



## ***BELTCON 3***

Belt Conveyor Holdbacks

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## BELT CONVEYOR HOLDBACKS

A CONSIDERATION OF DESIGN ASPECTS AND CONCEPTS  
OF SIZE SELECTION FOR VARIOUS CONVEYOR DRIVE  
APPLICATIONS

Clarence A Seaton  
Marland One-Way Clutch

1. SUMMARY

Since the purpose of a backstop is to prevent reverse rotation of an inclined conveyor, this paper will discuss the reasons why reversal should be prevented.

We will briefly discuss various types of backstop designs, spending additional time on the over-running clutch design.

We will present examples of various backstops and how they should be installed and sized. Where multiple drives and/or shafts are shown, we will discuss location of the backstops, the load sharing aspects of the installation and how we size select for these applications.

## 2. INTRODUCTORY REMARKS

The continuing world demand for energy resources and raw materials from the earth creates a requirement for a rapid means of transporting the material from the mine to where it is used. Several methods for transporting these materials are utilized, depending upon the type of material, quantity, distance to be moved, location and surrounding environment to deal with. These methods include pneumatic, pumped slurry, vertical skips, tram-ways, large, heavy duty trucks and belt conveyors. Though each of these has certain advantages for specific applications, heavy duty trucks and belt conveyors are the principal means used to transport the heavier, bulkier ores such as iron, copper, uranium, molybdenum, manganese, etc., as well as coal.

The higher initial cost of the belt conveyor is off-set by the higher, on-going expense of the truck, such as the cost of truck mechanical maintenance and tires, the driver and the distance related cost of the fuel. Both have their use and the purpose of this paper will not be well served by debating the cross-over point of cost or use between trucks and conveyors. Needless to say, belt conveyors serve a vital link in the movement of numerous types of materials. They are used for many and various applications where the drives vary from low, single digit KW to several thousand kilowatts. They run cross-country, up steep inclines from underground or open pit mines, to and from storage areas, to processing operations, to load-out devices or end use points.

## 3. BACKSTOPS

Inclined conveyors require an anti-runback device to prevent reverse movement of the belts. Such a device is referred to as a backstop, or holdback. Though backstops are most likely to be found on inclined conveyors, they are also employed on flat, overland conveyors to avoid the unusually severe shock loading on start-up where the loaded belt sags between idlers. This paper will direct its attention to backstops installed on inclined conveyors.

### 3. BACKSTOPS continued

Without a backstop, a reversing conveyor can rapidly accelerate to a runaway condition, which can kill or injure personnel, damage or destroy drive train components, tear or rip expensive belting, or cause considerable other damage. A backstop is essentially a safety device which acts to prevent reversal thereby protecting against any of the above from occurring, as well as the massive clean up of material spillage than can occur.

Backstops can be classified either for low-speed or high-speed use. In the US, consulting engineering firms generally specify the use of low-speed backstops on all inclined conveyors where the motor KW exceeds 30 to 40. In addition to wanting to avoid the problems noted above, ours is a highly litigious society with large personal injury and product liability settlements. For this reason, the consultants want to minimize the chance for trouble, thus specifying low-speed backstops. It is to these low-speed devices that we will be directing our attention.

### 4. LOW-SPEED BACKSTOP DESIGN TYPES

There are three basic backstop designs that are or have been used to prevent anti-runback throughout the many years of conveying materials;

- 1 ratchet and pawl
- 2 differential bandbrake and
- 3 the overrunning clutch design

The advantages and disadvantages of these units is best shown in the table below:

	<u>Ratchet and Pawl</u>	<u>Differential Bandbrake</u>	<u>Over-running Roller</u>	<u>Sprag</u>
Subject to wear	YES	YES	NO	YES
Affected by dirt	YES	YES on most	NO	NO
Requires adjustment	NO	YES	NO	NO
Backlash	YES	YES	NO	NO
High Stress Concentration	YES	NO	NO	NO
Price	LOWER	LOWER	HIGHER	HIGHER

#### 4. LOW-SPEED BACKSTOP DESIGN TYPES continued

The over-running clutch type backstop is designed for precision operation, automatically engaging to transmit torque when relative motion is in the driving direction and freewheels when relative rotation is in the opposite direction. This design provides a wider operating speed range than other types of backstops and much greater torque ranges - in excess of 700,000 Newton-metres.

There are two basic types of over-running clutch style backstops; roller on inclined planes and sprag clutches.

##### 4.1 Roller-Clutch

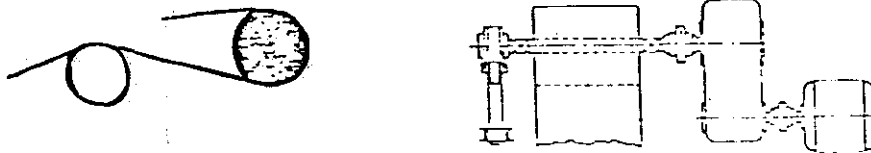
The roller on inclined plane design consists of two concentric races, one cylindrical (the outer race) and the other precision machined with a series of inclined planes or wedge-shaped surfaces equally spaced around the circumference (the cam). Precision ground rollers are installed between the inclined planes and cylindrical race and it is the wedging action of the rollers between the two surfaces that transmits the torque. The rollers are separated from both surfaces by an oil film during freewheeling so no wear occurs in this mode of operation. When the clutch slows down as the pulley shaft decelerates, the spring loaded rollers overcome the viscous shear of the oil bringing the rollers up the inclined plane to insure automatic backlash free engagement when the pulley shaft stops and tries to reverse.

##### 4.2 Sprag Clutch

The sprag clutch design consists of circular inner and outer races and a complement of non-cylindrical, irregularly shaped wedging elements or sprags. The sprags are installed in the annular space between the two cylindrical races. During freewheel, the sprags must be retained in position to engage, so these elements rub on the races. Since a backstop freewheels most of the time, this constant rubbing of the spring loaded sprags will produce wear both on the races and the sprags. When the sprags rotate to wedge between the races to transmit torque, they always engage on the same contact point of the sprag, unlike a roller which has an infinite number of points of contact.

## 5. LOCATION FOR BACKSTOP INSTALLATIONS

A low-speed backstop generally refers to units that are running at conveyor drive pulley speeds. Most frequently, low-speed backstops are mounted directly on the extended head pulley or drive pulley shaft opposite the drive, as shown below.



This provides the most positive means of controlling belt reversal. Further, it also allows necessary service work on the drive components (i.e. reducer, couplings and motor) to be performed with ease as no reverse torque is present.

If space or some other factor is a problem for locating as above, then an alternative location would be to mount the backstop on the double extended low-speed reducer shaft. Mounting the unit in this fashion does subject the backstop to the inherent vibration in the reducer shaft together with higher operating temperatures. Either of these conditions could increase maintenance on the backstop. This does provide a more convenient location for servicing the unit than if it were located between the pulley shaft bearing and the low-speed coupling. With the backstop mounted on the reducer, should a failure occur in the low-speed coupling, the conveyor would run back as the backstop would not then be connected to the pulley shaft.

## 6. BACKSTOP SIZE SELECTION

The vast majority of recognized engineering firms, both in the US and abroad, use the breakdown or stalled torque of the driving motor(s) to size the backstop. We concur completely with this method of sizing. This ensures that the backstop will not be damaged in the event the belt becomes jammed or stops due to an overloaded condition. Since there is no backlash in the backstop, the torque that it must be capable of withstanding is the equivalent breakdown torque of the driving motor(s) at the head shaft which will be present in the system as stored energy or rubber band effect.

Most manufacturers of low-speed backstops have a prescribed method for size selection based on the torque rating of the backstop in conjunction with a recommended service factor to be used based on the maximum torque characteristic of the driving motor(s). The accompanying table shows typical service factors to be applied when size selecting backstops. Torque rating of the unit would include a motor torque characteristic of 175% as built in; i.e. a service factor of 1.0.

TABLE

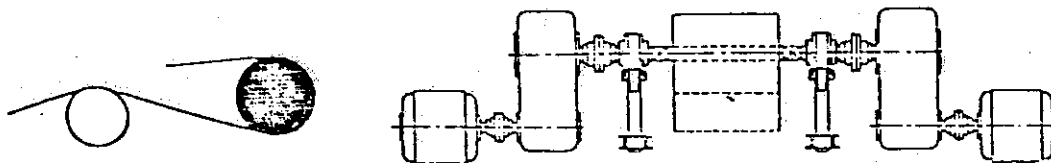
Maximum Breakdown or Stalled Torque	Service Factor
% of Normal Motor Rating	
175%	1.00
200%	1.15
225%	1.30
250%	1.50



## 6.1 Dual Drives

If two backstops are located on the same shaft, they will not initially share the load equally.

### DUAL DRIVE ON HEADSHAFT WITH 2 BACKSTOPS



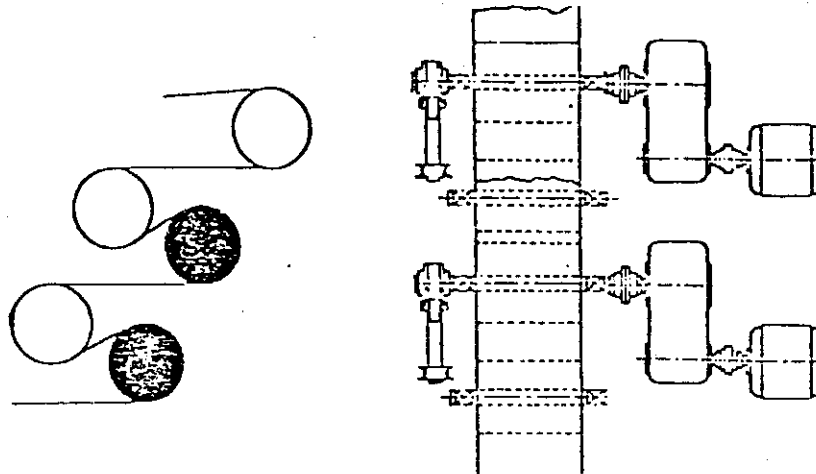
How the backstops are installed, the positioning of the torque arm, bore and key fits, and maintenance of the backstops are among the factors that affect load sharing. Therefore it is recommended that each backstop have a capacity of at least 70% of the total calculated reverse torque of the drive shaft. In addition, this provides some extra factor of safety over the theoretical 50% minimum torque required and should one backstop be required to carry a higher than normal torque load, it is fully capable of doing so without incurring any damage.

## 6.2 Tandem Drives

When the pulley arrangement calls for a primary and secondary drive, (two power driven pulleys), the backstop(s) on the primary pulley shaft must be sized to have a torque capacity equal to the total of the primary and secondary drive motors - the total possible torque the shaft could see.

## 6.2 Tandem Drives continued

### TANDEM DRIVE ARRANGEMENT WITH PRIMARY AND SECONDARY DRIVES AND BACKSTOPS



The backstop on the secondary pulley shaft needs only to be sized for the breakdown or stalled torque of the drive motor(s) on the secondary pulley shaft. If this tandem drive arrangement calls for two backstops on either the primary shaft or both shafts, the backstop size selection for the respective shaft(s) with dual backstops would require the same 70% sharing safety factor of the total calculated torque for the appropriate shaft(s).

Load sharing of backstops on conveyors with multiple drives is an extremely important factor. Although the over-running clutch backstop has no backlash and in theory one would expect it to share the load on a calculated percentage basis, our 30 plus years of experience dealing with these multiple drive installations dictates to the contrary. The 70% factor that Marland recommends has proven to be quite satisfactory over a long period of years and many successful dual driven installations.

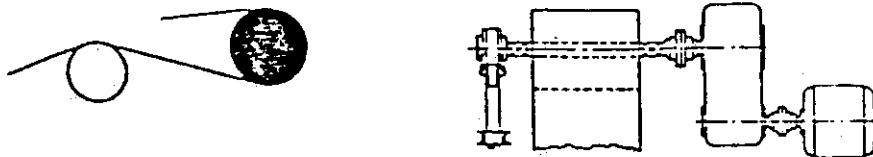
## 7. EXAMPLES OF SIZE SELECTION

Several examples of size selecting the backstops for various applications are included for discussion.

SIZING EXAMPLE NO 1

APPLICATION : Single drive on Head Pulley - Single Backstop  
Union Camp Paper Company, Franklin, VA., USA

ILLUSTRATION :



DATA : Drive Motor = 22.4 KW  
Head Shaft = 18.48 RPM  
225% Electric Motor = 1.30 S.F.

SOLUTION :  $\frac{22.4 \text{ KW}}{18.48} \times 9543 \times 1.30 = 15037 \text{ Newton Metres}$

Backstop Size Selection = BC-12 MA

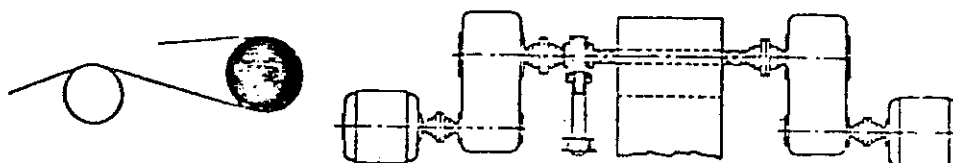
Having a torque capacity of 16260 N-m

with 115 mm max bore

SIZING EXAMPLE NO 2

APPLICATION : Dual drive on Head Pulley - Single Backstop  
National Steel Pellet Company, Keewating, MN., USA

ILLUSTRATION :



DATA : Drive Motors = 2 @ 750 KW

Head Shaft RPM = 30.16

175% Electric Motor = 1.00 S.F.

SOLUTION :  $\frac{1500 \text{ KW}}{30.16} \times 9543 \times 1.0 = 474618 \text{ Newton Metres}$

Backstop Size Selection = BC-375 MA

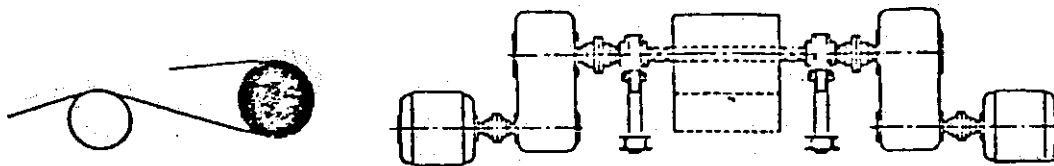
Having a torque capacity of 508125 N-m

with 460 mm max bore

SIZING EXAMPLE NO 3

APPLICATION : Dual drive on Head Pulley - Dual Backstop  
Copper Mine - British Columbia, Canada

ILLUSTRATION :



DATA : Drive Motors = 2 @ 1500 KW

Driveshaft RPM = 47

200% Electric Motor = 1.15 S.F.

Recommended Factor  
for sizing dual drive  
backstops = .7

SOLUTION :  $\frac{3000}{47} \times 9543 \times 1.15 = 700497$  Newton Metres  
Total Torque

$700497 \times .7 = 490348$  Newton Metres  
per backstop

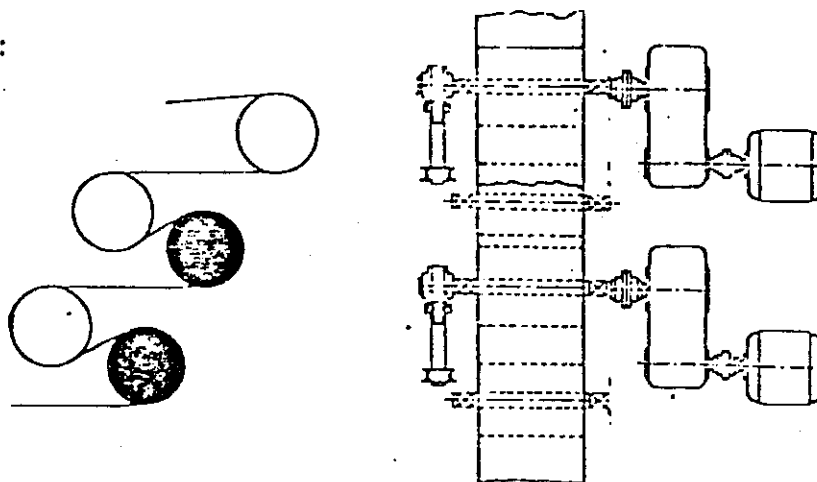
Backstop Size Selection = Two (2) BC-375 MA

Having a torque capacity of 508125 N-m  
with a max bore of 460 mm

SIZING EXAMPLE NO 4

APPLICATION : Tandem Drive - Primary and Secondary Tandem Backstops  
Amax Coal Company, Eagle Butte, WY., USA

ILLUSTRATION :



DATA : Primary Drive Pulley = 597 KW  
Secondary Drive Pulley = 597 KW  
Drive Pulley RPM = 62.57  
200% Electric motors = 1.15 S.F.

PRIMARY BACKSTOP SIZED FOR TOTAL DRIVING KW

SOLUTION :  $\frac{1194}{62.57} \times 9543 \times 1.15 = 209421$  Newton Metres  
Total Torque

Backstop size selection = BC-180MA.  
Having a max. torque capacity of 243900 N-m  
with 300 mm max bore

SIZING EXAMPLE NO 4

SECONDARY BACKSTOP SIZED FOR SECONDARY DRIVING KW

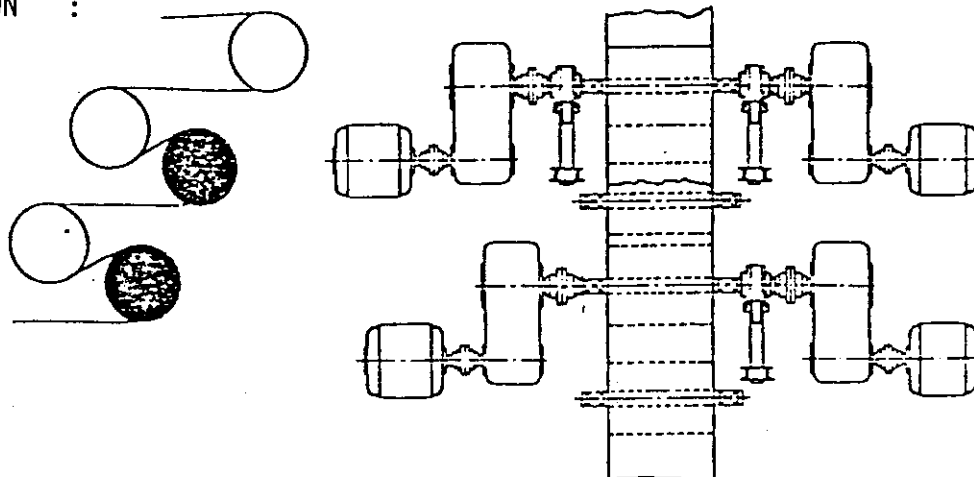
SOLUTION :  $\frac{597}{62.57} \times 9543 \times 1.15 = 104710$  Newton Metres

Backstop size selection = BC-90 MA.  
Having a maximum torque capacity of 121950 N-m  
with 235 mm max bore

SIZING EXAMPLE NO 5

APPLICATION : Dual Tandem Drive  
 Primary and Secondary Tandem Backstops  
 CVRD, Porto Madeira, San Luiz, Brasil

ILLUSTRATION :



DATA : Primary Drive Motors = 2 @ 1120 KW  
 Secondary Drive Motors = 2 @ 1120 KW  
 Drive Shaft RPM = 42  
 175% Electric motor = 1.0 S.F.  
 Recommended Factor  
 for Sizing Dual Backstops= .7



PRIMARY BACKSTOP SIZED FOR TOTAL DRIVING KW

SOLUTION :  $\frac{4480}{42} \times 9543 \times 1.0 = 1017920 \text{ Newton Metres}$   
 Total Torque

$1017920 \times .7 = 712543 \text{ Newton Metres}$   
 per backstop on the Primary Shaft

Backstop size selection = BC-180MA.  
 Having a max. torque capacity of 781700 N-m  
 with a 540 mm max bore

SECONDARY BACKSTOP SIZED FOR SECONDARY DRIVING KW

SOLUTION :  $\frac{2240}{42} \times 9543 \times 100 = 508960 \text{ Newton Metres}$

Backstop size selection = One (1) BC-375 MA.  
 Having a maximum torque capacity of 508125 N-m  
 with 460 mm max bore