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Steel wire ropes with variable lay lengths for mining applications

If a steel wire rope is installed horizontally in a tensile test machine with both rope ends prevented from twisting and then subjected to a tensile force, every rope element will be stressed by forces corresponding to the line pull and by moments proportional to the external load and the nominal rope diameter (Figure 1).

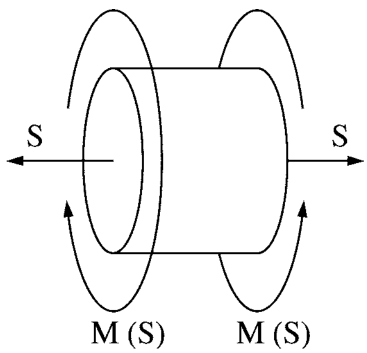

Figure 1: Horizontal rope element

We have equilibrium of forces,

$$S - S = 0$$

and we have equilibrium of moments:

$$M\,(S) - M\,(S) = 0$$

The moment of the rope can be written as:

$$M = c_1 \cdot d \cdot S \qquad (1)$$

where:

M	Rope moment
c_1	Constant
d	Nominal rope diameter
S	Rope line pull

If the rope is being twisted before loading, its moment changes to:

$$M = c_1 \cdot d \cdot S + c_2 \cdot d^2 \cdot S \cdot \omega + c_3 \cdot G \cdot d^4 \cdot \omega \qquad (2)$$

where:

c_2, c_3 Constants

G Shear modulus

$\phi = \omega \cdot l_R$ Angle of rope twist

ω Angle of rope twist per unit length

l_R Rope length

The moment now consists of three components, two of which depend on the angle of rope twist per length unit ω, and two of which depend on the rope line pull *S*. For the rope user this has the following consequences:

If the rope will not be twisted during rope installation or during rope service, the angle of rope twist per unit length ω will be zero, and Equation (2) can be reduced to Equation (1). Equation (1), however, only depends on the rope line pull *S*. In other words: whenever the line pull is zero, the moment of the rope will also be zero.

If, however, the rope will be twisted during rope installation or during rope service, the angle of rope twist per unit length ω will be not be zero, and Equation (2) will apply. In order to determine the moment in a twisted rope in an unloaded condition, we must set the line pull in this equation to zero, and we obtain the following equation:

$$M\,(S{=}0) = c_3 \cdot G \cdot d^4 \cdot \omega \qquad (3)$$

Obviously, a twisted rope can still display a considerable moment even in an unloaded condition. Therefore, whenever twisted steel wire ropes becomes slack, it will show a tendency to form a loop, which at subsequent loading might be pulled tight and form a kink or in the worst case cause the rope to break.

It is therefore recommended in every wire rope users manual to avoid any procedure during wire rope handling and installation which might twist the rope. In addition, the permissible fleet angles for wire ropes travelling over sheaves and drums are limited by the Standards in order to minimise rope twist during service.

It is a common understanding that wire rope twist cannot be avoided if a wire rope is hanging down a vertical shaft, even if both ends are secured against twisting. If such a wire rope is subjected to an outer force (e.g. the conveyance weight), every rope element will on the lower side be subjected to the line pull *S* created by the external force plus the weight of the rope hanging underneath it, on the upper side by the same force plus the weight δS of the rope element itself (Figure 2).

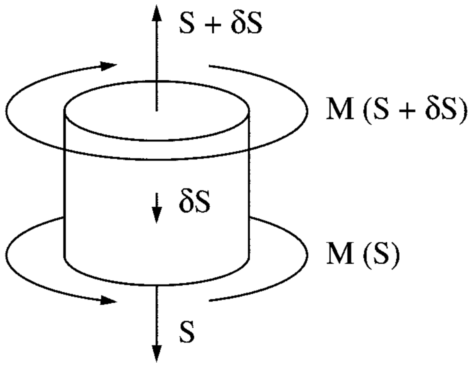

Figure 2: Vertical rope element

We still have equilibrium of forces,

$$S + \delta S - (S + \delta S) = 0$$

but with the exception of a totally rotation resistant rope, where the moment is zero regardless the line pull, we no longer have equilibrium of moments:

$$M\,(S) - M\,(S + \delta S) \neq 0$$

The wire rope will unlay in the direction of the greater (upper) moment. Because both rope ends are fixed, however, the number of strand helixes in the rope cannot change. Therefore the rope lay length will increase in the upper parts of the rope and decrease in the lower part, the total number of rope lays remaining the same.

There will be three sections with the original rope lay length: two at the rope ends and one close to (but normally not exactly in) the middle of the rope. Every section of the rope will deform in such a manner that ω, its angle of rope twist per unit length, will change corresponding to the changing line pull S so that Equation (2) is satisfied. Figure 3 shows a typical distribution of the lay length along the rope length.

In the upper part of the shaft, the wire rope will be twisted considerably in the opening sense, in the lower part it will be twisted severely in the closing sense. This has a number of consequences:

1. Whenever slack rope occurs, such a rope is in danger of forming loops and kinks. Several rope breaks have been reported as a consequence of this mechanism.
2. The wires forming the strands of the rope have different lengths, depending on their position in the strand (the innermost being the shortest) and depending on the lay lengths (the shorter the lay lengths, the longer the wires). An undisturbed lay length will guarantee relatively even load sharing between the wires. Any change in lay length will create length differences between the wires, leading to uneven load sharing and to premature failures of the over-proportionally loaded wires. This is why deep shaft mining ropes never achieve fatigue lives as high as what could be expected from their load history.
3. Parallel lay strands (such as Seale, Filler, Warrington or Warrington-Seale strands) have shown far superior fatigue performance compared to single layer or

two operation strands in crane applications, because in these designs wire crossovers can be avoided. In deep shaft mining applications, however, parallel lay strands exhibit greater differences in wire lengths than single layer or two operation strands when subjected to changes in rope lay length and therefore fail prematurely. As a result, the most fatigue resistant strand designs cannot be used in deep shaft mining operations.

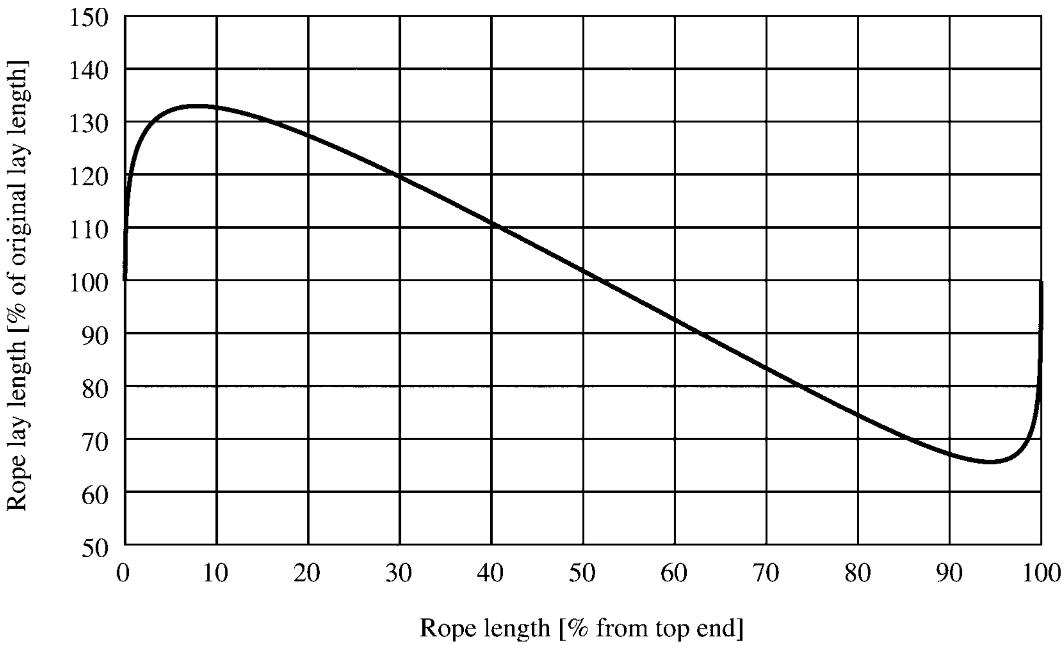

Figure 3: Rope lay length along the rope length (schematic)

All these problems could be avoided if the rope did not have to change its lay length in order to achieve a state of equilibrium. How could that be achieved?

Obviously, the moment of the rope would have to be constant along the rope length from the beginning.

$$M = \text{const.} \tag{4}$$

If that was to be achieved without twisting the rope ($\omega = 0$), Equation (2) for the moment could be reduced to:

$$M = c_1 \cdot d \cdot S = \text{const.} \tag{5}$$

The first solution to this equation would be a totally rotation resistant rope ($c_1 = 0$). In such a rope, the moment would be constantly zero, no matter how big the rope diameter d and no matter by how much the line pull S would increase along the rope length. Rotation resistant ropes, however, may exhibit other disadvantages in mine hoist applications, and therefore non rotation resistant ropes are normally preferred.

With the line pull S increasing towards the top of the shaft, a constant moment could be obtained by decreasing the rope diameter correspondingly. That would leave us with the strange solution that we would have a tapered rope with a large diameter at the bottom of the shaft (where the forces are low) and a small diameter at the top of the shaft (where the forces are the highest) (Figure 4).

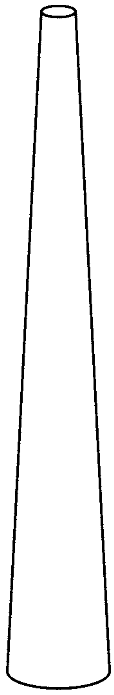

Figure 4: Tapered rope with constant moment along the rope length

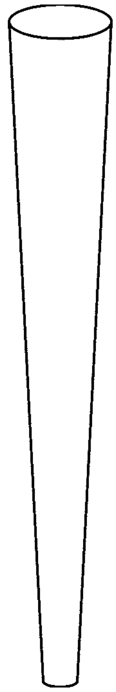

Figure 5: Tapered rope with constant safety factor along the rope length

If we wanted the same specific stresses in every cross-section (constant safety factor along the rope length), we would end up with exactly the opposite solution: a tapered rope which would have a small diameter at the bottom of the shaft (where the forces are low) and a large diameter at the top (where the forces are the highest) (Figure 5).

Obviously, the two solutions exclude each other.

With the additional restriction to keep the rope diameter d constant, the only other solution to Equation (5) is to reduce the factor c_1 along the rope length proportionally to the increasing line pull S. The factor c_1, however, depends on the rope geometry and varies mainly with the lay length of the wire rope. The most convenient solution therefore is to change the wire rope lay length continuously (or in steps) during rope production in such a way that for the average loading conditions of the rope factor $c_1(x)$, multiplied with the line pull $S(x)$ is constant for every position x along the rope length.

If e.g. the average rope forces vary between 80% of a medium figure (at the bottom end of the rope) and 120% (at the top end), the torque factor c_1 of the rope should vary between 120% (at the bottom) and 80% (at the top) of the "normal" torque factor. At first sight, manufacturing such a rope seems to be difficult. In the past, the rope lay length has been kept constant by coupling the cage of the rope closer with its pullout capstan by means of a gearbox. Changing the cogwheels of the gearbox would allow the manufacture of ropes with different, but always constant lay lengths.

In a modern wire rope machine, however, the closing cage and the pullout capstan are no longer mechanically connected, but synchronized electronically. Here, the rope production can e.g. be started with a lay length much shorter than normal (bottom end of the rope), which can then continuously be increased until at the end of the rope (top end of rope) the longest required lay length is achieved.

If the closing machine only allows for a given number of steps in between the minimum and the maximum required lay length, the rope can be manufactured in sections of increasing or decreasing lay length. The rope twist in service will then be reduced approximately by a factor corresponding to the number of steps, but it will not be entirely eliminated.

Optimum strand and rope core diameters will vary with the rope lay length. Figure 6 shows how the clearance between the outer strands of an 8 strand rope will vary if the diameters of the rope core and of the outer strands are kept constant, the rope being closed with varying lay lengths. In this example, the outer strands will touch each other at lay length factors below 5.5, and the clearance will increase to about 0.045 rad. with a lay length factor of 12.

Figure 7 shows the cross section of such a rope with the cross sections of the outer strands shown for lay length factors 5.5 (contact) and 12 (clearance 0.045 rad).

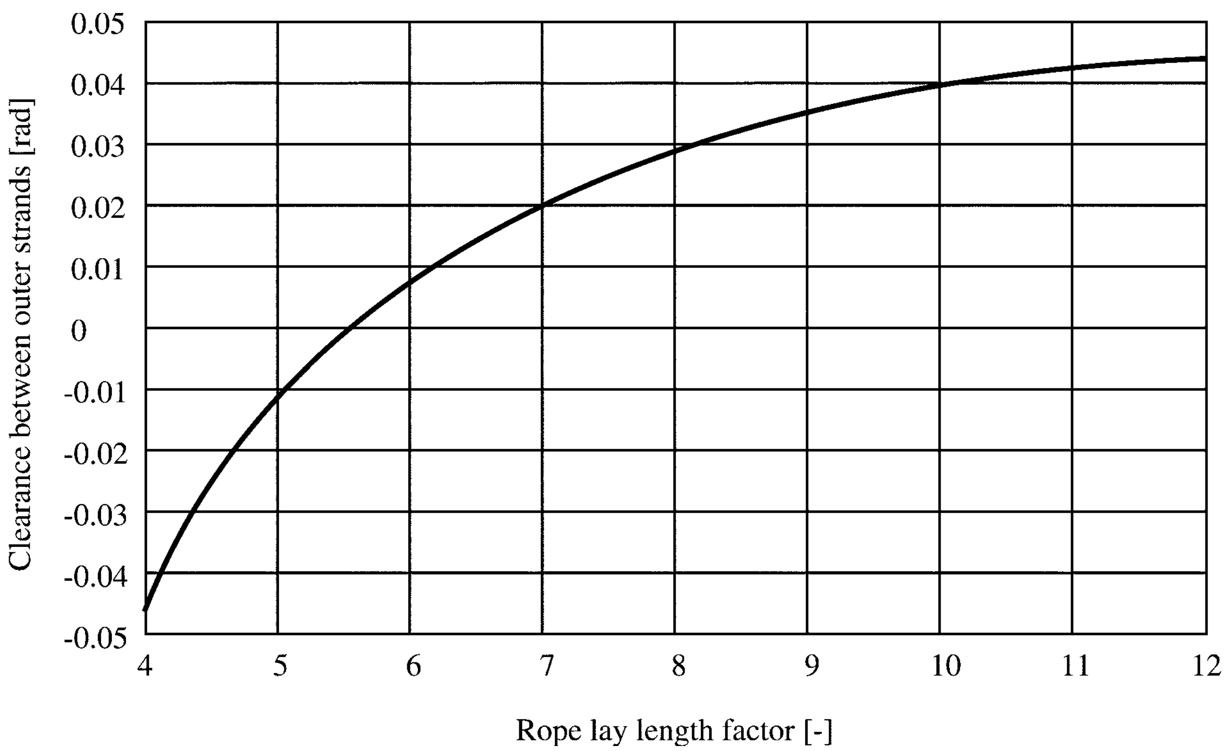

Figure 6: Clearance between outer strands depending on the rope lay length

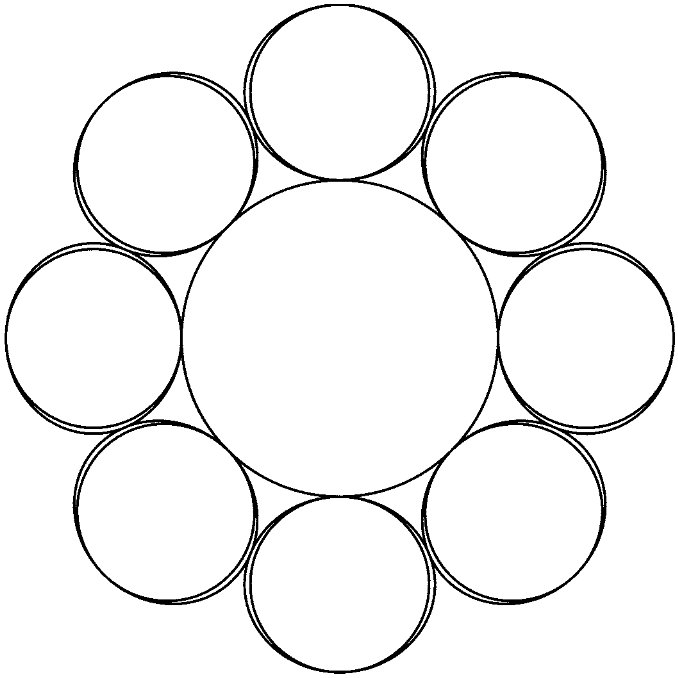

Figure 7: Rope with 8 outer strands closed with lay length factors 5.5 (contact) and 12 (clearance)

In a very favourable execution a wire rope with variable lay length can have a plastic layer around the steel core, separating the outer strands from each other no matter how large the clearance gets with increasing lay length. This execution will guarantee stability even for great clearances between the outer strands.

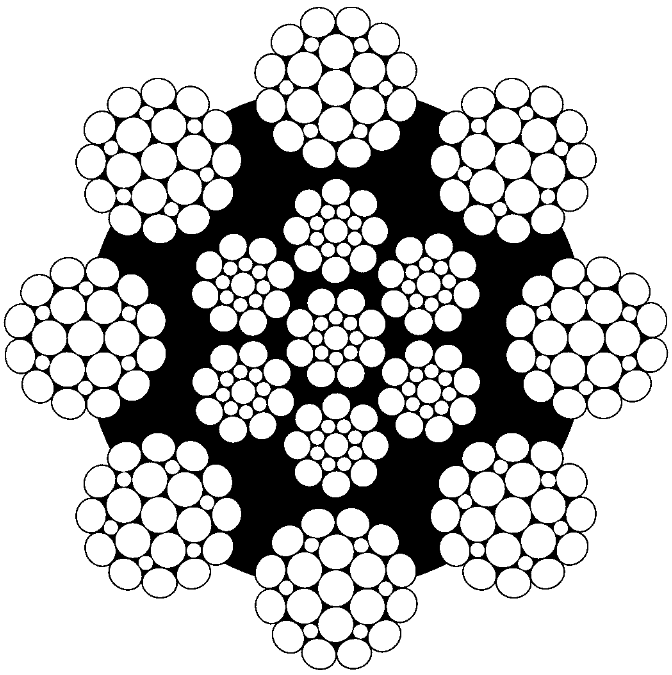

Figure 8: Rope with 8 outer strands and a plastic layer separating the outer strands from each other regardless how great the clearance (Casar Stratoplast)

The concept of steel wire ropes with variable lay length looks very promising. It will considerably increase the fatigue life of deep shaft mining ropes by allowing the use of fatigue resistant parallel lay strands. In addition, it will eliminate the danger of rope breaks under slack rope condition. The first ropes have already been produced and are currently being tested.

There is one important drawback for ropes with variable lay length, however: The concept will only work on drum winders. On friction winders, the rope would also have to work upside down half of the time. And then all its advantages would turn into disadvantages.

References

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