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# The Fatigue and Degradation Mechanisms of Hoisting Ropes


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## ABSTRACT

Based upon specialised experience of rope mechanics spanning over 20 years, this paper reviews the processes of degradation and fatigue that are relevant to hoisting ropes in mines. The review is brought up to date with an account of the most recent work in this field, which identifies a torsional fatigue process and quantifies the impact of degradation upon the residual service life. A proper understanding of these processes is important in determining how different parameters of hoist design and operation interact to determine rope life. This knowledge is also important in informing decisions relating to rope discard based upon observed condition, as well is identifying the critical features that must be quantified reliably during inspection.


## INTRODUCTION

In the context of past, present and foreseeable technology, it is an accepted fact that rope used for hoisting in mines has a finite life. The continual process of degradation associated with operational service will ultimately lead to failure, and a hoisting rope must therefore be replaced before the risk of such failure becomes unacceptable. The process of service degradation is complex and different for each installation, reflecting the local operating parameters and the characteristics of the rope employed.

Serious damage, sufficient to warrant discard, may be induced by single events such as a major slack rope incident or a lightning strike. But in the long run, and in well maintained and operated installations, fatigue will play a role. However, fatigue of rope is in practice rather different from, and more influenced by other processes than, the comparatively straightforward 'crack propagation' of a fatigue crack in a contiguous metal component under fluctuating stress.

The great majority of safety critical rope applications involve fatigue coupled with other degradation processes, which together determine a finite service life. This combination is reflected in the inspection and discard policies employed on rope systems, as well as in system design and operation.

So for enhanced safety in hoisting, for improved efficiency in rope utilisation and for better design and maintenance of hoisting systems, a better understanding of rope fatigue is desirable. This paper reviews available knowledge of rope fatigue processes, introduces results of recent investigations and presents the whole in a context of mine hoisting.

## FATIGUE OF WIRE ROPE

In essence, the process of fatigue in metals involves crack propagation from some stress concentrating defect, by mechanisms that involve local plasticity under the influence of a fluctuating load. In practice there are often exacerbating environmental factors, and the 'defect' may be some geometrical stress raiser such as a step in a shaft, or some seemingly minor blemish. Very localised surface stresses (fretting) are often linked with the initial stages of fatigue crack propagation.

In the fatigue of engineering components it is unusual to have other than a single crack, and in laboratory experiments it is common to observe very significant scatter in the fatigue endurance recorded in similar fatigue tests. The parameter that is found to dominate the endurance, or number of cycles to failure, is load range.

Wire ropes are constructed of an assembly of steel wires. Typically, the steel used has a very high strength, which may be a factor of five greater than the strength of typical structural steels. This high strength is achieved by using a plain carbon steel with high carbon content and a very fine grain structure achieved through isothermal transformation (patenting), and work hardening by successive drawing. The division of the load bearing capacity between many 'parallel' wires (in the sense of redundant load paths) has two essential benefits: it assures the essential combination of high axial strength and stiffness with bending flexibility; and allows the structural use of an essentially brittle steel at very high stresses by subdividing the structure to isolate local fractures in much the same way as a fibre composite achieves toughness. The latter point is important in ensuring that a wire rope is 'tough' in the sense that it is tolerant of local damage, particularly in the form of broken wires.

Wire ropes operate at high stress levels and are almost invariably subject to fluctuating loads. In running ropes a significant source of stress fluctuation is the repeated bending and straightening as ropes run over sheaves, and on and off drums. Axial loads also fluctuate, for various reasons. Given time and a sufficiently high fluctuation in stress range, fatigue is inevitable. However, in a wire rope, due to the loose coupling between wires, complete failure of the rope requires that many wires are broken in fairly close proximity. But the fatigue of a single wire in the rope is invariably more than a simple matter of fluctuating stress; there is usually some other process that exacerbates and accelerates the fatigue and which focuses the process to specific locations. This process might be fretting between wires, or be linked to another degradation mechanism such as wear or corrosion.

In practice therefore, rope fatigue involves a large number of fatigue processes going on in series (at different locations along each wire) and in parallel (similar processes along each of the many wires) and rope failure occurs when the accumulation of wire breaks in a locality is sufficient to precipitate total failure. One interesting consequence of this requirement for multiple wire fatigue failures is that whilst there is a characteristically wide statistical scatter in the numbers of cycles for the isolated failure of a single *wire*, there will be many broken wires distributed throughout the rope before *rope* failure. This leads to an 'averaging' process that results in strikingly little scatter in the cycles to *rope* failure for samples from one manufactured length. The Palmgren-Miner cumulative damage approach to evaluating the cumulative effect of cycles at different levels of loading also works well with wire rope, arguably for the same reasons. But there are exceptions to this and it is important to be sure when using this approach that the fatigue damage being summed takes place at the same location in the rope.

The primary mechanisms responsible for stress fluctuations in ropes can be grouped under four headings: tension-tension, bending-over-sheaves, free bending and torsion. Each of these will be considered in turn, and then in combination.

### Tension-tension fatigue

This category of rope fatigue is probably the simplest, involving stress fluctuations resulting from changes in axial tensile loading. In any fixed rope, such as the stay for a tall mast or for a crane jib, it may be the sole class of fatigue. It is a major consideration for mooring ropes. It is also relevant to any lifting or hoisting


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application (including mine hoisting) where attached mass changes and acceleration are the primary sources of axial load fluctuation. For this type of fatigue, dominant parameters are:

- tensile load range;
- mean load (but note that the requirement that load is always tensile limits this effect);
- rope construction and wire grade;
- environment (including lack of effective lubricant, or exposure to corrosion); and
- manufacturing quality.

Of these, the dominant parameter is load range, and a good model for tension-tension fatigue performance is provided by a simple power law equation, as commonly used for fatigue processes, whereby:

Number of cycles to failure, $N = (\text{constant/load range})^m$

and the power $m$ typically has a value of about five, but may be much higher for rope with small wires.

Note that provided rope temperature is not increased by energy dissipation to a level that affects lubrication, frequency is not a consideration. In addition, terminations should not influence rope performance provided a suitable system is used (eg resin sockets) and it is correctly applied. The issue of rope quality relates to the variation in load sharing both between wires and along any one wire, which results from the dynamics of the manufacturing process. This can have a very significant influence on the relative performance of nominally identical ropes (from different makers, or from different lengths), affecting fatigue life by as much as an order of magnitude (Chaplin, 1995). The quantitative issues of this feature of rope manufacture have now been thoroughly explored by Chaplin, Ridge and Zheng (1999a) and Evans, Ridge and Chaplin (2001). The mechanism also provides the basis for explaining the beneficial effects of overloads which, by generating a more uniform load distribution, can enhance tension-tension fatigue endurance, especially in a rope of initially poor quality.

## Bending-over-sheaves fatigue

This is the name traditionally used to describe the process of repeated bending under constant tensile load: it is a topic which has been the subject of considerable experimentation, especially at Stuttgart by Feyrer (1995) and before him Muller (1961), but also notably by Scoble (1930) and Gibson (1980). The primary sources of the stress fluctuations in this mechanism are the local changes in wire curvature as the rope adapts to the radius of a sheave or drum. However, the restriction of the source of fatigue stresses to changes in wire curvature requires that wires can slide with respect to one another. Any constraint on this freedom, for example by ineffective lubrication or internal corrosion, can impair fatigue endurance.

The principle parameters are:

- D/d ratio, the ratio of sheave diameter to rope diameter (which would relate more closely to the bending stress magnitude if defined as sheave to wire diameter ratio);
- tensile load;
- angle of wrap (arc of contact);
- bending length (or bending stroke);
- fleet angle;
- groove geometry;
- rope construction and wire grade;
- environment;
- lubrication; and
- rope quality.

Of the parameters listed above, for a well maintained system, the first two are normally the most important and their effect is illustrated in Figure 1. However, very short bending lengths or low angles of wrap can lead to a significant increase in life because the rope does not fully conform to the sheave curvature. Adverse combinations of fleet angles and groove geometry can cause additional degradation, which in turn compromises bending fatigue performance, causing wear, or introducing turn.

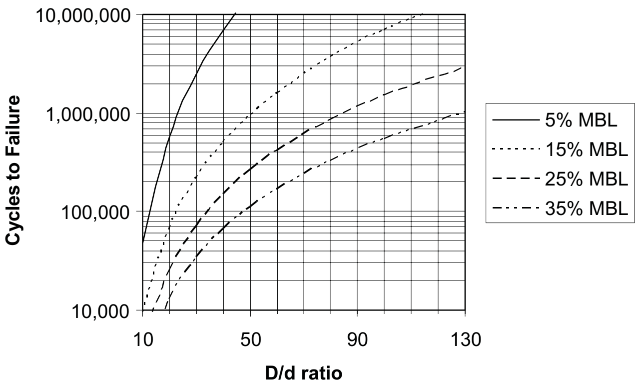


FIG 1 - The mean number of cycles to failure in bending-over-sheaves fatigue as predicted using a numerical model presented by Feyrer (1995). The plots are for a 6 × 36 IWRC, ordinary lay 32 mm rope, as a function of D/d ratio for constant tensile loads, in terms of manufacturer's minimum breaking load (MBL).

There are several geometrical parameters that determine whether a rope will contact the flank of the sheave groove. These are: groove root radius (usually specified in terms of nominal rope diameter), groove flank throat angle, actual rope diameter (in relation to nominal diameter) and D/d ratio. Figure 2 shows the result of geometrical analysis to determine limiting fleet angles as a function of D/d ratio. Apart from the effects of different recommendations for groove profile (here as per BSI or API), the need for lower limits on fleet angles at higher D/d ratios is marked. It should also be noted that actual rope diameters, as supplied, are generally greater than 'nominal', with typical tolerances allowing up to four per cent over the nominal diameter. Because this can depress the fleet angle at which contact is avoided, this can have a significant effect upon performance through abrasive wear or induced rotation.

Rope construction can be a significant issue in bending fatigue particularly because of the response to transverse loading on sheaves or drums. In ropes with 'equal lay' constructions and a single layer of outer strands, forces are transmitted between wires by line contacts, whereas with compound strands or multi-layered rotation-resistant constructions the transmission involves point contacts between wires. In the latter case this tends to induce internal wire fatigue failures that are not externally visible: a feature which must be considered when defining inspection methods and discard policy.

The rope quality issues described in the context of tension-tension fatigue are also relevant in bending, but their effects are less significant due to the predominance of the D/d bending ratio in this mode.

Where bending is induced as rope runs on and off a multi-layer winch drum, life may be expected to be significantly reduced in comparison to that found when running over a sheave that has a nominally similar D/d ratio. Recent work at the University of Stuttgart (Weiskopf, 2005), simulating winding on and off drums on cranes, indicates endurances may be reduced by factors as high as 50. But this comparison is somewhat simplistic: in addition to the far more severe contact situation when supported

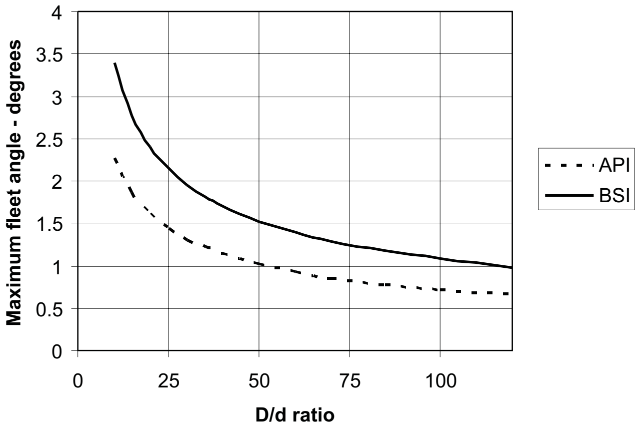


FIG 2 - The maximum fleet angle to avoid contact between rope and sheave groove as a function of D/d ratio and for grooves to BS6570 (upper line) and API RP 9B (lower line). The BSI groove has an included angle between flanks of 52° and root radius at nominal +10 per cent, while the API groove has an included angle of 30° and groove radius at nominal +8 per cent. In both cases it is assumed that the rope diameter is at a maximum tolerance of +4 per cent above nominal.

by rope in the layer beneath, which Weiskopf recognises, there is also the issue that at the crossovers, which typically occur twice per turn, the D/d ratio is effectively reduced to a much lower value (Chaplin, 1994). Furthermore, the D/d ratios investigated are quite severe in comparison to mine hoists (cranes typically <35, whereas mines >80), and the crossovers typically more abrupt. There are further factors that can affect rope running on and off drums, which introduce additional degradation mechanisms: these are discussed below in the context of mine hoisting.

## Free bending fatigue

Free bending fatigue involves fluctuating bending deformation of the rope, which does not involve contact with another body and which is typically excited by system dynamics. A useful qualification is that the curvatures developed are seldom as severe as those typical of ropes running over sheaves, or on and off drums, although frequency might be high. In fixed rope applications this type of bending often takes place adjacent to a termination, which introduces additional local problems and life may then be a concern. Lateral oscillation of the cables of cable-stayed suspension bridges is an application where this category of fatigue can be significant (Siegert and Brevet, 2003).

## Torsion fatigue

The construction of a typical wire rope, with a large number of wires combined so they share the tensile load, results in overall properties that combine axial strength and stiffness with bending flexibility. An unintended consequence of the construction is that the rope also has a low torsional stiffness in comparison to structural components of similar axial properties. A further consequence of the geometry of many categories of rope is that they are torsionally active, generating a torque in response to the tensile load when the ends are constrained against rotation, or conversely twisting about their axes when one end is not constrained.

There are of course rope constructions designed to minimise this tendency to rotate but often these have other disadvantages, which might include being less robust, having a propensity to break up internally (Dohm, 2000; BS 6570, 1986) and high cost. It is also the case that in many applications a combination of fixed ends and lack of significant variation of tension along the rope means that even for torsionally active ropes there is no rotation. However, in some applications, especially where components with different torsional characteristics are connected end to end, torsional oscillations can be induced in response to tension fluctuations. An application where such problems arise is in the moorings of floating offshore systems with hybrid mooring lines combining polyester fibre rope and torsionally reactive, six-strand wire rope (Chaplin, Rebel and Ridge, 2000). Under these conditions the wire rope may experience a torsional mode of fatigue for which the dominant parameter seems to be twist amplitude (Chaplin, 2005).

Little is known about this kind of fatigue, and current work at the University of Reading is investigating the quantitative role of different parameters. The majority of tests performed at Reading to date have combined torsional oscillation with tensile fluctuations, the coupling being achieved by attaching the six-strand wire test rope in series with a torque balanced polyester fibre rope. By adjusting the relative lengths different combinations of twist were obtained. The full details of this work are awaiting publication, but the salient features are indicated in Figure 3. The primary location of wire breakage seems to be the contact between outer wires of the outer strands and the core of the rope, and broken wires become apparent as they loop out of the construction. Although yet to be investigated it would seem reasonable to expect a better life in torsion when the core is fibre rather than IWRC (independent wire rope core), which has been the focus of Reading's work to date.

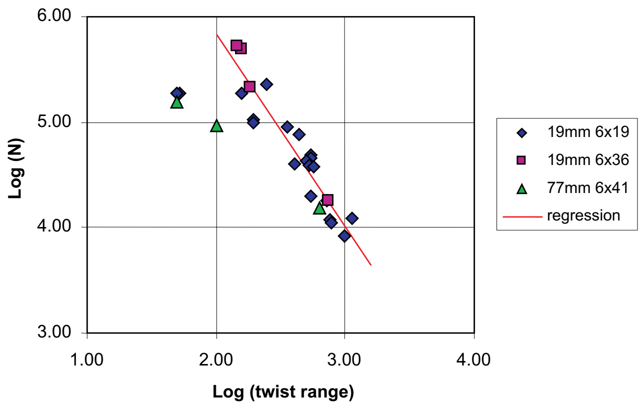


FIG 3 - The cycles to failure in torsional fatigue of six-strand wire ropes of different diameters and constructions plotted as a function of twist range. The regression line does not include the four left-hand data points where the endurance is considered to have been determined by tensile load amplitude.

What is very clear from the investigations to date into this mechanism, is that where the torsional oscillations are a consequence of tensile load fluctuations, the endurance can be very substantially reduced from the level associated with tension-tension fatigue. Figure 4 shows results from Figure 3, but with life expressed as a proportion of the tension-tension fatigue endurance at the same tensile loading, in relation to the level of cyclic twist. Whilst at low levels of twist the life tends to that of the sample with full torsional restraint, at high levels of twist fatigue endurance is reduced in some instances by a factor of 50.

## Combined modes

Laboratory testing inevitably tends to idealise the nature of the fatigue loading imposed upon ropes, and this is helpful in aiding our understanding of the roles of different parameters, but in service things are seldom so simple. The practical operating

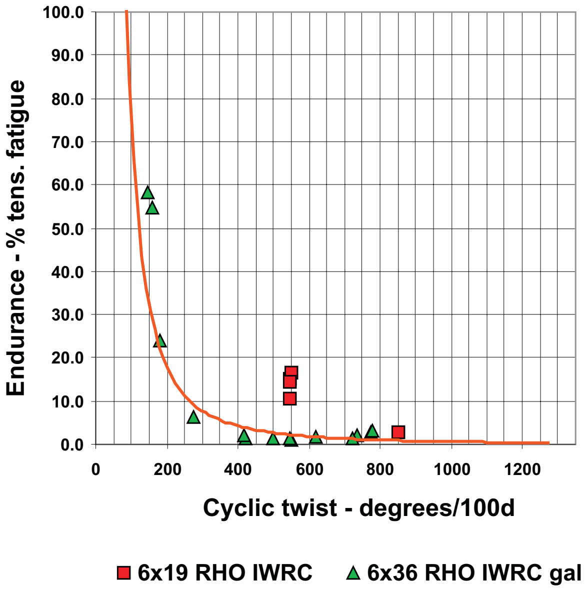


FIG 4 - Fatigue endurance for 19 mm diameter six strand wire rope of two different constructions, but both ordinary lay and with IWRC. The tests involved axially coupled wire and non-rotating fibre rope (different levels of rotation being achieved at different load levels and with different sample lengths). The endurance is expressed as a percentage of the tension-tension life for the same rope at the same axial fatigue load. The abscissa is the range of cyclic twist.

conditions of ropes are such that not only do different types of fatigue mechanism operate in combination, but also the fatigue parameters are not consistent. Furthermore there are usually additional processes, such as wear and corrosion, introducing degradation that physically alters the rope and its fatigue performance. In some instances it is possible to make simplistic assumptions that allow a safe (ie exaggerated) estimate to be made of such effects, but this should be approached with caution.

## *Fatigue modes in combination*

Lifting operations typically involve different tensile loads at different stages, as well as running over sheaves and onto drums. Feyrer (1995) gives a method for modifying the bending-over-sheaves prediction to incorporate tension changes, but these must be at the same frequency. An alternative is simply to employ a cumulative fatigue damage summation, on the assumption that the fatigue damage induced by the different mechanisms is co-located but independently determined.

A rather more subtle set of circumstances exists when bending length (the movement of the rope on and off a sheave) is very short. Here there is a transition range between a modified bending-over-sheaves regime, with accentuated tensile load, and tension-tension at small amplitudes (Chaplin and Potts, 1991).

Experimental work on the combination of tension-tension and torsion suggests these mechanisms are essentially independent, except for an ill-defined transition at low twist amplitudes (see Figure 4).

## *Fatigue damage distribution*

Different points along a running rope are typically subjected to different levels of fatigue damage. This is best illustrated in the context of a mobile crane. When lifting a given load different parts of the rope will go round different numbers of sheaves, depending on the reeving but also depending on the jib extension and the height of the lift. Some parts of the rope will only go onto the drum. This results in a very uneven distribution of bending fatigue damage along the rope. The next lift may be quite different, generating another uneven distribution of fatigue damage, which is then superimposed upon the previous damage.

A careful review of the spatial distribution of accumulated damage can be informative in optimising the kind of 'slip and cut' policy adopted in many offshore applications, such as drilling line or riser tensioners (Bradon and Chaplin, 1999).

## *Allowance for the influence of degradation*

Degradation such as abrasive wear or corrosion can reduce the load bearing area of a rope. Discard decisions should be based upon a 'safe estimate' of residual service life, which is obviously affected by degradation. One approach to dealing with this is to calculate the *effective* loss of the cross-section, load bearing area of the rope. This is not quite the same as strength loss, since at normal operating loads the effective area lost will take into account slack wires, for example. The concept embodied in this approach is to estimate the effective increases in operating stresses induced by degradation, whether that be actual area lost or elements which through local deformation no longer share the load under operating conditions.

Consider a $6 \times 36$ rope in which external abrasive wear has reduced the outer wires along the strand crowns by 50 per cent of their diameter. As a 'snap shot' of the cross-sectional this represents a loss of some 4.5 per cent of the cross-sectional area, as illustrated in Figure 5. But if the wear is uniform over a few rope diameters and around the circumference, every one of the outer wires will in turn come to the strand crown position and suffer the same loss, so it must be assumed that the effective load bearing area has been reduced by the total of all outer wires losing 50 per cent. This amounts to some 22 per cent of the total area (the outer wires constituting some 44 per cent of the total in this construction).

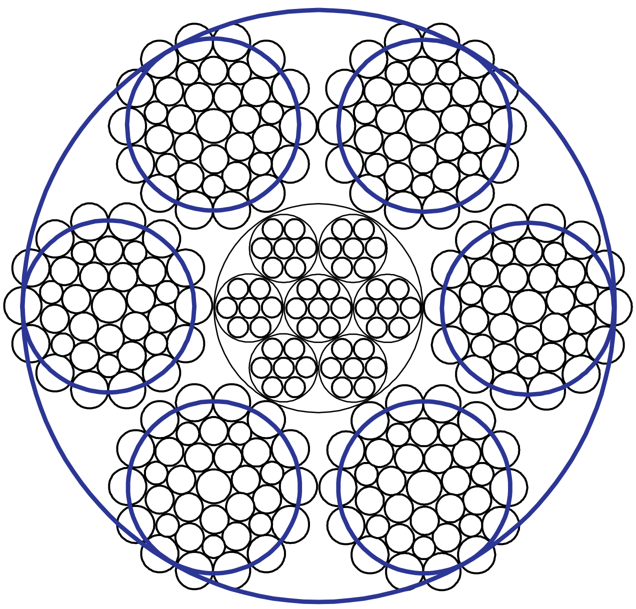

FIG 5 - The idealised cross-section of a $6 \times 36$ rope with IWRC, in which the circumscribing circle represents the physical boundary of uniform external abrasive wear. The wear illustrated has removed steel around a cylindrical surface to half the diameter of the outer strand wires along the strand crowns, representing a nominal loss of cross-sectional area of some 4.5 per cent. The *effective* loss of area, in which all outer wires have the same level of wear, is equivalent to the situation illustrated by the six inner circles, which corresponds to some 22 per cent of the area.

Thus the impact of this apparent area loss of 4.5 per cent of area is *effectively* 22 per cent, which might therefore be considered as a proportionate increase in stresses of 28 per cent (ie 22/(1 - 0.22) per cent). Were such an increase in loading applied to the original rope the impact upon fatigue can be calculated using the appropriate equation (Chaplin, 1992). For tensile fatigue governed by a simple power law equation, and assuming that the relevant power is five, then the proportionate reduced life is $(1 - 0.22)^5$, which is 29 per cent, or a loss of 71 per cent. A similar procedure can be employed to calculate the effects on bending fatigue using Feyrer's (1995) equations for endurance. In practice the loss may be greater, and some part of the fatigue life would normally have been used before the degradation developed, so in as much as residual life is concerned this can only be an upper bound. Nevertheless the scale of the effect can be calculated and has been found to provide an estimate that correlates quite well with test results for new rope containing artificial degradation of different types (Chaplin, Ridge and Zheng, 1999b; Ridge, Chaplin and Zheng, 2001).

## FATIGUE OF MINE HOISTING ROPE

In comparison to other 'running' rope applications, the characteristics of mine hoisting relevant to rope fatigue are:

- high tensile loads and a consequently low factor of utilisation (the ratio of strength to maximum static tension ~ the so-called 'factor of safety');
- fairly high load range (especially for rock hoists);
- high D/d ratios;
- potentially very significant suspended lengths, such that rope weight contributes appreciably to tension;
- high speed operation, with potential associated dynamics; and
- possibility of exposure to corrosive media.

Inevitably, there are combinations of different fatigue mechanisms as well as interactions with other degradation mechanisms. These issues, and especially the interactions, are best discussed in the context of the two main classes of winding mechanism: Koepe (or friction) winders and drum winders.

### Koepe winders

In a Koepe (friction) winder the concept is that given a well matched tail rope, the tension difference across the driving sheave is attributable to the payload only plus acceleration and deceleration components. This minimises the driving torque to the sheave, but whether the drive is mounted on the ground or tower-mounted directly above the shaft, the extremities of the head rope are subjected to the full tension range, and the rope is effectively 'end-for-ended' twice during each full cycle (Berry and Wainwright, 1962; Hermes and Bruens, 1957). Where a shaft is particularly deep this will add appreciably to the tension range, the primary factor in tension-tension fatigue. Koepe systems generally have multiple ropes acting in parallel, providing opportunities for rope tensions to become unbalanced, adding further potential to the load range of the most heavily laden ropes. There are additional considerations, however, for torsionally active ropes.

The weight of a hoisting rope suspended in a deep shaft adds to the tension associated with supporting the conveyance and tail rope, where appropriate. So with the conveyance at the lowest level, this creates a linear rise in tension from the conveyance to the sheave in the tower or head frame. The torque in the rope between the conveyance and the sheave must be constant. In a torsionally active rope this equilibrium torque is achieved by a more or less parabolic distribution of rotation along the length of the rope, with tightening of the lay at the bottom and increase at the top (Hermes and Bruens, 1957). In a Koepe system, as the conveyance rises the twisted head rope is transferred across the driving sheave to the other compartment. The twist distribution changes a little as the weight of the tail rope is progressively added beneath the conveyance, but remains essentially the same.

As the head rope crosses the driving sheave there is a change in tension associated with the difference in conveyance weight, but also a change in torque and corresponding equilibrium rotation distribution. So as the rope approaches the tangent point, in addition to the incremental relative axial slippage between rope and groove (essential to the tension changes in any friction drive (Chaplin, 1994)) rotational slippage will also occur. Due in effect to the torsional stiffness of the rope being low in relation to its axial stiffness, the actual displacements associated with rotation will be greater than the extensional changes. This dictates the direction of slip in the groove, which since it departs appreciably from a circumferential direction, will lessen the effective friction coefficient available for the friction drive, risking the possibility of full rope slip in extreme cases.

Furthermore, when the wind is completed the part of the head rope just below the sheave, in the compartment where the conveyance is at the bottom, will have been just above the conveyance in the other compartment at the start of the wind. Thus this part of the head rope will have transferred from a state of low tension and lay tightening, to high tension and lay lengthening. The range of this tension change, and more particularly the range of the torsional change, is critical from a fatigue point of view.

Triangular strand ropes, and other torsionally active constructions, can be used very successfully on Koepe winders, but only to a limited depth. This was demonstrated when the depth of shafts serving the gold mines of South Africa was extended to ever greater limits and problems were quite suddenly encountered as depth was increased from some 1100 m to 1400 m, and rope life fell by an order of magnitude (Berry and Wainwright, 1962). Rotational slippage of the rope in the groove was reported to be a significant factor, especially as regards wear of the friction lining. MacMillan (1960) describes the adverse experience using triangular strand ropes on the Koepe system at Stilfontein Gold Mine, not only identifying the reversal of the vertical twist profile, but also describing the resulting deformation as 'torsional fatigue'.

It was only with the development of well balanced rotation resistant rope designs that these problems were fully overcome (Fuller and Wainwright, 1967).

In light of recent studies of torsional fatigue it is worthwhile re-examining the circumstances of these problems. Using the model for rope torsion published by Feyrer and Schiffner (1987) with constants derived from test data for a triangular strand rope manufactured by Haggie Steel Wire Rope, the twist distribution as a function of shaft depth has been calculated. The results, with the same attached mass in each case and considering the extreme positions only, are presented in Figure 6, from which it is clear that the twist range experienced by the rope is very high†. The actual magnitudes calculated for all hoist depths are well beyond any threshold level apparent from the data presented in Figure 4.

† In describing problems with triangular strand head ropes for the 1350 m shaft at Stilfontein, MacMillan (1960) mentions that 'the ropes twisted through 40° per ft'. Assuming 1 ¾ inch ropes this translates to 580°/100d, which is a lot less than calculated here. However, the total spin reported by MacMillan is 130 turns, which accords well with a prediction of 126 turns. But the circumstances under which the observation was made, and whether the observation is directly comparable with the values calculated is not clear. The calculations also involve a number of assumptions, including that the twist profile established when the conveyance is at its lowest position is frozen as the hoist proceeds.

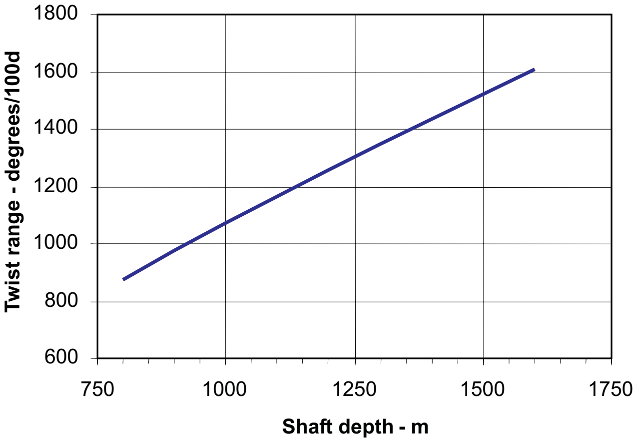


FIG 6 - Prediction of twist range in parts of Koepe head ropes close to attachment to the conveyance, but which pass over the driving sheave at the extremity of each wind. An empty 12 tonne conveyance and 44 mm rope are assumed throughout.

But those results are for ropes having metallic cores rather than the fibre core of the triangular strand rope, and, as already indicated above, a fibre cored rope is expected to have a superior performance in torsion fatigue, and thus the threshold for transition to torsion fatigue would be expected to be at a higher level.

So to achieve a reasonable performance from a Koepe system at moderate depths, a torque balanced, non-rotating rope is essential. To maintain strength and good load sharing between the strands in counter-laid layers it is essential to avoid the introduction of twist into such ropes. This requires that mines pay special attention to grooves and fleet angles, both all the more critical due to the large D/d ratios of sheaves.

The main benefit of the Koepe system is the low motor torque required. Provided the torsion problems can be overcome by rope design and good maintenance, then the main disadvantage with increasing depth is the higher load range experienced by the extremities of the head ropes.

The issue of fatigue in the tail ropes of Koepe systems is a little different, but some of the issues are the same. There is no attachment, so the minimum load is very low, even if tail sheaves are installed. A low minimum can exacerbate problems associated with ‘rope quality’ since at very low loads parts of some wires can be slack and the stress ranges experienced by other wires increased accordingly (Chaplin, 1995). Tail ropes can also be exposed to other exacerbating factors such as corrosion and the ingress of abrasive particles.

Both MacMillan (1960) and Fuller and Wainwright (1967) give detailed recommendations on installation and maintenance of ropes for friction hoists in the context of the evolution of solutions for the use of such systems for depths approaching 2000 m. As far as fatigue of ropes in such systems is concerned, provided torsion can be controlled to an acceptable level, the dominant factor will be tensile load range, and that will increase with depth, being effectively equal to the sum of the weights of the suspended head rope and the payload accentuated by accelerations.

Bending is inevitably an additional consideration, especially when the drive is not tower-mounted and two head-frame sheaves are included. However D/d ratios are generally so high that bending fatigue is not a major consideration, nevertheless the small bending stresses that result can be additive to fatigue stresses from other sources. An issue here is consistency of rope orientation as it goes over each sheave: a torsionally stiff rope (such as a full locked coil) tends to maintain the same axial orientation and so bending effects will be additive in the same locations; ropes that are torsionally more active will generally rotate so that orientation varies almost randomly and bending effects are distributed.

Bending in tail ropes is another consideration. This can occur in the form of free bending adjacent to terminations just below the conveyance. The rope is also bent in the sump either in free bending or bending over a sheave, if fitted. Especially if in a free loop, the effective bending diameter is almost inevitably less than over sheaves in the head frame, but tension is very low and for the free loop there is no transverse contact pressure, which greatly enhances life.

## Drum winders

In a drum winder there is, typically, no tail rope. In comparison to a Koepe winder of similar performance this leads to requirements for greater motor torque and power even when pairs of overlaid and underlaid drums are mounted on a common shaft, but especially when drives are electrically coupled to obtain fleet angles that promote good coiling. When the conveyance is at its lowest position the distribution of tension and twist along the rope is the same as described at the start of the wind in a friction system, but the all-important changes as hoisting proceeds follow a very different pattern.

As hoisting commences tensions will temporarily increase in line with accelerations, and second order dynamics. Such fluctuations contribute to fatigue range, and in so far as acceleration changes the gradient of tension increase up the rope, also have an influence upon twist distribution. But this effect, even in combination with loading, is small in relation to the static twist distribution (Rebel, 1997; Rebel, Chaplin and Borrello, 2000). In essence, the state of tension and rotation is locked into the rope as it winds onto the drum; the only change is associated with an element of rope relaxation resulting from radial deformation of the drum barrel, combined with a ‘Poisson’s ratio’ effect of compressive load of outer wraps imposed on inner layers of rope.

Considering the rope response for a rock hoist, the hoisting will typically be for a fully laden skip, and rope tensions therefore maximum. With the conveyance at maximum height, once the skip has discharged its payload, rope tension between the conveyance and the drum falls, but apart from the outermost few metres, the tension in the rope on the drum will be unchanged. The tension adjacent to the tangent point will rise exponentially from the low level in the catenary to the ‘stored’ level on the drum (Chaplin, 1994). This relaxation of the rope is accompanied by elastic recovery and relative motion of the rope around the drum, which has been termed ‘back-slip’ and is considered to be the principal contributor to wear of outer wires. This wear, or ‘plastic’ wear since it involves plastic deformation rather than abrasive removal of material, is worse where contact pressure is higher, especially at the turn crossovers, where the ‘hump’ of rope crossing from one valley between wraps to the next reduces the effective drum radius.

So apart from some accentuation due to dynamics and small additions from bending, the load range for each part of the rope is dominated by the difference in attached weight (the payload) between hoisting and lowering. The typical magnitude of load ranges for drum winders for deep shafts is therefore less than for equivalent Koepe hoists, and unlikely to exceed 12 per cent of rope breaking load (Wainwright, 1973; van Zyl, 2000). A ‘capacity factor’ limiting attached mass is clearly effective in limiting load range. This level of tension load range is just below the level identified as a fatigue limit for wire ropes of a range of constructions (Chaplin and Potts, 1991) although it may not eliminate fatigue failures of individual wires entirely (van Zyl, 2000).

One of the characteristics of rope degradation on drum winders, already mentioned above, is the flattening, or plastic wear, of wires along the exposed crowns of strands. This is especially noticeable in triangular strand ropes. This plastic deformation disrupts the structure of the material and creates potential locations for fatigue crack initiation, as reported by Chaplin (1994) and as shown in Figure 7. However, van Zyl (2000) reports fatigue tests on used drum-winder hoist rope in which only two per cent of observed wire breaks were in such locations: the majority were at the inter-strand valleys where adjacent strands contacted each other. Indeed the latter is the most common location for outer-strand wire breaks in tension-tension fatigue of wire ropes (Chaplin and Potts, 1991) but it does require that strands are in contact. This condition will be a function of the details of the rope construction and particularly its state of torsional deformation, as well as progressive collapse of the core during fatigue. This makes generalisation on this point impossible, and it must be recognised that fatigue fractures of individual wires may take place in either location as well as within strands where secondary bending, induced by the compound strand construction, may occur.

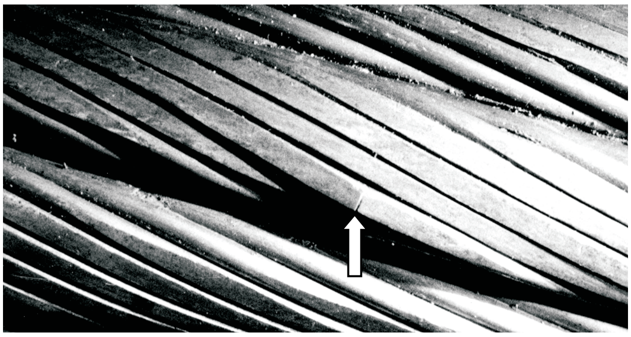

FIG 7 - Service degradation of plastic wear on a triangular strand, drum-hoist rope, with fatigue fracture (arrowed).

It is worth recording that fatigue crack initiation from induced plastic wear on outer wires has been demonstrated in the laboratory, with significant loss of tension-tension fatigue endurance in relation to rope without degradation (Chaplin, Ridge and Zheng, 1999b).

A further specialised form of fatigue found in ropes operating on drum winders is 'splitting fatigue' (Figure 8). These longitudinal internal cracks can develop when there is a significant level of plastic wear, sometimes with a 'Y' pattern in section leading to peeling of the deformed surface. Wainwright (1973) attributes this mechanism to the repeated application of high surface pressure between ropes in different layers on the drum. It is obviously worst at crossovers where the effective drum radius is reduced and contact pressure therefore very much higher. Absolute magnitude (as opposed to dynamic range) of rope tension and effective D/d ratio are the major factors that determine inter layer contact forces. Speed may also contribute according to Wainwright, but the wire heat treatment and drawing sequence are also important in determining the residual stress distributions that contribute to this process. Another parameter that affects the contact force between ropes is the pitch of the grooves on the drum in relation to actual rope diameter: as the spacing increases the direction of the contact forces between layers flattens and the magnitude of the resolved components increases (Chaplin, 1994).

It is further worth noting that compression of a cylinder across a diameter generates an internal transverse splitting tension, and failure along the boundaries of slip line fields can also generate the classic 'Y' pattern (Johnson and Mellor, 1962).

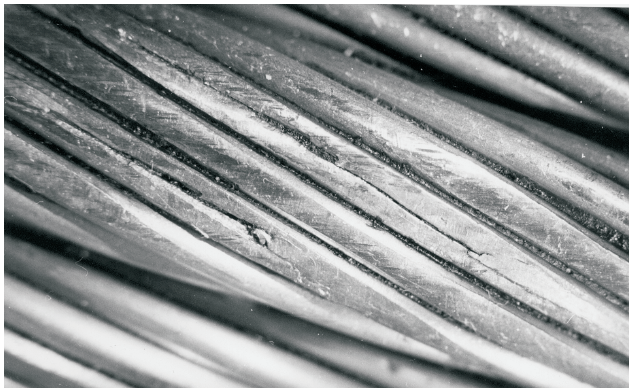

FIG 8 - Service degradation of splitting fatigue of two outer wires in a discarded triangular strand, drum-hoist rope.

As with plastic wear itself, split wires are most likely to form at the areas of highest pressure between rope in different layers, so the detail of the crossover geometry, imposed by the grooves on the drum barrel, is important. However, the localisation of damage in this way is at least a factor that can be mitigated by maintenance procedures to move the zones of greatest damage along the rope.

The processes of external wear are an important aspect of rope life. Attention above has focused upon the drum, where, with rope bearing upon rope, contact pressures are inevitably very high. The rope also contacts the sheave, and over time the groove will wear. If lateral rope oscillations are high and fleet angles adverse, wear can be rapid and asymmetrical. This introduces a further source of rope wear and possible distortion. The former accelerates fatigue but the latter especially can imbalance load sharing between components of the rope construction, further impairing fatigue performance.

## DISCUSSION AND CONCLUDING COMMENTS

Mine hoisting ropes work at high stresses in a demanding environment. They are subjected to a range of load patterns depending on the details of the hoisting mechanism and the shaft duty cycle. Degradation is inevitable in the long run, but assurance of safety is vital.

An understanding of the basic fatigue processes applicable to rope, how they can interact, and how they are affected by system parameters and operation, are important at the design stage in optimising potential life for ropes.

This is relevant to considerations of the effect of increasing depth on the difference in load range when choosing between drum and Koepe winders, where in the former case the range tends to fall, whilst in the latter case it increases.

Torsion fatigue is a problem that has been evident for some years, but it is only recently that our knowledge of the mode and analysis of the deformation began to allow some quantitative evaluation to be made.

An understanding of the degradation processes, and especially how the rates of fatigue deterioration are influenced by damage of one kind or another, is important in realising the potential rope life. This understanding is important in the context of inspection and discard.

The reliability of inspection can be enhanced through knowledge of how the details and operation of a hoisting system affect the types of fatigue and the characteristics and location of associated wire breakage. Quantitative evaluation of degradation in terms of wire breaks, wear, corrosion, etc can help to evaluate the future rates of deterioration and the remaining safe life. In this context, the definition of a discard criterion in terms of a perceived percentage loss of rope strength (be that seven per cent, ten per cent or 15 per cent) is not an acceptance that such a strength loss is in itself dangerous, but a tool for the recognition of a state and rate of degradation that raises concern as to the continued integrity of the rope until the time of the next inspection.

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