

Torque Limiting High Speed Backstop Solutions for the West Angelas and P.T. Freeport Indonesia Conveyor Projects

Abstract

This paper will discuss the application of high-speed torque limiting backstops in two large overland conveyor projects. Backstop selection and drive integration will be discussed for the West Angelas high capacity conveyor drives. Special attention will be given to integration of load-sharing backstops in multiple drive conveyors at P.T. Freeport Indonesia.

1 Introduction

The demand for high powered conveyor drive systems continues to increase with improved conveyor belt technology. Power demands of today can be such that a single cost effective drive system is simply not available. To solve this dilemma, manufactures are utilizing high powered dual, triple, or quad drive packages on a single conveyor. The latest conveyor drive projects at West Angelas and P.T. Freeport Indonesia are evidence of this trend. On multiple drive systems the challenge becomes finding the proper load holding device that will be economical, while at the same time providing reliable service and long life. This was the challenge for multiple drive conveyors installed and operating at West Angelas and P.T. Freeport Indonesia.

2 Details of the Two Conveyor Projects

2.1 West Angelas

The West Angelas iron ore project is located in the Eastern Pibara of Western Australia. The West Angelas project consist of an open pit mine; a crushing and screening ore processing plant producing lump and sinter fines iron ore, as well as stockpiling, reclaiming and train-loading facilities. West Angelas contains an estimated 455 million tons of Marra Mamba (a higher grade of ore) [13]. The West Angelas project required numerous overland bulk handling conveyors, with the largest two conveyors requiring two 800 kW drive systems each. The 800 kW drive solution shown in Fig. 1 consist of a bevel helical gear drive with an integral torque limiting backstop.

The torque limiting high speed backstop was selected over standard high speed and low speed backstops for a variety of reasons. The main one being the high speed backstop provided an economical solution over a higher cost low speed backstop. In addition, the torque limiting feature of the FXRV backstop allowed the backstop to be sized for just the backstopping torque required, not peak loads. See Fig. 17 for reference.

The FXRV backstop inner-ring and lift-off sprag assembly is mounted on the intermediate shaft of the gear drive, and the outer torque limiting ring is bolted to the side wall of the drive

Fig. 1: West Angelas gear drive with backstop mounted to intermediate shaft



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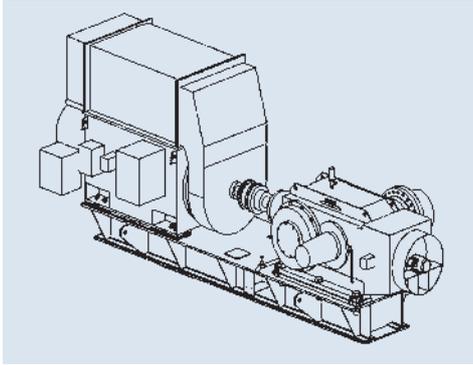


Fig. 2: ▶ West Angelas 800 kW drive system



▲ Fig. 3: P.T. Freeport Indonesia 1 600 kW drive system

housing. The lift-off sprag assembly and friction linings share the same lubrication system as the gear drive.

The West Angelas drive system also included an 800kW main drive motor, rigid low speed coupling, electric fan cooling, swingbase, and torque arm pad as show in Fig. 2.

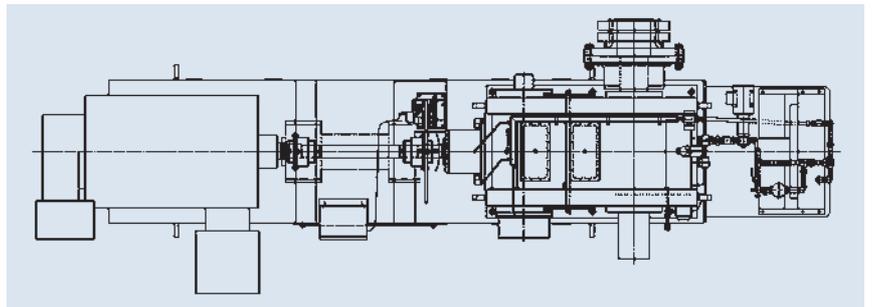
West Angelas Conveyor Drive Specifications

- Two 800 kW (600 hp) drives
- Double reduction bevel helical drive
- 1000 rpm input
- 12.37:1 ratio
- Output torque rating 146 000 Nm (107 700 lbs.ft.)
- Internal torque limiting backstop
- Slip torque 42 500 Nm (31 500 lbs.ft.)
- High speed disc brake system
- Rigid flange output coupling
- Electric fan cooling
- Swingbase
- Torque arm pad
- Total drive system wt. 17,600 kg (38,800 lb)

2.2 Freeport, Indonesia

P.T. Freeport Indonesia’s copper, gold, and silver mining operations are located in Papua, Indonesia. P.T. Freeport Indonesia’s operations consist of mineral exploration and development, mining and milling of ore containing copper, gold and silver, and the worldwide marketing of concentrates containing those metals [12]. P.T. Freeport Indonesia installed an expansive overburden handling system that consisted of several high powered overland conveyors, with the largest requiring four 1 600 kW (2 100 hp) drive packages, see Figs. 3 and 4.

The original 1 600 kW drive included a variable fill fluid drive for soft starting and a disc brake system for holding. Proper synchronization of the fluid drive and disc brake system was found difficult to achieve and maintain. To alleviate this problem, conventional high speed and low speed mechanical backstopping options were proposed as solutions. Torque limiting high speed backstops were selected because it was an economical solution over low speed backstops. Four of the largest known cataloged low speed backstops would have been required, in addition to major drive or pulley rework to allow the backstops to be used. The overburden handling sys-



▲ Fig. 4: 1 600 kW drive system with intermediate backstop shown

tem is remotely located at 4 000 m elevation, which limited the ability to perform in-place drive modifications.

Torque limiting style backstops were selected over standard high speed backstop due to their ability to load share between the four drives. The torque limiting feature also ensured the vital components of the drive would not be subjected to unexpected, potentially damaging, shock loads produced by the conveyor.

P.T. Freeport Conveyor Drive Specifications

- Four 1 600 kW (2 100 hp) drives
- Two primary and two secondary drives
- Triple reduction bevel helical design
- 1145 rpm
- 24.19:1 ratio
- Output torque rating 720 000 Nm (5 311 000 lbs.ft.)
- Internal, torque limiting backstop
- Slip torque 33 900 Nm (25 000 lbs.ft.)
- Variable fill fluid drive
- High speed disc brake system
- Rigid flange output coupling
- Cooling system
- Swingbase
- Torque arm
- Total drive system wt. 48 500 kg (107 000 lb)

2.3 Important Features Of The Drive Trains

- 1 Rigid low speed connection which reduces elasticity in the drive system.
- 2 Gear drive with proper service factor for the application.
- 3 Internal backstops sealed to the drive which eliminates the risk of contamination due to seal failure.

- 4 Torque limiting feature to facilitate load sharing between drives and eliminate drive exposure to peak loads.
- 5 Proper backstop rating versus slip torque rating.

3 Design of High Speed Backstops

3.1 The Most Advanced and Reliable Design: Centrifugal Lift-Off With Rotating Inner Ring

With the invention [1] in 1970 of the centrifugal lift-off with rotating inner race, it became possible for the first time to arrange small dimensioned backstops with low lift-off speeds directly onto fast running shafts and integrated in the gearbox. Since then much 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 backstop, see [4] and [8]. The new generation provides for high torque capacity at very low lift-off speeds. The clever geometrical design of the sprags can even compensate for a large tolerance between shaft and outer race.

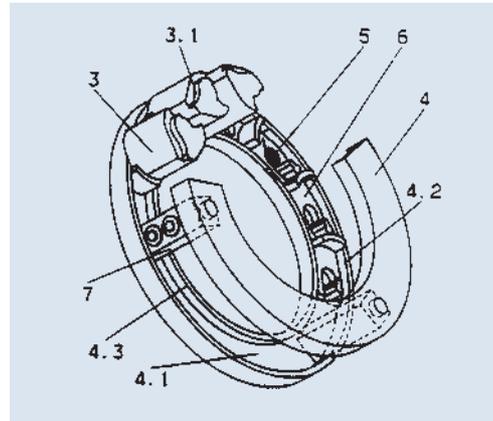
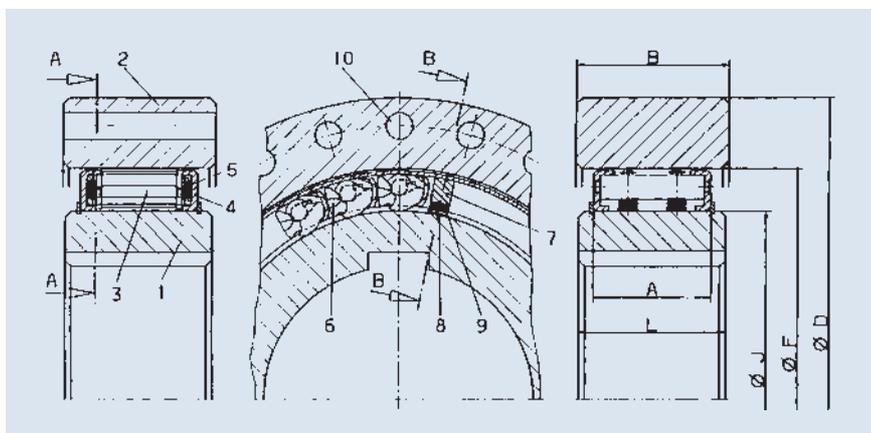
3.1.1 Specifications of Backstops with Centrifugal Lift-off X

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

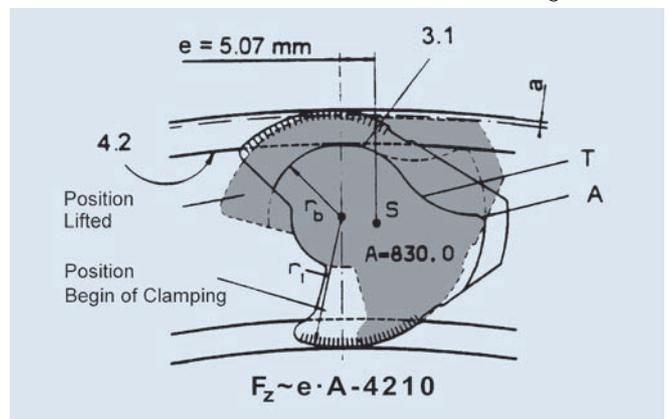
- simple design with as few components as possible
- minimum dimensions with maximum torque capacity
- no wear and no maintenance
- sturdy construction
- high temperature stability
- large misalignment capability of the shaft in the bearings
- large tolerances of adjacent parts

In order to meet all of these specifications the most common solution today is provided by the freewheel with centrifugal lift-off "X" with rotating inner race. This necessitates the use of cage-supported sprags. The requirement for sturdiness and temperature stability can only be met by metallic materials. The power transmitting parts must be made of hardened steel to satisfy the requirement for small dimensions.

▼ Fig. 5: Built-in backstop FXM with lift-off type cage



◀ Fig. 6: 3D drawing of the cage "X"



▼ Fig. 7: Sprag with centrifugal lift-off

Centrifugal lift-off "X" with rotating inner race is the most elegant method for building an extremely sturdy, reliable and maintenance free backstop with minimum costs. An inner race together with the sprag cage is fitted to a gear shaft end. The effective outer race is bolted to the gear housing, either directly or via an intermediate flange – see Fig. 5. During normal operation, i. e. rotation of the gear shaft, the centrifugal effect causes the sprags to swing inward so that above the lift-off speed they rotate within the outer race without any contact – see Fig. 7. This means a virtually unlimited life for such backstops.

3.1.2 Design of the Cage-Freewheel with Lift-off X

Figs. 5, 6 and 7 show the design of the built-in backstops using the patented cage with centrifugal lift-off "X" at a rotating inner race which fulfils all specification in 3.1.1.

Circumference Support of the Sprags

Fig. 6 shows that the freewheel cage consists of two sturdy U-formed races (4) which are riveted together with connecting-pins (7). At the bottom (4.1) of the rings (4) are welded guide pockets. These guide pockets are very close together so that a maximum number of sprags may be placed in the cage. Care must be taken to lock the guide pockets to hold the sprags in axially parallel position. This is necessary to insure maximum torque capacity. At the same time, the

sprags need proper 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, (see Figs. 6 and 7), of both sprag ends is important for the lift-off process and the centrifugal lift. This contour has a radius r_b in the virtual contact area with the inner radius of the cage races (4.2). This is concentric to the inner sprag radius r_i . This contour transmits the centrifugal force which is acting on the sprag to the cage ring and allows 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 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 pocket 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. A minimum of metal is cut from the sprag. 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

Single springs are arranged in guide pocket on both sides of the sprag. The guide pockets (6) are formed in such a way that they have an opening for the sprag spring (5) and a 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 spring causes the spring force to act on the sprag basically in a radial direction, i. e. the spring force is directly opposed 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 toward the lift off position.

Contour of the Sprag. Figs. 6 and 7 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 maximum load is applied to the sprag during a stop.

Synchronisation of the Cage

For a perfect centrifugal lift-off with rotating inner race 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 race. This is achieved by the synchronising pin (8) which is pressed into each of the cage connecting pins (7). (See Fig. 5.) They have mushroom heads and are made of hardened steel pressed against the inner race 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 between the cage and the inner race during acceleration or deceleration. On the other hand this frictional synchronous connection does permit a very slight relative

movement between the cage and the inner race during the roll-in movement of the sprags at torque transmission.

4 Requirements for Backstops in Multiple Drives

4.1 Why Load Sharing in Multiple Drives?

On large conveyors with multiple drives and backstops on each drive a perfect operating load sharing system is essential. For details see [2], [4] and [5] If a sharing system is not used each backstop and each gearbox must be able to handle the full conveyor torque, e. g. this is required in [7]. This would increase the cost of the drive equipment significantly. Unequal load on the gearboxes and backstops are inevitable. This is mainly created by different spring characteristics and different masses of inertia in the many components of the conveyor.

Therefore, many designs have been tried to solve the problem of a reliable load sharing device or system. The following section will show the most common mechanisms and discuss their characteristics and problems. The next section will show a very popular system, the damper spring. This is a simple but dangerous design. Under many conditions this spring damper system will actually amplify the peak torque instead of damping the torque seen by the backstops.

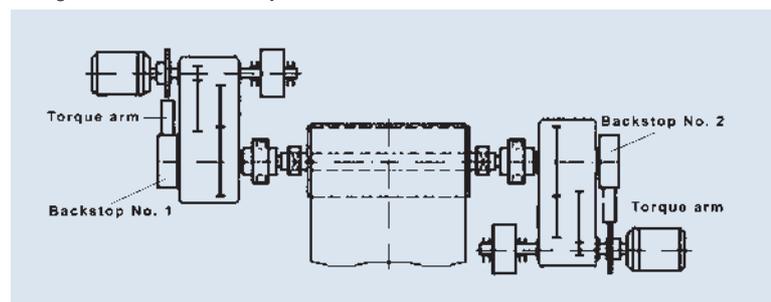
After this discussion a most reliable load sharing system with an integrated torque limiter will be presented and discussed. This design has operated successfully for more than 25 years in hundreds of difficult applications. The British Coal Board mainly required this type of system in the 1980s. There had been a number of serious accidents in the coal mines in Britain. Some people were killed in some of the accidents. Today most of the gearbox manufacturers supply this torque limiter backstop system for multiple drive conveyors.

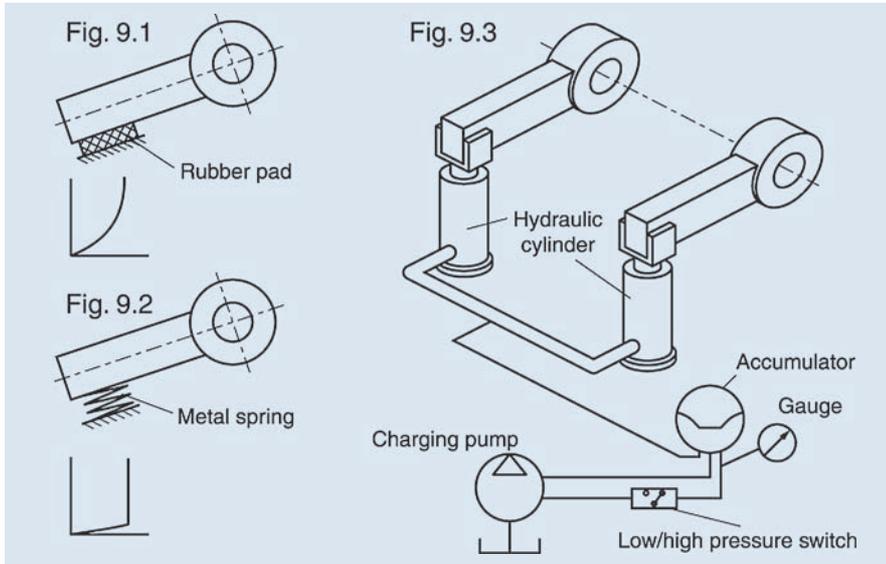
4.2 Load Sharing Devices at Backstops of Plants With Multiple Drives

In the past backstops were nearly always designed as external backstops – see Fig. 8.

A solid support of the torque arm is very important because of the enormous forces occurring on the foundation. It must be ensured that the supporting structure will not be destroyed by the reaction force that builds up when the backstop is

▼ Fig. 8: Dual drive conveyor





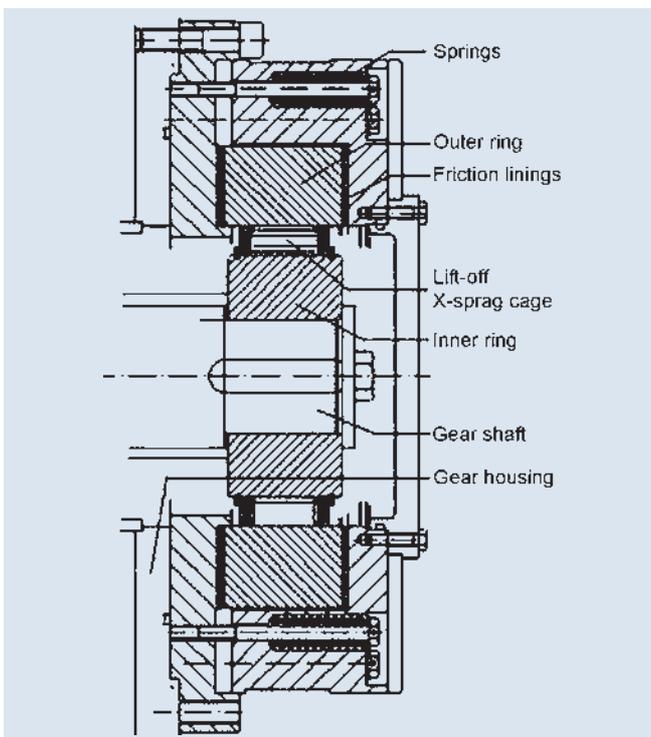
▲ Fig. 9: Traditional load sharing devices

engaged. Manufacturers of backstops recommend supporting structures as stiff as possible. There have been many attempts over the years to force multiple drive backstops to share the load or to limit the peak torques seen by the backstop.

Fig. 9 shows three examples of elastic load sharing. Fig. 9.1 shows a simple rubber bumper. Fig. 9.2 is a schematic drawing of a spring pillar arranged between the foundation and the torque arm. The disadvantage or sometimes the danger of such methods is shown in detail in the next Paragraph 4.4.

Fig. 9.3 is a schematic diagram of a hydrostatic equalizer for a multiple drive of a conveyor belt. Such installations were

▼ Fig. 10: Backstop FXRV with build-in torque limiter for ideal load sharing



produced for tests. They did not gain acceptance, however, for various reasons; the main reason was their unsatisfactory short service life.

Since the load sharing systems described above never operated satisfactory a new system was invented with a build-in torque limiter. Fig. 10 shows the backstop Type FXRV with Integral torque limiter.

Here the same backstopping principle applies as with the high speed inner race overrunning built in backstops of Fig. 5 with the exception that the outer race is not screwed on but is held captive via bilaterally arranged friction linings. These linings are pressed on by a spring pack. This design allows for the reduction of peak torques and by a small amount of slipping in the locking direction facilitates compensation between the individual backstops. This distributes the load

evenly among all of the backstops. This allows smaller backstops with lower torque capacities to be used. The main purpose of the torque limiter installed on multiple drives is, however, to achieve ideal load sharing. Other load share methods cannot insure that two or more backstops will share the torque evenly. When the limiting torque is reached there is a relatively small turning movement in that backstop, and then the next backstop becomes effective and takes up the rest of the required torque.

This backstop with torque limiter shown in Fig. 10 could have a finely controlled release facility so that in case of locking the tension in the conveyor belt can be reduced where necessary.

4.3 Shock Loads in Conveyor System During the Backstopping Mode

As described earlier in detail, the spring characteristics of the total conveyor mechanical system (wind-up) seen at the head pulley is always a nonlinear curve. Therefore the created vibration or the first shock load seen by the backstop is not a sine or cosine curve. It is a typical, nonlinear vibration curve.

▼ Fig. 11: Photo of a backstop with build-in torque limiter, 1 of 9 units at torque setting for the West Angelas project



During the stopping mode a completely different mechanical system has to be used than that used during the normal operating, conveying operation. This is shown in Fig. 12. The dotted line at 50 000 lbs-ft is the static load torque in the conveyor. The solid line at approximately 32 000 lbs-ft is the torque reduced by the friction in the conveyor during the first run back cycle. Curve 1 shows the calculation result as described in [2] with a completely analytic method. Curve 2 is the result of a computer dynamic torque simulation with a multi mass system (same data as for curve 1).

Based on this it is simple to find an approximation (without friction damping) for the first torque peak using the nonlinear wind-up characteristic, (see Fig. 13) of the system reflected to the head pulley.

The static torque M_{STAT} is known. The area (1) has to be calculated. This area represents the potential energy which is stored in the system when it is stationary. At the stopping mode a vibration is created and area (2) has to be equal to area (1). Since the wind-up function is known the maximum torque M_{MAX} can be calculated by integration.

With the same method a system with multiple drives can be analyzed. The individual wind-up characteristics are not identical for various reasons. The worst case is when one motor stops before the other. This drive will completely relax. This will happen in a few milliseconds. When the other motor stops and the conveyor begins to run back, the second drive is still loaded and the torsional characteristic is very stiff. The torsional characteristic of the first drive is very soft, because of clearance in the gearbox and unloaded rubber couplings. These two characteristics are shown in Fig. 14

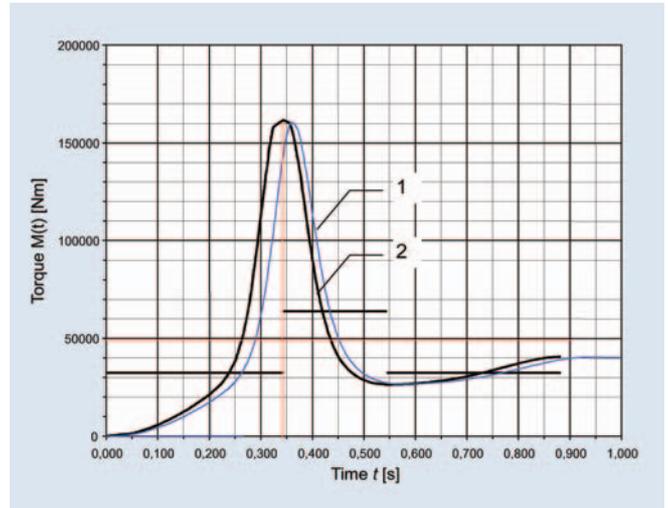
It is obvious that with this soft, nonlinear characteristic for the same static load the peak torque will become much higher than with a stiff characteristic. For this system the peak torque of the soft wind-up was app. 60 % higher than that of the stiff wind-up curve.

4.4 Reason for Torque Amplifying with Rubber or Spring Supports of the Torque Arm

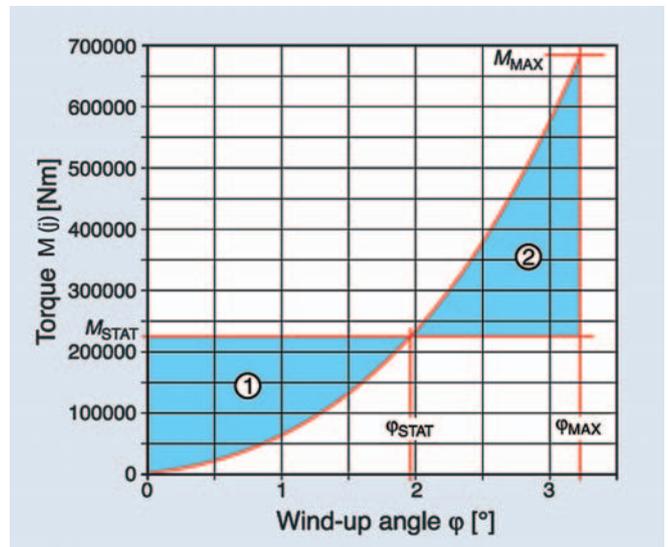
Based on these mechanical facts, it is simple to show why a rubber or damping spring system causes problems. This system does not dampen. The opposite is reality. This type of system amplifies the shock at the very beginning of the stopping mode.

Fig. 15 shows the principal of a common design of a more sophisticated damping device with Belleville springs. The individual torsional characteristics of the backstop (diagram in upper right corner) and of the "damper" are shown. The spring in this damper is relatively soft but at a certain angle it is and has to be very stiff. The curve looks like a wall. See lower diagram in the middle.

These two characteristics will be combined as shown in [2] The result of this combination is shown in the diagram in the lower right corner in Fig. 15.

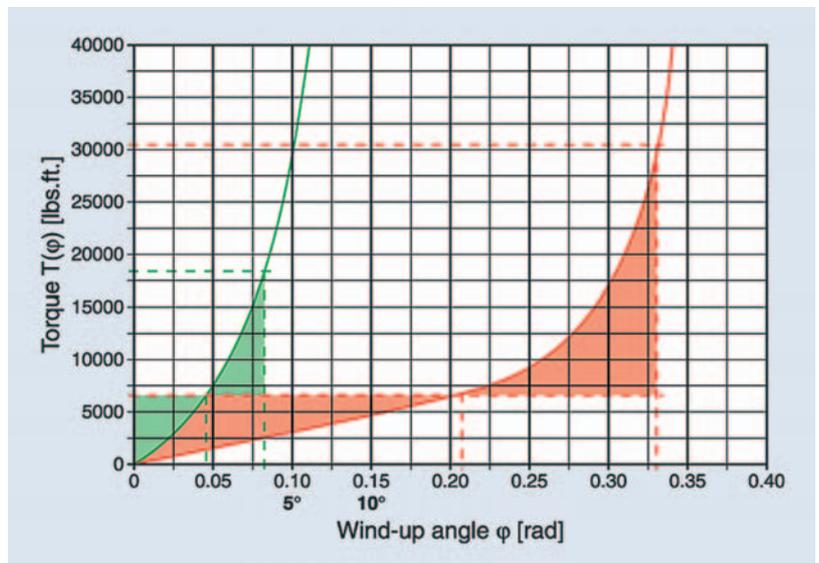


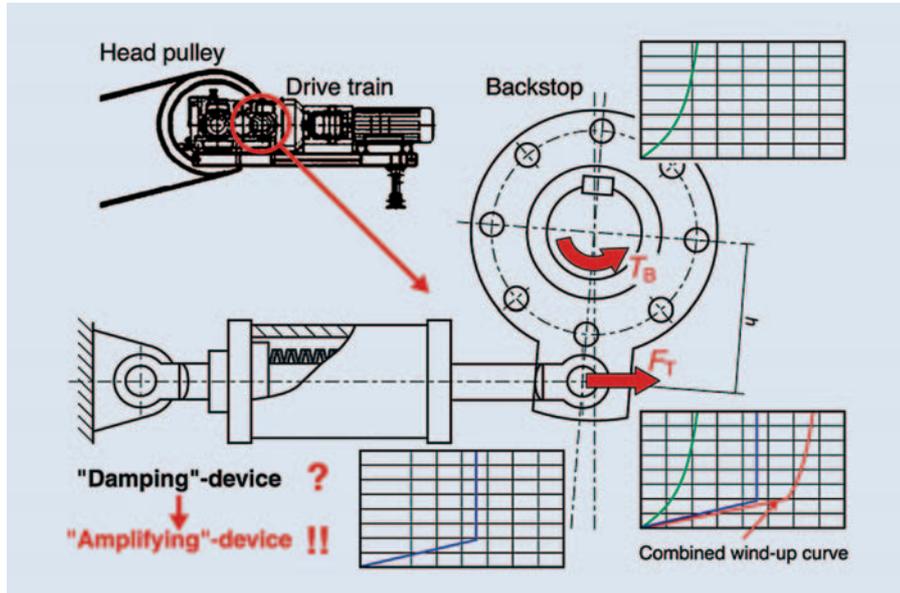
▲ Fig. 12: Dynamic torque versus time in a backstop at the very first actuation with friction damping in the conveyor



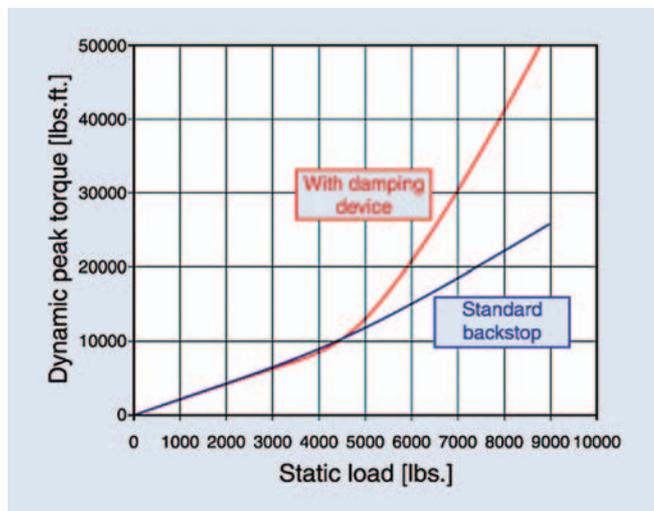
▲ Fig. 13: Nonlinear wind-up characteristic of a conveyor system

▼ Fig. 14: Two real wind-up characteristics and maximum torques at the same static load in a conveyor



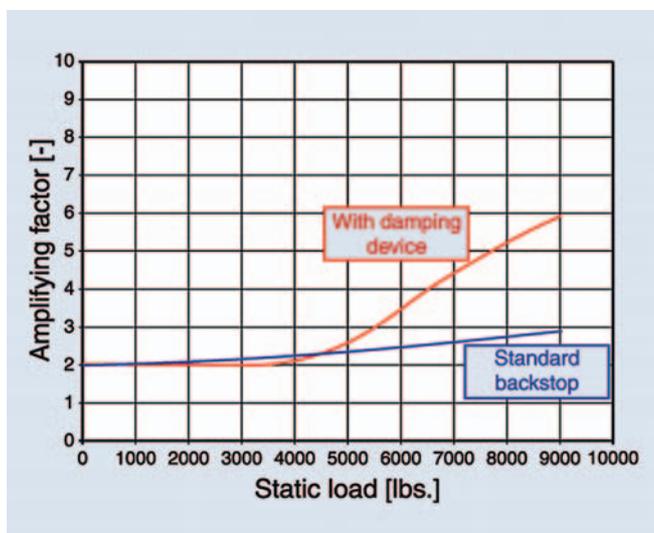


▲ Fig. 15: Backstop "Damping"-system with Belleville springs



▲ Fig. 16: Dynamic peak torques for two backstops with "Damping" device and standard backstop for various static loads

▼ Fig. 17: Dynamic amplifying factor for two backstops with "Damping" device and standard backstop



Figs. 16 and 17 show the dynamic peak torques and the amplifying factor of both systems.

It is self evident that this curve will create a much higher torque than the stiff characteristic of the backstop and a stiff support.

Fig. 17 shows a static load up to approximately 4 000 lbs. The amplifying factor for both backstops is only 2 since the wind-up curve is linear for this load. At the full load of approximately 9 000 lbs. the amplifying factors for a standard backstop is 3 but for a backstop with a "Damping" device 5.8. For this reason such devices are better called "Amplifying" devices!

4.5 Ideal Load Sharing with Integrated Torque Limiter in the Backstop

Fig. 18 shows the torque in two backstops. One with a stiff characteristic. The other with a soft characteristic. Both with build-in torque limiters. Some installations use only one backstop with build-in torque limiter to reduce the peak torques in the system. In both backstops the torque is limited by the same slipping torque set point. Also in this system area (1)=(2) and (3)=(4) For this stiff system the backstop slips thru the torque limiter approximately 5°. The soft system slips about 20°. After the torque limiter slips the torque is dissipated and the conveyor comes to rest. In either case the maximum torque seen by the backstop will not exceed the set point torque of the torque limiter.

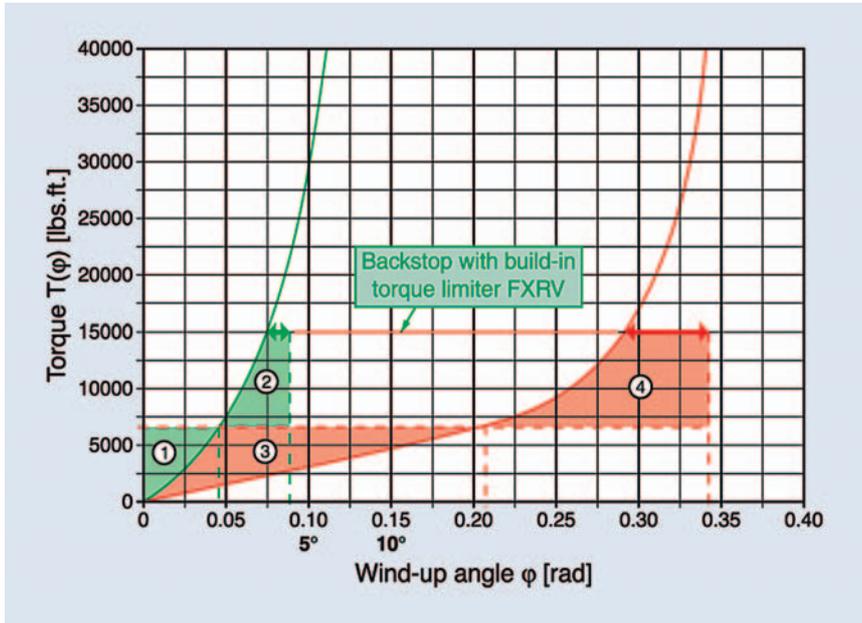
In a multiple drive conveyor a load sharing system utilizing is essential. For such a drive (Fig. 8) we can assume that the head pulley has a very high torsional stiffness. The two stub shafts on the drum always rotate with the same angle. But the torsional wind-up in the driveline elements with the backstops and head pulley is always different as shown in 4.1.

Fig. 19 shows for such a multiple drive the two wind-up curves reduced to the head pulley. Backstop 1 has a stiff characteristic. Backstop 2 has a soft characteristic. The torque limiter in both backstops is set to 16 500 lbs.ft. The head pulley always has the same amount of rotation on both sides. Therefore we must sum up the torques at the same wind-up angle. This is the curve Σ in Fig. 19.

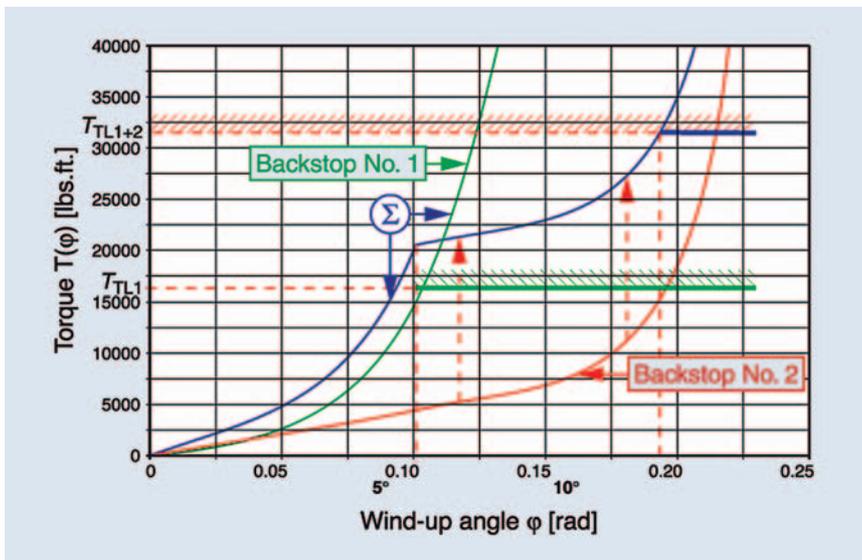
At the angle of approximately 6° in Fig. 19, the torque limiter in backstop 1 starts to slip at a torque T_{TL1} of 16 000 lbs.ft. At 6° backstop 2 only carries 4 500 lbs.ft. If the load torque is higher, backstop 2 will carry the additional torque up to the slipping torque of 16 000 lbs.ft. This is shown in Fig. 19 in the Σ curve for wind-up angles larger than 6°.

The static back driving torque of the installation (even with an overload) must never be allowed to reach the proportional set point slipping torque of the individual backstop.

Fig. 19 further shows that the maximum total backstopping torque is twice the set point torque of one backstop T_{TL1+2}



▲ Fig. 18: Backstop with build-in torque limiter, torque versus wind-up angle for two wind-up characteristics



▲ Fig. 19: Torque versus wind-up angle for two backstops in a multiple drive

(32 000 lbs.ft.). If there is no torque limiting backstop installed the maximum torque is only 20 500 lbs.ft. This is the Σ curve less than 6° . It is assumed that this backstop has the same torque rating as the version with the torque limiter. We also see, that for such a pair of backstops the load sharing ratio is 78 % to 22 %.

5 Conclusion

In a high powered conveyor with multiple drives the utilization of a load sharing system is essential. To achieve ideal load sharing on a multiple drive conveyor a torque limiter must be incorporated with the backstop. Reliable service and long life are requirements at West Angelas and P.T. Freeport Indonesia, and the high speed lift-off backstop has proven to provide just that.

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