Custom Designed Couplings

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Custom Designed Couplings

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Gear Spindle

Abstract

Flexible couplings are too often thought of as a piece of hardware, rather than a vital part of a power transmission system. Flexible couplings are often bought on price rather than suitability or functionality.

Many applications of rotating equipment do not require much thought about the coupling that connects the driver and driven equipment, but there are many applications that are critical and very sophisticated. These usually require custom designed couplings to accommodate the system. The applications that require custom designed couplings are not limited by horsepower or speed considerations, but are dictated by the need for a coupling that will provide dependable operation under specified conditions.

Custom designed couplings are manufactured for specific equipment, versus the so called standard or stock couplings, which are mass produced and selected from the coupling manufacturers catalog.

This paper will help the selector and/or user of couplings to understand some of the applications for custom designed couplings. Applications discussed are steel mill drives, aircraft drives and the industrial gas turbine drives. Also discussed are some of the important design considerations used when designing gear and flexible membrane couplings.

Steel Mill Applications (Figure 1 & Figure 2)

One of the toughest coupling applications in the steel mill industry is for hot roughing mills. These mills are primary mills that supply hot rolled slabs, plates, or coils to many other finishing mills. If the mill does go down, productivity in other mills is greatly affected.

The spindles used on these drives are limited by the minimum roll size used and they must handle very high continuous loads. These spindle couplings must also handle high misalignment angles due to the constant change of material thickness and roll sizes. This job is also made difficult by the high misalignment angles (up to 6 degrees) for and during roll change that must be handled by these spindles.

If the motor torque was the major criteria to select and size these spindles, a gear spindle designer could easily design a spindle to survive. However, history has shown that these spindles must be sized to handle impact loads due to T.A.F. (Torque Amplification Factors). These T.A.F.'s are one of the culprits that break up spindles and shorten their lives. Torque amplification factors are multiplication factors on normal torque seen by the components of a train due to the system's inertias, backlash and loading cycle of the total system.

The key to the recent success with these applications is the simple

fact that the strongest and biggest spindles possible are used, rather than the smallest and cheapest.

The difficult applications use gear spindles made of high nickel alloy steels with high core hardness and carburized gear teeth. The use of these high strength alloys does not solve all the problems, however, it significantly increases the life of the spindle and has well justified the initial investment.

It must also be acknowledged that these spindles are getting more tender loving care at the mill from the maintenance crews due to the fact that they are paying closer attention to lubrication, maintenance, and repair cycles.


Cutaway of a gear spindle

Figure 1
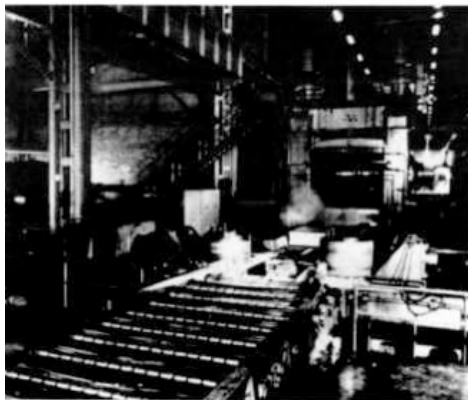
Application of Gear Spindle (4-Hi, 30"/36" and 55" X 96") Flexible gear spindles transmit 12,300,000 in-lbs torque continuously at misalignments up to 3°. Maximum OD is 27".
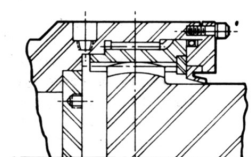
Figure 2

Gear spindles are very special and usually incorporate many special features, some of which are:

Splined Hubs - Figure 3 - The male hubs are usually connected to intermediate shafts through tight fitting splines usually with pilots. Normally the hubs are retained externally to eliminate the possibility of any broken bolts working into the gear teeth.

Insert Gear Rings - Figure 4 - This component is connected to a sleeve or adapter through a tight fitting spline which usually employs pilots. If the gear rings are damaged they can be replaced much more economically and in less time than it takes to replace adapters.
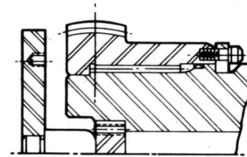

Splined Hubs

Figure 3

Insert Gear Rings

Figure 4

Spring Loaded Thrust Buttons - Figure 5 - Many spindles incorporate a spring loaded thrust button in the roll end coupling, which holds the roll end coupling in line with the roll shaft during roll change. The design also provides thrust surfaces that are located on the centerline of the gear mesh.

Splined Replaceable Sleeve - Figure 6 - This basically is an adapter which fits onto the roll or pinion shaft and intermediate sleeve, which is splined to the adapter and has an internal gear which meshes with the hub. This provides a quick disconnect feature.
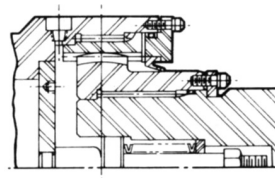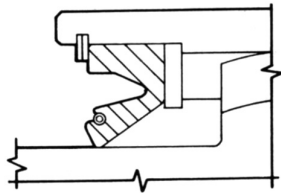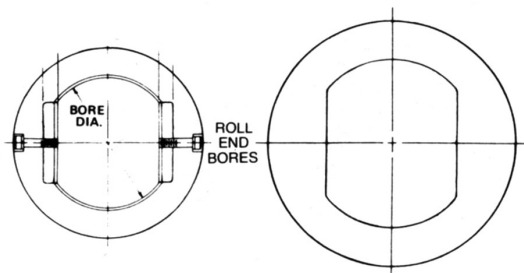

Spring Loaded Thrust Button

Figure 5
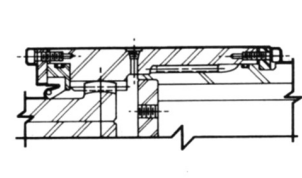
Splined Replaceable Sleeve

Figure 6

Lipseals - Figure 7 - This is the most common type used. The seals are usually a special integral one-piece molded design.

Lip Seals

Figure 7

Helicopter main drive between engine and transmission 305 HP @ 6,000 RPM, Continuous misalignment 2.5°, Intermittent misalignment 4.5°. Coupling made of nitralloy. Nitrided lapped teeth.
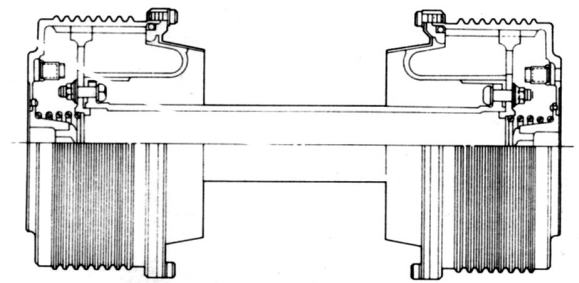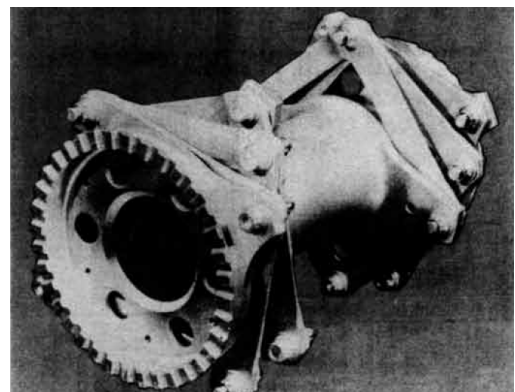
Figure 9

Roll End Bores - Figure 8 - These are usually furnished with developed or with replaceable wear keys. Keys and bores can be through hardened or surface hardened. The tougher applications tend to use surface hardened developed bores.

Helicopter drives have also used nonlubricated couplings such as flexible frame, contoured diaphragm, and multiple convoluted diaphragms.
(See Figures 10 A, B, C)

Replaceable Spindle Key   Developed Bore

Figure 8

Flexible Frame Coupling

Figure 10A

Aircraft Applications

Helicopter drive trains are mounted on very soft structures which twist and turn under loads. This movement requires a flexible coupling that can operate at high speeds and angles (18,000 RPM - 7-1/2° misalignment). Helicopter gear couplings for these applications are made from high strength nickel alloys with nitrided gear teeth to minimize distortion. These couplings are usually seal lubed and incorporate specially designed seals to retain lubricate (See Figure 9). A seal lubed or grease packed gear coupling is used in this type of application because special lubricants can be used in these couplings that can handle the high compressive loads experienced.

Contoured diaphragm coupling, 3900 In-lb @6,000 RPM, 2° misalignment with ± .100 inches axial travel.

Figure 10B

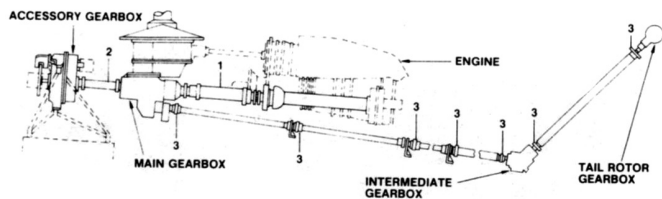Multiple convoluted diaphragm transmitting up to 530 HP @ 6,000 RPM.

Figure 10C

Helicopter Drive Train
Figure 11A

Applications for gear couplings on helicopters have encompassed main drives, auxilary drives and tail rotor drives. (See Figures 11A & 11B)

Turbine Helicopter Drives

Illustrated in Figure 11A is a simplified drawing of the drivetrain of a typical turbine-powered helicopter utilizing three types of high performance gear couplings. These types are listed and described briefly below: (Figure 11B)
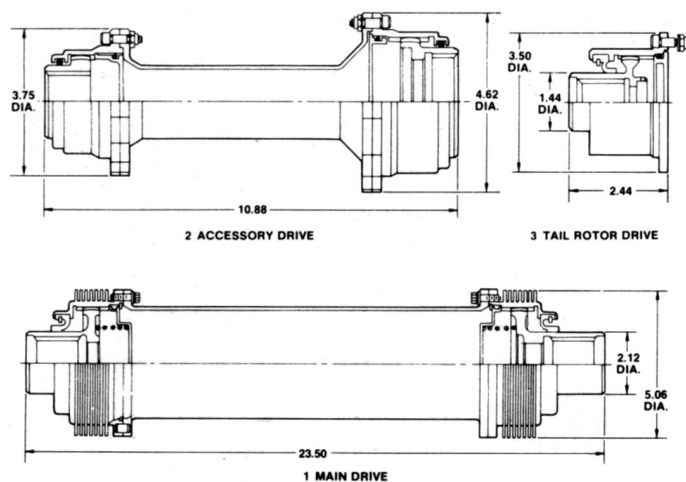
1. Main Drive - Connects engine to rotor gear box. Fins on sleeves aid heat dissipation and stiffening effect. 1,650 HP at 6,500 RPM with ± 4° angular misalignment per gear mesh.
2. Accessory Drive - Connects turbine engine driven main gear box to accessory gear box. 130 HP at 6,600 RPM with ± 2° angular misalignment per gear mesh.
3. Tail Rotor Drive - 100 HP at 2,800 RPM with ± 4° angular misalignment per gear mesh. A series of these single engagement couplings are used in the tail rotor drive system to compensate for air frame deflection.

The new technology that is now being applied to helicopters using composite materials and super alloys will require greater demands on couplings for these applications. Couplings for these applications will be required to be smaller and more compact, and they will probably use light weight super alloys and/or composite materials in their construction.

Turbine Helicopter Couplings
Figure 11B

Industrial Gas Turbine

Many Industrial gas turbines are derivatives of the Aircraft gas turbine or use many of the principles. These units operate at a high speed (3600 - 15000 RPM). They usually require couplings with large axial travel capabilities to over 1/4 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 12 featured at the bottom of this page is a typical heavy duty industrial gas turbine. The turbine shown employs two gear couplings, an accessory drive coupling, (Figure 13) and a load coupling (Figure 14).

Many applications for this type of gas turbine also use diaphragm couplings, Figure 15 shows an accessory diaphragm coupling and Figure 16 shows a diaphragm coupling used as the primary load coupling.
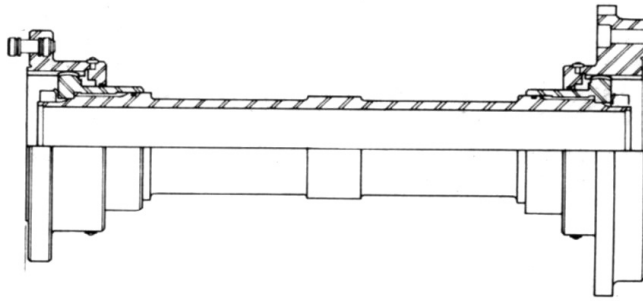
Accessory Drive Gear Coupling - Normal Torque 15,000 In-Lbs @ 6,000 RPM

Figure 13
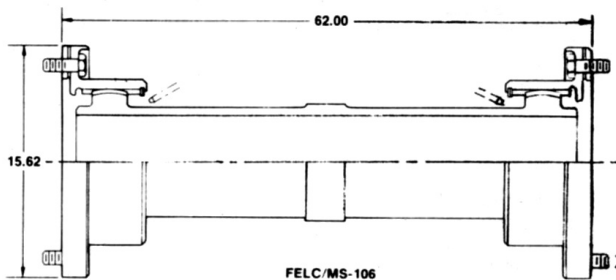
Continuous Lubed Generator Drive Gear Load Coupling, 534,000 In-Lbs @ 6,000 RPM

Figure 14

Multiple Convoluted Diaphragm Coupling for Accessory Drive of a Gas Turbine

Figure 15

Industrial Gas Turbine

Figure 12

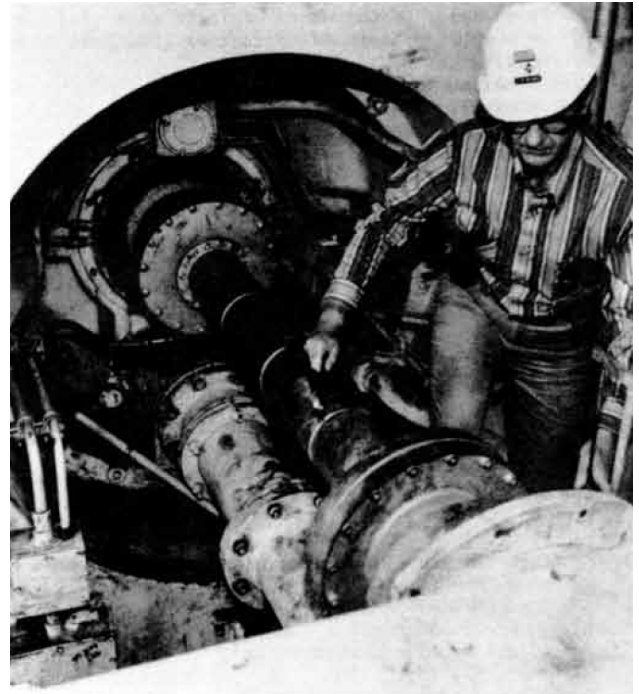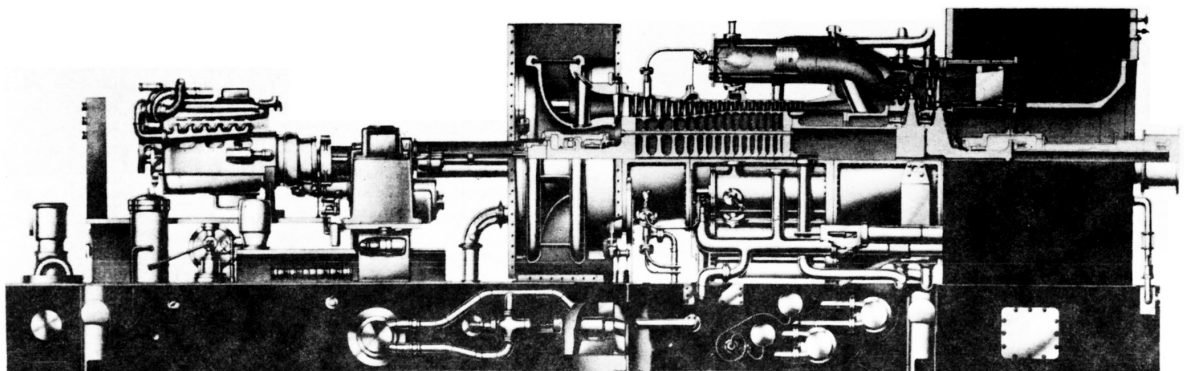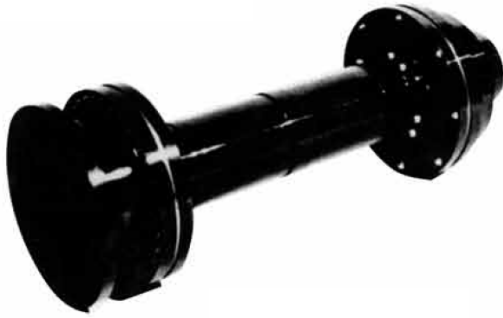Multiple Convoluted Diaphragm Coupling For Load Drive
of a Gas Turbine

Figure 16

Major Considerations When Selecting Custom Gear Spindles

The torque capacity of gearing for spindle type couplings can be expressed as a function of:

1. Material and type of heat treatment.
2. Operating misalignment.
3. Outside diameter of spindle (OD).

The following is an approximation and can be used for preliminary sizing and/or selection of a custom designed gear spindle.

Most gear spindles are sized by outside diameter restrictions. Therefore, spindle OD is usually set by minimum roll diameters. Knowing OD, find base torque (T_B) from capacity graph (Figure 17). Next, select the misalignment factor "M" (Figure 18). Misalignment angle is the maximum operating angle expected. Now calculate the material factor or "S":

$$S = \frac{T_O}{T_B \times M} \qquad (1)$$

T_O - Operating torque (In-Lbs) including service factor.

T_B - Base torque capacity as a function of OD.

M - Misalignment factor, based on empirical data and is developed from the fact that in a gear coupling fewer teeth are loaded as misalignment imposed on it is increased.

Once "S" has been calculated using Figure 19, select appropriate material for the application.

EXAMPLE:

Max OD of gear spindle = 10"
Operating torque including 1.25 SF = 315,000 In-Lbs
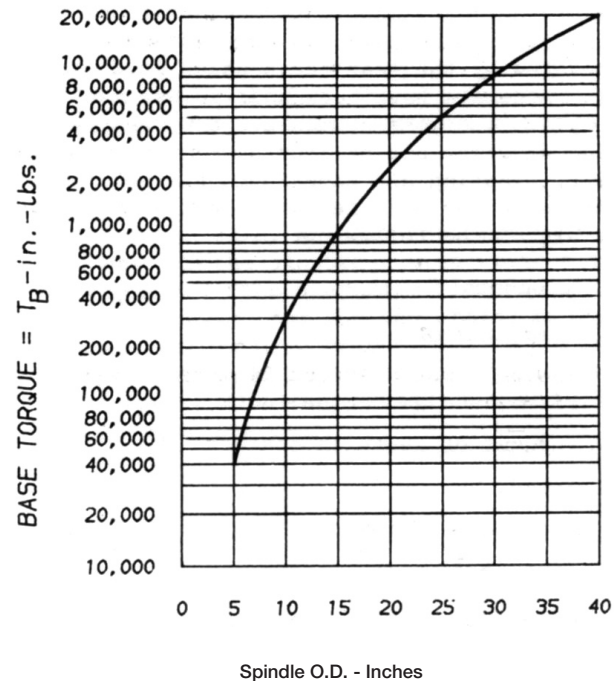Maximum operating angle = 2°
T_B for 10" Figure 17 = 300,000 In-Lbs
M for 2° misalignment Figure 18 = .7

$$S = \frac{315,000}{300,000 \times .7} = 1.5$$

From Figure 19 selection is a gear spindle made from A1SI-4140 steel with flank hardened gear teeth.

Spindle O.D. - Inches

*Figure 17

* Figure 17, 18 and 19 are empirical and based on twenty years of experience by Zurn Industries, Inc.

MISALIGNMENT FACTOR

MAX. OPERATING ANGLE	1°	2°	3°	4°	5°	6°
MISALIGNMENT FACTOR M	1	.7	.5	.4	.3	.2

Figure 18

MATERIAL FACTORS

SIZE LIMIT	MATERIAL* & HT	S = NORMAL FACTOR @ MAX. OPERATING ANGLES		
		‹4°	5°	6°
All	1045 Flank Hardened (Flame or Induction)	1	1	1
12"-28" 4° MAX.	1045 Full Contoured Induction Hardened	1.5		
All	4140 Flank Hardened (Flame or Induction)	1.5	1.5	1.5
12"-28"	4140 Full Contoured Induction Hardened	2.75	2.75	2.50
TO 13"	4140 & 4340 Nitrided	2.75	2.5	2.25
All	8620, 9310, 3310 Carburized	3.5	3.5	3.25

Figure 19

*Steel (AISI Designations)

Major Consideration When Selecting Custom Designed High Speed Gear Couplings

One of the most important considerations in the design of high speed gear couplings is the amount of heat generated by the gear teeth under load and misalignment. Attempts have been made to use many limits and criteria. The most common are compressive stress or sliding velocity but neither have proven to be sufficient. Testing and experience has shown that a design criteria for gear teeth of couplings should consider all of the following:

1. Load
2. Misalignment
3. Tooth Geometry- (Pitch Diameter, Diametral Pitch, and amount of tooth crown)
4. Percent of teeth in contact
5. Speed
6. Material, hardness of teeth, and finish
7. Type and quality of lube

Compressive Stress - Axial Sliding Velocity Criteria: (SV Criteria For Gear Couplings)

A criterion used to design gear couplings closely resembles the PV criteria used in bearing design. This criteria has been used successfully on thousands of couplings

(Zurn Industries, Inc. for twelve years). Let's call this criteria "SV". "S" for compressive stresses (PSI) and V for sliding velocity of the gear teeth when misaligned (IPS). Custom designed gear couplings have been designed and operated successfully with velocities as high as 120 inches/second (IPS) and compressive stresses up to 125,000 pounds/square inch (PSI) and with the combined SV's shown below:

T	=	Torque (In-Lbs)
PD	=	Pitch Diameter (In)
D	=	Diameter of Curvature of Tooth Face (In)
C	=	Number of Teeth In Contact as a Decimal % (Figure 20)
N	=	Maximum Speed (RPM)
∝	=	Misalignment (Degrees)
FW	=	Width of Tooth Face (In)
S	=	Hertz Compressive Stress (PSI)
V	=	Maximum Axial Sliding Velocity (IPS)

$$S = 3600\sqrt{\frac{T}{PD^2 \ X \ D \ X \ C}} \quad (PSI) \quad (2)$$

$$V = \pi \ X \ PD \ X \ Sin\propto \ X \ \frac{N}{60} \quad (IPS) \quad (3)$$

$$SV = S \ X \ V = PSI - IPS \ (Pounds/In^2 - Inches/Seconds) \quad (4)$$

Approximate C Values at Various Misalignment

Figure 20

∝	C
0°	.8
1°	.5
3°	.35
6°	.10

The allowables SV's change with material, tooth geometry, tooth finish, and type of lubrication and are based on test and field experience.

Some SV's for couplings that have operated successfully:

Material: Lubrication Method: Misalignment: SV (PSI - In/Sec):	AISI-1045 Grease Packed 3/4 Degree 1,500,000
Material: Lubrication Method: Misalignment: SV (PSI - In/Sec):	AISI-4140 W/Nitrided Teeth Continuously Lubed 1/4 Degree 750,000
Material: Lubrication Method: Misalignment: SV (PSI - In/Sec):	Nitralloy N W/Nitrided Teeth Grease Packed 4-1/2 Degrees 6,000,000
Material: Lubrication Method: Misalignment: SV (PSI - In/Sec):	3310 W/Carburized Teeth Grease Packed 3 Degrees 500,000

Major Considerations When Selecting A Custom Designed Flexible Membrane Coupling

In applying flexible membrane couplings to industrial applications the most important consideration when designing the flexible membrane coupling (disc, contoured diaphragm, flexible frame, and multiple convoluted diaphragm) is the operating stresses in the flexible membrane. These stresses must be designed to be under the endurance limit of the material used.

The type of stresses that must be considered are shown below. A multiple convoluted diaphragm is used as an example. Most flexible membrane couplings have similar stresses and the approach is very similar.

Diaphragm Stresses

To understand the diaphragm stresses it is important to understand how the diaphragm reacts to the various types of misalignment. Figures 21, 22 and 23 are exaggerated in order to more clearly demonstrate the diaphragm reaction to various forces. It is important to also 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 21), torque transmission, centrifugal effects and thermal gradient effects.
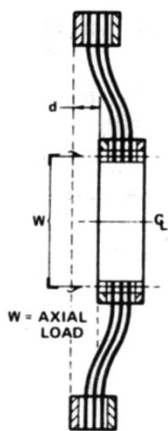
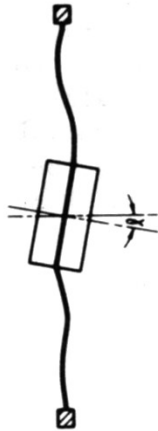Axial Stress (Figure 21)

(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

(τ) occurs when torque is transmitted through the diaphragm pack and is dependent upon the size, number and thickness of the diaphragms. Shear stress (τ) 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) 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) 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 22) and stresses due to offset deflection (Figure 23).

Flexure Stress (Figure 22)

(S_F) is the result of the angular misalignment of the coupling. ($\propto$) is misalignment angle in degrees.

Figure 21   Figure 22
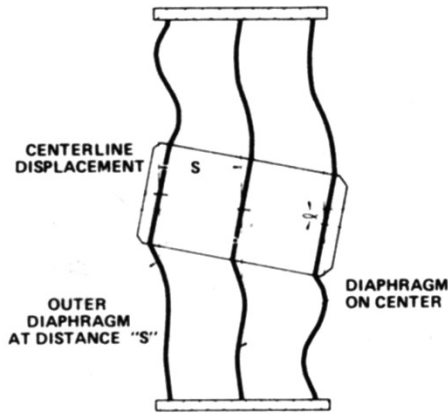
Figure 23

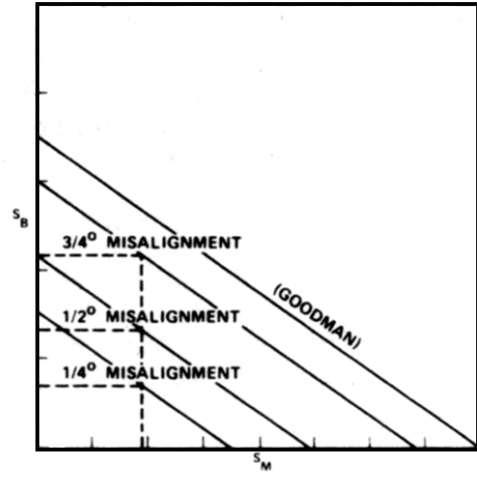Modified Goodman Line

Figure 24

Offset Stress (Figure 23)

    (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:

$$S = S_A + S_t + S_c \qquad (5)$$

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

$$S_M = \frac{S}{2} + \sqrt{\left(\frac{S}{2}\right)^2 + \mathcal{T}^2} \qquad (6)$$

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

$$S = S_O + S_F \qquad (7)$$

If cyclic torque is present:

$$S_B = \frac{S_O + S_F}{2} + \sqrt{\left(\frac{S_O + S_F}{2}\right)^2 + \mathcal{T}^2} \qquad (8)$$

    Finally, the mean stress and the alternating stress resulting from bending of the diaphragm can be plotted on a MODIFIED GOODMAN LINE for various misalignments and total coupling axial displacements. Using a typical Goodman Equation, the value of the design factor can be calculated. The results are shown in Figure 24.

$$\frac{1}{N} = \frac{S_M}{S_{ult}} + \frac{S_B}{S_{end}} \qquad (9)$$

N = Safety Factor

S_{ult} = Ultimate strength of material (PSI)

S_{end} = Endurance strength of material (PSI)

Summary

The applications and designs discussed are only a few of the thousands of applications and various types of custom designed couplings available. The trend today is that more and more couplings are being used and specifically designed for specific applications. There are many more design considerations that are used when designing custom gear and membrane couplings than discussed.

A designer of couplings must also consider:

1. All the available application information such as: load, speed, environment, misalignment, experience, etc.
2. Analysis of various components of a coupling, such as: bolt loading, shear stresses of components, bending and tensile loads on components, etc.

References

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