

# HIGH SPEED BACKSTOPS IN CONVEYOR INSTALLATIONS

Applications, Design, Selection and Calculation of Dynamic Peak Torques\*)

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## 1. Backstops - Useless Machine Elements, or...???

During normal operation of conveyor installations the motors, gears, bearings and many other components are working continuously, thus 'earning their money'. Backstops, on the other hand, remain inactive for most of the operating time and can therefore be thought of as a useless, unnecessary accessory. This statement is refuted by the manufacture of tens of thousands of backstops each year, and their attachment to conveyor installations where they do their job reliably and mostly unnoticed, only into action when a load needs to be stopped immediately after a drive has failed in order to prevent reverse running. There is no other machine element today which, like the modern backstop, operates completely automatically, is of a simple construction and, with appropriate selection, maintenance free.

Despite of this there is a tendency to unnotice the importance of backstops so that these are often badly chosen and under-dimensioned. In the past this has led to several serious accidents caused by backstop failures. This is not necessary because for every application there is the right solution. There is no shortage in the market of reliable, proven designs by competent manufacturers. This article gives a comprehensive description of applications, types and the selection of modern backstops for conveyor installations.

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\*)This article is dedicated to Dr. Christof Maurer, Bad Homburg, in honour of his 60th birthday.

## 2. Applications of Backstops

All kinds of Conveyor installations are of immense importance for the infrastructure all over the world. Apart from belt conveyors, elevators etc. for the transport of bulks, there are pumps and ventilators serving as the transport medium for fluids or gases. Automatic backstops are provided in many of these conveyor installation to prevent the accidental reverse movement of the load. Sometimes this is even a legal requirement.

The most important conveyor installations for bulks are inclined conveyors, elevators and 'ski lifts'. Inclined conveyors are built to a capacity of 40,000 t/h. The most important conveyor installations for fluids are piston pumps, centrifugal pumps and spiral pumps. Often backstops are also used in installations where several pumps are operating in parallel with the possibility of stopping some of them temporarily. Conveyor installations for gases are compressors, ventilators and vacuum pumps where backstops are also used often.

However, the using of backstops do not end with these applications in conveyor technology. Backstops are also used in the most diverse machines, like textile machines, tool machines, presses, paper processing machines and printing machines. In these the backstops are often applied in indexing devices to prevent the reverse movement of the material to be transported. Nowhere is their application more numerous than in the automatic gears of motor cars in the torque converter. Backstops are also used in mechanical clocks, in cable winches, in office machines etc.

Analogies to mechanical backstops are found in hydraulics in the form of non-return valves and in electronics in the form of diodes.

Backstops should be considered as safety components in inclined conveyor installations, in particular. Backstop manufacturers have to be very conscious of product liability laws and must take whatever precautions are necessary because backstops are components which, if they fail, can cause serious damages.

## 3. Designs of Backstops

From the view of machine elements, the backstop function is one of three possibilities how the machine element 'freewheel clutch' can work. Freewheel clutches are working depend of the rotation direction. The oldest type of which was known even in antiquity in the form of a ratchet

clutch. These were used in hoists and weapons. Today they are still used in cable winches on sports sail boats for example, and also on bicycles. However, ratchet clutches are no longer of much practical use in modern machines because their torque capacity is low but most of all because they have a relatively large idle running angle of rotation in the locking direction before the load comes to a stop. The most common types of backstops nowadays are the sprag and roller backstops. Generally, the advantage of the sprag principle regarding power transmission is the greater compactness of the sprags. However, the most important factor is that only with the sprag principle a centrifugal lift-off can be achieved of. Centrifugal lift-off means operation without contact and therefore no maintenance. [3] and [5].

Depending on the way backstops are fitted to motors and gears, they differ between floating backstops (Fig. 1) and built-in backstops (Fig. 2). Floating backstops according to ill. 1 are mounted on an extended shaft end of a motor or gear. These backstops are independent machine elements, with own bearings, oil filling, seals and a torque arm.

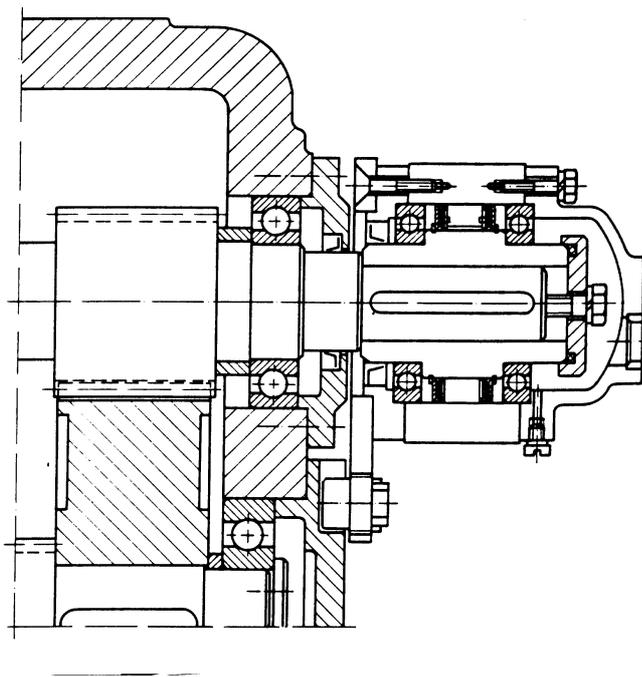


Fig. 1 Floating Backstop

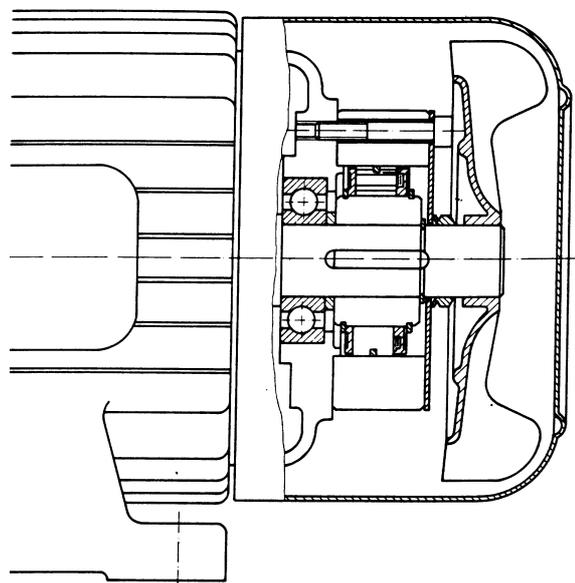


Fig. 2 High-Speed Built-in Backstop

The built-in backstops as shown in Fig. 2 have become the dominant type in Europe, particularly in connection with electromotors and gears because this backstop design is the simplest with the lowest costs. Here the inner ring with the sprag cage is fitted on the B-end of the shaft of the electromotor or on an extended shaft end of the gear. The outer ring is screwed directly to the housing of the motor or the gearbox. Often a cover is screwed on in such a way that the backstop

becomes an integral part of the gear. Nowadays such backstops are always designed with sprags capable for centrifugal lift-off - more about this in chapter 4.

Relative to the speed there are differentiates between the types of slow and fast running backstops, [3] and [6]. In particular, this means that slow running backstops are arranged on the head drum shaft of - for example - belt conveyors, whereas fast running backstops are arranged on the second or third drive shaft with correspondingly higher speeds, see ill. 2. Slow running backstops must always have considerably larger dimensions because according to the law of conservation of energy low speeds mean high torques, thus making these backstops correspondingly large and expensive.

In the case of multimotor drives on belt conveyors and elevators, for example, one is always confronted by the problem of even load distribution over the built-on backstops. There are many mechanisms to make the load distribution more even but there is only one design which truly meets this particular demand: the backstop with integral torque limiter [3] shown in Fig. 3.

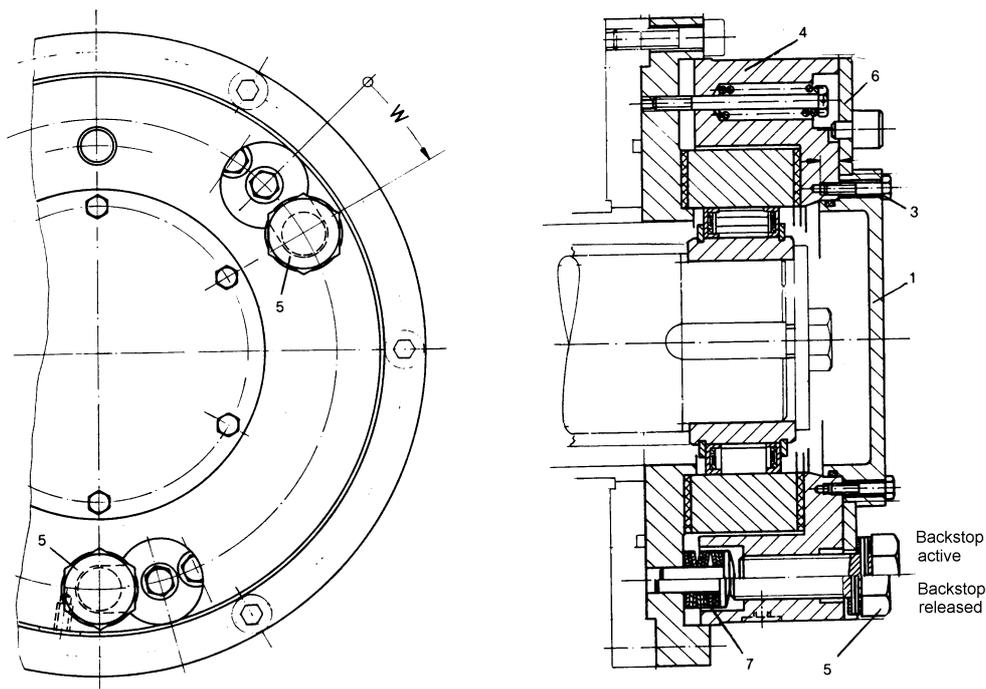


Fig. 3 Backstop Type FXRT with Integral Torque Limiter and Release Facility

Here the same backstopping principle applies as with the fast running built-on backstops of ill. 2, with the exception that the outer ring is not screwed on but is held captive via bilaterally arranged friction linings which are pressed on by a spring pack. This design allows the reduction of peak torques and by brief slipping in the locking direction facilitates a slipping compensation between

the individual backstops. This distributes the load evenly over all the backstops which can therefore be designed to smaller dimensions.

This backstop with torque limiter shown in ill. 3 has a finely controlled release facility so that in case of locking the tension in the conveyor belt can be reduced where necessary. Such backstops can also be supplied without the release facility.

#### 4. The Most Modern Type of High-Speed Backstop

##### 4.1 Centrifugal Lift-off with Rotating Inner Ring

With the invention [1] in 1970 of the centrifugal lift-off with rotating inner ring, it became possible for the first time to arrange small dimensioned backstops with low lift-off speeds directly onto fast running shafts. Since then many experience has been gained from tens of thousands of differently sized backstops of this type, resulting in the development of a completely new generation of backstops [5] where optimum utilisation of the material achieves with high torque capacity at very low lift-off speeds. The clever geometrical design of the sprags can compensate even a large tolerance between shaft and outer part.

##### 4.2 Specifications of Backstops with Centrifugal Lift-off

In brief the specifications for backstops can be summarised, as follows:

- simple design with as few components as possible
- minimum installation dimensions with maximum torque capacity
- no wear and no maintenance
- robust construction
- high temperature stability
- large misalignment capability of the shaft in the bearings
- large tolerances of adjacent parts
- unimpaired function with oils with friction-reducing additives
- cost effective

In order to meet all these specifications the most common solution nowadays is provided by the 'freewheel with centrifugal lift-off with rotating inner ring'. This does necessitate the use of cage-supported sprags. The requirement for robustness and temperature stability can only be met by

metallic materials whereby the power transmitting parts must be made of hardened steel to satisfy the requirement for small dimensions.

Centrifugal lift-off 'X' with rotating inner ring is the most elegant method for building an extremely robust, reliable and maintenance free backstop with minimum costs. An inner ring together with the sprag cage is fitted to a gear shaft end. The outer ring is screwed to the gear housing, either directly or via an intermediate flange - see Fig. 4. During normal operation, i. e. rotation of the shaft, the centrifugal effect causes the sprags to swing inward so that above the lift-off speed they rotate within the outer ring without any contact - see Fig. 6. This means a virtually unlimited life for such backstops.

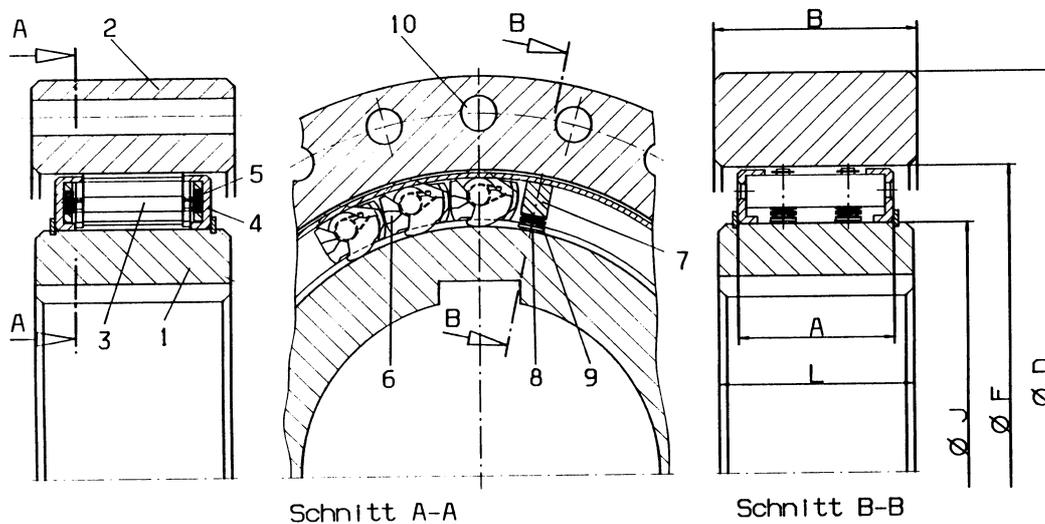


Fig. 4 Backstop FXM with New Type of Cage

The brief and rarely periods of starting and stopping do not normally cause much wear because in most cases the backstop is at least splash-lubricated by the oil from the gear. At low speeds there is practically no contact between metal parts. In this context it is important to note that many gear oils contain friction reducing additives which needs be taken into consideration for the design of the clamping geometry.

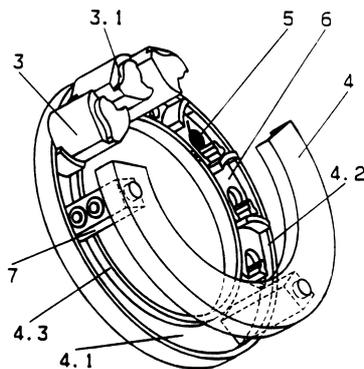


Fig. 5 Perspective Drawing of the Cage

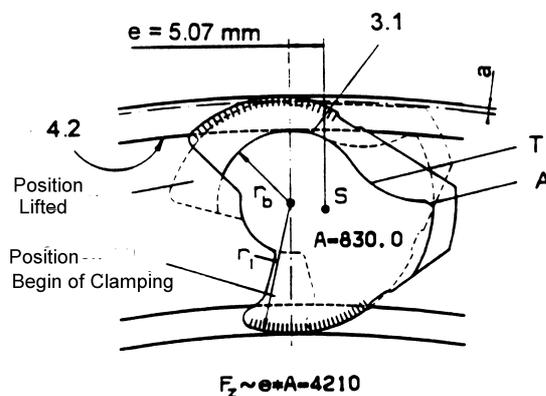


Fig. 6 Sprag with Centrifugal Lift-off 'X'.

### 4.3 Design of the new cage-freewheel

Fig. 4,5 and 6 show the construction of the new build-in backstops with the new patented cage with centrifugal lift-off "X" at a rotating inner ring.

#### Circumference Support of the Sprags

As Fig. 5 show the freewheelcage consist two sturdy U-formed rings which are riveted together with connecting-pins 7. At the ground 4.1 of the rings 4 are welded guide pockets. These guide pockets are very close together because a maximum number of sprag must be placed in the cage. At the beginning of locking the guide pockets are taking care to hold the sprag in axially parallel position because this is necessary for a maximum torque capacity. On the other hand the sprag need a support to tilt in the right lift-off position without additional forces.

#### Radial Support of the Sprags

The efficient design of the profile 3.1 of both sprag ends is important for the lift-off process and the centrifugal lift-off - Fig. 5 and 6. This contour has a radius  $r_b$  in the virtual contact width with the inner radius of the cage rings 4.2 which is concentric to the inner sprag radius  $r_i$ . This contour transmits the centrifugal force which is acting on the sprag to the cagerings and allow the sprag to rotate at the same time. This rotation - caused by the effect of the centrifugal force - is limited by stop A which is found further along the contour and which also acts against the inner radius of the cage rings 4.2 when the sprag is tilted into the disengaged position. The recess T serves as a for the hook of the sprag spring between the sprag contour and the supporting surface of the cage ring.

These two support contours are milled to exact specifications with a minimum of metal cutting work in suitable devices. This process is less complicated than the former standard procedures when large sections of material had to be cut away from the ends of the sprags.

#### Spring Action of the Sprags

Arranged in guide pocket on both sides of the sprag are single springs. The guide pockets 6 are formed in such a way that they have an opening for the sprag spring 5 and the slot for the spring hook support. Therefore no extra space is needed for the spring. The other hook of each spring engages with each side of the sprag and holds it in the locked position. This particular arrangement of the springs causes the spring force to act on the sprag basically in a radial direction, i. e. the spring force directly opposes to the centrifugal force which is acting on the sprag. This prevents additional friction forces which might hinder the free movement of the sprag as it tilts.

## Contour of the Sprag

Fig. 5 and 6 show a recess on the left side of the sprag. This recess was chosen to gain the largest possible pitch spacing  $e$ . Furthermore, the contour was extended as far as possible to the right in order to increase the cross sectional area of the sprag and the weight. These two features ensure that the result of centre of gravity distance, multiplied by the cross-sectional area, is as large as possible to minimise the lift-off speed. Of course the recess is shaped to ensure that the sprag retains its full stability if load is applied in case of loading.

## Synchronisation of the Cage

For a perfect centrifugal lift-off with rotating inner ring during the accelerating phase and to ensure the re-engagement of the sprags during the deceleration phase, the cage must have a synchronised connection with the inner ring. This is achieved by the synchronising bolt 8 which is pressed into each of the cage connecting-pins 7. They are made of hardened steel and are pressed against the inner ring with a high radial force via plate springs 9; no additional axial space is needed. This connection must be dimensioned to prevent any relative movement of the cage against the inner ring during acceleration or deceleration. On the other hand this frictional synchronous connection does permit a very slight relative movement of the cage against the inner ring during the roll-in movement of the sprags.

## 5. Determination of the right size of Backstops

### 5.1 General

As with all machine elements, when deciding the size of backstops, two factors need always to be taken into consideration:

- maximum **transmissible** torque of the backstop
- maximum **occurring** torque of the machine or plant

It is of course important that in order to avoid torque of the backstop, the maximum **transmissible** torque capacity of the backstop must at all times of its operation be **higher** than the torque **occurring** in the machine or the plant at the backstop. The currently safest method of calculation according to [2] to determine the maximum transmissible torques of backstops - even with eccentric races - is described in chapter 5.2

It is difficult to determine the actual torques in the machine or plant for the backstopping situation. It is often a highly dynamic process where high peak torques occur, as for example the stopping of an inclined belt conveyor under load, an elevator or a pump. The safest method of predicting these torques in the event of a locking action is by a torsional vibration calculation of the whole system, although this requires that all the important data regarding torsional mass and torsional stiffness as well as the damping characteristics of the different elements of the system are known. Chapter 5.3 goes into this in more detail.

As is usual in the machine calculation, safety factors are used when deciding the dimensions for backstops. In this context the basic rule is that as part of the whole system the backstop must never be the weakest link in the chain. If damage does occur, it would always be the worst when a backstop breaks down. The total potential energy stored in the load would then be released and transferred into kinetic energy - with catastrophic consequences. For example, in the case of a backstop/gear combination it may be better for the operator of a conveyor plant if an overload causes a gear teeth deformation rather than backstop failure because the probability of the load still being held by a damaged gear is considerably greater.

A further consideration in the calculation of torques in the machine or plant must be given to the fact that the dynamic system present during the locking situation is different to that which applies under normal operating conditions. It is therefore completely wrong to dimension a backstop by using the nominal torque of the motor. More about this in chapter 5.3, 5.4 and 6.

## 5.2 Maximum Transmissible Torques of the Backstop

### 5.2.1 Correct Calculation of the transmissible of Torques by Freewheels with Sprags and Rollers

The calculation of the maximum permissible torque of backstops is relative difficult. An exact calculation must take into consideration all geometric and elastic conditions which are subject to considerable changes during the load process. This is where the method for the reliable calculation of maximum permissible torques of freewheels and their torsion characteristics published in [2] has been proved for over twenty years now.

This calculation method for the transmissible torque  $M$  of backstops, with concentric inner and outer rings is based on the simple equation:

$$M = z \cdot R_i \cdot F_{Ni} \cdot \tan \varepsilon_i$$

In this equation  $z$  represents the number of sprags or rollers,  $R_i$  is the radius of the inner ring race,  $F_{Ni}$  is the normal force between inner ring and sprag or roller and  $\tan \varepsilon_i$  is the tangent of the inner clamping angle  $\varepsilon_i$ , see ill. 7. As one can see from the above equation,  $\tan \varepsilon_i$  is a decisive value for the transmission of torque.  $\tan \varepsilon_i$  is the ratio of circumferential force  $F_{Ti}$  to the radial force  $F_{Ni}$  on the point of contact on the sprag. Under load, through elastic deformations, the position of the sprag in the clamp clearance changes and therefore also the clamping angle. Ill. 8 shows the clamping angle position of a normal sprag in relation to the height of the clamp clearance  $\Delta H$ .

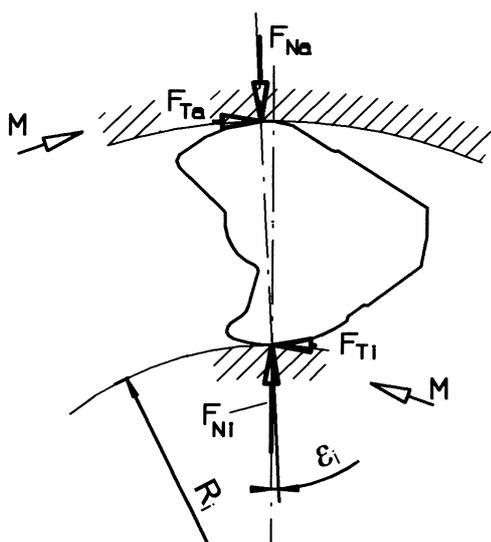


Fig 7 Forces at the Sprag

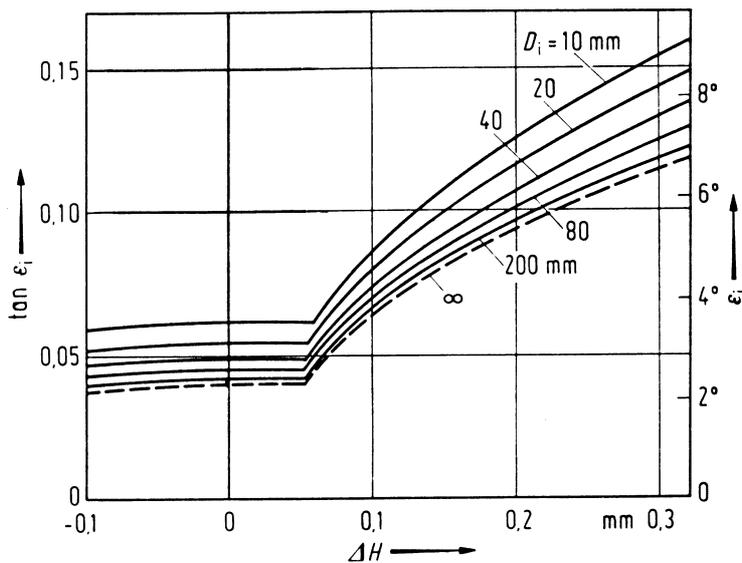


Fig. 8 Course of the Claming Angle  $\tan \varepsilon = f(\Delta H)$

The geometric principle applied to modern sprag design means that with an unloaded freewheel the clamping angle  $\varepsilon_i$  has a value of ca. 3°. Therefore, secure gripping (oil film on the races) during the changeover from freewheeling operation to locking is guaranteed. When the sprags have gripped safely, the clamping angle may be increased which, under otherwise equal conditions, causes the transmissible torque to increase correspondingly. The use of such sprags increases the torque capacity considerably, or it is possible to use fewer sprags. Furthermore, the  $\tan \varepsilon_i$  rising in line with the sprag height effects a stiffer torsion characteristic of the backstop.

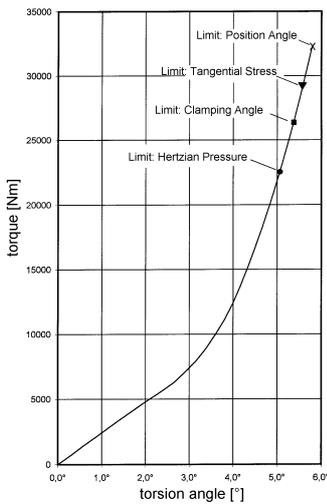
The above mentioned equation for the calculation of torque only applies for concentric rings because only then are the normal Forces  $F_{Ni}$  and the clamping angle the same on each sprag. As the torque load increases the sprag clearance height is altered due to the elastic deformation of all components, so that the position of the sprag is changed too. As this change is continual the calculation is made as follows:

1. Pre-selection of a normal force  $F_{Ni}$
2. Calculation of the sprag clearance height caused by the deformation of the sprag and the rings (polygonal deformation of inner and outer ring)
3. Using the clearance height to calculate the effective clamping angle
4. Using the clearance height to calculate the angular position of the sprag
5. Calculation of the hertzian pressure at the virtual contact point of the sprag
6. Calculation of the total angle of rotation  $\alpha$  between inner and outer ring by changes in the sprag clearance height
7. Calculation of the tangential stress in the outer ring
8. Calculation of the transmissible torque
9. Increasing the normal force  $F_{Ni}$  and back to 1.

After each calculation the limits which cause a backstop failure are checked, as follows:

1. The **Hertzian pressure** which appears between sprag and the races. According to the results of FVA research projects the limit for hard sprags and steeling rings is min. 4000 N/mm<sup>2</sup>
2. The friction ratios may be used only up to the **adherence limit** to prevent a slipping effect. This means that the tangent of the clamping angle must be lower as 0,105.
3. The **tangential stress** in the outer ring which is produced by the high normal forces must be lower than the limit of the material.
4. The elastic deformations cause the sprags to roll under load up to the end of the geometry of the sprags. This borderline case is very important when there is a great sprag clearance at the beginning of clamping, this is the limit of the **position angle**. The additional elastic deformations bring the sprag to an over tilt.

The maximum transmissible torque occurs when one of these limits is reached.



If the force steps were kept low in the calculation algorithm, then each torque is given with its own angle of rotation  $\alpha$ . By joining these points in a diagram the torsion characteristic of the backstop is drawn. It is practical important because it influences the dynamic behave of the plant during sudden load.

Fig. 9 Wind-Up Diagram with Limits of Calculation

Fig. 9 shows the torsion characteristic of a standard backstop. In this case the calculation was continued after reaching the first limit and one can see quite clearly the sequence of arriving at each limit value. With an ideal design all the limits described should be reached simultaneously. That however is rarely possible because for economical reasons one sprag size has to be utilised for a wide range of ring diameters.

### 5.2.2 Torque Transmissibility with Eccentric Races

The procedure in 5.2.1. is based only on perfectly concentric freewheel rings. Eccentricity between the inner and outer races of the backstop can be caused by bearing clearance, shape positional and dimension tolerances of the centerrings. This deviation from the ideal condition in a different sprag clearance height at each point of the circumference. Each sprag adopts a different position. Therefore, as shown in Fig. 10 and 11 work on each sprag a different clamping and position angle at the torque transmission.

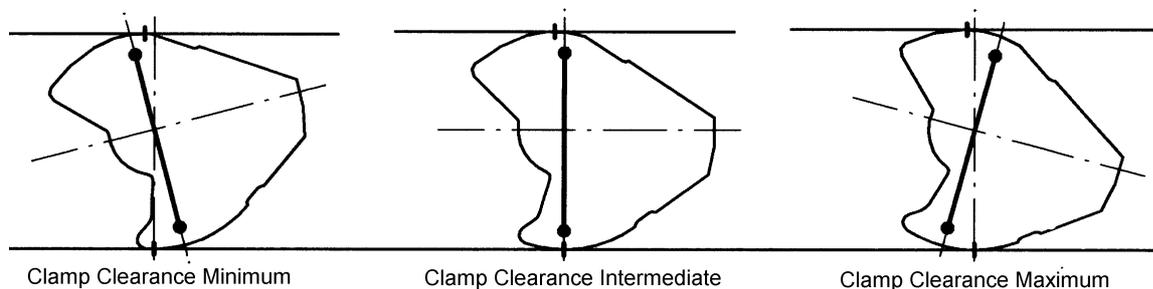


Fig. 10 Position of the sprag in different clamp clearances

This is put simply in Fig. 10 by showing parallel clamping races with clearance heights in three stages. With eccentric freewheel rings the clamping clearance when viewed in circumferential direction increase continually from a minimum to a maximum value and then decrease to the minimum value. This is clearly shown in Fig. 11. Here the inner and outer ring race are eccentric EX to each other so that the same sprag profile has different position- and clamping angles on the circumference. Fig. 11 shows schematically the clamping angle curves with the respective values for the clamping angle for the sprag at the top, the right and the bottom.

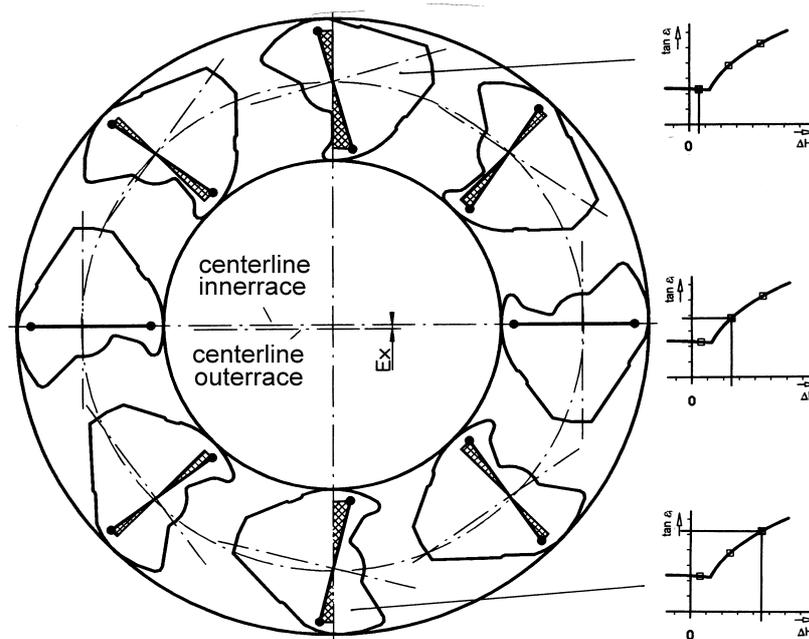
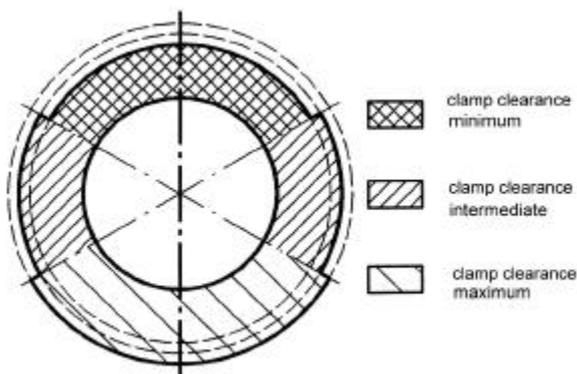


Fig. 11 Sprags between Eccentric Races at Medium Loading

An exact but extremely time-expensing method to calculate the torque capacity with eccentric freewheel rings is described in [2]. The basis for this calculation is very stiff inner and outer rings in the circumferential direction and therefore the **same angle of rotation  $\alpha$**  is forced upon all sprags. With this assumption and the simplified method described below it is possible to reduce the calculation time considerably.



The simplification consists the selecting of 2 x 3 concentric ring areas with 3 different clearance heights in exchange for the eccentric rings, the division is shown in ill. 12. 2 x 3 means that with a vertical axis of symmetry in Fig. 12 ,to the right and left three segments each of 60° with the same clearance height are used.

Fig. 12 Principle Sketch of the Calculaton with Eccentric Races

The three segments each have a minimum, a medium and a maximum clamping clearance which are selected with the eccentricity taken correctly into account. The clamping clearance is easily calculated by addition or subtraction of the eccentricity of the medium clamping clearance.

For the three clearance heights three part-torque-calculations for centered freewheel rings according to 5.2.1 are now carried out. The part-calculation whereby first one of the four limits is reached, gives the torsion characteristic  $M_1 = f(\alpha_1)$ . Then by selecting appropriate angular steps the torques applicable to the angle of rotation  $\alpha_n$  of the other two clearance height calculations are ascertained through interpolation. After that all 3 torques are added up for the same angle to get the arithmetical mean value, thus giving point by point the torsion spring characteristic and the maximum transmissible torque according to the first limit reached in one segment. The other two segments are generally unlikely to reach one limits.

By applying the above calculation method to a standard backstop and by varying the eccentricity, one will see a definite decrease of the torque with increased eccentricity - see Fig. 13.

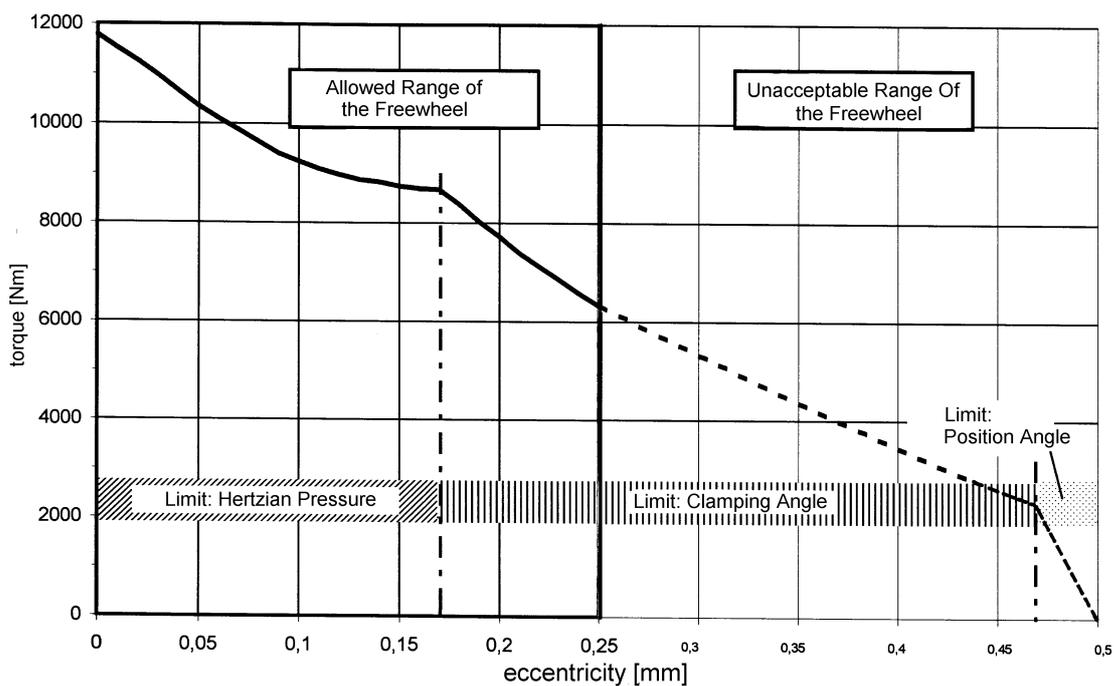
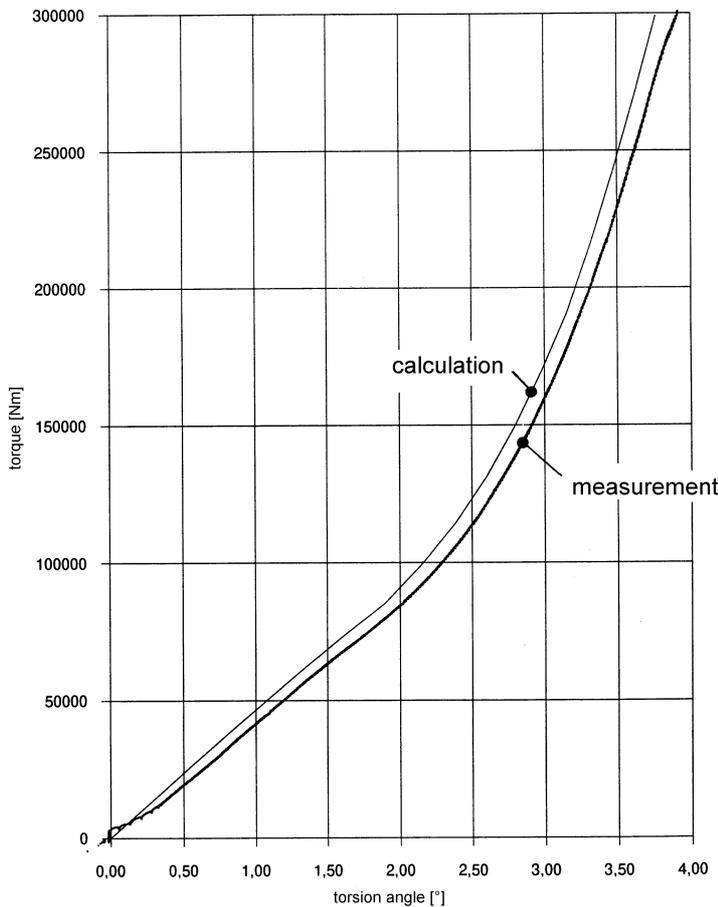


Fig. 13 Permissible Torque of a Standard Backstop over the Eccentricity of the Races

This illustration shows how the different limits take effect and that once the position angle has been reached, torque can, of course, no longer be transmitted. The calculation is on the side of caution because this result is already at the end when the limit of the position angle has been reached in one segment. Another fact shown up in Fig. 13 is that the transmissible torque is not linear. This means that a linear interpolation should never be done between two catalogue values. To ensure a safe design the user must select the lower value or refer to the manufacturer.

The designs discussed in this chapter show that eccentricities between inner and outer ring races must always be taken into consideration of the calculation of the transmissible torques of backstops. Unfortunately, not all manufacturers make this connection clear in their catalogues, although they always use sprag profiles with ascending clamping angle curves.



Finally, shown on Fig. 14 is the calculated and the measured torsion spring characteristic, with eccentricity known.  $EX=0.1$  mm, for one of the largest backstops built so far, with centrifugal lift-off with rotating inner ring, for a large conveyor plant. It is a good illustration of the close match between calculation and actual test results.

Fig. 14 Measure and Calculated Torsion Characteristic

### 5.3 Plant Torques affecting the Backstop

As mentioned in 5.1, the stopping of a conveyor plant involves considerable dynamic processes during which high peak torques occur. The article [4] shows an accurate vibration calculation of the whole system when backstopping. The most significant fact to emerge from this is that non linearities in the torsion spring characteristics of the individual elements of the system can cause unexpectedly high peak torques. Replacing it with a simplified mechanical system (Fig. 15) and the combined torsion spring characteristics (Fig. 16) gives a dynamic torque process in the backstop, taking the damping system into consideration (Fig. 17).

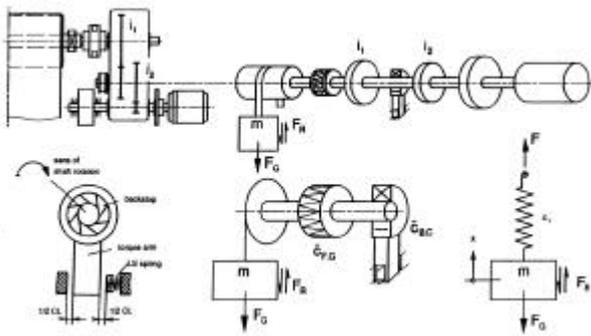


Fig 15. Mechanical model of the drive arrangement

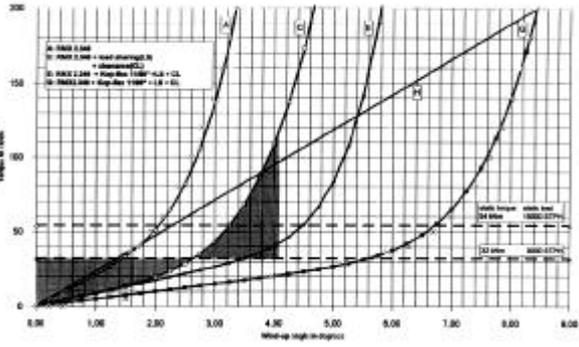


Fig. 16 Wind- up characteristics

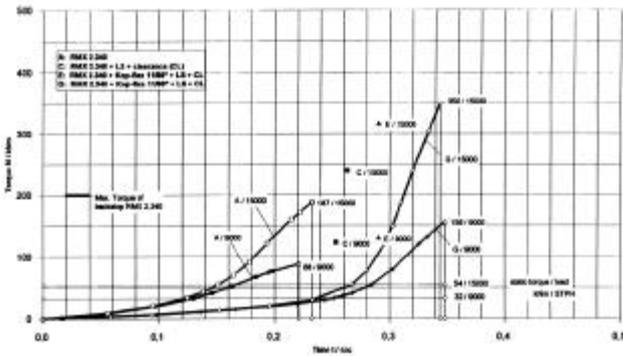


Fig. 17 Dynamic Torques

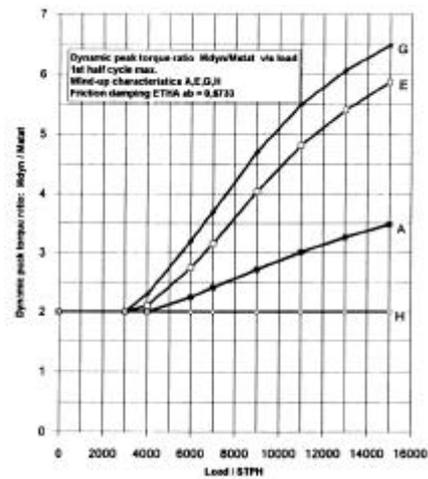


Fig. 18  $M_{dyn}/M_{stat}$

Due to the non linearities the dynamic peak torque is not a constant with regard to the static torque of the load, as would be the case with linear torsion spring characteristics (factor 2), but the dynamic factor is load-related. This applies to the torsion spring characteristics A, E, H and G in Fig. 18, with H describing the linear case.

5.4 Problems of Load Distribution with Multiple Drives (Multiple-Backstops)

When conveyor plant are fitted with several drives it is normal to mount one backstop to each of the individual drives, the general assumption being that different backstops would take an equal share of the load. Theoretic tests and measurements have proved, however, that this is not correct. Equal distribution of load is prevented by clearances, uneven elasticities etc., on the other hand large differences may occur when the different drives are stopped at different times, even if only for fraction of a second. The first drive to lose power becomes mechanically slack very quickly and continues to rotate free from load for a certain time until the last of the drives comes to a halt. It is in the final drive that the mounted backstop is subjected to full load while the

backstops of the previously stopped drives are scarcely under load at all because the elastic elements between the drives couldn't tensioned enough.

The result of the calculation for the case with two drives is shown in Fig. 19. This illustration shows that when the first drive is relief to the half the share of the backstops is 90% to 10%. This means that the first load backstop (in the last stopped drive) will be destroyed if the selection is based on the nominal gear torque. More details are in [4].

An effective method to ensure a good distribution of load is the using of backstops with integral torque limiter, according to chapter 3, Fig. 6.

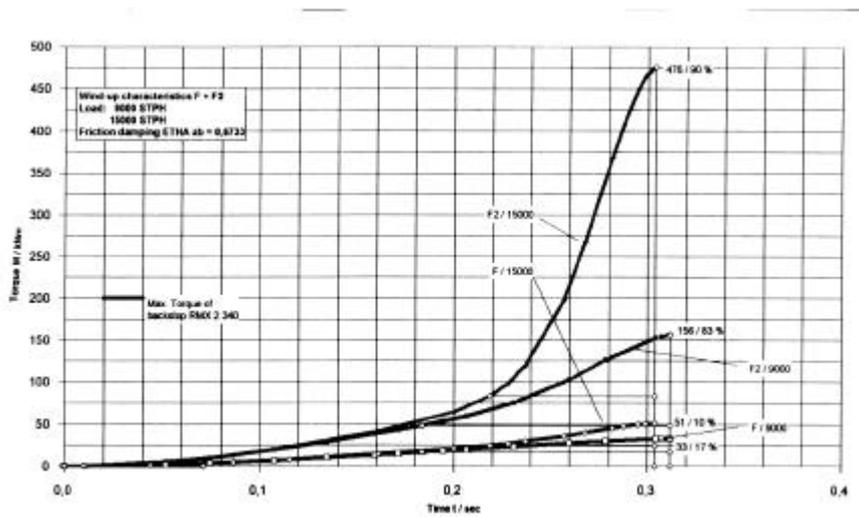


Fig. 19 Dynamic Torque by bad load sharing

## 6. Recommendations for Dimensions - Simplified Version

An exact oscillation calculation is often either too time-expensing or even impossible. A proven method arrived at after many years of experience (in close co-operation with manufacturers of gears and installations) prescribes approximate dimensions for backstop to comply with the following:

$$M = 3.5 * \eta_{ab} * M_L$$

In this formula  $M$  is the transmissible peak torque of the backstop, as explained in 5.2.1 and 5.2.2.  $M_L$  is the static backdriving torque of the load in relation to the backstop shaft. The factor 3.5 is a dynamic factor which takes into consideration all the dynamic characteristics of the installation.  $\eta_{ab}$  represents the efficiency of the conveyor installation (= lifting capacity / (lifting capacity + power loss)). Further details are given in [6 RINGSPANN]. As mentioned, this is an approximate calculation which does not take the correct conditions of individual cases into account.

## 7. Summary

Brief detailed descriptions of applications for the very important application example of backstops in conveyor installations were followed by explanations of the design considerations applicable to the very latest type. The currently best and safest method of calculating the torque transmissibility of backstops including the case of eccentric races were described.

Also explored was the topic of peak torques created by dynamic influences which can occur in machines or installations. As this essay demonstrates, it is perfectly possible nowadays to determine safe dimensions for backstops which will prevent damage in case of load.

During their main function of freewheeling the backstops operate completely without contact through the centrifugal lift-off effect, and therefore maintenance free, so that the operating life of the backstop can exceed the life of the whole system.

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