GEAR COUPLING MISALIGNMENT INDUCED FORCES AND THEIR EFFECTS ON MACHINERY VIBRATION

by

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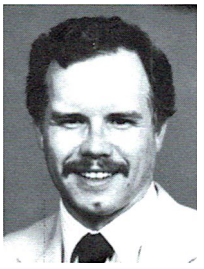
Dr. Palazzolo's expertise is in machinery and structural vibrations, rotordynamics and deflections, and stress. For simulations, he employs transfer matrices, and finite and boundary elements. He has also been extensively involved with field troubleshooting of mechanical malfunctions in rotating and reciprocating machinery. Dr. Palazzolo has presented papers at ASLE and ASME Gas Turbine and Vibration Conferences, and has published many papers in various engineering journals. His current research includes cryogenic vibration dampers, active vibrations control, fluid film bearings, shaft currents, gear couplings, magnetic bearings, and annular seals.



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Prior to this, Mr. Locke was assigned to a turbomachinery consulting group in Wilmington. During his first 10 years at Dupont, he was assigned to a petrochemicals plant where he provided technical assistance to operations and maintenance, participated in several plant startups and the commissioning of several large process compressors.

Mr. Locke graduated from Purdue University (1972) with a B.S. degree in Mechanical Engineering. He has written two other papers on turbomachinery and is a member of ASME.

Michael M. Calistrat has outstanding experience in the field of power transmission equipment, which he has accumulated working with oil drilling equipment, gearing and flexible couplings. He also has a solid background in industrial gas turbines.

He is active in engineering societies and manufacturing associations. He chaired many technical committees for ASME, ASTM, and ASLE, and was Chairman of the International Conference on Power Transmission and Gearing. For the last three years, he was associate editor of the ASME Journal of Machine Design.

He has been a coauthor on three engineering manuals and has written many technical papers and articles in the United States, Japan, France, Canada, Holland, Italy, and Taiwan.

Mr. Calistrat was Manager of Research and Development for the Power Transmission Division of Koppers Company, where he was instrumental in the development of many products. Presently, he heads a consulting company which specializes in rotating machinery.

Mr. Calistrat received his M.S. degree in Mechanical Engineering from the University of Bucharest and holds 17 patents.

ABSTRACT

Gear couplings can produce large static forces and moments that can affect the vibrations of turbomachinery, even with nearly perfect alignment. Research, testing and case histories have verified this theory and are reported. Methods are suggested to control the direction of these forces for reduced vibration and enhanced turbomachinery reliability.

INTRODUCTION

Gear couplings are very reliable, lightweight and used extensively in turbomachinery. An important, but little recognized factor in using these couplings, is the presence of static forces and moments which can be quite large. Understanding these effects can help improve and/or explain the vibration behavior of turbomachinery. These forces have the potential to significantly alter bearing loads, stiffness and damping and result in high vibration. Observations of related behavior in plant machinery led Locke to initiate this study.

The role of the ideal gear coupling is to transmit torque from the driven machine, while accommodating some degree of parallel and angular misalignment between the two shafts. In reality, the torque about the axis of the shaft may produce bending moments about the perpendiculars to this axis, by the following three mechanisms:

- Projection of the shaft torque vector onto the perpendicular of the misaligned shaft,
- Friction forces due to torque and sliding of the misaligned teeth during rotation and,
- Offset between contact points of the teeth on opposite sides of the coupling. These bending moments are reacted by the fluid film bearings supporting the shafts. The additional preloads on the fluid film bearings have a significant influence on their stiffness and damping. Critical speeds, unbalanced response, and stability are in turn directly influenced by these bearing properties.

One very surprising characteristic of gear couplings is that the amplitude of the frictional moments and forces do not decrease with improved alignment. Nearly perfect alignment can make the direction of the forces uncertain with normal machine thermal changes. Methods are suggested to control the direction of these forces for reduced vibration and enhanced turbomachinery reliability.

MISALIGNMENT INDUCED MOMENTS IN GEAR COUPLINGS—THEORY

The literature contains several references on the subject of gear coupling moments. Notable among these are those by Mancuso [1], Gibbons [2], and Crease [3]. Mancuso derives equations for couplings with crowned teeth and assumes that the torque is transmitted solely by two pairs of contacting teeth, separated by 180 degrees. Clark [4] generalized these results by deriving equations in which the number of teeth in contact could be varied from two to "all." His work was guided by the earlier tests and analyses of Calistrat.

Working from the diagrams in Figure 1, Clark derived the following equations:

Out of Plane Moment (About X_r or X_s)

$$M_X = \sum_{i=1}^{n} \left(\frac{D_p}{2}\sin\theta_i(F_N\sin\theta_i\sin\psi - \tilde{\mu}F_N\cos\psi) - R\sin\psi\sin\theta_i(F_N\cos\theta_i)\right) \quad (1)$$

In Plane Moment (About Y_r or Y_s)

$$M_Y = \sum_{i=1}^{n} \left(-R\sin\psi\sin\theta_i(F_N\sin\theta_i\cos\psi + \tilde{\mu}F_N\sin\psi) - \frac{D_p}{2}\cos\theta_i(F_N\sin\theta_i\sin\psi - \tilde{\mu}F_N\cos\psi)\right) \quad (2)$$

where;

$$\tilde{\mu} = \begin{cases} \mu, \ if \ 0<\theta_i<\pi \\ 0, \ if \ \theta_i=0,\pi \text{ or } 2\pi \\ -\mu, \ if \ \pi<\theta_i<2\pi \end{cases} \quad (3)$$

and

$$F_N = \frac{2\times T}{D_p \sum_{i=1}^{n} (\cos^2\theta_i + \sin^2\theta_i\cos\psi + \tilde{\mu}\sin\theta_i\sin\psi} \quad (4)$$

If it is assumed that μ and ψ are very small, i.e., the quantities $\sin 2\psi$ and $\mu \sin\psi$ are approximately zero and cos is approximately 1, that $\sin\psi << \mu$, and that all teeth are symmetrically arranged around the gear, the above equations reduces to:

Out of Plane (Friction) Moment

$$M_f = \frac{\mu T}{n}\sum_{i=1}^{n} |\sin\theta_i| \quad (5)$$

In Plane (Kinematic) Moment

$$M_K = \frac{TR\sin\psi}{nD_p/2}\sum_{i=1}^{n} \sin^2\theta_i \quad (6)$$

Experiments performed by Calistrat showed that, in most instances, all teeth are in contact. The literature, however, contains many references that assume only two teeth in contact. If it is

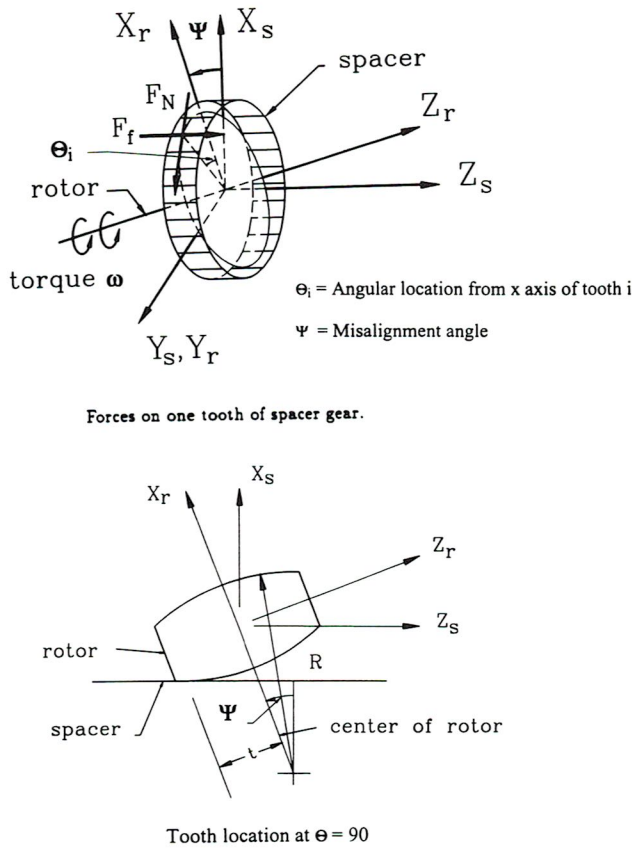


Figure 1. Kinematic/Kinetic Diagrams for Moment Equation Derivation.

assumed that only two teeth are in contact, Equations (5) and (6) reduce to:

Out of Plane (Friction) Moment

$$M_f = \mu T \tag{7}$$

In Plane (Kinematic) Moment

$$M_K = \frac{TR\sin\psi}{D_p/2} \tag{8}$$

Equations (7) and (8) are the form for the moment equations which commonly appear in the literature. The moment M_f is labeled as the friction moment, because it is due solely to the coefficient of friction μ, whereas the kinematic moment M_K is solely due to the crown radius (R), which displaces the contact point when considered at opposing sides of the gear. Some common ranges of μ quoted in the literature are Mancuso (0.004 to 0.045), Gibbons (0.05 to 0.35) and Crease (0.005 to 0.30) [1, 2, 3].

A comparison is shown in Figure 2 between predictions by Clark's equations and Mancuso's (1971) test data, using the following test parameters;

D_p = pitch diameter = 4.2 in.

R = tooth curvature radius = 3.2 in.

P = diametral pitch = 10

Torque = T = 4500. in. lbs.

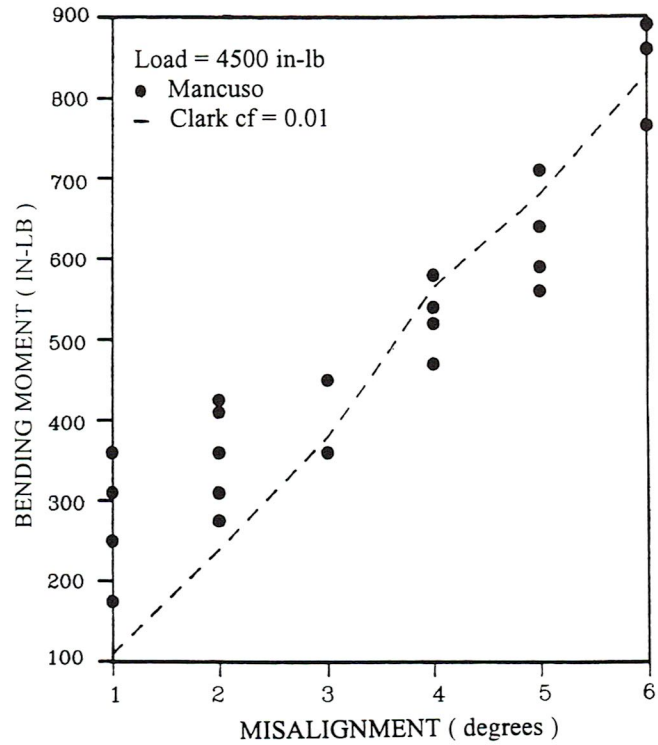


Figure 2. Comparison Between Mancuso's Measured Moments and Clark's Equation.

The agreement between the measured and predicted moments is seen to be very good.

MISALIGNMENT INDUCED MOMENTS IN GEAR COUPLINGS—MEASUREMENT

A four-square test rig was designed by Calistrat and fabricated and assembled at Texas A&M University, to experimentally measure gear coupling bending moments and friction coefficients. In Figure 3, photographs are presented of the test rig showing the reverse indicator alignment bars, the center stand, which is intentionally misaligned and supports the test coupling, the load cells which measure the bearing reaction forces, and the torquemeter. The rig is driven by a 10 hp electric motor through a 2:1 speed reducing belt system to the test (low speed) shaft. The 1:2 speed increasing gear box also drives the high speed shaft which includes the torquemeter, measuring torque Γ. Note that

$$T = 2\Gamma \tag{9}$$

in Equations (1-8). Properties of the industrial gear couplings employed in the testing are summarized in Table 1. The low speed (test) shaft spins at 1800 rpm, and the axial float (at zero torque) of each coupling was set to be at least 60 mils to allow them freedom to flex. The couplings were lubricated with turbine-grade oil or coupling grease and the torque was varied by pretwisting a high speed shaft line coupling with a slotted bolt circle.

The purpose of the testing is to investigate the influence of coupling misalignment on shaft forces and moments. Therefore, in setting the test rig, shafts are intentionally misaligned at various predetermined values. These misalignment values must be tightly controlled in order to obtain useful information. Misalignment of the gear coupling test rig is attained utilizing the reverse indicator method of alignment. This method involves measuring the extension of one shaft relative to the other using dial indicators. The dial

Table 1. Properties of the Industrial Gear Couplings for the Low Speed (Test) Shaft Line.

SIZE: 1 1/2"
LOAD RATING: 27 HP/100 RPM
RATED TORQUE: 17,000 in.lbs.
MAX SPEED: 12,000 RPM
MAX. MIS-ALIGNMENT PER GEAR MESH: 1 1/2°
TOOTH PITCH DIAMETER: 3.5 in.
NO. OF TEETH: 49

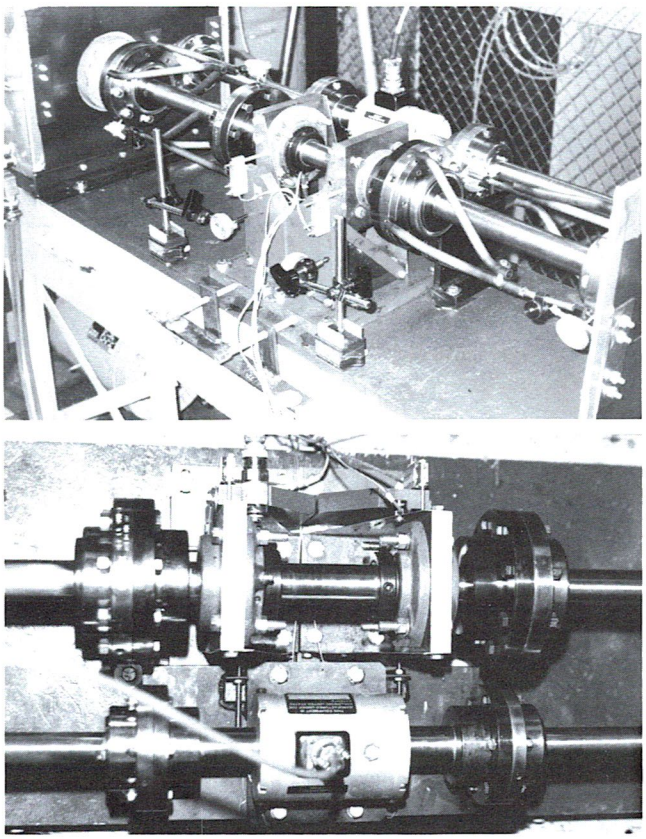
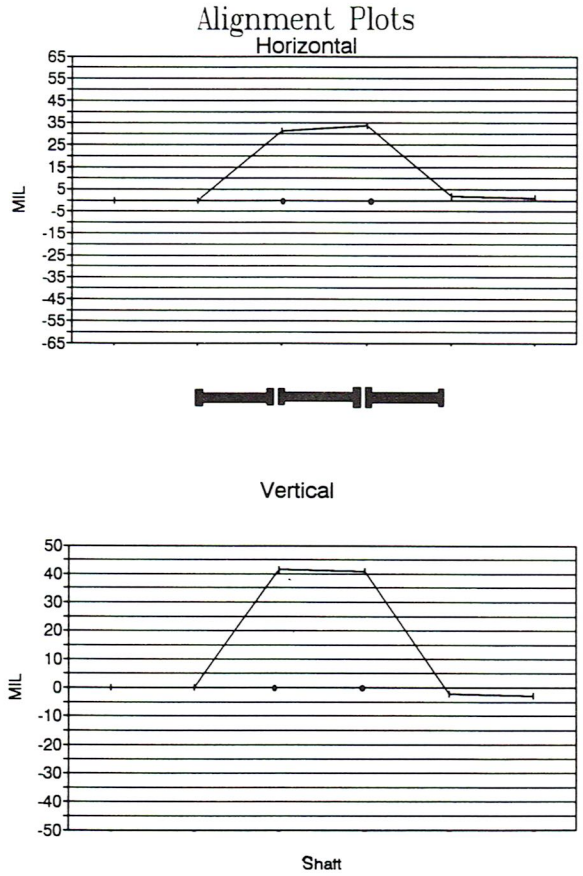


Figure 3. Four Square Gear Coupling Test Rig and Closeup View of Alignment (Center) Stand and Torquemeter.

Date 3/23/92 Time: 3:00 pm Torq= 0 in-lb

Alignment data						
Horizontal =	31.625	mil =	0.8033	mm =	0.1647	Degrees
Vertical =	42.125	mil =	1.07	mm =	0.2194	Degrees



Alignment Readings (mil)

Horizontal	-60	63	63	-67
Vertical	-84	83	85	-85

Figure 4. Typical Intentional Misalignment Plots in Horizontal and Vertical Planes.

indicators are attached to the assembled couplings via a reinforced indicator extension bar. Readings are taken 90 degrees apart and indicate the relative position of the machine shafts in both horizontal and vertical axes. Desired misalignment of the two center stands, relative to the otherwise aligned gear boxes, is induced through the utilization of dual jackscrews on opposing sides of each stand. Misalignment readings are taken from the dial indicators and then computer plotted, as illustrated in Figure 4. Proper adjustments are made to the center stand via screw travel. Jackscrews are used to align the gearboxes, and to assure that the gear boxes are held in proper position.

Although the misalignment procedure is conceptually simple, it becomes complicated due to flexure of the center stands. The center stands are held in place on the gear coupling test rig using four high-strength bolts attached to each stand. During tightening of these bolts, shaft misalignment may fluctuate to an intolerable degree. This occurrence somewhat complicates the bolt tightening procedure. To compensate for this movement, the ideal method of bolt tightening is to tighten all four bolts at once. Therefore, the procedure for exact misalignment requires an iterative approach. Once the desired misalignment has been attained, the coupling rig goes through a 24 hour period of "stress relief" during which, the misalignment changes by an average of three to five percent. Following this period, the test rig is ready for dynamic testing and further misalignment changes.

The following list details the rig modifications that were employed to improve the test's validity:

- A gearbox foot was cracked while attempting to align the outboard side gearbox. A new hole was drilled for another gearbox bolt, which holds down the cracked section with a flat steel bar.
- Jack-bolt screws were placed in four locations surrounding the center stand and both gearboxes in order to assist with the alignment procedures.

• During alignment procedures of the test rig, deviations in vertical alignment were noted while jack-bolts were being tightened. To alleviate this:

· spherical cap nuts were placed on the end of all jack bolts, and

· reinforcement "trusses" were designed and welded to the bottom of the test stand.

· flat reinforcement bars contained in the bearing houses were replaced with angle iron.

• The "table top" plate deformed and caused skewed alignment readings. This plate was rigidly welded to the test stand resulting in a significant reduction in its deformation.

• The outboard side gearbox was disassembled for routine inspection. This inspection revealed a spinning outer race and a partially damaged bearing. A new bearing and race were installed with thread adhesive.

• Play in the gearbox shafts was reduced by giving a higher preload to their bearings by removing approximately 24 mils of axial shim.

Moments were measured following these mechanical integrity confirming procedures.

The friction moment is determined by imposing a symmetrical vertical misalignment and then by recording the center stand bearing's horizontal reaction forces (R^f_{X1}, R^f_{X2}) as shown in Figure 5. Equations relating the reaction forces and friction moment are obtained by considering symmetry, static equilibrium and the friction moment directions, yielding:

$$R^f_{X1} = R^f_{X2} = R^f_X \tag{10}$$

$$M_f = -\frac{L}{4}(R^f_{X1} + R^f_{X2}) = -\frac{L}{2}R^f_X \tag{11}$$

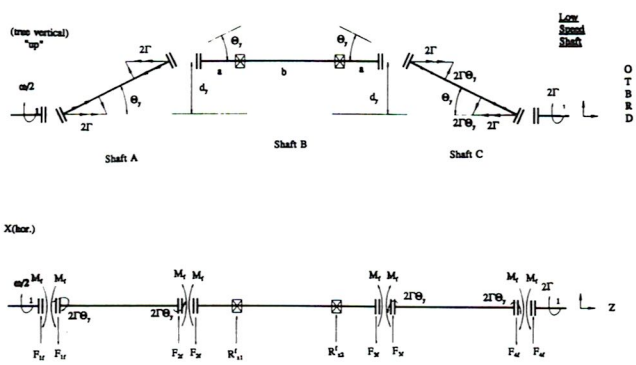


Figure 5. Force and Moment Directions Used in Deriving Equations (10) and (11).

An estimate of the friction coefficient is then obtained by solving either Equation (5) or Equation (7) for μ.

The kinematic moment is determined by simultaneously imposing both symmetric horizontal and symmetric vertical misalignments, and then measuring the center stand bearing's horizontal reaction forces (R^k_{X1}, R^k_{X2}), as shown in Figure 6. Equations relating the reaction forces and kinematic moment are obtained by considering symmetry, static equilibrium and the kinematic moment directions, yielding:

$$R^K_{X1} = R^K_{X2} = R^K_X \tag{12}$$

$$M_K = \frac{L}{2}R^K_X + M_f \mathrm{Sin}\beta \tag{13}$$

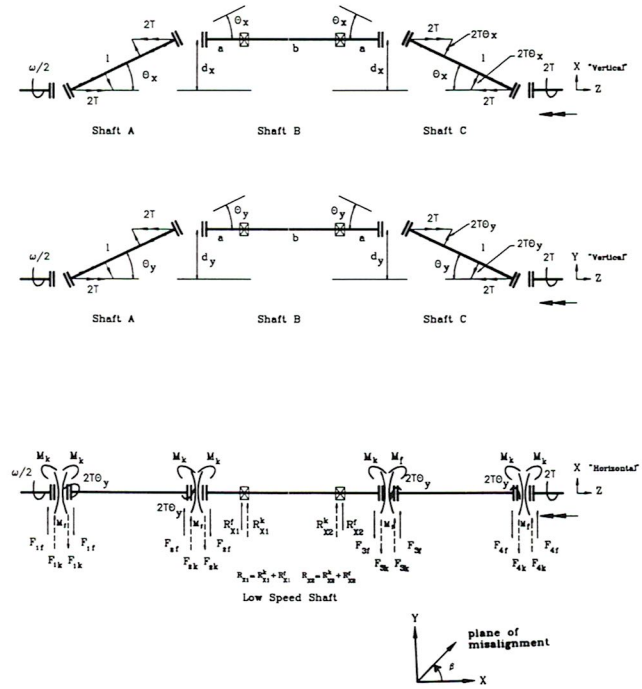


Figure 6. Force and Moment Directions Used in Deriving Equations (12) and (13).

It is assumed that the magnitude of M_f only varies slightly between the pure vertical misalignment and simultaneous horizontal and vertical misalignment cases. Therefore, the term M_f in Equation (13) is known from the pure vertical misalignment case.

Testing was performed with both turbine oil and coupling grease as lubricants. Figures 7, 8, 9, 10, and 11 correspond to the turbine oil tests. A typical plot of the horizontal reaction forces *vs* low speed torque for a pure vertical misalignment is shown in Figure 7. The outboard and motor end reaction forces are seen to be nearly equal, as predicted by Equation (10), over the torque range employed. The friction moments, as determined from Equation (11) and the measured reaction forces, are shown in Figure 8 for three

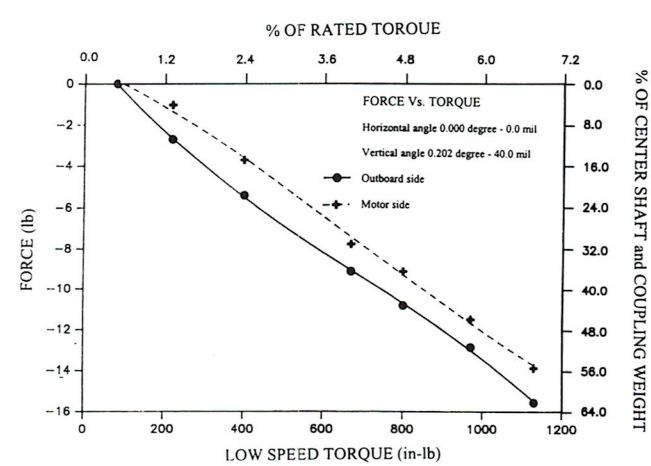


Figure 7. Reaction Forces for a 40 Mil Pure Vertical Misalignment. (Turbine Oil Lubricated).

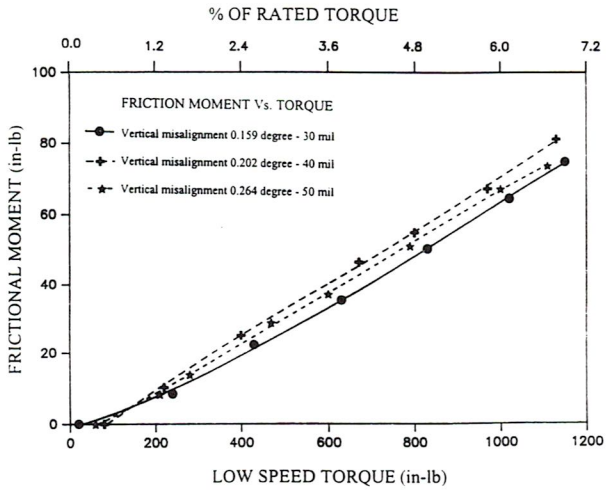


Figure 8. Friction Moment Vs *Torque for Several Pure Vertical Misalignments. (Turbine Oil Lubricated).*

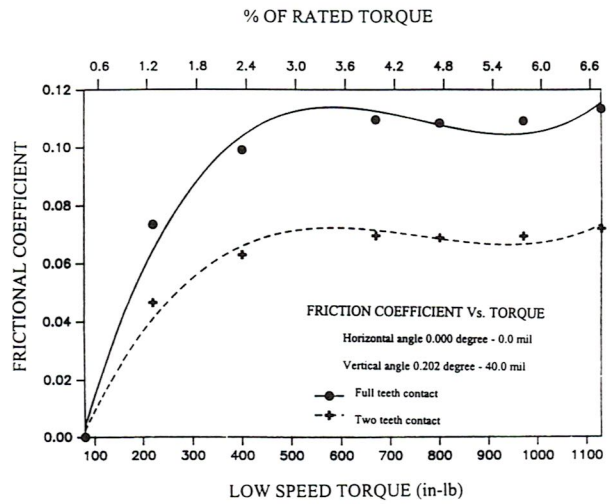


Figure 9. Friction Coefficient Vs *Torque Assuming Two Teeth and All Teeth in Contact. (Turbine Oil Lubricated).*

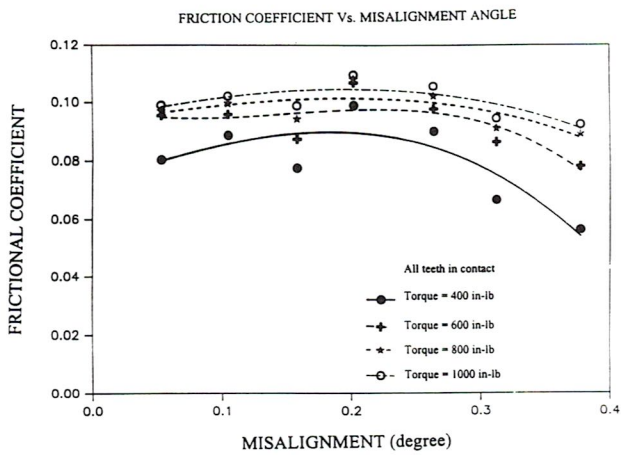


Figure 10. Friction Coefficient Vs *Misalignment Angle for Various Torques. (Turbine Oil Lubricated).*

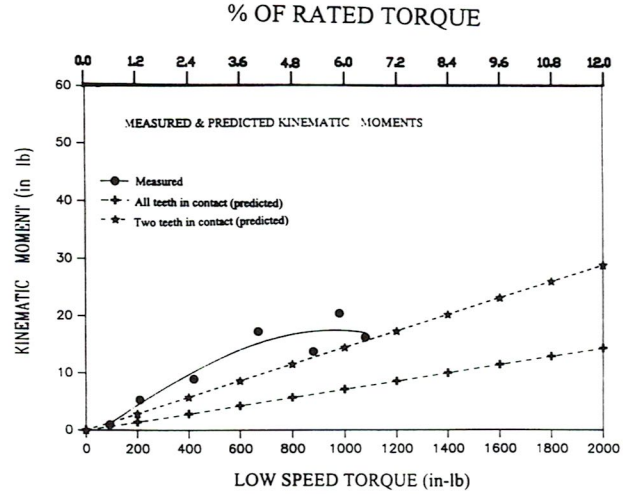


Figure 11. Kinematic Moment-Theory and Experiment Vs *Torque. (Turbine Oil Lubricated).*

levels of pure vertical misalignment. The friction moment is seen to increase with torque but does not have a monotonic relationship with misalignment.

The friction coefficient is estimated by solving Equation (5) or (7) for μ. Plots of μ *vs* torque obtained by assuming either "two teeth" or "all teeth" in contact are shown in Figure 9. It is noteworthy to compare the values shown in Figure 9 to those employed by Mancuso (0.004 to 0.045), Gibbons (0.05 to 0.35), and Crease (0.005 to 0.30) [1, 2, 3]. The friction coefficients in Figure 9 are seen to stay relatively constant above torques of 400 in/lb. The friction factors shown in Figure 10 are relatively insensitive to misalignment angle over the measured range (0.05 degrees to 0.4 degrees). Typical predicted and measured kinematic moments are shown in Figure 11. The predicted values are calculated with Equations (6) and (8) using values of R, D_p, and n, obtained from the coupling manufacturer. The measured values are obtained from Equation (13) by assuming that the friction moment (M_f) does not significantly vary from its value at zero horizontal misalignment. The agreement between the predicted and measured kinematic moments is significantly improved by assuming that only two teeth are in contact, as shown in Figure 11. The disagreement between theory and measured results in this case may be due to uncertainty in the face radius, and small violations in the symmetry assumptions made in deriving Equations (10, 11, 12, 13).

The effects of lubricant type, turbine oil *vs* coupling grease, on the coefficient of friction are compared in Figure 12. Both lubricants have the same misalignment angle and "all teeth in contact" is assumed in calculating μ. The effects of lubricant type on the kinematic moment for the same misalignment angle are compared in Figure 13.

EFFECTS OF GEAR COUPLING MISALIGNMENT INDUCED MOMENTS ON ROTOR RESPONSE—SIMULATION

A diagram of an air compressor rotor supported by three-lobe journal bearings, and coupled to the remainder of the train with gear couplings is shown in Figure 14. Parameters used in the rotordynamic simulation of this rotor are presented in Table 2. Coupling moments, bearing rotordynamic coefficients, bearing eccentricity, stability, modeshapes, and unbalance response were calculated for this rotor with software developed at Texas A&M for the Turbomachinery Research Consortium. The effects of

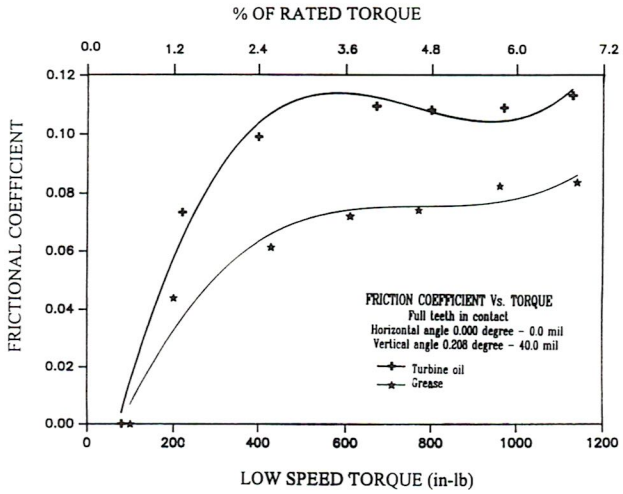


Figure 12. Friction Coefficient Vs *Torque for Turbine Oil and Grease Lubricants.*

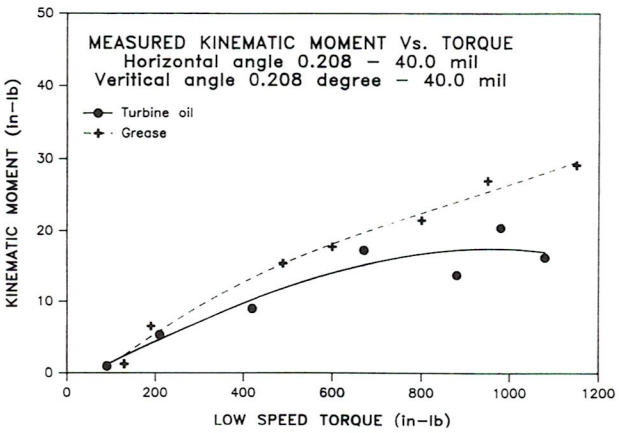


Figure 13. Kinematic Moment Vs *Torque for Turbine Oil and Grease Lubricants.*

varying the angle of misalignment, plane of misalignment (Figure 15), and torque were examined in this study.

The predicted journal equilibrium position is shown in Figure 16 in the right bearing *vs* torque, as it is increased by 20 percent increments of the values given in Table 2. The corresponding bearing loads, stiffnesses dampings, and equilibrium positions are shown in Table 3. Increasing torque has a significant effect on stabilizing this machine as is shown by the plot of the real part of the eigenvalue in Figure 17. The change in the journal equilibrium position is shown in Figure 18 for various misalignment planes. The corresponding bearing loads and equilibrium position coordinates are shown in Table 4. The effects of misalignment plane on

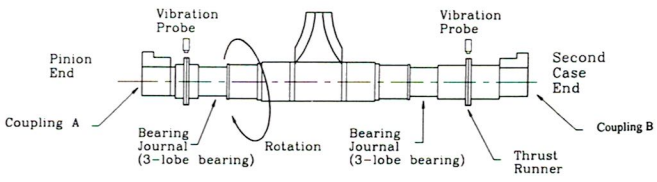


Figure 14. First Stage Air Compressor (Figure 20) Rotor Used for Simulation Study.

Table 2. Shaft and Coupling Data for the Rotordynamic Simulation.

ROTOR LENGTH: 59 in.
BEARING SPAN: 36 in.
ROTOR WEIGHT: 1033 lb.

	Coupling A	*Coupling B*
PITCH DIAMETER:	7.5 in.	7.5 in.
TOOTH CROWN RADIUS:	100.0 in.	100.0 in.
LENGTH:	15.0 in.	6.4 in.
APPLIED TORQUE:	147,000 in.lb.	123,000 in.lb.
NUMBER OF TEETH:	50	50

unbalance response is illustrated by the results in Figure 19. The angle of misalignment (0.08 degrees) was held constant along with the friction coefficient (0.04) for this case. To obtain this plot, the misalignment plane was rotated to 0 degree, 80 degrees, and 200 degrees from the vertical direction. The bearing reactions were determined by static force and moment balances of the coupling bending moments and shaft weight. These reaction forces were inserted in a bearing simulation code to determine the stiffnesses and dampings at various rotor speeds. The shaft vibrations shown in Figure 19 are very sensitive to the plane of misalignment of the gear couplings. The first critical speed, apparent at the 0 degree orientation, is seen to be nearly eliminated with misalignment planes at 80 degrees and 200 degrees. In addition, the second critical's amplification factor is significantly reduced when the misalignment plane is rotated from 0 degrees to 200 degrees.

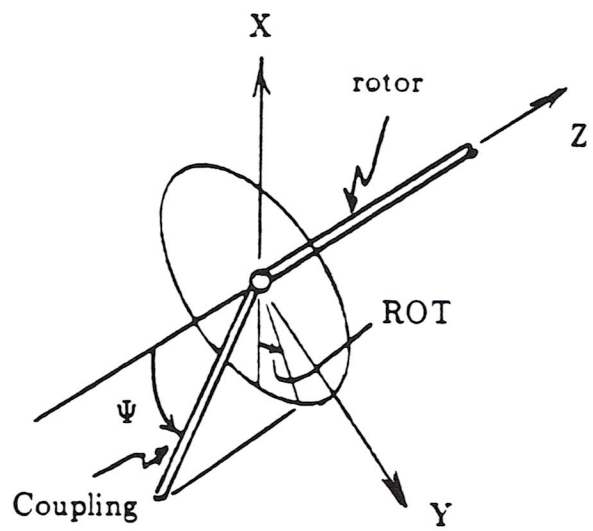


Figure 15. Angle and Plane of Misalignment at Gear Coupling.

EFFECTS OF GEAR COUPLINGS ON ROTOR RESPONSE—CASE HISTORIES

The preceding sections have been based entirely on either theory or lab testing. The initial impetuses for this work were the observations and hypotheses made by Locke based on *field test data.* This emphasizes the importance of studying case histories of coupling connected machinery with vibration problems. The following case histories provide guidance and illumination into this phenomena.

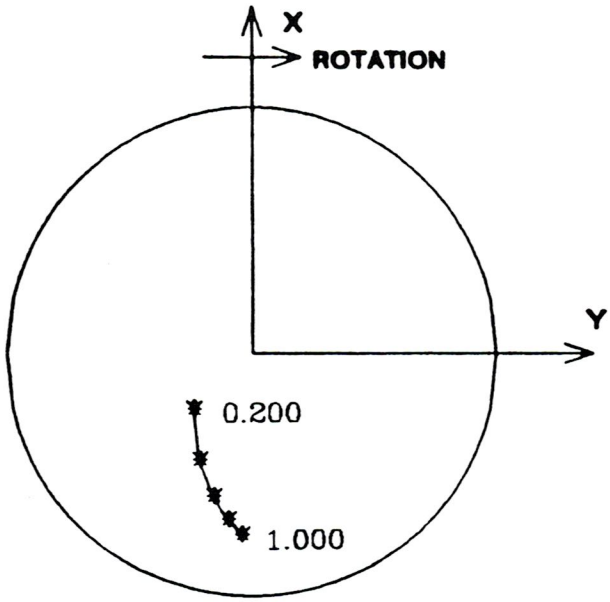


Figure 16. Right Journal's Equilibrium Position Vs *Torque Attenuation Factor.*

Table 3. Bearing Load, Stiffness, Damping and Equilibrium Position Vs *Torque Attenuation Factor (A.F.) (Torque/ Table 2 Torque).*

LOAD ON BEARING (lbs)			DISTANCE in × 10^{-2}	
A.F.	FX	FY	X	Y
0.200	–546.	27.	–0.057	–0.061
0.400	–733.	291.	–0.110	–0.055
0.600	–920.	558.	–0.146	–0.041
0.800	–1107.	821.	–0.171	–0.026
1.000	–1294.	1086.	–0.187	–0.012

DAMPING lb.sec./in.					STIFFNESS lb./in × 10^{-3}			
A.F.	Cxx	Cxy	Cyx	Cyy	Kxx	Kxy	Kyx	Kyy
0.200	2269.	788.	788.	912.	259.	1113.	–305.	511.
0.400	4006.	861.	607.	751.	759.	1729.	–335.	496.
0.600	5911.	579.	–1.	642.	1629.	2267.	–497.	374.
0.800	8192.	134.	–802.	693.	2824.	2818.	–847.	262.
1.000	10795.	–461.	–1804.	867.	4377.	3347.	–1387.	159.

Table 4. Bearing Loads and Equilibrium Position Vs *Misalignment Plane.*

Load on Bearing (lbs)			*Position × 100*	
ROT	FX (lbs)	FY (lbs)	X (in)	Y (in)
0.000	–755.	354.	–0.119	–0.051
40.000	–485.	431.	–0.121	–0.037
80.000	–225.	307.	–0.087	–0.032
120.000	–96.	39.	–0.017	–0.019
160.000	–158.	–247.	0.058	–0.048
240.000	–665.	–393.	0.063	–0.130
280.000	–873.	–164.	–0.047	–0.086
320.000	–906.	110.	–0.093	–0.069

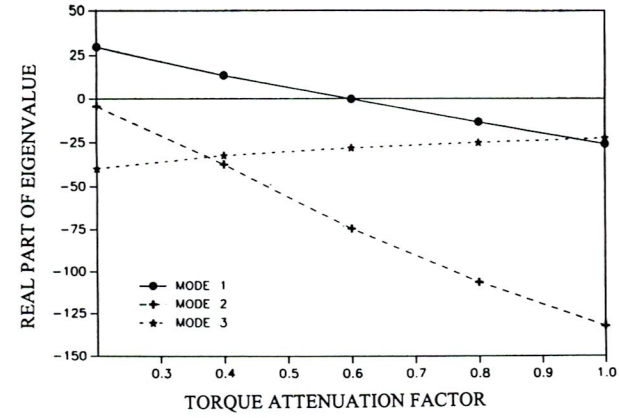


Figure 17. Real Part of Eigenvalue Vs *Torque Attenuation Factor.*

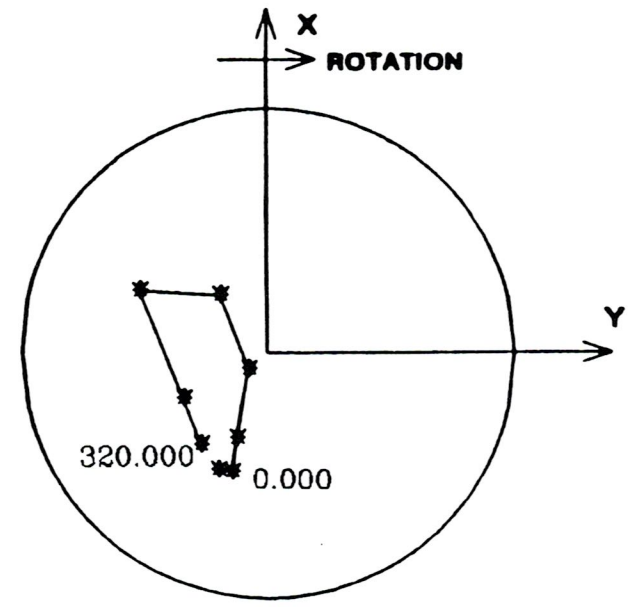


Figure 18. Left Journal's Equilibrium Position Vs *Plane of Misalignment.*

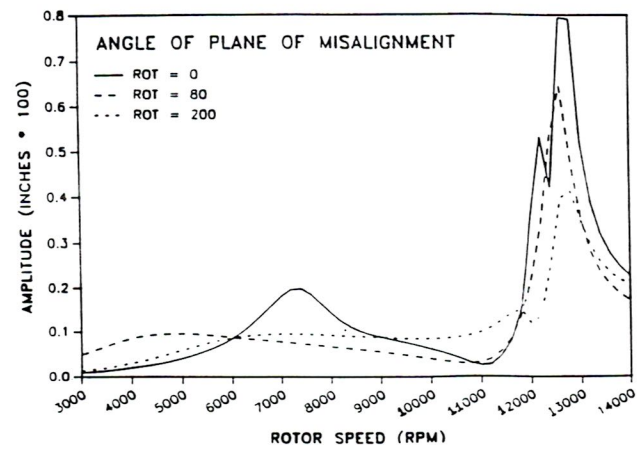


Figure 19. Simulated Unbalance Response Vs *Speed and Plane of Misalignment.*

Case 1

A large utility company experienced an odd situation, in which vibrations increased after it was decided to replace all gear couplings driving boiler-feed pumps, with nonlubricated couplings.

In order to help the engineering group select the best type of coupling to use, it was decided to try a few types. Based on suppliers' selections, a number of pumps were retrofitted, with a variety of coupling types. To eliminate the possibility of bad performance due to misalignment, laser equipment was used to perform as perfect an alignment as possible.

In all cases, the vibration levels of the pumps increased significantly after the gear type couplings were replaced! Lacking a solution to solve the problem, the old gear type couplings were reinstalled, and the pumps again worked with acceptable levels of vibrations!

What happened? The cause of the increased vibrations with perfectly aligned nonlubricated couplings was the *unloading* of the pump bearings. Nonlubricated couplings operating at *small* misalignments create very small reaction forces, which allowed the shafts to orbit unrestrained inside the bearings. Gear couplings, on the other hand, created sufficient loads on the journal to allow smooth operation.

The conclusion in this case history was not meant to imply that gear couplings must be used in boiler feed pumps. The problems in this case are not the couplings, rather, the plain cylindrical bearings. EPRI papers have shown the need of pressure-dam or tilted-pad bearings for boiler feed pumps.

Case 2

A paper plant experienced a situation in which vibration worsened after a turbine repair.

The rotor of a large gas turbine driving a generator through a gear reducer had to be replaced because of damage caused by an ingested inlet guide vane. Before restarting the unit, maintenance people decided to check the misalignment, which was found to be slightly larger than the one considered acceptable by the OEM's instruction manual.

The machines were carefully aligned. Upon startup, the pinion bearing had very high vibration levels. The third author analyzed the problem and concluded that, originally, the OEM had intentionally misaligned the machines to avoid vibrations.

Fortunately, the holes for the original dowel pins were still available, and the turbine was moved in the original position. The unit started and operated with no problems!

The explanation was in the pinion bearing, which was not sufficiently loaded by the forces in the gear mesh, or by its weight. Misaligning the machines created a stabilizing force in the bearings, which allowed a smooth operation.

The type of coupling in this particular case was a *rigid* coupling (no flexible elements). As it is generally known, rigid couplings oppose misalignment with very large reactions forces, and because of this relatively small change in alignment, help in stabilizing the vibrations.

Case 3

Dupont had high vibration on a large process air compressor (Figure 20). The air compressor rotor was the one simulated in Figures 14, 15, 16, 17, 18, and 19. Although the vibration was not extremely high, the investigators had mild wiping on the first case pinion side bearing on two occasions and could not identify the cause. Nearly all of the vibration was at running speed. Even high speed balancing of the rotor did not change the vibration, while the opposite end of the first case rotor and the pinion ran smoothly. Then they discovered that the vibration, probe gap, and alignment changed with ambient temperature. This compressor was optically aligned and the hot alignment was known very precisely. The investigators traced the changing horizontal alignment to a loose anchor bolt on the steel skid, and corrected the problem by installing a centerline guide between the skid and the concrete pier. The vertical changes in alignment of the first case had to be addressed differently. Using the equations described herein, they found the highest forces at the shortest coupling, between the first and second case. *Although the misalignment forces were over three times the journal gravity loads,* they had no problems with these bearings! But, the net misalignment force on the problem bearing turned out to be nearly equal to the gravity load.

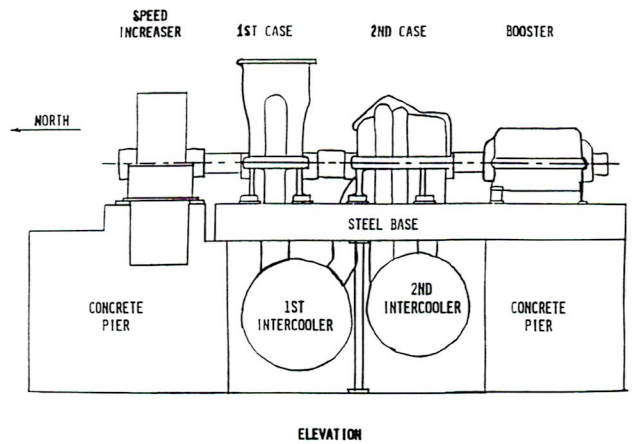


Figure 20. Industrial Air Compression Train With Gear Type Couplings.

The alignment on the coupling mesh next to the problem bearing was very nearly perfect when ambient temperature was hot, as shown in Figure 21. With this alignment, the normal changes in the first case elevation from changing inlet temperature allowed the direction of the forces to completely reverse. When the ambient temperature was high, as shown in Figures 22 and 23, the shaft ran in the bottom of the bearing and vibration was lower. When ambient temperature was low, the shaft ran in top of the bearing, but it was lightly loaded and had higher vibration.

The solution was to control the DIRECTION of the forces with direction of the misalignment. The magnitude of the friction moment depends only on friction and torque, not the amount of misalignment. They shifted the first case to the left of the pinion by

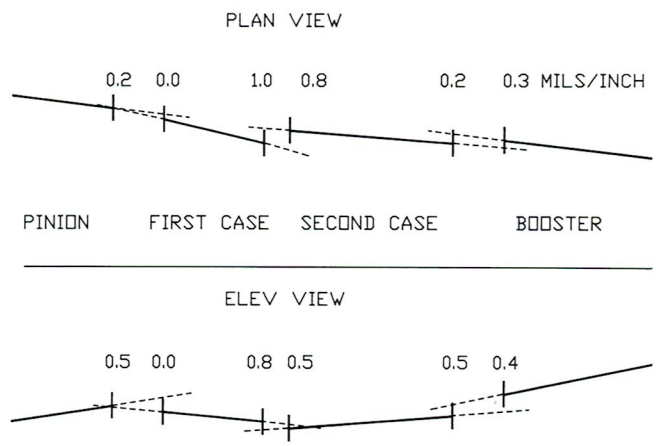


Figure 21. Diagrams for Initial Alignment of Air Compression Train.

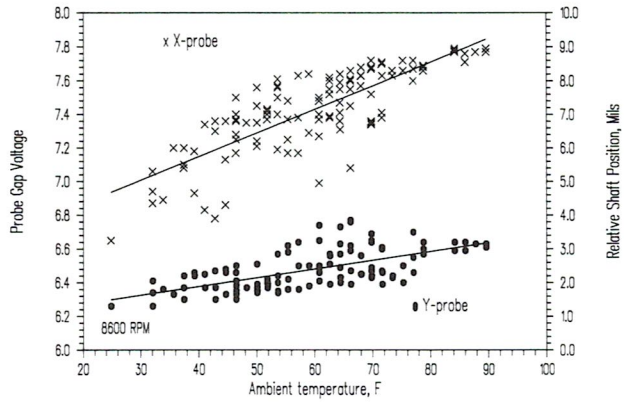


Figure 22. Shaft Movements in the Downward Direction Vs *Ambient Temperature for the Initial Alignment Case.*

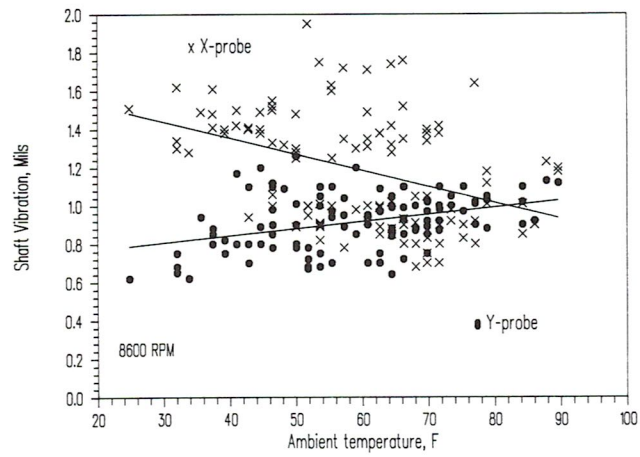


Figure 23. Shaft Vibration Vs *Temperature at Constant Speed (8600 RPM) for the Initial Alignment Case.*

about a half of a mil/in, as shown in Figure 24. This accommodated the normal casing thermal elevation changes and ensured a fairly constant direction of misalignment force down on the bearing, rather than at times opposing gravity. The results in Figures 25 and 26 show the shaft position was stabilized and vibration was

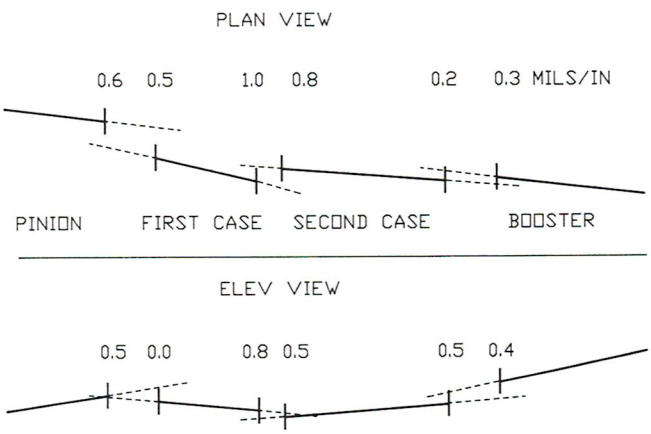


Figure 24. Alignment Diagrams Showing Applied Selective Directional Misalignment.

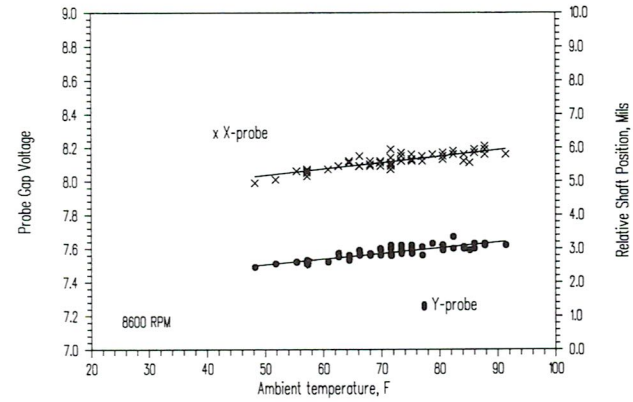


Figure 25. Shaft Movements in the Downward Direction Vs *Ambient Temperature for the Selective Directional Misalignment Case.*

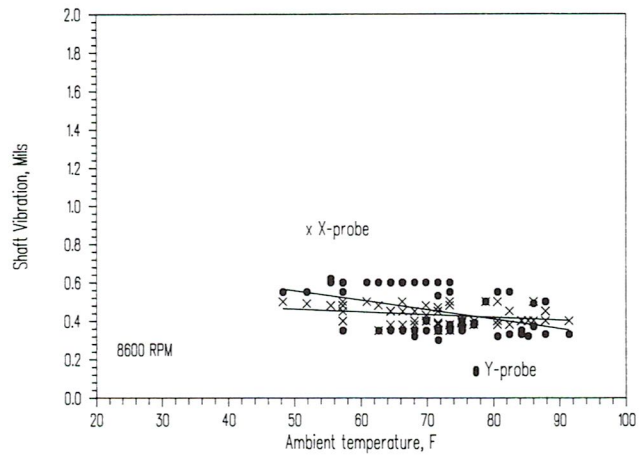


Figure 26. Shaft Vibration Vs *Temperature at Constant Speed (8600 RPM) for the Selective Directional Misalignment Case.*

reduced. The investigators found it surprising that the forces could be so large, and that even nearly perfect alignment would not reduce the forces. They subsequently asked the Texas A&M Turbomachinery Research Consortium to help verify their conclusions.

Case 4

A very dramatic reduction of vibration was made on a Dupont single stage overhung blower by controlling the misalignment force direction. This machine has been in operation for over 30 years and usually pounded out bearings in the blower and turbine about once a year, and occasionally forced the plant into an unplanned outage. They found that the location of the turbine could either greatly unload the blower bearings, or help preload them as shown in Figure 27. They set the machine for one mil/inch of misalignment with the turbine to the left of the blower, as shown in Figure 28. Typical of overhung rotors, the inboard blower journal normally runs in the top half of the bearing. The friction moment acts out of plane with the misalignment, as shown in Figure 29. Using a horizontal offset then results in vertical forces as shown in Figure 30. Choosing this alignment helped to increase the normal gravity loads on both the blower and the turbine. The results, shown in Figure 31 were a reduction of vibration levels of up to 1.0 in/sec down to a maximum of 0.15 in/sec.

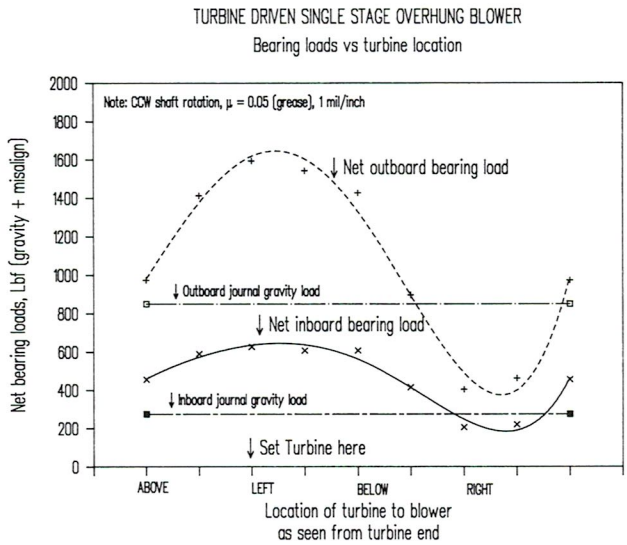


Figure 27. Total Downward Loads on the Overhung Blower Bearings Due to Gravity and Gear Coupling Moments.

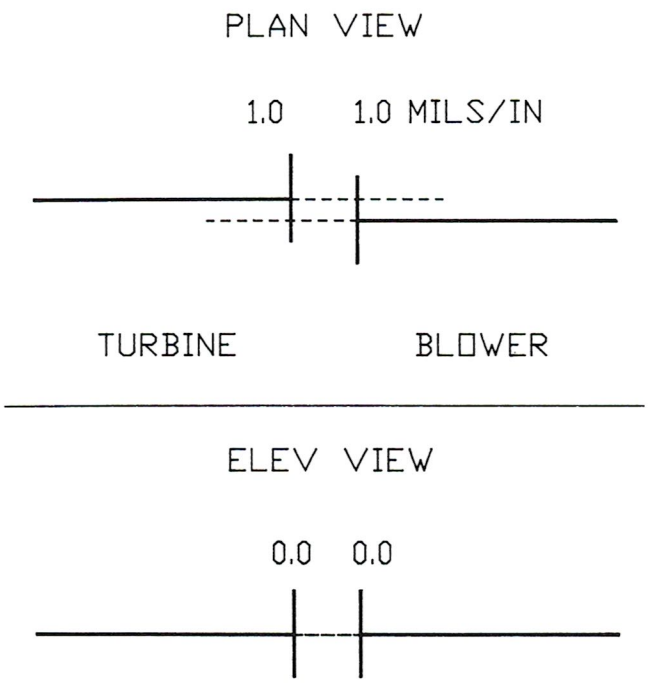


Figure 28. Selective Directional Misalignment Applied to the Overhung Blower Train.

VIBRATION RELATED ALIGNMENT GUIDELINES

The theory, test data, and case histories presented thus far were utilized to establish guidelines for recognizing and diagnosing coupling related vibrations. These guidelines are summarized below.

Couplings have a strong influence on machinery vibrations, and this has been known for some time; however, before actual studies and tests were performed, there was a lot of misunderstanding about how, and by what mechanism couplings can increase or decrease the amplitude of vibrations. Perhaps the best known "old wives tale" about machinery vibrations was that resonance at twice synchronous speed is caused by misalignment.

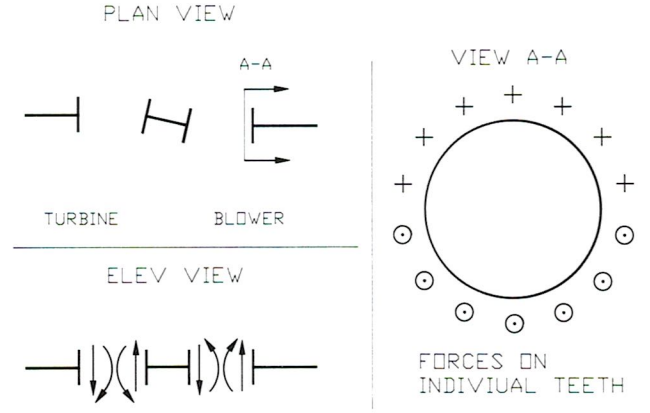


Figure 29. Diagram to Illustrate How Horizontal Misalignment Produces Vertical Loading.

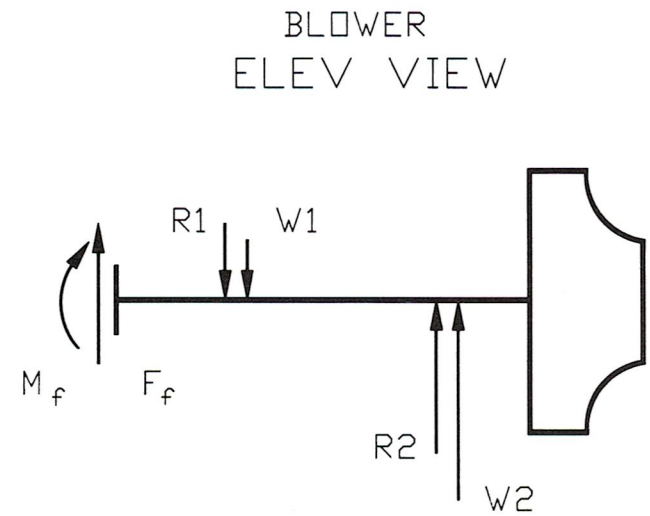


Figure 30. Vertical Reaction Forces on the Blower Rotor Due to the Selective Directional Misalignment.

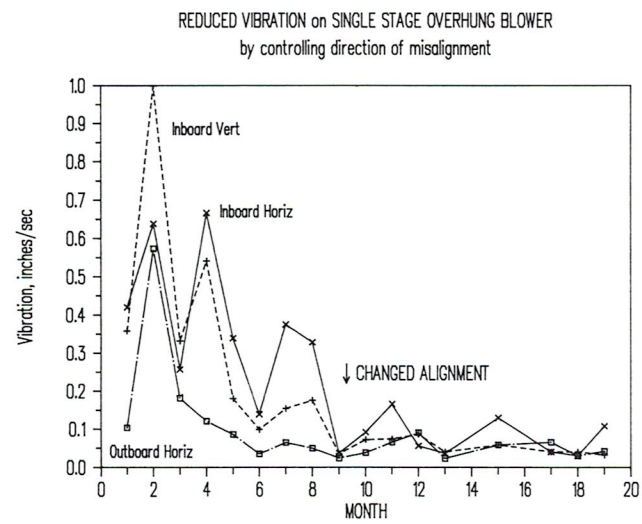


Figure 31. Vibration Reduction Achieved by Applying Selective Directional Misalignment to the Single Stage Overhung Blower.

• *Couplings oppose misalignment.* This has been shown both qualitatively and quantitatively by the tests and studies described herein. The reactive forces and moments are only to a very small extent a function of the amount of misalignment (Figure 10).

• *Couplings do not* create *vibrations.* It is through the forces imposed on bearings that misalignment can either stabilize or destabilize the orbit of a journal in its bearing. In fact, forces generated by misaligned couplings can modify the shape of an orbit such as either the amplitude of vibration decreases, or the frequency of vibration acquires a two times synchronous component. Two such conditions are exemplified in Figures 32 and 33. In highly loaded bearings, the orbit of the shaft's centerline is an elongated ellipse, as shown in Figure 32.

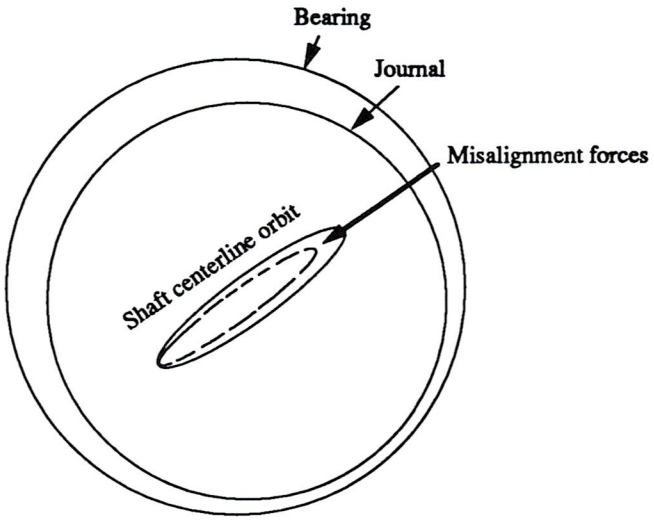


Figure 32. Misalignment Force Direction to Reduce Vibration in Elliptical Orbit.

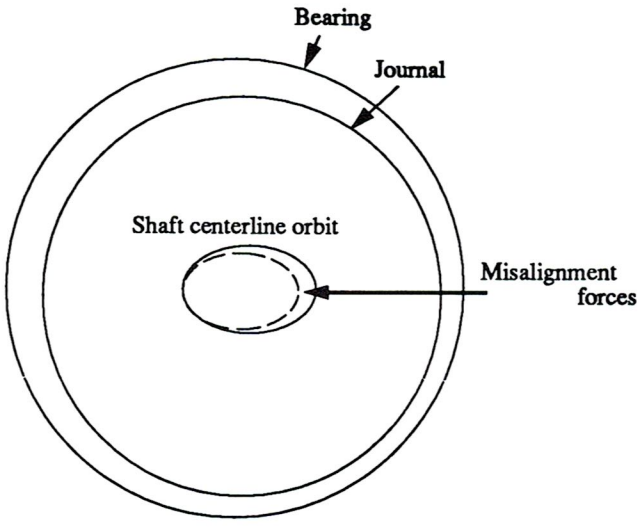


Figure 33. Vibrations in a Lightly Loaded Bearing.

• *Coupling forces may alter vibrations.* Assuming that the forces generated by coupling misalignment are in the direction of the long axis (Figure 32), the amplitude of vibrations decreases (dotted curve). Therefore, in such a case it is helpful to misalign the machines; however, only a study of the orbit will indicate the correct direction of misalignment! Assuming that the forces generated by misalignment are in the direction of the short axis (Figure 34), the ellipse becomes banana-shaped (dotted curve), which shape has two peaks per revolution. Therefore, the vibration has a twice synchronous component! It should be noted that in this case the forces generated by misalignment did not affect the amplitude of vibration, it only altered the shape of the shaft's orbit.

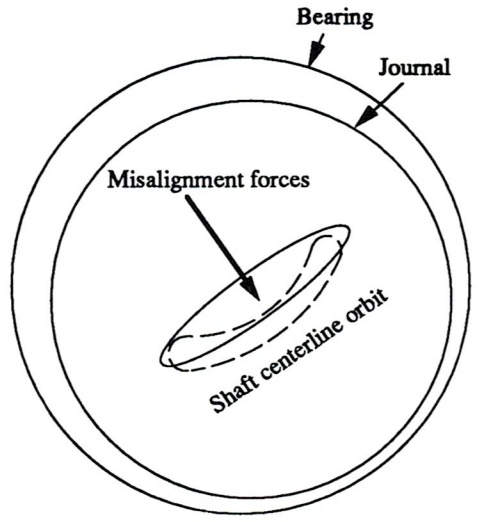


Figure 34. Misalignment Force Distorting an Elliptical Orbit into a Banana Shape.

In a lightly loaded bearing (such as a pinion or boiler-feed pump), the orbit of the shaft's centerline is shown in Figure 33. The orbit's shape is close to a circle because the unbalance forces are only slightly restrained by the forces acting on the shaft. Misalignment forces, independent of their direction, decrease the amplitude of vibration (dotted curve). Therefore, in such a case, it is helpful to misalign the machines. Obviously, the most benefit is realized if the forces created by misalignment are in the direction of the long axis. As these examples have shown, it is necessary to study the shaft's orbit to use misalignment as a vibration reduction tool. Therefore, it is necessary to equip machines with X-Y proximity probes, at least at the bearing next to the coupling.

• Misalignment and heat. The heat generated by misalignment of gear couplings imposes a restriction on the practice of intentional misalignment for vibration control. The friction moment depends only on friction and torque, not alignment. But misalignment in high speed turbomachinery must be kept small to avoid excessive friction heating, shown in Equation (14), derived by Locke. The other two moment terms do depend on alignment and approach zero with very good alignment. Because of the need to limit heating, the friction moment will generally be the dominant moment for turbomachinery.

$$hpf = 4 * \mu * \sin(\psi) * HP \tag{14}$$

• *Coupling spool length.* Shear forces are required for equilibrium as shown in the free body diagram in Figure 35. These shear forces are a direct function of the spool length per Equation (15), and can impose much larger reaction forces on the rotors than the friction moments!

$$sf = 2 * M_f/L \tag{15}$$

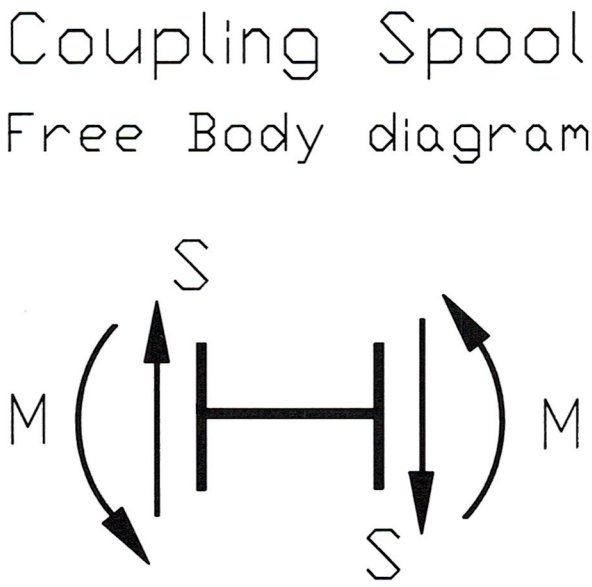


Figure 35. Shear Force Required to Balance Misalignment Induced Moments.

Although pure angular misalignment would in theory cancel the shear forces, pure angular alignment would be very difficult to sustain on a machine built for collinear alignment. So, whatever misalignment exists will most likely be parallel misalignment.

As an example, estimate the forces on the first case rotor from the pinion side coupling from case history 3. Assume the transmitted torque is 147,000 in-lbf and the coefficient of friction 0.05. The spool is 15 in between mesh centerlines, rotor span is 36 in and the shaft overhang 14 in. The friction moment and resultant forces on the rotor bearings are then:

$$Mf = 2/3 * \mu * T = 4900 \text{ in-lb} \tag{16}$$

$$Rf = 4900 \text{ in-lb}/36 = 136 \text{ lbf} \tag{17}$$

For a pure parallel offset alignment, the friction moments at the ends of the coupling spool will be equal and additive. These must be counterbalanced by the shear forces and bearing reaction forces:

$$Sf = 2 * Mf/L \tag{18}$$

$$= 2 * 4900 \text{ in-lb}f/15 = 653 \text{ lbf}$$

$$R1f = 906 \text{ lbf, pinion end}$$

$$R2f = 253 \text{ lbf, opposite end}$$

The opposite end of the first case rotor has a much shorter 5.5 in spool that produces frictional shear forces about three times larger! The net bearing loads on the first case depends on the alignment direction at both ends of the rotor. Using an API standard 18 in coupling spool would not reduce the bearing reaction forces below the normal journal gravity loads. Even much longer coupling spools would still produce very large forces and may have flexible shaft critical speeds.

• *Checking for misalignment forces.* Large misalignment forces will produce flattened or banana shaped orbits. The location of the orbit in the bearing indicates the direction of the actual hot misalignment forces.

• *Calculate just the shear force component.* The coefficient of friction for grease is typically 0.05, and 0.10 for oil. Forces more than half of the journal gravity load, may interfere with reliable operation. The actual net forces imposed on each bearing in a multicase machine has forces produced by each of the couplings involving all three moment terms and corresponding shear forces.

• *Controlling force direction.* The best approach to control the DIRECTION of the forces seems to be to use a SMALL amount of parallel offset misalignment in a known direction so that the most critical bearings are accommodated. Where possible, make the forces additive to the normal gravity loads or other machine loads, such as gear forces.

• *Amount of misalignment.* The amount of misalignment should be kept reasonable to limit heat generation in the teeth. On the machine in case history 3, there is normal wear on the couplings with 1 mil/inch of misalignment, which is a sliding velocity of about 3.5 in/sec at 8600 rpm. This has been sufficient misalignment to override changing alignment from normal thermal variations.

• *Type of misalignment.* Friction forces act out of plane with the misalignment. Thus, parallel misalignment in the plan view loads one bearing vertically up, and the other vertically down. This works well for an overhung machine with one bearing gravity loaded up, or where a gear runs in the top half of its bearing. It will also work in for between bearing machines where the forces are not the similar in magnitude to the journal gravity loads. The direction of rotation determines which way the driver should be offset to the driven, as discussed in case history 4. Elevation view misalignment will load the bearing on opposite sides, and may be beneficial for multicase machines, where the forces are similar in magnitude to the gravity loads.

CONCLUSIONS AND FUTURE WORK

The results in this study include:

• Presentation of new equations for misalignment gear coupling bending moments with an arbitrary number of teeth in contact.

• Results showing that the predicted bending moments depend on the assumed number of teeth in contact.

• Analysis and test configuration design for experimentally measuring gear coupling bending moments and friction factor.

• Test results for friction and bending moments, and for friction factor.

• Computer simulation results showing the effect of the bending moments on the vibration of an industrial rotor supported on three-lobe fluid film bearings.

• Case histories illustrating how gear couplings can affect, and can be employed to control vibrations.

• Guidelines for plant maintenance personnel and machinery designers to employ for vibration diagnosis and control.

Gear couplings are very reliable, lightweight, and used extensively in turbomachinery. Nearly perfect alignment does not reduce the friction force, degrades tooth lubrication and makes the misalignment direction uncertain. Frictional shear forces will usually dominate and depends on the coefficient of friction, spool length and direction of misalignment. Good alignment is necessary for long life and low tooth heat generation. *The authors do not advocate LARGE misalignments to control the direction of the forces.* For many machines, 0.5 to 1.0 mil/in of residual misalignment in a KNOWN DIRECTION will provide reasonable control of misalignment forces and enhance turbomachinery reliability.

Finally, when the misalignment forces are large compared to gravity loads, one should definitely consider the following:

- Bearing Capacity
- Bearing Stiffness and Damping, Rotordynamics
- Desired Bearing Load Angles and Required (Mis)alignment

NOMENCLATURE

D_p	Pitch diameter of gear
F_N	Normal force on gear tooth
HP	Horsepower
hpf	Horsepower loss due to friction
L	Spool piece length
M_f	Friction induced moment
M_K	Kinematically induced moment
M_X, M_Y	Moments about x and y axes, respectively
P	Diametral pitch
R	Face (crown) radius
R_{X1}, R_{X2}	Reaction forces on load cells in the test stand
S_f	Shear force
T	Torque on coupling
$\tilde{\mu}$	see equation 3
Γ	Torque measured in the high speed shaft of the test rig
θ_i	Angle of ith tooth
μ	Friction coefficient
ψ	Misalignment angle
β	Angle of the misalignment plane relative to the x (horizontal axis) in Figure 6

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ACKNOWLEDGEMENTS

The authors gratefully acknowledge the funding provided for this project by the 17 member company Turbomachinery Research Consortium at Texas A&M University. Appreciation is also extended to Mr. Robert Munyon of Kop-Flex for providing the test couplings at a discounted rate. Special thanks is offered to Ms. Wendy Harding for her assistance in preparing the manuscript.