

Warner Electric

Boston Gear

TB Wood's

Formsprag Clutch

Wichita Clutch

Marland Clutch

Industrial Clutch

Bauer Gear Motor

Nuttall Gear

Warner Linear

Delroyd Worm Gear

Stieber Clutch

Svendborg Brakes

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Lamiflex Couplings

Ameridrives Power
Transmission

Flexible Couplings for Gas Turbine Applications

Presented at the
**Gas Turbine & Aeroengine
Congress and Exposition**
June, 1989



An Altra Industrial Motion Company

Flexible Couplings for Gas Turbine Applications

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Gear Coupling

Abstract

The gear coupling was the primary flexible coupling for industrial gas turbines for almost twenty-five years. As advancements in the industrial gas turbine have occurred so have advancements in gear couplings. There are many variables in gear couplings that can direct their characteristics, including tooth design, materials, and lubrication methods. All couplings react on connected equipment. The system designer must consider these reactions when designing the system. If a gear coupling is chosen, there are many characteristics which can make a designer's life easier. This paper compares the characteristics of diaphragm couplings versus gear type couplings used in industrial gas turbine applications including applications of couplings for turbine to generator, turbine to mechanical drives (compressors, pumps, and gear) and also couplings used between the turbine and accessory gears. The methods used to analyze the design and the forces and moments generated by both the gear coupling and the diaphragm coupling are also provided. These analysis are used to show that the forces and moments generated by a diaphragm coupling are not only predictable, but usually lower than those of a gear coupling. The paper shows that a diaphragm coupling can be more predictable and reliable than a gear coupling for most industrial gas turbine applications.

Introduction

In today's technology, higher horsepowers and higher speeds continue to advance the art of rotating equipment design. This is especially evident in new gas turbine technology where reliability, weight, maintenance, and reaction loads on system components are of greater importance. Both the diaphragm coupling and the gear tooth coupling can usually be used for these high performance applications. When selecting couplings for these applications the system designer must look at the inherent design characteristics of the candidate couplings. Consideration should also be given to how well the flexible coupling will accommodate the application requirements without unleashing forces and moments that become very difficult to handle.

Some of these examples include load couplings for generator and mechanical drives and also accessory drive coupling.

All couplings react on connected equipment components when subjected to misalignment and torque. These reaction forces are greater for some couplings than others and if not considered can cause failure of shafts, bearings, and other equipment components. When these reactions are large or unpredictable the impact on the system is usually increased size and weight.

Types of Couplings

Couplings can be basically categorized as one of two types, the rigid

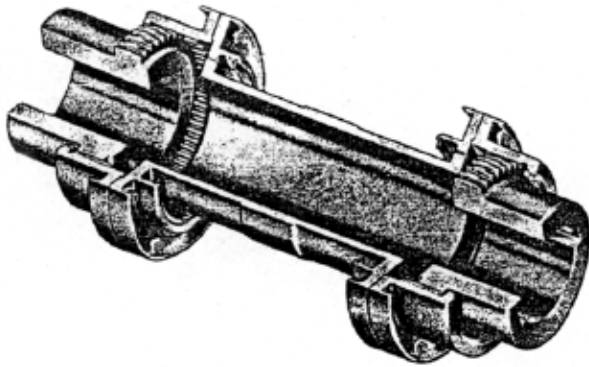


Figure 1: High Performance Gear Couplings

coupling and the flexible coupling. Rigid couplings are usually used to connect equipment that experiences very small misalignments, since most applications require the use of flexible couplings. Flexible couplings are categorized as one of four types:

1. Mechanical flexible
2. Elastomeric
3. Metallic Membrane
4. Miscellaneous

For the applications being compared, a mechanically flexible gear coupling or a metallic membrane coupling are most often considered because of torque, misalignment, speed, and environmental requirements. The gear coupling is a type of mechanically flexible coupling and the diaphragm is a type of metallic membrane coupling. Both types of couplings have unique characteristics that are suitable for the applications being considered.

Gear Couplings

A gear coupling (Figure 1) accommodates misalignment and shaft end float by the misalignment and sliding of the gear teeth on the hub in the sleeve (Figure 2). Due to this mechanical motion, these couplings must be lubricated. The most common mode of failure for a gear coupling is wear. It is one of the most common and simplest couplings used today. Due to the number of variables that can affect its successful operation it is usually difficult to design and evaluate. Some of the variables affecting its design and characteristics are:

1. Tooth Design
 - A. Straight or the type and amount of crown
 - B. Pressure angle of tooth
 - C. Amount of backlash
 - D. Accuracy of tooth spacing

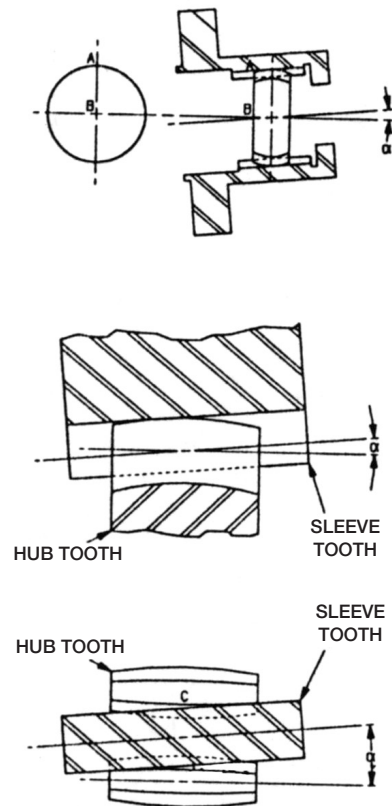


Figure 2: Misalignment of Crowned Tooth Coupling

2. Material
 - A. Type of material(s)
 - B. Type of core heat treatment
 - C. Type of surface treatment
3. Lubrication
 - A. Oil
 - B. Grease
 - C. Sealed lubrication
 - D. Continuous lubrication
- E. Method of lubrication

A gear coupling has its most significant effect not only on itself, but on system components from the forces and moments generated when it slides and/or misaligns. When a gear coupling accommodates the shaft float from thermal growth, foundation deflection, etc., axial forces react back onto the thrust bearings and other equipment. When misaligned a gear coupling will produce a bending moment that will load equipment shafts, bearings, and other system components. Both the axial forces and bending moments are significantly affected by the lubrication and the coefficient of friction between the mating gear teeth members.

Diaphragm Couplings

A diaphragm coupling (Figure 3) is one type of metallic membrane coupling. A diaphragm coupling consists of one or more metallic membranes which are attached at the outside diameter of a drive flange and transfers torque radially through the diaphragm to an inside diameter attachment. The other type of metallic membrane coupling is the disc coupling. A disc coupling will usually consist of several flexible metallic membranes which are alternatively attached with bolts to opposite flanges (Figure 4).

Metallic membrane couplings were developed because of a desire to eliminate the problems associated with a lubricated coupling. A metallic membrane coupling relies on the flexure of metallic materials to accommodate axial travel and misalignment of equipment shafts. The axial force imposed on bearings and equipment is a factor of the deflection imposed on the diaphragm and the axial stiffness of the diaphragm flex unit. The bending moment that is transferred to the shafts, bearings, etc., is a function of the diaphragm bending stiffness of the flex unit and the amount of misalignment imposed.

Couplings For Industrial Gas Turbines

Many industrial gas turbines are derivatives of the Aircraft gas turbine or use many of their principles. These units operate at a high speed (3600 -15000 RPM). They usually require couplings with large axial travel capabilities to over 1/2 inch. These applications require couplings that are usually custom designed and use aircraft technology in their design and construction.

These couplings are made of high strength alloys and are manufactured and constructed to specification and tolerances employed in making aircraft components.

Figure 5 is a typical heavy duty industrial gas turbine. The turbine shown can and does use both gear couplings and diaphragm couplings.

Accessory Coupling Requirements (Figures 6 & 7)

Normal Torque	18,000 IN-LB
Peak Torque	194,400 IN-LB
Speed	5,500 RPM
Misalignment	$\pm 1/4$ Degrees
Axial	$\pm .500$ IN

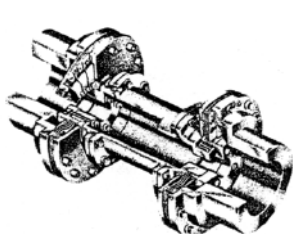


Figure 3: Diaphragm Coupling

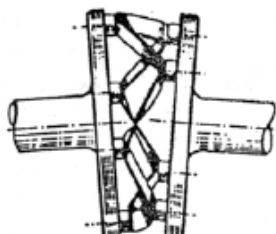


Figure 4: Disc Coupling

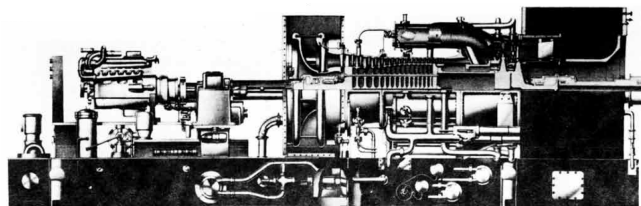


Figure 5: Heavy Duty Industrial Gas Turbine

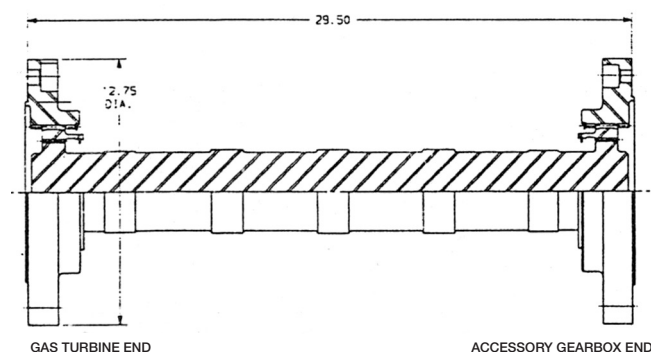


Figure 6: Gear Accessory Coupling

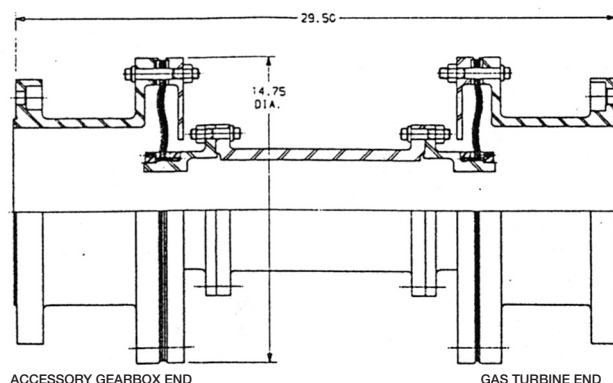


Figure 7: Diaphragm Accessory Coupling

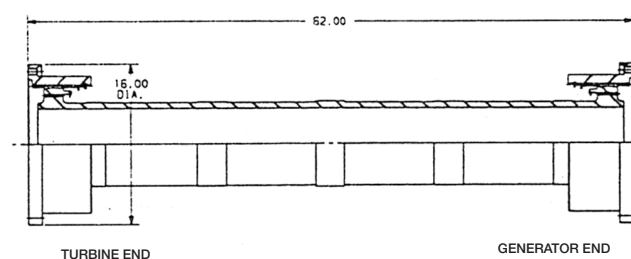


Figure 8: Gear Loading Coupling

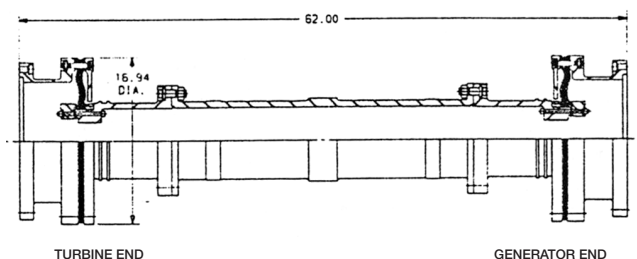


Figure 9: Diaphragm Loading Coupling

Load Coupling Requirements (Figures 8 & 9)

Normal Torque	534,000 IN-LB
Peak Torque	2,640,000 IN-LB
Speed	6,000 RPM
Misalignment	$\pm 1/4$ Degrees
Axial	$\pm .300$ IN

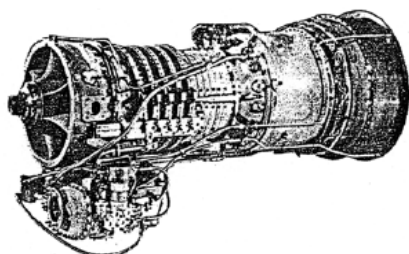


Figure 10: Turbine Requiring High Speed Coupling

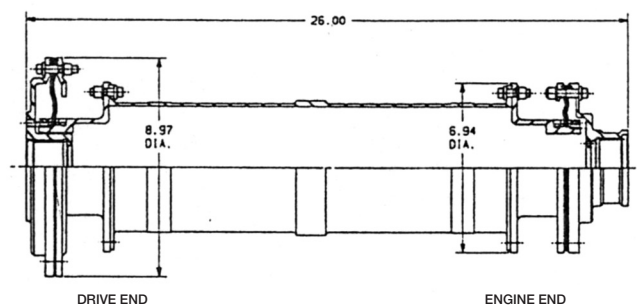


Figure 11: High Speed Gas Turbine Diaphragm Coupling

Many advanced gas turbines have been designed to use diaphragm couplings only.

High Speed Gas Turbine (Figures 10 & 11)

Torque	43,300 IN-LB
Speed	11,500 RPM
Misalignment	$\pm 1/4$ Degrees
Axial	$\pm .125$ IN

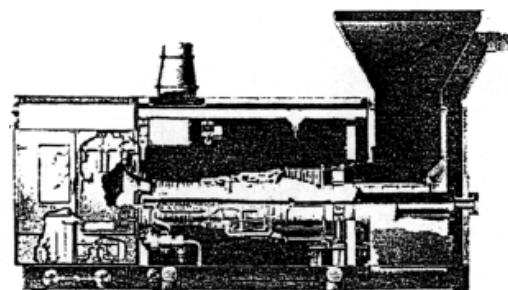


Figure 12: Gas Turbine with Coupling in Exhaust

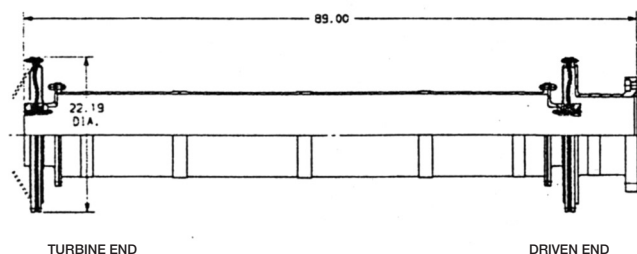


Figure 13: High Temperature Gas Turbine Coupling

High Temperature Gas Turbine (Figures 12 & 13)

Torque	528,000 IN-LB
Speed	3,600 RPM
Misalignment	$1/2$ Degrees
Axial Travel	$\pm .400$ IN
Operating Temp.	600 Degrees F

Gear couplings can be classified in many ways, but the most common classification is by the type of tooth form. All gear couplings have straight-sided internal gear teeth. These internal teeth are involute formed, usually at a 20 degree pressure angle. Most of the backlash required to misalign a gear coupling is generally cut into the internal teeth. The two types of hub tooth form used to classify gear couplings are:

1. Straight Sided (as described above - Figure 14)
2. Crowned Tooth (Figure 15)

A crowned tooth is crown hobbled or shaped with a cutter on a cam. For small misalignment angles of $1/8$ degree or less, a straight tooth works almost as well as a crowned tooth. For angles greater than $1/8$ degree, a crowned tooth should be used as the crown allows greater tooth contact during misalignment which reduces stresses.

One of the most important considerations in the design of gear couplings is the amount of heat generated by the gear under load and misalignment. There have

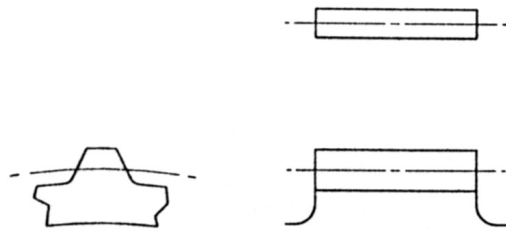


Figure 14: Straight Sided Hub Tooth

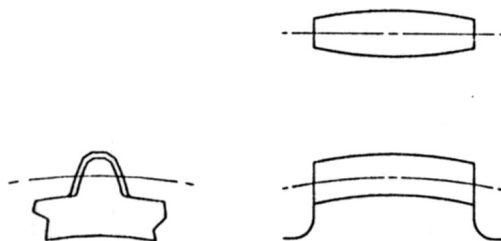


Figure 15: Crowned Hub Tooth

been many attempts to establish limits and criteria. The most commonly used is compressive stress or sliding velocity, but neither has been proven to be sufficient. Testing and experience have shown that a design criteria for the teeth of gear couplings should consider the following:

- Load
- Misalignment
- Tooth geometry (pitch diameter, diametral pitch, and amount of tooth crown)
- Percent of teeth in contact
- Speed
- Material, hardness of teeth and finish
- Type and quality of lube

The compressive stress-sliding velocity criteria ($S_c V$) can be used to compare equivalent couplings and applications. Limits on ($S_c V$) depend on the type of lubrication and the material used for a particular coupling.

Compressive Stress - Axial Sliding Velocity Criteria

- | | | |
|----|---|---|
| T | = | Torque (IN-LB) |
| PD | = | Diameter of Pitch Circle of Gear Tooth (IN) |
| h | = | Active Tooth Height (IN) |
| D | = | Diameter of Curvature of Tooth Face (IN) |
| DP | = | Diameter Pitch of Gear Tooth |
| N | = | Maximum Speed (RPM) |
| a | = | Full Load Misalignment Angle (Degree) |
| FW | = | Gear Tooth Face Width (IN) |
| R | = | P.D./2 (IN) |
| n | = | Number of Teeth in Coupling |
| c | = | Percentage of Teeth in Contact |

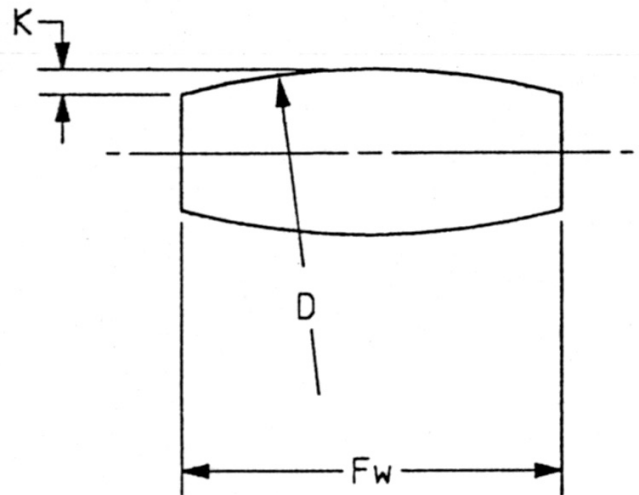


Figure 16:

For crowned teeth:

$$S_c = 2290 \sqrt{\frac{T}{(PD/2) \times (D/2) \times c \times h \times (DP \times PD)}} = \text{PSI}$$

If tooth curvature is expressed in amount of crown:

K = Amount of Crown: - (IN) (See Figure 16)

$$D = \frac{(FW/2)^2}{K} = (\text{in.})$$

$$h = \frac{1.8}{DP} = (\text{in.})$$

$$S_c = 3600 \sqrt{\frac{T}{(PD)^2 \times D \times c}} = \text{PSI}$$

For straight-tooth couplings:

$$S_c = \frac{T}{RnA} = \text{PSI}$$

$$A = \frac{1.8 \times FW}{DP}$$

Maximum sliding velocity (V) of gear coupling:

$$V = \pi \times PD \times \sin \alpha \times \frac{N}{60} = \text{IPS}$$

The $S_C V$ criteria can be used to compare equivalent couplings and applications:

$$S_C V = S_C \times V = \text{PSI} - \text{IPS}$$

Some typical values based on testing and field experience are given in Figure 17.

How Gear Couplings Affect The System

The most significant effect that a gear coupling has on the system comes from the moments and forces generated when it glides and/or misaligns.

1. AXIAL FORCE - The formula for the axial force reacting on the thrust bearings and other system components is:

$$\text{FORCE} = \frac{T \times \mu}{R} = \text{LB}$$

Where T = Torque (IN-LB)

μ = Coefficient of friction

R = Pitch radius of gear (IN)

Various coupling manufacturers use the following typical values from test data for the coefficient of friction:

Sealed Lubricated Gear Couplings:

$$\mu = .05$$

Continuous Lubricated Gear Couplings:

$$\mu = .075$$

Values of coefficient of friction higher than the above can be experienced, although if they are present for any period of time, the coupling is no longer flexible which would more than likely precipitate the failure of one of its own components or some component, of the coupled

Compressive stress/sliding velocity comparison	
Material	AISI 1045
Lubrication	Grease packed
Misalignment	3/4 Degrees
ScV (PS1-IPS)	1,500,000
Material	AISI 4140 with nitrided teeth
Lubrication	Continuously lubed
Misalignment	1/4 Degrees
ScV (PS1-IPS)	750,000
Material	Nitralloy N with nitrided teeth
Lubrication	Grease packed
Misalignment	4-1/2 Degrees
ScV (PS1-IPS)	6,000,000
Material	AISI 3310 with carburized teeth
Lubrication	Grease packed
Misalignment	3 Degrees
ScV (PS1-IPS)	500,000

Figure 17: Compression Stress/Sliding Velocity Comparison

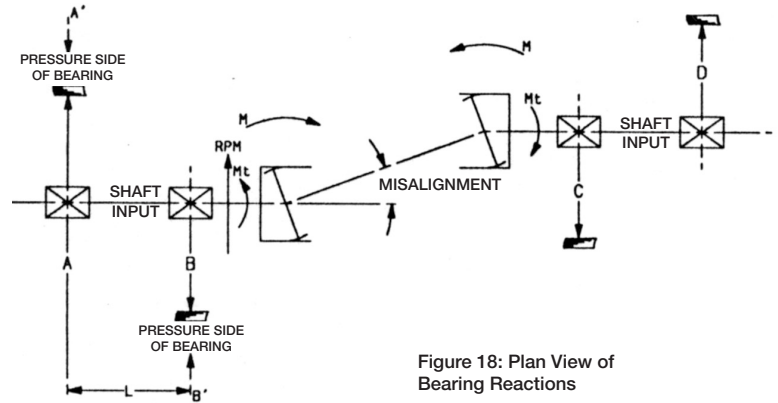


Figure 18: Plan View of Bearing Reactions

equipment. If a coupling is mechanically locked due to sludge or wear, the forces could be increased seven to eight times those normally expected.

There is still much discussion of how large a value to use for a safety factor when designing a system or even a gear coupling. Due to the high reliability necessary for gas turbine application, American Petroleum Institutes Specifications require thrust bearings to be designed for the maximum gear coupling thrust load with the conservative coefficient of friction of = .25 for continuous operating conditions.

2. Bending Moment - The moments produced in a gear coupling will load equipment shafts and bearings (see Figure 18) and also change the operating characteristics of the equipment.

The three basic moments in a gear coupling as referenced in Figure 19 are:

A. Moment generated from transmitted torque and angle (rotated around the Z - Z axis)

B. Moment generated from frictional loading (rotated around the Z-Z axis)

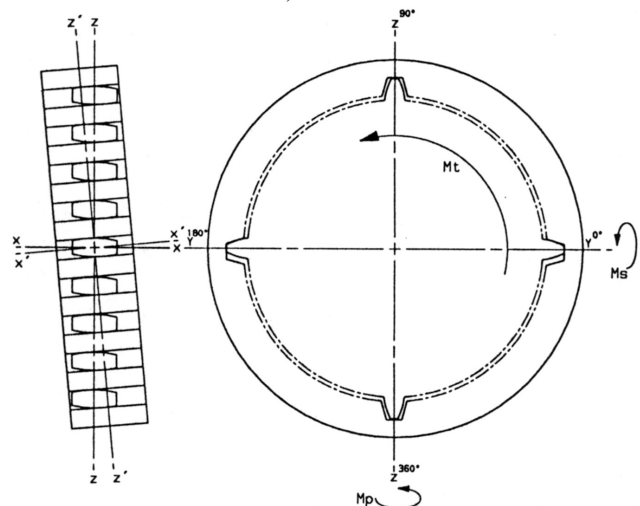


Figure 19: Coordinate System for Gear Coupling

C. Moment generated from displacement of the load from its center rotated around the Y - Y axis

As in axial forces, the value of the coefficient of friction significantly affect the moments generated. The following is a detailed analysis of the moments in gear type couplings.

Bending Moments

First, one must define a coordinate system for a gear coupling. In Figure 19, the line X' - X' defines a cut through the plane of misalignment. Moments that tend to rotate the coupling around the Z - Z axis will be designated M_z . Moments that tend to rotate the coupling around the Y - Y axis will be designated M_y . T is torque in IN-LB.

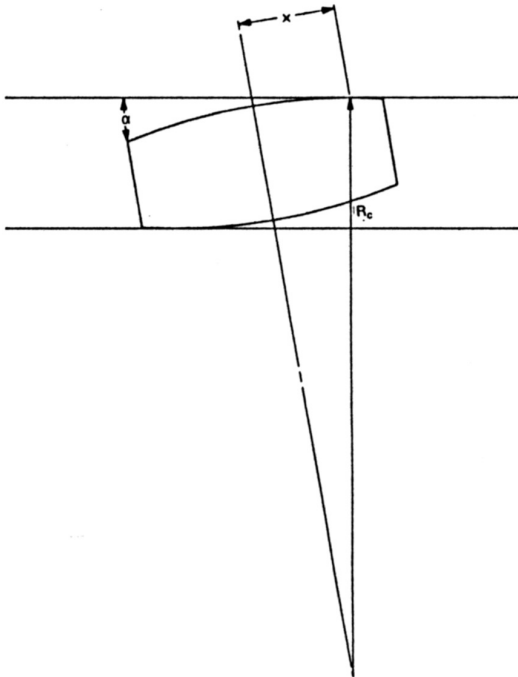


Figure 20: Gear Tooth Sweep

Gear Tooth Sweep:

$$\sin \alpha = X/R_c$$

Where: X = Distance from the centerline of gear tooth to the point along the flank of the gear contact at α misalignment (Figure 20).

R_c = Radius of curvature of gear teeth (IN)

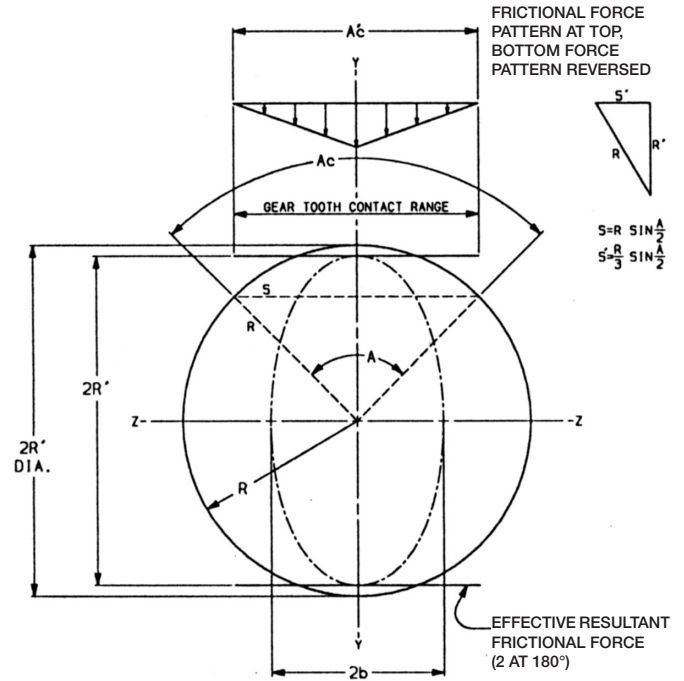


Figure 21: Average Radius for Misaligned Gear Coupling

Average radius for misaligned coupling (Figure 21):

$$R' = \sqrt{R^2 - \left[\frac{R}{3} \sin \frac{A}{2} \right]^2}$$

Where: R = Pitch radius of gear (IN)

A = Angle for gear tooth contact range (Degrees)

The moments causing rotation around the z axis:

$$M_z = \frac{T(\sin \alpha - \mu \cos \alpha) \sqrt{R^2 - \{(R/3) \sin (A/2)\}^2}}{R(\cos \alpha - \mu \sin \alpha)}$$

Moments causing rotation around the Y - Y axis:

$$M_y = \frac{M_z x}{R}$$

Resultant moment is the vectorial combination of two moments:

$$M_z = \sqrt{(M_z)^2 + (M_y)^2}$$

at approximately $\partial = \tan^{-1} (2R/Dc)$ from Y in the ZY plane.

Final combined moment equation:

$$M = \sqrt{\left[\frac{T(x)}{R} \right]^2 + \left[\frac{T(\sin \alpha + \mu \cos \alpha) (R')}{R(\cos \alpha - \mu \sin \alpha)} \right]^2}$$



Figure 22:

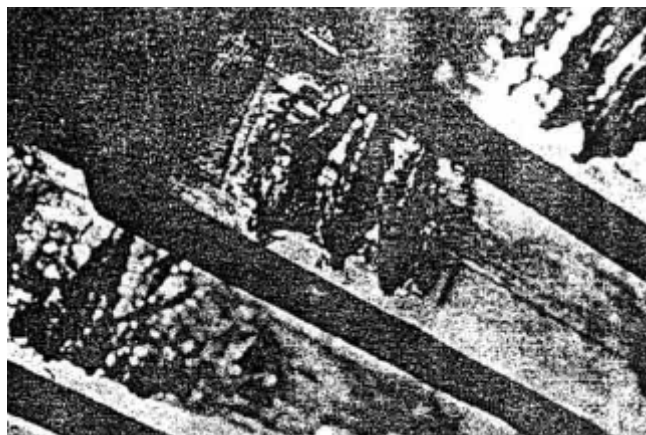


Figure 24:

Failure Mode For Gear Couplings

For gear couplings the most common type of failure is due to tooth wear and distress. Tooth distress is most commonly caused by:

1. Inadequate lubrication (Figure 22)
2. Improper tooth contact (Figure 23). In this case the coupling ran close to a torsional critical and the cyclic load caused localized tooth distress only on those teeth loaded.
3. Worm Tracking - Cold flow or welding occurs more frequently on continuous lubed couplings. This occurs near the end of the teeth when misalignment approaches the design limit of the crown and the lubrication film breaks down and causes metal-to-metal contact. High, localized tooth loading or lubricant deterioration can cause this type of failure as described in number 2 above (Figure 24).



Figure 23:

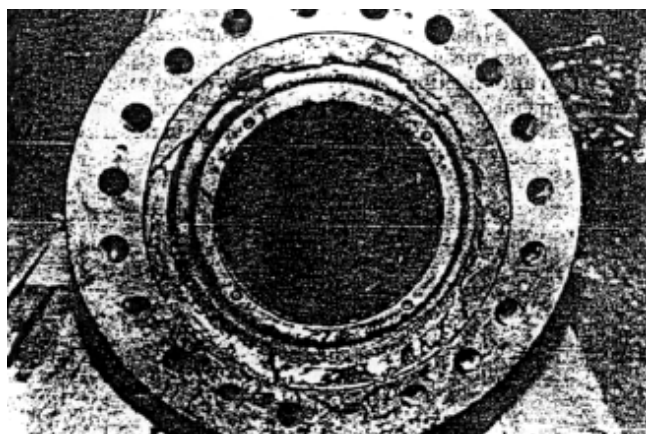


Figure 25:

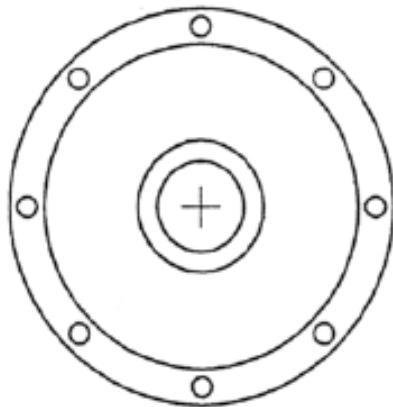
4. Sludge build-up can cause failure of almost any system component such as shafts or bearings (see Figure 25). Sludge also will collect corrosive residue, which can corrode coupling parts and act as a source of crack initiations for a fatigue-propagated failure of a part.

Diaphragm Coupling Design Considerations

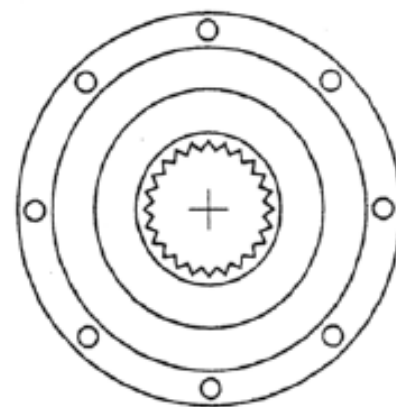
As stated earlier, there are two basic types of flexible metallic membrane couplings, the diaphragm and the disc. Either type can be used singularly or as a pair separated by a spacer or spool piece.

Metallic membrane coupling use was limited through the 1940's to low torque, low speed applications where only limited amounts of misalignment were required. In the late 1940's the appearance of the small gas turbine produced the need for the thin contoured diaphragm, which saw usage in aircraft applications. The progress and acceptance of this coupling in industrial and marine applications was greatly hindered by the inability to accommodate high misalignment and large axial movements.

For many years gear couplings have been used on



Tapered Contoured Diaphragm



Multiple Convolved Diaphragm

Figure 26: Typical Diaphragm Coupling

steam turbines, gas turbines, compressors and pumps. When the horsepower, speeds, and operating temperatures increased, many problems with gear couplings developed. The need for lower moments, forces, and noise characteristics has pushed the advanced development and usage of diaphragm couplings in thousands of applications. Because of this developed technology, diaphragm couplings have been successfully used since the mid 1970's for gas turbine applications.

Many manufacturers and users of rotating equipment have increased their list of coupling requirements to include the following:

- A. No lubrication
- B. Higher torque capability without an increase in coupling size
- C. Accommodation of greater misalignments
- D. Accommodation of greater axial motions
- E. Suitable for high temperature operation
- F. Adaptable to all types of connections: splines, taper shafts, flanges, etc.
- G. Produce low moments and forces
- H. Produce predictable moments and forces
- I. Easily balanced
- J. Operate for years without maintenance or problems
- K. Produce low vibratory inputs into equipment

Diaphragm couplings for these applications are

available in two basic forms (Figure 26):

- A. Tapered contoured
- B. Multiple convoluted diaphragm

Both shapes have some type of profile modification that helps reduce size, increase flexibility and control stress concentrations. For example, in order to increase flexibility, tapered contoured diaphragms are designed for a constant shear stress from the inner diameter to the outer diameter.

Couplings using multiple diaphragm designs, with a number of thin plates in parallel rather than a single thick one, have improved flexibility and usually lower stresses. Diaphragm stresses, moments and forces increase with the cube of the material thickness (t). Therefore, several thin diaphragms will produce lower stress values than a single thick one.

Since diaphragm couplings are usually on high performance equipment which require high reliability, coupling components are usually made of high strength alloys with good fatigue properties. Typical materials are AISI 4100 or 4300 steels coated for corrosion protection, PH stainless steels or high strength nickel alloys.

Diaphragm couplings, like all metallic membrane couplings, are usually designed for infinite life. In applying membrane couplings to gas turbine applications the most important design consideration (in relation to the flexible membrane) is the operating stresses in the flexible membrane. The stresses in the diaphragm or disc are

W = AXIAL
LOAD

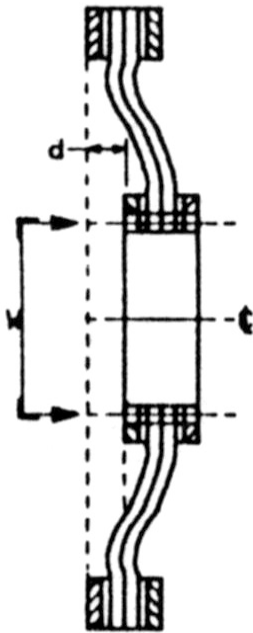


Figure 27: Angular Deflection of Diaphragms

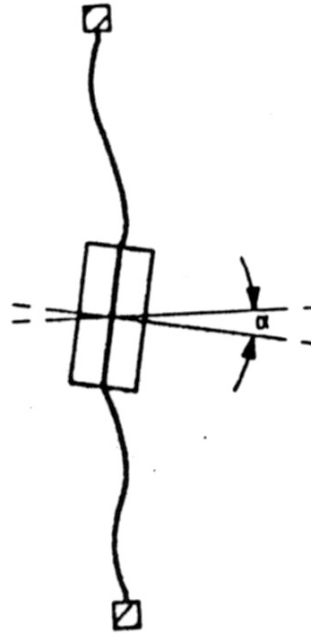


Figure 28: Angular Deflection of a Single Diaphragm

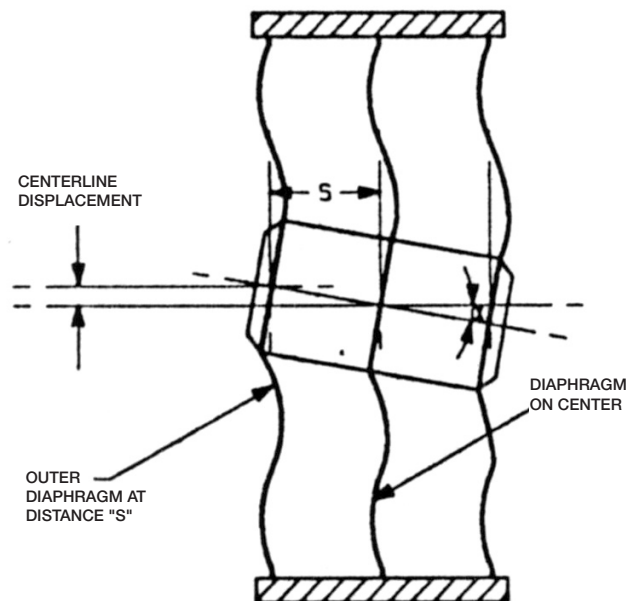


Figure 29: Deflection in Diaphragm Due to Offset

designed to be less than the endurance limit of the material used with some factor of safety.

Diaphragm Stresses

As an example, a multiple convoluted diaphragm will be used, but the approach is very similar for most flexible membrane couplings as most have the same type of stresses.

To understand diaphragm stresses it is important to understand how the diaphragm reacts to the various types of misalignment. Figures 27, 28, and 29 are exaggerated in order to demonstrate more clearly the diaphragm reaction to various forces. It is also important to realize that a diaphragm on the centerline of the diaphragm pack reacts differently than a diaphragm off the centerline, when angular misalignment is imposed on the entire pack. Some of the stresses resulting from the diaphragm deflection shown are continuous during the entire period of operation, and these are termed "Steady State". On the other hand, some of the stresses not only vary, but go through complete reversals during each revolution. These are termed "Alternating".

Steady State Stresses

The steady state stresses are considered to be the stresses which result from: axial displacement of the diaphragm (Figure 29), torque transmission, centrifugal effects and thermal gradient effects.

Axial Stress (S_A) (Figure 27)

(S_A) is determined by the amount of deflection imposed on the diaphragm. W is axial force, d is distance moved. The stiffness of pack is equal to: $K_A = W/d$. The stiffness of a coupling is equal to: $K_A = W/2d$.

Shear Stress (T)

T occurs when torque is transmitted through the diaphragm pack and is dependent upon the size, number and thickness of the diaphragms. Shear stress T is highest at the inside diameter of the diaphragm. NOTE: In cases where torque is cyclic or reversing this stress or part of it must be combined with other dynamic stresses.

Centrifugal Stress (S_C)

(S_C) always results when the coupling is rotated and this rotational effect on the diaphragm must be combined with the other steady state stresses.

Thermal Gradient Stress (S_t)

(S_t) applies only where there is a temperature differential across the surface of the diaphragms and/or where there is a coefficient of expansion difference. If this case exists the combined thermal stress must be calculated.

Alternating Stress

(S_b) two cases are considered which contribute to the total alternating stress: angular flexure stress (Figure 28) and stresses due to offset deflection (Figure 29).

Flexure Stress (S_F) (Figure 28)

(S_F) is the result of the angular misalignment of the coupling. (α) is misalignment angle in degrees.

Offset Stress (S_o) (Figure 29)

(S_o) is caused by angular deflection of the inside diameter of the diaphragm with respect to the outside diameter. The diaphragm elements which are spaced axially away from the centerline of flexure experience a stress proportional to the distance that they are removed from the centerline. Figure 23 shows a cross section of an exaggerated pack and how the outer diaphragms are compressed or stretched due to the distance(s) removed from the center of misalignment.

Combined Diaphragm Stress

All the stresses presented above are calculated at the inside of the diaphragm and for the furthest diaphragm from the centerline which are then combined in the following manner to give the highest stress point in the diaphragm pack. Axial stress, thermal stress and centrifugal stress are additive. Total steady state normal stresses (S) are computed as follows:

$$S = S_A + S_t + S_C$$

Coupling Comparisons

Coupling Type	O.D. (in)	Weight (lbs)	Continuous Torque Condition (in-lbs)	Continuous Axial Travel (in)	Axial Force at Continuous Conditions (lbs)	Continuous Angular Misalignment (Degrees)	Bending Moment at Continuous Conditions (in-lbs)
Gear Accessory Coupling ($\mu = .075$)	12.75	190	18,000	—	440	$\pm .25$	1,900
Gear Accessory Coupling ($\mu = .25$)	12.75	190	18,000	—	1,450	$\pm .25$	4,800
Diaphragm Accessory Coupling	14.75	250	18,000	$\pm .500$	1,050	$\pm .25$	400
Gear Load Coupling ($\mu = .075$)	16.00	480	534,000	—	6,870	$\pm .25$	56,300
Gear Load Coupling ($\mu = .25$)	16.00	480	534,000	—	22,880	$\pm .25$	140,800
Diaphragm Load Coupling	16.94	580	534,000	$\pm .300$	4,050	$\pm .25$	5,600

* - Gear type couplings can be designed for unlimited axial capacities

Figure 30: Coupling Comparisons

These are then combined with the shear stress to produce the combined steady state stress (S_M).

$$S_M = \frac{S}{2} + \sqrt{\left(\frac{S}{2}\right)^2 + \tau^2}$$

Total alternating stress (S_B) is conservatively determined by the simple summation of the offset and flexure stresses. Where no cyclic torque is present:

$$S_B = S_o + S_F$$

If cyclic torque is present:

$$S_B = \frac{S_o + S_F}{2} + \sqrt{\left(\frac{S_o + S_F}{2}\right)^2 + \tau^2}$$

Finally, the mean stress and the alternating stress resulting from bending of the diaphragm are combined by the Soderberg equation.

$$\frac{1}{N} = \frac{S_M}{S_{yld}} + \frac{S_B}{S_{end}}$$

N = Safety factor

S_{yld} = Yield strength of material, psi

S_{end} = Endurance strength of material, psi

1. Axial Force - The axial force reacted back to the bearings is:

$$F = K_A (d)$$

K_A = Axial stiffness of the coupling (LB/IN)

d = Axial deflection (IN)

2. Bending Moment - The bending moment reacted to system shafts and bearings is:

$$M = K_b (\alpha)$$

Where:

K_b = Bending stiffness of the metallic membrane (IN-LB/DEGREES)

α = Angle of misalignment (Degrees)

How Diaphragm Couplings Affect The System


The forces and moments produced by a diaphragm coupling are caused by the axial and angular deflection of the metallic membrane.

Comparison Between Gear Couplings And Diaphragm Couplings

Figure 30 lists a comparison of load and accessory drive diaphragm and gear couplings for a typical industrial gas turbine. Coupling diameter, weight, torque conditions, bending moment and axial force are listed.

The forces and moments are calculated for two different coefficients of friction to demonstrate the impact of this variation for gear couplings. The coefficient of friction will vary with the tooth design, type and quantity of lube, the types of material and tooth finish. For a diaphragm coupling the moments and forces are predictable and are simply related to the stiffness of the metallic membrane. Although a gear coupling will typically have a smaller diameter and a lighter weight for very high axial travel requirement applications, the bending moment imposed by the gear coupling on the system is still up to 10 times higher than the equivalent diaphragm coupling. The axial force imposed by the gear coupling can be equivalent to the diaphragm coupling dependent upon the coefficient of friction. Under peak torque conditions (5 to 10 times the normal operating torque) the axial force can be 9 times greater for the gear coupling and the bending moment 50 times greater than a comparable diaphragm coupling.

Figure 30 also compares calculations based on the



requirements of API specifications (coefficient of friction equals .25). The impact of these moments and forces on system design is even more significant and can require larger shafts, bearings, thicker flanges, etc., with the gear coupling.

For gas turbine couplings, reliability and maintenance considerations also show an advantage for the diaphragm coupling. Although gear couplings have been known to last twenty-five years or more, periodic sludge removal for continuously lubed couplings and replacement of the lubrication for sealed couplings are standard maintenance requirements.

Summary

Both the gear coupling and the diaphragm coupling can be used in industrial gas turbine applications. The system designer must analyze the characteristics such as the reaction forces and moments produced by each type of coupling. The diaphragm coupling forces and moments are not only lower than a gear type coupling, but more predictable.

References

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