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WHY FLEXXORS?

I will discuss here, some of the reasons why FLEXXORs are designed differently from all other flexible couplings, and why they have such an excellent record for solving difficult drive problems.

In recent years, many new demands have been made on flexible shaft couplings. Years ago, most centrifugal compressor applications were limited to low pressure air and low-pressure refrigerants. Shaft torques and speeds were relatively low compared to those used today where high density gases are the common thing for compressors, and the torques and speeds have increased dramatically. This has resulted in much higher torque and higher speed requirements for flexible couplings. In addition, today many gases being handled require tight shaft seals, and the temperature differences between machines have tended to increase. All of these factors help to make flexible coupling requirements more stringent than ever before.

With these increasingly difficult requirements for couplings, the problems that have developed in the field have been more and more frequent, and more and more severe. The FLEXXOR was developed to overcome the many problems and difficulties which occurred with commonly used gear type, or multiple strip flexing couplings.

NO WEARING PARTS – FRICTIONLESS

One of the most important characteristics of the FLEXXOR design is that it is frictionless. In any flexible coupling where there is misalignment between shafts, there is necessary relative motion of the parts within the coupling. If these parts slide on each other, such as they do in the gear coupling, or in a multiple strip stack coupling, then there is bound to be wear as these parts slide on each other. Eventually, this must result in failure of the coupling. This is a common cause for couplings to wear and fail. This also wastes energy. The FLEXXOR has no rubbing parts whatsoever, since all flexible disks are separated from each other, and relative movement between the parts simply bend the disks within the tolerance of allowable fatigue stresses.

With no friction and no wear, no lubrication is required in the FLEXXOR. This means that there is no maintenance required for these couplings.

If there is friction in a coupling, then the working of the parts generates heat. This heat must be dissipated to the atmosphere as a loss of energy. In the case of a multiple stack of strips used in couplings such as the Thomas, the friction between these strips causes wear, but also the outer strips are cooled by the atmosphere in the high-speed rotating coupling. The inner strips in the pack necessarily become hotter than the outer strips. This means that the inner strips expand slightly and do not carry the same load as the outer strips. Therefore, the outer strips tend to carry a higher share of the load than they should, and eventually are likely to fail because of the differential stresses created. Since the FLEXXOR has no heat generated, the stress carried by the different disks remains essentially constant throughout, and all disks uniformly share the load.

FORCES AND MOMENTS GENERATED BY GEAR COUPLINGS

In a gear type coupling the torque is carried by high local forces between the gear teeth. In order to move the coupling sleeve with respect to the hub, it must slide on the teeth. In order to slide it must overcome the friction load created by the high force between the teeth to transmit the torque. If we take as an example a size 2 ½" gear coupling transmitting 351 HP per 1,000 RPM at a gear pitch diameter of 5", a tangential force of 8,850 lb is produced. Ignoring the effect of the tooth pressure angle, and assuming only 0.15 for a friction coefficient, then a force of 0.15 times 8,850 = 1,330 lb would be required to slide the coupling axially. This is a typical problem with gear couplings and results in a high transmission of thrust from one rotor to the other. This is commonly called coupling lock-up, and can result in thrust bearing failures. It often causes shutdown of the machines, because thrust bearing clearances are indicated by the detecting instrument as being too small. In many cases, it is considered to be advantageous to have some misalignment of a gear coupling, so that the surfaces are continually working and thereby tend to prevent this axial lock-up condition. Obviously misalignment will also cause wear of the coupling, so that this is not a desirable way to prevent lock-up.

In addition to the friction causing high thrust loading in a gear coupling, it also causes high moments on the shaft end, and transverse forces transmitted from one shaft to the other. This can be illustrated by studying the diagram below, showing the position of a gear coupling sleeve with offset misalignment.

Note that point A on the coupling must move axially to point B position in one half revolution. Therefore, it must wobble from A to B and back to A in each revolution. During this cycle, the gear teeth must slide on each other. At the same time, they are heavily loaded to transmit the torque. Therefore, the angular wobble requires a force proportional to the friction required to slide the teeth under load. Since point A moves to the right and point moves to the left as they turn from the plane in view, then the friction forces create a bending moment on the joint. If we assume that the friction load is uniform around the circle, then the bending moment would occur at 90 degrees from the plane of misalignment, and would produce reactive forces at the supporting shaft and the coupling shaft.

We can make an estimate of the moments created, and the reactive forces, by using as an example a 2 ½" gear coupling transmitting a torque of 351 HP per 1,000 RPM. This produces a tangential force of 8,850 lb at a gear pitch diameter of 5 inches. At an assumed friction coefficient of 0.15 this means that the axial force needed to move each sleeve relative to the hub would be 0.15 times 8,850 = 1,330 lb.

The moment that is generated by the axial wobble is the friction force between the teeth times the distance of the teeth from the wobble axis. By integrating this moment around the full circle, the moment is calculated to be Fd/π, where F is the friction force = 1,330 lb, and d is the pitch diameter = 5 inches. Then $M = 1{,}330 \times 5/\pi = 2{,}117$ in-lb. This is the moment at the gear teeth, which literally bends the shaft end supporting the gear hub.

The moment on the shaft end must be opposed by a reactive moment in the coupling sleeve. This moment must be created by a transverse force at the opposite end of the sleeve, where the gear teeth touch the adjoining shaft hub. This equals M/L where L is the length of the coupling between gear teeth. In the foregoing example the gears on the hubs are 8.62 inches apart. The transverse would then equal 2,117/8.62 = 245 lb. Note that this is far greater than the coupling weight, and could occur in any direction, depending on the plane of misalignment.

While the above calculations give an approximate idea of the forces generated by friction, the actual situation is much more complex. On the diagram it is obvious that the relative axial velocity of the teeth at point A and point B is zero, and that it is maximum at C. Since the friction coefficient is higher at zero velocity than when friction surfaces are sliding it is obvious that the friction coefficient carries around the circle. Also, the tooth pressure is not uniform around the circle. At point A and B, the tooth spaces are parallel, but at point C they are not. Therefore, the tooth pressure changes. Ideally, all the teeth are perfectly machined on both the male and female gear sets so that at point A and B where the relative sliding stops and changes direction, each tooth successively imparts its portion of moment to the shaft. If there were 36 teeth for instance, and we had accurate sensing equipment, we might sense 18 small moment impulses in each revolution as the teeth at point A and B (assuming they are 180 degrees apart) break from static to sliding. Most likely, however, two teeth carry more load so there is more friction at one set of teeth (180 degrees apart) than others. Then as these teeth break free at points A and B, there is a noticeably higher moment impulse which occurs once of twice a revolution (depending of the friction force between teeth varies with direction of sliding as it might as the result of a wear pattern).

Clearly, a much higher moment impulse is felt on the shafts. This reaction may excite a natural frequency in one of both rotors, or it may force enough movement of the shaft so as to trigger proximity of vibration probes. From this discussion it is obvious that misalignment of gear couplings generates forces considerably higher than the weight of the coupling, and that are very probably variable and cyclical in nature. They are also very probably higher than unbalance forces.

The energy input per revolution is proportional to the deflection or offset misalignment. This explains why gear couplings wear out, and why they are so sensitive to misalignment. It also explains why they cause vibration when misaligned, and why they transmit vibration from one shaft to the other.

As a help to understand this, interlace your fingers and bend one hand with respect to the other. The interlaced fingers must slide between each other, just as gear teeth must slide within the matching teeth on the coupling sleeve. Next, tighten the finger grip and note how it becomes more difficult to slide the fingers. Now, imagine how much force it must take if the force between fingers is increased to 8,850 lb, as illustrated above.

ENERGY LOSS

Since rubbing of gear teeth on each other generates heat and wear, this energy is lost and is a direct loss in power. Let us calculate the loss in a typical situation such as that for the preceding section where the force on each shaft was 245 lb. If we assume this coupling is operating at an offset misalignment of 0.020", which would be typical, then the work input would be $W = F \times 2\pi r = 245 \times 2\pi \times 0.020 = 30.78$ in-lb/revolution on each end for a total of 61.6 in-lb/revolution. If the speed is 8,000 rpm, then the power loss is $61.6/12 \times 8{,}000/33{,}000 = 1.244$ HP. This is only 0.04 percent of the power transmitted, but does cost money that is being dissipated to the atmosphere, and wears out the coupling teeth.

The coupling reaction forces must be supported by the bearings. In case of horizontal misalignment the reaction forces are vertical and the vertical load on one bearing is increased by 245 lb, while the vertical load is decreased by 245 lb for the other bearing. Presumably then the friction loss is decreased in one bearing and increased in the other, with virtually no net friction loss.

If the offset misalignment is vertical, as is typical, then the forces on the bearings are horizontal, and the 245 lb force adds friction loss to both bearing.

An approximate bearing diameter for this size coupling would be 3". A typical friction coefficient in a hydrodynamic bearing is 0.002. The friction loss in each bearing is then $245 \times 0.002 \times 3\pi/12 \times 8{,}000/33{,}000 = 0.0933$ HP. If we add the loss for one bearing to the coupling internal loss, then the total friction loss is $1.244 + 0.093 = 1.337$ HP. The electric power loss is $1.337 \times 0.745/0.92 = 1.083$ kW. If the machine runs 8,000 hours per year and electricity cost 6.5 cents per kWH, the yearly cost is $1.083 \times 8000 \times 0.065 = \$563.00$.

The foregoing calculations can be combined and simplified into a single equation for yearly cost of coupling friction power.

Yearly cost = 86.4 x HP x y/L
In the foregoing example HP = 351 x 8000/1000 = 2808 HP
Y = 0.020" offset misalignment
L = 8.62 length between coupling gear teeth
Yearly cost = 86.4 x 2808 x 0.020/8.62 = $563.00.

The following chart shows yearly cost of the friction loss in gear couplings plotted against offset misalignment for various horsepowers. For example, if a plant is using 20,000 HP at an average misalignment of 0.004 inches/inch, then the yearly friction power cost is $6,900.00. It does not take long for a frictionless coupling to pay for itself.

By eliminating friction, and by reducing misalignment forces it becomes easy to see how FLEXXORs virtually eliminate losses, as well as the wear that produces those losses.

Doesn't it seem a little ridiculous to pay for power that is used only to wear out couplings?

FORCES AND MOMENTS GENERATED BY THE FLEXXOR

Since a FLEXXOR uses flexible disks to accommodate offset misalignment, there is no friction involved, and the forces are approximately proportional to the amount of misalignment. Typically a size 200 FLEXXOR, transmitting 416 HP per 1,000 RPM, has an angular spring rate of 70 in-lb per degree. At a distance between shafts of 6.97 inches and typical offset deflection of 0.020 inches, the angular deflection would be 0.164 degrees, and the bending moment would be 0.164 x 70 = 11.5 in-lb. The transverse force would then be only 11.5/6.97 = 1.65 lb.

While the action of multiple strip flexible coupling is not exactly the same as that of a gear coupling, the friction between the disks also resists movement, and we have a bending force situation somewhat similar to that of a gear coupling. The FLEXXOR, with no friction, avoids this problem almost completely.

FLEXXORs have repeatedly demonstrated a reduction in vibration compared to the gear couplings that they have replaced. The foregoing explanation helps to show why this should occur, because there is no friction in the coupling element. It also shows why FLEXXORs have reduced vibration to a small fraction of that occurring with the previously used gear coupling on the same assembly.

Not only is the FLEXXOR without friction, but also its axial spring rate is much lower than that of other flexible element couplings, such as multiple strip couplings or couplings with single flexible disks. For example, a size 200 FLEXXOR has an axial spring rate of 228 lb/in of movement. Compared to this a typical single disk coupling such as the Yorkflex has a spring rate of 21,000 lb/in. Thus at a given relative axial movement between the shafts, the Yorkflex develops a thrust 92 times as high as that of the FLEXXOR. Other single disk couplings are similarly stiff.

The angular spring rate of the FLEXXOR is also quite low compared to that of other disk couplings, again because of the use of multiple disks rather than single disks for the flexing element.

The FLEXXOR disks are designed with holes in the disks so that the flexibility is much greater than that of a disk without holes. The torque is transmitted through a series of struts between the holes in the disk. This means that the flexibility is improved dramatically as shown above, and the torsional flexibility is also greater, as are the axial and angular flexibility. This means that the FLEXXOR can be designed with low torsional spring rates, and these low spring rates make it possible to prevent high frequency vibrations from being transmitted from one rotor to the other. The FLEXXOR design using a quill shaft makes it possible to reduce the torsional stiffness to almost any degree that is desirable to cause torsional vibration isolation between the two rotating elements.

HIGH MISALIGNMENT TOLERANCE

As a corollary to the very low spring rate achieved in the FLEXXOR, the FLEXXOR also has very high misalignment capabilities. Typically, the standard FLEXXOR will have up to four times as much misalignment tolerance as other couplings. The large diameter "M" series have even higher misalignment tolerances.

High misalignment tolerance also means that alignment of the shafts is less critical, and more easily done than it is with other couplings.

Oft times the critical misalignment tolerances for some couplings require many days work to align the machinery. The much higher misalignment tolerances for FLEXXORs shorten required alignment time to a mere fraction of that required for other couplings. Usually, much more than the price of the coupling is saved by the reduction in machinery down tome for alignment.

LIGHT WEIGHT

By very carefully designing all parts of the FLEXXOR so that the maximum use is made of the material, we have achieved much lighter weight coupling than other designs. This insures less effect of misalignment on unbalanced forces, less lowering of the critical speeds of the shaft on which they are mounted, and lower rotative inertia in the coupling. All of these factors tend to make for smoother running, longer lasting machinery.

REDUCING OVERHUNG WEIGHT

Not only is the FLEXXOR lighter than other couplings, but also the special shaft mountings which have been developed, and which are available for the FLEXXOR, permit shorter overhangs from the supporting bearings. It also helps to keep the critical speed of the rotors higher, with less amplification of unbalance forces, so that the result is a much smoother running system.

UNITIZED CONSTRUCTION

FLEXXORs are designed with unitized construction so that all elements are tightly fitted together and well centered. This is turn means that very high speeds are attainable, and particularly high speeds are possible with the new FLEXXOR design which eliminate the external clamping rings.

The flexing elements in a FLEXXOR are made from ultra high strength stainless steel, which helps to prevent corrosion, and are also specially treated to produce extremely high fatigue strengths. Therefore, the disks are designed with an ample factor of safety to allow for acceptable misalignments and for continuous high loads. The fatigue limits on these disks are so designed that under allowable limits of loading as specified in the design brochure the life of the elements should be virtually unlimited.

FATIGUE CONTROLLED AND DESIGN FAILURE POINTS

FLEXXORs are designed so that the maximum stresses occur in the disk elements, and if accidental over-torque occurs, the disk elements act as failure points. This often protects damage to a second rotor if one of the rotors seizes or has a sudden failure. In other couplings it is often the case that if one rotor should seize or be damaged by foreign material striking the blades, then the coupling will actually break a shaft and wreck not only the one rotor but both rotors. The FLEXXOR can be designed so that this almost never happens. The FLEXXOR acts essentially like a shear pin in an outboard motor propeller shaft, to protect the coupled rotor from damage. If the disks fail from overload the coupling sleeve cannot fly out, but it is safely contained.

Fatigue failure in a FLEXXOR is almost non-existent, but where there are cases that excessive misalignment occurs, then the many holes in the disk of the FLEXXOR act as multiple failure points. For example, in a typical FLEXXOR there are as man as 768 points of failure. This means that any fatigue failure is likely to occur very gradually starting at only a few points so that inspection of these failures is easily possible before a final or catastrophic failure occurs. This ease of inspection is important where couplings are very critical to the drive, for example, in a helicopter drive, and the ease of inspection and detection of these failure points becomes a very important feature.

FLEXXORs can be designed with redundant drive simply by putting pins through the holes in the disks of the adjacent member, or by jaws between sleeves and hubs. If the disk fails in fatigue then the pins going through the holes can drive the coupling temporarily. This can be important in some types of process industry applications.

FLEXXOR INTERCHANGEABILITY

The flexible elements of FLEXXORs are interchangeable. This becomes of great practical importance in the case of an accident, in which the flexible element is designed to be the most likely part of the coupling to fail. The elements can be removed and replaced using ordinary high-strength socket wrenches. This means that assembly and disassembly of the elements either for replacement, or for accessibility to bearings and seals is very convenient, and saves a great deal of time. In many cases shutdown time is extremely costly. The time taken for assembly and disassembly of a coupling or for its replacement becomes a critical factor, since it is both the starting point and the finishing point for any repair jobs that take place on coupled machinery.

ADAPTABLE TO MANY SHAFT MOUNTINGS

FLEXXORs are adaptable to many types of hub mountings. This has made it possible to replace many types of couplings in the field, where vibration problems and wear problems due to misalignment were causing rather frequent necessity for replacement of the couplings.

A typical mounting is a keyed shaft hub, either straight bore or tapered bore. FLEXXORs are easily adapted to keyed hub mountings, and where necessary can be mounted on reversed hub mountings as shown in the FLEXXOR brochure. They are also easily adapted to flanged shaft mounting. The rather large diameter of the bolt circle mounting used in the FLEXXOR permits it to be mounted on many difference diameters of shafts, and this makes it especially convenient to design FLEXXOR elements to replace couplings in almost any situation.

The keyed shaft hub has been the traditional mounting for coupling, but has some serious disadvantages, which have been overcome by newer designs available for mounting the FLEXXOR. The keyed shaft hub necessarily produces high stress concentrations in the hub mounting. This in turn required the shaft mounting to be a larger diameter than a mounting that does not have the high stress concentrations created by a key. The large diameter and the length required by the key also require that the length of the shaft overhang is greater than desirable. This necessarily moves the coupling mass farther away from the supporting bearing.

When a keyed shaft hub is mounted by the usual method of shrinking on the shaft, the stress level in the hub at the thin part where the key cuts into the hub is much higher than that in the rest of the hub. This means that this part of the hub stretches more than the rest of the circumference. For this reason, mounting a keyed hub on a shaft will cause eccentricity of the mounting and unbalance of the entire coupling, because the hub is eccentric to the shaft. While this may not be a large factor, it does tend to increase the vibration forces cause by the coupling being mounted eccentrically.

Another disadvantage of the shrunk and keyed shaft hub is simply the great amount of time required for removing the hub or remounting it. This invariably amounts to many hours of time, and increases the time out of operation for any two machines, because the coupling mounting or dismounting must always occur at the ends of the assembly or disassembly operation.

THE ANDERSON HUB MOUNT

Coupling Corporation recognized a long time ago the many disadvantages of the conventional types of keyed hubs, and developed the patented Anderson Hub Mount to overcome these disadvantages.

The Anderson Hub Joint is an important new breakthrough for mounting hubs on shafts. In its simplest form it can be thought of as two collars thread onto a shaft. The first collar is threaded onto the shaft on a thread that is slightly larger than the second thread, at the end of the shaft. One thread is left hand, and the other is right hand. The second thread, at the tip of the shaft is smaller than the first thread, so that the inner collar can slip over it, and be thread into the first larger diameter thread, even though the hand is opposite to that at the shaft tip.

The inner collar has a number of set screws arranged in a circle. The second collar or coupling hub is threaded onto the tip of the shaft. This hub is turned until a desired axial position is located.

Next, the first collar is turned back on its thread until it touches or comes very close to the coupling hub. The collar is adjusted until the set screw holes are in lines with the dimples in the coupling hub. The set screws are then tightened uniformly so as to attempt to push the collar and hub apart. This puts high pressure on the tapered faces of the threads, and creates enough frictional force on the thread, so that the hub can resist very high torques, far above the required rated torque on the coupling.

The threads are asymmetrical, with a slight slope on one side and a steep slope on the other. The collar pushes the hub up the slight slope, so as the center the hub on the shaft quite positively. In this sense the joint is similar to a tapered hub mount. However, the helical thread permits the hub to be adjusted to a desired axial position, which is important when installing a coupling between two shafts.

The taper on the threads is designed so that a high centering and friction force is generated, but it is easily backed off the removing the hub, when necessary.

An additional feature is the use of “Friction-eeze” in the thread joint. This specially developed compound creates a high friction coefficient, but at the same time prevents galling or threads corroding together after long time service.

The Anderson Hub Joint provided outstanding advantages in strength, safety, convenience, and ease of manufacture. There are no keys to cause stress concentration. This permits the use of a smaller shaft. With no keyway in the hub, it can be both shorter and lightly. The hub is perfectly centered on the shaft. There is no eccentricity caused by keyways.

No shrink fits are used. This eliminates the need for torches – often dangerous in a plant atmosphere.

No special tools are required for assembly or disassembly. Ordinary sockets keys are used for assembly and tightening the hubs. Axial adjustment is convenient and easily made. This saves time during coupling installation.

Thread machining is done on the same lathe on which the shaft is machined. All that is required is the specially shaped thread chasing tool. There is no need for expensive key seating. Diameter tolerances are much less severe than those required for shrunk or tapered hubs. Hub mounting is quick and easy. There is no hand fitting of keys to hubs or shaft.

QUILL SHAFT DESIGNS

The quill shaft with a single FLEXXOR shown in the brochure is another introduction by Coupling Corporation, which makes very small shaft seals possible and also very easy accessibility for changing shaft seals. This type of mounting has been used in thousands of refrigeration compressors, and has saved tremendous amounts of money by providing extremely small mechanical shaft seals, which have low rubbing velocity and smallest possible leakage areas. The type of shaft mounting has an outstanding record of success in refrigeration compressors and special gas compressors throughout the world. It has been copied in Europe, Japan, and Russia.

The quill shaft mounting in conjunction with the Anderson Hub Mount has the additional advantage of easy accessibility requiring short shutdown time for changing seals, and also permits easy longitudinal alignment adjustment.

The quill shaft design makes it possible to minimize transmission or torsional pulsations from one machine to another. It can be designed with a low enough spring rate to smooth out vibrations such as those produced by reciprocating engines or compressors, and has been very successful in many such applications.

FLEXXOR FEATURES

The many features of FLEXXORs make it easy to see why we can claim that it is the world's best flexible coupling.

To summarize:

> FLEXXORs are frictionless, therefore, friction generated transverse misalignment forces are negligible.

FLEXXORs are frictionless, therefore no heat is generated to create differential stresses that can cause premature failure.

FLEXXORs are frictionless; therefore there is no wear.

FLEXXORs are frictionless, therefore they need no lubrication.

FLEXXORs have no loose parts, and no backlash that can cause torsional impact stresses.

FLEXXORs are lightweight, and create minimum unbalance forces.

FLEXXORs are all steel, and adaptable to extremes in temperature.

FLEXXORs have very low axial spring rates and transmit almost no thrust forces from one shaft to the other.

FLEXXORs have extremely high axial, angular, and offset misalignment tolerances.

FLEXXORs are easily adapted to many kinds of shaft or flange mountings.

FLEXXORs are available for extremely high speeds.

FLEXXORs make it possible to design small shaft seals with minimum wear, rubbing speed, and vibration.

FLEXXOR elements are interchangeable.

FLEXXORs can be designed for shaft mounts requiring no shrink fits, and for easy and quick disassembly with only ordinary hand tools. This shortens down time for replacing bearings, seals, etc.

Special FLEXXORs can be furnished for extremely high misalignment capability.

FLEXXORs can be tuned to isolate torsional pulsations from coupled machines.

FLEXXORs can be designed for failure to protect coupled machinery from damage due to failure of a high-speed rotor.

FLEXXORs save energy. With no friction, low spring rates, and lightweight, the reduced power loss can often reduce power costs by more than $1000 per year.

FLEXXORs save enough time in assembly and alignment to pay for themselves in just one season.

Coupling Corporation can furnish 24-hour replacement service for FLEXXORs, if customer wishes. This can eliminate the need for stocking spares.

J. Hilbert Anderson, President
COUPLING CORPORATION OF AMERICA
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