OVERRUNNING CLUTCHES
APPLICATION MANUAL

RegalRexnord™

FORMSPRAG CLUTCH™

Formsprag Clutch and Stieber Clutch are manufacturers of the highest quality overrunning clutches in the marketplace. Formsprag is also a supplier to the commercial and military aircraft markets. The quality documentation systems and procedures necessary to support this level of quality products has facilitated Formsprag's certification for ISO-9001 in October 1993 and Stieber's ISO-9001 in October 1997.

Formsprag Clutch and Stieber Clutch have been designing, manufacturing and delivering dependable, long-lived, precision power transmission products for over 50 years, providing one of the broadest lines of overrunning clutches in the world.

Formsprag Clutch and Stieber Clutch overrunning clutches deliver thoroughly proven, dependable power transmission. Careful selection of highest quality materials, precision manufacturing by experienced craftsmen, conscientious assembly, and rigid adherence to detail guarantee a reliable, quality product.

Overrunning clutches handle torque loads up to 700,000 lb.ft. (949200 Nm). Other sizes and designs are available by special order.

Designs Feature:

- Instantaneous action, no backlash
- Infinitely changing wear points
- More torque in less space with sprags
- Full sprag and roller complements
- Formchrome® sprags
- "Free-action" retainer
- PCE positive continuous engagement
- Inner Race C/T "Centrifugal throwout" sprags
- Outer Race C/T "Centrifugal throwout" sprags

FIND IT FAST AT WWW.FORMSPRAG.COM

Use our online interactive eCatalog to find what you need just in a few clicks. Input your application requirements using many helpful aids to identify performance criteria and the right product will power its way to the top of the list.

- Exclusive "Competitor Interchange" feature lets you enter a competitor part number and find the specific replacement Formsprag model.
- Fill out the Application Data Form to get assistance in determining your selection.
- Submit an online RFQ to the distributor of your choice.
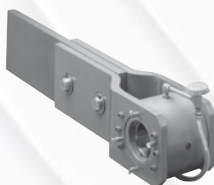
Wide range of styles and sizes to choose from...
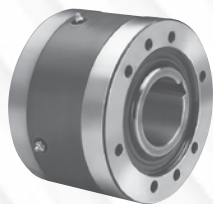
General and Special Purpose Clutches
- Overrunning
- Indexing
- Backstopping

Special Purpose Clutches
- Overrunning
- Backstopping

Holdback Clutches
- Backstopping
- Holdback

Clutch Couplings
- Overrunning

TABLE OF CONTENTS

PREFACE

All Formsprag clutches described in this manual are "overrunning" clutches, i.e., they will drive in one direction but overrun (free-wheel) in the other direction. The preparation of a manual such as this can be undertaken only by a company having a very broad experience in all aspects of the use and application of overrunning clutches. Formsprag Clutch is such a company and the information given in this manual is based on the accumulation of many years of experience in the design, construction and application of overrunning clutches. In physical appearance the Formsprag Clutch Catalog model overrunning clutches are all very much alike. However, different types of applications will involve differing load characteristics and will call for variations in the technical details of clutch construction. For this reason the clutches are divided into the three basic types of applications for which they are intended.

The three basic applications are:
- Overrunning
- Indexing
- Holdbacks or Backstops

These three categories are described in greater detail under those headings in this manual.

APPLICATIONS

Formsprag clutches can be found in every corner of the world with applications in commercial, aircraft and military equipment. Our clutches are available for three basic types of applications:

■ Overrunning   ■ Indexing   ■ Backstopping

MARINE LIFTING
Bi-Directional Backstopping

Form-Lock (FL) bi-directional clutches are used on winch drives of davits on ships to provide optional manual drive to the normal drive system to smoothly raise and lower the life boats. The Form-Lock is a mechanical clutch that can be driven in both directions while automatically providing a holding brake function when the driving torque is stopped.

Additional uses include the smooth raising and lowering of gun barrels in artillery, and other loads being raised and lowered smoothly with ball screws. Contact Formsprag for information and assistance.

Torque Range: 100 to 4,800 lb. ft.

Model FL
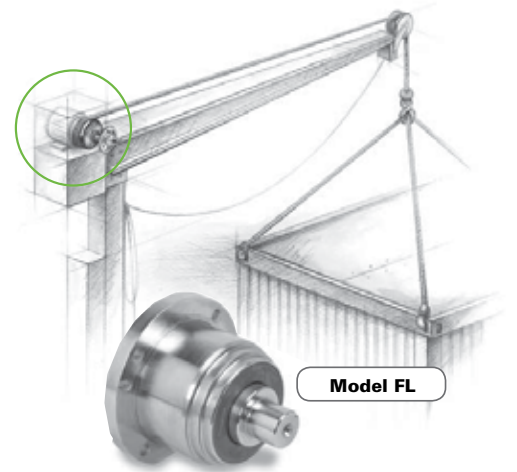
TEXTILE PROCESSING
Overrunning

FSO overrunning clutches allow the feed drive to power the rollers at a low speed when threading the leading edge of a new roll of fabric. When the fabric is fully threaded and ready to be pulled at production speeds, the FSO un-clutches the feed drive allowing those same rollers to be rotated at faster speeds without back driving the lower speed feed drive. The clutch automatically disconnects the lower speed feed drive during higher speed operation and prevents possible damage to the feed drive if back driven at the higher speeds.

Bore Range: 0.5 to 7.0 in.
Torque Range: 275 to 27,000 lb. ft.

Model FSO
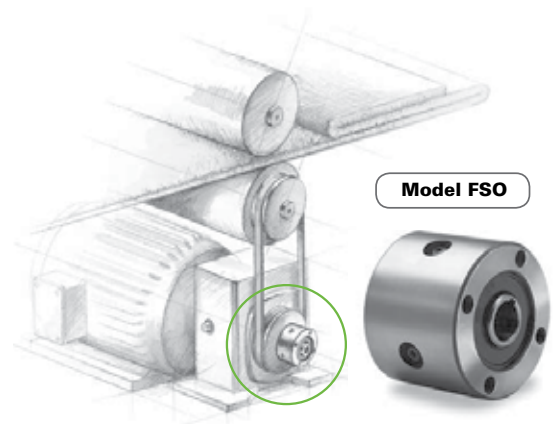
FOOD CONVEYOR INDEXING DRIVE
Indexing

HPI clutches are mounted on the head shaft and translates the reciprocating motion from the crank mechanism into incremental motion in one direction, advancing the conveyor a uniform distance with each drive stroke and then overrunning in the opposite direction on the return stroke. These clutches are available with FDA approved food grade greases and a variety of surface coatings required for wash down duty.

Bore Range: 0.5 to 7.0 in.
Torque Range: 275 to 27,000 lb. ft.

Model HPI
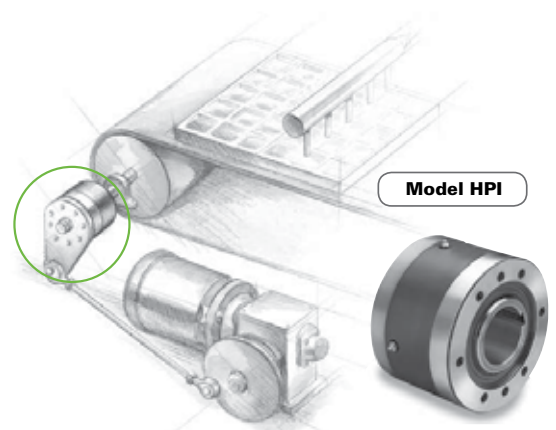
APPLICATIONS
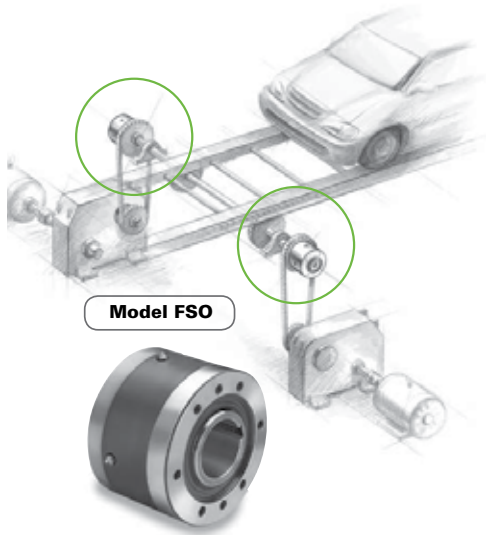
ASSEMBLY LINE CONVEYOR
Overrunning

FSO overrunning clutch is mounted on the head shaft of a conveyor system to provide instantaneous switching from the primary drive to a stand-by drive to reduce down time. Mounting the FSO clutch between each drive and the conveyor provides the automatic clutching and de-clutching whe switching from one drive to another without any utility or input to the clutch. Using the FSO clutch on dual drive applications reduces installation and operating costs by eliminating controls or actuation mechanisms.

Bore Range: 0.5 to 7.0 in.
Torque Range: 275 to 38,000 lb. ft.

Model FSO

DUAL DRIVE
Overrunning

FW clutch coupling is used on in-line mounting of dual drive systems of fans and pumps to provide a smooth transfer of power from one drive/power system to another (electric motor to steam turbine and gear reducer). The model FW clutch coupling mounted between the output of a steam turbine reducer and the fan allows this fan to be initially started with an electric motor without back driving the steam turbine. When steam becomes available the FW allows the steam turbine drive to come up to speed (over any amount of time) and automatically transfer power. When the speeds are matched, the starting electric motor can be turned off to save its utility cost.

Bore Range: 0.6 to 7.0 in.
Torque Range: 107 to 27,000 lb. ft.

Model FW

TWO SPEED CONVEYOR DRIVE
Overrunning

FWW clutch coupling is used on inline mounting of creep drive conveyor systems for low speed belt inspections or cold weather low speed weekend creep operation. The FWW clutch coupling is mounted between the main drive and the low speed creep drive and provides for smooth clutching when switching from one drive to the other. Cost savings are realized at both the initial installation because no controls are required to perform the clutching and declutching function; and during operation because no set-up or wear adjustments are required so that regular maintenance costs are reduced.

Bore Range: 0.6 to 4.4 in.
**Torque Range: 276 to 7,000 lb. ft.**

Model FWW
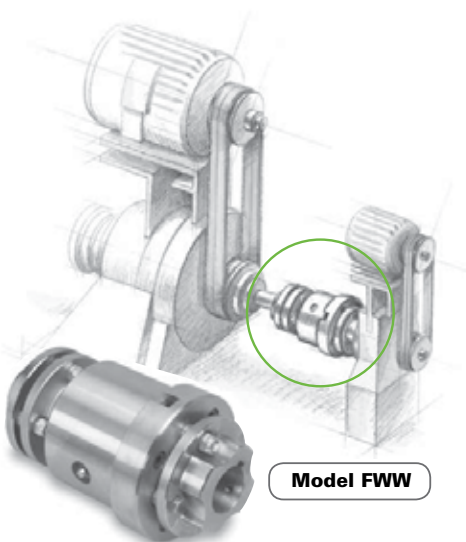
APPLICATIONS

PUNCH PRESS FEED DRIVE
Backstopping

FSR clutches are used as a backstopping clutch on a press feed drive by mounting the clutch on one end of the drive roller and grounding the outer race with a torque arm. This installation allows the drive roller to rotate in only the desired direction. Press feed rollers for applications using steel and other stiff materials pulling from a coil need to resist the material's natural tendency to spring backwards or re-coil.

Bore Range: 0.3 to 2.19 in.
Torque Range: 40 to 1,800 lb. ft

Model FSR
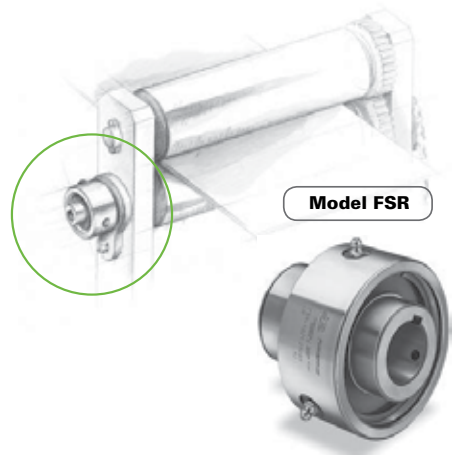
FAN DRIVE
Backstopping

FB clutches can be mounted directly on the shaft of the fan drive and grounded with a torque arm; the unit provides a backstop function for the fan drive. When the fan is turned off, wind flow from other fans in the system or the environment can cause the fan to rotate opposite the operating direction and the FB can eliminate this problem thereby improving the life of the fan drive systems.

Bore Range: 0.3 to 1.57 in.
Torque Range: 2.6 to 284 lb. ft

Model FB
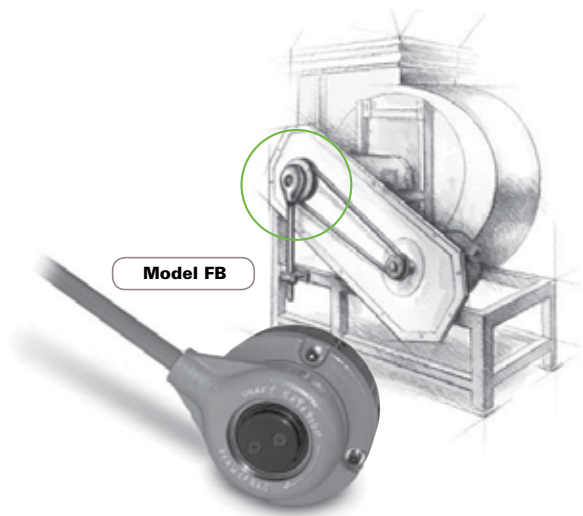
COOLING TOWER FAN DRIVE
Backstopping

HSB (High Speed Backstop) clutches are used on cooling tower fan drives to prevent rotation in the opposite direction from the normal driving direction. When the fan drive is off, wind pressure from other fans or ambient breezes can apply sufficient force to rotate fan blades in the opposite direction. The HSB clutch allows the fan to freely rotate in the desired driving direction while preventing any rotation in the opposite direction.

Bore Range: 0.43 to 7.0 in.
Torque Range: 300 to 27,000 lb. ft.

Model HSB
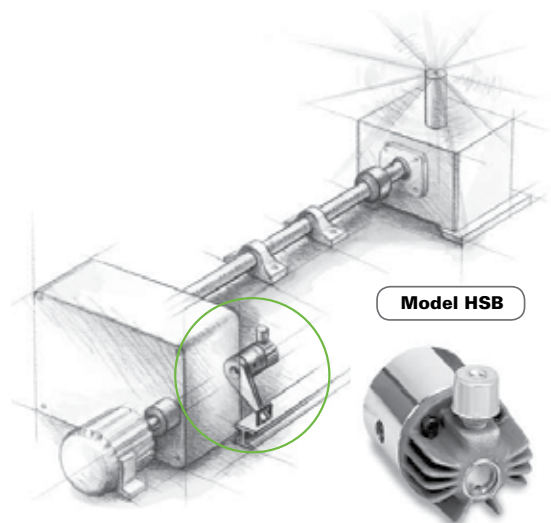
APPLICATIONS
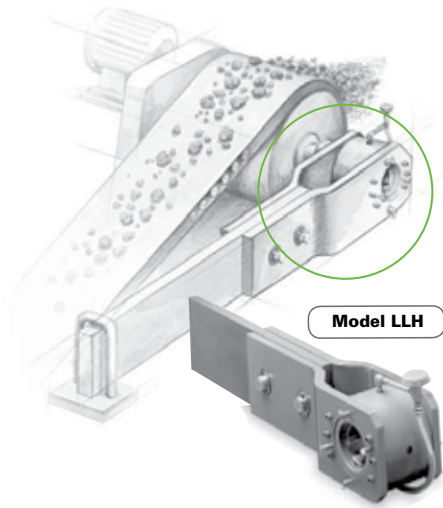
BULK HANDLING CONVEYOR DRIVE
Backstopping

LLH holdback clutches are mounted on the head shaft of the conveyor, typically on the side opposite the electric motor and gear reducer.

The LLH clutch allows the headshaft to freely rotate in the driving direction while preventing any rotation in the opposite direction.

Bore Range: 1.87 to 20.0 in.
Torque Range: 4,000 to 700,000 lb. ft

Model LLH
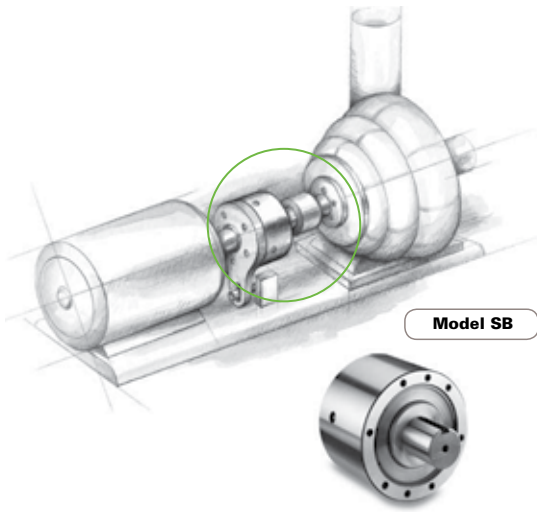
PUMP BACKSTOPPING
Backstopping

SB Clutches are used on fluid pumping equipment to provide a backstopping function preventing anyreverse rotation. In a simple electric motor/pump drive system, the SB clutch is mounted on the electric motor shaft with the output shaft of the SB clutch coupled to the pump's input shaft. The SB clutch with a torque arm attached to the outer race allows the pump to rotate only in the driving direction.

Bore Range: 1.87 to 20.0 in.
Torque Range: 4,000 to 700,000 lb. ft.

Model SB
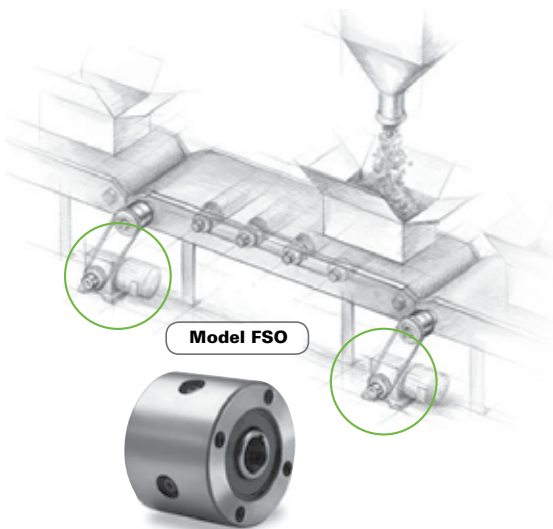
CONVEYOR DRIVE
Overrunning

FSO clutches are used on multi-speed, one direction only conveyors for non-destructive accommodation of speed differentials in powered rollers and belt drive systems. The FSO clutch allows the slower speed zone to over-speed during the transitional period without back driving the slower drive system.

Bore Range: 0.5 to 7.0 in.
Torque Range: 275 to 27,000 lb. ft.

Model FSO

PRINCIPLES OF DESIGN & OPERATION

There is nothing new or unique about the sprag principle as defined in Webster's Dictionary. What is new and unique is the advancement that Formsprag Clutch has made to that principle in the application of the sprag principle to overrunning clutches (Figure 1). The very name, "Formsprag" is descriptive of the product. Formsprag Clutch refined the sprag wedging principle and gave the sprag its own distinctive shape or "form" (Figure 2). Thus the Formed Sprag that Formsprag Clutch uses in its clutches gave birth to the company name "Formsprag Clutch".

The form of the sprags is precisely calculated and shaped to give the most efficient possible sprag action for an overrunning clutch.

Figure 1

Figure 2

Clutch Design

Stripped of such items as gears, splines, bearings, oil seals and other attachments a Formsprag overrunning clutch consists basically of a cylindrical inner race, a cylindrical outer race surrounding it, with an annular space left between the two races, and a full complement of accurately-formed sprags filling the annular space between the two races.

Each sprag is essentially a strut fulfilling the dictionary definition placed between the races in such a manner that it prevents rotation in one direction just as the wagoner's sprag prevented backward motion on his vehicle. In the overrunning clutch the sprag permits free and unimpeded rotation in one direction and drives in the opposite direction.

If one race of the overrunning clutch is securely affixed to a grounded member and the other race is free to be turned, the free race will turn freely in one direction but will be locked to ground in the opposite direction (just as the wagon was locked to ground by the sprag used by our ancestors). If the grounded race is connected to a moving mechanism—rather than to ground—then the overrunning clutch will free-wheel and transmit no torque to the moving mechanism in one direction but will transmit torque to the moving mechanism in the opposite direction of rotation.

In theory each sprag is simply the line of action between the contact point of the sprag against the inner race of the clutch, and the contact point of the sprag against the outer race of the clutch (Figure 3).
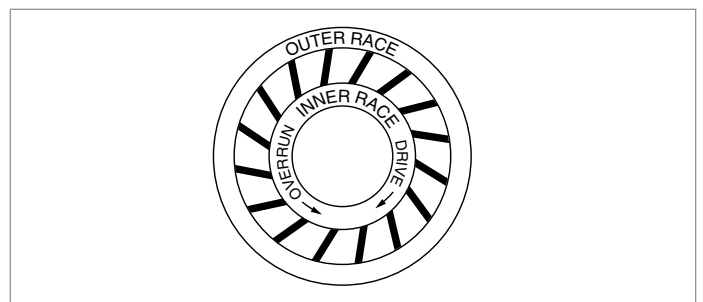
Figure 3

The line of action of the sprag is inclined at an angle from a true radial line in such a manner that it permits free relative motion of the races with respect to one another in one direction, and will wedge and transmit torque from one race to the other in the opposite direction of rotation (Figure 4). Either race can be the driving member and either race can be the overrunning member. The angle between the line of action of the sprag and a true radial line passing through the inner race contact point represents a gripping angle (G.A.) of the sprag (Figure 5).

The gripping angle is one of the most important aspects of sprag design. If the tangent of the gripping angle exceeds the coefficient of friction of the metals involved, the clutch will not drive.

Figure 4   **Figure 5**

PRINCIPLES OF DESIGN & OPERATION

A triangle (Figure 6A) composed of the line of action (hypotenuse), an extension of the radial line passing through the inner race contact point (cotangent), and a line from the outer race contact point normal to the extension of the inner race radial line (→A) represents the force vector -A- for a sprag. The tangent of the gripping angle is the load carrying component and the cotangent (→B) of the gripping angle represents the separating force or pressure against the race.

If the gripping angle were to be increased (Figure 6B) and the separating force (B) or pressure against the race left constant, the load carrying component (→A) would be increased proportionately. Conversely, if the load carrying component (A) were left constant, an increase in gripping angle (Figure 6C) would result in a decrease in the pressure (B) against the races.
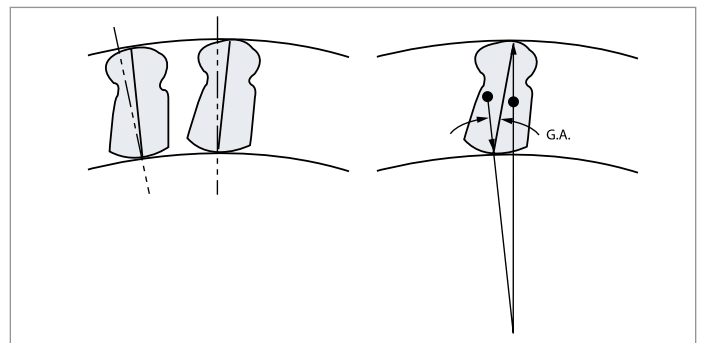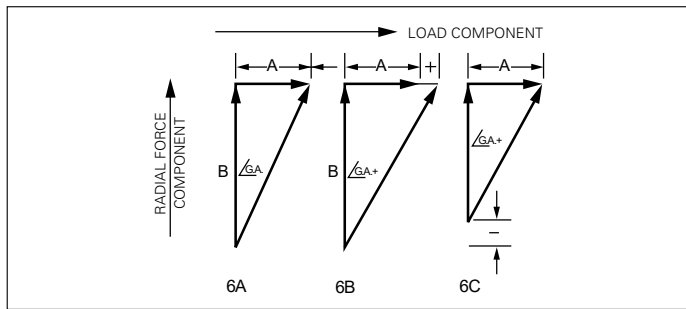

Figure 6

An inherent characteristic of a Formsprag Clutch sprag is the increase in gripping angle as the sprag rotates about its own center under the application of a torque load. The horizontal and vertical displacement of the centers of the inner and outer cam radii produces a change in the sprag geometry as the sprag is rotated about its own center.

At rest, during overrunning, and at the instant of the initial torque engagement, the sprag is inclined to a position that results in contact between the inner and outer cam surfaces of the sprag at points near one end of those cam surfaces. This results in a sprag geometry producing a gripping angle as shown as G.A.1 (Figure 7).
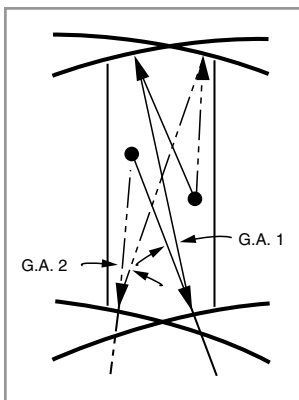

Figure 7

As torque is applied, the sprag is obliged to rotate about its own center in response to the torque load and within the limitation of the elasticity of the materials.

At full torque the sprag assumes the position where its cam surfaces are now contacting the races at points near the opposite ends of the cam surfaces. This change in contact point results in sprag geometry that produces a gripping angle such as that shown as G.A.2 in the figure.

As readily shown in Figure 7, G.A.2 is much larger that G.A.1. Because of this characteristic increase in gripping angle, the sprag geometry (force diagram produces a larger load carrying component for the same separating force as the sprag is caused to rotate about its own center. Thus at the point of initial engagement, the sprag has a very low gripping angle (and very low tangent of gripping angle) to insure instantaneous engagement with no possibility of slip. Yet as the sprag is subjected to load, the load carrying ability increases as is shown by the increase in the tangent of the steadily increasing gripping angle.

The torque carrying capacity of the Formsprag overrunning clutch is calculated from the Hertzian equations for compressive stress. The Hertzian equations for compressive stress cover the conditions of a cylinder in a cylinder (Figure 8A), a cylinder on a flat surface (Figure 8B), and a cylinder on a cylinder (Figure 8C).
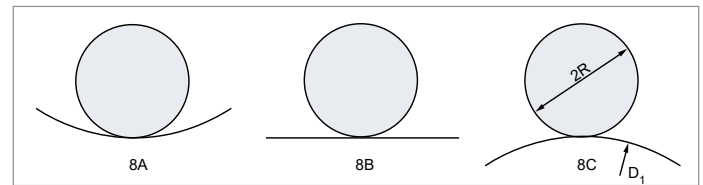
Figure 8

The Hertzian equation for a cylinder on a cylinder represents the contact between the inner race cam surface of the sprag and the inner race of the clutch. The Hertzian equation for a cylinder in a cylinder represents the contact between the sprag outer race cam radius and the outer race of the clutch. Since the non-conforming surface condition of a cylinder pressing on a cylinder results in the greatest compressive stress condition, it is this quation that is used in calculating torque carrying capacities of the Formsprag overrunning clutches.

The Hertzian equation for a cylinder on a cylinder is:

$$Sc = .591 \sqrt{PE \frac{D_1 + 2R}{D_1 (2R)}}$$

Sc equals the compressive stress. P equals the separating force per linear inch of sprag against inner race. E is the modulus of elasticity (30,000,000 PSI). D1 is the diameter of the inner race and R is the radius of the inner race cam of the sprag.

Brinelling or permanent deformation of the mating surfaces will occur in the presence of a Hertz stress in excess of 650,000 PSI.

In calculating the capacities of Formsprag clutches, Formsprag Clutch uses a Hertz stress of 450,000 PSI.

As shown in the equation, the stress varies as the square of the pressure. Since the square of 650,000 PSI is twice the square of 450,000 PSI, Formsprag's use of 450,000 PSI as a permissible Hertz stress gives the Formsprag clutch an automatic 100% safety factor on the calculations.

DESIGN ADVANTAGES

A sprag type of overrunning clutch delivers more torque for any given size than can be delivered by a roller type of overrunning clutch or a ratchet. The ratchet (Figure 9) is obviously limited in capacity because of the contact between the ratchet and pawl and a single tooth of the ratchet. In order to keep the stresses on the ratchet-pawl and the tooth to a reasonable level, the ratchet must be very large as compared to the size of either a roller clutch or a sprag type of overrunning clutch. The ratchet is also limited by the number of the teeth as compared to the infinite divisions available with a roller clutch or a sprag clutch.

It could never achieve the versatility of the Formsprag clutch which is infinitely variable in both rate and degree of index. The roller clutch (Figure 10) can deliver the same amount of torque in a much smaller envelope than a ratchet—yet it is still much larger than a sprag type of overrunning clutch. The roller clutch must be larger than a sprag type of clutch because the ramps are on one of the races rather than being a part of the contour of the sprag. The space taken by the ramps and the cage or other energizing means for the rollers is space which would be devoted entirely to sprags in a sprag type of overrunning clutch.

As a result, Formsprag clutches (Figure 11) can transmit a proportionately larger torque for any given clutch size than a roller clutch can transmit. Although the rollers of a roller clutch may engage in an infinite number of points on one race (thus giving infinite divisions as does a sprag clutch), the rollers engage always at the same point on the ramp of the other race. This continual engagement at the same point will cause an indentation which in time can result in complete loss of the wedging action and a failure to transmit torque when called upon to do so.

The Formsprag clutch engages in an infinite number of points on both races and, thus, distributes the wear over the entire circumference of both races—as well as giving an infinite number of engaging points.

In other types of overrunning clutches, wear is repeatedly imposed on certain fixed points, such as the single tooth of the ratchet or the same spot on each ramp of the roller clutch. In the Formsprag clutch the sprags engage both races at constantly and infinitely changing points of contact. As a result of this, as long as the Formsprag clutch is operating, it operates precisely because the contour of each sprag allows it to rotate to compensate for sprag wear.
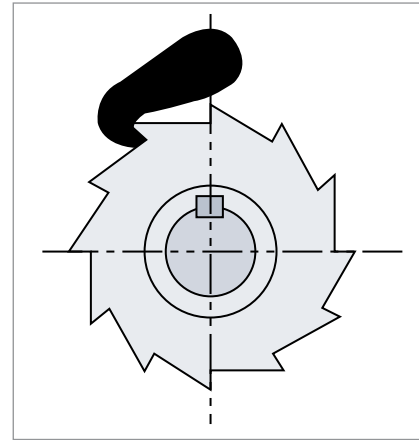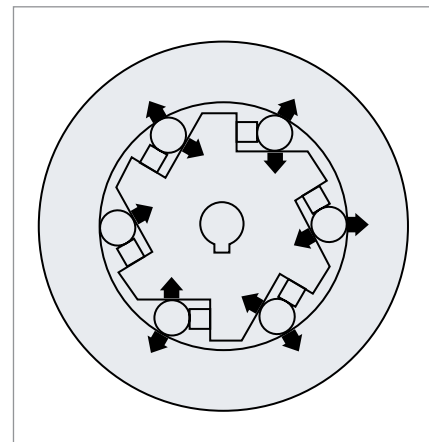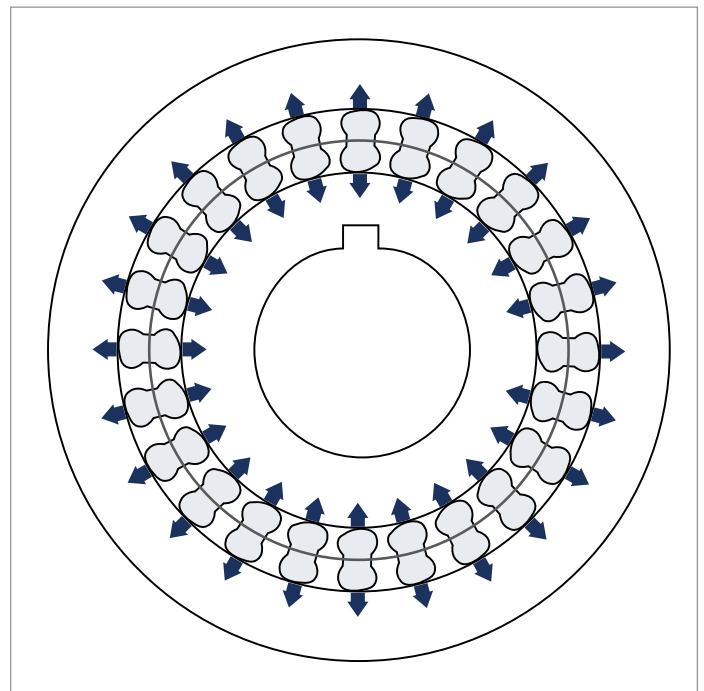

Figure 9

Figure 10

Figure 11

DESIGN ADVANTAGES

Not only does the Formsprag clutch have an infinite number of contact points on the races for each sprag but each Formsprag clutch also carries a full complement of load-transmitting sprags or wedges. This means that the load is being transmitted by the maximum ractical number of sprags which spreads the load more completely over the clutch and, as a result, allows the clutch to deliver far more torque capacity for its size than any other type of overrunning mechanism.

Each sprag in the Formsprag overrunning clutch is independently energized so that it is always in contact with both races at all times. Because of the fact that the sprags are always in contact with both races, there is never any relative motion required between the sprags and the races in order to transmit torque. The Formsprag overrunning clutch engages instantly because the sprags are always in contact and will disengage instantly. Because the Formsprag clutch engages instantly, there is never any backlash between the races when torque is transmitted. There will be a certain amount of torsional windup due to the elasticity of all materials but as long as the load applied remains constant, the torsional windup will remain constant and can be canceled out during the initial setup of a mechanism.

Each sprag in every Formsprag overrunning clutch is independently energized. The sprags are energized by springs that act upon each end of each sprag. Formsprag Clutch has developed several different types of energizing springs, such as contracting springs, expanding springs or torsional type springs (Figure 12). In each overrunning clutch the type of energizing spring used will reflect Formsprag Clutch's broad experience in the design and application of overrunning clutches in the choice of a method of energizing best suited for the particular design of clutch.

In all cases, whether the spring is an expanding spring, a contracting spring, or a torsional spring, the spring design energizes each sprag individually without transfer of motion or effect from one sprag to the next.

The cam surface of each sprag in a Formsprag overrunning clutch is actually a section of a cylinder having a diameter far greater than the annular space between the inner and outer races (Figure 13). This results in a contact surface for the working radii of the sprags far greater than is possible with any other type of overrunning clutch. This increased contact surface also results in lower stresses on both the sprags and the races with the result of greater resistance against the possibility of brinelling in the presence of maximum torque loadings.
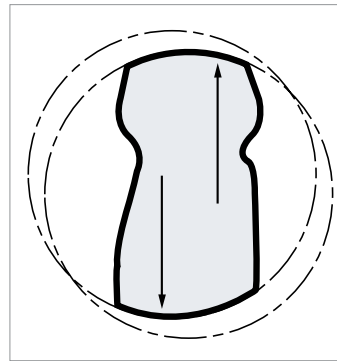
Figure 13

The portions of the cylinders that form the cam surfaces of the sprag have their centers displaced from one another in both the horizontal and vertical planes. The displacement of these centers results in the change in gripping angle previously described and also results in a remarkable tolerance for wear.

As the sprags begin to show wear as the result of long periods of overrunning, they will stand a little bit straighter with respect to the radial line. The offset relationship between the centers of the cam surfaces of the sprags results in a sprag having a greater dimension across the "load corner" than across the "overrunning corner" of the sprag. Thus as the sprag does show wear, it simply stands a little straighter and finds a new portion of its cam surface which still continues to fill the annular space between the two races.

The energizing springs, of course, will keep the sprags in contact with both races at all times.
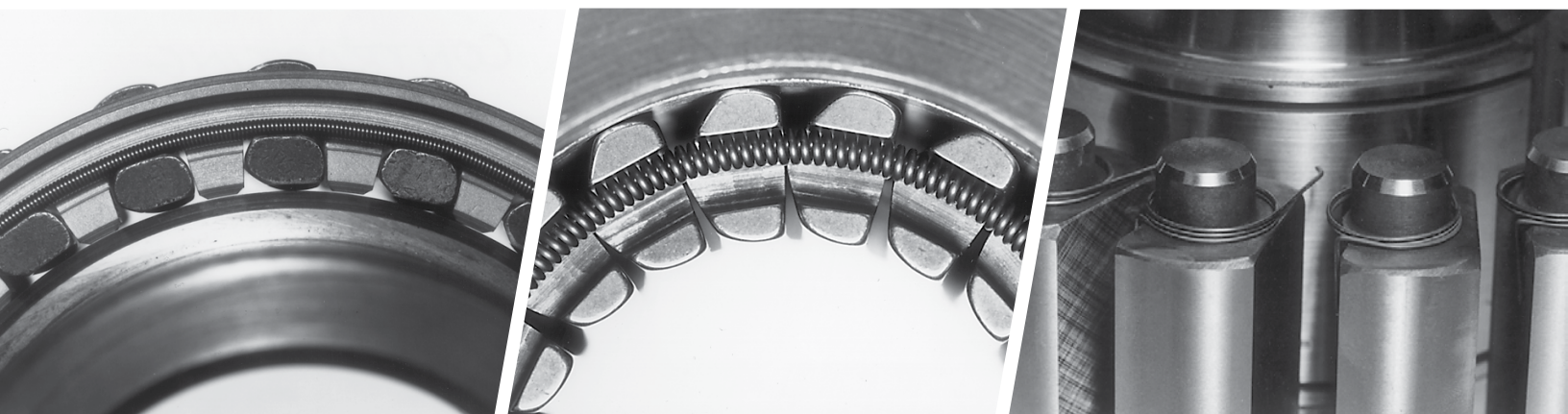
| Contracting | Expanding | Torsional |

Figure 12

DESIGN ADVANTAGES

Formsprag Clutch makes sprags in many different cross sections (Figure 14). The D-shape cross section (Figure 14A) is used in clutch designs that do not require a Sprag retainer such as the model FS-04s. The other Sprag cross sections (Figures 14B, 14C and 14D) are used with a Sprag Retainer. These have been developed to meet every conceivable type of overrunning, indexing, and backstopping (holdback) application specifications.
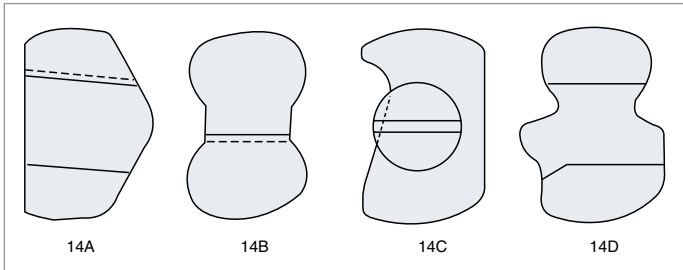
Figure 14

To bring the sprag clutch to the current state of the art, Formsprag Clutch recognized the advantages of using some type of sprag retainer as far back as 1954. Formsprag Clutch's first approach was to utilize a double cage arrangement based on the assumption that the double cage would phase all sprags into or out of engagement in unison. This approach was pursued and double cage retainer was developed, prototyped, and patented by Formsprag Clutch. Because of the preliminary work with this initial design, it was learned that it was necessary to give sprags freedom within the annular space to assure that all sprags were operative at all times. The current design offered by Formsprag and Stieber apply this principle, with the exception of the Stieber DC.

The Stieber DC "double cage" retainer design (figure 15) all sprags are obliged to move in unison to ensure simultaneous engagement and lock up while maintaining sufficient clearance within the cages to allow for eccentrics between the races. Thus in the presence of run out which occurs even in the most precise mechanisms positive lockup is achieved with simultaneous sprag engagement.

In addition, it was learned that in order to control sprag wear and hence prolong clutch life, it was desirable to have the sprag pivot about a point as close to the outer race as possible. This early program led to the current Formsprag Clutch "free action" retainer.

In the Formsprag Clutch "free action" retainer (Figure 16) all sprags are permitted to have free and independent action. During overrunning this allows each sprag independently to adapt itself to any variations in annular space caused by runout or by foreign matter which may inadvertently get inside the clutch.

Since each sprag operates independently, it cannot transfer the effects of variations from one sprag to the next. With all sprags in uniform engagement at all times, the load is evenly distributed. The free action principle also distributes wear evenly for a minimum of wear on all components.
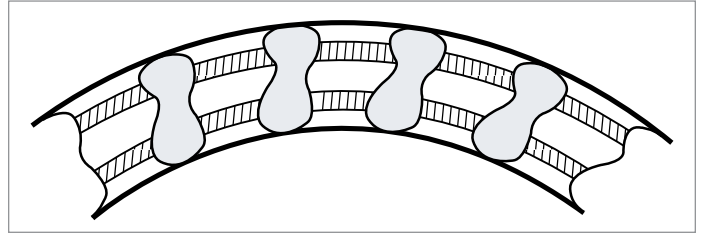
Figure 15
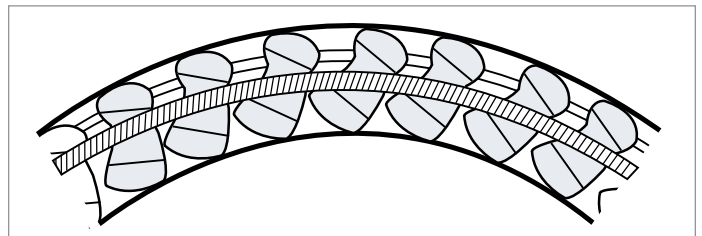
Figure 16

PCE Positive Continuous Engagement Retainer Assembly

The Formsprag PCE (Positive Continuous Engagement) design prevents sprag rollover from momentary torque overload, yet does not interfere with normal retainer engagement or overrunning. It possesses all the avantages of Formsprag Free-Action operation.

The patented PCE retainer was originally developed to meet the demands of high performance aircraft and helicopter applications. The PCE design provides reliable performance, even under extremes of torsional vibration and severe transient overload. PCE retainers are now available for a wide range of applications.

FORMCHROME SPRAGS

Wear is one of the unwanted design characteristics built into any moving mechanism, and the problem is to alleviate this condition as much as possible. Wear is a particularly important factor in the life of an overrunning sprag clutch, because the sprag geometry must be preserved if clutch life is to be prolonged and proper performance secured. The sprags used in Formsprag clutches have wear tolerance built into them as a feature of the Formsprag Clutch design.

The inherent tolerance for wear can be increased through the chromallizing process used on "Formchrome" sprags. The chromallizing process diffuses chromium into the surface of the high carbon steel sprags to form chromium carbides. Chromallizing is the first and only method that allows Formsprag Clutch to achieve high "super hard" sprags at a practical and economical level.

The base metal used in Formchrome sprags is SAE-52100 steel. The chromallizing of this high carbon content steel results in a chromium-carbide surface that has extreme hardness (Figure 17). Sprags are heat treated after the chromallizing so that the surface hardness of approximately 75 Rockwell C (1500 Vickers) is backed up by a hard steel core of 61-63 Rc. Thus, the sprags not only have unusual wear qualities, resistance to abrasion and corrosion, but they also have unusual strength characteristics throughout their entire cross section.

Chromallizing is not a plating process, but it is an alloying of two metals. Instead of depositing chromium on the surface of the metal, Formchrome is alloying chromium with the base metal to form a chromium carbide surface layer. The presence of this hard chromium-carbide alloy in the sprag surface enhances abrasion resistance, which is an important factor when Formsprag clutches are overrunning.

Chromium diffused into the base metal forms a chromium-carbide alloy with the carbon steel base and thereby becomes integrally fused. The surface alloy formed falls into the general group of high chromium steels.

In applications subjected to torsional vibrations, strong energizing is needed to resist the effects of the vibrations. In some applications, strong sprag energizing is required to separate a viscous-oil film at below freezing temperatures.

Stronger energizing is required under these conditions so that sprags will properly engage the races. The hard Formchrome surface permits the use of strong energizing springs with minimum wear. Also, the Formchrome sprags have the durability to withstand high spring energizing pressure during overrunning in warm months, or in the winter after the clutch has warmed up and the oil is thinner or less viscous.
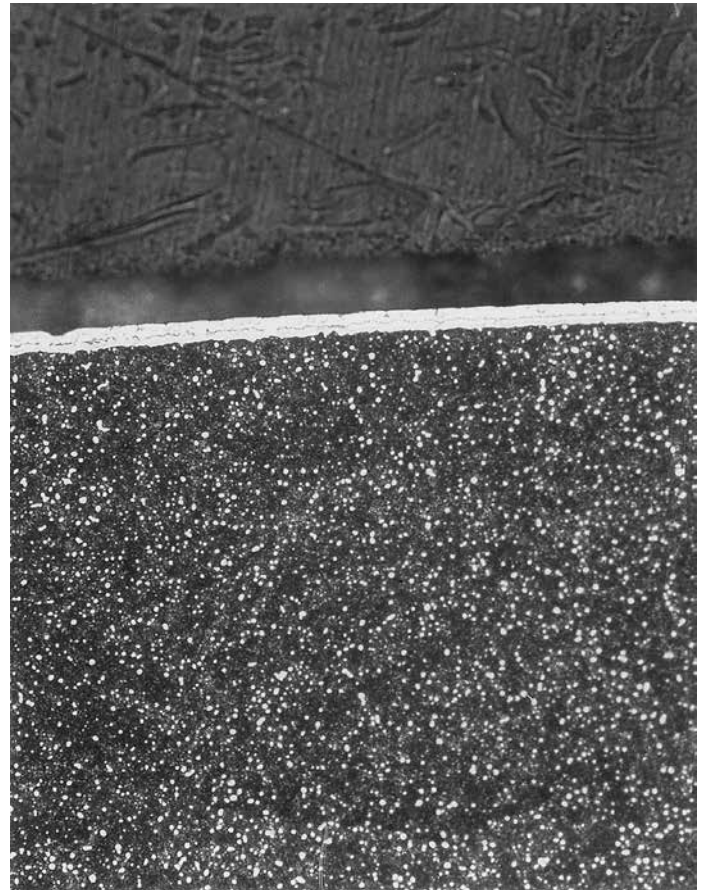
Figure 17

Micro-photo (500 x) showing chromium-carbide wear surface (1500 Vickers 75 Rc) of Formchrome sprags. Chromium is diffused into high-carbon alloy steel sprag (62Rc)

OTHER TYPES OF CLUTCH DESIGNS

Roller
Figure 18

Ratchet & Pawl
Figure 19

Ratchet & Pawl Clutch
Figure 20

Roller Clutch

The roller clutch (Figure 18) like Formsprag Clutch, also employs a wedging principle. However, the wedging action is obtained by causing a roller to wedge between a ramp and either the inner or outer race. While the roller may engage at an infinite number of points on the race, it always engages at the same point on the ramp, eventually causing an indentation which prevents the wedging action. In the roller clutch, each ramp requires a relatively large segment of the race circumference, reducing to a comparative few the number of rollers which can be used, compared to the number of sprags in the same diameter. Formsprag clutches can therefore transmit a proportionately larger torque for any given clutch diameter, again making a more compact, more efficient installation (Figure 20).

Ratchet and Pawl

The ratchet and pawl (Figure 19) is the oldest and most common device for indexing and backstopping. It is severely limited by the pitch of the teeth, and by centrifugal forces acting on the pawl. To reduce motion, tooth pitch must be reduced; this makes a weaker tooth, with resultant low torque capacity. The effect of smaller tooth pitch can be obtained by using multiple pawls, but this makes a more complex, cumbersome installation. It can never achieve the versatility of the Formsprag Clutch, which is infinitely variable in both rate and degree of index. Also, the full torque-carrying capacity of the complete complement of sprags is always operating, regardless of the degree or rate of index, whereas the ratchet and pawl device imposes the full torque load on one tooth for each pawl. One Formsprag clutch will cover the full range of many different sizes of ratchet and pawl assemblies, and in a much smaller assembly (Figure 20).
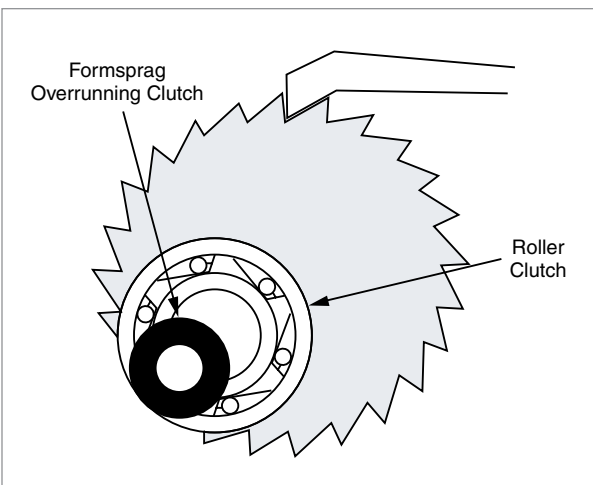
OTHER TYPES OF CLUTCH DESIGNS

Outer Race | Sprag | Double Cage Retainer | Ribbon Energizing Spring | Inner Race
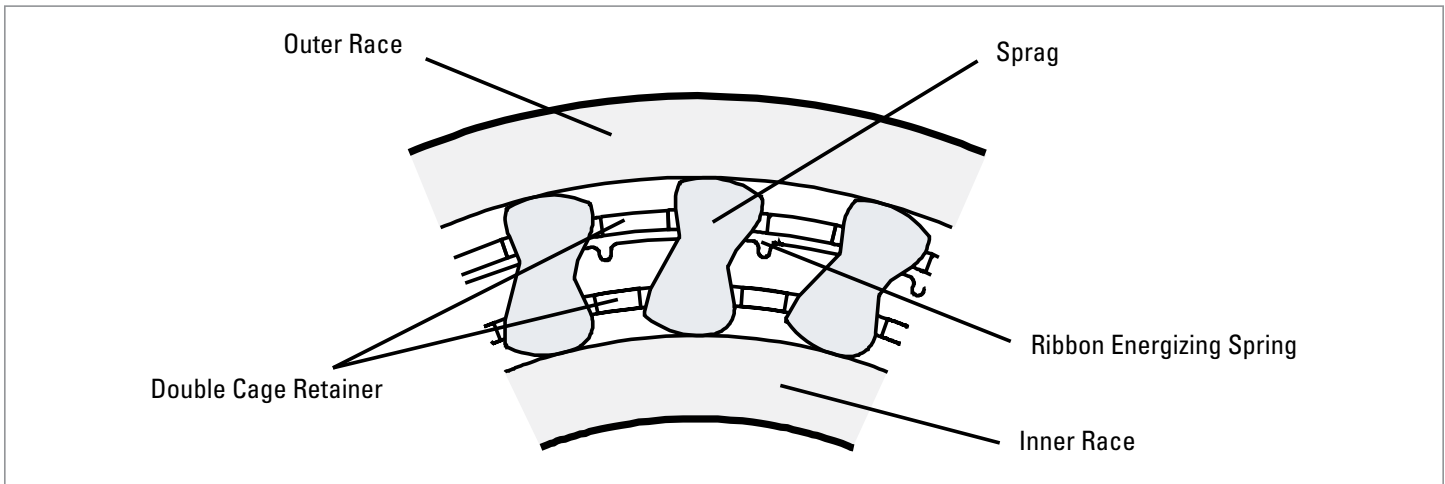
DC Sprag Retainer Assembly

DC (double cage) Sprag retainer assemblies have been used in industrial markets for over 20 years. The initial designs were dictated by the demands of the automotive industry for which they were originally developed. Since that time, many unique designs for the industrial market have been developed. The basic design includes the following components:

Sprags

The enhanced design of the DC Sprag results in giving the clutches more flexibility in lubrication and mounting than many other designs. On the surface the latest generation of DC clutches looks much the same as previous generations but there are major technical improvements incorporated.

- The DC Sprag gripping angle has been developed to provide a more stable initial engagement so that the sprags will not slip during the transition between overrunning and driving modes. This results in a robust design that reduces possible Sprag poping under extreme operating conditions.

- The DC Sprag design is compatible with all current lubricants used in power transmission equipment including those containing EP (extreme pressure) additives.

- The improved Sprag cam geometry has been designed to allow the DC clutches to operate with looser concentricity tolerances. The allowable TIR value has been increased by 33% over original designs.

- The cross sectional thickness of the DC Sprag has been increased to provide improved Sprag fatigue life.

- DC Sprag can be provided with "Formchrome" to extend the wear life in the most demanding high speed or continuously overrunning applications.

Double Cage Retainer

The double cage retainer was developed to provide full phasing through the use of two cage retainers to synchronize the movement of the full Sprag complement. The outer cage controls the sprags near the point of contact with the outer race and the inner cage controls the sprags near the point of contact with the inner race. The advantages are:

- Full phasing of all the sprags, assuring the same design engagement angle.

- A robust design, providing easier handling and reducing problems when being assembled into the races.

- Longer wear life, through heat treatment/hardened cages.

Ribbon Energizing

The ribbon spring has been designed to provide constant and independent Sprag energizing over a wide range of operating conditions.

- The energizing is designed to work with all types of lubrication including EP additives.

CLUTCH BEARINGS

Formsprag Clutch ball bearing clutches are able to carry both radial and thrust loads. Often it is necessary to check the radial loading of the bearings for an application where the clutch is subjected to radial loads imposed by drive chains, gears, sprockets or V-belts. The radial loads imposed by high tension, multi-V belts are particularly high.

The load that can be applied to a ball bearing clutch is dependent upon the bearings used in the clutch and the recommended bearing load rating as specified by the bearing manufacturer. Table 1 gives the Maximum Permissible Load (lbs) for radial and thrust conditions for Formsprag Clutch ball bearing clutches sizes 300 through 1027. These loads are based upon a calculated L-10 bearing life of 10,000 hours (50,000 hrs. avg. bearing life). Higher loads are possible at lower speeds.

Condition #1 (A) is the Maximum Permissible Load (lbs) for radial loads centered between the bearings.

Condition #2 (B) is the Maximum Permissible Load (lbs) radially applied above the end face of the clutch.

Condition #3 (C) is the Maximum Permissible Load (lbs) radially applied which can be offset or overhung from the end of the clutch.

Example: Determine the Maximum Permissible Load (C) that can be radially applied to a stub shaft adapter 10 inches from the end of a FSO-700 clutch.

Using the formula:

$$\text{Load (C)} = \frac{(A)\ (L)}{2\ (d + D + L)}$$

$$\text{Load (C)} = \frac{2520\ \times\ 3.060}{2\ (10 + .925 + 3.060)}$$

Load (C) =   276 lbs

L-10 bearing lives for loads and speeds other than those listed in Table 1 for each clutch may be calculated by using the following formula;

$$(L\text{-}10)_0 = \left(\frac{A}{X_0}\right)^3 \times \left(\frac{N}{N_0}\right) \times 10,000$$

where:

(L-10) = the new L-10 life in hrs.

X_0 = new load in lbs.

A = load from Table 1 in lbs. (note: B and C can be used in place of A for Conditions #2 and #3 as required).

N = overrunning (O/R) speed from Table 1.

N_0 = new O/R speed.

Condition #1
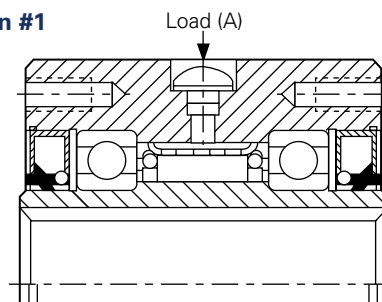
Load (A)

Condition #1 has force or Load (A) in center of clutch between the two ball bearings. (See Table 1)

Condition #2
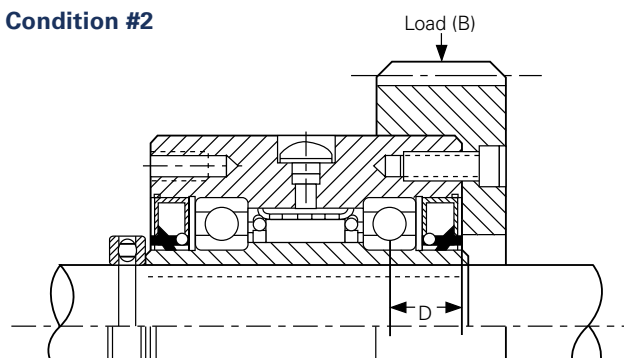
Load (B)

Condition #2 has force or Load (B) exerted on clutch bearings at end of clutch. Distance "D" is the distance from the centerline of the ball bearing nearest the load to end of clutch. (See Table 1)

Condition #3
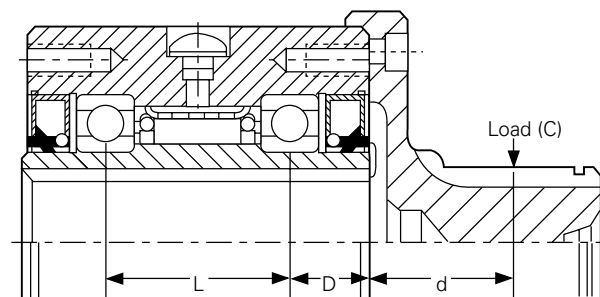
Load (C)

Condition #3 has force or load applied "d" distance from face of the clutch. (To be determined by individual application.) To calculate the Maximum Permissible Load (C) a distance "d" from the clutch face, use the following equation:

$$\text{Load (C)} = \frac{(A)\ (L)}{2\ (d + D + L)}$$

(Refer to Table 1 for values of A, D and L)

CLUTCH BEARINGS

FS and
FSR Series
Figure 25
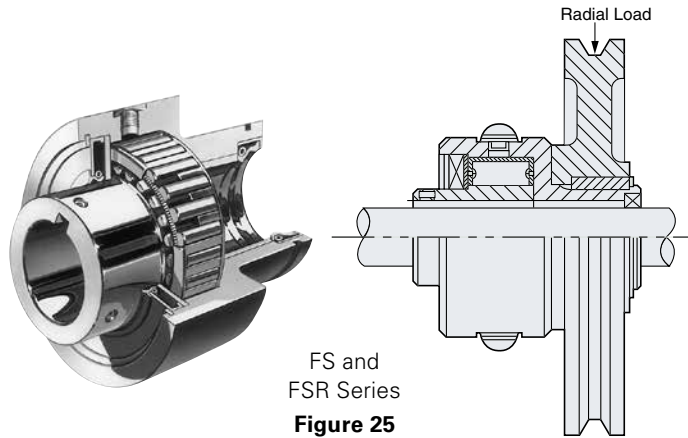
Radial Load

Example: Determine the maximum permissible load that can be radially applied between the bearings of a FSO-750 with an overrunning speed of 600 RPM that will result in a L-10 bearing life of 10,000 hours.

Since the load is applied between the bearings the value (A) for Condition #1 is used for this calculation. Also, because the bearing life is 10,000 hours, the new L-10 remains at 10,000 hrs.

Using the bearing life formula:

$$(L\text{-}10)_0 = \left(\frac{A}{X_0}\right)^3 \times \left(\frac{N}{N_0}\right) \times 10,000$$

Substituting values into the equation:

$$10,000 = \left(\frac{2040}{X_0}\right)^3 \times \left(\frac{1800}{600_0}\right) \times 10,000$$

$$X_0{}^3 = \frac{2040^3}{600} \times \frac{1800}{10,000} \times 10,000$$

$$X_0 = \sqrt[3]{2040^3 \times 3 \times 1}$$

$$X_0 = 2942 \text{ lbs}$$

Answer: The new maximum permissible radial load that can be applied is 2942 lbs.

The clutch thrust capacity at Max. O/R speed given in Table 1 is the **Maximum permissible load** applied in an axial direction to the end of the clutch. The clutch thrust capacity listed in Table 1 is without any radial load applied to the clutch. For applications that have both thrust and radial loads consult Formsprag Clutch engineering.

Formsprag Clutch sleeve bearing clutches, models FS-02 through FSR-16, are equipped with oil-impregnated bronze bearings (Figure 25). The bearings are designed to provide proper support for radial loads imposed on the clutch hubs, however, they are not designed to accept axial loads.

Table 2 gives the radial load capacity for each sleeve bearing model. The bearing capacity shown is rated at the maximum overrunning (O/R) speed of the inner race for each clutch model.

Higher radial loads are possible at lower speeds. In such cases please consult Formsprag Clutch engineering.

Table 1

Clutch No.	Maximum Permissible Load		D	L	Thrust Cap.@ Max, O/R Speed	Max O/R Speed RPM
	Cond. #1 (A) lb	Cond. #1 (B) lb	in	in	lb	
FSO-300	452	165	.516	1.381	226	3600
FSO-400	450	151	.675	1.376	225	3600
FSO-500	755	275	.745	1.990	378	3000
FSO-600	1248	437	.836	1.950	624	2400
FSO-700	2520	968	.925	3.060	1260	2000
FSO-750	2040	755	1.247	3.550	1020	1800
FSO-800	2555	944	1.251	3.542	1277	1500
FSO-900	3030	1134	1.257	3.740	1515	1350
FSO-1027	500	51	1.446	3.355	0	1100
HPO-720	2196	837	.925	2.96	1098	3000
HPO-750	1795	663	1.247	3.53	897	2600
HPO-800	2273	840	1.251	3.55	1136	2100
HPO-900	2712	1014	1.257	3.73	1356	1850
HPO-1027	639	230	1.446	3.72	319	1500

Table 2

| Clutch No.	Bore (Ref.)	Radial Load Capacity (lbs) RPM	Max O/R Speed RPM
FS-02	.250	12	3450
FS-04	.375	15	2800
FS-04	.500	15	2800
FS-05	.625	30	1800
FSR-3	.375	40	1950
FSR-3	.500	40	1950
FSR-5	.500	45	1950
FSR-5	.625	45	1950
FSR-6	.750	70	1950
FSR-8	.875	110	1650
FSR-8	1.000	110	1650
FSR-10	1.125	130	1250
FSR-10	1.250	130	1250
FSR-12	1.375	190	1150
FSR-12	1.500	190	1150
FSR-14	1.625	250	950
FSR-14	1.750	250	950
FSR-16	1.875	260	950
FSR-16	2.000	260	950

TORSIONAL WINDUP

An unworn, or undamaged Formsprag Clutch will never slip in the driving direction. The sprags are always held in contact with both races, and do not have to move in order to drive. Because of this, the assumption of drive is instantaneous. This is of particular importance in indexing applications where instantaneous response is essential for accurate indexing.

Although the Formsprag Clutch will not slip when load is applied, there will be a certain amount of torsional windup due to the elasticity of the parts. This windup is always directly proportional to the load, and is a constant for any given load. Since the windup is a constant, it may easily be cancelled out in the initial setup. The following curves show the torsional windup for standard catalog overrunning clutches under varying torque load conditions. Curves of similar slope are applicable to all sizes of Formsprag clutches.

FS series windup curves

FS-02 Windup Curve

FS-04 Windup Curve
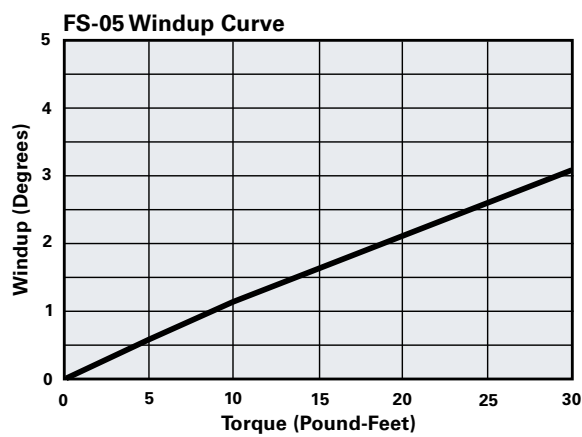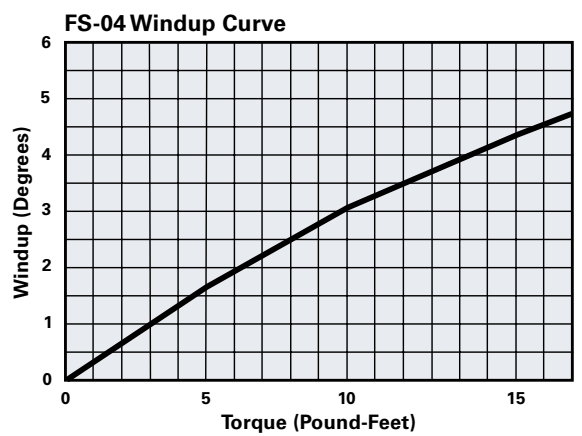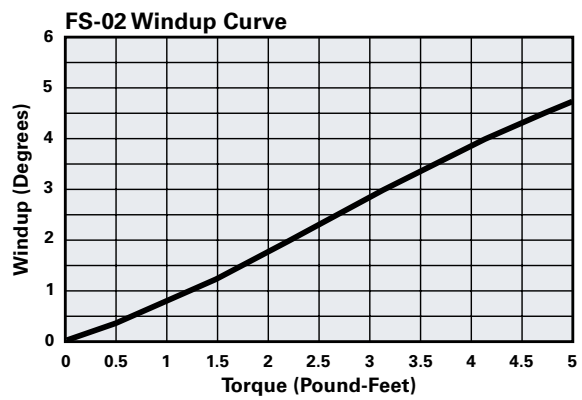
FS-05 Windup Curve

TORSIONAL WINDUP

FSR series windup curves

FSR-3 Windup Curve

FSR-10 Windup Curve

FSR-5 Windup Curve

FSR-12 Windup Curve

FSR-6 Windup Curve

FSR-14 Windup Curve

FSR-8 Windup Curve
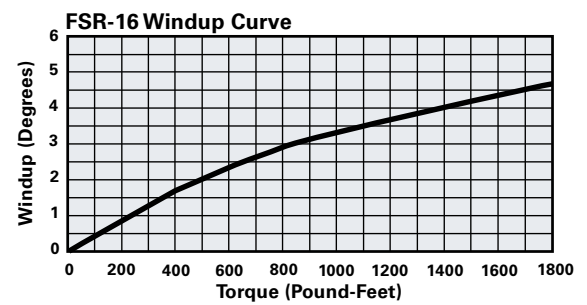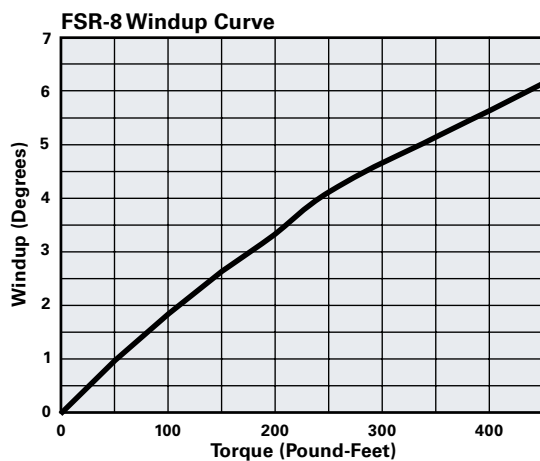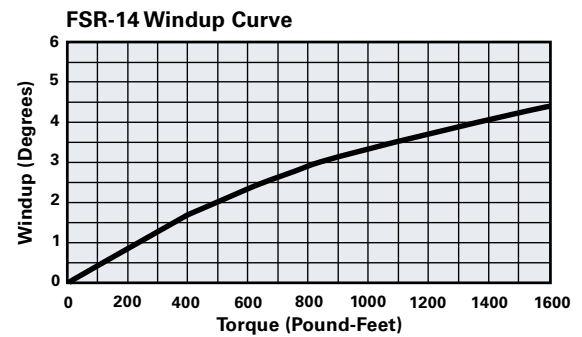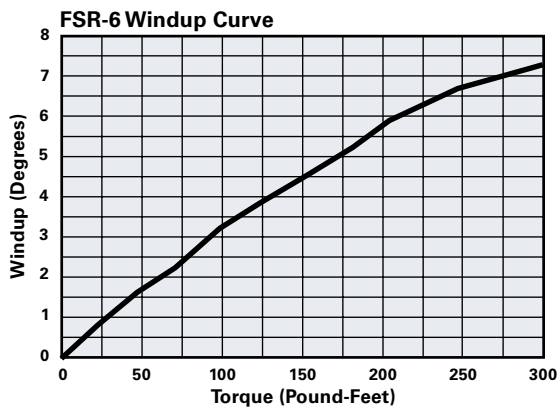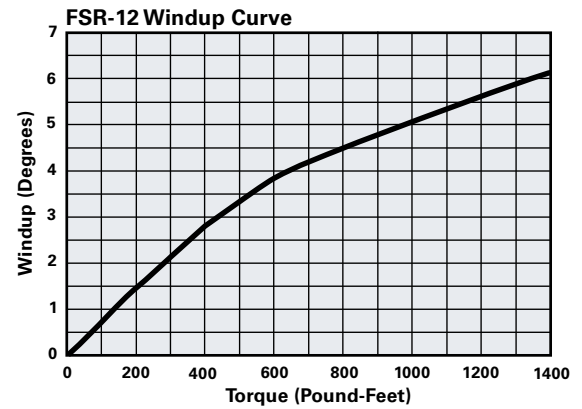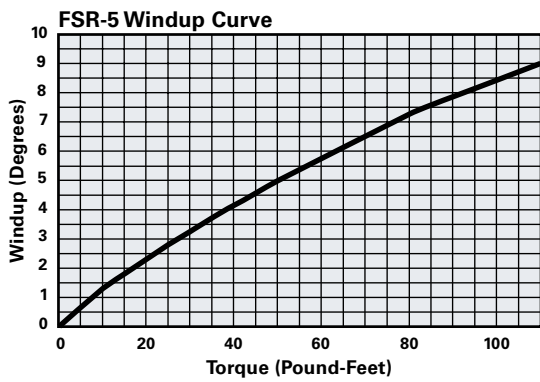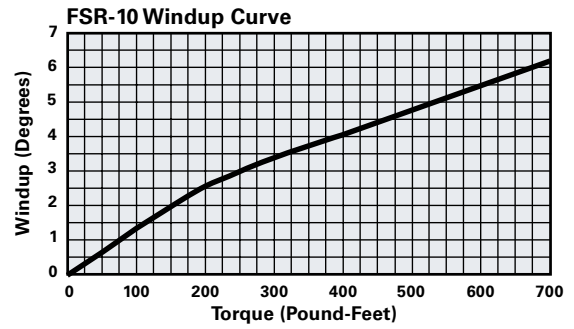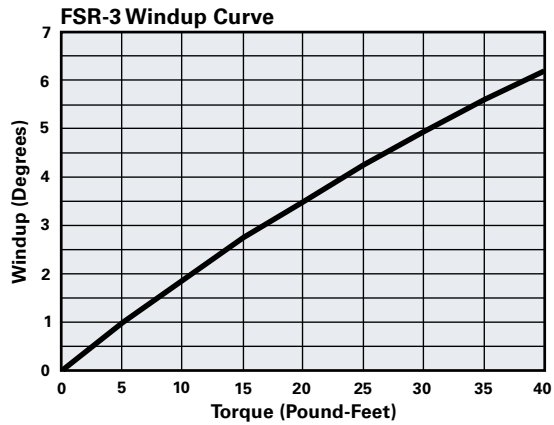
FSR-16 Windup Curve

TORSIONAL WINDUP

FSO/HPI series windup curves

Series-300 Windup Curve

Series-700 Windup Curve

Series-400 Windup Curve

Series-750 Windup Curve

Series-500 Windup Curve

Series-800 Windup Curve

Series-600 Windup Curve

Series-900 Windup Curve
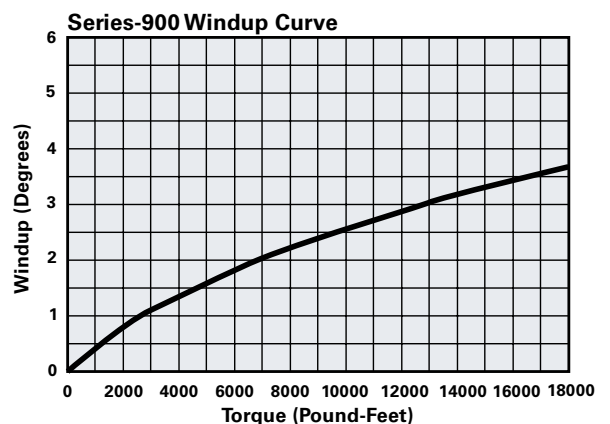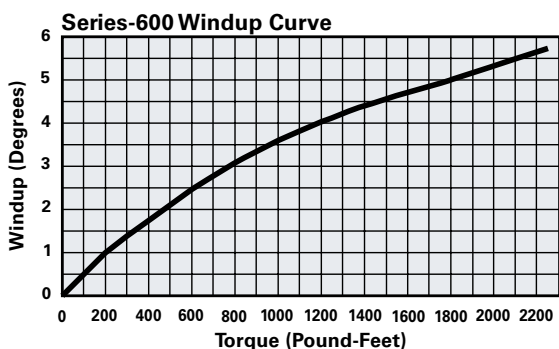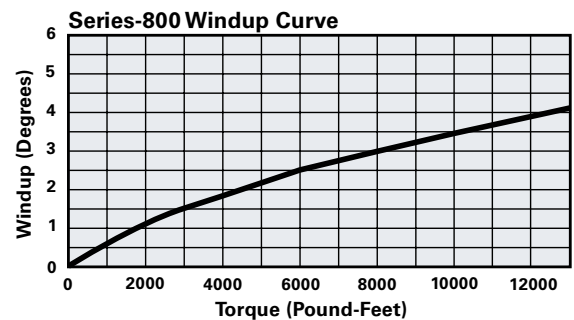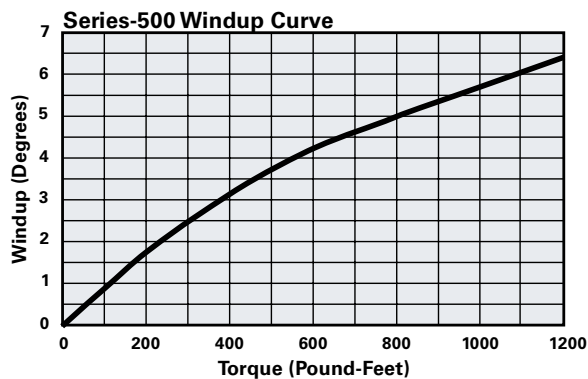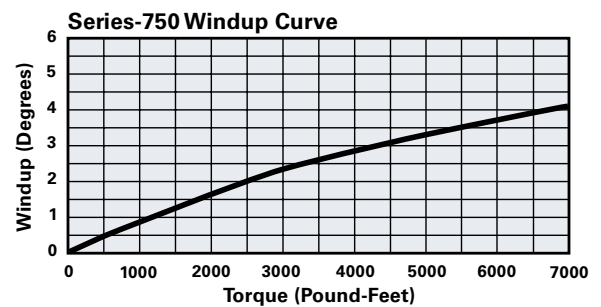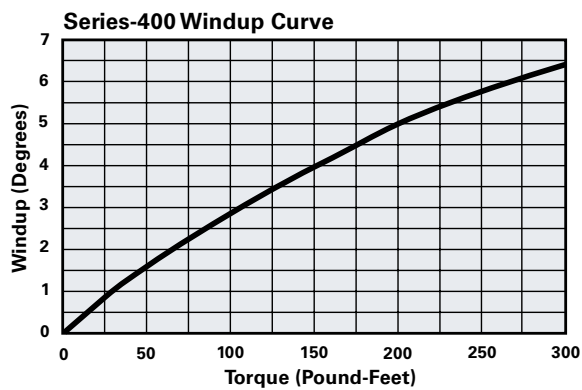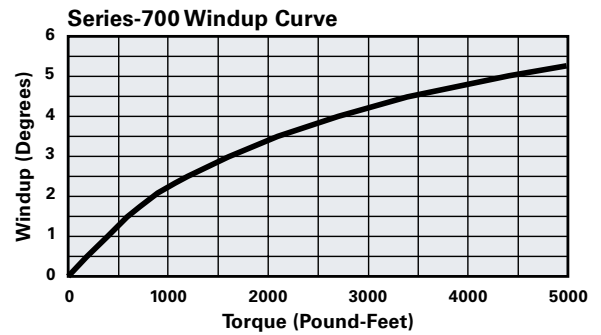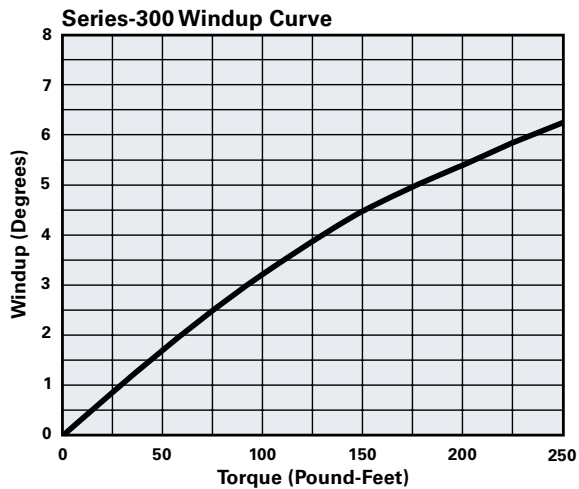
OVERRUNNING APPLICATIONS

Definition

An overrunning application is one in which neither race is permanently grounded. At various times during the operation cycle, both races will be rotating (Figure 26).
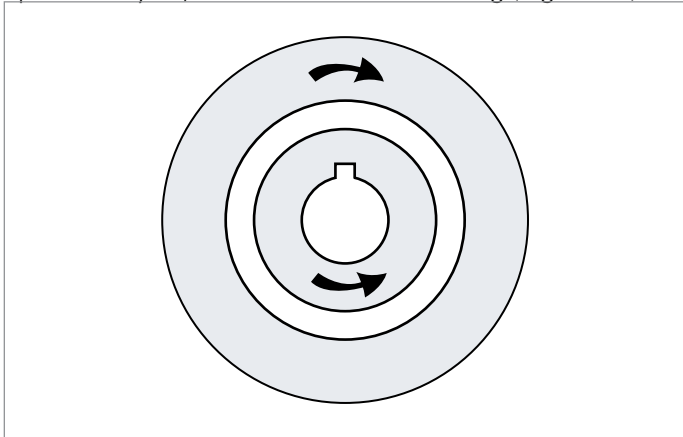
Figure 26

This class of application is typified by standby and compound drives. For example, a steam turbine and a standby electric motor may be connected to a single driven shaft through overrunning clutches. The shaft can then be driven by either the turbine or the motor or both, with no further modification of the installation. The turbine drive clutch automatically engages when the turbine starts to drive, but automatically overruns when the load is transferred to the electric motor.

Service Factors

The torque capacity shown for all standard catalog model Formsprag Clutch overrunning clutches is based on a steady state load gradually applied and without shock or pulsation. When applying the clutch to overrunning applications, the torque should first be established on the basis of the torque absorbed by the driven mechanism if this information is known. If not known, the torque can be determined from the standard torque equation—

$$T = \frac{HP \times 5250}{RPM}$$

T = torque, lb. ft.

HP = horsepower at the clutch location

RPM = revolutions per min.

This equation gives the torque at the clutch under a steady load condition at the particular speed and horsepower used in the equation. Since the equation does not take into consideration the type of load or method of load application, a service factor should be applied to the result in order to get the design torque which will be used in making the clutch selection.

In overrunning applications the service factors may vary from 1 to 6 depending upon the nature of the application and the type of loading.

1.0	Steady load, gradually applied—no shock
1.5	Steady load, applied through chain or gears (minor shock)
1.5	Pulsating loads such as fans, blowers, pumps, conveyors, etc.
2.0 to 3.0	Critical applications such as hoists, or personnel safety
3.0 to 4.0	Machine tools—arbitrary for long machine tool life
5.0 to 6.0	High torque motors, heavy shock applications such as jogging duty

When torsional or linear vibration is present, use a FSO series clutch and increase the service factor at least 50% (multiply by 1.5). For severe vibration, a greater service factor increase is necessary. To conform with coupling manufacturer's recommendation, use a minimum service factor of 1.5 on all Clutch Couplings.

The use of an internal combustion engine with an overrunning clutch drive will complicate the selection of a service factor. Whereas an electric motor or turbine produces a steady non-pulsating flow of power to the driven mechanism, an internal combustion engine inherently will produce a pulsating load of power. The fewer the cylinders—the greater the pulsation and therefore the higher the service factor that must be used with internal combustion engines. (Clutch couplings require higher service factors than clutches alone. The higher factors are needed to protect the flexible coupling elements against the effects of fatigue.)

Four cylinder or 2 cycle engines:	
4.0	Clutch
6.0	Clutch Coupling
Six cylinder or 2 cycle engines:	
3.0	Clutch
5.0	Clutch Coupling
Eight cylinder or 2 cycle engines:	
2.0	Clutch
4.0	Clutch Coupling

Service factors should be compounded if an internal combustion engine is used with a pulsating type of load. For example, a pulsating load with a pump which normally requires a service factor of 1.5—when driven by a two cycle engine which requires a service factor of 4—would call for a service factor of 6.0 at the clutch (1.5 x 4.0 = 6.0).

OVERRUNNING APPLICATIONS

Relative Overrunning Speeds

The maximum overrunning speeds shown in the Formsprag Clutch catalog are the maximum speed for the race shown with the other race presumed to be stationary. In applications where both races are rotating during the overrunning cycle, there will be a relative overrunning condition. This condition will fall into one of three categories:

A. Inner race rotating with outer race rotating at a faster speed in the same direction. In this case the outer race would be overrunning the slower moving inner race. The relative overrun speed would be the speed of the inner race subtracted from the speed of the outer race.

B. Outer race rotating with inner race rotating in the same direction at a higher speed. In this case the inner race would be overrunning the slower moving outer race. The relative overrunning speed would be the speed of the outer race subtracted from the speed of the inner race.

C. Both races rotating but in opposite directions. In this case each race is overrunning and the relative overrunning speed is the sum of the speeds of both races.

The curves on the following pages show the relative overrunning speeds for clutch models FSR 6, FSR 14 and FSR 16, also FSO 400 and FSO 700. As shown on these curves, the relative overrunning speed for all three conditions described above may be picked from the curves.

Category: C

Races rotating in opposite directions with O.R. at 400 RPM. Then I.R. may run at any speed up to 1020 RPM. (1020 + 400 = 1420 max. relative speed)
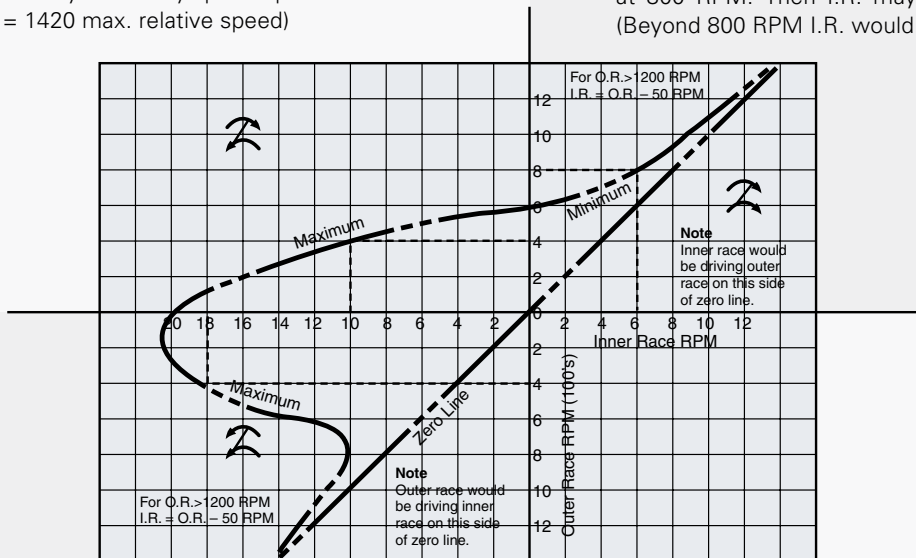
Category: A

Both races in same direction with O.R. as O/R member at 800 RPM. Then I.R. may run at 600 to 800 RPM. (Beyond 800 RPM I.R. would drive O.R.)

Category: B

Both races in same direction with O.R. at 400 RPM and I.R. as O/R member. Then I.R. may run at 400 to 1820 RPM. (Below 400 RPM O.R. would drive I.R.) (1820 - 400 = 1420 rel.)

Relative Over-run Speeds

Select outer race speed first and use curve to find limit of inner race speed.

Plotted values are for maximum or minimum inner race speeds as noted on curve.

Area inside curve covers safe relative over-run speeds. Outside of curve the relative speeds are too high. Outside of zero line clutch would drive as noted and no overrunning would occur.

The relative overrunning speed curves on the following page are typical curves for plain bearing clutches and ball bearing clutches. All relative overrunning curves have a similar configuration. Curves for sizes FS-02 through FSO-700 are available upon request.

OVERRUNNING APPLICATIONS

Relative Overrunning Speeds

Relative Over-run Speed
FSR-6

For O.R. = 1400 - 1600 RPM
I.R. = O.R. − 150 RPM
For O.R. > 1600 RPM
I.R. = O.R. − 50 RPM

For O.R. = 1400 - 1600 RPM
I.R. = O.R. + 150 RPM
For O.R. > 1600 RPM
I.R. = O.R. + 50 RPM
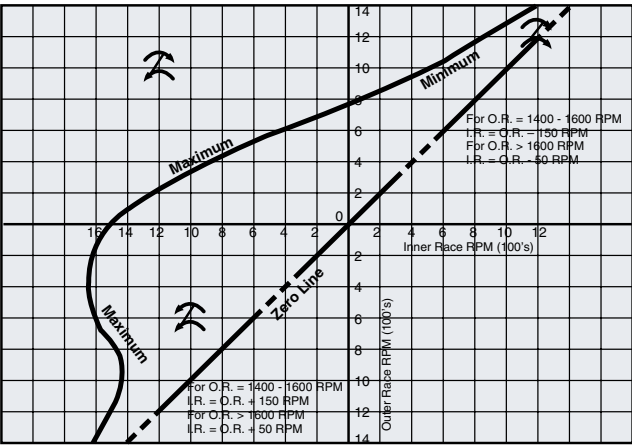
Relative Over-run Speeds
FSO-400
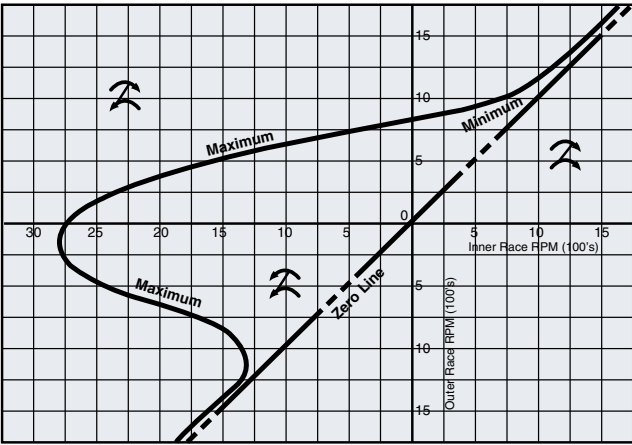
Relative Over-run Speed
FSR-14 & FSR-16

For O.R. > 400 RPM
I.R. = O.R. − 50 RPM

For O.R. > 400 RPM
I.R. = O.R. + 50 RPM
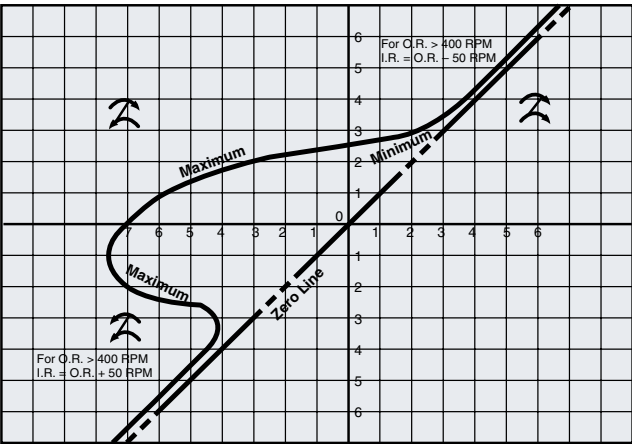
Relative Over-run Speeds
FSO-700
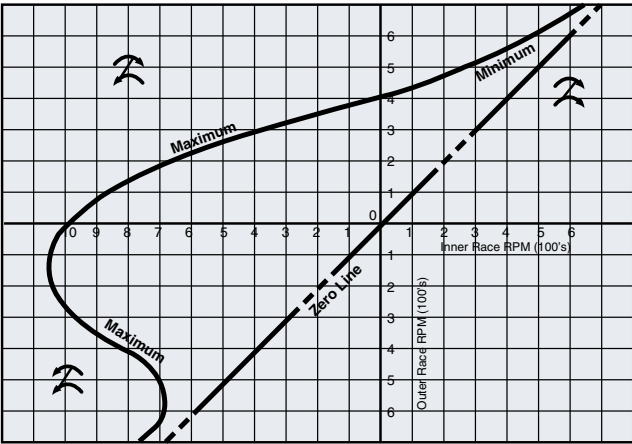
OVERRUNNING APPLICATIONS

Selection Considerations

The initial tentative selection of a Formsprag Clutch overrunning clutch will be based on three considerations:

1. Design torque including service factors
2. Overrunning speed and member
3. Shaft size

Before accepting the initial tentative selection as the final selection for the application, the design as a whole should be considered for its possible effect upon the choice or application of the overrunning clutch.

The following aspects of the design as a whole should be considered in arriving at the final clutch selection:

1. Location—where the design as a whole permits the choice of location for the clutch, all possible locations should be reviewed for the effect upon clutch operation at each location. In general a location calling for the lower overrunning speeds will call for higher torque and larger shaft size and, hence a larger clutch. Conversely, a smaller clutch can be used on a lower torque, smaller bore size application at the expense of a higher overrunning requirement.

    Select an FSO clutch for both inner race and outer race overrunning conditions if at all possible. If the application, however, requires outer race overrunning but the outer race overrunning speed exceeds the catalog limit of an FSO, then select an AL, GFR or HPO clutch design.

2. The design as a whole should be reviewed for the possible presence of axial and/or thrust load requirements imposed upon the bearings within the overrunning clutch. When such bearing load conditions do exist, the bearing capacities should be reviewed based on the information given elsewhere in this manual.

3. If the design as a whole imposes differential or relative overrunning speeds on the clutch, these speeds should be reviewed and compared against the maximum permissible speed shown on the relative overrunning speed curves.

4. Mounting considerations, such as vertical position or exposed locations will call for special attention when installing a clutch. In cases where the clutch is mounted on a vertical shaft, the use of grease lubrication has been found to give better results than the standard oil lubrication. In such cases, oil would tend to flow away from the upper bearing and leave it dry—whereas grease because of its heavier body and tendency to adhere to surfaces will remain on the upper bearings and keep them lubricated.

In exposed locations extra sealing provisions may be necessary and extra lubrication provisions may also be necessary in order to protect the internal members of the clutch. In all cases of critical mounting requirements, please refer to the factory before making the final selection.

Lubrication

Adequate provision for lubrication (oil holes or grease fittings) is provided on all clutches.

1. The following types of lubrication are recommended.

Lubricated for life	Oil	Grease
FS-02	*FSO 300/700	*FSR 5/16
FS-04	*HPI 300/700	FSO 750/1027
FS-05	*FS 750/1027	HPO 720/1027
FSR-3	HPI 750/1027	
	*FSA 1051/5000	
	FSD 27/300	
	*LLH 700/5000	
	CDS 600 & 700	
	CDU 500/1051	
	HSB 400/1027	

* If a clutch in the field is to be converted from oil lubrication to grease lubrication the clutch should be returned to Formsprag Clutch for proper installation of grease seals. Also, it should be noted that if a customer requires grease lubrication when ordering a new clutch, this should be so specified on the purchase order so that grease seals may be properly installed. Grease lubrication is satisfactory for ambient temperatures +20° to 150° F.

2. The following amounts of lubrication are recommended:

    Indexing Applications—7/8 FULL

    Overrunning Applications—HALF-FULL

    Holdbacks and Backstops—HALF-FULL

Caution:
Do not use lubricants of the E.P. Type (Extreme pressure characteristics) or those containing slippery additives. Fill to recommended level.

3. Use Mobil DTE Heavy-Medium oil for temperatures +20° to 150°F.

    When ambient temperature is below +20°F., use Mobil Gargoyle Arctic "C" Heavy to -10°F.

    For temperatures of -10°F. to -40°F., consult Formsprag Clutch.

OVERRUNNING APPLICATIONS

4. Grease Lubrication—For FS, FSO and LLH use Lubriplate Low Temp (Fiske Bros.). For HPO 720 – 1027 use Mobilith #21: Fill until grease flows freely from around seals in both end caps.

5. When ball bearing clutches are mounted vertically, use grease lubrication to assure adequate lubrication of top bearing.

For additional information on clutch maintenance, contact Formsprag Clutch or visit our website.

Flushing

1. Oil lubricated clutches should be flushed periodically every six months with mineral spirits such as Mobil Solvasol or equal. Do not use carbon tetrachloride. Flush more often if clutch is subject to severe operation or abrasive dust.

2. Flushing Procedure: Fill clutch with mineral spirits and operate for 5 to 10 minutes, then drain and relubricate.

3. Grease lubricated clutches do not normally require flushing. When clean grease is pumped in, old grease is forced out through seals.

4. Clutches which have been out of use for six months or longer should be flushed to remove any wax or gum formation resulting from vaporization of old lubricant.

Special Design Advantages

While the wide range of sizes and capacities covered by Formsprag Clutch's catalog line of clutches will cover substantially all industrial needs, it is sometimes necessary to design and build clutches to meet specialized requirements.

Special designs can also take advantage of a self contained lubrication system or lubricant contained within the customer's machine. In such cases, lubrication could be introduced through the inner race or through clutches assembled without seals to permit a free flow of oil through the clutch. The increased quantity and flow of lubricant in such cases would greatly increase the life of the clutch during overrunning periods. Such improved lubrication facilities will also permit a clutch to operate at much higher speeds than it could as a standard self-contained item.

Centrifugal Throwout Designs C/T Sprags

Special designs for overrunning clutches include designs which are "tailor made" to suit a specific application and clutches which are modifications of standard catalog items. In either category, centrifugal throwout construction "C/T" is available to permit high speed overrunning with the outer race as the overrunning member. With centrifugal throwout construction the C/T sprags are lifted completely clear of the inner race so that no rubbing occurs during overrunning. As a result, the overrunning speed is limited only by the speeds permissible on the bearings, and overrunning wear can occur only during the brief transition periods before the sprags lift clear of the inner race during overrunning.

The C/T retainer assemblies are designed for higher speed overrunning and lower speed drive conditions. C/T sprag retainers are available in all model sizes for the Series FS and FSO 300 through 1051. Models 300 through 700 are available with the PCE and C/T sprag combination.

Formsprag Clutch's years of research and development in this specialized field, are at your service to help solve all special overrunning clutch applications. In many cases a standard clutch can be adapted to a special use. You can be sure of a clutch that will meet your exact requirements, simply by supplying the following information:

1. Overrunning speed required

2. Desired torque capacity

3. Method of lubrication

4. Which race will overrun

5. Any special features you need

Overrunning Drag Torque

In an overrunning clutch the drag torque is listed as resistance after run-in in the product catalog No. P-956. The clutch drag is a result of the additive values of seal drag, bearing drag and sprag energizing drag. The drag (resistance after run-in) torque values are listed for each model and series in their respective catalog. New clutches will have a higher drag torque at first, but after 12 to 24 hours of overrunning at standard motor speeds will reduce to catalog listed values.

When a clutch is overrunning the drag torque is exerted generally on the lower speed race and any attached drive components.

CLUTCH COUPLINGS

Figure 27

FW Series

The FW Series clutch coupling is designed for INNER RACE OVERRUNNING. Mount the clutch half of the unit on the driven shaft.

Figure 28

FW C/T Series FW Series

The FW C/T Series clutch couplings are designed for OUTER RACE OVERRUNNING. Mount the clutch half of the unit on the driving shaft.

Figure 29

FWW Series

The FWW Series clutch coupling is designed for INNER RACE OVERRUNNING. Mount the driving coupling on the driving shaft and the driven coupling on the driven shaft. The clutch and adapter are then mounted with the adapter connected to the driving coupling.

Note: Mounting is reversed for C/T Series.

Figure 30

In all sizes except the FWW series, the clutch can be combined with either of two sizes of couplings. A high capacity coupling for applications demanding greater ruggedness and imposing higher operating loads, or a more economical unit for operating loads that are light and where bore size becomes the controlling factor in the selection of the coupling portion.

Light load applications should be selected on the basis of bore size. Torque requirements should determine the choice for heavy duty installations.

Formsprag Clutch Couplings

A clutch coupling is required when two shafts are coupled end-to-end and an overrunning provision is required in the installation. An overrunning clutch can not accept any angular or parallel misalignment and therefore requires the use of a coupling. A clutch coupling is a Formsprag overrunning clutch combined with a flexible coupling into a complete unit.

The Formsprag catalog model clutch couplings consist of the Formsprag overrunning clutch, an adapter plate and a disc coupling (Figures 28, 29 & 30). In series FW-403 through FW-1018 the Formsprag FSO-400 through FSO-1027 clutch is used in connection with the disc couplings. In series FWW-420 through FWW-755 the Formsprag FSO-400 through FS-750 clutch is used with two single flex couplings, one on each side of the overrunning clutch.

The disc coupling has the following features:

All metal construction, no wearing parts, no lubrication, wide temperature range, high torsional stiffness, no backlash, accepts limited axial movement and provides smooth and constant rotational velocity.

In dual drive applications the clutch drag (resistance after run-in) torque should be compared to the unpowered drive motor drag torque to determine if the unpowered motor will rotate (due to the drag difference). Slow rotation of the drive motor may be desirable to prevent false brinelling of the motor bearings due to vibration. If the possible rotation of the unpowered motor is a problem, then a small brake will be required. The brake torque rating should be sized larger than the clutch drag (resistance after run-in).
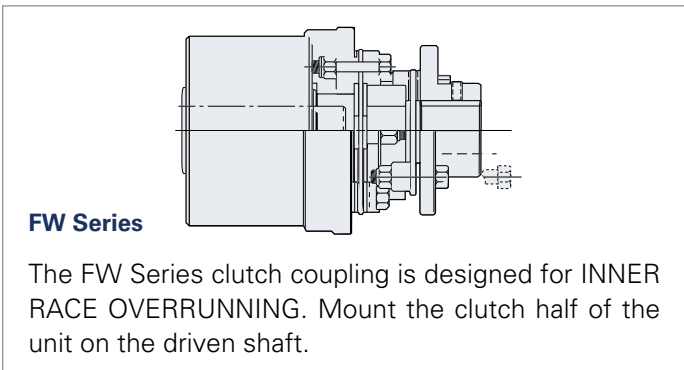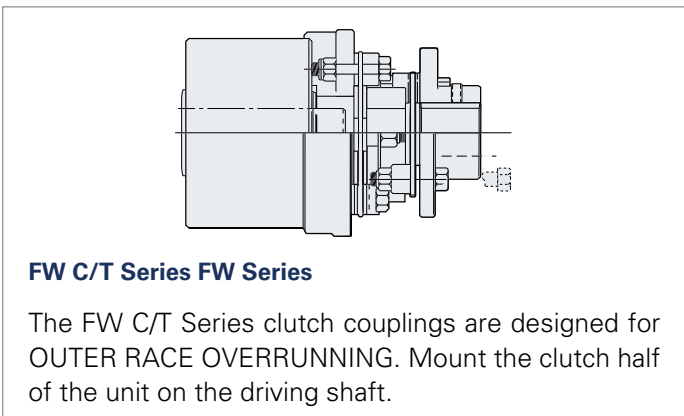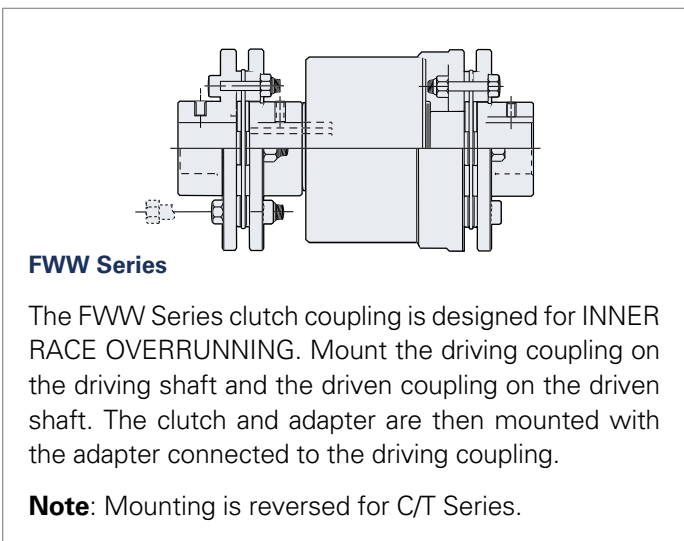
CLUTCH COUPLINGS

CK Series

Although not carried as a catalog standard item, the Formsprag-Gear line of clutch couplings is available. These consist of a Formsprag Clutch overrunning clutch, an adapter plate, and a modified Gear coupling, CK series (Figure 31).
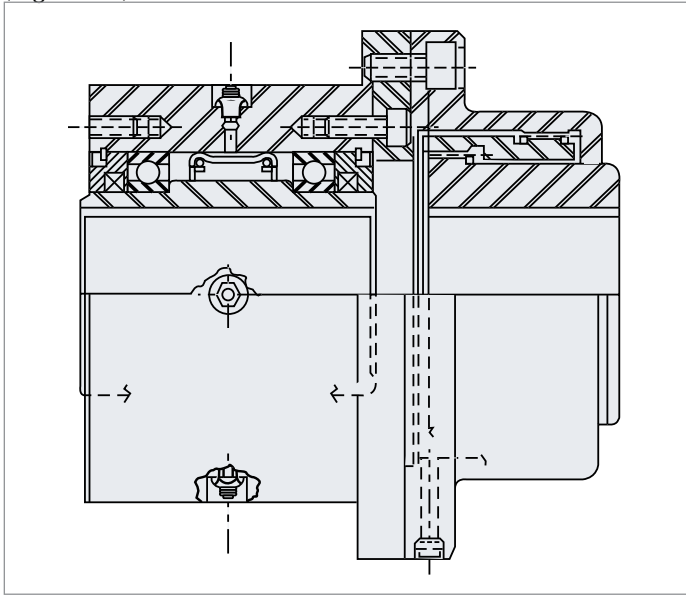
Figure 31

The Formsprag-Gear clutch coupling is available from sizes 700 through 1027. In all cases these are designated by the prefix CK. Following is a tabulation of the Formsprag clutch and Gear coupling combinations which are available:

Model	Formsprag Clutch	Gear Coupling (Type DE)
CK-700	FSO-700	3 1/2 AF
CK-750	*750	4 AF
CK-750	*750	4 1/2 AF
CK-800	*800	4 AF
CK-800	*800	5 AF
CK-900	*900	4 1/2 AF
CK-900	*900	5 AF
CK-1027	*1027	5 AF
CK-1027	*1027	5 1/2 AF

* FSO, AL or GFR Series may be used depending on the overrunning requirements of the application.

Note that on sizes 750 through 1027 two different coupling sizes are available for each clutch size—a high capacity coupling for applications demanding maximum torque capacity and a smaller more economical unit for the operating loads that are light and where bore size becomes the controlling factor in the selection.

The Formsprag Clutch CK series is also comprised of an FSA-1051 through FSA-5000 clutch and a suitable gear coupling to provide a torque range of 38,000 lb.ft. to 500,000 lb.ft. Consult Formsprag Clutch with application details.

Service Factors

In the Formsprag Clutch catalog line of clutch couplings the service factor requirements are based on the service factors required by the coupling portion of the unit.

1.5	Minimum service factor for any clutch coupling
2.0	Pulsating loads such as compressors, bucket elevators, and pumps
2.5	Heavy pulsating loads such as forced draft fans
3.0	Heavy variable loads such as induced draft fans and kilns
Four cylinder or 2 cycle engines:	
4.0	Clutch
6.0	Clutch Coupling
Six cylinder or 2 cycle engines:	
3.0	Clutch
5.0	Clutch Coupling
Eight cylinder or 2 cycle engines:	
2.0	Clutch
4.0	Clutch Coupling

Accumulate service factors if internal combustion engine is used with a pulsating load.

Example: Pump driven by a 2 cycle engine – 6.0

Use minimum service factor of 1.25 on ll Formsprag-Gear clutch couplings. Following is a brief listing of some typical service factors to be used with this line.

1.25	Minimum Service Factor
1.5	Compressors, Pumps, etc.
2.0	Forced Draft Fans
2.5	Induces Draft Fans
5.0	4 Cylinder Engines
4.0	6 Cylinder Engines
3.0	8 Cylinder Engines

INDEXING APPLICATIONS
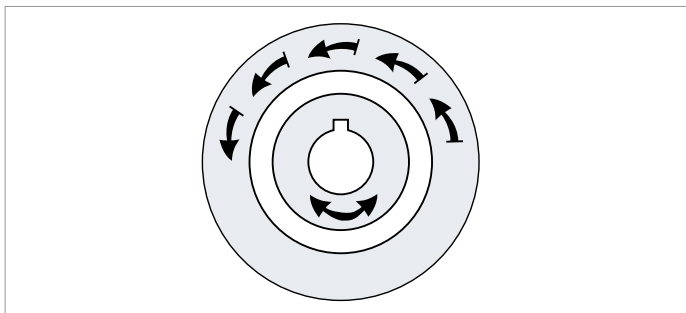
Figure 32

In this type of application, reciprocating motion applied to the driving race is transformed into intermittent motion in only one direction at the driven race (Figure 32). For example, if a pinion is connected to the driving race, a rack meshing with the pinion can give reciprocating motion to the driving race. The clutch will then advance, or "index" the work (Driven race) on each forward stroke of the rack, but will not return or back-up on the return stroke of the rack.

An indexing application is one in which the clutch is used to convert reciprocating or oscillating motion into intermittent rotary or intermittent linear motion. The input reciprocating motion may be applied in any one of several ways:

Typical methods are a rack and pinion or other linkage, a cam or eccentric, and pneumatic or hydraulic pistons (double-acting or spring return). The precision design and manufacture of Formsprag clutches and their unique freedom from backlash makes them especially reliable for use in precise indexing applications.

Consideration of Actuators

Indexing mechanisms may be operated by many different means but the operating means can be divided into two main categories: harmonic motion and non-harmonic motion.

Harmonic motion is characterized by smooth acceleration from zero to full speed followed by smooth deceleration back to zero as shown in Figure 33.
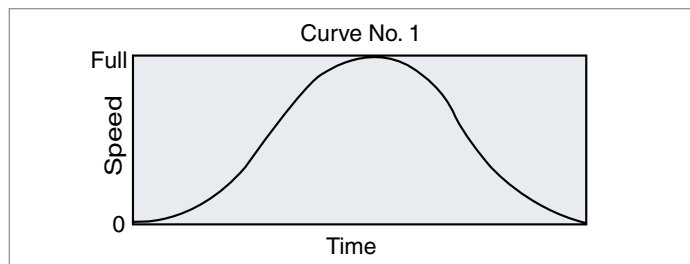
Figure 33

This motion is given by a crank mechanism and the variation in rate reflects the varying positions of the crank as it travels from top dead center to the midpoint, and then to bottom dead center.

The motion from the crank may be transmitted to the clutch by means of a linkage with a lever arm fastened on the clutch, a linkage operating a rack and pinion drive with the pinion being mounted on the clutch, or a combination of these means. A typical example is shown in Figure 34 in which the crank arm imparts harmonic motion. The minimum service factor for any indexing application is 2.0 and that minimum factor would apply to an application such as this.
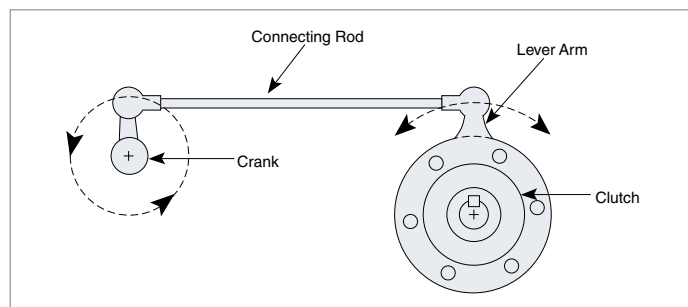
Figure 34

Because of the smooth acceleration curve with harmonic motion, the indexing clutch normally is not subjected to shock loading and the minimum service factors may be used.

Non-harmonic motion is characterized b a very abrupt acceleration from zero to full speed unlike the curve given by harmonic motion. This is shown in Figure 35.
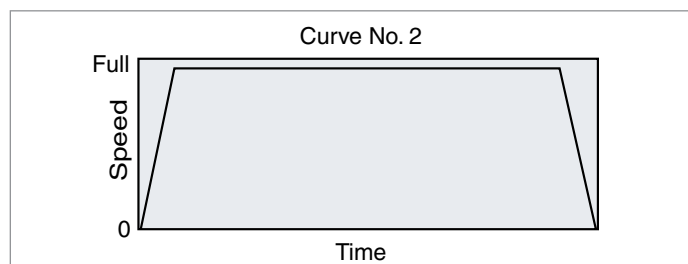
Figure 35

Non-harmonic motion is imparted by hydraulic or pneumatic pistons. The piston attempts to accelerate the mass being indexed up to full speed as soon as pressure is applied to the piston. This can result in very high shock loads at the indexing clutch. Because of the very rapid acceleration with resulting shock loads, the maximum service factors should be used on uncushioned piston-operated indexing mechanisms.

The abrupt action of the piston-operated mechanism may be cushioned by means of metering valves to control the initial admission of pressure to the piston and thus control the rate of acceleration and by dashpots to cushion or control the deceleration at the end of the stroke. The use of such cushioning devices will tend to approximate the action found in a harmonic motion drive and will permit the use of a reduced service factor.

INDEXING APPLICATIONS

Service Factors

The service factors used for indexing applications will depend upon the nature of the load applications. A load which is smoothly applied, uniformly accelerated, and smoothly released (such as results from the harmonic motion on the input) will require the smallest service factor. A load that is applied abruptly but which is cushioned by means of a dashpot will require a higher service factor, and an abruptly applied load without the cushioned effect of a dashpot will require the highest service factor.

Also on installations using the plain bearing series FS-02 through FSR-16, a higher service factor would be required than a similar installation using a ball bearing HPI series clutch.

Following is a tabulation of service factors applicable to indexing installations.

Harmonic motion

Ball bearing HPI series—2.0
Plain bearing FS-02 to FS-16—3.0

Abrupt motion (Piston)

Ball bearing HPI series
    With dashpot—2.5
    Without dashpot—3.0
Plain Bearing FS-02 to FS-16
    With dashpot—3.0
    Without dashpot—4.0

Selection Considerations

The proper selection of indexing clutches requires the careful evaluation of more variables than for either overrunning or backstopping applications.

Indexing applications normally require extreme accuracy such as in a mechanism feeding strip stock to a punch press. An overfeed of stock would represent wasted raw material and underfeed would deliver insufficient raw material for a complete part.

The Formsprag clutch will faithfully transmit the exact motion imposed on it to assure an accurate indexing system. Proper selection of the indexing clutch requires a consideration of the total torque imposed on the clutch and selection of an adequate service factor consistent with the rate of indexing and the desired life.

Formsprag clutches are designed to transmit as much torque as the shaft on which the clutch is mounted can carry, subject to the usual safety factors for shaft stress. Therefore, as a general rule, a clutch selected on the basis of shaft diameter will be adequate for the load. However, it should be remembered that torque requirements increase rapidly, in proportion to the increase in the rate of acceleration of the inertia load.

The torque load to be transmitted by the clutch can be determined by using the formula:

$$T = \frac{HP \times 5250}{RPM}$$

Brake Torque. If a brake is to be used in the indexing system, the clutch must operate against the resistance of the brake; this resistance must therefore be calculated in terms of torque added to the other torque values in the mechanism.

Stock Load. If, as in a punch press feed device, the indexing mechanism must pull the stock from a coil, the force required to do this must be added. This can best be determined by actual measurement.

Service Factor. In indexing applications, the service factor will range from 2 to 4, depending on the rate and magnitude of index, operating loads and the series of clutch selected. Determine the applicable service factor from the table.

Speed and Stroke. For slow speeds and short strokes the FS series is usually appropriate. However, at higher speeds or where the stroke exceeds 90°, the HPI series must be used. Select the appropriate model from the engineering data tables in catalog P-956.

Shaft Diameter. This is necessary to select a clutch of the correct bore size. Bore size of the clutch cannot be changed in the field.

6 Steps for Selecting an Indexing Clutch

1. Determine torque requirements from formula

    $$T = \frac{HP \times 5250}{RPM}$$

2. Select and apply proper service factor from table.

3. Determine shaft size and bore requirement of clutch.

4. Select model size based on torque and bore requirement.

5. Determine series on basis of number of strokes per minute and degrees per stroke.

6. Specify the proper series, model and bore.

INDEXING APPLICATIONS

Check Clutches

In an indexing operation the material being indexed is fed forward during the feed portion of the cycle and should stand still while the index clutch overruns during the return portion of the indexing cycle. In cases where the stock fed is very light and offers very little resistance or where the stock being pulled into the mechanism offers tension and resistance, the material may have a tendency to go backward during the return stroke instead of standing still. In such cases a check clutch should be added to the index mechanism (Figure 36).

The check clutch is essentially a backstop with one race grounded to a stationary member of the mechanism and the other race connected to the index drive shaft. During the feed portion of the cycle, the indexing clutch will drive and the check clutch will overrun in that same direction. During the return portion of the cycle, the indexing clutch will overrun and the check clutch will hold stationary to prevent any tendency toward backward motion on the part of the material being indexed.

The check clutch can normally be a much smaller clutch than the index clutch. The index clutch must allow for the inertia of all parts being indexed, the torque required to index the material in question, and must provide for shock loads. On the other hand, the check clutch need only have enough capacity to overcome any tendency toward backward motion under static conditions.

Overthrow Brakes

In any indexing mechanism the material being indexed will display a tendency to over-throw. In the case of a harmonic motion drive, the inertia of the parts being indexed will tend to travel at the same rate during the deceleration portion of the index cycle. Thus, the material being indexed will attempt to overrun the clutch as the clutch decelerates.

In the case of non-harmonic drives (whether cushioned or uncushioned) the material being indexed will tend to over travel as a result of its own inertia when the indexing mechanism reaches the end of its stroke.

Here again, the material being indexed will tend to overrun the clutch. In either case the tendency toward overthrow can be controlled by the addition of a drag brake in the indexing mechanism. The simplest installations use a spring-loaded pony brake on the extended shaft of the feed roll. The drag brake should be adjusted to absorb the inertia of the parts being indexed and thus cause the indexing clutch to drive to the end of the feed portion of the cycle. The brake should be tightened no more than necessary to absorb inertia. Excessive braking will shorten brake life and add un-needed load to the clutch.

The addition of an overthrow brake will increase the torque load imposed upon the clutch and this additional load must be considered when establishing the size of the indexing clutch.
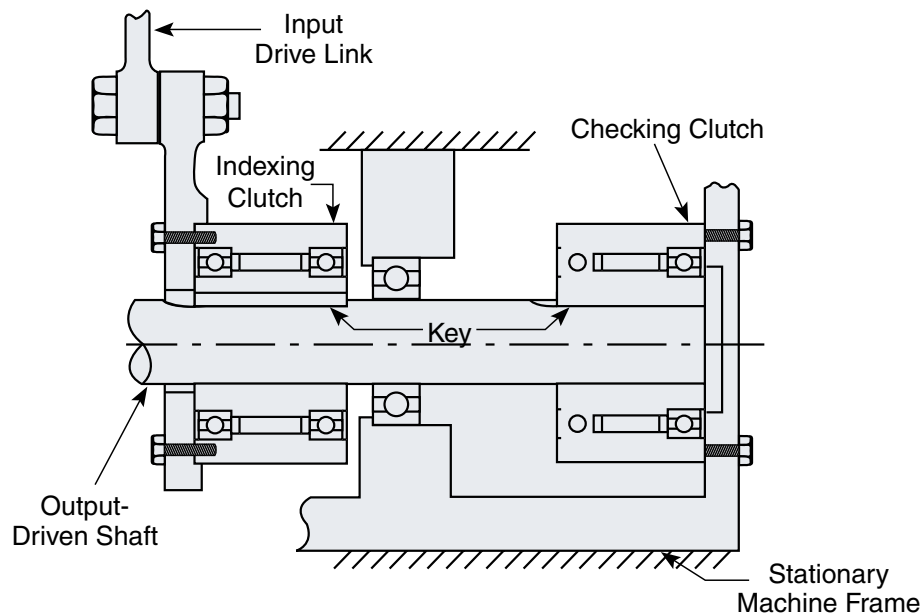
Figure 36

INDEXING APPLICATIONS

Accuracy

The Formsprag Clutch overrunning clutch will faithfully transmit whatever signal is given it. There is no slip or lost motion in any Formsprag Clutch overrunning clutch because the sprags are always in contact with both races at all times. Since the sprags are always in contact and ready to assume the torque load, there is no lost motion or lost impulse necessary to permit parts to move into position. All parts are elastic. The components of the Formsprag Clutch overrunning clutch are subject to torsional windup due to the natural elasticity of the parts. This torsional windup, of course, is very small and is directly proportional to the amount of load applied. Since the windup is directly proportional to the load, it remains a constant for any given load and is automatically canceled out during the initial set-up. Because the torsional windup is constant and can be cancelled out during set up, it should not be construed as lost motion in any sense. The overall accuracy of the complete indexing installation depends upon the fits and clearances ithin the entire train of the indexing mechanism.

Small variations in indexing accuracy which may range from overfeed to underfeed are never the result of improper performance within the Formsprag Clutch indexing clutch. Small "over-and-under" variations of that nature could be the result of looseness or wear upon the other elements comprising the indexing system.

Sources of Inaccuracy

The possible sources of such inaccuracies would be looseness, wear, or backlash in the linkage, lever arms, or gears feeding the signal to the clutch and the fit of such gear or lever arm to the outer race of the clutch.

Fit of Shaft and Key

The fit of the clutch bore to the shaft and the fit of the key between the clutch inner race and shaft can affect the accuracy of the system.

Other Areas

Other possible sources of minor indexing variations can be found in clearances or looseness in other elements of the indexing system removed from the area of the clutch itself. An exaggerated situation is shown in Figure 37, in which there are 11 possible sources of inaccuracy.
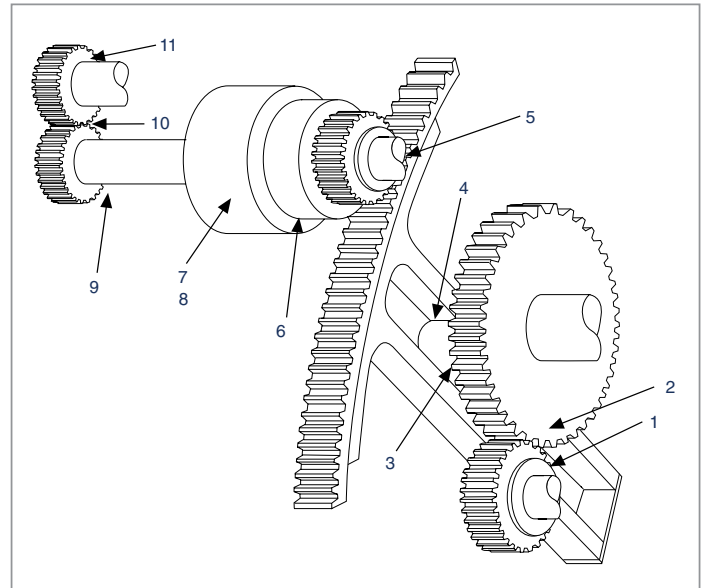
Figure 37

11 Possible Sources of Inaccuracy

1. Fit and mounting of drive pinion to shaft
2. Backlash between gears
3. Adjustment of crank throw
4. Fit of crank in gear sector slot
5. Backlash between gears
6. Fit and mounting of pinion to clutch
7. Fit of clutch on shaft
8. Fit of key between clutch and shaft
9. Fit and mounting of pinion on shaft
10. Backlash between gears
11. Fit and mounting of pinion on shaft

Torsional Windup

As discussed in the section on accuracy, there will be torsional windup in any Formsprag Clutch overrunning clutch while it is carrying load. This windup is directly proportional to the amount of load applied and will not vary for any given torque load. (See Torsonial Windup)

INDEXING CLUTCH SELECTION

Nomographs

On the following pages are nomographs which simplify the selection of indexing clutches.

The selection steps described on page 27 are arranged in chart form on the nomographs. The three nomographs are based on the total inertia of all parts being indexed.

1. **Inertia up to 10 lb. in. sec.²**
2. **Inertia up to 100 lb. in. sec.²**
3. **Inertia up to 1000 lb. in. sec.²**

Example:

Assume:

I = 50 lb. in sec.²

100 indices per minute at 15° per index

Brake Torque (External) - 1500 lb. in.

(If a Formsprag clutch were used as a check device instead of a brake, external brake torque would be zero.)

Use Nomograph No. 2 (Up to 100 lb. in. sec.²)

Solution:

1. Draw line from 50 lb. in. sec.² (I) through 15° per inde (Ø) to "Turn Line".

2. Draw line from that point on "Turn Line", through 100 per minute (N) to Inertia Torque Line (T_i).

3. Draw line from Inertia Torque (T_i) to Brake Torque (T_s) to find point on Total Torque Line (T_t).

4. Transfer the value from the Total Torque Line (T_t) to the Sloping Total Torque Line (T_t).

5. Draw line from Service Factor (F_s) through Sloping Total Torque (T_t) to Rated Static Torque Line (TR). See page 27 for Service Factors.

6. Read clutch required opposite Rated Static Torque (TR).

Selection: Model FSR 12

Calculation:

See previous example.

$$TR = F_s \left(\frac{I\emptyset N_2}{5225} \right) + T_B$$

$$TR = 3 \left(\frac{50 + x\ 50 \times 100_2}{5225} \right) + 1500$$

TR = 8806 lb. in.

Selection: Model FSR 12

Formulas

Clutch torque based on known speed and horsepower

$$T = \frac{HP \times 5250}{RPM}$$

in which

T = torque, pound feet

HP = horsepower

RPM = revolutions per minute

Clutch torque — angular harmonic motion — (indexing)

$$T_t = \frac{N_2 \emptyset}{5225}\ I + \mathbf{T}_B$$

in which

T_t = Total torque, pound inches

N = Indexes per minute

Ø = Angular motion of clutch per index, in degrees

I = Mass moment of inertia of load, pound inches sec.,² (see below)

$\mathbf{T}_B$ = Brake Torque, pound inches, not included in first term, engaged brakes, feeding strip, friction, etc.

INDEXING CLUTCH SELECTION

Mass Moment of Inertia

For Solid Cylinder or Roll

$$I = \frac{\pi \, d \, L \, R^4}{2g}$$

For Hollow Cylinder or Roll

$$I = \frac{\pi \, d \, L \, (R^4 - r^4)}{2g}$$

in which

I = mass moment of inertia, pound inches sec.^{2}
d = density, lbs/in.^{3}
L = cylinder length, inches
R = outside radius, inches
r = inside radius, inches
g = acceleration due to gravity = 386 inches per sec.^{2}

For Train of Cylinders or Rolls

$$I \text{ effective} = I_1 + I_2 \frac{r_1^2}{r_2^2} + I_3 \frac{r_1^2}{r_3^2} + \dots + I_\eta \frac{r_1^2}{r_\eta^2}$$

in which

I_1 = mass moment of inertia of first roll, pound inches sec.^{2}
I_2 = mass moment of inertia of second roll, pound inches sec.^{2}
I_η = mass moment of inertia of ηth roll, pound inches sec.^{2}
r_1 = radius of first roll, inches
r_2 = radius of second roll, inches
r_η = radius of ηth roll, inches

Torsional Shaft Stress—solid shaft

$$S_s = \frac{5.1 \, T}{D^3}$$

Torsional Shaft Stress—hollow shaft

$$S_s = \frac{5.1 \, TD}{D^4 - d^4}$$

in which

S_s = torsional shear stress, PSI
D = outside diameter of shaft, inches
d = inside diameter of shaft, inches
T = torque, pound inches

Feed-Speed Proportioner

To use a nomograph as a feed-speed proportioner, work the nomograph in reverse from the point of rated capacity on the right hand scale for the model used.

Points can be established on the inertia torque line and the load inertia line for a given installation. Any pair of lines between these two points will establish a satisfactory combination of index rate and index angle. Knowing a desired index angle, the maximum index rate is determined. Knowing a desired speed, the maximum index angle is determined.

INDEXING NOMOGRAPH NO. 1

FS-02 to HPI-500
Model Selector
Feed-Speed Proportioner

$$TR = F_S \left(\frac{I\emptyset N^2}{5225} + T_B \right)$$

Typical Selection
I = 2 lb. in. sec.²
5∞ Per Index
800 Indexes Per Minute
400 lb. in. Brake Setting

The dash lines
illustrate the six
steps to the solution
of the typical selection

Moment of
Inertia of
Load, I
lb. in. sec.²

Angle of
Index Ø,
Degrees

Turn
Line

Indexes
Per
Minute,
N

Inertia
Torque,
T_i,
lb. in.

INDEXING NOMOGRAPH NO. 1

Formsprag Clutch Required

Total Torque, T_t, lb. in.

Brake Torque, T_B, lb. in.

Service Factor, F_s

Sloping Total Torque, T_t, lb. in.

Rated Static Torque Required, TR, lb. in.

Rated Static Torque Required, TR, lb. in.

INDEXING NOMOGRAPH NO. 2

HPI-400 to HPI-750
Model Selector
Feed-Speed Proportioner

$$TR = F_S \left(\frac{I\emptyset\,N^2}{5225} + T_B \right)$$

Typical Election
I = 50 lb. in. Sec.²
15∞ Per Index
100 Indexes Per Minute
1500 lb. in. Brake Setting

The dash lines
illustrate the six
steps to the solution
of the typical selection

HPI or FS

HPI Only

Moment of
Inertia of
Load, I
lb. in. sec.²

Angle of
Index, Ø,
Degrees

Turn
Line

Indexes
Per
Minute,
N

Inertia
Torque,
T_i,
lb. in.

INDEXING NOMOGRAPH NO. 2

Formsprag Clutch Required

Total Torque, T_t, lb. in.

Brake Torque, T_B, lb. in.

Service Factor, F_s

Sloping Total Torque, T_t, lb. in.

Transfer

FSR-5
FSR-6
FSR-8
FSR-10
FSR-12
FSR-14
FSR-16

HPI-400
HPI-500
HPI-600
HPI-700
HPI-750

Rated Static Torque Required, TR, lb. in.

INDEXING NOMOGRAPH NO. 3

HPI-400 to HPI-1027
Model Selector
Feed-Speed Proportioner

$$TR = F_S \left(\frac{I\emptyset N^2}{5225} + T_B \right)$$

Typical Selection
I = 200 lb in. Sec.^{2}
8∞ Per Index
290 Indexes Per Minute
2000 lb. in. Brake Setting

The dash lines
illustrate the six
steps to the solution
of the typical selection

Moment of
Inertia of
Load, I
lb. in. sec.^{2}

Angle of
Index, ø,
Degrees

Turn
Line

Indexes
Per
Minute,
N

Inertia
Torque,
T_i,
lb. in.

INDEXING NOMOGRAPH NO. 3

Formsprag Clutch Required

Total Torque, T_t, lb. in.

Brake Torque, T_B, lb. in.

Service Factor, F_s

Sloping Total Torque, T_t, lb. in.

Transfer

Rated Static Torque Required, TR, lb. in.

324,000 lb. in. Rating

HPI-400
HPI-500
HPI-600
HPI-700
HPI-750
HPI-800
HPI-900
HPI-1027

HOLDBACK OR BACKSTOP APPLICATIONS

Figure 38

In backstopping or holdback applications, one race is always stationary (Figure 38). The function of the clutch is to permit rotation of the inner race in one direction only, and to prevent any rotation in the reverse direction at any time. This is therefore basically an overrunning installation, applied to the one job of "holding back," usually as a safety device on conveyors, gear reducers, and similar types of equipment.

Service Factors

The service factors for holdbacks will vary from 1.0 to 2.5 depending upon the nature of the application.

Infrequent, non-functional, non-critical loadings—1.0

Frequent, non-functional, non-critical loadings—1.5

Frequent, functional, but not critical loadings—2.0

Frequent, functional, and critical loadings—2.5

Infrequent, non-functional, non-critical loadings are those such as would be found on conveyors which are stopped only at the end of the day or at the end of a shift, and in which the backstop is merely called upon to hold the load until the beginning of the next drive period.

Frequent, non-functional, non-critical loadings are those in which the conveyor would be stopped several times a day but in which the loss of load due to conveyor runback could result in inconvenience but not in damage to personnel or equipment.

Frequent, functional but not critical loadings are those in which the conveyor is stopped frequently as a functional part of a system. For example, in loading a series of cars the conveyor will drive while loading one—then stop and hold the load while the next car is brought under the conveyor and repeat.

Frequent, functional, and critical loadings are similar to the last one with the addition of the important fact that conveyor runback could cause damage to equipment or injury to personnel if load runs backward on conveyor.

Selection Considerations

Holdback and backstop applications represent the simplest selection category for Formsprag clutches. Normally, inertia and dynamic torques are not a consideration since the mechanism using the clutch must come to rest before torque is applied. Additionally, since the Formsprag clutch has no backlash, the clutch is subjected to torque at zero rpm, which excludes any buildup of inertia torque.

Care must be used in calculating the torque requirements for a backstop or holdback clutch. Clutches must be selected for maximum conditions rather than average or typical conditions. These maximum conditions require a thorough review of all possibilities of the entire system on which the clutch is used.

Backstop clutches may be selected in the same manner as holdback clutches. However, many backstop clutches are incorporated as an integral part of a speed reducer and are incorporated into the reducer at the time of manufacture. Because the actual holdback torque cannot be calculated, it is common practice to use a holdback clutch selected on the basis of the maximum horsepower motor which can be used with the reducer.

The torque requirements for backstops may be found using the standard formula—

$$T = \frac{HP \times 5250}{RPM}$$

In the case of holdbacks used on inclined conveyors, the horsepower required to drive the conveyor must be able to overcome friction losses, in addition to lifting the load on the conveyor. The friction losses actually assist the holdback and, as a result, a holdback calculated on the basis of full motor horsepower would be far larger than is necessary.

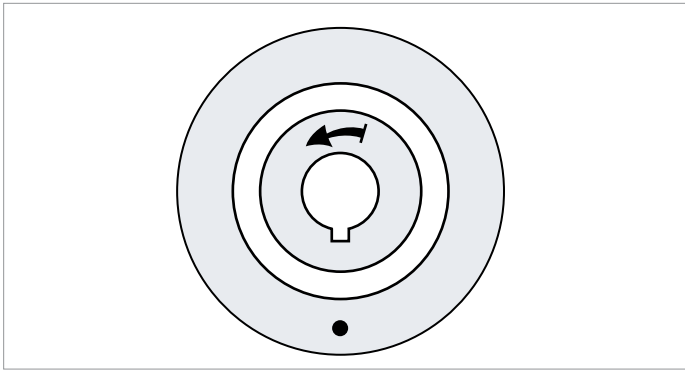For this reason the torque calculations for holdbacks used on inclined conveyors is rather complex in that it takes friction losses into consideration. Following are the considerations used in calculating the holdback torque requirements for inclined conveyors.

Slow speed holdbacks for inclined belt conveyors are normally selected based on calculations of the reverse torque generated by the design peak load.

Overloads resulting from operations beyond design capacity, improper electric control and oversized motors can result in holdback damage unless the holdback is selected for the more severe conditions.

There are several ways of calculating the amount of torque that might be imposed upon a holdback. Illustrated on the following pages is one method. Each case should be determined upon its own set of circumstances.

HOLDBACK OR BACKSTOP APPLICATIONS

Factors such as the possibility of maximum belt bulk loading to the spill or overflow point, may have to be considered in specific cases because this additional load would reflect in the holding power required by the holdback.

Although the conveyor system may be designed to operate at only a percentage of full theoretical capacity, the holdback should be selected on the basis of full capacity to allow for jamming or over-filling. The full capacity should be based on angle of repose of bulk materials capable of being heaped above the normal level of the conveyor.

Also it may be necessary in a specific installation, to consider the possibility of belt stretch due to jamming of the conveyor or jogging a fully loaded conveyor causing belt stretch. The instant the motor is stopped, the "trapped energy" in the belt reacts upon the holdback adding additional load.

These are some of the examples that the conveyor engineer should consider when calculating a holdback torque. It is not intended to imply that these considerations should be used for all selections. Each case must rest upon its own particular set of specifications and circumstances.

Formsprag Clutch does not assume the responsibility for determining the amount of torque which will be imposed upon the holdback. Nor does it accept any responsibility for the Speed and Idler Factors published on the following pages although it believes them to be accurate. The method illustrated should be considered as a guide. The conveyor design engineer should calculate and specify to FORMSPRAG CLUTCH the required maximum holdback torque.

Selection Procedure

Conveyors

There are basically two ways to size conveyor backstops or holdbacks, either according to CEMA (Conveyor Equipment Manufacturers Association) standards or based on Motor Breakdown/Stall Torque. Formsprag Clutch typically uses the Motor Breakdown/Stall Torque method. The CEMA method may be used as an alternative but could result in a selection of a smaller holdback. Formsprag Clutch recommends that the selection be made on whichever value is greater.

Motor Breakdown/Stall Formula

Motor breakdown or stall torque could be imposed on the backstop if the conveyor is jammed or frozen, or when the conveyor is overloaded. In this condition the motor will produce stall torque which is significantly higher than the motor nameplate rating. This will cause the belt to stretch; and when the motor stalls, the stall torque produced by the motor will be imposed on the backstop. This torque will be increased if there is a load on the belt. Therefore, the maximum loading on a backstop occurs in the rare case on a moving belt being gradually overloaded until the motor reaches stall or breakdown torque. The torque on the backstop will be the motor torque from the stretched belt plus the torque from the load moving in the reverse direction due to the force of gravity.

To select a holdback based on motor breakdown torque, calculate motor torque using the following formula:

$$T_{motor}(\text{lb. ft.}) = \frac{(SF)(\text{Motor Nameplate HP})(5250)}{(\text{Headshaft RPM})}$$

The clutch may be selected on the basis of installed nameplate horsepower without using a service factor (S.F.), provided the motor breakdown torque does not exceed 175 percent of the nameplate rating. A minimum service factor based upon the ratio of motor breakdown torque to 175 percent of nameplate torque is required in order to preclude serious damage to the installation or to the holdback.

CEMA Formula

The CEMA formula allows the design engineer to consider friction as a partial aid in preventing reverse rotation, thus reducing the torque required for the clutch. Selection by this method does require the use of a service factor (S.F.). The basic CEMA formula for design torque is:

$$T_{CEMA}(\text{lb. ft.}) = \frac{(SF)(\text{Lift HP} - 1/2 \text{ Frictional HP})(5250)}{(\text{Headshaft RPM})}$$

Formsprag Clutch recommends a minimum service factor of 1.5 when sizing with this formula. See CEMA Calculation Example on page 40.

Torque Limiting Device

If a torque limiting device is used and is less than 175 percent of the motor nameplate torque, then the holdback selected should have a capacity of not less than 1.5 times the CEMA runback torque.

Bucket Elevators

When selecting and sizing a holdback on a bucket elevator application, friction may or may not be considered because it is usually only a small fraction of the lift horsepower (HP) required. Similarly, motor breakdown HP is usually not considered if traction wheel drives are used because the elevator will usually slip before the breakdown HP is reached. For these reasons, the recommended design torque is based on lift HP alone. To calculate the torque required for a bucket elevator use the following formula:

$$T_{B}(\text{lb. ft.}) = \frac{(SF)(\text{Lift HP})(5250)}{(\text{Headshaft RPM})}$$

A minimum service factor (S.F.) of 2 recommended. If additional assistance is needed, consult Formsprag Clutch Applications Engineering.

HOLDBACK OR BACKSTOP APPLICATIONS

CEMA Calculations

Example: A conveyor 42″ wide is moving taconite which has an average weight of 130 lbs. per cubic foot at a rate of 1800 tons per hour. The conveyor is driven by a 30″ diameter head shaft pulley at 51 rpm to give a conveyor speed of 400 feet per minute. The conveyor is inclined at an angle of 18° and has a total lift of 32 ft. It is desired to mount a holdback clutch on the 3/4″ diameter head shaft. The conveyor is for intermittent duty. (Figure 39)
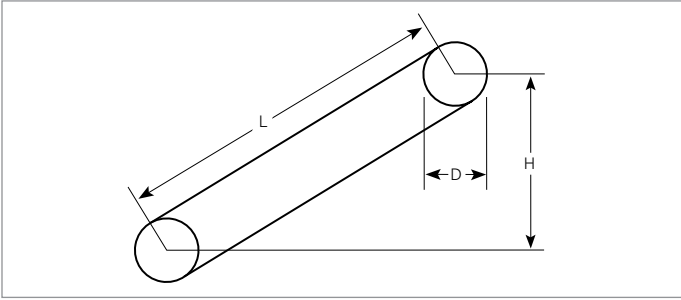
Figure 39

The following steps are used to select a holdback clutch by the CEMA method.

1. Ascertain that all data required is given or available.

In this example the Conveyor Length (L) can be determined as 104 feet from the Angle of Inclination and the Total Lift (H) of the conveyor using the formula:

$$L = \frac{H}{\text{Sine of Inclination}}$$

Since the conveyor will be used for intermittent duty having frequent stops, select a Service Factor (SF) at 1.5.

See Service Factors on page 38.

Select the Speed Factor (F) and Idler Factor (C) from Table 3 on page 41. Note that the Speed and Idler Factors depend upon the Height divided by the Length ratio, in this example the H/L ratio = .308.

Therefore, F = .036 and C = .027 for a 42″ Belt Width.

2. Enter the data into the appropriate equation and perform the calculations.

A. Power to lift load (P1):

$$P1 = \frac{W\,H}{990}$$

$$P1 = \frac{(1800)\,(32)}{990}$$

$$P1 = 58.2 \text{ HP}$$

Data Required	
Conveyor Length	L = 104 Feet
Belt Speed	S = 400 FPM
Short Tons Per Hour	W = 1800 TPH
Total Lift	H = 32 Feet
Headshaft Pulley Dia.	D = 2.5 Feet
Belt Width	= 42 inches
Material Weight	= 130 Lb/Ft³
Speed Factor	F = .036
Idler Factor	C = .027
Service Factor	= 1.5

B. Power to move empty belt and idler (P2):

$$P2 = \frac{.5\,L\,S\,F}{1000}$$

$$P2 = \frac{.5\,(104)\,(400)\,(.036)}{1000}$$

$$P2 = .75 \text{ HP}$$

C. Power to move loaded belt (P3):

$$P3 = \frac{.5\,L\,W\,C}{990}$$

$$P3 = \frac{.5\,(104)\,(1800)\,(.027)}{990}$$

$$P3 = 2.55 \text{ HP}$$

D. Total Holdback Power (HBP):

$$HBP = P1 - P2 - P3$$

$$HBP = 58.2 - .75 - 2.55$$

$$HBP = 54.9 \text{ HP}$$

E. Holdback Torque (T):

$$T = \frac{(HBP)\,(\pi D)\,(5250)\,SF}{S}$$

$$T = \frac{54.9\,(\pi 2.5)\,(5250)\,1.5}{400}$$

$$T = 8{,}489 \text{ lb.ft.}$$

3. Select the holdback on the basis of:

A. Torque requirement. In this case: 8,489 lb.ft.

B. Overrunning Speed. In this case: 51 RPM

C. Shaft Size. In this case: 4.750 in.

Answer: LLH 900

Note: The LLH 800 was not selected because the maximum bore size in only 4.437 in. per Catalog P-956.

HOLDBACK OR BACKSTOP APPLICATIONS

Table 3 (Conveyor under 500' Long)

Based on Material Weight (W), Total Lift (H), and Conveyor Length (L)

Belt Width	Material Weight (Lb./Ft.³)												H L (Ratio)
	30 lbs.		50 lbs.		75 lbs.		100 lbs.		130 lbs.		200 lbs.		
	"F"	"C"	"F"	"C"	"F"	"C"	"F"	"C"	"F"	"C"	"F"	"C"	
18″	.017	.050	.017	.050	.018	.050	.018	.050	.018	.050	.020	.037	.0 − .105
	.015	.040	.015	.040	.016	.040	.016	.040	.018	.034	.020	.033	.105 − .310
24″	.021	.050	.020	.043	.020	.045	.020	.041	.020	.045	.022	.037	.0 − .105
	.020	.035	.019	.033	.019	.033	.020	.030	.020	.030	.020	.030	.105 − .310
30″	.025	.044	.025	.043	.022	.037	.022	.041	.025	.038	.022	.037	.0 − .105
	.024	.030	.025	.028	.022	.028	.023	.025	.025	.025	.025	.025	.105 − .310
36″	.030	.040	.030	.038	.030	.037	.030	.038	.032	.038	.030	.039	.0 − .105
	.027	.030	.027	.030	.027	.030	.027	.030	.032	.029	.030	.023	.105 − .310
42″	.030	.040	.036	.036	.030	.040	.030	.040	.036	.040	.038	.040	.0 − .105
	.031	.030	.031	.030	.035	.027	.036	.027	.036	.027	.038	.027	.105 − .310
48″	.038	.036	.038	.038	.038	.039	.043	.038	.045	.040	.052	.044	.0 − .105
	.038	.030	.038	.030	.040	.027	.043	.027	.045	.027	.052	.027	.105 − .310
54″	.040	.037	.040	.040	.045	.042	.046	.042	.050	.047	—	—	.0 − .105
	.040	.030	.041	.028	.047	.028	.051	.028	.051	.030	—	—	.105 − .310
60″	.042	.037	.042	.040	.052	.045	.052	.049	—	—	—	—	.0 − .105
	.042	.030	.042	.028	.052	.029	.052	.030	—	—	—	—	.105 − .310

F = Speed factor for empty belt   Factor used depends on H/L Ratio (Sine of angle of Slope)

C = Idler factor for loaded belt

ACCESSORIES

Torque Arms — The sizes of torque arms given have been selected to correspond to the torque capacity of the holdback itself. Torque arms smaller than listed may result in damage to torque arm under the application of full torque. Torque arms for LLH and HSB series should be restrained but should not be rigidly secured to the stationary framework. (Figure 40) It is important that some free play between the torque arm and its stops be maintained (1.0″ free play is recommended in either direction). Free play for the torque arms is recommended to prevent any possibility of a fight between the bearings in the clutch and the bearings supporting the shaft upon which the clutch is mounted.

Auxiliary Seals — LLH and HSB clutches with auxiliary seals are available for locations in extremely dusty or dirty locations. The auxiliary seals provide an additional grease-filled labyrinth to provide additional protection for the seals in the clutch itself. The auxiliary seals may be easily cleansed by pumping additional grease into the labyrinth area to purge out contaminated grease to insure continued protection of the clutch.

The HSB models come complete with a large capacity reservoir which has an oil sight gauge and a breather cap built on to it. For LLH models a sight gauge attachment incorporates a breather and permits a visual check of the oil level at all times. The sight gauge attachment is a mandatory requirement on LLH-700 through LLH-5000 (Figure 40).
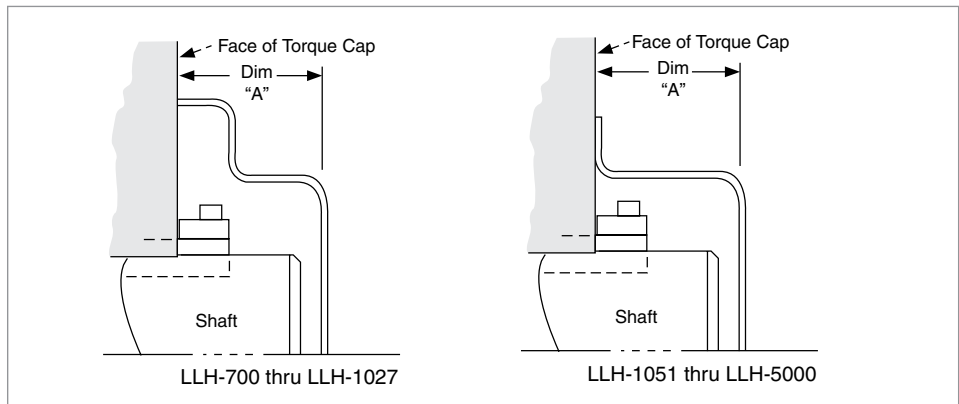
OSHA Cover Kit

OSHA Covers — End cover kits for shaft end mounted LLH units are available. These cover kits provide not only the stationary cover enclosure required by OSHA, but additional protection for the LLH from abrasive environments as well (Figure 41).

Clutch	End Cover Part No.	Dimension "A" Inches
LLH-700	CL-7891	2.00
LLH-750	CL-7892	2.00
LLH-800	CL-7893	2.00
LLH-900	CL-7894	2.00
LLH-1027	CL-7895	2.50
LLH-1051	CL-7896	2.50
LLH-1250	CL-7897	2.50
LLH-1300	CL-7898	2.50
LLH-1375	CL-7899	3.00
LLH-2000	CL-7900	3.00
LLH-2400	CL-7901	Consult
LLH-3500	CL-7902	Formsprag
LLH-5000	CL-7902	Clutch

Dimension "A" is distance allowed from face of torque cap to inside face of end cover.

Stub Shaft Adapter

Stub Shaft Adapters — Stub shaft adapters are available to fit clutch models 400 through 1027 (Figure 42). They are secured to the clutch by fastening to the tapped holes at the end face of the outer race. The stub shaft adapters may be used to mount gears, pulleys or similar attachments when the item to be mounted is too small to mount on the O.D. of the clutch, and to mount couplings when the customer desires to use a coupling other than the Formsprag clutch coupling assemblies.

2″ Total Free Play

Figure 40

Face of Torque Cap
Dim "A"
Shaft
LLH-700 thru LLH-1027

Face of Torque Cap
Dim "A"
Shaft
LLH-1051 thru LLH-5000

Figure 41

Keyseat
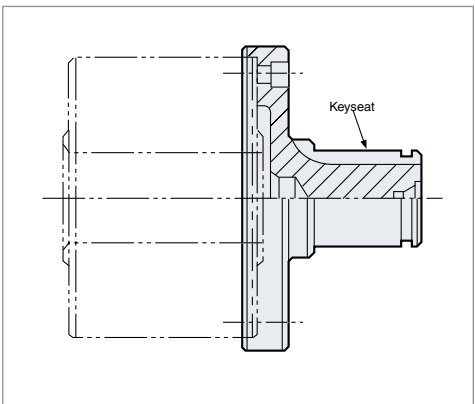
Figure 42

ACCESSORIES

Restraint Keys

Axial Restraints — All holdbacks must be axially restrained when mounted on the head shaft. The recommended bore to shaft fits are loose to facilitate installation and removal. Holdbacks installed without axial restraint can shift on the shafts, causing bearing loads which can significantly reduce bearing life. Set collars along with single and double block restraint keys are available to provide a simple means to restrain a holdback on a head shaft (Figure 43).

Series 20 and 50

Formsprag Clutch backstops may be installed on extended shafts of gear reducers, or they may be built into a reducer at the time of original assembly. The Formsprag Clutch backstops as built into reducers may be designed expressly for particular reducers, or they may be built around the Formsprag "Series 20 and 50" line of built-in backstops (Figure 44). The Series 20 and 50 line of built-in backstops is available to original equipment manufacturer of reducers for use as built-in backstops.

The Series 20 and 50 line of backstops are comprised of an outer race, a set of sprags with energizing springs, and snap-rings and washers to hold the sprag complement in the outer race.

The customer's reducer shaft must be hardened to act as an inner race for the Series 20 and 50 clutch and concentricity is provided by the bearings which support that shaft. The Series 20 and 50 design is not a complete self-contained clutch.

Information on the Series 20 and 50 line or a specific use on built-in backstops is available from the Formsprag Clutch factory.

Check Clutches

A specialized field for backstops is in the use of clutches for check clutches and indexing mechanisms (Figure 45). An overrunning clutch used as a backstop or check clutch in an indexing mechanism can customarily be far smaller than the clutch used for indexing the same mechanism. Although the indexing clutch must be able to handle the inertias of the entire system being indexed, plus overcoming stock pull and any friction brake used for prevention of overthrow—the check clutch need only be large enough to resist any tendency toward backpull on the part of the material being indexed.

The check clutch will hold intermittently when resisting such backpull, and will overrun during the feed portion of the indexing cycle. For best results,

the check clutch should be mounted on the feed shaft of the indexing mechanism.

Figure 43

Figure 44

Figure 45

BORE SIZE/SHAFT TOLERANCES

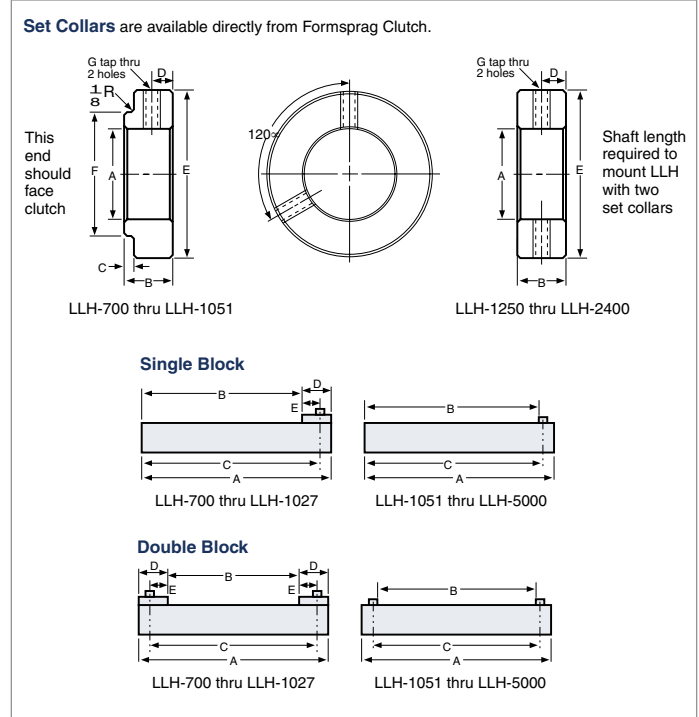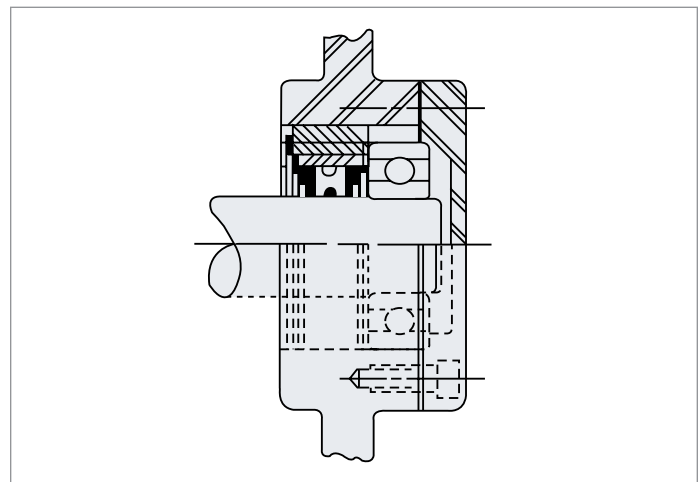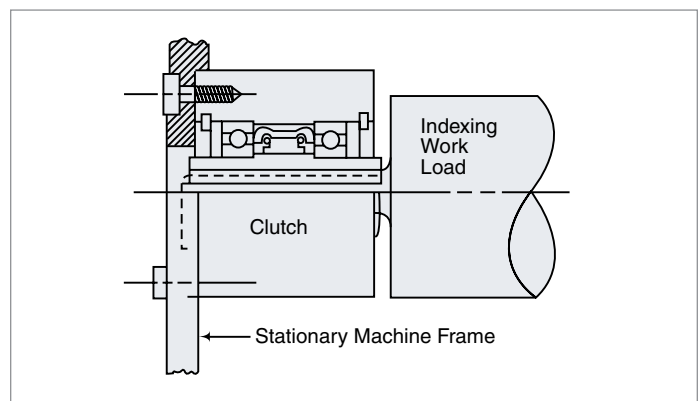The clutch bore size and shaft tolerance listed in Tables 4 and 5 are the recommended fit for each size clutch. It should be noted that the bore and shaft size used with Formsprag clutches do not necessarily coincide with the tolerance recommended by other sources. If a press fit is necessary, never exceed .001 inch interference fit. For all bore and shaft sizes not listed below use same tolerance and fit as the next larger size.

Note: To obtain the metric equivalent for the dimensions listed in Tables 4 and 5 multiply by 25.4.

Table 4 - Sleeve bearing clutches

Series	Nominal Bore	Bore Diameter	Shaft Diameter
FS	.250	.250/.252	.250/.249
	.375	.375/.377	.3745/.374
	.500	.500/.502	.4995/.499
	.625	.625/.626	.6245/.6240
FSR	.375	.375/.376	.3745/.374
	.500	.500/.501	.4995/.499
	.625	.625/.626	.6245/.624
	.687	.687/.688	.6865/.6860
	.750	.750/.751	.7495/.749
	.875	.875/.876	.8745/.874
	1.000	1.000/1.001	.9995/.999
	1.125	1.125/1.126	1.1245/1.1235
	1.250	1.250/1.251	1.2495/1.2485
	1.375	1.375/1.376	1.3745/1.3735
	1.500	1.500/1.501	1.4995/1.4985
	1.625	1.625/1.626	1.6245/1.6235
	1.750	1.750/1.751	1.7495/1.7485
	1.875	1.875/1.876	1.8745/1.8735
	2.000	2.000/2.001	1.9995/1.9985
	2.187	2.187/2.188	2.1865/2.1855

Table 5 - Ball bearing clutches (excpet LLH series)

Series	Nominal Bore	Bore Diameter	Shaft Diameter
	.500	.499/.500	.499/.498
	.625	.624/.625	.624/.623
	.750	.749/.750	.749/.748
	.875	.874/.875	.874/.873
	1.000	0.999/1.000	.999/.998
	1.125	1.124/1.125	1.124/1.123
	1.250	1.249/1.250	1.249/1.248
	1.312	1.311/1.312	1.311/1.310
	1.375	1.374/1.375	1.374/1.373
	1.500	1.499/1.500	1.499/1.498
	1.625	1.624/1.625	1.624/1.623
	1.750	1.749/1.750	1.749/1.748
	1.937	1.9365/1.9375	1.9365/1.9355
5	2.000	1.999/2.000	1.999/1.998
	2.125	2.124/2.125	2.124/2.123
	2.250	2.2485/2.2500	2.2485/2.2475
	2.437	2.4360/2.4375	2.4360/2.4350
1	2.437	2.4365/2.4375	2.4365/2.4355
	2.500	2.4985/2.5000	2.4985/2.4975
	2.625	2.624/2.625	2.624/2.623
	2.750	2.7485/2.7500	2.7485/2.7475
	2.937	2.9360/2.9375	2.9360/2.9350
	3.000	2.9985/3.0000	2.9985/2.9975
	3.250	3.2485/3.2500	3.2485/3.2475
	3.437	3.4360/3.4375	3.4360/3.4350
	3.500	3.4985/3.5000	3.4985/3.4975
	3.750	3.7485/3.7500	3.7485/3.7475
	3.937	3.9360/3.9375	3.9360/3.9350
6	3.937	3.9355/3.9370	3.9355/3.9345
	4.000	3.9985/4.0000	3.9985/3.9975
2,6	4.000	3.998/4.000	3.998/3.997
	4.250	4.248/4.250	4.248/4.247
2	4.250	4.2485/4.2500	4.2485/4.2470
	4.437	4.4355/4.4375	4.4355/4.4345

Series	Nominal Bore	Bore Diameter	Shaft Diameter
2	4.437	4.4360/4.4375	4.4360/4.4350
	4.500	4.498/4.500	4.498/4.497
	4.750	4.748/4.750	4.748/4.747
	4.937	4.9355/4.9375	4.9355/4.9345
2	4.937	4.9360/4.9375	4.9360/4.9345
	5.000	4.998/5.00	4.998/4.997
4	5.000	5.002/5.004	5.001/5.000
	5.250	5.248/5.250	5.248/5.247
	5.437	5.4355/5.4375	5.4355/5.4345
3	5.437	5.435/5.437	5.435/5.434
	5.500	5.498/5.500	5.498/5.497
	5.750	5.748/5.750	5.748/5.747
	5.937	5.9355/5.9375	5.9355/5.9345
3	5.937	5.9360/5.9375	5.9360/5.9350
	6.000	5.998/6.000	5.988/5.997
	6.250	6.248/6.250	6.248/6.247
	6.437	6.4355/6.4375	6.4355/6.4345
3	6.437	6.436/6.438	6.436/6.435
	6.500	6.498/6.500	6.498/6.497
	6.750	6.748/6.750	6.748/6.747
	6.937	9.9355/6.9375	6.9355/6.934
	7.000	7.003/7.005	7.002/7.001
	7.500	7.504/7.506	7.503/7.502
	7.750	7.754/7.756	7.753/7.752
	8.000	8.004/8.006	8.003/8.002
	8.250	8.254/8.256	8.253/8.252
	8.500	8.504/8.506	8.503/8.502
	8.750	8.754/8.756	8.753/8.752
9.000	9.004/9.006	9.003/9.002	
	9.250	9.254/9.256	9.253/9.252
	9.500	9.504/9.506	9.503/9.502
	9.750	9.754/9.756	9.753/9.752
	10.000	10.004/10.006	10.003/10.002
	10.500	10.504/10.506	10.503/10.502
	11.000	11.004/11.006	11.003/11.002
	11.500	11.504/11.506	11.503/11.502
	12.000	12.004/12.006	12.003/12.001
	12.250	12.254/12.256	12.253/12.251
	12.500	12.504/12.506	12.503/12.501
	13.000	13.004/13.006	13.003/13.001
	13.250	13.254/13.256	13.253/13.251
	13.500	13.504/13.506	13.503/13.501
	13.750	13.754/13.756	13.753/13.751
	14.000	14.004/14.006	14.003/14.001
	14.250	14.254/14.256	14.253/14.251
	14.500	14.504/14.506	14.503/14.501
	14.750	14.754/14.756	14.753/14.751
	15.000	15.004/15.006	15.003/15.001
	15.250	15.254/15.256	15.253/15.251
	15.500	15.504/15.506	15.503/15.501
	15.750	15.754/15.756	15.753/15.751
	16.000	16.004/16.007	16.003/16.001
	16.250	16.254/16.257	16.253/16.251
	16.500	16.504/16.507	16.503/16.501
	16.750	16.754/16.757	16.753/16.751
	17.000	17.004/17.007	17.003/17.001
	17.250	17.254/17.257	17.253/17.251
	17.500	17.504/17.507	17.503/17.501
	17.750	17.754/17.757	17.753/17.751
	18.000	18.004/18.007	18.003/18.001
	18.250	18.254/18.257	18.253/18.251
	18.500	18.504/18.507	18.503/18.501
	18.750	18.754/18.757	18.753/18.751
	19.000	19.004/19.007	19.003/19.001
	20.000	20.004/20.007	20.003/20.001

1 Model 750 only
2 Model 900 only
3 Model 1027 only
4 Model 1051 only
5 FWO-600/2.00 tolerance 2.000/2.001
6 Model 800 only
Note: On Models 750 thru 5000, Formsprag Clutch may elect to supply a stepped key in the event of keyseat distortion during heat treat of inner race.

EQUATIONS

Clutch torque based on known speed and horsepower $T = \dfrac{HP \times 5250}{RPM}$	**in which:** T = torque, pound feet HP = horsepower RPM = revolutions per minute
Clutch torque — angular harmonic motion — (indexing) $T_t = \dfrac{N^2 \emptyset}{5225}\; I\; +\; T_B$	**in which:** T_t = Total torque, pound inches N = indexes per minute Φ = angular motion of clutch per index, in degrees I = mass moment of inertia of load, pound inches sec.^{2} (see below) T_B = Brake Torque, pound inches, not included in first term, engaged brakes, feeding strip, friction, etc.
Mass Moment of Inertia For Solid Cylinder or Roll For Hollow Cylinder or Roll $I = \dfrac{\pi\, d\; L\; R^4}{2g}$ $I = \dfrac{\pi\, d\; L\; (R^4 - r^4)}{2g}$	**in which:** I = mass moment of inertia of load, pound inches sec.^{2} d = density, lbs/in.^{3} L = cylinder length, inches R = outside radius, inches r = inside radius, inches g = acceleration due to gravity = 386 inches per sec.^{2}
For Train of Cylinders or Rolls I effective $= I_1 + I_2 \dfrac{r_1^2}{r_2^2} + I_3 \dfrac{r_1^2}{r_3^2} + \ldots + I_\eta \dfrac{r_1^2}{r_\eta^2}$	**in which:** I_1 = mass moment of inertia of first roll, pound inches sec.^{2} I_2 = mass moment of inertia of second roll, pound inches sec.^{2} I_η = mass moment of inertia of ηth roll, pound inches sec.^{2} r_1 = radius of first roll, inches r_2 = radius of second roll, inches r_η = radius of ηth roll, inches
Torsional Shaft Stress — solid shaft $S_s = \dfrac{5.1\, T}{D^3}$ **Torsional Shaft Stress — hollow shaft** $S_s = \dfrac{5.1\, TD}{D^4 - d^4}$	**in which:** S_s = torsional shear stress, PSI D = outside diameter of shaft, inches D = inside diameter of shaft, inches T = torque, pound inches
Ball Bearing L-10 Life $(L\text{-}10)_0 = \left(\dfrac{A}{X_0}\right)^3 \times \left(\dfrac{N}{N_0}\right) \times 10{,}000$	**in which:** $(L\text{-}10)_0$ = New L-10 life, hours X_0 = New load, pounds A = Rated bearing load of a ball bearing clutch, pound N = Maximum overrunning speed of clutch, revolutions per minute N_0 = New overrunning speed of clutch, revolutions per minute

CROSS REFERENCE

Formsprag Clutch™ is the first name in overrunning clutches because Formsprag Clutch is the leading manufacturer of these unique devices. Formsprag Clutch offers superior design, products manufactured to the highest standards and the most complete line of overrunning clutches available, all at competitive prices. This cross reference will help you to specify Formsprag Clutch models when customers are requesting off-brand replacements. Formsprag Clutch unique exchange program is described on the next page. Your customers can start receiving the benefits of superior Formsprag Clutch performance immediately by turning in their old off-brand overrunning clutches for rebuilt Formsprag Clutch units with a new product warranty. If you have any questions about these products or the exchange program, please contact your Formsprag Clutch Customer Service Representative at 810-758-5000 or 1-800-927-3262.

Bold type designates units which can be rebuilt on Formsprag Clutch Exchange Program.

Cross Reference Part Numbers				
Formsprag Clutch	**Morse™**	**Renold**	**Ringspann**	**Marland™**
FS-02	—	SA-02	—	—
FS-04	—	SA-04	—	—
FS-05	—	SA-05	—	—
FSR-3	PB-3A	SB-3	—	—
FSR-5	PB-5A	SB-5	—	—
FSR-6	**PB-6A**	**SB-6**	—	—
FSR-8	**PB-8A**	**SB-8**	—	—
FSR-10	**PB-10A**	**SB-10**	—	—
FSR-12	**PB-12A**	**SB-12**	—	—
FSR-14	**PB-14A**	**SB-14**	—	—
FSR-16	**PB-16A**	**SB-16**	—	—
FSO-300	**MG-300A**	—	**FRSG-300**	—
FSO-400	**MG-400A**	***SO-400**	**FRSG-400**	****RMS-14**
FSO-500	**MG-500A**	***SO-500**	**FRSG-500**	****RMS-21**
FSO-550	**—**	**—**	**FRSG-550**	**RMS-26**
FSO-600	**MG-600A**	***SO-600**	**FRSG-600**	****RMS-32**
FSO-650	**—**	**—**	**FRSG-650**	**RMS-40**
FSO-700	**MG-700A**	***SO-700**	**FRSG-700**	****RMS-47**
FS-750	**MG-750A**	***SO-750**	**FRSG-750**	****RMS-55**
FS-800	**MG-800A**	***SO-800**	**FRSG-800**	****RMS-72**
FS-900	**MG-900A**	***SO-900**	**FRSG-900**	**—**
FS-1027	**MG-1000A**	***SO-1000**	—	—
HPI-300	**MI-300A**	—	**FRS-300**	**—**
HPI-400	**MI-400A**	**SX-400**	**FRS-400**	**RMS-14**
HPI-500	**MI-500A**	**SX-500**	**FRS-500**	**RMS-21**
—	—	—	**FRS-550**	RMS-26
HPI-600	**MI-600A**	**SX-600**	**FRS-600**	**RMS-32**
—	—	—	**FRS-650**	RMS-40
HPI-700	**MI-700A**	**SX-700**	**FRS-700**	**RMS-47**
HPI-750	**MI-750A**	**SX-750**	**FRS-750**	**RMS-55**
HPI-800	**MI-800A**	**SX-800**	**FRS-800**	**RMS-72**
HPI-900	**MI-900A**	**SX-900**	**FRS-900**	**—**
HPI-1027	**MI-1000A**	**SX-1000**	**—**	**—**
FS-100	HT-10	—	—	—
FS-200	HT-20	—	—	—
FS-300	HT-30	—	—	—
FSO-750	M0-750A	—	—	—
FSO-800	M0-800A	—	—	—
FSO-900	M0-900A	—	—	—
FSO-1027	M0-1000A	—	—	—
HPO-720	—	—	—	—
HPO-750	MR-750A	—	—	—
HPO-800	MR-800A	—	—	—
HPO-900	MR-900A	—	—	—
HPO-1027	MR-1000A	—	—	—

* Also available for end face GR lubrication, F/S has no interchange.
** Also available Model RMZ

EXCHANGE PROGRAM

- Return any competitive unit listed below in a standard bore size with a purchase order and shipping instructions and receive a superior Formsprag clutch.

- All Formsprag rebuilt clutches come with a new clutch warranty at a reduced price.

- Refer to the Distributor order book for Formsprag rebuilt clutch pricing.

- Contact Formsprag Clutch at 586-758-5000 for additional information on the Formsprag Clutch Exchange and Rebuild Programs.

Part Numbers		
Formsprag Clutch	**Morse**	**Renold**
FS-100	HT-10	N/A
FS-200	HT-20	N/A
FS-300	HT-30	N/A
FSR-6	PB-6	SB-6
FSR-8	PB-8	SB-8
FSR-10	PB-10	SB-10
FSR-12	PB-12	SB-12
FSR-14	PB-14	SB-14
FSR-16	PB-16	SB-16
FSO-300	MG-300	N/A
FSO-400	MG-400	SO-400
FSO-500	MG-500	SO-500
FSO-600	MG-600	SO-600
FSO-700	MG-700	SO-700
FSO-750L	MG-750	SO-750L
FSO-800	MG-800	SO-800
FSO-900	MG-900	SO-900
FSO-1027	MG-1000	SO-1027
HPI-300	MI-300	N/A
HPI-400	MI-400	SX-400
HPI-500	MI-500	SX-500
HPI-600	MI-600	SX-600
HPI-700	MI-700	SX-700
HPI-750	MI-750	SX-750
HPI-800	MI-800	SX-800
HPI-900	MI-900	SX-900
HPI-1027	MI-1000	SX-1027

REBUILD AND OVERHAUL SERVICE

Formsprag Clutch's Service Department can refurbish your overrunning clutches. We take your worn, scaled, seized, scored, or rough running clutches and make them like new…for substantially less than the replacement cost.

As a major manufacturer of overrunning clutches, we have the people, facilities, and experience to rebuild and replace all components as needed. The same standards of excellence that exist in the manufacture of original equipment clutches are closely adhered to in the overhaul procedure of every rebuilt clutch.

The Rebuild Service Department will perform the following procedures to restore your overrunning clutches to original specifications with a new warranty:

- Clean
- Inspect
- Rework worn races or…
- Replace unsalvageable races
- Replace sprags
- Replace bearings and other internal components
- Retrofit with design improvements

We're also interested in helping you prevent problems and failures. We can recommend a preventive maintenance schedule for all your clutches.

Contact your Warner Electric representative or our Formsprag Clutch Plant for details on our cost-saving program. **1-800-927-3262**

Overrunning Clutches Faxable Application Data Form

Fax (586) 758-5204   For Application Assistance call 1-800-927-3262

Date _____

_____
Company name

_____
Address

_____
City   State   Zip

_____
Name of contact   Title

_____
Phone   Fax

Type of equipment

_____

_____

Type of application

- ☐ Overrunning
- ☐ Backstop
- ☐ Indexing
- ☐ Clutch Coupling

Maximum torque at clutch

_____ pound-feet, or

_____ H.P. at _____ RPM

Power Source

- ☐ Electric motor
- ☐ Turbine
- ☐ Gasoline engine
- ☐ Diesel engine
- ☐ Air cylinder

Load Application

- ☐ Smooth
- ☐ Moderate
- ☐ Shock

Lubrication

- ☐ Runs in oil
- ☐ Accessible for lubricating
- ☐ Not accessible

Type or specification of lubricant

_____

For overrunning or backstop applications

Inner race speed during overrunning _____ RPM max.

Outer race speed during overrunning _____ RPM max.

If both members are rotating during overrunning, are they rotating in the

- ☐ Same direction
- ☐ Opposite directions

Time cycle of Formsprag clutch

Drive_____minutes   Rest_____minutes

Over-run_____minutes

For indexing applications

Indexes per minute_____max.

Degrees per index_____max.

Clutch operating time

_____ hours per day

Shaft diameter (give limits)

_____

_____

Size of keyseat in shaft

_____

Environment

Temperature range _____°F to _____°F

Exposed location? _____

Radiation? _____

For non-symmetrical clutches and clutch-couplings

a. Identify the end from which the clutch is viewed

_____

b. The (inner, outer) member (drives, over-runs) in the (clockwise, counter-clockwise) direction

Anticipated quantity required

a. For this application

_____

b. Annually

_____

Supply a sketch of your installation.

Note: A clutch is not a coupling. When necessary to couple two shafts, a coupling must be used with the clutch. Ambient temperature should not exceed approximately 150°F. It is requested that prints of installation be supplied if available.

_____
Completed by

FORMSPRAG CLUTCH™

Regal Rexnord

www.formsprag.com

North America - USA
AmericaUSA23601 Hoover Road
Warren, MI 48089, USA
586-758-5000

Europe - Germany
Hatschekstraße 36
69126 Heidelberg, Germany
+49 (0) 6221-3047

Asia Pacific
Australia: +61 2 9894 0133
China: +86 21 5169-9255
Hong Kong: +852 2615 9313
Singapore: +65 6487 4464
Taiwan: +886 2 2577 8156
Thailand: +66 2322 5527

regalrexnord.com

RegalRexnord™