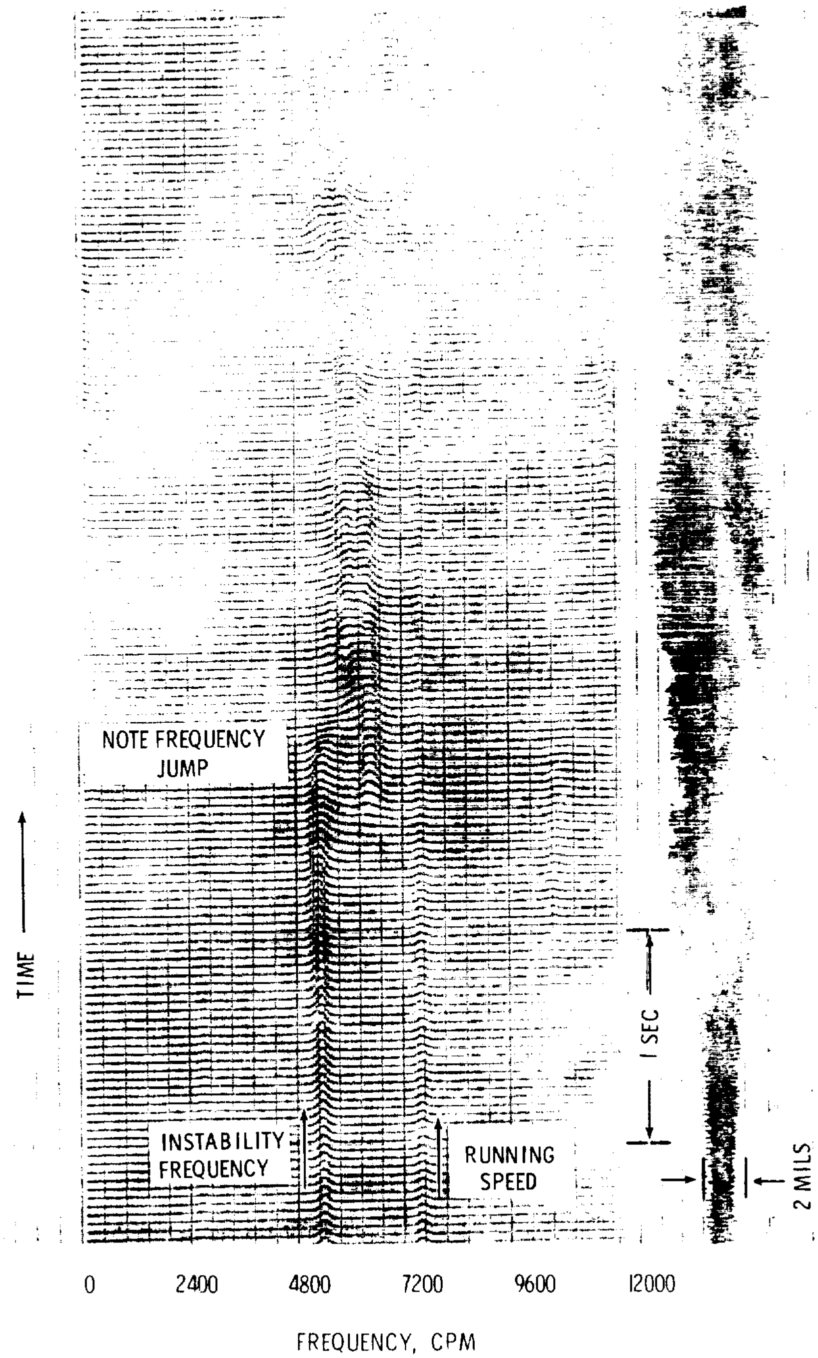


Figure 4

SCHEMATIC OF COMPRESSOR TRAIN

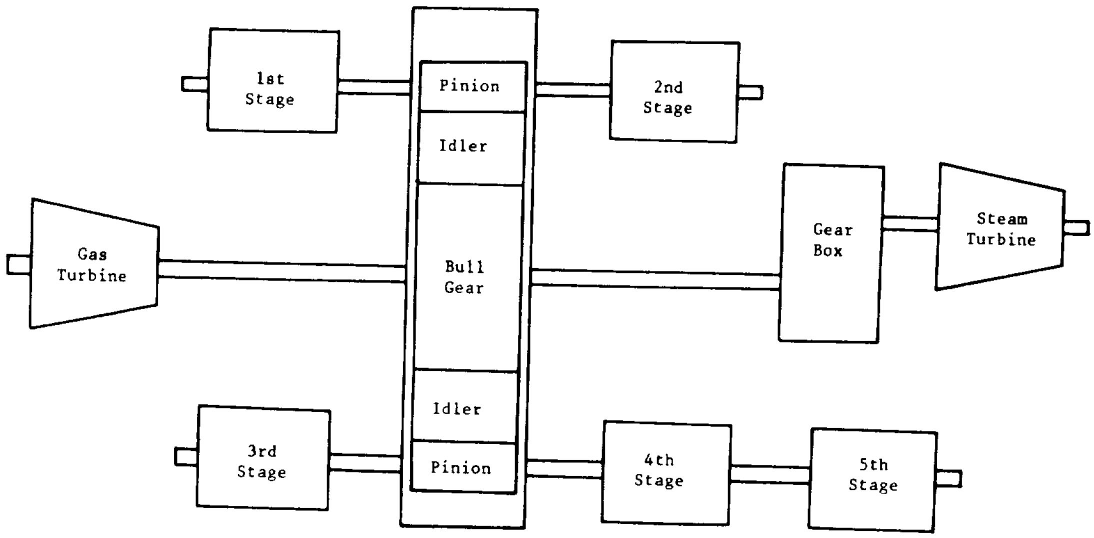


Figure 5

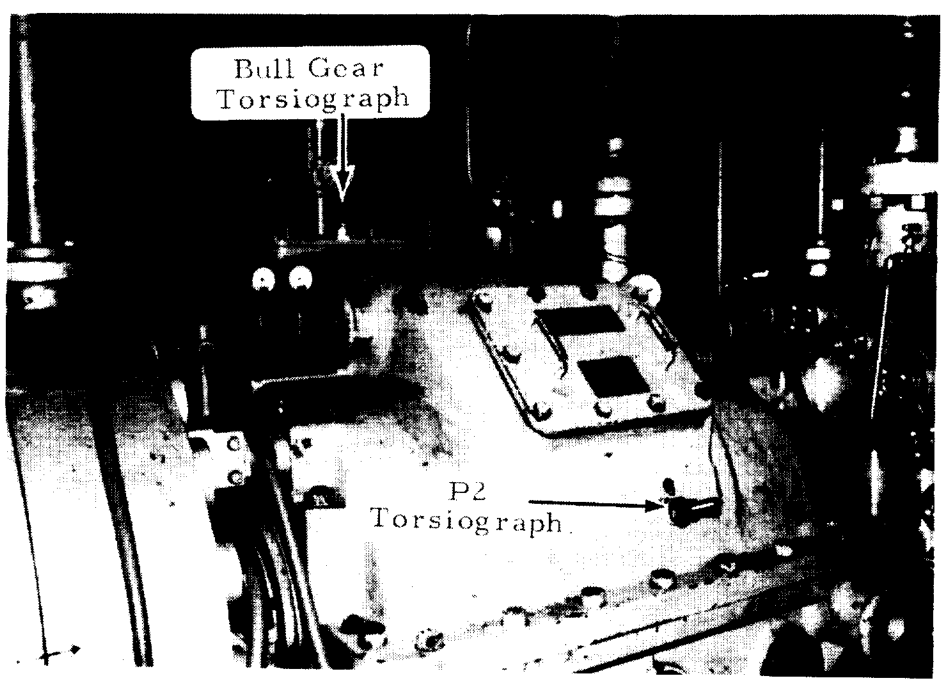
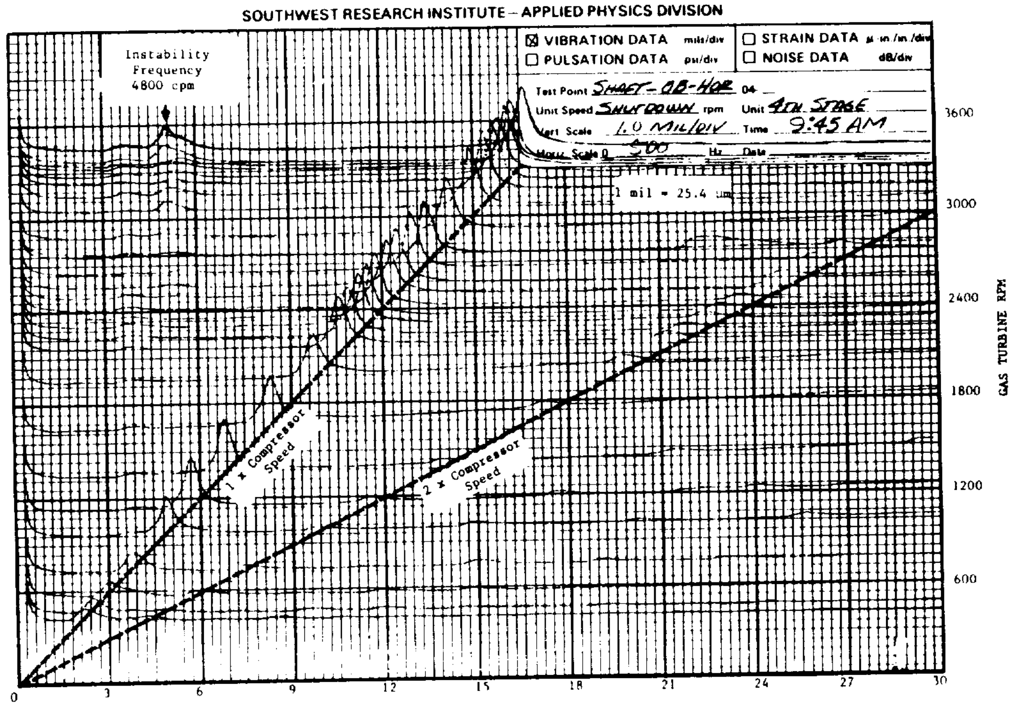


Figure 6

CAMPBELL DIAGRAM
OF FOURTH STAGE COMPRESSOR VIBRATIONS SHOWING INSTABILITIES

SOUTHWEST RESEARCH INSTITUTE – APPLIED PHYSICS DIVISION

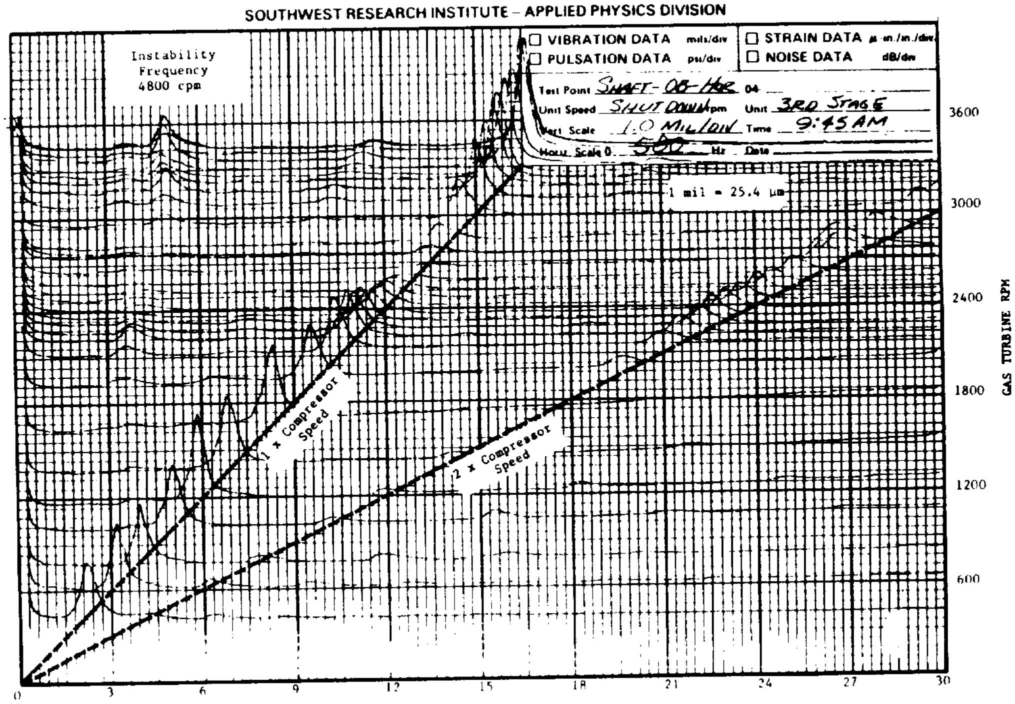


CPM IN THOUSANDS

Figure 7

LATERAL VIBRATIONS OF THIRD STAGE COMPRESSOR

SOUTHWEST RESEARCH INSTITUTE – APPLIED PHYSICS DIVISION



CPM IN THOUSANDS

Figure 8

BULL GEAR TORSIONAL VIBRATIONS

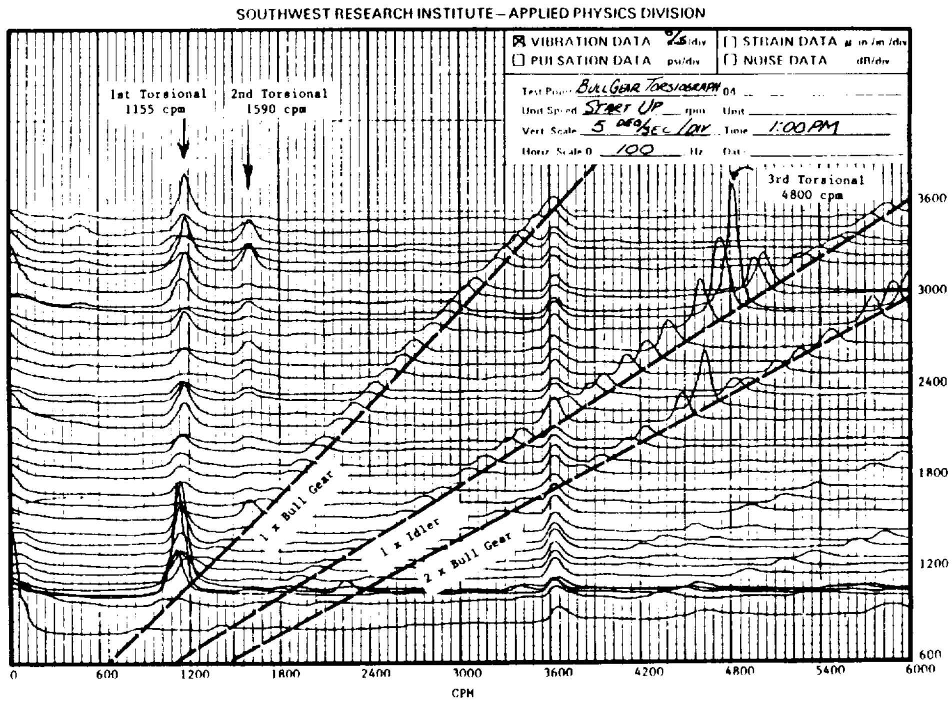


Figure 9

PEAK-STORE BULL GEAR TORSIONAL VIBRATIONS

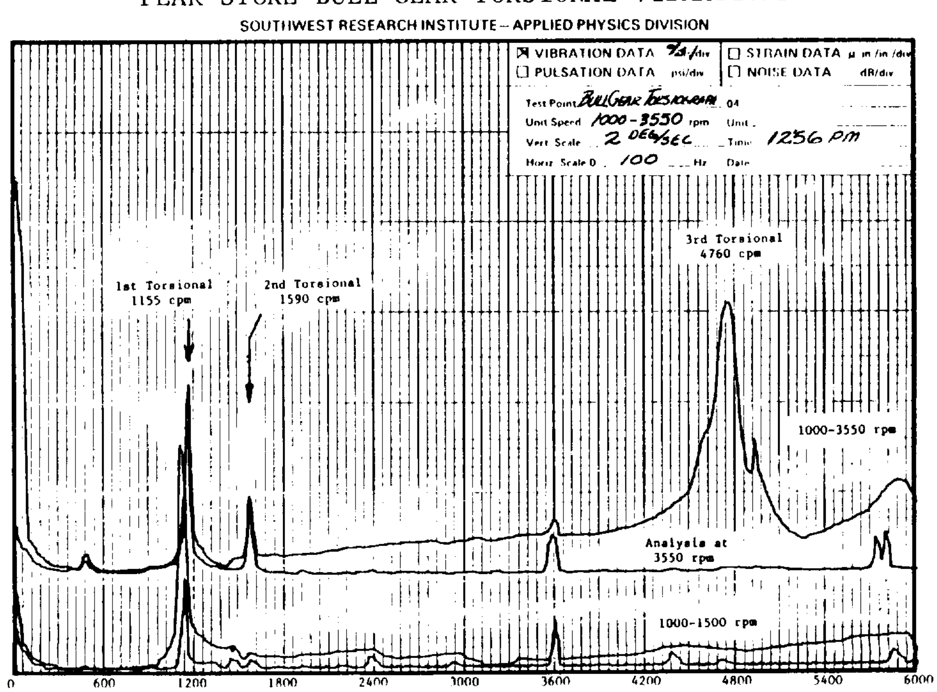


Figure 10

TORSIONAL VIBRATIONS MEASURED ON PINION

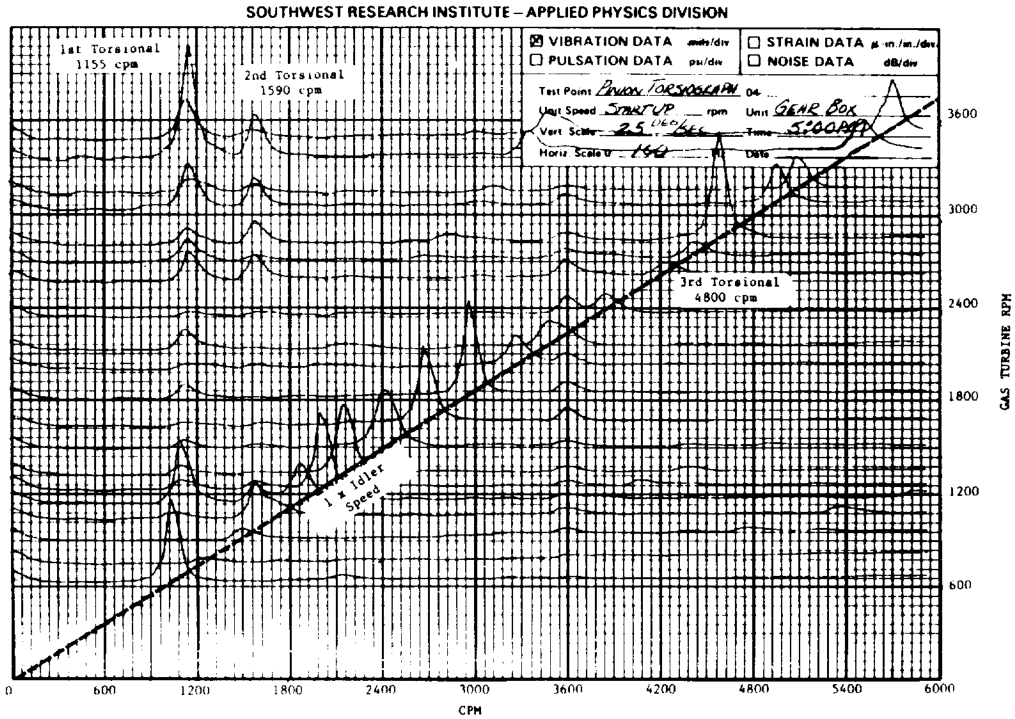


Figure 11

CRITICAL SPEED MAP

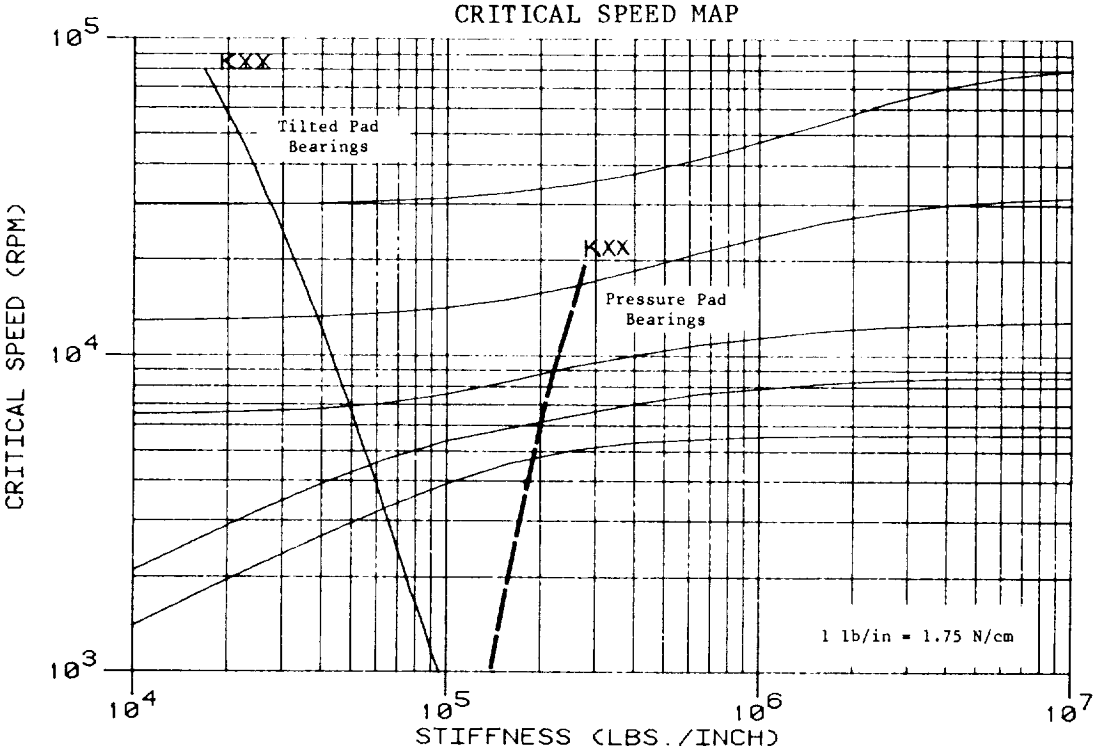


Figure 12

EFFECT OF AERODYNAMIC
CROSS COUPLING ON STABILITY

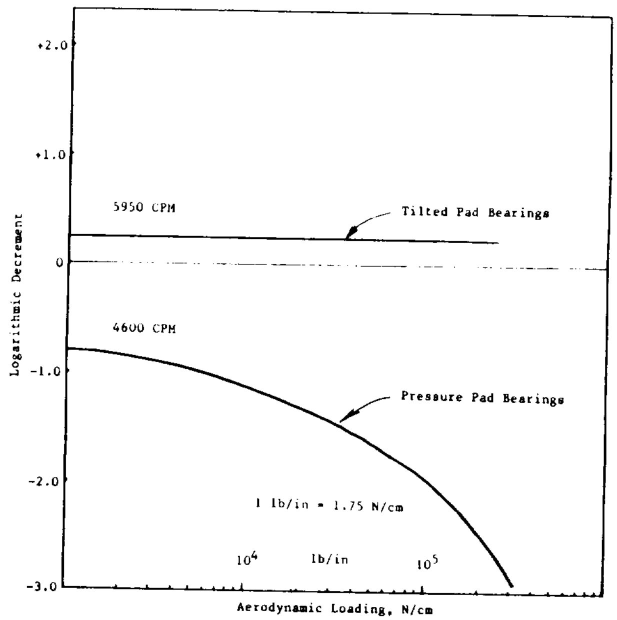


Figure 13

FOURTH STAGE COMPRESSOR VIBRATIONS
WITH TILTED PAD BEARINGS

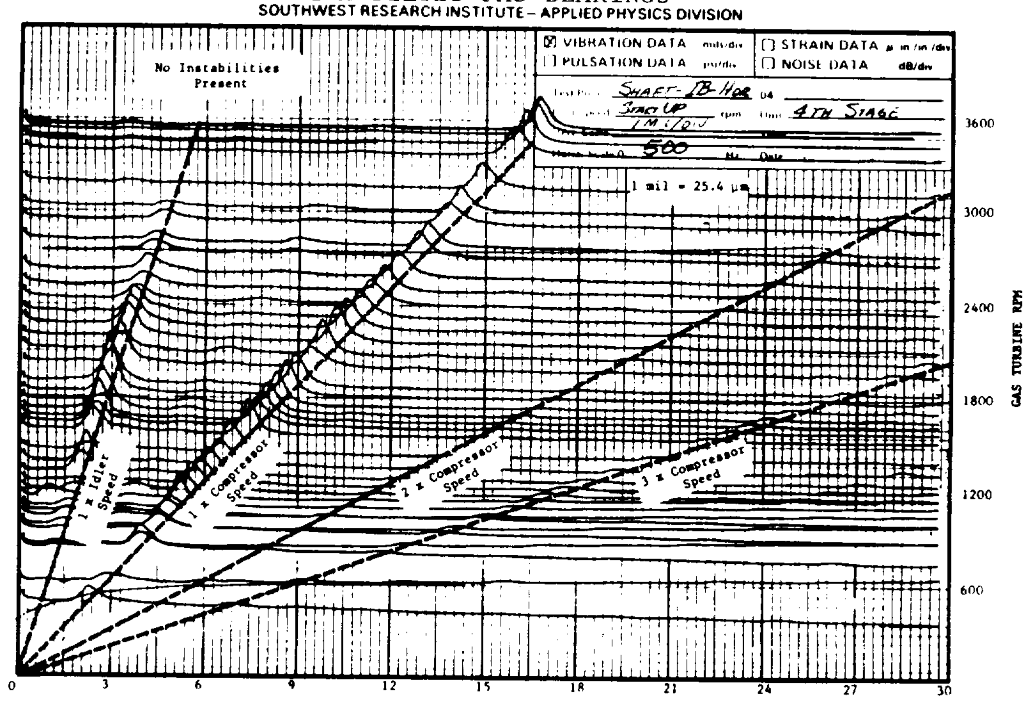


Figure 14

COMPRESSOR SUBSYNCHRONOUS VIBRATIONS
EXCITED BY IDLER GEARS

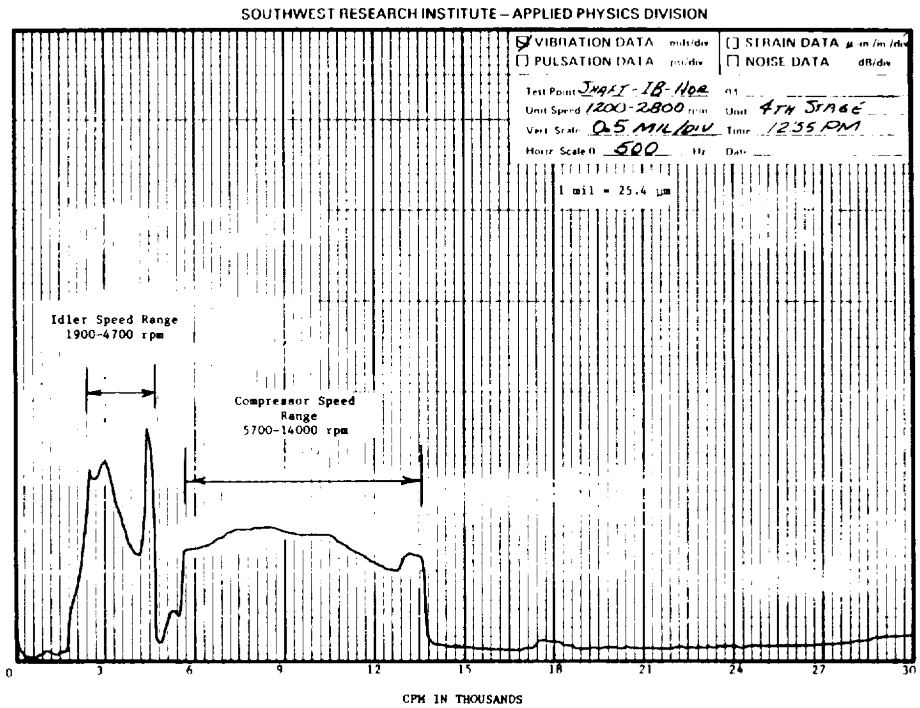


Figure 15

PINION LATERAL VIBRATIONS
CAUSED BY TORSIONAL-LATERAL COUPLING

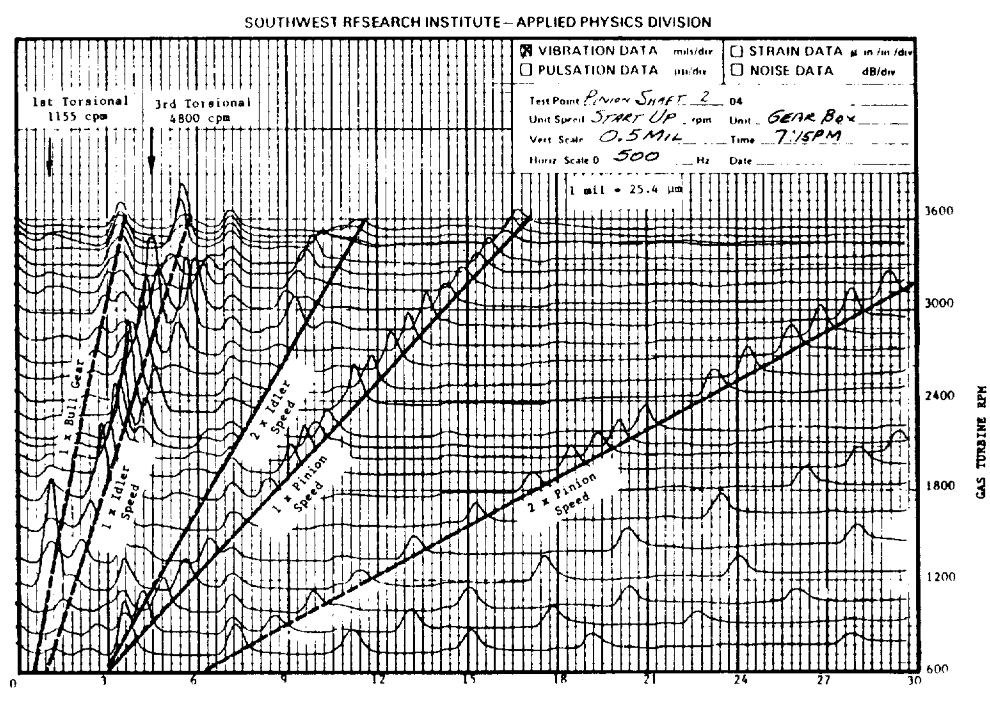


Figure 16

ORDER TRACKING OF PINION VIBRATIONS

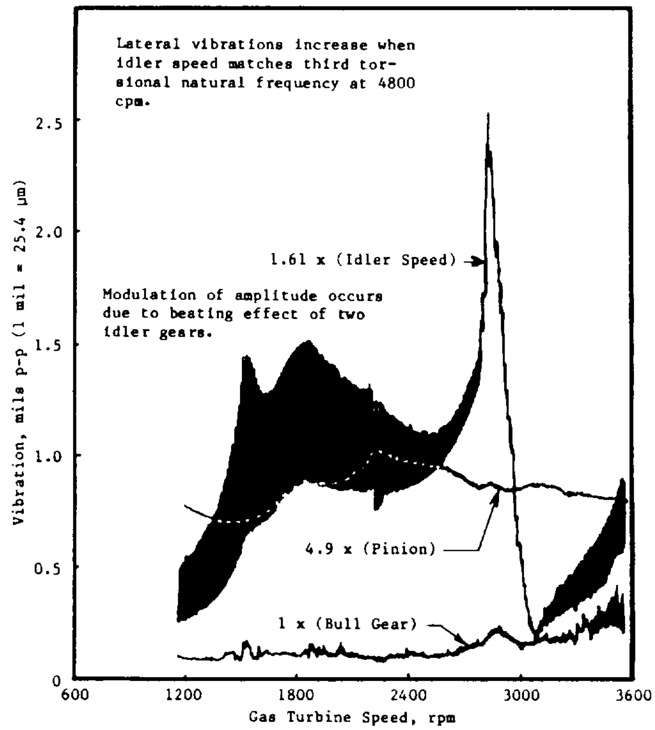


Figure 17

FIRST STAGE PINION LATERAL VIBRATIONS

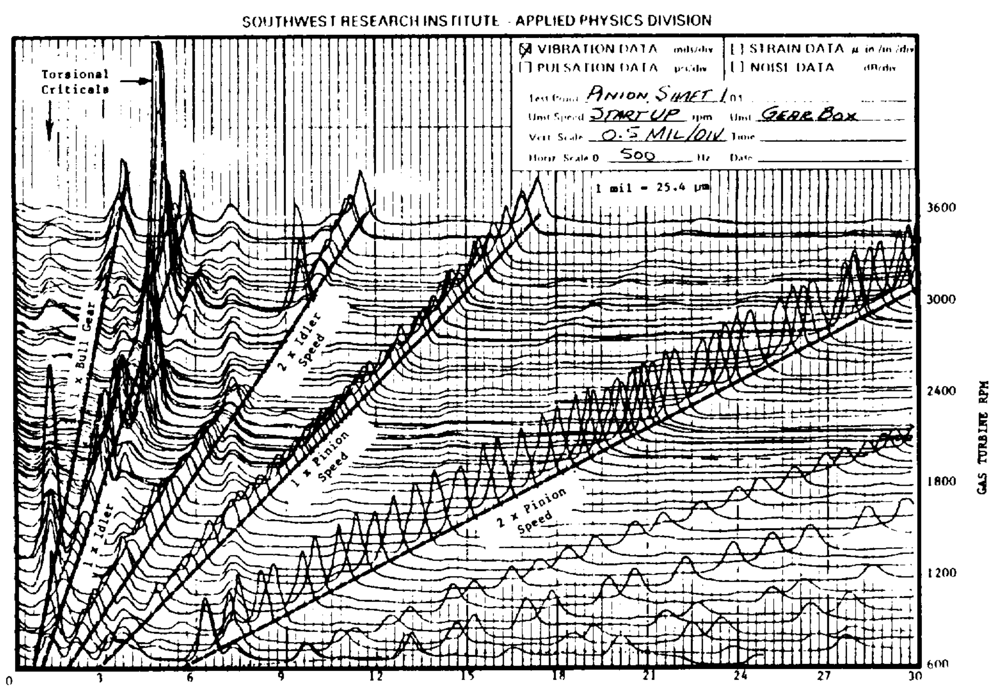


Figure 18

IDLER GEAR LATERAL VIBRATIONS

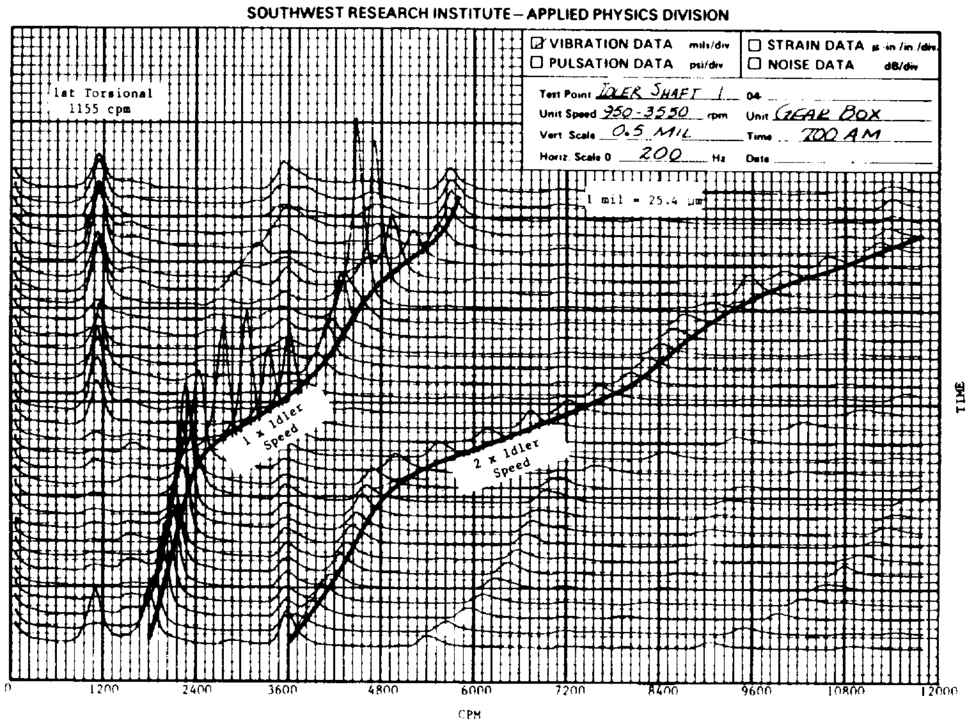
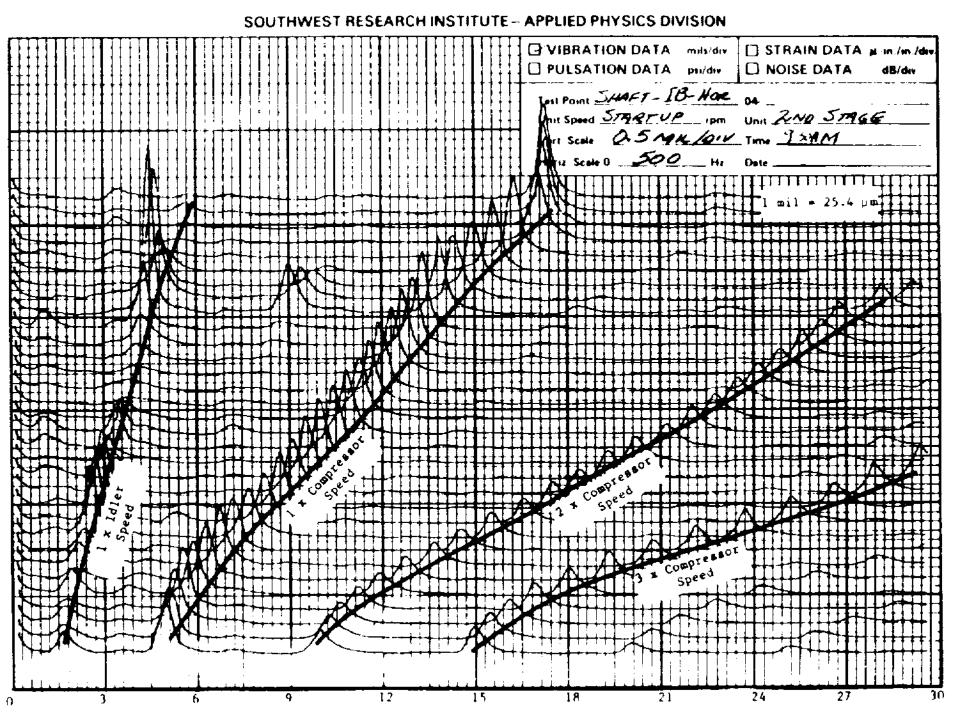


Figure 19

SECOND STAGE COMPRESSOR VIBRATIONS



CPM IN THOUSANDS

Figure 20