Euroflex Transmissions Ltd
www.euroflextransmissions.com
email: sales@euroflextransmissions.com
Tel: +44 (0)1494448880

ANSWERS TO FREQUENTLY ASKED QUESTIONS REGARDING FLEXIBLE COUPLINGS

INDEX TO QUESTIONS

01 - WHY SHOULD I HAVE A FLEXIBLE COUPLING?

In many engineering situations, there arises the need to couple together two items of rotating machinery to enable the transmission of power from one to the other (e.g. motor-pump or turbo-gen sets) and, under normal circumstances, each machine will have it's own rotor rigidly mounted in bearings (the obvious exception being single bearing electrical machinery). In an ideal world, it would be possible to simply join the two machines directly together and for them to operate with no problems. Unfortunately, as we are all too well aware, the world is far from ideal and, in reality, the simple solution is rarely practical.

In practice, the two machines can, firstly, only be positioned and aligned within finite limits. Secondly, during operation, machinery temperatures can vary, causing differential growths. Finally, vibration and time can lead to settling of foundations, a point especially critical when the machines are mounted on independent bases. [Fig.1.1].

The outcome of applying the "ideal" direct connection solution to the practical problem will, at best, result in premature failure of the machinery bearings due to extreme forces and, at worst, lead to the catastrophic failure of one or both of the machinery shafts.

The answer is to introduce a shaft between the two machines that will permit transmission of the required power and flex sufficiently to accept all the relative movements between the two machinery shafts with imposed or transmitted forces kept sufficiently low to avoid reducing the predicted life of the machines. Thus enters the flexible coupling, in one of its many varied forms.

The reason for a flexible coupling!

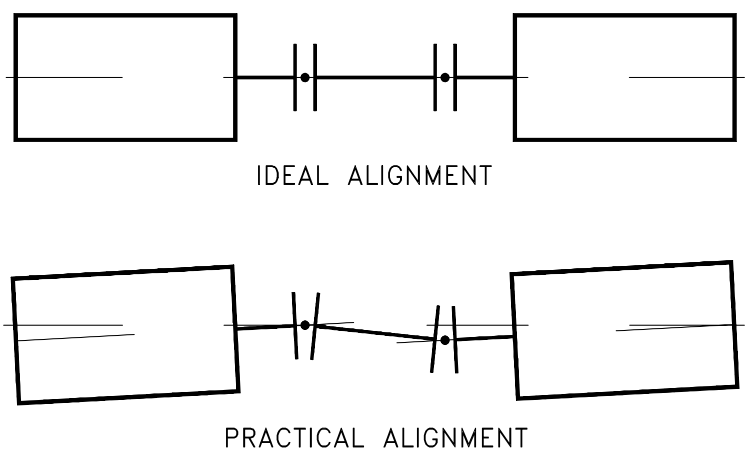

Fig.1.1

In finishing this brief introduction as to the reasons for flexible couplings, it would be remiss not to include reference to the special problems associated with reciprocating machinery where it is often considered necessary to include couplings with a level of torsional flexibility to avoid possible fatigue failure of, say, a diesel driven pump shaft. Whilst there is undoubtedly a role for the specialist torsionally flexible coupling in such applications, it should be understood that the proper application of a, so called, "torsionally stiff" coupling, with it's many added advantages, will often be more than adequate to fulfil the requirements, especially when correctly "tuned" and designed in conjunction with modern torsional analysis calculations.

02 - WHAT TYPES OF FLEXIBLE COUPLINGS ARE AVAILABLE?

It is intended that this document basically cover couplings that fall into the "Torsionally Stiff" category and, thus, the description of "Torsionally Soft" couplings will not be covered.

2.1 QUILL SHAFT COUPLINGS

Principally the oldest and, potentially, the most primitive of flexible couplings is the use of a shaft that is of sufficient length to bend in such a way as to permit misalignment between machines [Fig.2.1]. Nowadays, high-tech design and manufacturing methods have meant that such units have moved far beyond their primitive ancestors and now are very sophisticated.

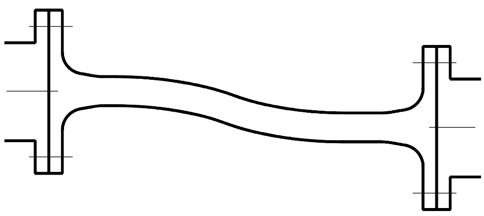

Fig.2.1

The principle of the design is to reduce the major length of the shaft to the minimum possible diameter commensurate with the required torque transmission, such that bending forces are kept to a minimum. As will be appreciated, stress levels must be kept extremely high to achieve this and, thus, misalignment capability is very limited.

Connection to machinery shafts is critical in this design, usually being achieved by a splined connection [Fig.2.2]. Because of the high stresses involved, the change in section between the shaft and the ends, where stress concentrations occur, is the most critical area, both in terms of the machined profile and surface finish, especially with regard to the fatigue life of the unit.

Typical Splined Quill Shaft Connection

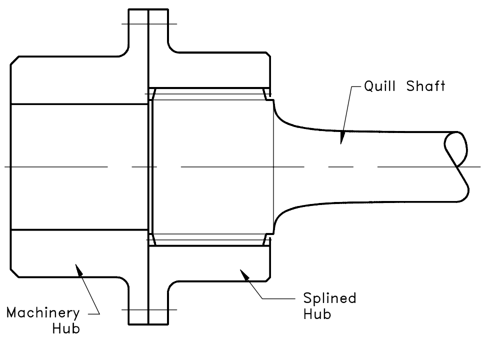

Fig.2.2

Whilst such units can be designed to accept misalignments in the form of bending, they are unable, unless associated with another form of coupling, to accept any axial movement between the two shafts.

Since the units are principally a single shaft, the level of balance possible, and it's maintenance, is excellent. However, the use of a thin section does impose limitations on shaft length at higher speeds, due to the reduced lateral critical speed.

Because these units are very specific in their usage, they will, generally, be given little more than a cursory consideration in much of the following discussions. This is not intended to denigrate their place in the flexible coupling environment but would simply require volumes in their own right.

ADVANTAGES

- Good balance characteristics.
- Very light weight.

DISADVANTAGES

- Complex design & manufacture procedures.
- Very high stresses, leading to the use of "higher grade" materials and a delicate unit.
- Very limited misalignment capability.
- No axial movement permitted without connection to another form of coupling.
- Limited length at high speed due to lateral critical speed.

2.2 HOOKE'S UNIVERSAL JOINTS

Undoubtedly the most widely recognised form of flexible coupling, by virtue of it's common use on cars and lorries, the Universal Joint has had a long history of low speed use in a vast range of applications.

TYPICAL UNIVERSAL JOINT

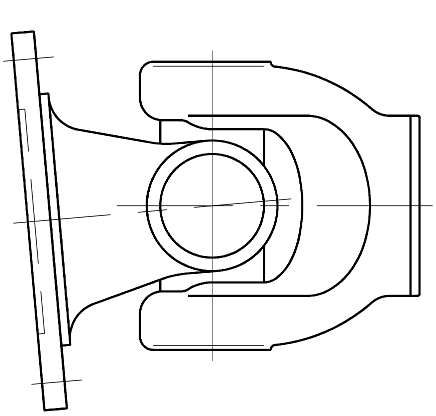

Fig.2.3

The most common form comprises two "Yoke" sections connected via two perpendicular rods which themselves form a "cross". These rods are free to rotate in bearings fitted in the yoke pieces [Fig.2.3]. This permits the two sections to move to give an angle between their relative axes.

In the conventional design of unit there is an efficiency loss and a non-constant velocity transfer when a single unit operates at an angle, the degree of these effects increasing with the angle. However, the use of two joints connected by an intermediate shaft returns the unit to a constant velocity coupling whilst also permitting a radial, as well as angular, misalignment between the two connected machinery shafts these being known as "Constant Velocity Joints". (It should be noted that there are variations on the "conventional" design which will give constant velocity in a single unit but these will not be discussed at this point).

The major advantage of the Universal Joint type of coupling rests with the extremely large angular misalignment that the unit can accept which, when using a double unit, also means very high radial misalignment. However, speeds are, for various reasons, limited to relatively low values and, as for the Quill Shaft units, any axial deflection between shafts (including the apparent length variation from large radial misalignments in a double unit) requires the incorporation of another form of coupling. Conventionally, this axial movement in a Universal Joint is accommodated in a sliding spline positioned in the intermediate shaft [Fig.2.4] which not only affects the

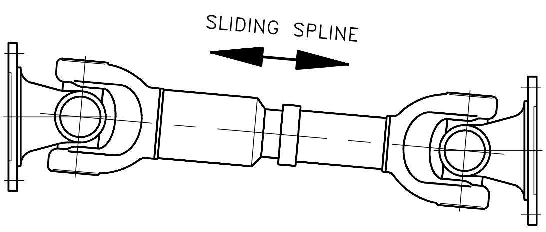

Fig.2.4

operating speed but also introduces a wear and maintenance factor.

Because of the limits on speed, the need for regular maintenance and the high axial loads imposed on shaft bearings (due to the forces required to move the spline during operation), the use of conventional Universal Joints tends to be limited to the more “robust” applications or those where the need for extreme misalignment cannot be avoided.

ADVANTAGES

- Extremely high radial/angular misalignments can be accommodated.
- Smaller units can prove cost effective.

DISADVANTAGES

- Units have a finite life.
- The units normally require regular maintenance.
- Axial movement commonly depends on the use of a splined “sliding” joint which imposes large axial forces when movement is required during operation and also will suffer from wear.
- The units are, generally, limited in speed when compared with other types of coupling.
- Because of the need for lubrication, there are temperature and environmental limitations on effective use.
- In the event of failure, no emergency drive is available.

2.3 GEAR COUPLINGS

The oldest general form of power transmission coupling stems from the use of gears. Two straight, intermeshing, gears can be moved axially relative to one another whilst in operation. In addition, if clearances between the teeth are sufficient, the two gears can be offset angularly. Hence is born the idea of the Gear Coupling.

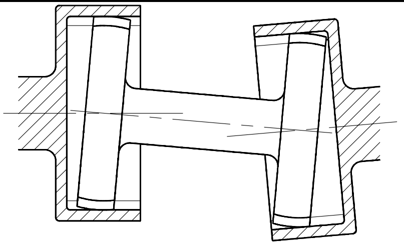

Fig.2.5

To perform the above movements on straight gears requires relatively large forces and creates rapid wear. However, if the teeth are appropriately profiled and careful consideration is given to lubrication, an effective flexible coupling can be produced [Fig.2.5].

For low speed applications, developments of both the tooth profile and lubrication are nowadays such that grease packed couplings can give a reasonable degree of misalignment with a, usually, acceptable level of transmitting force and a reasonable operating life [Fig.2.6].

TYPICAL LOW SPEED GEAR COUPLING

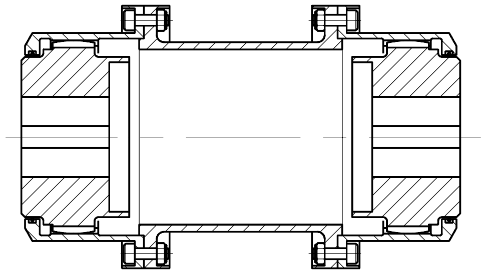

Fig.2.6

In the case of medium and high speed units, tooth profile requires to be far more complex, often involving the use of crowned and barreled teeth. In addition, to give any reasonable life span with regard to wear, continuous lubrication is required and radial/angular misalignments are limited.

With the ever increasing operating speed of modern compressor drives, the lubrication system required is itself becoming very complex, involving extremes in filtration to avoid premature wear of teeth. In addition, maintaining effective lubrication under extreme centrifugal forces is drawing a limit to the operating speed of Gear Couplings [Fig.2.7].

HIGH SPEED - LUBRICATED

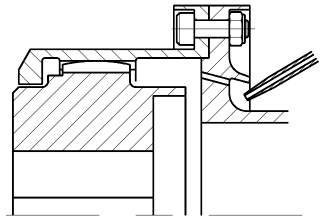

Fig.2.7

Whilst low speed units have a great advantage in offering an almost unlimited axial travel, the design of high speed teeth tends to limit this more than one might expect. Also, since the contact load on the intermeshing teeth tends to be high, especially on the higher speed units, the loads required to achieve relative movement are higher than one may initially assume and are also uneven due to what may be called the "sticktion" effect.

Finally, however sophisticated the coupling design, the principal of operation involves the sliding of teeth. This in turn must lead to some degree of wear. Since the concentricity of the components must depend upon the fit between teeth, it is easy to appreciate that there is a great potential for increased out-of-balance of the units as time progresses.

ADVANTAGES

- Outer diameter of the units can be kept relatively small, hence leading to light weight.
- Large axial movements of shafts can be accommodated, especially in low speed applications.
- For the lower speed applications, cost can prove beneficial.

DISADVANTAGES

- Units have a finite life, with high replacement costs.
- Maintenance at regular intervals is required, especially for grease packed units.
- Angular/radial misalignment is limited to prevent excessive wear.
- In high speed units, expensive lubrication and filtration systems are required.
- In the event of failure, it is very difficult to maintain an emergency drive.
- Due to the restrictions of the lubricants, there are temperature and environmental restrictions on effective use.
- A certain level of misalignment must be maintained to prevent "micro-fretting" which may lead to "tooth-lock".
- Balance of the units may be affected after a period of time.
- Whilst the level of shaft loading resulting from misalignment may, initially, be acceptable, these may increase in time as wear effects cause an increase in the coefficient of friction.
- The transmitted loads are not smooth due to the slip/stick ("sticktion") effect.

2.4 DIAPHRAGM COUPLINGS

These, in their most primitive form, represent the oldest type of Dry, Torsionally Stiff, flexible couplings. The principal of their design is to transmit torque by shear in a plate fixed on one side to the outer periphery and the other at some inner circumference. Flexibility is obtained by distortion of this plate or "Diaphragm" between it's two anchor positions [Fig.2.8].

DIAPHRAGM COUPLING UNDER ANGULAR AND RADIAL MISALIGNMENT.

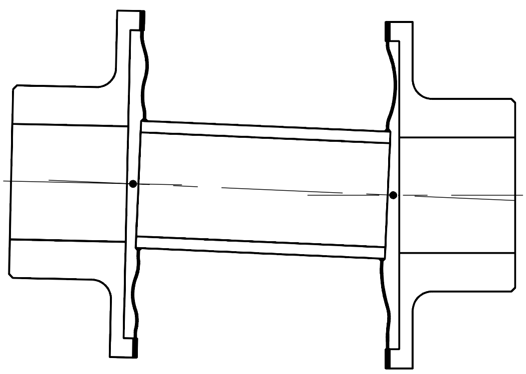

Fig.2.8

The fact that these units depend on the use of shear to transmit torque, makes them somewhat complex to design since, as every engineer knows, shear values are far harder to define for material than tensile.

Since flexibility of the diaphragm coupling depends on bending of the plate over the section between inner and outer anchor regions, the difference between these two must be kept as large as possible. The larger this difference, the greater the flexibility and the lower the resulting transmitted forces. However, the inner anchor region tends to be on a diameter that is pre-defined by dimensional constraints and the physical requirement to transmit the torque in shear. Hence the outer diameter of the coupling is itself, defined by the need for a "flexing span" giving acceptable deflections for reasonable forces. The outcome being a large outer diameter of the coupling.

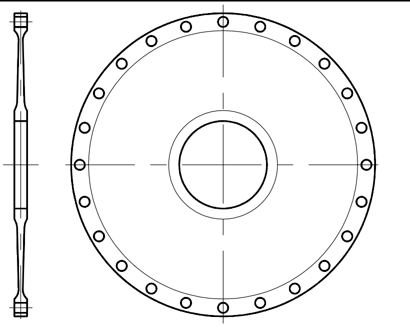

Fig.2.9

In an attempt to keep forces to a minimum without resorting to very large diameters, solid diaphragms have been developed to have complex profile whereby the plate thickness reduces non-linearly, as the diameter increases [Fig.2.9]. In addition, the designers are forced to take stress levels within the diaphragm to very high values. The result of these factors is an expensive to produce unit of not very robust nature, often requiring the use of special, costly, materials.

Alternative to the solid diaphragm is the laminated diaphragm. In this type, the "diaphragm" is made up of a series of thin plates, often with some material removed to give the effect of spokes. This results in reduced forces and cheaper production but often at the cost of size and weight [Fig.2.10].

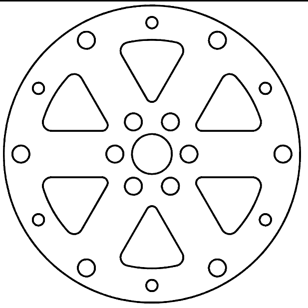

Fig.2.10

After the standardisation by many machinery manufacturers on hydraulic expanded hubs, coupling manufacturers had the problem of accommodating this expansion in their diaphragms, without the need to introduce additional "adaptor" plates. As a result, the "Wavy" diaphragm was evolved. In this design the plate, seen in section, takes on a corrugated form which permits such expansion by acting like a radial "bellows" [Fig.2.11].

"WAVY" DIAPHRAGM

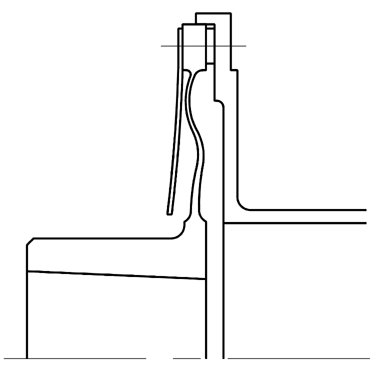

Fig.2.11

As an additional advantage of this "Wavy" diaphragm, the unit can accommodate extended axial deflections with low transmitted forces. However, it will be appreciated, the cost of manufacturing such items, which must still be highly stressed, is high. Also, by virtue of their profile, transverse stiffness of the coupling is reduced and, in some instances, this seems to have led to problems in maintaining dynamic balance during operation.

ADVANTAGES

- Thinner component flanges can result in light weight units.
- In the case of "Wavy" diaphragms, relatively large axial deflections can be accommodated.
- No lubrication is required and, hence, operating costs are lower than gear type units and are not environmentally sensitive.
- Diaphragm Couplings are fundamentally, maintenance free and, if operated within their design limits, substantially of infinite life.

DISADVANTAGES

- To enable useful misalignments with standard diaphragms, large diameters are required.
- In many cases, stress levels are taken to extremes, resulting in "delicate" units.
- In the case of standard diaphragms, flexibility is limited.
- Overload, emergency, drive requires the use of a separate drive system within the coupling.
- In most cases, the diaphragm design forces expensive manufacture and, thus, relatively high selling costs.
- "Wavy" diaphragm units, specifically of the laminated type, require great care in balancing.

2.5 DISC COUPLINGS

The last major form of torsionally stiff coupling is of a design whereby torque is transmitted through a series of "links" joining two components with attachments on a common pitch circle diameter. In it's most basic form, the "links" would be chords of two pairs of holes equispaced in a circular ring, or "disc", with two diametrically opposed holes being bolted to one flange and the other two to the second flange. Torque is then transmitted in tension through two spans. Flexibility is obtained by the deformation of these spans in a manner akin to the bending of beams [Fig.2.12].

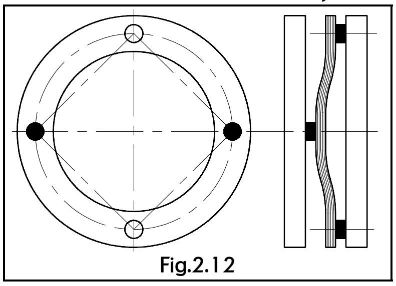
Fig.2.12

The above, simplified, form of Disc Coupling has many problems, not least of which is the fact that, being circular, the line of tension for torque transmission falls outside the material of the "disc" and, hence, leads to additional stresses due to the "straightening" effect. To overcome this, the next stage of development is to use a square profiled "disc" such that the lines of tensile stress act directly between adjacent bolts [Fig.2.13].

This process can now be developed further by increasing the number of bolts to any even number greater than four. Commonly, six or eight bolts are used, giving three or four lines of tension. As can be easily appreciated, the greater the number of bolts (and, hence, spans), the greater the torque transmittable within a given overall size of unit [Fig.2.14].

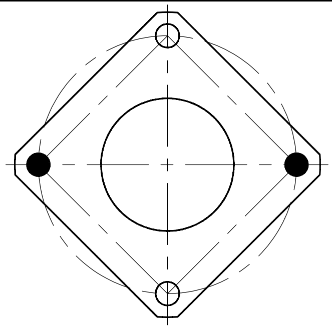
Fig.2.13

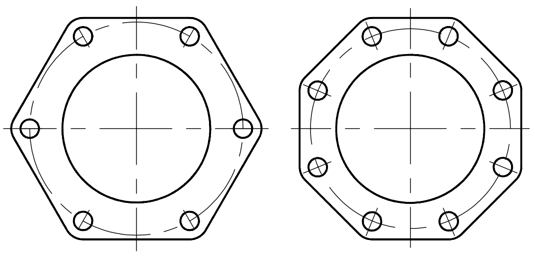
Fig.2.14

However, for a given pitch circle diameter, as the number of bolts increases, the length of span between them decreases and so, consequently, does the flexibility of the unit. Thus, in any design of Disc Coupling, a compromise is required between the size of the unit (i.e. the number of bolts) and the flexibility.

While some manufacturers use "solid" discs, it is far more common for the flexible elements to comprise of a number of blades or laminations. This laminating has the effect of dramatically reducing the forces induced by flexing, or misaligning, the unit.

Whilst the actual profile and proportions of the "discs" varies between manufacturers, as does the bolting system, the overall principle of operation remains the same. However, certain of the variations in design principles are discussed in later sections.

ADVANTAGES

- Angular and radial misalignment capability tends to be greater than for the three previously discussed units.
- The units are far more robust and able to withstand greater levels of misuse than Diaphragm Couplings.
- No lubrication is required and operating costs are therefore lower than gear type units.
- Disc Couplings are, fundamentally, maintenance free and, if operated within their design limits, substantially of infinite life.
- Overload, emergency, drive in the event of disc failure can, unlike other types of coupling, be inherent in the basic design.
- Manufacturing methods are relatively simple when compared with diaphragm couplings and, hence, the price is, generally, lower.
- Modern design methods mean that, under most circumstances, there need be no weight penalty when using a Disc Coupling in preference to any other form of unit.
- These couplings are not particularly sensitive to environmental conditions and can, with some modification, be used in extremes of temperature and atmosphere.
- Due to the simplicity of design Disc Couplings are easily "tailor-made" to suit particular applications.

DISADVANTAGES

- The capability for axial deflection, whilst being proportionally greater than for standard diaphragm units and generally more than adequate for the majority of applications, is not as generous as for Gear or "Wavy" Diaphragm Couplings.

03 – WHAT CREATES LIMITS TO FLEXIBLE COUPLING MISALIGNMENTS?

Before attempting to answer this question, it is important to obtain some clarification as to the types of misalignment faced and, indeed, a terminology used for these [Fig.3.1].

AXIAL misalignment, probably the easiest to understand, is simply the variation in the axial distance between the shafts of a driving and driven machine.

ANGULAR misalignment is the effective angle between the two shafts and is usually quantified by measuring the angle between the shaft centre lines if they were extended to intersect. If the shafts are flanged, it is simply the enclosed angle between them if they were bought to a position of contact.

RADIAL (or PARALLEL) misalignment is the transverse distance between the two shafts and is quantified by measuring the distance between the centre line of one shaft if it were extended to overlap the other.

In the case of all the above, the misalignment may simply be a variation between the actual position and that intended during design stage (due to the practical restraints in positioning equipment) or due to shaft movement/growth during operation (e.g. thermal growth of a Turbine shaft & casing). In reality, it will be a combination of these and other factors.

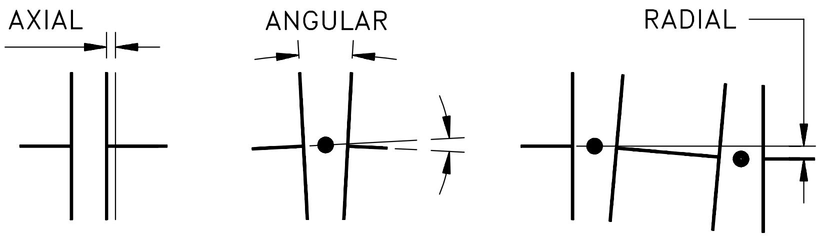

Fig.3.1

Practical applications will, normally, involve all three of the misalignment types.

If one is considering a Spacer or Double-Engagement type coupling, a Radial misalignment manifests itself, as far as the flexing joints are concerned, as an angle at each joint. This is additional to any Angular misalignment present and, depending on the circumstances, the resulting, combined, angle at each joint may be different [Fig.3.2].

If one is considering a Non-spacer or Single-Engagement unit, only Axial & Angular misalignment is accepted in both the Gear & Dry type couplings being considered here.

Having described the types of misalignment, we can now look at their effect on couplings or, more correctly, the ability of various coupling types to cater for these whilst transmitting torque. The limitations of various coupling types arise for different reasons and, as such we will consider the limits under various sections.

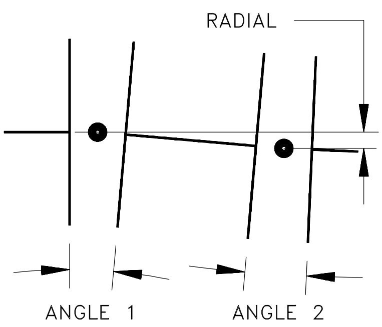

Fig.3.2

3.1 QUILL SHAFT COUPLINGS

As described in "Question 2", the ability of a Quill shaft coupling to accept Angular & Radial misalignment depends on it's ability to physically deform as a shaft [Fig.3.3]. This, firstly, indicates that an effective Quill shaft unit requires a reasonable length to give any useful degree of misalignment and, secondly, shows that the level of allowable misalignment depends on the stresses within the shaft itself. When the unit is rotating, Angular/Radial misalignment represents itself in a cyclic manner and, hence, the stresses in the Quill shaft will need to be considered as Fatigue stresses which obviously have offer a considerable limitation to the misalignment of the coupling.

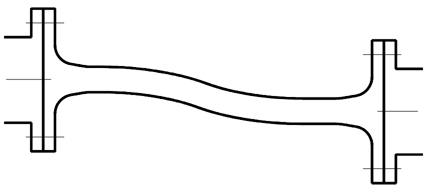

Fig.3.3

Also, as mentioned earlier, a Quill shaft coupling cannot accept any degree of Axial misalignment unless it is connected to another form of flexing joint.

3.2 HOOKE'S UNIVERSAL JOINT COUPLING

As we saw in "Question 2", this type of unit will accept extremely large angular misalignments and, when combined as a pair to form a Spacer type unit, give correspondingly large Radial misalignments. With this type of unit, there is always an effect of wear within the coupling, but this does not vary dramatically with the level of misalignment.

Axial misalignment depends entirely on a sliding spline system within the unit shaft (only applicable in a Double flexing coupling) [Fig.3.4]. Whilst this sliding system can be designed to give large movements it is designed to avoid any bending of the shaft and, as such, is usually of the simple straight sided form. The sliding spline is intended, principally, to allow for axial length variation during set-up and, in reality, it will not move when the unit is operating under torque without the use of extremely high axial forces to overcome joint friction. If it is forced to move during operation, an amount of spline wear will occur and, as a result, the fit of the splines will deteriorate and the unit will no longer be stable.

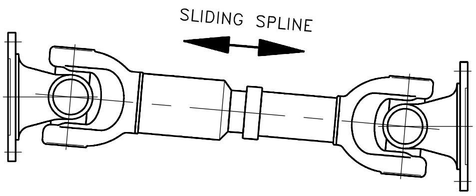

Fig.3.4

3.3 GEAR COUPLINGS

The only real limitation to the Axial movement of a Gear coupling is that of the physical limitations of the meshing gears and this offers one of their few technical advantages in modern engineering.

Under Angular or Radial misalignment, the flexing joint sees a cyclic movement varying from, say, extreme opening to extreme closing and back to extreme opening. In the case of Gear type couplings, this involves the relative motion of the meshing gears. When the unit is transmitting torque, the meshing teeth are under contact pressure and, to obtain the above movement, the teeth must overcome the transverse force resulting from the pressure. The magnitude of this force will depend upon the effectiveness of the lubrication, the surface finish of the teeth and the resultant coefficient of friction between the sliding parts. However good the lubrication may be or how good the surface finish, wear may ultimately result with the inherent deterioration of the couplings ability to perform it's required function.

Consequently, in an attempt to limit the "rubbing" speed of the teeth, and hence the wear, a strict limitation is imposed on the degree of Angular & Radial misalignment that a Gear coupling is permitted to accept. This limit is dependent on the operating speed of the unit and the ability to lubricate the teeth but, in all but the slowest speed applications, the level of permitted misalignment is considerably less than the design may at first suggest and, in the majority of cases, less than could normally be accepted by other coupling types (with the exception of the Quill shaft unit).

In addition to the limitation on misalignment, Gear couplings have the "peculiarity" of requiring a slight Angular offset in the engagement to enable Axial movement. This angle enables the intermeshing gears to "walk" during rotation and hence accommodate Axial movement without producing massive end forces. This slight angular offset is also required to prevent fretting of the teeth surfaces under extended operation.

3.4 DRY TYPE COUPLINGS (DISC & DIAPHRAGM)

In the case of both Disc and Diaphragm couplings, the degree of misalignment in the Axial, Angular or Radial sense is limited by the imposed stresses in the flexing material.

Axial misalignment of a dry coupling imposes tensile bending stress in the flexing component. In the case of a Diaphragm unit this is the bending of the outer to inner attachment circles [Fig.3.5]. In the case of a Disc coupling, this is the bending of a "beam" between anchor points [Fig.3.6]. In normal cases, such as thermal growth of machinery shafts, this bending stress, once applied, is steady and can thus be "added" to the steady stresses due to torque and speed (these being both tensile in Disc couplings and shear & tensile in Diaphragm couplings).

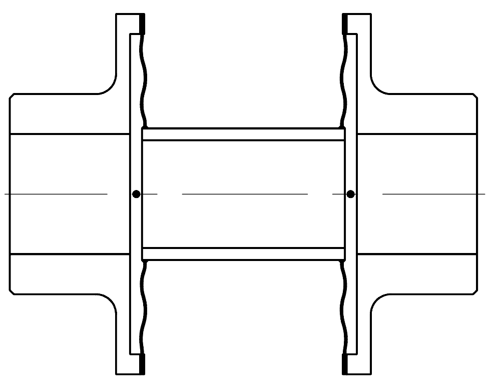

Fig.3.5

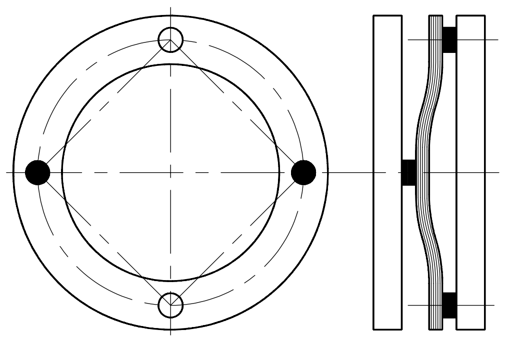

Fig.3.6

Hence, by itself, the limitation on Axial misalignment of a dry type coupling is defined by the bending stress within the flexing element (in combination with shear stresses in the case of Diaphragm couplings). This in turn is dependant on the length of the bending span.

Angular or Radial misalignment again introduces a bending in the "span" of the flexing element but, as for the Gear coupling, it is cyclic in form. In dry couplings however, unlike gear tooth wear, the factor of greatest concern is the bending fatigue in the material. Hence, the limitation to Angular and Radial misalignment is within the level of the allowable cyclic, fatigue, stress introduced. The level of stress induced by deformation, and hence the permitted level of misalignment, is different for Diaphragm and Disc type couplings (and indeed between designs of these units) but, in all cases, is a function of the geometry of the flexing element.

The one exception to the above is the “wavy” diaphragm where the effects of the corrugation permit large movements with the addition of extremely low bending stresses and hence permit higher levels of misalignment than the comparative “standard” diaphragm.

As stated earlier, the Axial misalignment stress is, generally, steady and could be considered as additive to the torque and speed stresses. However, in reality, where all forms of misalignment are encountered, and where the torque and speed of a unit are pre-defined, it is most practical to consider the stresses due to these as fixed and allow a relationship to exist between the bending stresses due to Axial and Angular/Radial misalignment to exist. Due to the inter-dependence on forms of deformation, it is common to offer an Axial misalignment that increases as the level of Angular/Radial diminishes (generally in graphical form), thus permitting the best selection of a unit to suit a particular application [Fig.3.7].

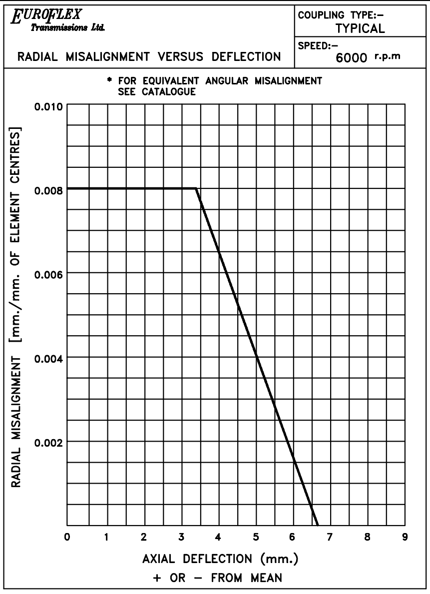

Fig.3.7

As a note of caution, it should be remembered that, when a coupling catalogue publishes, for example, a maximum permitted Axial and Angular misalignment (as most do), these are probably not inter-related and the maximum Axial will be at zero angle and the maximum Angular at small axial. It is advisable, if the supplier has not furnished the details of inter-relation of the misalignments, to request confirmation of the full misalignment capability.

3.5 GENERAL

Having ascertained the maximum recommended working limits for the coupling, the supplier will, usually offer his recommendation for initial alignment limits based on the expected movements during operation. Whether the user cares to ignore these in favour of his own, more stringent, alignment values or not, it is important to remember that a flexible coupling is utilised to cater for misalignments, both initial and operating. With the aid of modern technology for machinery alignment (e.g. Laser equipment) it is fairly simple to give a more than adequate alignment at installation but, unfortunately, far to many users insist on wasting time and money on trying to achieve perfect alignment. This, in most cases, is not necessary and more reliance on flexible couplings to perform their intended function, could save money and inconvenience.

04 – SHOULD I USE SINGLE OR DOUBLE FLEXING?

Whilst certain torsionally flexible couplings can permit a parallel or radial offset between shafts in a single flexing member, torsionally rigid couplings (with the exception of the Quill shaft type which has it's own rules) tend only to allow Angular and Axial deformation at a single flexing joint (again the Hooke's Universal joint being an exception in only permitting Angular misalignment). For a torsionally rigid coupling to accept Radial misalignment between machinery shafts, two flexing joints are required.

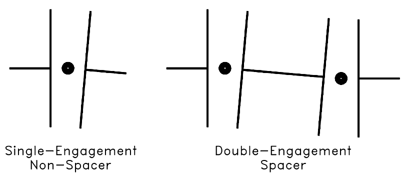

Fig.4.1

For a Gear coupling, such units are described as Single- or Double-Engagement respectively. In the case of Disc or Diaphragm units, the terminology is generally Non-Spacer or Spacer type [Fig.4.1].

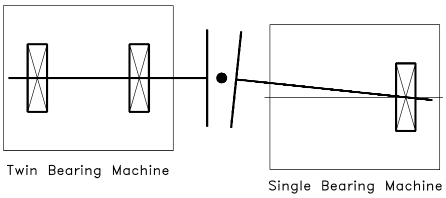

Fig.4.2

Single-Engagement or Non-Spacer couplings have a limited amount of applications due to the lack of Radial misalignment capability. Generally their use is restricted to applications where the driving or driven machine shaft or rotor is located in a single bearing such that only Angular deformation at the joint is required [Fig4.2].

However carefully machinery is aligned, accuracy can only be within finite limits and machines will, undoubtedly, move during operation. Hence, fitting Single-Engagement or Non-Spacer couplings between two machines whose shafts are mounted in pairs of bearings will lead to severe transverse forces being transmitted between the shafts with the likely event of early bearing failure.

Such machinery trains should be fitted with Double-Engagement or Spacer type couplings which, since they permit Angular deflection at each joint, will give acceptance to Radial offset between two machinery shafts. Where space is critical, and one may initially consider a Single-Engagement/Non-Spacer unit is the only solution, it is normally preferable to consider the use of a "Close-Coupled" unit which has the benefit of two flexing joints whilst keeping a small distance between machinery shaft ends. Whilst this may involve additional initial cost, the savings on down time due to bearing/shaft problems will usually far outweigh this [Fig.4.3].

TYP. "CLOSE-COUPLED" DISC COUPLING

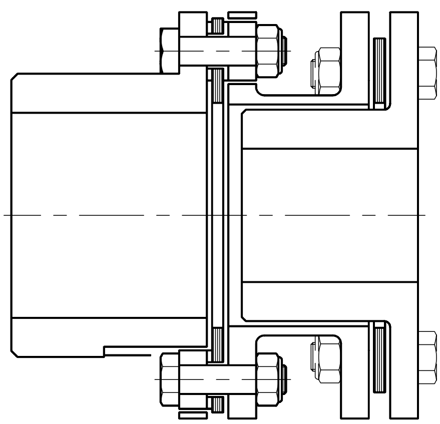
Fig.4.3

Because these types of couplings are designed to operate on the principle of Angular deflection at the flexing points and since the level of such deflection is defined by design, the amount of Radial offset is directly proportional to the distance between these flexing points and thus the length of the unit [Fig.4.4].

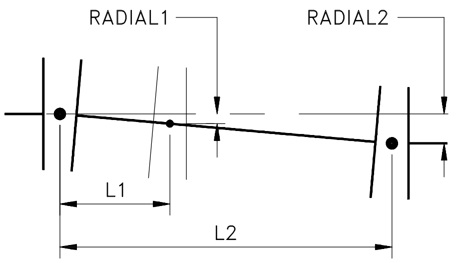

Fig.4.4

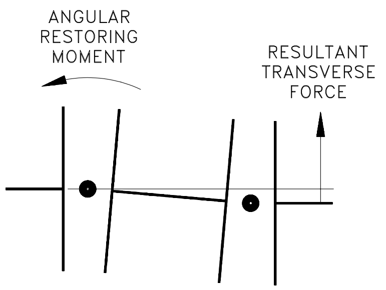

Fig.4.5

It should be noted that, whenever any of the dry type couplings are deformed angularly, a restoring force is imposed back onto the shafts although these can, generally, be neglected since they are usually of extremely low magnitude [Fig.4.5].

05 - WHAT IS DYNAMIC BALANCING?

Excessive vibration in rotating machinery can cause unacceptable levels of noise and, more importantly, substantially reduce the life of shaft bearings. Hence, the ideal would be to remove all causes of vibration and run the unit totally “smooth”. Unfortunately, in practice, the ideal cannot be achieved and, whatever one does, some inherent cause of vibration, or unbalance, will remain. The best one can do is to reduce this unbalance to a level that will not adversely effect the bearing life and will reduce noise levels to an acceptable level.

The process of reducing the out-of-balance forces that cause vibration in rotating machinery is called “Balancing”. The unbalance is caused by an effective displacement of the mass centre line from the true axis caused by some mass eccentricity in the unit [Fig.5.1]. The process of “Balancing” is the removal or addition of weight to the unit such that this effective mass centre line approaches the true axis. A multitude of books and papers has been written about this complex subject and, as such, the following is intended to be no more than a brief outline with specific reference to the balancing of flexible couplings. Where balancing grades or levels are referred to here, and in subsequent sections, they are referenced to ISO 1940.

Mass eccentricity / Unbalance

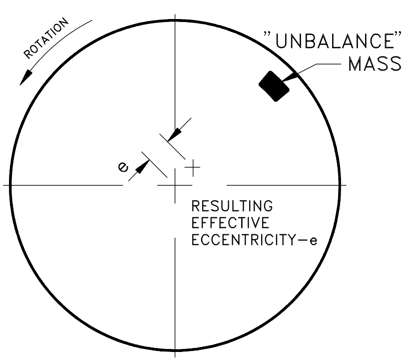

Fig.5.1

The simplest form of, “Static”, balance involves placing the unit on low friction bearings and allowing it to rotate and “settle” with the “heaviest” point falling to the bottom. Material is then removed from this point (or added at the top point) and the unit gently rotated until, when stopping, the new “heavy” point again falls to the bottom. This process is then repeated until no obvious “heavy” point seems to exist.

Static / “Single-Plane” Balance

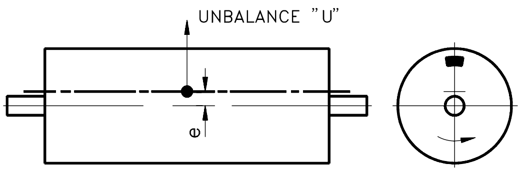

Fig.5.2

Advancing this one stage further, the unit can be mounted on a purpose built “Balancing Machine” which has it’s bearings connected to sensors (displacement or acceleration type depending on the design of the machine) which detect the “heavy” point, in relation to a datum on the unit, whilst it is being rotated. This increases the sensitivity and, hence, the accuracy of the balance. If one considers correction at a single position along the length of the unit, the balance is said to be “Single-Plane” [Fig.5.2].

If the unit being balanced is very short in relation to it’s diameter Single-Plane Balance will, normally, be very acceptable. However, if the unit has any appreciable length, correction at a single plane, say in the centre of the unit, will probably give a dangerously false correction. If you consider a shaft with two flanged ends, it is quite likely that the major unbalance will arise in the two flanges, probably caused by the inherent concentricity machining errors. The two “heavy” points may fall in precisely the same angular position but, more likely, they will not and, thus, the displacement of the mass centre lines from the true axis will be a different orientation in both ends, as will the size of this displacement. If the unit is now being balanced by the Static or Single-Plane method, as it can easily be, the overall effect would appear to be correct on the Balancing Machine but, in operation, since no account has been taken of the variation between the two ends, a “Force Couple” will exist which will again introduce vibration in the two machinery shafts [Fig.5.3].

To overcome this problem, a “Multi-Plane” or “Dynamic Balance” must be performed. In this case the balancing machine will have both sets of bearing pedestals connected to sensors and the unbalance at the two planes can be independently identified and, thus, corrected. By the use of electronics and computer control, the actual correction planes can be at any convenient position along the length of the unit relative to the running bearings and, in the case of very long units, more than two planes can be considered, although this is rarely an advantage unless the unit is to run at speeds above it’s lateral critical speed (“Super-Critical”).

“Dynamic Balance”

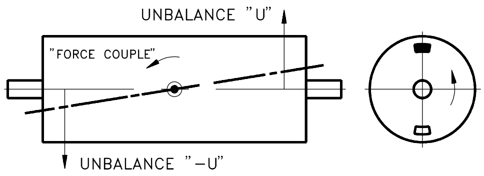

Fig.5.3

For units which will operate well below their lateral critical or “whirl” speed and which are considered “stable” with regard to the speed of operation (a category into which all but the longest flexible couplings fall), there is no necessity for the balance operation to be performed at the actual running speed. Because Balance Grade, whether it be specified as Q0.6 or 4W/N for example, considers operational speed when the calculation of the actual permitted physical amount of out-of-balance is made, balance at a speed much below the operational speed will result in the same final level of balance.

If the unit is to run “Super-Critical” then the actual characteristics of the unit, after passing through the critical speed, may well be different to those below it, Hence, such units should be balanced at running speed.

A final complication arises if a unit is to operate at a speed close to it’s critical lateral frequency. In this case there may be some finite movement of, say, the shaft section of a coupling and, hence, it becomes important to balance such a unit at the final running speed. In addition, if this unit is to be subject to multi-speed operation, such as might occur in a two-speed motor driving a fan, then balance must be performed at both speeds with a compromise between the two unbalance forces being made.

It should be appreciated that however close to the ideal a coupling is balanced on the balancing machine, this will change when mounted between driving and driven machines - albeit only slightly. As it is impossible to achieve zero Unbalance, all rotating elements will have an inherent error so that, when conjoined, each element will have an influence to increase or decrease the overall balance of the train by a small amount.

06 - DO I NEED MY FLEXIBLE COUPLING BALANCED?

The actual requirement for balancing of a coupling will depend on the magnitude of the inherent unbalance of the coupling and the sensitivity of the machinery to which it is to be fitted. In real terms this will depend upon such factors as the quality of the coupling manufacture, the speed of operation and the mass of the coupling relative to the masses of the machine rotors.

The above statement is, without doubt, an over simplification of the real situation (although correct in it's own right) and it thus seems prudent to examine the considerations involved in flexible coupling balance (without reference to the actual balance procedures which may vary between companies and designs).

Dynamic Balance of rigid shafts or, indeed, Turbine & Pump rotors, etc, whilst requiring a high degree of expertise is, generally, relatively simple. These shafts are, principally, single items that run in bearings. Mounting them on a balancing machine with the sensor bearings located on their running journal surfaces enables a fairly true representation of the actual operating condition.

Flexible couplings, on the other hand, are constructed from a number of individual components and are connected to the machine bearings through one or more interfaces. In addition, they are, in their free state, inherently "flexible" at the flexing joints, a condition which must be overcome to make it possible to balance as a rigid shaft on a conventional balancing machine..

The number of "major" components (excluding bolts, nuts, etc) in the simplest Spacer coupling will rarely be less than five and, in the case of a reduced moment high speed coupling, may be seven or more. Each of these components must be individually produced to practical tolerences and connected to it's neighbour in some form of interface or joint. Add to this the fact that some, or all, of these joints will require to be broken for unit installation and one can start to appreciate the complexity in both coupling design and balancing. Balanced couplings will, normally, have joints which are to be broken on site, marked or identified in such a way as to enable re-assembly in the same position. These marks are generally referred to as "Match Marks". It is usual for Hardware (e.g. Bolts) to be accurately "match" or "balance" weighed (as sets) and as such it is not generally essential for these items to be matched to a location provided the sets are not split.

Development of all types of coupling design has, over many years, led to considerable improvements in the production of coupling components, joint design, etc, such that the inherent unbalance and balance repeat of a multi-component coupling is, at least amongst reputable producers, excellent. Even in the case of Gear type couplings, where a natural clearance between the meshing teeth is required for misalignment, development of tooth profile has led to units that can, at least when new, exhibit excellent repeat of balance.

Despite the advances in design to achieve an acceptable balance, the breaking and re-assembly of any joint will lead to some variation in the degree of unbalance. Hence, in high speed and critical applications, it is common for the coupling supplier to specify certain joints as "factory-assembled" in an attempt to reduce as much as possible the installed unbalance of the unit. This does not mean that the disassembly and re-assembly of the joint will, necessarily, vary the level of unbalance outside acceptable limits but is, generally, merely a means to control what may occur.

To give some idea of the problem of repeat balance when components have been dismantled and re-assembled, to repeat a balance grade of Q0.6 the mass centre will have to be within 0.002mm of it's original position. This is, of course, impossible when allowing for a build up of machine tolerences. Repeat balance is, therefore, normally expected to be within 10X the original grade.

All reputable coupling suppliers will gladly discuss their own methods of balance in detail and will be able to supply written procedures against any particular coupling.

As a final note, no matter how well the coupling and machinery rotors are balanced individually, connection is through two independently produced interfaces and the finite machining eccentricities that will be present in these will mean that, on assembly, some degree of mass eccentricity will exist. If the sensitivity of the machines is very fine, a final "Trim-Balance" of the complete train once assembled may be necessary.

07 – WHY & WHEN DO I HAVE TO LIMIT THE END-FLOAT OF A COUPLING?

Under circumstances where machinery has shafts located with thrust bearings there is no reason why a coupling should have any form of limited end-float (unless operation is vertical - see relevant section). If, however, one of the machines has no rotor axial location and the float of that rotor is purely limited by end stops in the machine, then it may be necessary to consider some form of coupling end-float limitation to prevent the rotor "running" or "hitting" against these end stops. A typical example of such a situation arises with the use of "floating rotor" motors or generators.

Within a Gear Coupling, there is no inherent restraint to prevent movement of the inter-meshing gears within the limits of their travel and, hence, there is no natural tendency for the coupling to prevent the machine rotor from reaching it's end stops. At first sight it may appear that the axial force required to move the gear teeth axially, when transmitting the duty torque, would be ample to maintain the position of the rotor. However, any misalignment present in the unit will permit "walking" of the meshed gears, thus giving little or no restraint to the movement.

Whilst it may be that the natural restraint of Gear Coupling movement is within the limitation of the rotor float, it is, however, important that there is some form of specially designed limitation that will still permit radial/angular misalignment at the limit of travel.

Dry type couplings will not, under normal circumstances, require any form of additional end-float limitation to restrain the machinery rotor from reaching it's end stops. The flexing elements of either Disc or Diaphragm Couplings can be considered as springs having non-linear characteristics. Consequently, any axial movement of the rotor is restrained by a force of increasing magnitude. Under normal running, the rotor will endeavour to run on it's magnetic centre and any attempt to move it from this natural position will induce an axial restoring force increasing, linearly, in proportion to the axial movement. The restraint offered by the flexible elements is small for reasonably modest axial movements and the rotor will have no difficulty in overcoming this to operate in it's correct position [Fig.7.1]. When the machinery is stopped, the whole system will return to it's neutral position. This is not the case with Gear Couplings which will start up in the position in which they were when the machinery was stopped.

Comparison of Coupling & Rotor Axial Force against Deflection.

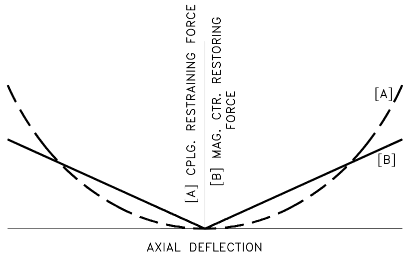

Fig.7.1

If the rotor should, for some reason, attempt to move significantly off it's magnetic centre then, under normal circumstances, the Disc or Diaphragm type coupling's increasing non-linear force will be sufficient to restrict the movement before it reaches the stops.

If the end-float limitation is to be included the coupling supplier will gladly provide that facility by the incorporation of one of a variety of systems. Most of these will be well proven but care must be taken to ensure that, under normal operating conditions of radial/angular misalignment, continuous contact with the end-float device does not occur.

08 - WHAT IF I REQUIRE TO OPERATE THE COUPLING VERTICALLY?

Certain applications involve the connection of two machines with their rotors in the vertical plane. In the majority of such applications, both the machine rotors will be supported in their own bearings. This then means that, provided the coupling hubs are adequately secured to the shafts (either by a shaft nut or a heavy interference) the only real concern to the coupling is the operating position of the central spacer section, assuming the use of a Spacer or Double-Engagement type unit. If one Non-Spacer or Single-Engagement coupling is employed, no special consideration is required.

Typical Vertical Gear coupling (showing support system)

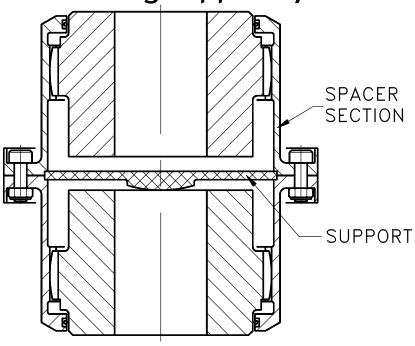

Fig.8.1

In the case of a Double-Engagement Gear type couplings, the nature of the design means that, to prevent the spacer section simply sliding on the teeth until it reaches a lower stop (and staying there), some form of additional support is required. This must be a true support which, whilst offering continual contact between it and the spacer section, will also permit the correct operation of the unit, allowing Angular and Radial misalignment to be accepted [Fig.8.1]. For relatively low speeds, the wear of the contacting parts will normally not cause any problem - a factor that has been proved by many years of successful operational experience of such systems.

By their design, Universal Joints, whilst depending upon the sliding of teeth to cater for axial displacement, do not require, any additions to the standard design since the axial joint is in a single plane and the ends are self supporting. Equally, the Quill Shaft coupling, being a single component is unaffected by the plane of it's operation.

Dry type couplings do not necessarily require a spacer support due to the inherent restoring force of the flexing elements. A Spacer type Disc or Diaphragm coupling, suspended in the vertical plane, will experience a "sag" of the spacer section directly proportional to the stiffness of the two flexing elements taken in "parallel" (not in "series" as is usually considered for calculation of imposed axial thrusts) [Fig.8.2]. The degree of this "sag" will vary with coupling size & design and with the length & weight of the spacer section. In addition, since the axial stiffness of a Dry coupling flexing element increases with speed, the level of "sag" will reduce as the unit starts to operate.

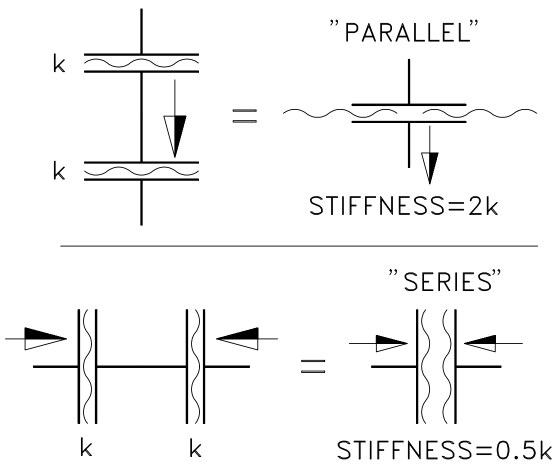

Fig.8.2

Hence, knowing the weight of the coupling spacer and the stiffness of the flexing element, both statically and at operating speed, it is possible to evaluate the resulting stresses in the element caused by the imposed misalignments and the "sag". This calculation can then define if a spacer support is required for the unit. In general, with "normal" length spacers, it is found that these supports are not required, even at very low speeds.

However, if the spacer section is very long (or heavy for any other reason) then a spacer support may well be required and specified. These supports must be positioned such as to allow a controlled amount of "sag" so that, with the stiffening effect of operational rotation, the spacer is raised clear, allowing normal operation without contact between the spacer and the support [Fig.8.3].

Typical Disc coupling in vertical operation.

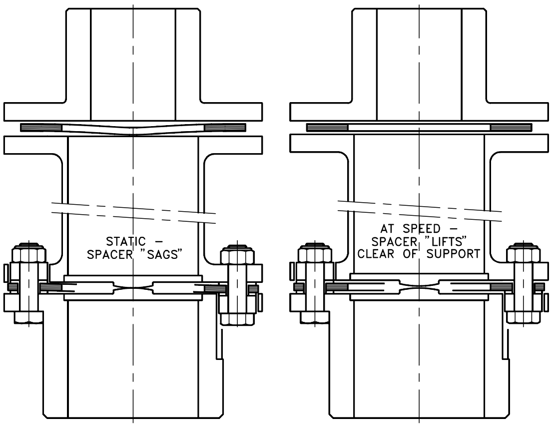

Fig.8.3

If it is deemed that a support is necessary, it is normal practice to produce this such that, at least one of the contacting faces, is spherical to permit Angular & Radial misalignment with the minimum of wear between the surfaces when they are in contact. In the case of a Gear coupling, by virtue of their standard designs, this support tends to be at the lower shaft end and, hence, remote from the gear teeth. The result of this is that, Angular misalignment will create a “rubbing” of the two surfaces and may lead to some wear [Fig.8.4]. In the case of Dry couplings, the support can be positioned at the central point of the flexing element and, as such, Angular or Radial misalignment only cause a “rolling” of the surface and create no major wear [Fig.8.5].

Support “Rub” in Gear coupling

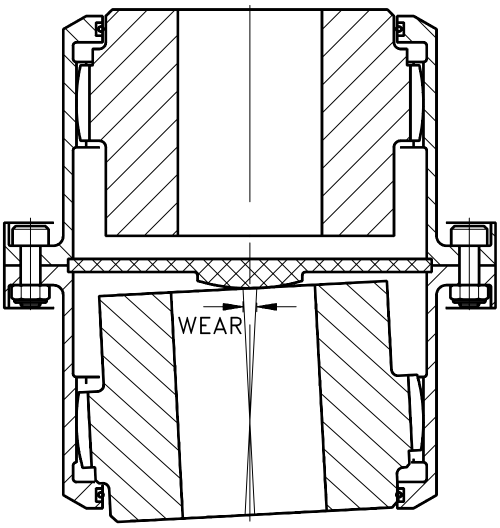

Fig.10.4

Support “Roll” in Dry (Disc) type coupling

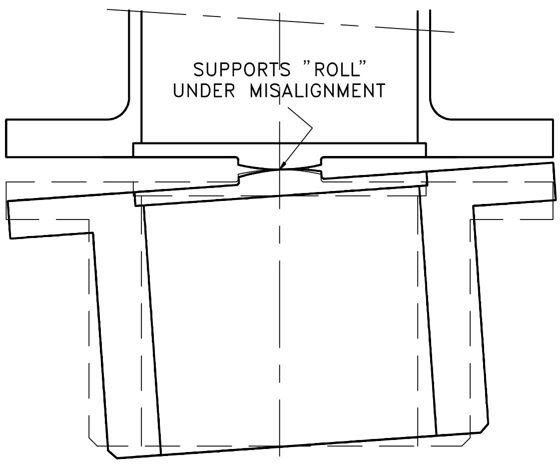

Fig.10.5

If either of the two machines does not have it’s rotor axially located in bearings then a more sophisticated design of coupling that will support the relevant rotor is required. This is usually achieved by the introduction of some form of bearing system and, due to it’s complexity, it is strongly recommended that specific requirements are discussed with your coupling supplier to enable the design of the most suitable unit.

09 - CAN THE TORSIONAL CHARACTERISTICS OF THE COUPLING BE VARIED IF I CALCULATE A PROBLEM?

The requirement to analyse the torsional characteristics of a machinery train will depend on it's complexity and the likelihood of the machinery to introduce an exiting force. However, with the complexity of modern machinery, the initial and down-time costs and the relatively simple nature of analysis using computers, torsional analysis of complete machinery trains is usually performed and recommended.

The results of such an analysis, if they predict a resonance within the running range, can lead to the dilemma of how to adjust the machinery, which may well have already been produced or ordered. To modify the torsional characteristics of a main item, such as a Turbine or Pump, could well prove difficult and costly, if not impossible. Hence one is forced to look to the more easily modified parts of the system such as the coupling.

Typical examples of design modification to a Disc Coupling to vary the Torsional Stiffness (showing related values).

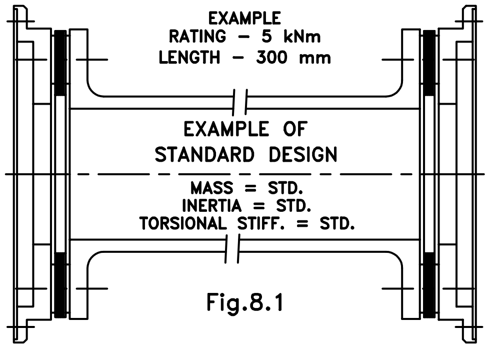

Fig.8.1

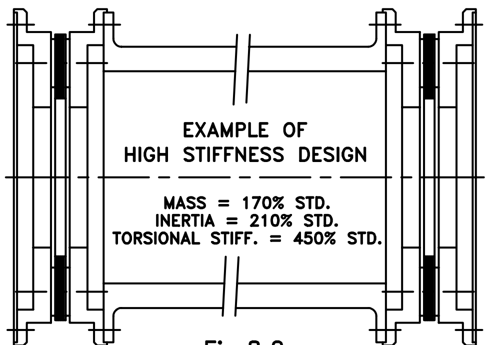

Fig.8.2

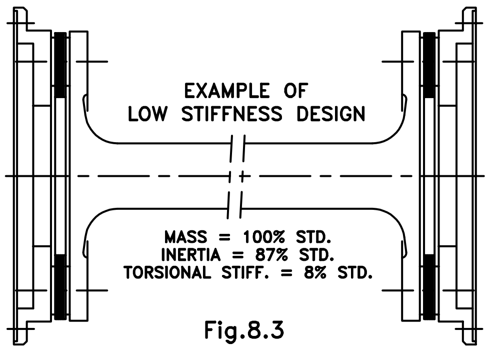

Fig.8.3

Whilst we are, in general, considering flexible couplings which are principally classed as “Torsionally Rigid”, they do, of course, exhibit some degree of torsional flexibility by virtue of the physical modulus of the material. This is distinct from units that are designed specifically to offer torsional flexibility by the use of rubber or springs but have their own limitations of speed, etc, which restrict their application.

It is, therefore, frequently possible to vary the torsional stiffness of any Spacer or Double-Engagement coupling sufficiently to adjust the overall stiffness and remove the train resonance from the problem area. The degree to which the variation can be achieved will, obviously, depend upon the design and size of the coupling but experience has shown that sufficient change is usually possible with most coupling types (with the exception of Quill shaft units), even if a compromise may, ultimately, be required.

Increase in a “standard” coupling stiffness [Fig.8.1] will, generally, be achieved by increasing the stiffness of the spacer component [Fig.8.2], limitation coming purely from the size restrictions imposed on the unit. However, it is important to note that, in general, increasing the stiffness of the coupling does increase it's weight. Hence, care must be taken to ensure that this weight increase does not adversely effect the lateral critical speed of the adjacent machinery.

Decreasing the stiffness of a coupling will again, primarily, involve an adjustment to the spacer tube dimensions, ultimately leading to a “Quill Shaft” type spacer design [Fig.8.3]. When considering a reducrion in stiffness there can also be a benefit in using alternative materials with a lower

modulus of rigidity, such as Titanium or Aluminium alloy, although these can often result in a substantial cost penalty.
In both cases it is worth noting that, if the problem is discovered at the design stage, an adjustment to coupling length will vary the coupling stiffness without the need for major design changes and will often prove more cost effective in terms of the coupling, if not the overall system. It should also be noted that a change in unit design to modify the stiffness will also modify the coupling Inertia which must be considered in any re-calculation.

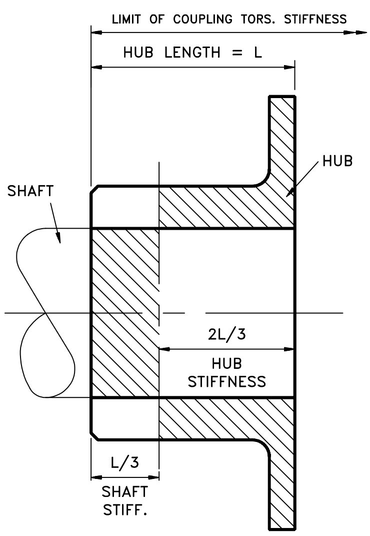

Fig.9.4

It is general practice for a coupling supplier, along with his primary bid, to specify the torsional characteristics of the coupling. This value is then used in the initial train analysis and, if a problem is then highlighted, an indication of the preffered stiffness can be given to the supplier and a re-design performed. Most coupling reputable coupling suppliers will be glad to offer a unit against these specified requirements.

Finally, as a word of warning, when evaluating the torsional resonance of the system, the extent of the torsional stiffness as indicated by the coupling supplier should be determined. When considering a shaft end to hub connection, it is common practice to assume a torsional stiffness for the coupling including "1/3 shaft penetration of the hub" (i.e. the value includes for the whole length of the coupling, allowing for the stiffness of the machinery shaft for 1/3 of the hub length and the hub stiffness for the remaining 2/3 [Fig.9.4]). Whilst this may be common practice, it is by no means a standard and must be checked. In addition, not all analysis programmes work on his premise and a value for "1/2 shaft penetration", etc, may be required in some cases.

NOTE: There is a formula commonly quoted to define Torsionally Rigid or Torsionally Flexible couplings as follows:

$$K = \frac{\text{Torsional Stiffness (Dynamic) [Nm/Rad]}}{\text{Nominal Torque Capacity [Nm]}}$$

K < 10 = Torsionally Highly Flexible
K 10-30 = Torsionally Flexible
K 30-100 = Torsionally Resilient
K 100-300 = Torsionally Stiff
K >300 = Torsionally Rigid

The Disc Coupling of Fig.9.5 was supplied for use between a Motor & Gearbox (on a Compressor drive) at 570 dbse. The unit is designed for a continuous torque of 3000 Nm and gives a Torsional Stiffness of 106000 Nm/Rad giving K = 35.

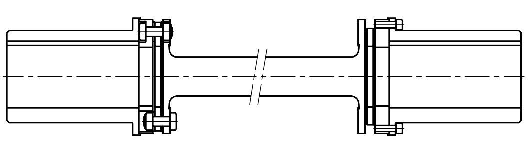
Fig.9.5

The above formula now defines this Disc coupling as "Torsionally Resilient", bordering on the "Torsionally Flexible". Hence, we would emphasise to the reader that, just because the types of Couplings discussed here are refered to as "Torsionally Rigid", it does not mean that they cannot be designed to suit applications where more torsionally flexible couplings are traditionally used.

10 - HOW DOES THE WORKING ENVIRONMENT EFFECT A FLEXIBLE COUPLING?

In general, couplings which are principally all metallic, i.e. those without rubber or plastic components, are very tolerant to fair extremes of environmental conditions. However, there is a difference in the way flexible couplings react to Chemical or Temperature environments and, hence, we will treat these separately.

CHEMICAL ENVIRONMENTS

The range of Chemical environments in which a coupling may be required to operate is very varied as indeed are the methods employed by coupling designers to cope with these. If one is selecting a unit to operate in extreme, or unusual, conditions, the overriding factor will be the effect of that environment on the lubrication in the case of Gear couplings or Universal joints and the effect on the metallic components in the case of all couplings.

The effect on lubrication of a Gear coupling or Universal joint will depend on the actual lubricant used and the degree to which it is open to the environment. For example, in the case of a "Grease Packed" Gear coupling, seals must be adequate to prevent any harmful atmosphere from contacting, and so possibly causing a breakdown of, the lubricant. The effect on the material of the seal must also be considered.

It is, obviously, much more difficult to prevent contact of the atmosphere with the metallic components of any coupling. Hence, when considering the operation in any extreme Chemical environment, care must be taken to "coat" all exposed surfaces or select materials for construction that are unaffected by the chemicals, etc., to which they are exposed.

The use of coatings on standard carbon or alloy steels is, generally, less expensive than using alternative base materials especially if it enables the use of standard components. However, there are certain, important, limitations to thier use. Firstly, the coating will, in all probability, be relatively thin and, hence, any surface damage caused by, for example, mis-treatment, will open the base metal to the atmosphere and, thus, defeat the object (at least locally). Secondly, the coating must cover all areas of the unit that could, conceivably, be exposed to the environment. This can, in certain circumstances where component fit or size is critical, be difficult to ensure and may even be overlooked at the design stage, potentially leading to corrosion in the worst possible areas.

It may, therefore, be worth considering, in cases of extreme environments and highly critical applications, the use of more suitable base materials. Since there are such a wide range of corrosion resistant materials available it is usually possible to find one, albeit, at a variable cost penalty, to suit the application. However, having made such a broad statement, it is important to realise that these materials may not be mechanically suitable and, hence, not applicable in all situations or components.

Thus, in reality, a compromise between coatings and special materials may need to be made (such as using stainless steel fitted bolts & nuts in a coupling with the major components produced in carbon steel coated with Epoxy Paint). In these case, other factors such as "Galvanic Corrosion" may have to be considered.

Due to the complexity of coupling design for extreme Chemical Environments, it must be emphasised that the coupling supplier should be furnished with as much detail as possible of the environment in which the unit is to operate and then be permitted, albeit in consultation with the purchaser, to use his experience to offer the most suitable materials and coatings in the design of the final unit.

TEMPERATURE

Lubrication in Gear couplings or Universal joints is, obviously, highly susceptible to the effects of extreme temperatures. The effectiveness of a lubricant at very high or low temperature is well documented and, hence, the strict limitation on Gear couplings and Universal joints in such cases is easily apparent. With a breakdown in the lubricant, even for a short duration, the rapid deterioration of, for example, the teeth in a Gear coupling can easily lead to premature unit failure, usually of a catastrophic nature.

In principle, the effects of temperature on the operation of a unit would seem easy to assess. However, in extremes, one requires a full understanding of the stress levels in all components of the coupling to cater for the variation of fatigue, creep, impact etc., that the materials will adopt under these temperature conditions. In addition, the designer must consider the effects of thermal expansion and contraction of the coupling, especially allowing for differential growth where two different materials may be present.

In general, if the unit is to operate in cryogenic or extremely high temperatures, such that an alteration to the standard material structure is likely to occur then the application of special materials is obviously required. If, in the application of these materials, an adjustment in stress levels, to account for Creep, etc., is required then the appropriate design changes must be made.

If the temperatures are not so extreme but are still sufficient to have any effect on the material performance, then consideration of reduced stress levels or a need to vary the material may be required. Under these conditions, the time for which the unit may be exposed to these temperatures may become critical (e.g. if high temperatures may only be the result of start-up or seal failure, etc.) since, under limited duration, a reduction in creep stress may not offer any substantial detriment to the operation of the unit. Where the situation is “border-line”, it may be that a slight “de-rate” of the coupling will permit the use of standard materials, hence saving complexity and cost.

In all cases, a degree of faith in the coupling supplier’s ability to understand the operating limitations of his product is required. The user will, thus, need to satisfy himself of the supplier’s experience and ability to offer a suitable unit.

11 - IS FLANGE BOLTING IMPORTANT?

The transmission of power between any driving and driven rotational machinery will require, at the very least, connection by means of a joint; probably in the form of two connected flanges, either integral or in the form of hubs fixed on the shaft ends. Obviously, without some form of connection between the two they are incapable of transmitting torque. By far the most common method of connecting the two flanges is by means of bolts which, when tightened, go part way to simulate an integral connection.

As has been discussed in earlier questions, such a simple joint has severe limitations in as much as it does not have the necessary properties to allow for misalignment of the two connected shafts. The introduction of a flexible coupling, which will comprise a number of individual components, will have a corresponding number of flanged joints all of which have to be securely joined. Unfortunately, the problems associated with the bolting at these joints often makes it considerably more complex than just joining two flat flanges in a satisfactory manner.

The primary aim in such a joint is to provide bolting of adequate strength to transmit the torque in one of two ways, both methods being in current use depending upon the design philosophy of the particular producer.

Firstly, the bolts can be sized to ensure adequate proportions to transmit the torque in shear. However, a bolt designed to this criteria is not necessarily capable of producing enough axial force to prevent slip between the mating surfaces of the joint. This problem is often overcome by the use of “fitted” or “body bound” bolts which are a close fit in both flanges, or by a combination of bolts and fitted dowels.

The second approach to the problem is to design the bolting of the joint to ensure that there is adequate pressure between the flange faces to prevent slip and so transmit the torque by friction. The dimensions of the bolts used in this solution will, undoubtedly, be larger than those used in the previous method but, by the careful selection of materials, even this seeming disadvantage can be minimised.

We now have two solutions to the problem of how to make an adequate bolted joint capable of satisfactorily transmitting torque. However, for reasons which will become obvious, in some coupling designs, e.g. the Disc coupling, the second solution offers a superior design when considering long term, successful, operation of the joint between the flexing element and it’s associated flanges.

In every type of all metallic flexible coupling other than the disc, that portion which does the flexing is in “free space”, unencumbered by adjacent flanges. What joints there are - and there may be a number - are face to face in contact over, what is usually, a relatively large diameter. In such cases, a considerable number of bolts can be used to ensure a satisfactory joint. Even with this type of joint, however, due consideration must be given to the extraneous forces to which the joint may be subjected - namely shock, overload & vibratory torque, over and above the normal duty.

The main joint in a disc coupling, i.e. the connection between the flexing elements and their adjacent flanges, is substantially different and poses a different set of problems.

The fundamental difference in the design of a Disc coupling is that it requires the flexing element to operate within the diameter of the flanges to which it is connected. To give the necessary freedom of space in which to operate, the flexing portion, namely the stack of

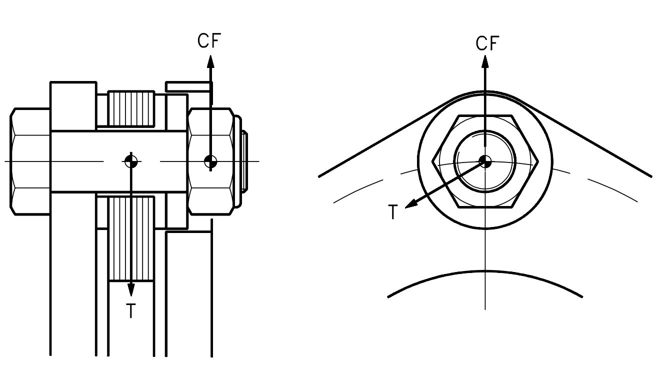

Fig.11.1

laminations, must be separated from the flange to which it is attached in order that it may flex. This is accomplished by inserting a washer between the laminations and the flange.

The design of the jointing arrangement is, therefore, somewhat different to that of a normal flange. Firstly, the number of bolts which can be utilised directly affects the flexibility, as well as the torque carrying capacity, of the unit and is therefore restricted. Next, considering the fact that the flexing laminations are overhung from the flange, the pull in the laminations transmitting the torque imposes a bending moment on the bolt. Added to this is the influence of the weight of the nut under the effect of centrifugal force, at higher speeds, which will also contribute to a tendency to bend the bolt [Fig.11.1].

[It may be worth pointing out at this stage that the speed in RPM is not necessarily the criteria that determines whether a coupling should be considered as high or low speed - the flange peripheral speed is a far better indicator].

Taking these, not insignificant, considerations into account it will be appreciated that the bolting arrangement for the flexing joint, based solely on adequate shear strength in the bolts to transmit torque, will be unlikely to cope with the full requirements. The ideal joint under these circumstances must be one that utilises bolts that will exert sufficient axial force to prevent both slip and bending.

Why such emphasis on the prevention of slip between the mating parts? Because the effect of such movement has caused many failures, not only in couplings, but in some very sophisticated machinery. The effect is that of “fretting”. When two surfaces slide over each other, albeit that the movement is extremely small, a minute amount of material is removed from the surfaces, particularly if the surfaces are restrained in close contact with each other as in a bolted joint. One slip due to, say, shock loading, although undesirable, is unlikely to be detrimental to the operation of the unit. If, however, such shock loading should be of a reversing nature, then the effect will soon manifest itself as a coupling failure as the “fretting” undermines the integrity of the flange. Also, in order to cater for angular/radial misalignment, the laminations in a Disc coupling will go through cyclic deformation which will involve ‘pushing’ and ‘pulling’ the laminations at the anchor points even without any cyclic variations in the torque being transmitted. Consequently under these conditions there could be a large number of sliding and, so, fretting surfaces if the joint bolting cannot be sufficiently loaded to fully clamp the flanges.

Once fretting has started, it’s progress is rapid as a minute amount of material is removed between the flanges & each lamination twice in every revolution. Everytime material is removed, the clamping distance of the bolted joint is reduced and, consequently, bolt tension is rapidly reduced towards zero, having the same effect as a bolt that was not tightened in the first place. Failure is then imminent as, due to the thinning of the laminations, they are no longer able to transmit the torque and the corners are torn out, or indeed the bolts break.

It will be appreciated, therefore, that any chance of slip under normal operating conditions must be eliminated and that this will depend on the use of bolting which is capable of clamping forces adequate to prevent this happening. The fact that the shear strength of such a bolt will be much more than adequate is merely an additional safeguard under overload conditions.

The correct torque tightening of the bolt is, consequently, critical and can be ascertained in several ways, one by applying the torque recommended by the coupling supplier or by measuring the stretch of the bolt which offers a more accurate control and is useful for the more critical applications.

To overcome the problem of fretting between laminations when using inadequately sized bolts, some coupling manufacturers advocate the use of a coating on each individual lamination. The type of such coatings varies but, in essence, they are merely dry lubricants which prevent metal to metal contact. The adoption of such a practice would tend to indicate that some movement between the laminations has been observed or is expected. In the short term such coatings may prolong the life of the coupling but, in the long term, may not provide the extended life which can, and should, be expected.

Many years of operational experience has proved that a correctly bolted joint, designed to drive in friction, does not suffer the problems associated with fretting.

Other methods of making a joint capable of transmitting torque are, of course, possible. Such methods will include riveting and welding. Of these, welding is far superior and is used to effect where, as in a Diaphragm coupling, the inner circumference is connected at a relatively small diameter making the use of bolting difficult, if not impossible. Riveting is only rarely used as it offers little control over the loading in the clamped joint when considering the actual flexing portion of the coupling. It is, however, sometimes used effectively in flange to flange joints of large diameter where a very large quantity of small rivets can be used.

The problem with both of these methods is that the resulting joint is permanent and, consequently, of no use where a coupling has to be dismantled to allow installation between driving and driven machinery.

12 – WHAT ARE THE EFFECTS OF SPEED ON COUPLINGS

Flexible couplings of whatever type are, generally, intended to rotate with the speed of the rotation being dictated by the application. Reputable manufacturers have understood the effect of this rotational speed on both the integrity and operating characteristics of couplings for many years. However, with the increasing speed requirements encountered in many sectors of industry, the depth of the understanding continually needs to improve.

The effect of the rotational speed can be split into two main areas:
- The windage effect created by the unit profile and protrusions.
- The centrifugal effect on the various components of the coupling.

The windage effects are covered in some detail in Question13 of this document and, as such, we will concentrate, in this section, on the centrifugal effects.

As with other sections of this document, we will divide the discussions between the main types of 'torsionally stiff' couplings and look at the effects the speed has on the various components. Some of the factors will be common between these and, rather than keep repeating sections, cross-reference will, in certain cases, be made.

12.1 GEAR COUPLINGS

As outlined in an earlier section, the design of Gear type couplings depends on the relative movement of the contacting surfaces of gear teeth to permit misalignment and, as such, the imposed forces and life of the unit will be highly influenced by the lubrication supplied at these surfaces.

The simplest form of lubrication is 'grease packing' in the area of the teeth, the ends being sealed to contain the grease. With the use of any grade of grease, there will be a limitation to the operating speed of the unit over which the centrifugal effects will start to separate the components of the lubricant. At this point, the effectiveness of the lubrication is substantially lost and any level of misalignment, and hence relative movement of the teeth, will cause excessive wear.

Under an angular or radial misalignment, the relative movement of the teeth is cyclic with each revolution of the coupling. As such, the higher the rotational speed and the greater the misalignment, the greater the rubbing speed of the teeth which means that there is a direct limiting relationship between the allowable speed & misalignment for a gear coupling to maintain an acceptable level of life for any lubrication condition.

CONTINUOUS OIL LUBRICATION

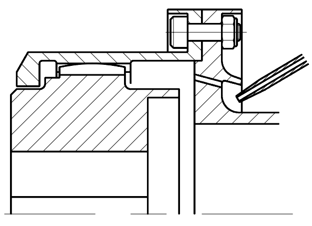

Fig.12.1

In addition, increased rubbing speeds mean increased temperatures that will, beyond a certain point, start to degrade the lubrication and, in turn, increase the coefficient of friction and further increase the temperatures. As such, beyond a certain point in the combination of misalignment & speed, there is a situation of 'self destruct' for grease packed gear couplings.

Beyond the upper boundary of speed resulting from the limits of temperature and centrifugal effects on grease, the coupling designer needs to consider 'Continuous Oil Lubrication'. Such a system is shown in Fig.12.1.

There are two major benefits with the use of this system that increase the speed capability of the gear coupling. Firstly the continuous flow of oil across the teeth means that the lubricant will not absorb the full heat generated from the meshing and, as such, is less likely to reach the position whereby it will 'breakdown'. Secondly, the heat it does absorb will be removed from the coupling and dissipated in the oil tank, meaning that the unit is, to some extent, cooled.

Despite these, and other, benefits with continuously lubricated gear couplings, there is still a final limitation in speed due to the centrifugal effects on the lubricant.

12.2 DIAPHRAGM COUPLINGS

The influence of speed on 'dry type' couplings (Diaphragm & Disc) is strictly related to mechanical effects. Fundamental to the design of any coupling, including gear couplings, running at speed are the stresses that become inherent in any flanges. The calculation of the hoop & radial stresses in such components has been understood for many decades and, with computers, finite element analysis, etc., the accuracy of predictions has increased dramatically and consideration of this will be normal amongst all high-speed coupling manufacturers.

Equally fundamental is the lateral critical speed of the coupling. The unit can be considered as a rotating central shaft suspended between the two Diaphragms with these normally, for calculation purposes, being considered as pinned joints. The result is that there will be a lateral critical speed (or whirl speed) that is dependant on the size and profile of the central shaft and the length between the flexing planes.

It is important that the design of the coupling is such as to ensure that critical speed will not occur too near that of operation. It is normal for the lateral critical speed calculated for the central shaft between the pinned joints to be at least 50% above the maximum operating speed (e.g. API671). The use of this factor is intended, firstly, to give a margin of safety that accounts for approximations and assumptions in defining material values and dimensions in the calculations. Secondly, it is intended to allow for the reduction in real operating lateral critical speed due to the 'sag' of the machinery shafts and the stiffness of the shaft bearings.

Whilst a simple analysis is sufficient for most applications, when the related machines may be considered sensitive, have long shaft end to bearing distances or have relatively heavy couplings attached, a full train analysis should be considered. Whilst it is rare for flexible couplings to be run 'super-critical', should it be considered, a full analysis should be seen as essential.

Even 'sub-critical', a coupling of substantial length can, if not correctly designed, manufactured and balanced, exhibit instability. To overcome this, any coupling of significant length should undergo a static and multi-plane dynamic balance.

With the centrifugal stresses introduced into flanges & diaphragms, comes radial growth. This growth will depend, amongst other factors, upon material constants and the outer and inner diameters of the components. As such, whenever one is considering flange interfaces, it is important to assess differential growth with regard to the possible extreme effect of locations disengaging. It is preferred, wherever possible, to use 'self-tightening' spigot/recess locations, that is, to ensure that the component flange with the spigot with grow more under centrifugal growth than that with the recess.

However, it is not always possible to have self-tightening locations and, in these situations, consideration needs to be given to the actual relaxation, compared with the installed interference of the mating parts, to assess if actual disengagement will occur. In addition, if one is considering joint interfaces which have substantial bolting giving a heavy friction between the faces then calculated differential growth will be reduced and, even if disengagement occurs, there is unlikely to be any radial movement of adjacent parts to cause unbalance & vibration.

When anything rotates, the centrifugal effect that causes it to grow radially also causes it to stiffen. An example of this is a simple grinding disc attachment for a 'pistol' drill. Hold it in your hand and it is completely flexible. Rotate it at high speed in the drill and it becomes rigid and stiff enough to cut through very solid objects. This same effect happens to Diaphragms when rotated at high speed. The radial expansion will 'stiffen' the diaphragm such that the axial stiffness of the coupling when stationary increases disproportionately with the speed of rotation.

As such, when reviewing the acceptability of loads imposed on machinery by couplings under conditions of misalignment, care must be taken to ensure that the forces stated relate to those at the appropriate rotational speed. Using only the loads related to a coupling in its static condition could easily be the downfall of a high-speed machine.

12.3 DISC COUPLINGS

Disc couplings require much the same consideration as given to Diaphragm couplings and, as such, even if focusing on disc couplings, the above section should be read first.

Lateral critical speeds and self-tightening of flanged joints occur in exactly the same way as above and the approach to design should be the same. Flange stresses at rotational speed are again covered by the same analysis. However, in Disc couplings it is normal for the main flanges (those adjacent to the flexing discs) to be produced with larger clearance holes for the 'free' end of the bolting. In this case, the effect of these holes should be considered when assessing the flange 'bursting' stresses.

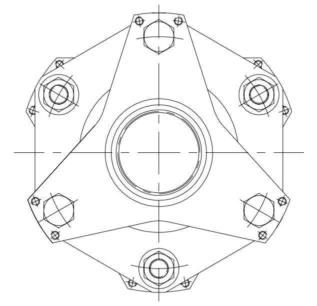

Fig.12.2

In certain designs of Disc coupling, rather than simply have large clearance holes, the complete 'redundant' section of the flange will be removed, leaving 'lugs' to which the main bolts will be attached and giving the flange a 'star' appearance [Fig.12.2]. In this case, the 'lugs' should be considered as free masses introducing stress into the remaining complete disc section of the flange.

The centrifugal growth of the disc coupling components again increases the stiffness of the flexing element. In the Diaphragm coupling this is simple to understand as the stiffening of a rotating 'disc' (in fact diaphragm). In the Disc coupling, the radial growth actually increases the bolt circle diameter. This increase in the diameters has the effect of increasing the chord distance between the bolts, which is translated into a tensile stress in the flexible disc section between the bolts [Fig.12.3].

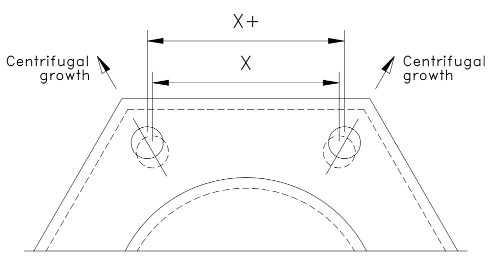

Fig 12.3

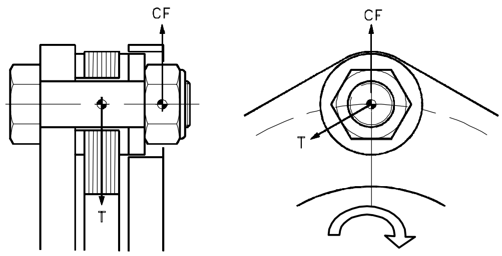

Fig.12.4

As with the Diaphragm couplings, consideration must be given to this 'stiffening' effect when sizing bearings, etc.

Whilst one could write volumes on the intricate analysis of speed effects on every component in the coupling, we will look at one last major effect. This relates to the centrifugal forces imposed on bolts/nuts when the unit is rotating at high speeds and was touched on in Quest.11. A typical bolted section for a disc coupling is shown in Fig.12.4 with 'CF' representing the centrifugal force resulting from the rotation of the coupling. The magnitude of this force will depend upon the speed of rotation, the radius of the bolt from the axis of rotation and the mass of the nut.

The centrifugal force on the nut will introduce a shear into the bolt but, more importantly, it will also introduce a bending stress into the bolt. The level of this bending stress is directly related to the magnitude of 'CF' and the distance from the point of action of the force and the point of 'anchor' for the bolt. The point of action of the load can, sensibly, be assumed as the mass centre of the nut. If, as shown in the sketch, there is an additional washer, then the 'CF' related to this need also to be considered, with its point of action at its mass centre.

Defining the point of 'anchor', the point along the bolt's length where it changes from fitted & located to free to bend, is not as easy. The definition of this point is dependent on how well the bolt fits into the holes and how much resistance is predicted from the pack of discs.

If the design is incorporating bolts that carry sufficient axial load such that the bending and shear cannot occur under the maximum operating 'CF' then the effects can be disregarded. However, whilst the actual load in the bolt, if correctly assembled, should not reduce during operation, it is safer to ensure that the bending effects will not cause damage under any circumstances, hence avoiding any potentially catastrophic problems.

Each Disc coupling manufacturer will have his own theories and using a reputable and well-established high-speed coupling supplier should ensure this is never a problem.

13 - WHAT IS WINDAGE & HOW CAN I REDUCE IT?

Any item moving through the air will meet with wind resistance. In the case of a body which is propelled, such as a car, this wind resistance results in a retarding effect on the vehicle which, in turn, has to be compensated for by increased power output of the engine. Modern day vehicles and other propelled bodies are designed in such a way as to be "streamlined" so as to reduce the effects of wind resistance and the resulting power loss with it's related manifestations such as noise.

A rotating body such as a coupling, or even a plain disc, undergoes the same form of wind resistance, with the same results. In this case, however, the surrounding atmosphere is, generally, retained within an enclosed guard with, maybe, only one or two relatively small vents to permit the movement of the air in and out of the guard. At high speeds, the rotating, enclosed, body will create turbulence within the "trapped" air which will not only lead to some degree of power loss but may also create temperature rises and pressure variations within the locality. The level of these effects will depend upon the speed of rotation, the viscosity of the atmosphere, the relative dimensions of the enclosure and the ability of the atmosphere to enter and exit the guard (the vent size).

Since, in terms of couplings, the effects of wind resistance or "windage" only become noticeable at high speeds, Universal Joints and grease packed Gear couplings, which are usually only utilised on low speed applications tend not to be concerned with windage effects.

Whilst there are various papers and suppliers guide lines on the level of windage created by rotating bodies and the resulting effects on power loss, temperature rise and pressure changes, these are of limited reliability - a statement best illustrated by the inconsistency of the results obtained using the different theories on the same application.

Coupling suppliers, likewise, have their own theories on windage effects, generally based on the published theorems adapted by a knowledge of their own designs and corrected using application and test data. However, the results obtained using these theories are equally inconsistent, a problem that is often compounded by the "lesser" coupling suppliers relying upon "copying" other's theories without fully understanding the principles or relevance to their particular design.

Because of the complex nature of the formulae used in evaluation and, indeed, the stated inconsistency between theories, it is not appropriate to discuss the points in great detail. However, a general discussion on cause, effect and the methods of reducing these effects will, at least in brief, be attempted in respect to high speed operation of couplings.

As already stated, even a plain disc or tube rotating at high speed may cause a noticeable degree of windage. Add to this the effects of irregularities such as holes and protruding bolt heads & nuts (and an "enclosed" atmosphere) and the effects of the windage can be greatly magnified. The resulting turbulence effects the surrounding atmosphere and manifests itself, principally, in the following three ways:

- POWER LOSS - In most coupling applications power loss (as opposed to Energy Loss which manifests itself as a combination of the three effects) is, thankfully, relatively small even for the most crude designs and, thus, can be ignored in normal applications. The power loss due to windage must not be confused with that arising from operation of a Gear coupling and resulting from the mechanical losses in intermeshing gears.

- TEMPERATURE RISE - Whilst the power loss is small, the dissipation of energy in a confined space, such as an enclosed guard, can lead to, what may at first seem, a disproportionate rise in the temperature of the coupling and it's surrounding atmosphere. The degree of this temperature increase is, obviously, dependant on the relative proportions of the guard & coupling (i.e. the volume of adjacent "air") and the total area and position of vents in the guard. These factors combine to establish the amounf of turbulence (at a given speed in a set particular atmosphere viscosity), the speed at which the heated "air" can be exchanged with that outside the guard through the vents and the rate of heat dissipation through the guard walls. The permitted temperature within the guard is usually defined for reasons of safety either as that acceptable for the guard or, for example, to avoid reaching the flash point of the oil containing atmosphere. It may seem patronising but it is, never the less important, to emphasise that it is pointless to operate in an ambient temperature of, say, 90 degrees Celsius and specify a maximum temperature within the guard of 85 degrees!

- PRESSURE VARIATION - Basic engineering or physics highlights the fact that turbulence causes pressure changes in it's locality. The greater the turbulence, the greater the pressure variation that will occur between the region of disturbance and that of the relative calm surrounding it. In relation to couplings, this effect is of most interest in the neighbourhood of the shaft bearings (i.e. at the ends of the coupling & their interface with the machinery shafts) where decrease in pressure at the coupling flange can cause substantial pressure decrease and, due to the resulting differential, "suck" oil from the bearings. This, whilst normally not being detrimental to the bearing or the coupling (metallic couplings are rarely affected by the presence of oil) it can, in extreme cases, create a problem in maintaining a suitable oil flow to the bearings. This situation can often be greatly improved by the introduction of labyrinth seals to the shaft, outboard of the bearings. This may not, however, always be practicable or, in the cases of extreme speed and dimensional restrictions, be sufficiently effective to reduce affect to acceptable levels.

High speed, continuously lubricated, Gear couplings, have the benefit of an inherent cooling oil supply. This is, however, limited to the region of tooth mesh which, whilst reducing some of the effect of temperature rise in that locality, is limited in its ability to reduce the overall windage effects. It is possible to take this principle a stage further and apply to all couplings by introducing an oil spray cooling system for the coupling and guard. Whilst this is, generally, no detriment to the coupling and will offer some reduction in the temperature, it does have the negative effect of increasing the viscosity of the atmosphere within the guard which in turn increases the windage and its other related effects.

Windage reduction by design is, obviously, the preferred solution to any detrimental effects that high speed coupling operation may cause. For any flexible coupling flange which contains bolting, various forms of "shrouding" can be introduced to reduce the overall effects of windage. A typical example is the, almost standard, shrouding over bolts & nuts in a Gear coupling flange [Fig.13.1]. The use of such shrouding is even more valuable in Dry type couplings where the flanges tend to be closer to the end of the guard and the bearing shafts and more specifically in Disc couplings, where the bolts tend to be larger than for Diaphragm couplings (although they, in turn, have problems due to larger flange diameter and thus higher peripheral speeds).

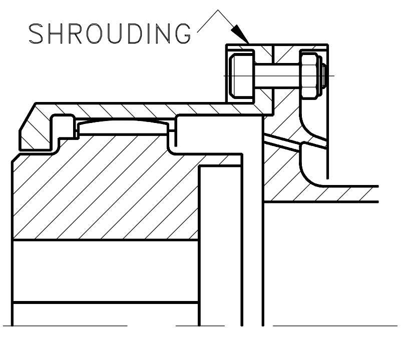

Fig.13.1

By way of an example, we can consider the various possible stages of shrouding typically available on a high speed, low moment, disc coupling. The "standard", unshrouded unit has protruding bolts & nuts which add to the windage effect of the flange & tube rotation effects

“Standard” Disc coupling

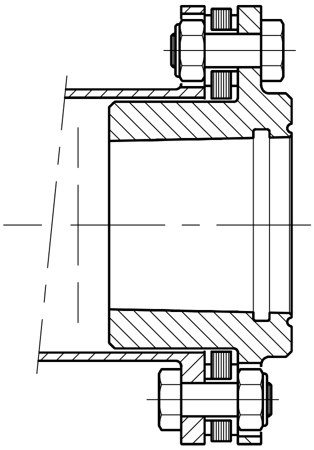

Fig.13.2

“Stage” one - Windage Shrouds

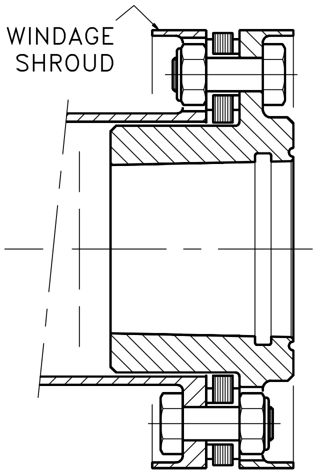

Fig.13.3

“Stage” two - Windage Plates

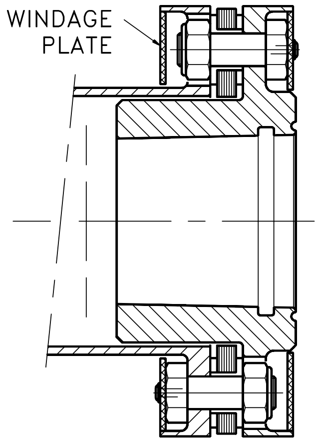

Fig.13.4

[Fig.13.2]. As a first stage of shrouding, the flange can be extended over the bolts & nuts to give the “conventional” shroud [Fig.13.3]. This form of shrouding improves the windage effect by a value in the region of 0.25 against the “standard” unit (Note this is a qualitative estimate of a typical case and is no way intended as an precise reflection of formulae used by suppliers nor does it represent a change in the overall effect of windage due to cylinder/flange rotation, it actually slightly increasing the effect of cylinder length).

The second stage is to totally enclose the bolt heads & nuts by use of a “windage plate” or “baffle” which is fixed to (and rotates with) the coupling flange [Fig.13.4]. This “hides” all the major protrusions and, on the basis of the earlier statement, can be considered to improve the windage effect due to the bolts & nuts by a factor of 0.1 against the “standard” unit.

Whilst the second stage shrouding results in a series of “smooth” faces and cylinders, the sectional profile of the coupling is still one of sharp corners and “steps”. To further improve the overall effect, most reputable coupling suppliers can offer a “streamline” coupling which not only offers a smooth surface to the unit but also eliminates rapid changes in profile and, hence, introduces the best possible “laminar” air flow around the rotating coupling.

Having discussed, albeit briefly, the effect and limitation of windage effects resulting from high speed coupling operation, it must be emphasised that the correct, or indeed incorrect, design of the coupling guard can have a greater influence on windage effects than any of the coupling modifications mentioned above. Hence it is recommended that guard design is treated with as much importance as the coupling design and readers are encouraged to refer to one, or more, of the learned papers written on this subject.

As a final note, if the reader has any concern about windage effects with regard to a coupling application, most reputable coupling suppliers will gladly discuss the subject in greater detail and be happy to advise on both coupling and guard design.