INSTABILITIES OF GEARED COUPLINGS - THEORY AND PRACTICE

R. Gordon Kirk, Roy E. Mondy, and Richard C. Murphy
Ingersoll-Rand Company
Phillipsburg, New Jersey 08865

SUMMARY

The use of couplings for high speed turbocompressors or pumps is essential to transmit power from the driver. Typical couplings are either of the lubricated gear or dry diaphragm type design. Gear couplings have been the standard design for many years and recent advances in power and speed requirements have pushed the standard design criteria to the limit. Recent test stand and field data on continuous lube gear type couplings have forced a closer examination of design tolerances and concepts to avoid operational instabilities. Two types of mechanical instabilities are reviewed in this paper: (1) entrapped fluid, and (2) gear mesh instability resulting in spacer throw-out onset.

Test stand results of these types of instabilities and other directly related problems are presented together with criteria for proper coupling design to avoid these conditions. An additional test case discussed shows the importance of proper material selection and processing and what can happen to an otherwise good design.

INTRODUCTION

In many instances of pre-order coupling selection, little consideration is given to either the dynamics of the coupling spacer as an individual body or its half weight effects on the shaft ends of the coupled units. Some efforts may be made to minimize weight but typically standard couplings are selected from a horsepower-speed aspect. This method is adequate in most cases because the spacer behaves as a rigid body with minimal influence on the connected shaft ends. It is this minimal influence on adjoining shaft ends and lack of response cross-coupling across the spacer which enables a single body analysis to accurately represent the individual unit's characteristics in the field.

The general technique for studying spacer tube dynamics is to consider it as a uniform beam with simple supports at each gear mesh as a worst case in the untorqued condition. As the torque level being transmitted through the spacer increases, the theory allows the coulombic gear mesh forces to restrain the spacer tube ends from rotating and effectively raise the spacer's critical speed. For some equipment strings the gear coupling's spacer tends toward this behavior. However, on units with flexible shaft ends, light journal loads, and large shaft end separation, the spacer cannot necessarily be considered to adhere to this proposed theory.

The amount of torque which a gear coupling is transmitting at a particular speed not only affects the amount of moment restraint which the adjoining shaft ends may induce upon the spacer but also greatly influences a mesh instability phenomena known as spacer throwout. Spacer throwout occurs when the mass unbalance forces on the spacer become sufficiently great as to cause the spacer to move off of the gear mesh pitch line creating a step function increase in the unbalance forces being exerted on the adjoining shaft ends.

In attempts to minimize the potential for throwout problems, the tendency is to specify a coupling with the lightest spacer possible which is capable of transmitting the required torque. Employing this philosophy may result in a spacer tube calculated critical falling near design speed. If the spacer tube critical is at least 120% of design speed, then satisfactory operation should result according to Staedeli and Vance (ref. 1 and 2). On large center hung machines with short, stiff shaft ends, this criteria may be acceptable. However, on machines where the shaft ends are flexible and the coupling half weight is a substantial percentage of the total journal loading, then the 120% criteria can result in unacceptable system, lateral dynamics.

In order to alleviate this potentially problematic coupling - shaft end configuration, the tendency is to increase both inside and outside spacer tube diameters while decreasing concentricity tolerances and gear mesh clearances. The result is a special design of coupling which may be significantly more expensive than the original, standard coupling would be. Although this coupling design would eliminate the amplification effects associated with operating near the spacer tube critical and reduce the maximum displacement possible due to throwout, the increase in spacer tube weight may nearly offset the gains of the coupling design change. Therefore, if the coupling is not properly specified, one set of dynamics problems may be simply traded for another set of problems. One major problem which may exist in this increased diameter spacer design is that of entrapped coupling lubricant within the spacer tube. Although this type of phenomena is documented in the literature (ref. 3,4,5) it can be difficult to diagnose during field operation due to limited vibration data acquisition capability. Another more important fact is that the vibration characteristics may not match the reported theories in all regards as this paper will illustrate. Therefore, efforts to preclude oil from entering the spacer should be included as part of the coupling design.

The remainder of this paper documents several case histories of turbomachinery vibration problems wherein the source of the problem was concluded to be inherent in the coupling design selected for each particular application. Further discussion on the theories and methods of calculations for coupling selection and analysis follows the review of the test results. Design standards are given and recommendations are made to assure the best possible dynamic response characteristics for high speed couplings.

REVIEW OF TEST STAND DATA RELATING TO GEAR COUPLINGS

The following case histories exemplify the basic problems described above. Each case history spotlights a particular problem encountered and what actions were required to correct the problem. Since all corrective actions were made on the coupling itself and none were made on either the test driver or driven unit, all problems were considered to be initiated by the coupling's dynamic characteristics and its interaction with the adjoining units.

Case 1 - Spacer Critical Speed and Balance

The first coupling to be considered is shown in Figure 1. The shaft end separation is 1727.2 mm (68 inches) and the coupling weight is 79.1 kg (174 pounds). The spacer is 149.2 mm (5-7/8 inches) O.D., 136.5 mm (5-3/8 inches) I.D. This coupling is rated for 11.6 MW (15,500 horsepower) at 95 Hz (5700 RPM) with a maximum continuous operating speed (MCOS) of 100 Hz (6000 RPM). The calculated spacer critical speed is 131 Hz (7860 RPM) or 1.31 times MCOS. In the field this coupling transmits power from an overhung design expander to a center slung design axial flow compressor. The compressor was tested using this coupling and at 100 Hz (6000 RPM), the compressor's bearing vibration adjacent to the coupling was approximately 0.025 mm (1 mil), Figure 2. This same coupling was then used to drive the expander for its mechanical run test.

The set-up for the expander test is shown schematically in Figure 3 where the variable speed motor drives the expander through a 5:1 speed increasing gearbox. The initial results of this test are depicted in Figure 4 which represents the initial run up to speed. As can be seen, vibration levels are acceptable at 83.3 Hz (5000 RPM), approximately 0.010 mm (0.4 mil), but the amplitude rises sharply above 83.3 Hz (5000 RPM) approaching 0.051 mm (2.0 mils) at 91.7 Hz (5500 RPM). The amplitude was nearly all at synchronous frequency. This rapid increase in synchronous response amplitude is usually indicative of a critical speed. A review of the rotor dynamics analysis of the expander, including the coupling half weight, did not indicate a critical speed in this speed range. Upon running an analysis of the test system, the indication was that a system critical existed which was primarily controlled by the coupling. It was necessary to field balance this coupling placing balance weights at the spacer midspan to enable a satisfactory vibration level to be achieved. The field balance required numerous trials due to the sensitive nature of the coupling spacer critical. Figure 5 shows the final balance condition with an acceptable level of 0.038 mm (1.25 mils) being achieved at 100 Hz (6000 RPM). The existence of the critical speed is indicated by the substantial shift in phase angle above 91.7 Hz (5500 RPM).

Case 2 - Gear Tooth Damping

Several proposals were considered to resolve the vibration problems of Case 1. One proposal was to coat the coupling gear teeth with a product which the coupling manufacturer recommends for the running in of new couplings. This coating is a polyphenylene sulfide resin placed on the flanks of the teeth to help the teeth share the load. After a reasonably acceptable level of balance was achieved by field balancing, this coating was applied to the coupling teeth. Figure 6 depicts the results. With the coating on the coupling, the vibration as measured by the expander's coupling end vibration probe reached .076 mm (3.0 mils) at 91.7 Hz (5500 RPM) as seen on Curve 1. The coating was removed by a light sandblasting; the balance of the coupling was considered to be unchanged. Curve 2, showing 0.025 mm (1.0 mil) at 91.7 Hz (5500 RPM), reflects the results upon rerunning the coupling on the test stand. Since the only change between these two runs was the removal of the coating, it was concluded that the difference in response was due to an increase in the coulombic damping in the gear meshes which resulted from the removal of the slippery coating that had been applied to the teeth flanks.

Case 3 - Entrapped Oil

Since test stand field balancing of the coupling was not considered as an acceptable permanent fix, an investigation of prior experience with similar configured machinery was undertaken. One result of this investigation was the discovery that on other jobs the ratio of calculated lateral critical speed of the coupling floating member to maximum continuous operating speed was substantially greater than on this application. This ratio was found to vary from about 1.9 to 6.3. It was concluded from this data that for a permanent fix, a coupling was required wherein the spacer critical to MCOS ratio would have a minimum value of 2.0.

The coupling design resulting from this study is shown in Figure 7. The shaft end separation is still 1727.2 mm (68 inches), but the weight has been increased to 97.7 kg (215 pounds). The main spacer has an O.D. of 235 mm (9.250 inches) and an I.D. of 223.9 mm (8.813 inches). The calculated critical speed is 207.6 Hz (12,457 RPM) or 2.08 times an MCOS of 100 Hz (6000 RPM).

In order to confirm the acceptability of this coupling, it was run on the next similar unit to go on the test stand. Figures 8-a through 8-d depict the initial results. An extremely high level of vibration would suddenly occur with a very small increase in speed. As can be seen from a chronological review of the figures, the speed at which this phenomenon would occur was lower with each successive start and the source of the high amplitude was a subsynchronous component varying from .94/rev. on the first start to .83/rev. on the fourth start. Although the amplitude at synchronous frequency was never higher than 0.015 mm (0.6 mil), the subsynchronous component reached levels approximating 0.241 mm (9.5 mils) at 84.8 Hz (5090 RPM) on the first start.

The first step in disassembly of the unit was the removal of the coupling. Upon breaking the flanges, a substantial quantity of oil was observed to have been entrapped in the spacer tube. In order to permit the oil to escape from the spacer, twelve 6.35 mm (0.250 inch) holes were drilled in the spacer; four at each end and four in the middle. The spacer assembly was check balanced and found to be acceptable. The unit was reassembled. Figures 9 and 10 show the results of the retest run. Figure 9 shows that the subsynchronous component, due to the entrapped oil, has been completely eliminated and Figure 10 indicates that the critical speed, previously seen at about 96.7 Hz (5800 RPM) has been eliminated. An additional modification has since been incorporated into this coupling design to preclude oil from entering the spacer from the gear mesh area. This modification consists of interference fitting dam plates in the ends of the spacer. The oil drain holes are retained.

Case 4 - Manufacturing Tolerances

Of necessity, couplings similar to that of Figure 7 consist of a number of pieces which must be assembled. Where possible, rabbet fits are used to maintain consistency in reassembly. However, fits may have clearance and may not be concentric with each other which can result in either constant or variable unbalance being manufactured into the coupling. This same problem can also exist for the fits and pilot diameter of the gear meshes.

The coupling of Figure 7 has a floating member weight of 84.1 kg (185 pounds) which results in an unbalance of 106.7 gm-cm (1.48 ounce inches) for each 0.025 mm

(1 mil) of floating member eccentricity. Figures 11 and 12 graphically depict what can happen as a result of manufacturing tolerances and eccentricities. Figure 11 indicates an overall amplitude of 0.081 mm (3.2 mils) at a speed of 96.7 Hz (5800 RPM). For this test run the coupling was assembled using the manufacturer's match marks. The assembled coupling was indicated with mechanical dial indicators and depending upon axial location, ran out as much as 0.127 - 0.152 mm (5-6 mils). When ignoring the manufacturer's match marks, and rotating components relative to each other, the run out was reduced to under 0.051 mm (2 mils). The test results for this configuration are shown in Figure 12 where the overall vibration level at 96.7 Hz (5800 RPM) is approximately 0.037 mm (1.45 mils).

Case 5 - Throwout of Mesh

As discussed in the introduction, gear type couplings require a minimim torque transmission in order to center the floating member. In the absence of this minimum torque, the residual unbalance in the floating member may cause it to move eccentric until the tooth tip clearance is reduced to zero in the heavy spot's radial direction. This situation creates an additional unbalance in the coupling.

This phenomena is graphically displayed in Figure 13a, where a speed reducing gear box is being driven by a gas turbine with the load compressor uncoupled. As long as the accelerating torque is applied, the vibration amplitude gradually increases to a level of 0.013 mm (0.5 mil) at 209.2 Hz (12550 RPM). Since the accelerating torque decreases as the unit approaches idle speed, the vibration quickly increases to 0.018 mm (0.7 mil) at 211.3 Hz (12680 RPM). When torque is reapplied through the coupling during acceleration to MCOS, the vibration amplitude immediately drops to .009 mm (0.36 mil) and gradually increases again as the driving torque decreases while the unit comes up to speed.

Figure 13b shows a separate test run for the subject equipment string. Both accel and decel signatures clearly show the response step increase associated with coupling spacer throwout. The speed-amplitude drift signature is included as additional information to show the absence of a response critical of any kind in this speed range.

Case 6 - Coupling Failure

Figure 14 depicts a peak hold plot of overall vibration during the startup of a unit similar to the one just described above. As can be seen, the amplitude starts rising rapidly above 120 Hz (7200 RPM) and peaks at about 0.330 mm (13 mils) at 160 Hz (9600 RPM). The unit was tripped and after coast down, the coupling guard was disassembled. Figure 15 shows the condition of the coupling spacer. Three large cracks were evident. The darkened area above the center balancing band was caused by the coupling rubbing the guard. The driven gear box showed no distress and the coupling was not transmitting load as the load compressor was not coupled for this start nor had it been coupled. Metallurgical investigation later determined that the cracks had been formed during the heat treatment cycle of the spacer.

THEORY AND GUIDELINES TO PROPER COUPLING DESIGN FOR ROTOR DYNAMICS CONSIDERATIONS

Entrapped Fluids

The analysis for the dynamics of a rotor partially filled with liquid was published by Wolfe (3,4) and Ehrich (5) in the mid-sixties in the U.S. Wolfe references earlier work from Europe dating to the late fifties. Other authors have more recently written on the subject of entrapped fluids (6,7) which refine the fluid dynamics and mathematics of the work by Ehrich. The published theory of entrapped fluids has been proven by experimental tests on vertical rotors wherein the system undergoes a zone of instability after passing a "reduced critical speed". This reduced critical is not a function of the fill ratio but is determined by calculating the system critical with an equivalent rotor mass. The equivalent rotor mass is found by adding the mass of the entrapped fluid required to fill the rotor full of liquid to the dry rotor mass. This theory indicates that higher and higher speeds may be reached as the rotor is filled with liquid before the instability onset occurs. The test stand data presented by Ehrich indicated the presence of a 0.87 x synchronous speed component that presented a speed zone of large response that the jet engine was able to pass through. The results of his analysis seemed to satisfy the test stand results of the jet engine for the one run he reported.

The results of the more extensive test runs with increasing volumes presented in this paper indicate that a mechanism of instability exists that produces a lowered threshold of instability as the volume of entrapped fluid increases and further that the whirl rate varies from almost synchronous for small volumes to 0.84 x synchronous as the volume increases. The difficulty in applying the simple theory to a real machinery situation is the identification of the critical of concern and the effective mass of the rotor. Additionally, the theory indicates a nonsynchronous component at 0.50 x synchronous for small volumes of entrapped fluid with whirl ratios increasing as the rotor is filled with more fluid. Wolfe noted that when his experimental rotor was unstable it whirled for the most part at frequencies below the reduced critical speed, the exact ratios not reported.

No attempt is made in this paper to apply the published criteria for instability onset since the nature of the observed instability differs from the theory as published to date. The practical solution is to prevent the admission of oil into a long spacer and further, to provide ample discharge outlets along the spacer length. The oil flow in gear couplings should not be oversimplified since the current design problem was considered by the manufacturer with the conclusion that it was impossible to get oil into the spacer. This indeed is logical if one does not consider axial pumping action or gear mesh oil ring instabilities that could increase the oil ring thickness. A minimum dam height at the spacer ends of double the normal gear mesh oil ring thickness is considered essential and preferably a solid blockage plate should be used on all coupling spacers that can entrap oil. The severity of the problem is amplified when lightly loaded coupling end bearings are present such as in the case of overhung expanders, turbines, compressors, or lightly loaded gear boxes.

When consideration is given to the potential imbalance from spacer I.D. nonuniformatives that can produce a nonuniform oil film in the spacer, the admission of oil into a spacer is quickly concluded to be highly undesirable.

Example 1

Consider the imbalance of 0.254 mm (10 mil) thickness of oil, 50.8 mm (2.0 inches) wide, along a 1524 mm (60.0 in.) spacer tube with an I.D. of 223.5 mm (8.8 in.), and an oil having a weight density of $\gamma = 0.0307$ lb/in³.

$$U = \gamma \times V \times D/2 = .0307 \text{ lb/in}^3 \times (2.0 \times 60.0 \times .010) \times (8.8) \times 16$$

$$U = 2.59 \text{ oz-in or } 186.7 \text{ gm-cm}$$

(A typical coupling component balance is made to a level of 0.1 oz-in).

Coupling Spacer Critical Speed and Amplification

Manufacturers of high speed couplings typically calculate a spacer lateral critical by considering it as a simply supported beam as the worst case (see Eq. 1). The general opinion is that the critical is typically higher due to the frictional constraint and damping arising from the torque condition for a gear coupling. This type of constraint is not present in a dry diaphragm type coupling.

For large center hung turbomachinery this calculation is typically adequate with the bearing reactions effectively giving a restraint to validate the approximate calculation. For lightly loaded coupling end bearing machinery, however, the coupling represents a large portion of the modal mass of the coupling end critical (typically the turbomachines 2nd mode) and hence the system sensitivity to imbalance on the coupling is important not only at the spacer critical but also at the rotor system criticals. The standard practice of using coupling half weights located at their respective c.g.'s is considered totally acceptable for machinery having the spacer critical ratio of approximately 2.0 or larger.

The analysis of turbomachinery having the coupling spacer critical speeds less than 2.0 times the maximum continuous operating speed should include a response analysis of the coupling joining the two adjacent rotor bearing systems. This type analysis is commonly referred to as a train or partial train analysis. The detailed train analysis results in the identification of the actual system critical speeds and response sensitivities. The justification for this is supported by the resulting amplification or deviation from a speed-square response as the spacer critical speed ratio N_{cr}/N becomes less than 2.0. This is shown in Table I where it is noted that a 33% deviation from a speed-square curve exists when $N_{cr}/N = 2.0$. A 176% deviation is indicated for $N_{cr}/N = 1.25$. This means that any residual imbalance force has been amplified by 176%. For $N_{cr}/N = 1.1$, an amplification of 425% would occur for the residual imbalance forces to the adjoining rotor bearing systems.

The proposed design procedure for preliminary selection of couplings is as follows:

1. Size coupling for torque capacity requirements.
2. Check placement of spacer Ncr by the following equation (American Units)

$$N_{cr} = \frac{4.73 \times 10^6}{L^2} \sqrt{\frac{D^2 + d^2}{1 + \left[\frac{9\ W_e}{L(D^2 - d^2)}\right]}} \text{ (RPM)} \qquad (1)$$

where

D = spacer nominal outer diameter, in.

d = spacer nominal inner diameter, in.

W_e = effective modal mass of flanges or torque meters referred to the spacer midspan, lb_m.

L = mesh to mesh distance, in.

Spacer is assumed to be of standard steel having a Young's Modulus, $E = 28.7 \times 10^6$ lb/in^2 and weight density, $\gamma = 0.283$ lb/in^3.

3. If the spacer is in the satisfactory region given in Figure 16 (Zones I and II, and the upper part of Zone III) for each of the adjoining machines journal reaction ratio, 1/2 Wcplg/Wjournal, the design specifications need not require a train analysis.

4. If the design falls in the marginal region of Figure 16, a train or partial train analysis should be performed for test stand and field conditions if the spacer cannot be redesigned to move into the satisfactory region.

5. If the design falls in the unacceptable region the coupling spacer should be redesigned to move the calculated critical as far as possible toward the satisfactory zone, analysis being made as noted above.

Coupling Throwout

The previous discussion on amplification is important to be able to fully appreciate the potential problems relating to the mesh throwout instability. According to the results given in reference (1), a gear coupling in perfect balance will center on the mesh pitch line when the torque transmitted is either equal to or larger than

$$T = M \times E \times \omega^2 \times D/4 \qquad (2)$$

where

M = mass of floating member

E = spacer eccentricity to be centered

D = mesh pitch diameter

ω = rotor speed, rad/sec.

This torque is typically very small for standard tooth tip clearances.

Example 2 (American Units)

Assume the following conditions:

E = .001 in. = .0254 mm

N = 6000 RPM; ω = 628.3 rad/sec.

$W = 160$ lb. $= 72.6$ Kg

$D = 9.0$ in. $= 228.6$ mm

Applying Eq. (2) and converting to horsepower gives

$$H.P. = \frac{2\ \pi\ T\ N}{33{,}000} = \frac{2\ \pi\ (W\ E\ \omega^2\ \frac{D}{4} \times 1/12) \times N}{33{,}000}$$

$H.P. = 17.52$ to center mesh at 6000 RPM.

Consider now the influence of the spacer imbalance on the centering action of the mesh. For a coefficient of friction of 0.15, it may be shown that the centering will not be maintained for an imbalance larger than

$$U_{T.O.} = \frac{1.07 \times 10^{10} \times H.P.}{D \times N^3 \times Q} \quad \text{(American Units)} \qquad (3)$$

where

- $H.P.$ = transmitted horsepower
- D = mesh pitch diameter, in.
- N = rotor speed, RPM
- Q = spacer amplification factor
- $U_{T.O.}$ = oz-in. imbalance to cause throwout

Example 3

- $H.P.$ = 500 = 372.5 kw
- D = 9 in. = 228.6 mm
- N = 6000 RPM
- Q = 5.0

$$U_{T.O.} = \frac{1.07 \times 10^{10} \times 500}{9 \times (6000)^3 \times 5} = 0.55 \text{ oz-in } (39.6 \text{ g-cm})$$

This imbalance represents an eccentricity of 0.43 mil TIR of a 160 lb. spacer. Hence, the likelihood of centering under unloaded or lightly loaded test stand conditions is very small for this example coupling.

A formula given by Staedeli (1) referred to as the Conti-Barbaran formula, which was developed from test stand experience, gives an estimate of goodness:

$$E = \frac{P \times 10^9}{1.36 \times M \times D^2 \times N^3} \quad \text{(Metric Units)} \qquad (4)$$

where

- P = power, KW
- M = spacer mass, Kg
- D = pitch dia., meters
- N = speed, RPM

For $E \geq 10$, stable and trouble free

$E \leq 5$, problems with mesh centering

Converting to units of H.P., lb_m, in.,

$$E = \frac{1.87 \times 10^{12} \times H.P.}{W \times D^2 \times N^3} \quad \text{(American Units)} \tag{5}$$

Example 4

Consider the calculation of Eq.(5) for the conditions of Example 3:

$$E = \frac{1.87 \times 10^{12} \times 500}{160 \times 9.0^2 \times 6000^3} = .334 \text{ --- Problems}$$

For H.P. = 10,000

$E = 6.68$ - - - Problems not likely

The above formula is similar to Equation 3 for the calculation of $U_{T.O.}$ The advantage of Equation 3 is that it gives a result that has physical meaning and gives the designer an imbalance level to achieve for best results.

The concern for mesh centering is two-fold in that gear wear would increase with the mesh off the pitch line and secondly, imbalance of the spacer is a function of the eccentric position of the floating member. The nominal tooth clearance at zero speed is increased at running speed and temperature to give typically 0.051 - 0.076 mm (.002 - .003 inch) diametral clearance. For the 72.7 Kg (160 lb.) spacer of the previous example, this would represent an additional imbalance level of 160 x 16 x .003/2 = 278.3 gm-cm (3.84 oz-in).

Typical balance standards would call for a balance of

$U = 4W/N = 7.21$ gm-cm (0.1 oz-in)

For 40 W/N; U = 1.0 oz-in (72.1 gm-cm)

Hence, the throwout imbalance is over 38 times the desired balance level and 3.8 times the practical balance level. This type of imbalance further amplified by proximity to a spacer critical is not desirable for turbomachinery.

Guidelines for Manufacturing Tolerances and Balance Requirements

The design and manufacture of long couplings required for machinery such as reported in this paper must be treated with due concern for the potential problems previously discussed and demonstrated by test stand experience. The design of such couplings typically necessitate a multicomponent design spacer with bolted flanges and close tolerance fits. It is therefore of utmost importance for couplings falling into Zone III of Figure 16, that the concentricities of the individual components be held below certain minimum standards. Component balance of the hubs, sleeves, and the floating member is required. The final coupling assembly, as mounted on the turbomachinery, must have a total indicator reading on each of the ground balancing journals on the floating member of less than .051 mm (.002 in). This requirement necessitates that the coupling manufacturer achieve runout levels of less than .038 mm (.0015 in) for each balance journal relative to its mounting bore or rabbet fit. This criteria is relaxed for couplings in Zone I and Zone II as indicated on Figure 16.

When the floating member balance journal runout levels cannot be reduced to the required levels by selective component assembly, it becomes necessary to require an arbor balance of the total coupling assembly. The concentricities of the balance arbor must be controlled to preclude significant arbor induced imbalance which can result in a built-in imbalance when a correction is made and the coupling is removed from the arbor.

For couplings having a spacer critical speed ratio larger than 2.0, 2 plane correction on the floating member is adequate. For couplings having the spacer critical lower than 2.0, correction on the floating member should be made in three or more planes along the spacer length, to reduce unwanted modal amplification of the residual imbalance.

CONCLUDING REMARKS

This paper has presented test stand data that points clearly to the importance for rigorous design standards on couplings for turbomachinery, with added precautions for machines that may have lightly loaded coupling end bearings. Turbomachinery likely to experience this condition includes power turbines and compressors of an overhung design. The major conclusions that may be drawn from this paper are:

1. Coupling designs that are below the marginal zone of Figure 16 should not be accepted for Turbomachinery application.

2. Coupling spacers should be designed to have a critical speed ratio of 2.0 or larger. Smaller ratios may be acceptable if adequate analysis is conducted to assure an insensitive design as outlined in this paper.

3. Design features of gear couplings should preclude the admission of oil into the spacer tube.

4. The manufacturing tolerances for couplings should meet specific minimum standards as outlined in this paper.

5. Gear couplings should be designed to have a minimum increase in mesh tip clearance at design speed and temperature.

6. Turbomachinery conditions having low horsepower requirements and utilizing gear couplings are particularly susceptible to larger than normal imbalance levels on the coupling due to spacer throwout.

7. Long spacer couplings having a critical speed ratio less than 2.0 should have at least three planes for balance correction.

8. The balance procedures outlined in this paper could apply to all couplings but are considered essential for those applications in Zone III of Figure 16.

9. Attention to proper material processing and heat treatment is essential in addition to the above considerations.

REFERENCES

1. Staedeli, O.: Toothed Couplings. MAAG Gear Wheel Company, Ltd. 8023 Zurich, Switzerland.

2. Vance, J.: Influence of Coupling Performance on the Dynamics of High Speed Power Transmission Shafts. ASME Paper 72-PTG-36, 1972.

3. Wolf, J. A., Jr.: Whirl Dynamics of a Rotor Partially Filled with Liquid. ScD. Thesis, Massachusetts Institute of Technology, Cambridge, Massachusetts, Sept. 1966.

4. Wolf, J.A., Jr.: Whirl Dynamics of a Rotor Partially Filled with Liquid. Journal of Applied Mechanics, Trans. ASME, Dec. 1968, pp. 676-682.

5. Ehrich, F. F.: The Influence of Trapped Fluids on High Speed Rotor Vibrations. J. of E. Ind., Trans. ASME, Nov. 1967, pp. 806-812.

6. Saito, S. and Someya, T.: Self-Excited Vibration of a Rotating Hollow Shaft Partially Filled with Liquid. Journal of Mechanical Design, Trans. ASME, ASME Preprint 79-DET-62.

7. Lichtenberg, G.: Vibrations of an Elastically Mounted Spinning Rotor Partially Filled with Liquid. Journal of Mechanical Design, Trans. ASME, ASME Preprint 81-DET-54.

TABLE I. - UNDAMPED MODAL AMPLIFICATION DUE TO APPROACH TO A CRITICAL SPEED

Speed Ratio $\frac{N_{cr}}{N} = \frac{1}{\Omega}$	Dimensionless Speed $\frac{N}{N_{cr}} = \Omega$	Speed-Square Ω^2	Undamped Response $\frac{\Omega^2}{1-\Omega^2}$	Percent Deviation from Speed-Square Curve
∞	0	0	0	0.0
4	.25	.0625	.0667	6.4
3	.33	.1089	.122	13.2
2.5	.4	.16	.19	19
2.3	.434	.189	.233	23.2
2.0	.50	.25	.333	33 (33.04)*
1.81	.55	.302	.434	43
1.67	.60	.36	.5625	56 (55.5)*
1.42	.70	.49	.96	96
1.25	.80	.64	1.77	176
1.11	.90	.81	4.26	425
1.0	1.0	1.0	∞	∞ (900)*

*Denotes value for typical damping value of Q = 10

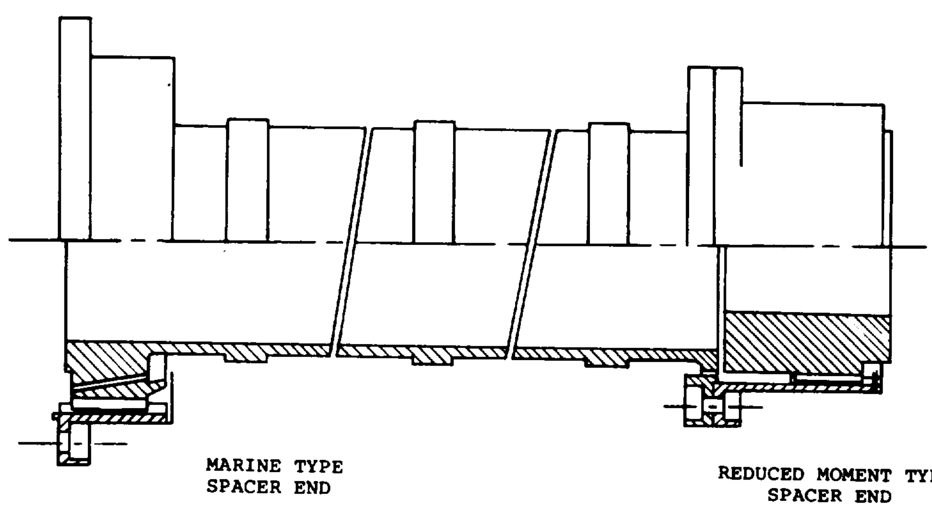


Figure 1. - Expander original coupling design employing both marine and reduced moment spacer design features.

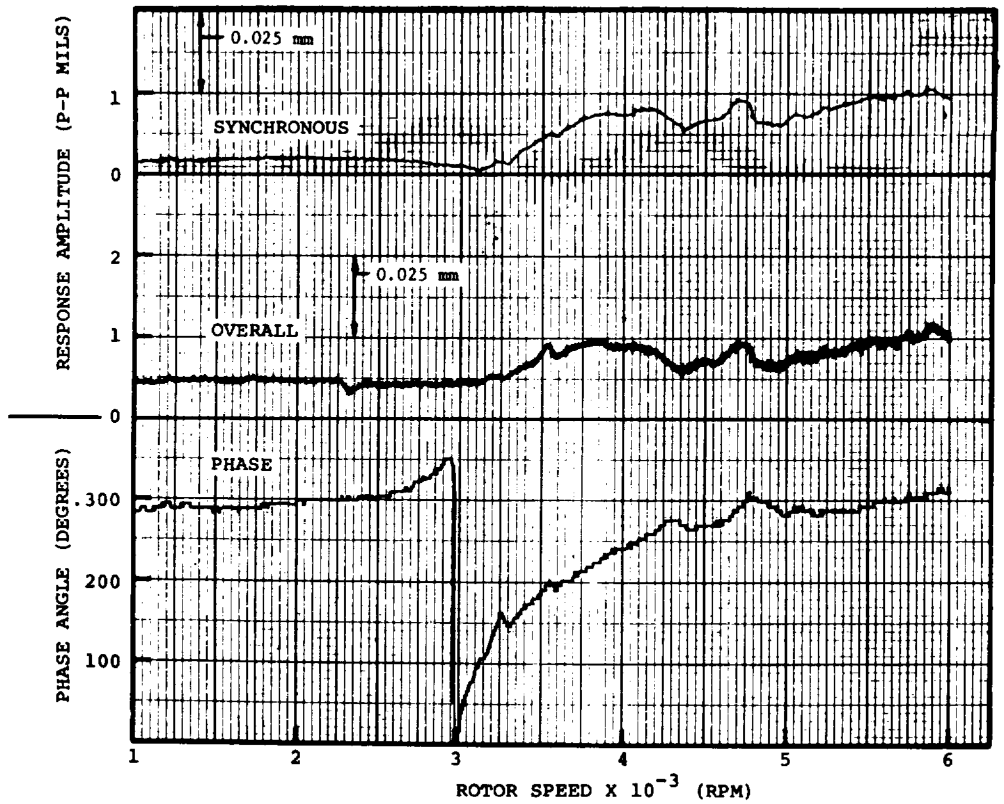


Figure 2. - Acceptable axial compressor vibration (speed vs. amplitude) signature. Four-hour mechanical test run.

TEST SET UP

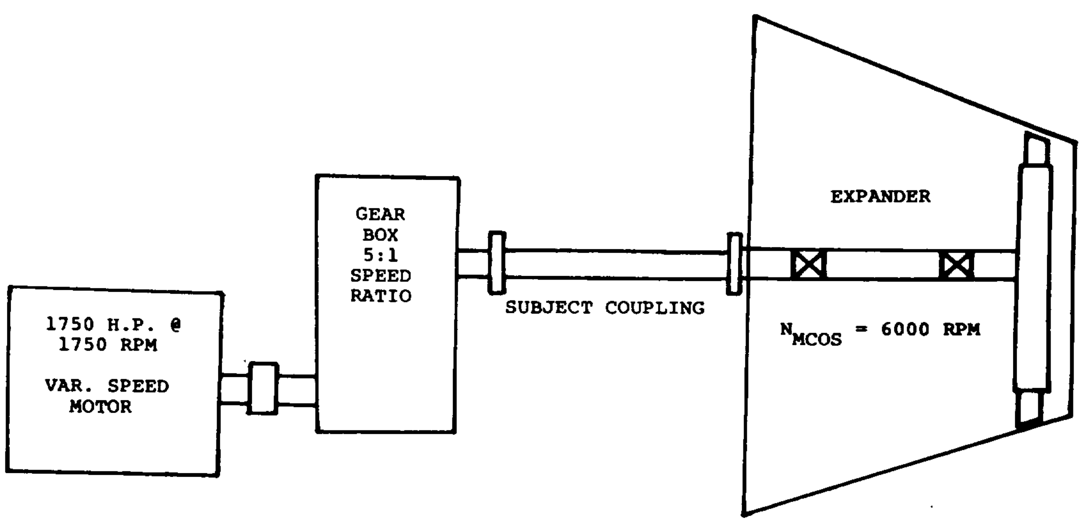


Figure 3. - General configuration of expander test stand assembly.

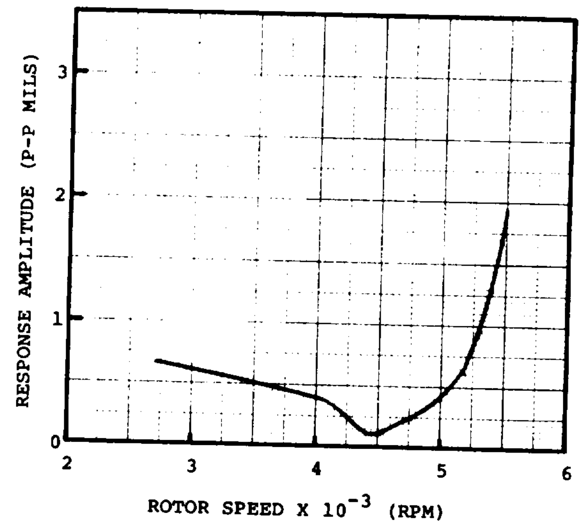


Figure 4. - Expander vibration signature for initial acceleration on test stand.

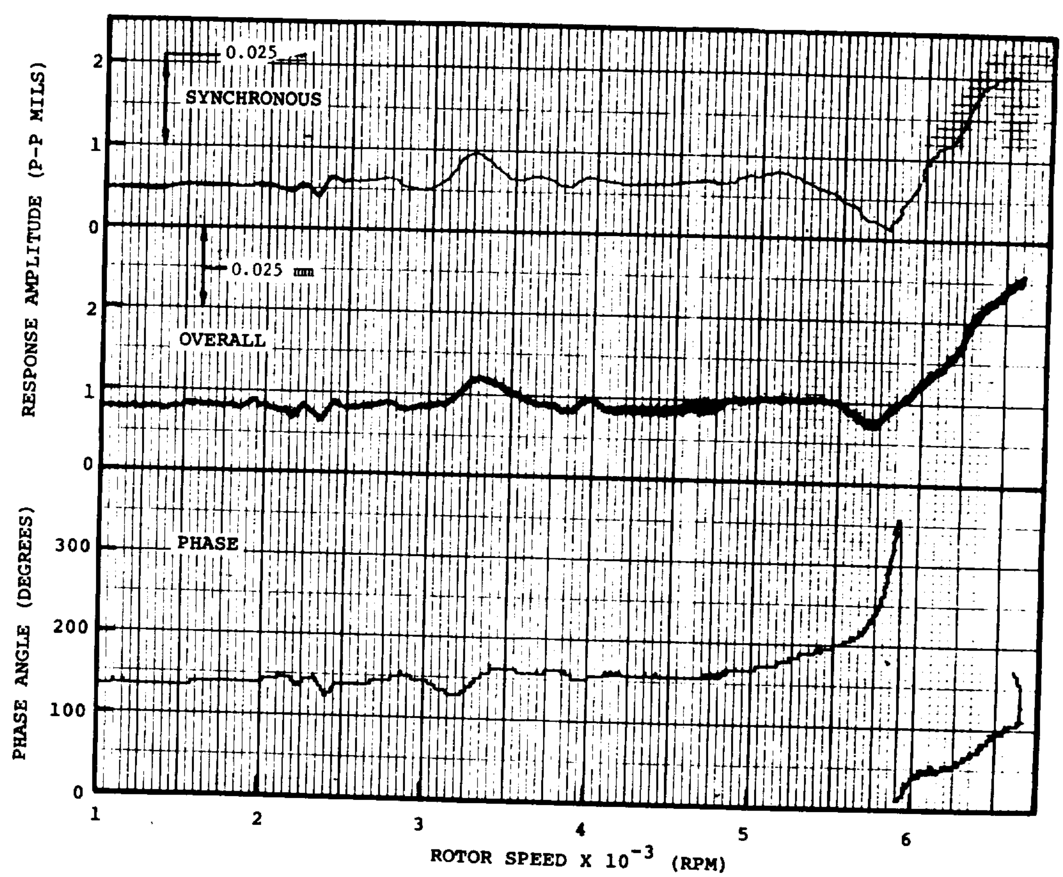


Figure 5. - Expander vibration signature for final balance condition of coupling on test stand during mechanical run.

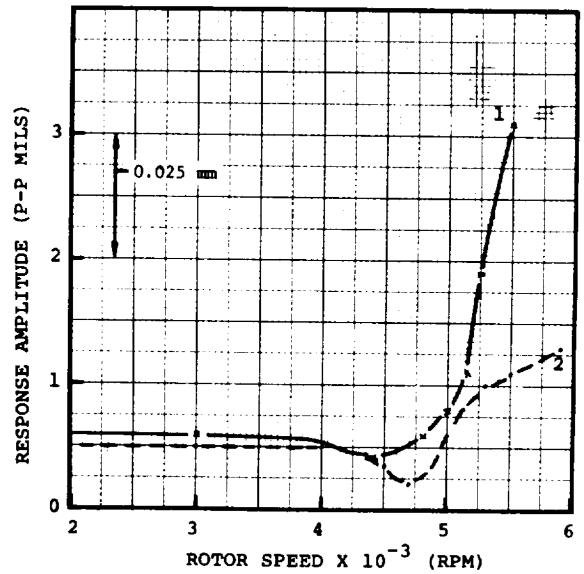


Figure 6. - Test stand vibration signature comparison for balanced spacer with and without polyphenylene tooth coating.

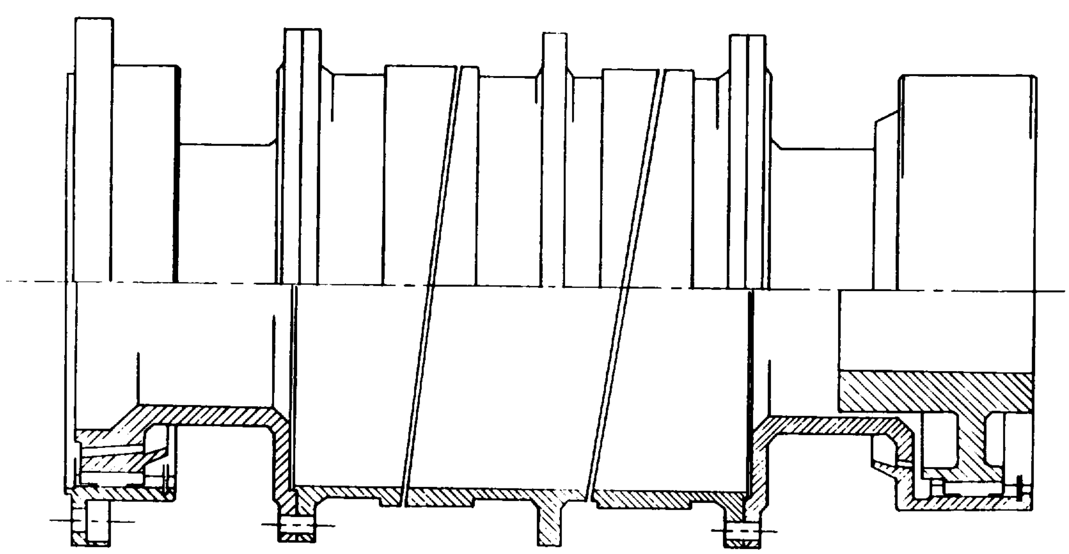
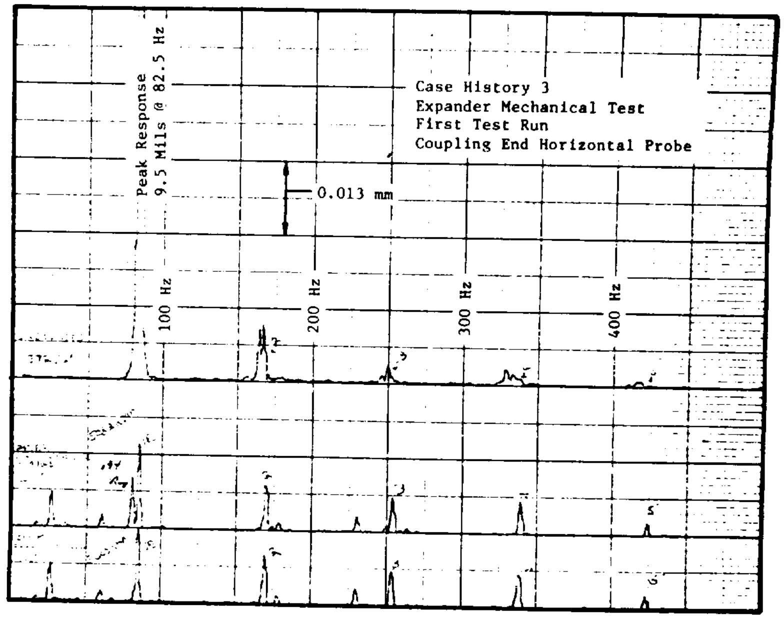
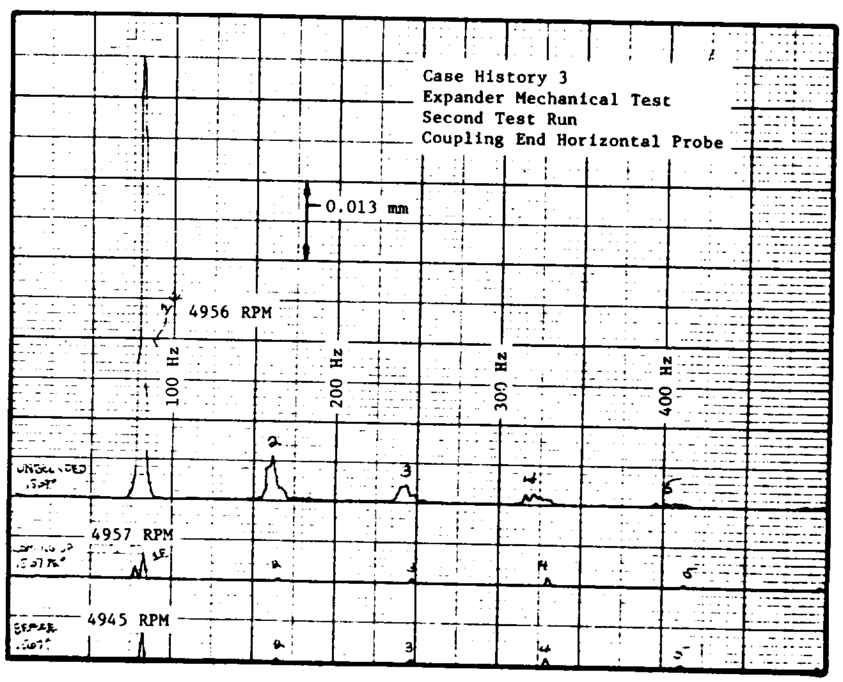


Figure 7. - Expander redesigned coupling with increased spacer tube diameters.

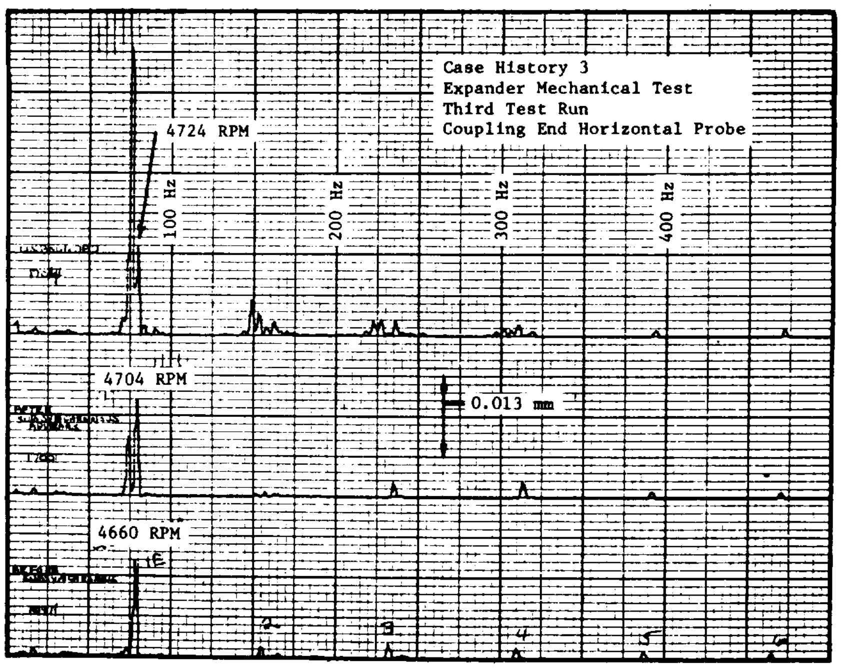


(a) First acceleration with entrapped oil.

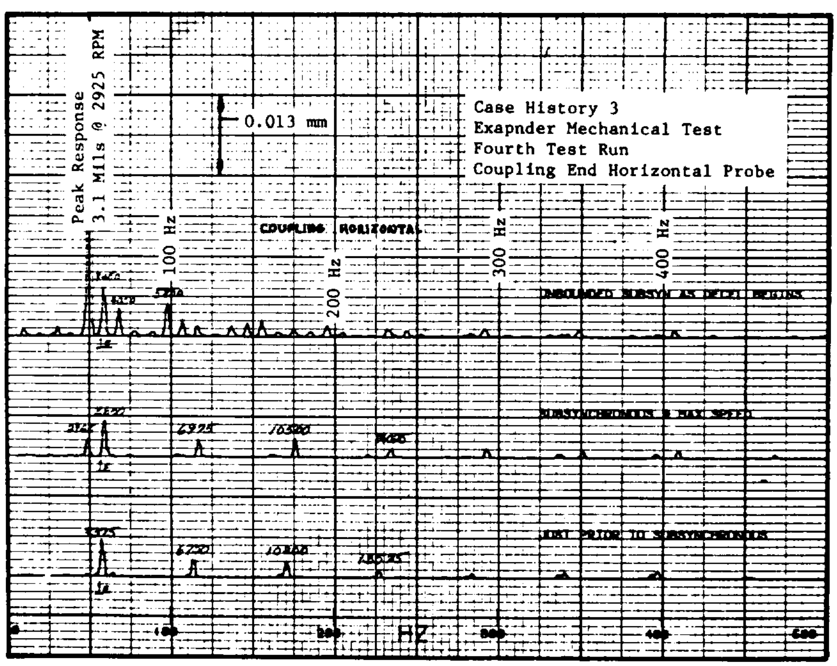


(b) Second acceleration with entrapped oil.

Figure 8. - Test stand vibration spectrum signature.



(c) Third acceleration with entrapped oil.



(d) Fourth acceleration with entrapped oil.

Figure 8. - Concluded.

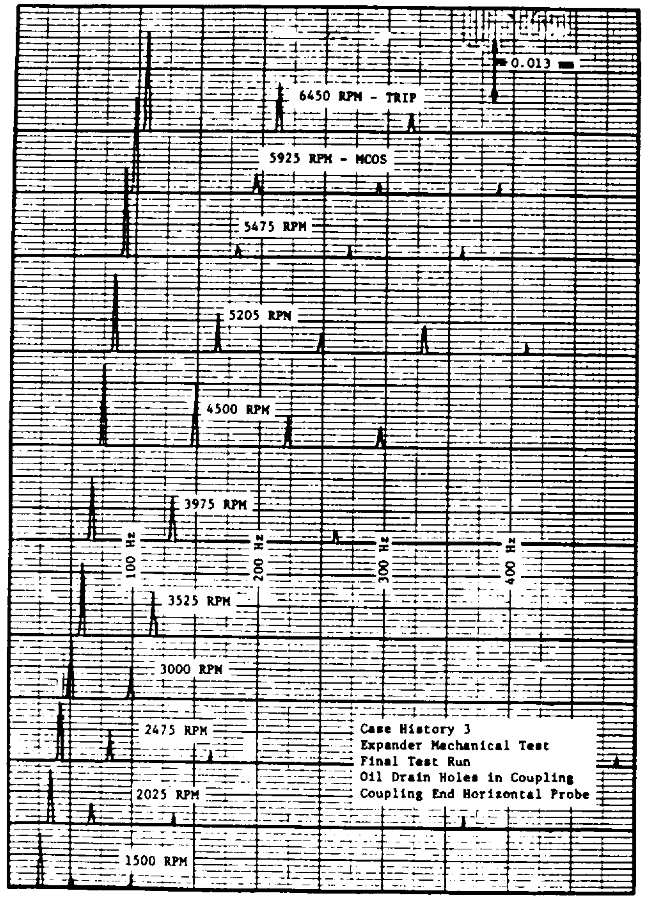


Figure 9. - Test stand vibration spectrum signature after oil drain holes were drilled in spacer.

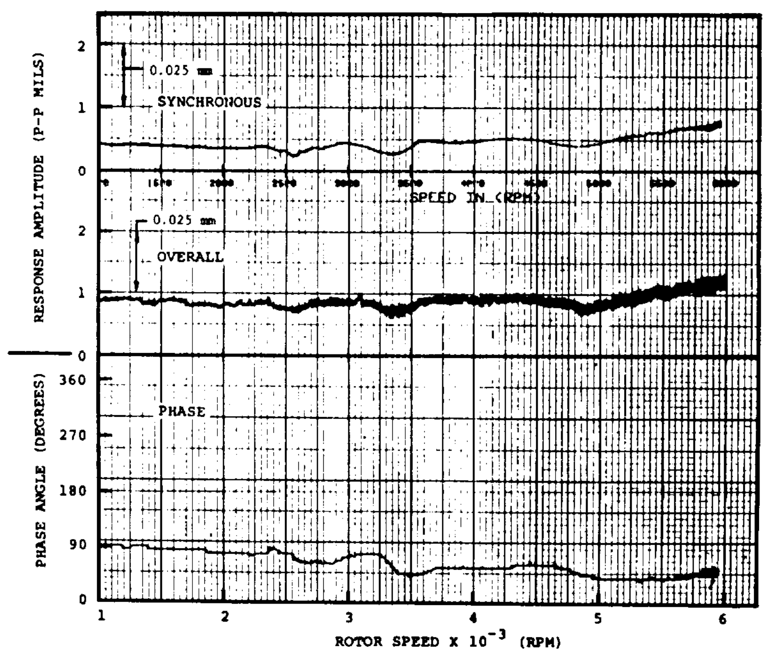


Figure 10. - Test stand vibration signature for retest run with oil drain holes in coupling spacer.

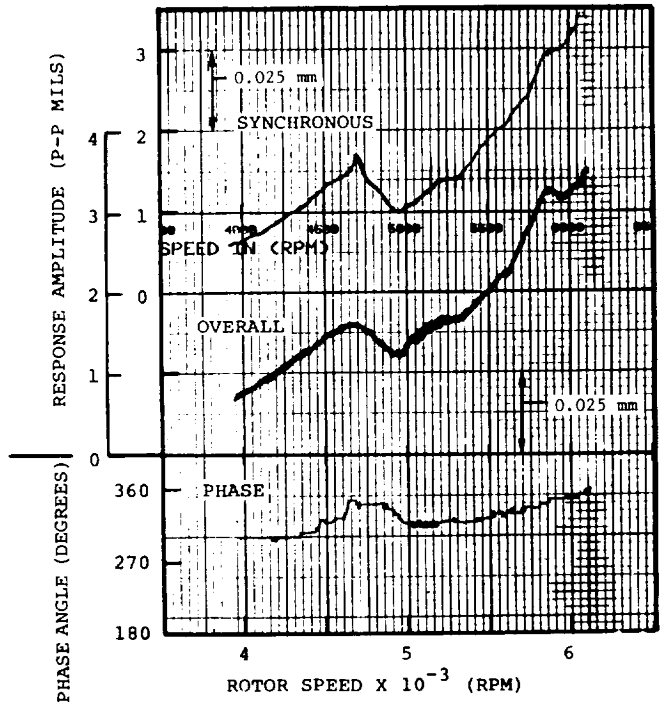


Figure 11. - Test stand vibration signature with coupling assembled using manufacturer's match marks (5-6 mils TIR).

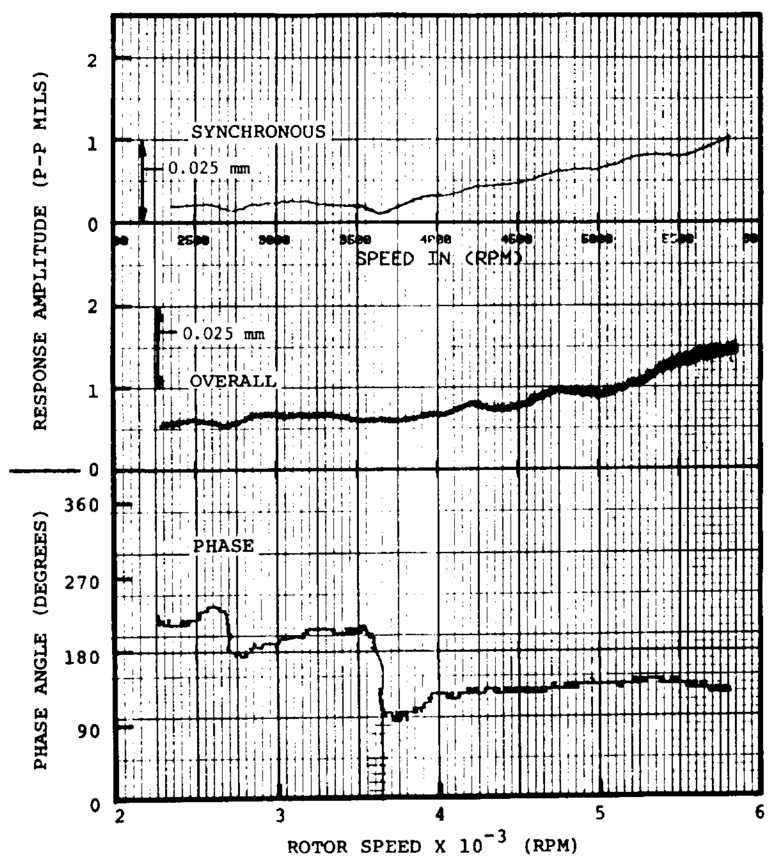
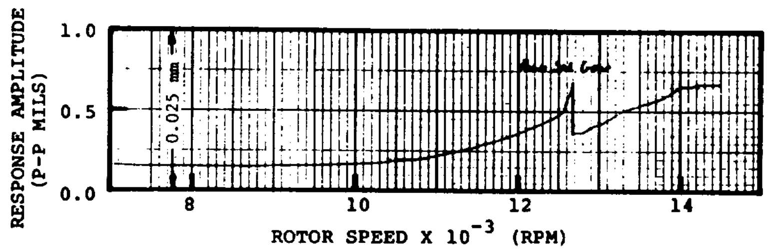
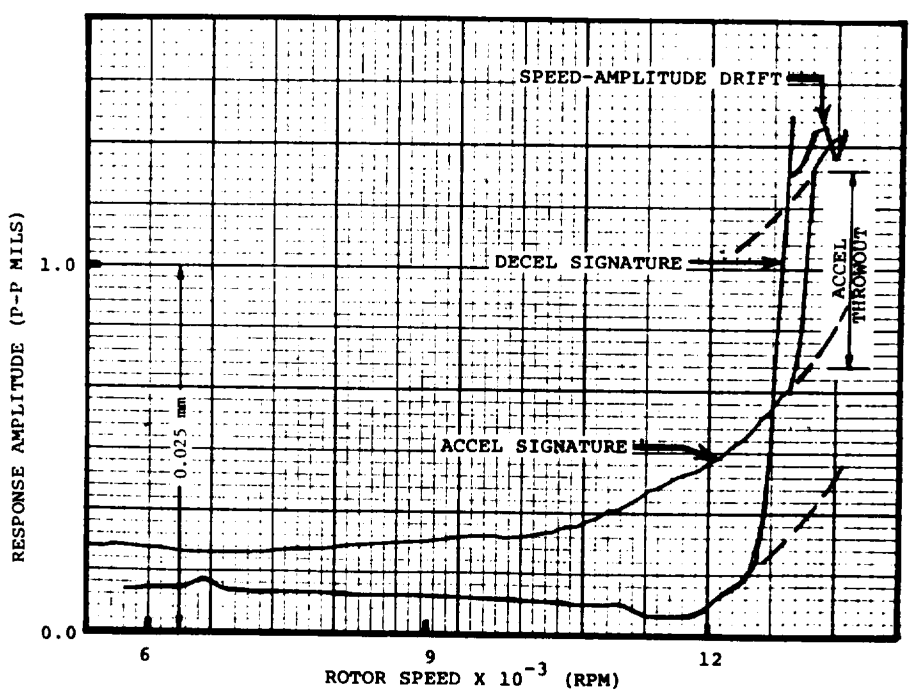


Figure 12. - Test stand vibration signature with coupling assembled to minimize coupling spacer runout imbalance.



(a) Exemplifying spacer throwout.



(b) For acceleration and deceleration showing spacer throwout.

Figure 13. - Gas turbine test stand vibration signatures.

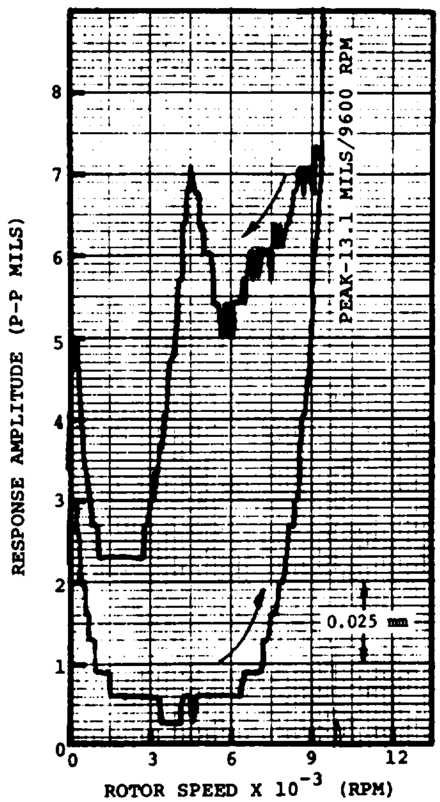
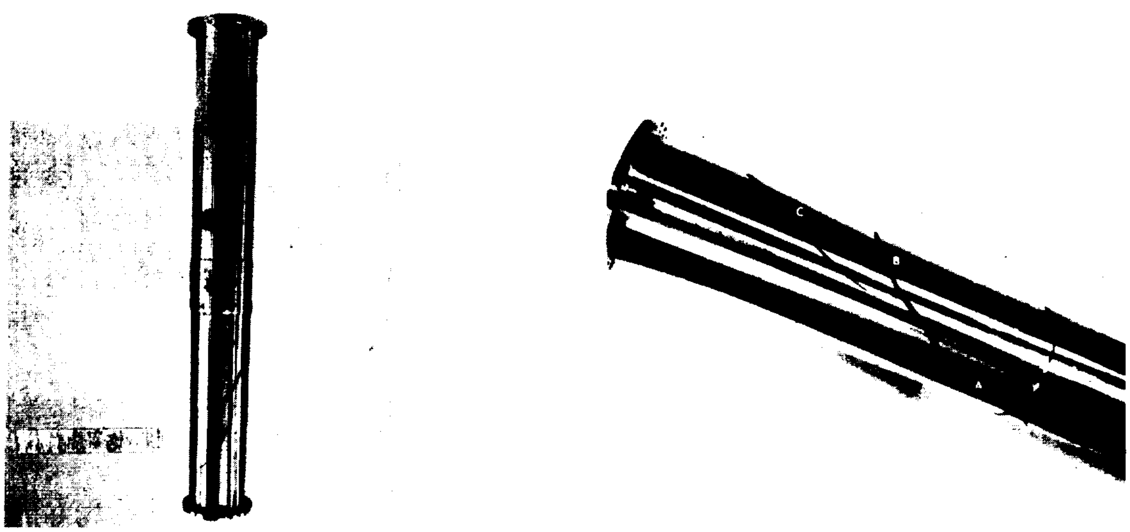


Figure 14. - Gas turbine test stand vibration signature for the condition of a cracked coupling spacer.



(a) (b)

Figure 15. - Failed coupling spacer.

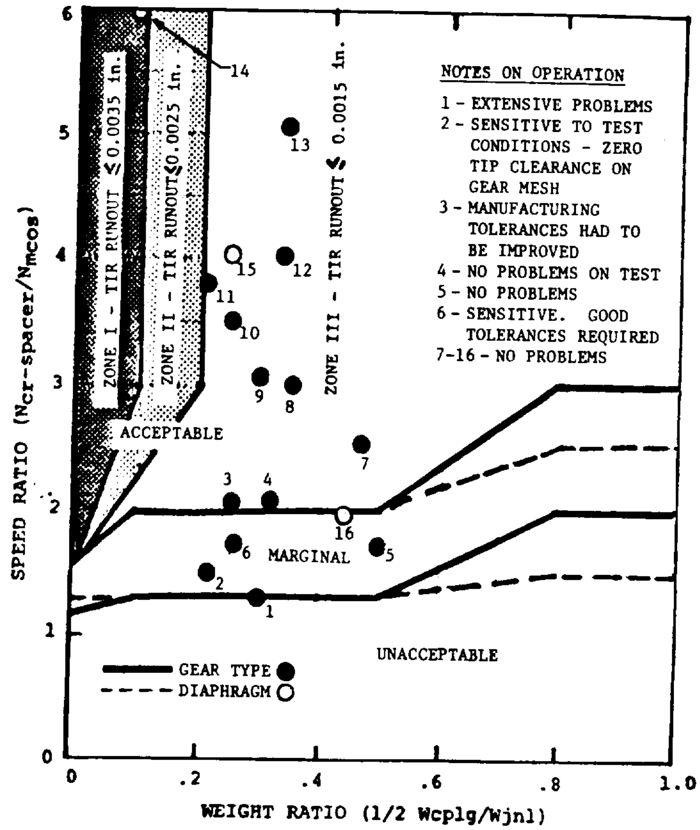


Figure 16. - Chart for coupling selection showing degree of acceptibility with overhung turbomachinery. Experience overplotted. Zones indicate runout level of balance bands relative to the mounting bore or rabbet fit.