LOAD SHARING METHODS OF BACKSTOPS IN CONVEYOR DRIVE SYSTEMS

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ABSTRACT

As shown in former publications there are problems in loadsharing of backstops in multiple drives of conveyors. Therefore, detailed calculation methods and design recommendations are required. In this paper various designs of loadsharing methods are described and a dynamic torque calculation method will be presented. With this method the loadsharing properties for low and high speed backstops are calculated and compared. These properties depend on the various spring characteristic between load and installation position of the backstop. Furthermore, the properties depend on the method of torquearm support and shock absorber configuration. Remarkable differences for external and internal backstops are shown

1. INTRODUCTION

The load sharing of backstops installed in multiple drives on inclined conveyors was described in short in [1] and [2]. In the Short Course of the SME of 1997 in Denver [3], the large possible differences with regard to the peak torque in the backstops were expressly pointed out in connection with the dynamic calculation of some conveyor installations. The present paper describes the calculation methods explained in [1] more comprehensively and in greater detail. Load sharing is of great interest, because of the high cost of large conveyor systems and components. An objective assessment, however, requires a reliable calculation of the entire system. As has been proven, it is decisive for such an assessment to know and correctly combine all elastic characteristic curves of the individual components between the head pulley and the solid foundation.

For the practical application it is most important to provide a simple calculation method that makes it easy to determine the maximum peak torques and the load sharing at the backstops. The method introduced by the present publication meets both requirements. The calculation can be performed by means of a simple PC within a very short time and a minimum of data input required. The method was tested thoroughly by means of a comprehensive computer program [4]. The peak torques and load sharing requirements were surprisingly identical.

The present paper first describes and comments in principle the most important designs of load sharing devices, and then gives a short explanation of the calculation method for load sharing and for the first dynamic peak torque. Examples of low and high speed backstops are also described and the calculation results are discussed. These explanations clearly show which of the load sharing methods are favorable and which are unfavorable, so that the designer has a clear guideline for the design and arrangement of backstops in multiple drive applications.

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2. LOAD SHARING DEVICES AT BACKSTOPS OF PLANTS WITH MULTIPLE DRIVES

2.1. Low Speed Backstops

The conveying velocity of conveyor belts, vary between 1.5 and approx. 6 m/s. Thus, the circumferential speed of the head pulley and of the drive pulley is predetermined. With the help of further plant parameters, in particular the stream of material to be conveyed, the rotational speed of the drive pulley can be determined. Normally, this speed amounts to 20 up to 100 rpm. The electric motors, that are mainly used, as well as internal combustion engines have drive speeds of far more than 1,000 rpm, so there must be a reduction gear between the driving motor and the drive pulley.

As a matter of principle, the backstop can be mounted on any shaft of the drive train, i.e. directly on the slow running drive pulley shaft as well as on any other intermediate gear shaft or even on the motor shaft itself. If the backstop is located on the shaft of the drive pulley or on the first successive gear shaft it is called a „low speed" backstop. In contrast to that, one speaks of „high speed" backstops if they are situated on the intermediate gear shafts that operate at higher speeds or directly on the motor shaft.

The advantages and disadvantages of these two arrangements are generally known. The main disadvantage of low speed backstops is their size and the costs because of the high torque requirements. Furthermore, the reaction forces require supporting arms of large dimensions, which are very expensive. In contrast to that, high speed backstops are rather small and as a consequence the price is very low; and it is easy to attach them or to provide appropriate torque support. The subject is explained in greater detail in [7].

Low speed backstops are nearly always designed as slip fix backstops, refer to [6], fig. 1. 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 engaged. Manufacturers of backstops recommend supporting structures of the largest stiffness 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.

The torque arm must be mounted in such a way that no force is applied on the bearings, otherwise, the bearings in the backstop would have to receive incalculable constrained forces. This could soon result in the damage of the bearings.

Fig. 1 shows three examples of elastic load sharing. Fig. 1a shows a simple rubber buffer whereas fig. 1b is a schematic drawing of a spring pillar arranged between the foundation and the torque arm.

Fig. 1c is a schematic diagram of a hydrostatic equalizer for a multiple drive of a conveyor belt. Such installations were produced for tests. They did not gain acceptance, however, for various reasons; the main reason was their unsatisfactory short service life. In addition, the operating security was impaired by the influences of temperature and by leakages.

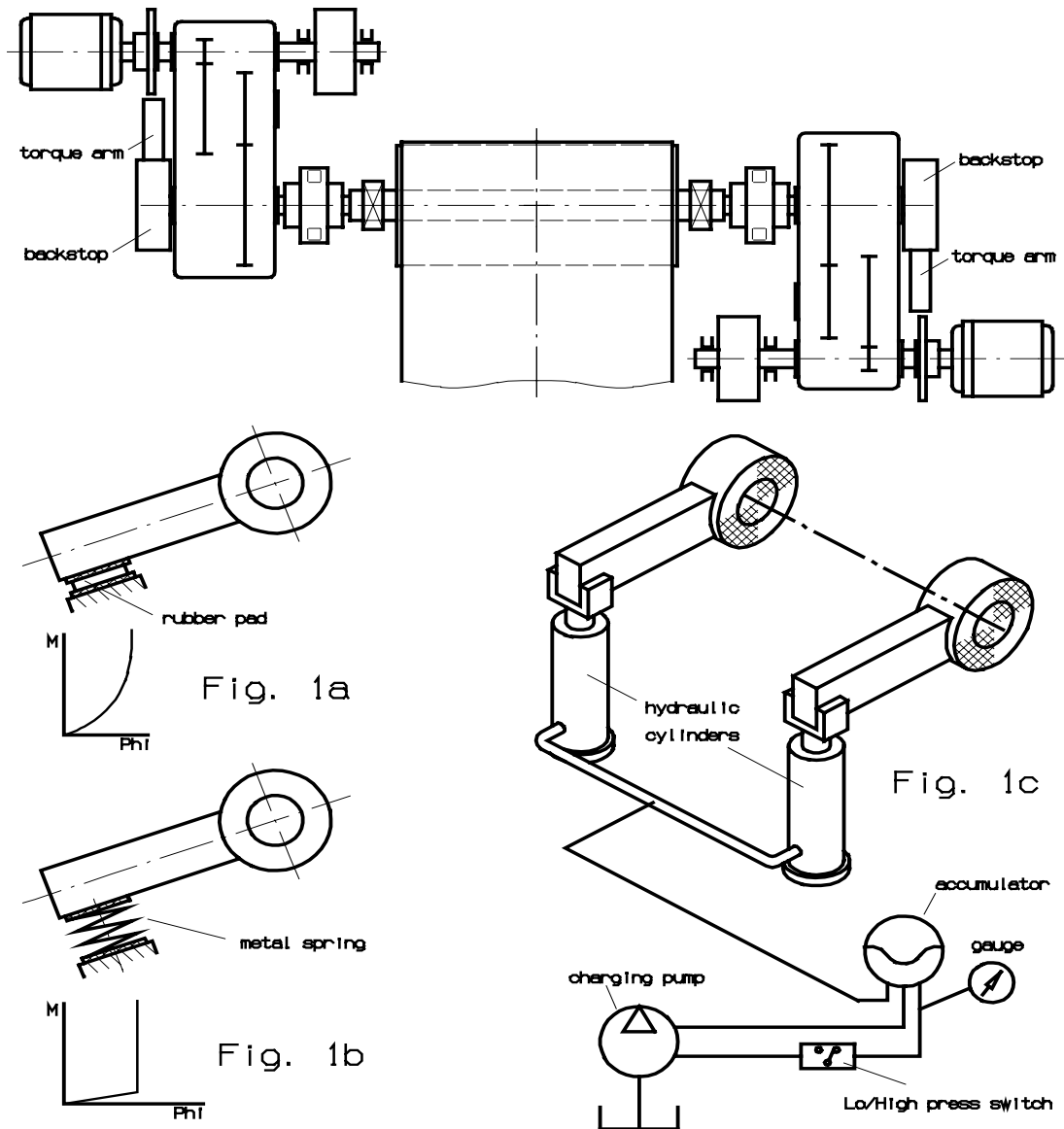Chapter 4 explains the influence of elastic elements on the load sharing and the peak torques.

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Fig.1 Conveyor double drive and load sharing devices

2.2. High Speed Backstops

As mentioned in chapter 2.1 high speed backstops are located either on high speed gear shafts or directly on the motor shaft. This design has been accepted - primarily in Europe - for more than 20 years. These backstops are small in size and thus have a lower cost. Most of these backstops are attached to the gear housing, refer to fig. 2. There are no torque arms or load sharing mechanisms. There are some high speed backstops installed on the free shaft end,

equipped with a torque arm. It was tried with this design to arrange elastic elements between the torque arm and the foundation in order to reduce the peak torque. In addition, the application of shock absorbers was tested. Non of these methods gained acceptance, however.
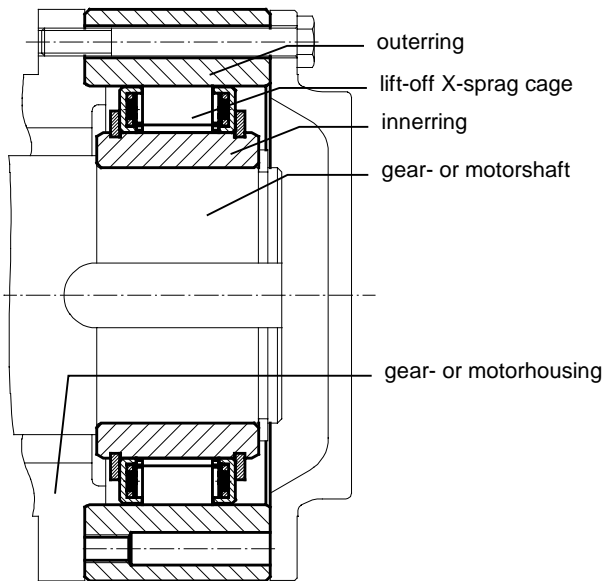
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Fig. 2 High speed backstop, type FXM

In the case of high speed backstops the best solution for safe load sharing in multiple drives is to use a backstop with an integrated torque limiter, see fig. 3.
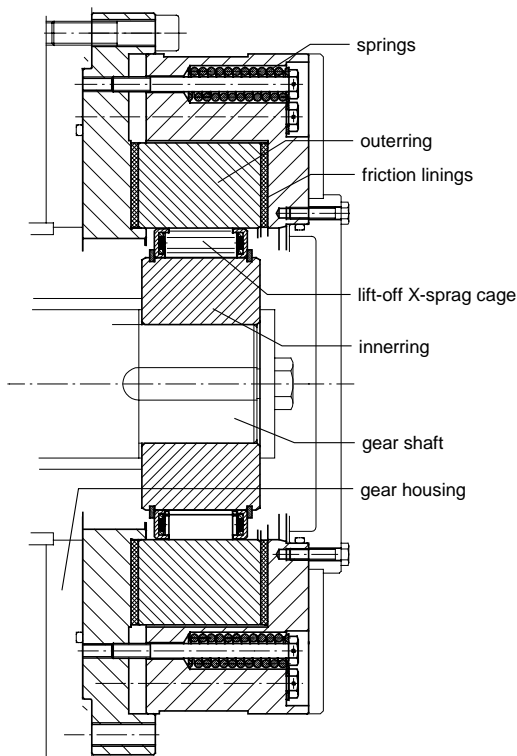

Fig. 3 Backstop with integrated torque limiter, type FXRV

This design has been accepted by the British coal mining industry for years and is gaining acceptence by gear manufacturers on the European Continent. As fig. 3 shows the backstop has the outer ring situated between friction linings that are engaged on by means of helical springs. As a consequence, this type of backstop can only transfer the sliding torque that is determined by the initial force of the spring and the friction linings. As a matter of principle, this torque is always smaller than the maximum torque capacity of the backstop. The backstop as well as the gear unit are protected against overload. 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 one backstop will secure full peak torque first. When the sliding torque has been 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.

2.3. A Comparison of Low Speed and High Speed Backstops

As explained above the high speed backstops are characterized by considerably smaller construction sizes and are thus less expensive. In addition, they are easier to assemble. Usually, they are attached directly onto the gear unit or the motor housing.

When considering the dynamic behavior of the entire system it has to be taken into account that there are a number of elastic elements between the load and the high speed backstop. Particularly with non-linear elements this can lead to higher peak torques. A reliable assessment of the situation, however, requires the exact calculation of the oscillations. By simplifying the entire system and by combining

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the various characteristic curves of the wind-up of the individual elements the author of the present paper managed to find a simple method, that can be carried out with the help of a PC, to calculate the dynamic peak torques of each individual backstop and, to calculate the amount of load sharing for all possible arrangements of the various elements of any system with multiple backstops.

3. DESCRIPTION OF THE CALCULATION METHOD

The basic principle of the calculation method has already been described in [1] and was introduced to the public at the SME-meeting in Denver in 1996.

The equation for the spring stiffness has been improved slightly and now reads as follows:

$$M(\varphi) = A \cdot \varphi + B \cdot \varphi^{n_1} + C \cdot \varphi^{n_2} \qquad (1)$$

As can be seen in this equation an additional exponential term was used in order to have better possibilities of approximating strongly progressive characteristic curves. The exponents n_1 and n_2 can be changed in the input data of the computer program to get an optimized approximation of the calculated wind-up curve related to the original wind-up curve.

In addition, the function of torque vs time is no longer calculated when determining the peak torques - as was shown in [1] - but the peak value is calculated by integration on the basis of the characteristic curves for the torsion springs above and below the static idle position.
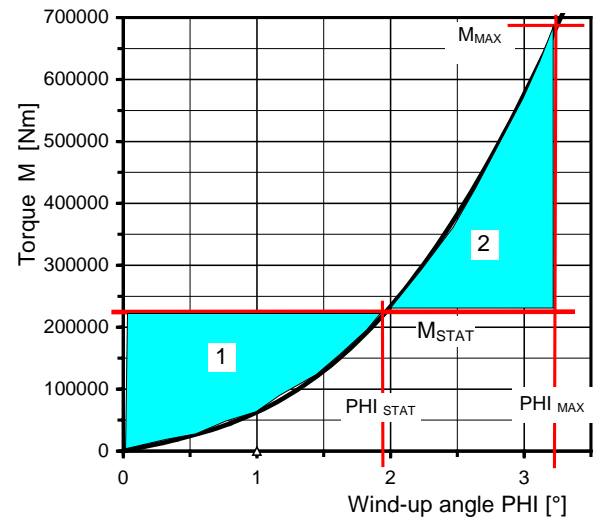

Fig. 4 Wind-up characteristic of a backstop

As it is clearly shown in fig. 4 the areas 1 and 2 are exactly identical by the law of energy. The horizontal line represents the static torque M_{STAT} in the backstop with the conveyor at rest.

Mathematically this is described by the following formula:

$$(M_{STAT} \cdot \varphi_{STAT}) - \int_{o}^{\varphi_{STAT}} M(\varphi) \cdot d\varphi =$$

$$\int_{\varphi_{STAT}}^{\varphi_{MAX}} M(\varphi) \cdot d\varphi - \left[M_{STAT} \cdot (\varphi_{MAX} - \varphi_{STAT}) \right] \qquad (2)$$

Transferring formula (1) in (2) and simplifying the new formula we get:

$$0 = \frac{A}{2} \cdot \varphi_{MAX} + \frac{B}{n_1 + 1} \cdot \varphi_{MAX}^{n_1} + \frac{C}{n_2 + 1} \cdot \varphi_{MAX}^{n_2}$$

$$- M_{STAT} \qquad (3)$$

With common approximation methods, i. g. Newton's method, with formula (3) φ_{MAX} is easy to calculate. Using φ_{MAX} and formula

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(1) we get the maximum dynamic torque M_{MAX} for this case.

The friction attenuation of the inclined conveyors must be taken into consideration for the static idle position. Additional investigations were carried out: With the help of the common calculation method for conveyor plants [5] and [6] the efficiency rates η of small, medium-sized and large conveyor belts were determined in dependence of the inclination angle of the belt. In summary the formula reads as follows:

$$\eta = 1 - f \cdot \left[\frac{m'_R}{m'_L \cdot \sin\delta} + \frac{\left(2 \cdot \frac{m'_G}{m'_L} + 1\right)}{\tan\delta} \right] \qquad (4)$$

The symbols stand for:

f   [1]   :   friction value of upper and lower belt

m'_R [kg/m] : load per meter due to the rotating idler roller on the upper and lower belt

m'_L [kg/m] : load per meter due to the load to be con-veyed

m'_G [kg/m] : load per meter of the belt weight

δ [°]   :   inclination angle of the conveyor

Fig. 5 shows the formula (4) as applied to small, medium-sized and large conveyor plants in the form of a graph.
As is shown in a simple way the efficiency rate is lower for conveyor belts with a small

inclination angle than for belts with a large inclination angle. The friction resistance
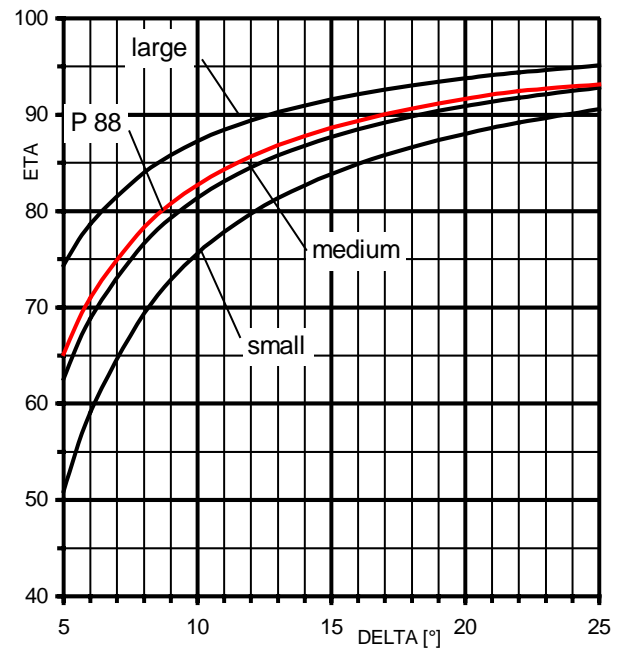

Fig. 5 Conveyor efficiency vs inclination angle

diminishes proportionally with the angle. By using the efficiency rate η the torque of the static idle position M_{STAT} can be calculated for the first reverse movement as follows:

$$M_{STAT} = (1 - \eta) \cdot M_L \qquad (5)$$

M_L in [Nm] is the torque generated by the load to be conveyed. With the help of this torque and the general wind-up characteristic of the system the peak torque can be calculated and the load sharing in systems with multiple backstops be determined by appropriate combination.

The comparative calculations described in Chapter 4 are carried out in such a way that the vibrational system is always reduced to the low speed pulley shaft. This method is permissible because the calculations will show that in the case of multiple drives only the backstops situated

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on the head pulley actually become effective. Backstops arranged downstream of the head pulley nearly no torque sec because the belt between the pulleys is far too flexible to allow any torque to build up on these other backstops. The backstops arranged downstream of the head pulley may only become effective in the very improbable case of the belt slipping at the head pulley as a result of an insufficient friction coefficient. If, however, the belt begins to slip at the head pulley it will soon slip at the subsequent pulleys, also. For calculations with regard to backstops on the head pulley, shaft mass reduction becomes superfluous. Only the spring stiffnesses on the pulley shaft must be adjusted - with the help of the transmission ratio i as described in [1]. For that purpose the wind-up characteristic must be arranged correctly in parallel or in series. If there are backstops on both sides of the head pulley it has to be taken into consideration that due to the large stiffness of the pulley and the large diameter shaft both backstops will see identical wind-up angles.

As the above explanations show the calculation method applied is very simple. Nevertheless, it has proven to be very reliable in test calculations performed with the help of the comprehensive computer program DRESP of the "Forschungs-vereinigung Antriebstechnik - FVA" (Research Association of the German Power Transmission Equipment Manu-facturers) [4]. DRESP differentiates to a much greater extent the various elements of the system of wind-up characteristics, damping devices and rotating inertias. As a consequence, the variations in time of the oscillations were calculated much more exactly, but the first peak torque that occurs at the backstop was still nearly identical with the result of the simplified calculation shown in this paper.

The applied calculation method deliberately leaves unobserved the influence of hydraulic couplings or fly wheels. Those installations are useful for plant operation: the hydraulic coupling particularly for the starting process, the fly wheels for the continuous operation in case of load variations. These elements may have a damping effect on the backstops, i.e. they may reduce the peak torques. For some specific situations, however, this is not true so that the applied calculation method is always safe.

4. COMPARISON CALCULATIONS

According to the calculation methods described in chapter 3 a number of low- and high speed backstop calculations where carried out. Therefore, in fig. 6 some explanations regarding the various wind-up characteristics are shown. There are different types of wind-up characteristics demonstrated in the graphs. As one can see, the stiffness of the torque arm or the foundation is relatively high; e.g. the curve of the torque vs the wind-up angle is steep. Compared with this the elasticity of the belt between the head pulley and the second pulley is very high, e.g. the curve of the torque vs the wind-up angle is flat. Backstops have a more (high speed) or less (low speed) progressive wind-up characteristic. Rubber type couplings or rubber pads always have a very progressive characteristic. At a certain wind-up angle both elements have an infinite torque respectively force.

For the vertical torque arm in fig. 6, the clearance CL is important. In this case the wind-up curve remains horizontal up to a certain angle, e.g. no torque built-up.
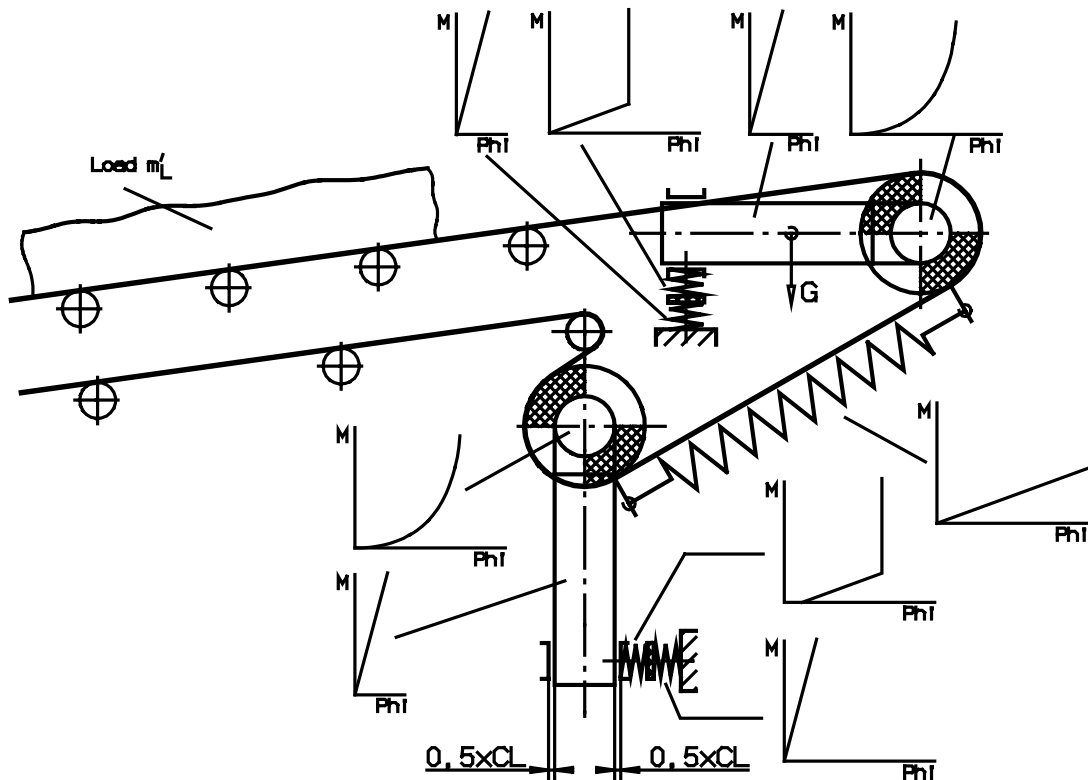
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Fig. 6 Conveyor and backstop elements with there related wind-up characteristics
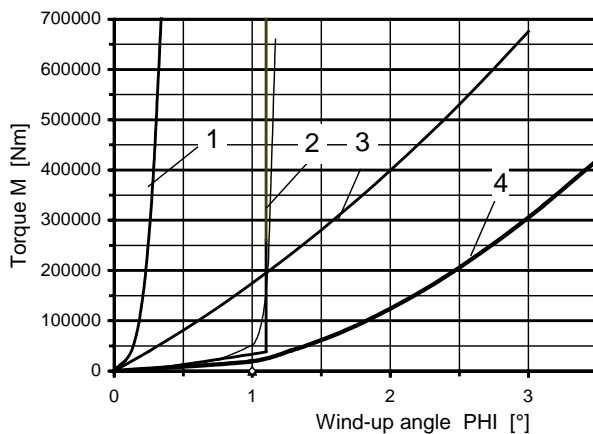

Fig. 7 Wind-up characteristics,
        calculation L14 in table 1

        1 Foundation
        2 Metal spring and clearance
        3 Low speed backstop
        4 Combination of 1 to 3

All wind-up curves in fig. 6 can be calculated with formulars (1) and (3). An example is shown in fig. 7. As one can see the superposition wind-up curve 4 is a series combination of the curves 1, 2 and 3 (angles are added up at the same torque). This curve 4 has to be combined with the superposition curve of the second backstop, in this case L14, curve 5 in fig. 8. Now we have to combine both curves in the parallel mode, e.g. add up torques at the same angle so we get the superposition curve 6. With this curve and curves 4 and 5 the load distribiution in both backstops can easily be calculated. As explained before all types of usual wind-up characteristics can be combined, e.g. superimposed for low and high speed backstops.
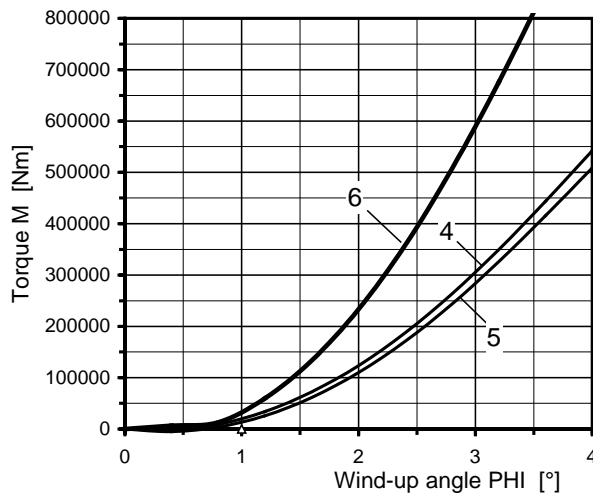
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Fig. 8 Combination of wind-up characteristics
   calculation L14 in table 1
   4 see Fig. 7
   5 Backstop No. 2
   6 Combination of 4 and 5

Table 1 give a short overview of various calculations with the same belt conveyor as shown in [1]. Typical wind-up characteristics where used:
- for low speed backstops Nos. L10 to L14
- for high speed backstops Nos. H10 to H13.

Table 1   Comparisons between dynamic peak torques and load sharing on low- and high speed backstops

Calcu-lation No.	Type of backstop	Type and combination of Wind-up characteristics								Torque ratio $\frac{M_{MAX}}{M_{STAT}}$	Load sharing % BST1/ BST2
		Belt elasti-city Fig. 1	Rubber coupling Fig. 1a	Rubber pad Fig. 1b	Metal spring Fig. 6	Clear-ance Fig. 6	Founda-tion Fig. 6	Back-stop 1 -5% Fig. 1	Back-stop 1 +5% Fig. 1		
L 10	Low speed	-	-	-	-	-	-	+	+	2,22	52,6/47,4
L 11	Low speed	+	-	-	-	-	-	+	+	2,21	81,0/19,0
L 12	Low speed	-	+	-	-	-	-	+	+	2,88	51,6/48,4
L 13	Low speed	-	-	+	-	-	+	+	+	2,58	52,5/47,5
L 14	Low speed	-	-	-	+	+	+	+	+	3,26	52,4/47,6
H 10	High speed	-	-	-	-	-	-	+	+	3,92	52,6/47,4
H 11	High speed	-	+	-	-	-	-	+	+	4,05	53,1/46,9
H12	High speed	-	+	-	+	+	+	+	+	4,50	52,8/47,2

The stationary load torque on the head pulley minus friction torque in conveyor system is always M_{STAT} = 221653 Nm. It was further considered that with manufacturing tolerances the wind-up characteristiv of two backstops of the same size have 10 % variation in the torque at the same wind-up angle.

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The two right columns indicate the dynamic torque ratio M_{Max}/M_{STAT} and the load sharing percentage BST1 / BST2. As one can obviously see at No. L 11 the belt elasticity causes very bad load sharing between the two backstops on the head pulley and the backstop on the second pulley: 91 % / 9 %. This is the same with high speed backstops. Therefore all other calculations where made only with two backstops on the head pulley.

At No. L 13 one can see that the rubber pad between torque arm and foundation is not necessary. The wind-up angle that this element allows is only 0.7° compared with the backstop maximum wind-up angle of 3.4°.

The dynamic ratio M_{Max}/M_{STAT} for low speed backstops is not as high as high speed backstops. The wind-up characteristic of low speed backstops is inmany cases not as progressive as the wind-up characteristic of high speed backstops. But the load sharing ratio is approximately the same. Nevertheless, the size of high speed backstops is still considerably smaller than the size for acompareable low speed backstop. Further the high speed backstops with centrifugal sprag lift-off are maintenance-free with unlimited service life because of their contact free overrunning operation.

5. CONCLUSION

Backstops installed in multiple drive of inclined conveyors require a precise calculation of the dynamic peak torques and load distribution. This paper describes various designs of load sharing methods. The calculation of Peak torque and load sharing for these designs is also detailed and explained. Some installations of low and high speed backstops are calucated and compared. The calculation shows that

the support of the output part of the backstop, e.g. the torque arm, has to be supported on the foundation as stiff as possible. For high speed backstops the attachment without any spacing devices for the gear housing is the best solution and couplings in the drive line must also be as stiff as possible in their wind-up characteristic. These results are completely in line with experiences and recommenadations of backstop manu-facturers.

6. REFERENCES

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