DEPTH LIMITATIONS IN THE USE OF TRIANGULAR STRAND ROPES FOR MINE HOISTING

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SYNOPSIS

Mine hoisting plant in South Africa are predominantly clutched, double drum winders having one or two Lang's lay triangular strand ropes per drum for hoisting in vertical shafts. Over the last hundred years the trend has been for ever increasing shaft depths with single lift shafts currently being sunk to 3200 m. The preferred construction for hoisting ropes has been the single layer triangular strand design as its properties include very good wear and abrasion resistance, ability to tolerate high crushing forces arising from multi-layer coiling on the drums, robustness and relatively low costs when compared to more sophisticated constructions. The rope design also allows for very efficient conversion of wire strength to rope strength. It is generally agreed that the performance versus cost ratio of these ropes is acceptable and a large amount of experience in the their manufacture, operation and inspection exists in the South African industry. The down side of the single layer Lang's lay construction is that it generates high torque under tensile load which results in considerable changes to the as-manufactured lay length of the rope. The extent of shortening of lay length at the splice end and lengthening at the drum end increases with depth and is also compounded by maintenance practices. It is accepted that at a certain depth these changes in lay length will eventually be so great as to prejudice the performance and safety of such ropes in service. Using previously developed and verified modelling procedures this paper investigates the depth limits of triangular strand ropes as a function of both manufacturing and operational parameters. Analysis supported by site observations has shown that seemingly small changes in rope design and the manner in which ropes are maintained or handled in service can have significant advantages in terms of the maximum depth of operation. The analysis indicates that, using current available technology, torsional deformations in triangular strand hoisting ropes can be kept within acceptable limits for winding depths of at least 3200 m.

Keywords :
deep shaft mine hoisting, wire ropes, torque, twist, lay length, Lang's lay.

1 INTRODUCTION

For at least the last ten years the torsional behaviour of Lang's lay triangular strand ropes has been of particular interest to engineers in the context of very deep shaft mine hoisting (drum winders). Figure 1 shows the cross-section of a typical triangular strand rope (TSR) as used for drum winding in South Africa. In a 1996 survey of 653 South African drum winders Rebel[1] found that 86 % of all ropes were of the compound triangular strand construction, all in the range of 6x26 – 6x34. The most common wire tensile grades were 1800 and 1900 MPa (together 70% of all ropes). Diameters in the range of 35 to 55 mm were the most widely used.

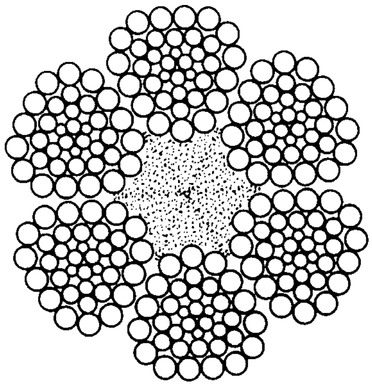


Figure 1 - Cross-section of a 6x33 Lang's lay compound triangular strand rope commonly used for drum winding applications in South Africa.

Operational experience has shown that these ropes have a shortened lay length at the conveyance end and an increased lay length at the sheave end when suspended vertically in a mine shaft. It is also accepted that the extent of the lay length changes increase with an increase in the suspended length. This means that as shafts become deeper, rope torsional stability issues require more careful consideration. Currently the deepest single lift shaft is some 2500 m and TSRs perform acceptably at this depth. The next generation of deep shafts will have suspended rope lengths of approximately 3200 m with permanent hoisting set to commence in the second half of 2001. Based on torsional behaviour modelling results, it is likely that triangular strand ropes will be used for these shafts. This paper gives a review of analysis which supports this decision and also addresses areas where improvements can be made to manage the changes in lay length more effectively and reduce risk.

1.1 Review of previous publications

The analysis of Hermes and Bruens[2] seems to be the earliest reference to methods of calculating rope rotation and lay length changes on drum winders (with the lower rope end fixed against rotation) and it formed the basis for further work. It is evident that refinements to their basic model were primarily in the form of more accurate descriptions of the rope torsional properties with less attention being given to detailed understanding of the in-service boundary conditions for rotation. At that time, Hermes and Bruens recognised that :

> *Spin in a Lang's lay wire rope causes the angle of lay of the strands to adopt various values along the rope. Starting from a linear theory for the spin phenomenon the variations from the mean angle of lay can be calculated. In the case of a drum winder it has been found that the angle of lay in the various rope parts decreases linearly with the distance from the cages but does not vary during the wind. The variations increase with the depth of the winding installation.*

Another significant example of early rope tension-torsion research is the investigation of Layland et al.[3]. The response of Lang's lay ropes to loading and twisting was examined experimentally. Both the free suspension method (torsion pendulum) and a torsion machine (modified lathe) were used to determine the torsional stiffness properties and relationships between load and torque.

Layland et al. stated that it was well known that a rope with one fixed end and the other free to rotate will do so in the unlaying direction until a condition of torsional equilibrium is established. In the case where the lower end is not able to rotate, a torque of a certain magnitude would result in the rope. They were aware that the South African and European approach of dealing with this torque involved fixing the conveyance end of the rope against rotation upon installation and never thereafter releasing turns except at the top of its run (bank), when re-shackling (re-terminating). This action resulted in minimum overall extension of the rope with a shortening of the lay length at the conveyance end. It was also suggested that the practice of preventing unlaying of the rope was favoured from a rope endurance point of view. Layland et al. indicated that mine rope operators at that time had a basic understanding of rope torsion and were aware that the removal of rotation from a rope in service could have a significant influence on the subsequent lay length variations. It was also appreciated that there could be a correlation between the state of unlaying of a Lang's lay rope and its endurance.

Shelly[4] used theoretical relationships derived from experimental tension–torsion results to predict the in-service behaviour of multi-strand non-spin Koepe balance ropes. Shelly named the constant C, of Equation [1], the "torque factor" and T, the "torsional stiffness".

$$M = C.F + T.\frac{d\phi}{dz} \qquad [1]$$

where :

M	=	rope torque(Nm)
$d\phi\,/dz$	=	rope twist relative to an initial lay angle (°/m)
F	=	rope load (kN)
C	=	constant of proportionality between load and torque (Nm/kN)
T	=	torsional stiffness constant (Nm/(°/m))

After examining typical torque versus load curves for varying twists, Shelly realised that neither C or T, as constants, could describe the general response of a rope over a range of loads and applied twists. An approach was subsequently adopted in which a computer algorithm calculated values of C and

T at fixed increments of load and applied twist. The measured test data formed the input to the algorithm. Similar approaches have since been applied by Yiassoumis[5] and Rebel[6].

As regards further research, Yiassoumis[5] recommended that efforts should be aimed at improving the quantitative accuracy of predictions of rope torsional behaviour. In particular, modelling of the torsional behaviour of the ropes should take into account hysteresis effects. It was expected that this would lead to a significantly more complex algorithm since the condition of each rope element will depend not only on the torque and tension in the element but also on the torque and tension history. Further investigation into the torsional properties of wire ropes was recommended. This included aspects such as the variation of the torsional properties of ropes along their length, and the effective torsional stiffness of wire ropes in service.

Hecker and van Zyl[7], discussed torque-tension rotation behaviour of Lang's lay ropes on drum winders. The explanation of the behaviour phenomenon was similar to that of Gibson[8] and the equation used to relate rope torque to load and twist was of the same form as the equations proposed by Feyrer and Schiffner[9] (which include scaling and constants for different constructions). The boundary conditions used in calculating the in-service rotation and lay length variations (48 mm Lang's lay rope) were based on the same assumptions as Hermes and Bruens[2]. The limitation of their analysis was that it neglected torsional hysteresis behaviour in calculating rope rotation during conveyance loading as well as the actual as-manufactured state of the rope. A certain position along a suspended rope was observed to have rotated 12.5 times during conveyance loading. Their analysis predicted 28 turns at the same point. In making lay length calculations, the rotation boundary conditions were assumed to be zero at the head sheave and at the conveyance. This assumption leads to errors when calculating the in-service lay lengths, as residual torque in the rope after manufacture is neglected. Using an analysis similar to that of Hecker and van Zyl[7], Rebel et al.[10] presented calculated results of in-service lay length and rotation for a 54 mm Lang's lay triangular strand rope. These results were subject to the same assumptions and therefore the same limitations.

Hecker and Van Zyl[7] also measured rope torque and load during the operation of a rock winder. Figure 2 shows a unique example of the measured parameters over a full cycle. Skip position, rope load at the skip and rope torque are plotted against time for a 2200 m length of wind. The rope used on the double drum rock winder had a diameter of 48 mm and was of the triangular strand construction. The skip mass was 9820 kg with a rock payload of 14 070 kg. The load and torque measurements were achieved by strain gauging and calibrating the skip draw bar (connection between skip and rope).

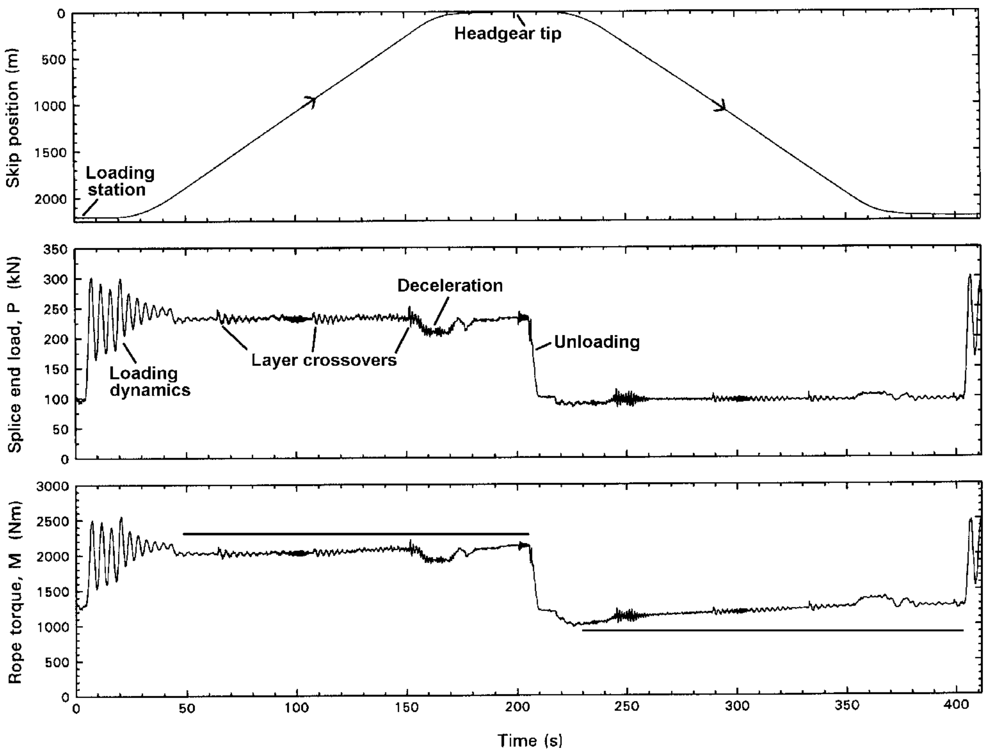


Figure 2 - Measured parameters of a 2200 m double drum rock winder during a winding cycle, adapted from Hecker and van Zyl[7].

The cycle commenced with the rock skip at the loading station where it was loaded. Since the equilibrium torque in the rope is related to the rope load, an initial step change in torque occurred (≈1200 Nm) and the rope would have immediately rotated about its axis to establish the new torsional equilibrium. The torsional properties of a suspended rope are such that any change in torque is associated with changes in the overall state of rotation of rope. The subsequent rope tension fluctuations, which resulted from the dynamics of drum acceleration, layer crossovers and deceleration of the system would have caused the rope to rotate accordingly throughout the trip. Since the dynamic tension fluctuations in every two winding cycles are never exactly the same, drum winder ropes in general pass over the sheave wheel and onto the winding drum with varying contact points, (as a result of load induced rotation) thereby uniformly distributing the wear and plastic deformation of the outer wires (Chaplin[18]).

Due to the complexity of wire rope constructions most modelling of in-service behaviour is reliant on laboratory measured rope properties for input data. The equipment used in this investigation to obtain rope properties has been described in some detail by Rebel and Chandler[11]. Typical measurements include rope torque as a function of tension at constant twist and rope torsional stiffness at various constant tensions (change in torque / change in twist).

1.2 Site observations on existing mines

Rope lay length data obtained from existing mines can give valuable insight into the lay lengths that might be expected for future deep shafts. Figure 3 shows such data collected for 144 ropes operating at different shaft depths, shortly after installation.

Least squares straight lines were fitted to the observed splice (conveyance) and sheave end values. Calculated predictions for a future deep shaft are also shown (squares at 3185 m, not included in the linear fits). Rebel[1] showed that the splice end lay length does not change linearly with depth but the number of factors which influence the observed data makes a linear fit the most sensible choice. The predictions were calculated independently of the observed shaft data using known rope properties and actual hoisting system design details (the calculation technique for which is described in more detail in Section 2).

Maintenance practices on most South African mines are such that rotation is inevitably lost (unlaying) from ropes in service. This leads to an increase in both the conveyance and sheave end lay lengths. Because of the non-linear helical geometry of the rope the sheave end lay length increases more rapidly than the conveyance end. This situation means that at installation in a very deep shaft, the shortened lay length at the conveyance end will most likely be the parameter of operational concern but with unlaying, this condition will improve. The converse is true for the sheave end lay length which, at installation, may be 40% longer than nominal but with time this value will increase (quite rapidly). Were it not for maintenance of high tension, this could result in operational instability in the drum end region of the rope (corkscrewing) but also to increased rope ovality.

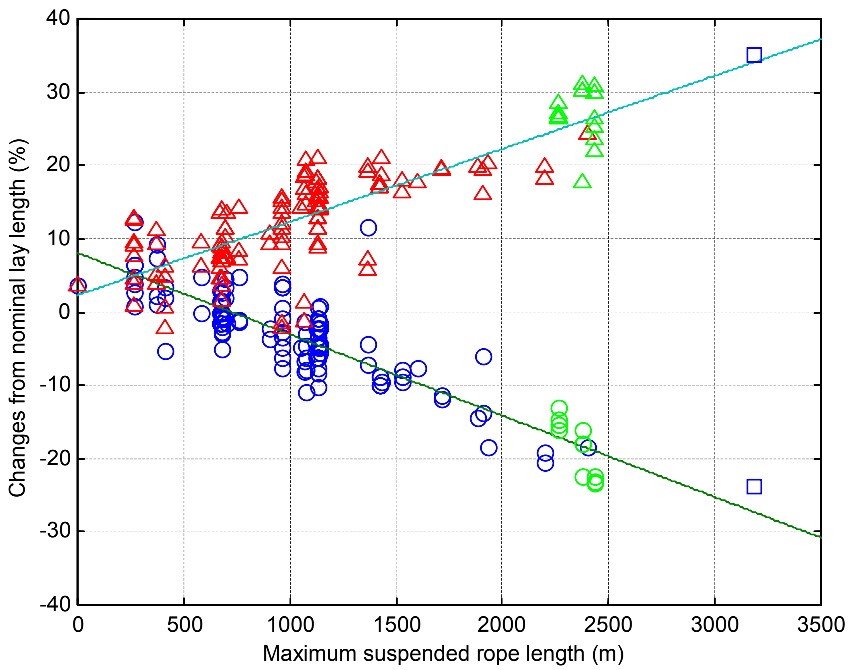


Figure 3 - Installation lay lengths for 144 Lang's lay triangular strand ropes of varying rope diameter, construction and shaft depth. o = conveyance end, Δ = sheave end, □ = independently calculated predictions for a future deep shaft at 3185 m.

1.3 Why is torsional behaviour important at greater depths ?

The first and most obvious issue is that of lay length changes from what is termed the “nominal lay length”. For the ropes with which this study was concerned, the nominal (or design) lay length is on average 7.56 times the nominal diameter. Effects of very short (nominal -20 % to -30 %) lay lengths at the conveyance end include increased likelihood of corkscrewing and kink damage at low rope loads. At the sheave end very long lay lengths (nominal +70 % to +100 %) can lead to ovality problems with possible complications for accurate drum coiling and reduced fatigue life. Preliminary tests by Coward[12] have shown that above an 80% increase in the torsionally unrestrained lay length, the ovality ratio (rope width on a flat drum / rope height) increases rapidly. The rope load for these measurements was 25 % of the minimum breaking load.

Greenway[13] examined the coupled extensional-torsional behaviour of triangular strand ropes in the context of deep level drum winding. A finite element approach was used to assess whether this dynamic coupling was of significance in analysing winding system dynamic behaviour. Although not the focus of Greenway’s study it was noted that there may be some correlation between the static and dynamic changes in rope torsion and rope deterioration.

A detailed investigation of the dynamic torsional behaviour of drum winder ropes was not within the scope of this study, but an experiment was conducted to determine the effect which depth can have on torsional oscillations during rock winder emergency braking. In two nominally similar shafts (760 m and 2000 m deep) an empty rock skip was hoisted from the underground loading station and the winder tripped at ≈10 m/s hoisting speed. Video filming of the ropes from the mid-shaft position showed that, in the 760 m deep shaft, the rope rotated twice to the left and then twice to the right. In the 2000 m case the rope rotated 26 times to the left and the 26 times to the right.

Yiassoumis[5] found that rope rotation on Koepe friction winders is a function of the square of the hoisting depth and for the purposes of this experiment it was assumed that a rope would exhibit a similar relationship on drum winders. It was also assumed that the rope rotation tends to zero as the maximum suspended length tends to zero. An equation of the form (constant . L^2) could however not adequately describe the observed shaft data and so Equation [2] was used.

$$\Delta\phi_{trip} = 4.617.10^{-8}.L^{2.651} \qquad [2]$$

where : $\Delta\phi_{trip}$ = max. dynamic rotation amplitude at mid shaft (turns)

L = maximum suspended rope length (m)

It is evident that the rope torsional response during emergency braking increases substantially with depth (Rebel[1]). In conjunction with a detailed quasi-static analysis Equation [2] was used to extrapolate the response for a 3000 m deep shaft. At this depth the rope is expected to rotate 76.2 times in one direction and then in the other under the same circumstances. It was mentioned earlier that the current limit of experience with triangular strand ropes is 2500 m suspended length. Increasing the suspended length by 500 m could result in a 62 percent increase in the amplitude of torsional oscillations associated with dynamic events. If this proves to be true then there may exist a transition depth where the coupled axial-torsional dynamics of the rope becomes severely detrimental to rope life (by inducing a torsional fatigue mechanism). Recent experiments by Chaplin et al.[14] on six strand right hand ordinary (RHO) and Lang's lay ropes with independent wire rope cores (IWRC) have shown that combined tension torsion fatigue can have a very serious effect on rope life. Depending on the amplitude of rotation it was shown through laboratory tests that rope life could decrease by a factor of ten or more when torsional oscillations were allowed to occur in phase with tension.

As a winding analogy, it is interesting to note that on Koepe hoisting systems there is a transition depth where Lang's lay triangular strand head ropes are no longer suitable. Craib[15] gave a detailed description of problems encountered with triangular strand head ropes on Koepe winders during 1958 and 1959. Modest increases in hoisting depth (e.g. 1070 m to 1350 m) lead to a different rope degradation mechanism and a dramatic fall in rope life. After six weeks of operation on one installation broken wires started to appear and after six months the ropes had to be discarded because of severe strand distortion due to rotation. It was only after a second set of ropes behaved in a similar manner that it was appreciated that there was a systematic problem.

While it is realised that the Koepe system, from a rope point of view, is significantly different to drum winding, the salient point in this example is that ropes which perform satisfactorily at a given depth may not do so in even moderately deeper shafts if increasing depth initiates a different failure mechanism. It is reasonable to expect that there is a critical depth for triangular strand ropes on drum winders at which the effect of torsional oscillations on rope life will become a deciding factor.

2 TECHNIQUE FOR CALCULATING IN-SERVICE LAY LENGTHS

When calculating the in-service lay lengths of drum winder ropes it is important to consider three key lay length definitions :

Nominal or design lay length (LL_{nom})which has a fixed ratio to the nominal or design diameter. For the ropes considered here the average is 7.56. All in-service changes in lay length must be referred to this nominal value for each particular rope.

As-manufactured lay length (LL_m) which is the lay length that the rope is closed at and which it arrives with at the mine. Quality control procedures usually specify that this lay length should be between 0 % to 5% greater than the nominal lay length.

Free or torsionally unrestrained lay length (LL_0), the lay length which the rope will have under zero load and torque conditions. The variation here can be considerably greater and values between 3% and 25% longer than nominal are on record.

2.1 Interpolated rope torsional data

From the standard torque-tension tests (torque versus load at constant twist), torque data files are created for a range of rope diameters. The properties are such that zero torque exists at zero load, at a lay length of 10 % greater than nominal (which is defined as *zero twist*). In practice it has been found that this *zero torque* / *zero twist* lay length is not consistent with values varying between 3 % and 25 % greater than nominal as mentioned above. The value of 10 % greater than nominal is however regarded as a realistic mean. Once created from the actual test data, the torque files are saved in the matrix format shown in Table 1 (i.e. one file for each diameter of rope tested) :

Table 1 - Configuration of torque data files. Such files are derived directly from the tension-torsion tests conducted on triangular strand rope specimens.

	% change from nominal lay length												
Load	80	70	60	50	40	30	20	10	0	-10	-20	-30	
0	..	..	..	..	..	..	..	0	..	..	..	..	
5	..	..	..	..	..	..	..	..	..	..	..	..	
10	..	..	..	..	..	..	..	..	..	..	..	..	
15	..	..	..	..	..	..	..	..	..	..	..	..	
…	..	..	..	..	..	..	..	..	..	..	..	..	
500 **kN**	..	..	..	..	..	..	..	..	..	..	..	..	**Nm**

If several torque files exist (for different diameter ropes) then it is possible to determine by interpolation or extrapolation what the value of torque would be for other diameters (i.e. at discrete values of load (kN) and % change from nominal lay length). Figure 4 shows one such interpolation for a 42 mm triangular strand rope.

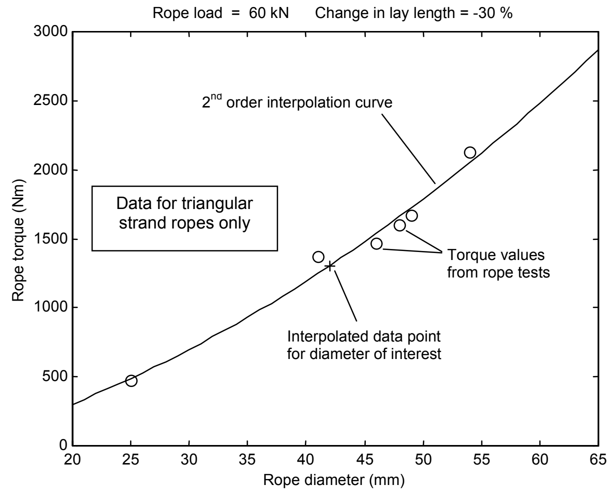


Figure 4 - Interpolation technique for an intermediate rope diameter torque value. The interpolated value is at a specific load (kN) and % change from nominal lay length.

Completing the interpolation for all useful load and % lay length change values results in a set of curves, Figure 5, describing the general full-slip torque-tension characteristics for the interpolated rope diameter of interest.

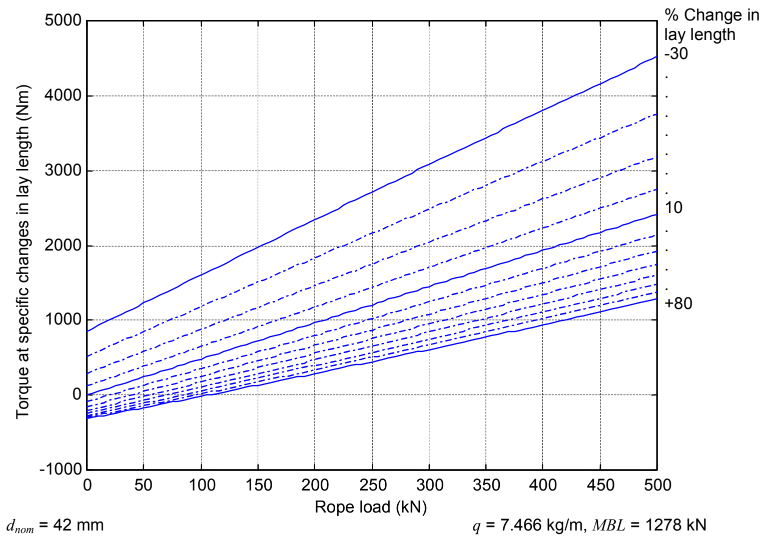


Figure 5 - Interpolated torque-tension curves for an intermediate rope diameter, 42 mm. Actual tests were only conducted on the diameters indicated by the circles in Figure 4.

Knowing the nominal lay length of the rope being modelled, it is possible to define the twists (°/m) associated with the % change in lay length curves, Equations [3] and [4] :

$$R = \frac{\arctan(\frac{\pi\, d_0}{LL}) - \arctan(\frac{\pi\, d_0}{LL_0})}{k_{LL}} \qquad [3]$$

$$k_{LL} = \frac{d_0}{108\,105} . \cos^2\left[\arctan(\frac{\pi\, d_0}{LL_0})\right] \qquad [4]$$

where :

where :	R	=	rope twist (°/m)
	LL	=	rope lay length (mm)
	LL_0	=	rope lay length at zero twist = LL_{nom} + 10 % (mm)
	d_0	=	zero twist rope diameter (mm), i.e. at LL_{nom} + 10 %
		=	1.025 *d_{nom} (from empirical data)
	k_{LL}	=	empirical lay length factor (1/(°/m))

The zero twist diameter, d_0, tends (on average) to be larger than the nominal diameter by 2.5 % (considering previous test data, Rebel[1]). With torque-tension data for the intermediate (interpolated) rope diameter of interest, d_{nom}, now in terms of rope twist, R, rope load, F, and rope torque, M, the curves in Figure 5 can be accurately represented by Equation [5] :

$$M = \begin{bmatrix} R^2 & R & 1 \end{bmatrix} \bullet \left[\begin{bmatrix} N_{11} & N_{12} & N_{13} \\ N_{21} & N_{22} & N_{23} \\ N_{31} & N_{32} & N_{33} \end{bmatrix} \bullet \begin{bmatrix} F^2 \\ F \\ 1 \end{bmatrix} \right] \qquad [5]$$

where :	M	=	rope torque (Nm)
	R	=	rope twist (°/m)
	F	=	rope load (kN)
	N	=	constants determined from 2nd order least squares fit to the interpolated data for the particular intermediate diameter

2.2 In-shaft rope rotation analysis

Static rope load, F, can in be expressed in terms of the vertical distance upwards from the conveyance, z (Equation [6]) :

$$F = P + z.q \qquad [6]$$

where : q = rope weight per unit length (kN/m)
P = rope end load (kN)

Knowing that the torque in a suspended rope must be constant along its length (in the absence of any externally applied torques, other than the end reactions) it is possible to determine the twist distribution $R = f(z)$ along the rope (which in turn gives the rope lay lengths $LL = f(R(z))$, Equations [3] and [4]). The rotation boundary condition at the conveyance is assumed to be zero and at the hawse hole it is consistent with the state of twist in the rope as delivered. The issue here is that the zero twist lay length may be, say, nominal +10 % but the rope is typically closed at nominal +3.7 %. This means that relative to the zero twist state, the rope is twisted up (shorter lay length) when it arrives at the mine ready for installation. Although the rope is effectively at zero tension, it does not necessarily have zero torque.

This calculation procedure has been applied to a Blair Multi-Rope (BMR) winding system, which is representative of installations for the next series of very deep shafts, using the following key data :

d_{nom}	=	42 mm	Nominal rope diameter
MBL	=	1278 kN	Minimum rope breaking load
L	=	3185 m	Maximum suspended rope length
q	=	7.466 kg/m = 0.0732 kN/m	Rope weight per unit length
P_e	=	39.20 kN	Rope end load (empty conveyance)

z_{max}	=	3185 + 350 m		Total rope length
LL_{nom}	=	d_{nom} * 7.56 mm		Nominal rope lay length
d_{0_ratio}	=	1.025	$[d_0 / d_{nom}]$	Zero twist diameter ratio
LL_{0_ratio}	=	1.100	$[LL_0 / LL_{nom}]$	Zero twist lay length ratio
LL_{m_ratio}	=	1.037	$[LL_m / LL_{nom}]$	As manufactured lay length ratio

Figures 6 and 7 show the results of the calculations for the rotation and lay length distributions after an equilibrium condition has been reached at installation.

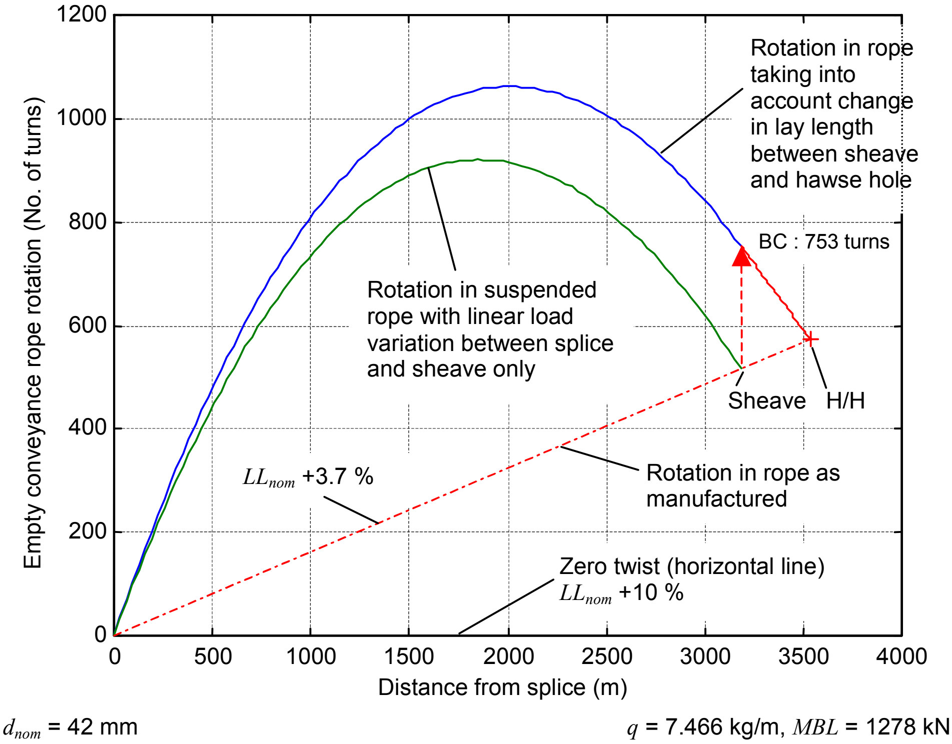
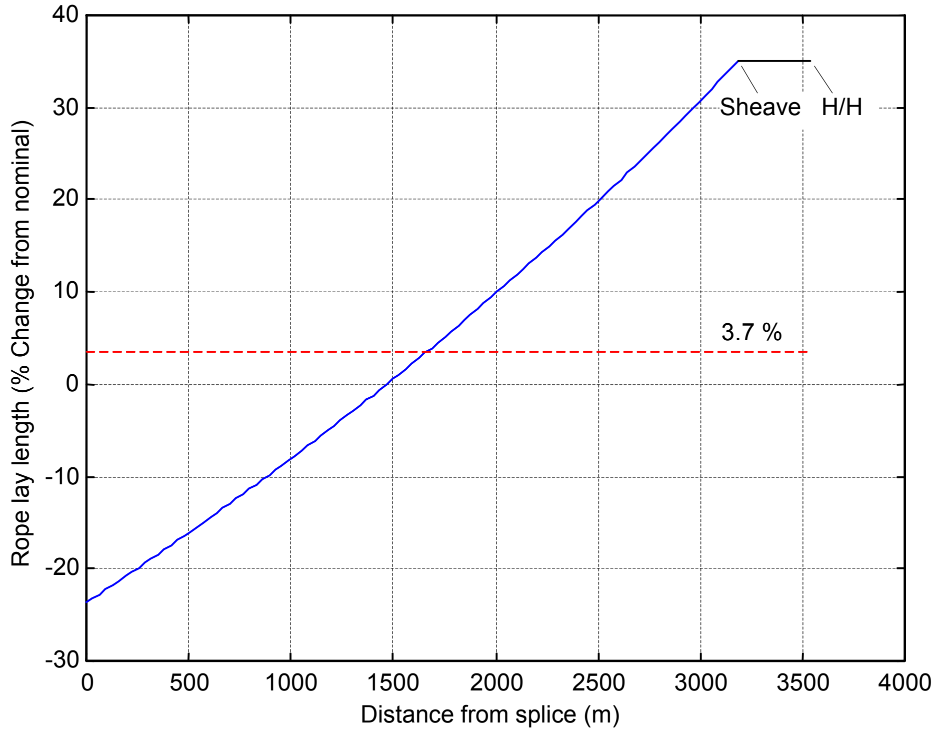


Figure 6 - Calculated rope rotation distribution after installation. The slope of the lines represents the twist in the rope at that distance from the splice. Positive twist (slope) is lay length shortening, negative twist (slope) represents an increase in lay length from the zero twist state (i.e. LL_{nom} +10 %). The upper curved line is used to calculate the lay length distribution along the rope after installation (i.e. rotation, ϕ, gives twist, R, which gives lay length, LL)



d_{nom} = 42 mm $\quad$ q = 7.466 kg/m, MBL = 1278 kN

Figure 7 - Calculated distribution of rope lay length for representative 3200 m winder. The dashed line indicates the lay length at which the rope was manufactured (i.e. closed at LL_{nom} +3.7 %). Note that the lay length between the sheave and the hawse hole with the conveyance at the bottom of the shaft is assumed to be equal to the sheave end lay length.

After achieving an equilibrium distribution at installation the following changes from the nominal lay length are expected :

Conveyance end = -23.7 %

Sheave end = +35.0 %

These values were shown earlier in Figure 3 in relation to site data collected from existing drum winder rope installations.

3 PARAMETERS AFFECTING IN-SERVICE ROPE LAY LENGTH

3.1 As-manufactured lay length and allowing rotation in-service

The lay lengths which are predicted for representative 3200 m deep shaft are not very different from those which have previously been observed at other deep shaft systems in South Africa (Figure 3). A conveyance end lay length shorter than -20 % may make the rope prone to corkscrewing / kink damage in that area. This problem of short lay length can be overcome by releasing some rotation from the rope during or after installation in the shaft. The other alternative is to manufacture the rope such that it has a closing lay length (LL_m) slightly longer than usual. Removing around 550 turns from the rope (in the unlaying direction) after installation would increase the conveyance end lay length from -23.7 % to -20 %, and the sheave end lay length from 35 % to 45 %. The same effect (in terms of lay lengths) could be achieved if the rope was closed at a lay length of nominal +10 % and not +3.7 % as in the initial calculations. The rotational interpretation of these two solutions are shown in Figure 8a and 8b respectively. Note that if the rope is closed with a longer lay length to simulate in-service unlaying then the strands in the rope also need to be closed (manufactured) with proportionally longer wire lay lengths. This arises from the fact that if say 25 turns are removed from a length of rope, the strands in that length are also being individually unlaid by the same amount.

If the lay length at which the rope is manufactured (closed) can be the same as the subsequent free (zero twist) lay length then such a rope may also be more easy to handle during installation, being effectively “dead”. This relationship between the manufactured lay length (LL_m) and the torsionally free lay length (LL_0) is largely a function of the preformation which the six triangular strands undergo during rope closing.

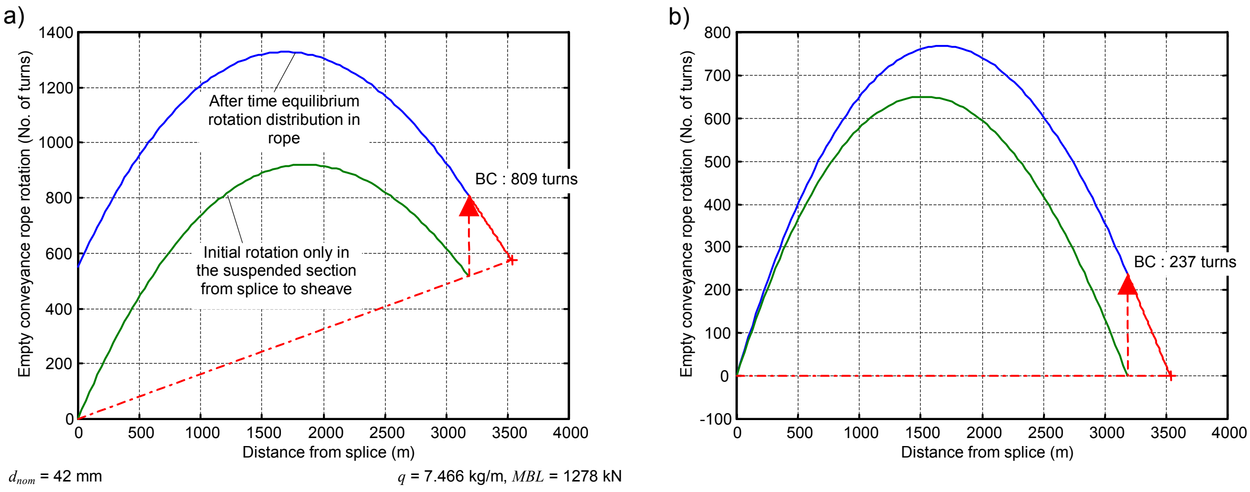


Figure 8 - Calculated rope rotation distributions after installation resulting in a conveyance end lay length of nominal -20 % and a sheave end lay length of nominal +45 %. a) Rope closed at nominal +3.7 % and 550 unlaying turns released after installation. b) rope closed at nominal +10 %, no rotation released after installation. The rotation boundary conditions (BC) at the sheave after achieving twist equilibrium from the sheave to hawse hole are shown.

Current operation experience suggests that it would be prudent to ensure a lay length at the conveyance end of the rope of not shorter than nominal -20 %. It remains a matter of debate as to whether it is better to manufacture the rope with a longer lay length such that it has the correct lay lengths at the time of installation or whether turns should be let out after installation to achieve the same effect. There are arguments suggesting that the less the rope construction is allowed to alter from the accurately controlled manufactured condition, the better its in-service endurance is likely to be.

To achieve sensible conveyance end lay lengths in very deep shafts after installation the sheave end lay lengths tend to be quite long (e.g. 45 % in the above example). This lay length will rapidly increase if the rope is allowed to unlay during maintenance operations, as is currently the practice on most shafts. There is not a great deal of experience (if any) in operating triangular strand ropes at low "rope selection factors" (RSF = 3.5) with lay lengths greater than say 50 % longer than nominal. It is therefore important that mining companies consider rope maintenance methods where rotation is either not released from the ropes (very difficult) or lost rotation is returned prior to reconnecting to the conveyance. This change in practice will prevent the potentially detrimental increase in sheave end lay length over the service period. Lay length extension limits could be introduced for the sheave end which would allow for some unlaying during maintenance operations but in a controlled and recorded fashion.

In another example of a circa 3200 m shaft, Figure 9 shows the effect which allowing the loss of rotation in-service has on the overall lay length distribution in the suspended rope.

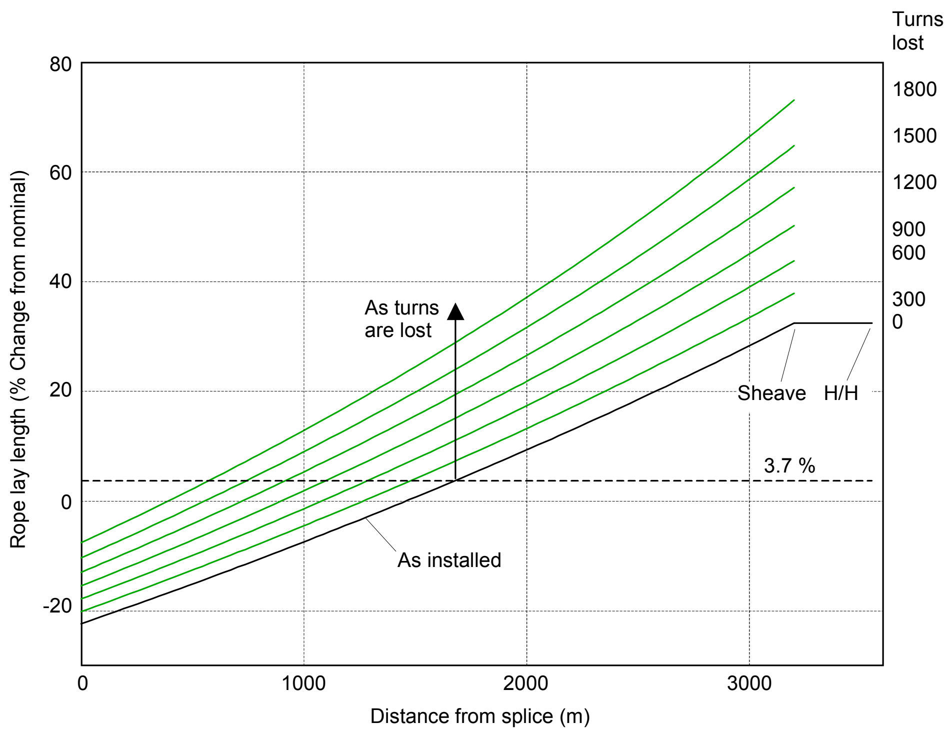


Figure 9 - Example of the effect of lost rotation (in steps of 300 turns) on the distribution of in-service lay length changes for a 3200 m deep shaft. Note that the sheave end lay length increases more rapidly than at the splice end and becomes the area of concern as the service period progresses.

Observations on existing mines have shown that it is not uncommon for several hundred turns to be released from triangular strand ropes over their service periods. This is how the much greater lay lengths seen in existing shafts have developed. Key factors here are the length of the service period, the general frequency of front and back end maintenance operations and also the headgear height. The actual free lay length of the rope, which historically can vary from one rope to another, also has a significant influence on this relaxation process. Man winder ropes will often show much larger sheave end lay lengths as they usually remain in service for a greater period than rock winder ropes. The headgear height affects the length of rope which is allowed to go to zero torque with each maintenance operation. The greater this length, the more turns will be lost for each operation.

Although not accurately quantified, indications are that significant unlaying of triangular strand ropes in service can effect endurance. For this reason, it would probably be sensible to prevent large increases in lay length (to say 80 % greater than nominal) on the 3200 m deep shafts even though there is some experience in operating with these lay lengths at depths of 2000 – 2500 m (but not with RSF = 25000/[4000+L] now allowed by SA Standards / Regulations[16]). It is also known that coupled tension-torsion oscillations in triangular strand ropes increase with an increase in mean lay length which could also have associated detrimental effects on performance as discussed earlier.

3.2 Effect of reducing rope mass per unit length

Lighter ropes with the same diameter and breaking strength are also desirable as the changes in lay length for the triangular strand rope construction are a direct consequence of the tension change with rope length (a function of the rope weight per unit length, q). Beneficial reductions in mass or weight per unit length are in the order of 10 to 20 %. Rebel[1] showed that within this range lay length changes could be delayed by between 500 and 1000 m (i.e. by reducing rope mass per unit length by 10% the same changes in lay length occur 500 m deeper and so on). Naturally there are other economic advantages in using lighter weight ropes of the same strength and diameter. The saving in rope weight can be transferred to increased payloads and higher shaft output with the given winding equipment.

4 DISCUSSION OF ALTERNATIVE SOLUTIONS AND CONCLUSIONS

The analysis of in-service rope behaviour has shown that triangular strand ropes can be operated within currently accepted lay length change limits at shaft depths of at least 3200 m. It is recognised that other multi-layer non-rotating rope constructions are available which would not be prone to such significant variations in lay length at this depth. However, Dohm[17] recently pointed out that the primary concern with using non-spin ropes for deep shaft drum winders is the effectiveness of assessing their level of deterioration with existing magnetic rope condition monitoring equipment. Magnetic testing of single layer triangular strand ropes on the other hand is well proven and known to be reliable. In general these ropes also deteriorate in a manner where the bulk of the broken wires appear on the outer surface of the rope which has been discussed in some detail by Chaplin[18]. These key issues of condition monitoring as well as the ruggedness of the triangular strand design for drum winding make them the most sensible option at this stage. If the lay length change situation is managed correctly it need not limit the application of triangular strand ropes in very deep installations.

Rope manufacturing technology is such that modest changes to the current preferred triangular strand rope design (e.g. strand and rope lay lengths) are feasible. This, in conjunction with the analysis presented in this paper, provides the possibility to tailor ropes for a specific installation. For example, in the case of the 3200 m winder described in Section 2.2, it would be possible to produce a rope which upon installation has acceptable conveyance and sheave end lay lengths. This will remove the necessity for the mine to have to release rotation and simultaneously reduce the risk of damage to the rope and hoisting system downtime. It was suggested earlier that from a performance point of view this would be a preferred approach as the rope would be operating as close to its manufactured condition as possible.

Problems are not foreseen as far as controlling lay length increases in service are concerned. This will require a change to current rope maintenance practices and more detailed record keeping of the amount of rotation released. Major South African mining companies are already investigating the feasibility of hydraulic equipment which would allow rotation to be returned to unlaid ropes in a safe manner.

Reductions in rope weight per unit length of the order of 20 percent could make suspended lengths of greater than 3500 m viable. This obviously does not take into account the amplitude of torsional oscillations associated with dynamic events which was discussed in Section 1.3. Figure 10 shows the cross-section of a triangular strand rope in which the "Plaited" triangular cores have been replaced by a polymer profile or polymer-fibre composite. McKenzie[19] proposed that extruded polymer profiles or polymer strands could be used to replace wires in the inner triangular strand ropes of multi-layer non-spin ropes. This concept is also applicable to triangular strand ropes on their own. It was suggested that stranded polypropylene (rope) cores with an extruded polypropylene jacket may be particularly suited for triangular strand ropes operating at great depths (McKenzie[20]).

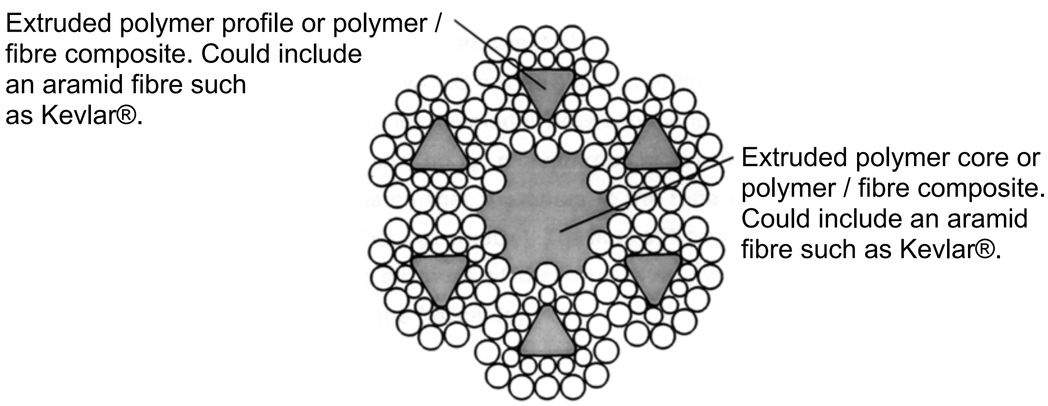


Figure 10 - Option for reducing the weight of triangular strand ropes while maintaining the breaking strength and diameter. Adapted from Rebel[1] and McKenzie[19].

It is probable that the weight of a triangular strand rope could be reduced by almost ten percent by replacing the "Plaited" core with a polymer profile and increasing the tensile grade of the remaining wires to maintain the breaking load. For a 42 mm 6x29 rope, the core represents approximately 9,7 percent of the metallic area. It is however expected that a twenty percent reduction in weight per unit length would require the addition of light weight, high strength material into the rope cross-section. In such a case, it would be best to maintain the outer wire diameters so as not to affect the abrasion resistance of the rope (required for drum winding). The diameter of the second layer of strand wires could be reduced to reduce the metallic content (may also be required for ten percent reduction). Both the extruded polymer strand core and the rope core could incorporate aramid type fibres such as Kevlar®. In the past, Kevlar® has been used for ropes particularly where rope weight is problematic. Kevlar® has a relatively low resistance to abrasion but this can be overcome by protecting the fibres with a suitable matrix. McKenzie[20] proposed that in the triangular strand rope the protective matrix could be formed by the outer extruded jacket of the strand and rope cores.

Attempts at including non-metallic components in a rope must not result in any significant loss in transverse stiffness. Considerable variations in diameter could negatively affect the coiling patterns on a winder drum. The introduction of non–metallic strength components into the rope cross-section could also complicate the condition assessment of the rope in service as conventional magnetic inspection techniques would not be able to detect broken fibres. A further concern would be the differences in moduli between the fibres and the rope wires as this would affect the load sharing for a given rope strain. A possible solution to this problem would be to ensure that the fibres in the rope core are almost parallel to the axis of the rope (i.e. virtually no helix geometry). The fibre stiffness would then be fully realised whereas the effective stiffness of the wires (forming the strands and rope) is reduced as a result of the helical lay and core compaction.

5 RECOMMENDATIONS FOR EXTENDING OPERATIONAL DEPTH LIMITS

The key technologies which could extend the depth limits for triangular strand ropes include :

1. **Either** making the triangular strand ropes with longer lay lengths so that there are no problems associated with lay length shortening at the front end just after installation, **or** developing reliable procedures for releasing controlled unlaying rotation from the rope during or immediately after installation. In the case of closing the ropes with longer lay lengths, the strand lay lengths would also have to be increased in a similar proportion to the rope lay length (as the same occurs when a ropes is unlaid in-service).

2. Development of maintenance procedures which prevent progressive unlaying of ropes in service. This would most likely be in the form of hydraulic or air powered equipment to return lost rope rotation, under controlled conditions, prior to reattachment to the conveyance.

3. The design of light weight ropes of the same basic Lang's lay triangular strand construction so as to reduce the extent of torsional deformation and also improve the overall efficiency of any given hoisting system (less dead weight and more payload). Target values for rope weight reduction are in the range -10 % to -30% from current design figures.

6 REFERENCES

1. Rebel, G., *The torsional behaviour of triangular strand steel wire ropes for drum winders*, PhD Thesis, University of the Witwatersrand, Johannesburg, South Africa, July 1997.

2. Hermes, J.M., Bruens, F.P., *The twist variations in a non-non-spin rope of a hoist installatio*n, Geologie and Mijnbouw (NW. SER.), 19e Jaargang, November 1957, pp 467-476 (in Dutch).

3. Layland C.L., Rao, B.E., Ramsdale, H.A., *Experimental investigation of torsion in stranded mining wire rope*s, Institution of Mechanical Engineers Proceedings (B) 1952-53, Vol 1B, pp 323-336.

4. Shelly, P.D., *An analysis of tail rope behaviou*r, The South African Mechanical Engineer, Vol. 24, December 1974, pp 1-18 (reprinted).

5. Yiassoumis, J.M., *Torsional behaviour of wire ropes for Koepe winder*s, M.Sc. Dissertation, University of the Witwatersrand, Johannesburg, South Africa, 1992.

6. Rebel, G., *Model of the torque-tension rotation behaviour of a 48 mm triangular strand steel wire rope operating on a drum winder*, B.Sc. Engineering Dissertation, University of the Witwatersrand, Johannesburg, November 1993.

7. Hecker, G.F.K, van Zyl, M.N., *Drum winder rope test facility : a feasibility stud*y, CSIR Contract Report MST(93)MC1233, January 1993.

8. Gibson, P.T., *Wire rope behaviour in tension and bendin*g, Proceedings of the First Annual Wire Rope Symposium, Denver Colorado, Published by the Engineering Extension Service, Washington State University, pp 3-31.

9. Feyrer, K., Schiffner, G., *Torque and torsional stiffness of wire rope Parts I and II*, Wire, 36(1986)8, pp 318-320, Wire, 37(1987)1, pp 23-27.

10. Rebel, G., Borello, M., Chandler, H.D., *On the torsional behaviour of triangular strand hoisting rop*e, Journal of the South African Institute of Mining and Metallurgy, Vol. 96, No. 6, November 1996, pp 279-287.

11. Rebel G., Chandler H.D., *A machine for the tension-torsion testing of steel wire rope*s, OIPEEC Bulletin No. 71, June 1996, pp. 55-73.

12. Coward, D.B., *Distortion of triangular strand ropes with elongated lay lengths on flat winding drums*, Haggie Rand Limited Product Development Department Report No. 0892, 3 July 1997.

13. Greenway, M.E., A *finite element study of the coupled extensional-torsional behaviour of winding rope*, Internal Anglo American Corporation Report No. 91-T-16, Johannesburg, October 1991.

14. Chaplin, C.R., Rebel, G., Ridge, I.M.L., *Tension/Torsion Interactions in Multicomponent Mooring Lines*, Offshore Technology Conference Proceedings, Paper No. 12173, Houston, Texas, 1–4 May 2000.

15. Craib, S., *Winding from deep levels in South Afric*a, Fourth International Mining Conference, Paper D2, 1965, pp 1-10.

16. South African Bureau of Standards, *SABS 0294 : Proposed code of practice for The performance, operation, testing and maintenance of drum winders relating to rope safety*, Pretoria, October 1998. Rope factor of safety equation included from Regulation 16.34.2 of the South African Mine Health and Safety Act, 1996 (Act 29 of 1996).

17. Dohm, M., *An evaluation of magnetic rope testing instrument defect detection capabilities, particularly in respect of low rotation, multi-layer rope constructions, OIPEEC Technical Meeting – The Non-Destructive Testing of Rope, Krakow, September 1999.*

18. Chaplin, C.R., *Hoisting ropes for drum winders - the mechanics of degradation*, Proceedings of the Third International Conference, Mine Hoisting 93, Royal School of Mines, London, 28-30 June 1993, pp 12.1.1-12.1.10.

19. McKenzie, I.D., *Patent specification : reduced mass rope*, South African Provisional Patent Application No. 89/8857, 21 November 1989, Applicant : Haggie Rand Limited.

20. McKenzie I.D., *Steel wire hoisting ropes for deep shafts*, International Deep Mining Conference : Technical Challenges in Deep Level Mining, Vol 2, Johannesburg, SAIMM, 1990, pp 839-844.

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