# Radial Pressure Damage Analysis of Wire Ropes Operating on Multi-layer Drum Winders

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**ABSTRACT:** The primary degradation mechanism of wire ropes operating on parallel grooved multi-layer drum winders is normally external wear and plastic deformation at the half turn and layer cross-overs towards the drum end of the ropes. This paper discusses the influence that (i) nominal radial crushing pressure on the drum and (ii) cyclic changes in rope load have on the rate of rope deterioration. Further factors such as actual rope contact areas and geometry are taken into account in the analysis and recommendations are made regarding the optimum rope designs for multilayer drum winders at different shaft depths. Specific examples of actual winder rope operating experience are given to validate the theoretical analysis.

## INTRODUCTION AND LITERATURE REVIEW

In mine hoisting applications, the most common winding systems are either of the friction / Koepe type or multi-layer drum installations. This paper is concerned with the deterioration of steel wire ropes on multi-layer drums. Of particular interest are the effects of radial crushing pressure and cyclic changes in rope load on rope damage accumulation at the parallel grooved half turn and layer cross-over areas.

The topic of drum winder rope deterioration has been well documented in the past including key effects such as tension, bending and torsion fatigue as well as plastic wear associated with backslip of the rope on the drum due to changes in skip loading during the winding cycle (Chaplin 1993; Chaplin 2005). These damage effects are not limited to wire ropes used in mining and are also found on multi-layer cranes designed for various industrial lifting applications (Verreet, 2003).

In South Africa, the Safety in Mines Research Advisory Committee (SIMRAC) funded a wide range of Gold and Platinum (GAP) mining rope related research programs over the past 20 years. The GAP 501 report on deterioration mechanisms of drum winder ropes (van Zyl 2000) discussed the effect of drum and headsheave sizes, number of rope layers, and the maximum dynamic rope load range on triangular strand rope deterioration. Of particular interest to this paper is the analysis given of the radial contact loads on drum winder ropes and the resulting contact stresses. The approach took into account varying rope tensile loading due to mine shaft depth and rope weight as well as the different spooling geometries between the parallel rope areas and the half turn cross-overs on the drum.

Rope radial and axial loads on multi-layer drums can be determined more accurately where the rope radial stiffness and drum construction details are taken into account (Dietz et al. 2009). This has the result that deformations of the drum and changes in rope geometry during spooling (diameter and length) can decrease the overall radial pressure experienced by the rope sections already on the drum. In general the method used by van Zyl is more conservative and results in higher nominal crushing loads on the drum as it ignores these dimensional changes.

Other investigations have considered the exact effects of rope maintenance practices, drum groove dimensions, groove pitching and filler positions and sizes on rope coiling behavior and damage in service for deep mining double drum and Blair multi-rope (BMR) hoists (Mostert and Musgrove 2007; Martin and Hein 2007).

Det Norske Veritas (DNV 2008) give maximum factors for radial pressure on a drum arising from multiple rope layers. DNV assumes that there is 1.75 times the pressure as a result of a second rope layer, that it will be 3 times the single layer pressure for 5 rope layers and that this will not increase for more than 5 layers. Recent offshore accidents involving drums with more than 5 layers proved that the last assumption may be incorrect. Therefore a more accurate method for estimating radial pressures in multi-layer rope systems is important.

In this paper an approach is presented for calculation of the nominal radial crushing pressure on the drum (and hence bottom rope layer) that is based on the more conservative van Zyl approach (van Zyl 2000). The results for 18 operational winding systems (with the parameters in Table 1) are presented taking into account the influence of cyclic rope load combined with radial crushing which show the correlation between the relative magnitude of these parameters and observed rope performance in service. The paper also considers the effect of increased rope surface area on rope damage through the reduction of radial crushing pressure. Recommendations are then made on optimum rope designs for multilayer drum winders.

**Table 1. Basic parameters for 18 operational drum winding systems, all ropes have 1770 MPa wire tensile grade, *T***

| **Winding system number** | **Rope construction** | **Rope diameter, *d* [m]** | ***D:d* ratio** | **Rope layers on drum, *n*** | **Minimum rope factor of safety** | **Payload per rope, *M* [% of Rope *MBL*]** |
|---|---|---|---|---|---|---|
| **1** | TURBOPLAST | 0.045 | 95 | 4 | 4.81 | 5.28 |
| **2** | TURBOPLAST | 0.056 | 97 | 4 | 4.91 | 6.31 |
| **3** | TURBOPLAST | 0.043 | 105 | 3 | 5.00 | 7.87 |
| **4** | TURBOPLAST | 0.034 | 26 | 2 | 5.00 | 0.00 |
| **5** | TURBOPLAST | 0.034 | 26 | 2 | 4.00 | 0.00 |
| **6** | TURBOPLAST | 0.051 | 108 | 4 | 4.67 | 5.71 |
| **7** | TURBOPLAST | 0.048 | 102 | 4 | 4.61 | 6.45 |
| **8** | TURBOPLAST | 0.051 | 108 | 4 | 5.00 | 0.00 |
| **9** | TURBOPLAST | 0.041 | 87 | 3 | 7.57 | 4.42 |
| **10** | TURBOPLAST | 0.054 | 90 | 2 | 6.10 | 7.64 |
| **11** | TURBOPLAST | 0.054 | 113 | 2 | 6.13 | 5.64 |
| **12** | TURBOPLAST | 0.051 | 96 | 2 | 6.00 | 7.61 |
| **13** | TURBOPLAST | 0.051 | 96 | 2 | 5.42 | 7.61 |
| **14** | TURBOPLAST | 0.026 | 79 | 5 | 13.66 | 4.28 |
| **15** | ULTRAFIT | 0.026 | 58 | 6 | 9.36 | 5.98 |
| **16** | ULTRAFIT | 0.037 | 81 | 3 | 5.25 | 8.18 |
| **17** | TURBOPLAST | 0.022 | 45 | 4 | 12.80 | 2.58 |
| **18** | ULTRAFIT | 0.026 | 35 | 5 | 10.89 | 3.76 |

## DRUM WINDER CONFIGURATIONS

Figures 1 and 2 show typical double drum and BMR winder layouts for mine hoisting applications. These types of winders are used to hoist both rock and men and materials depending on the mine shaft configuration and purpose. The conveyances normally run in fixed steel shaft guides but rope guided systems also exist. Shaft depths range from a few hundred meters to up to 3150 m in a single lift (Borello et al. 2005; Louw 2007). In deeper shafts with greater lifting requirements, as with rock winders, BMR machines are often preferred to keep the rope and hence drum diameters within sensible limits. The recommended drum to rope diameter ratio is 100:1.

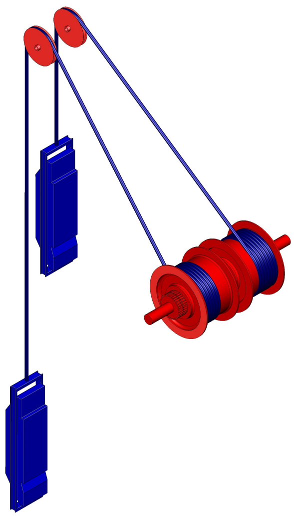


**Figure 1. Double drum winder configuration**

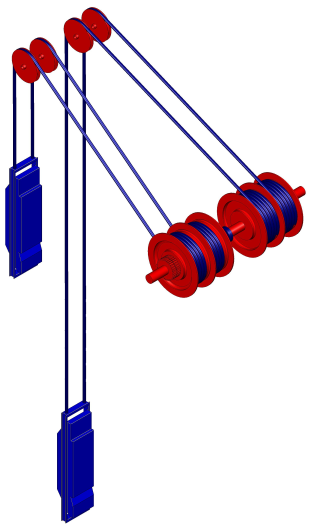


**Figure 2. Blair multi-rope (BMR) winder configuration**

Normally, ropes with either six triangular strands or eight round strands and an internal plastic layer are installed on drum winders as shown in Figure 3. The usual tensile grade of the rope wires is in the range 1770 MPa to 2160 MPa with either fiber cores or independent wire rope cores (IWRCs) (Wainwright 1994, Wainwright 1995). Rope diameters are mostly in the range of 25 mm to 55 mm (Rebel 1997).

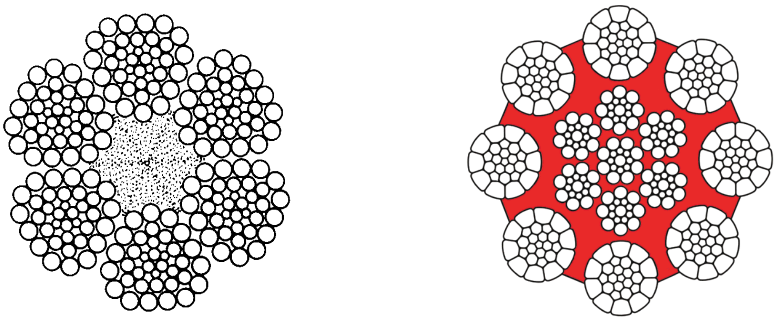


**Figure 3. Typical rope constructions used on drum winders**

## CALCULATION OF NOMINAL RADIAL CRUSHING PRESSURE ON THE DRUM

Using the simple radial load summation approach (van Zyl 2000) and taking the worst case that all layers of rope on the drum are subject to the same axial load equal to the maximum rope load in the system, the Equations (1b) and (3) result for the radial contact load per unit length and contact pressure seen by the bottom rope layer on the drum at the half turn cross-over regions (as per Figure 4).

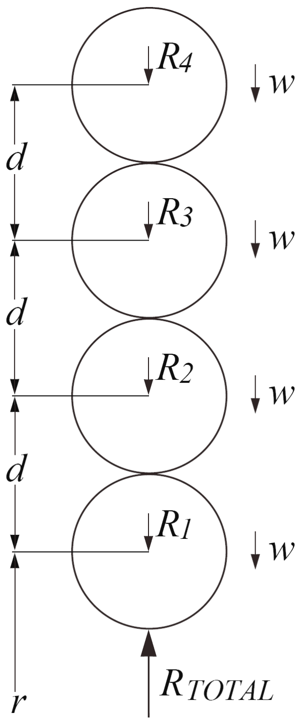



**Figure 4. General configuration of rope cross-sections on a drum at the half turn cross-over regions**

The total radial rope load per unit length is :

$$R_{TOTAL} = R_1 + R_2 + R_3 + R_4 + 4 * w \qquad (1)$$

where :

$w$ = rope weight per unit length [kN/m]

It can be shown (van Zyl 2000) that for small rope contact angles on the drum :

$$R_n = F_n / r_n \qquad (2)$$

where :

$R_n$ = radial rope load per unit length [kN]

$F_n$ = axial rope load [kN]

$r_n$ = drum radius for the given rope layer [m]

$n$ = rope layer number on the drum

Therefore and assuming that $F_n$ is constant for all rope layers, i.e. simply maximum rope load $F$ [kN], Equation (1) can be rewritten as :

$$R_{TOTAL} = F/(r+0 * d) + F/(r+1 * d) + F/(r+2 * d) + F/(r+3 * d) + \ldots + F/(r+(n-1) * d) + n * w \qquad (1a)$$

where :

$d$ = rope diameter [m]

Rearranging for $n$ rope layers on the drum gives :

$$R_{TOTAL} = F * \left( \sum_{i=1}^{n} \left[ 1 / (r + (i-1) * d) \right] \right) + n * w \qquad (1b)$$

From the total radial rope load per unit length, the total nominal pressure $P_{TOTAL}$ [Pa] seen by the bottom rope layer (and the drum) can be determined using Equation (3) assuming that the radial rope load is applied equally over the full projected contact area (as when calculating sheave nominal tread pressures) :

$$P_{TOTAL} = ( R_{TOTAL} * 1000 ) / ( 1 * d ) \qquad (3)$$

Dividing Equation (3) by 1000000 gives the result in MPa.

Figure 5 shows the results of Equation (3) applied to the 18 operational winding systems, all of the drum type with varying factors of safety, rope diameters, rope to drum diameter ratios and number of rope layers on the drums (see Table 1).

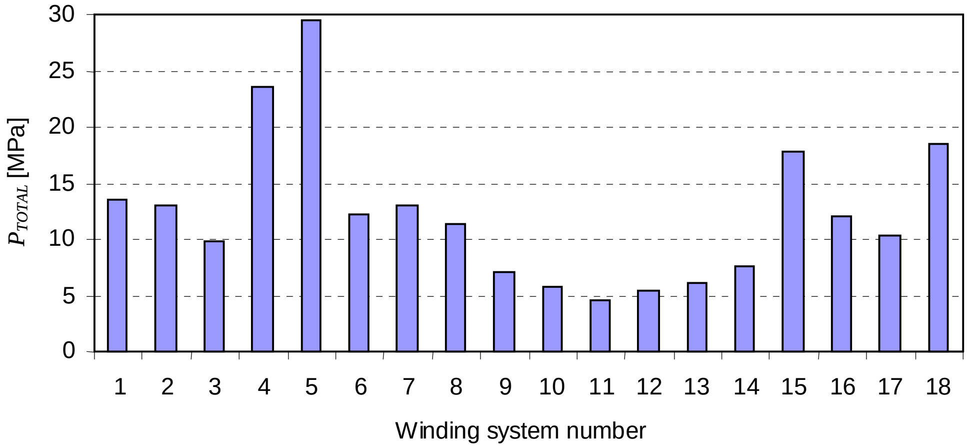



**Figure 5. Total nominal pressure on the drum for 18 operational winding systems based on Equation (3)**

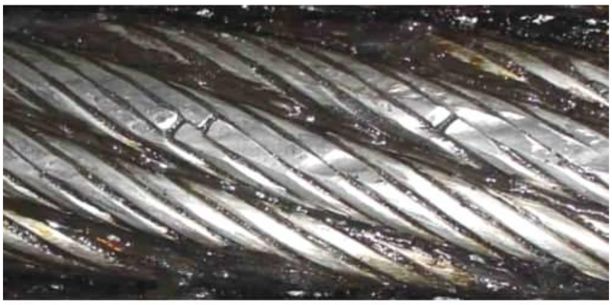
What is immediately apparent from the results in Figure 5 is the variation in magnitude and that all the values are significantly higher than the nominal tread pressure of 3.5 MPa that is recommended for headgear mounted sheaves for drum winder systems (Wainwright 1995). In general the wide variation in radial pressure on the drums leads to very different rates of deterioration of the ropes and it is therefore useful to consider radial pressure issues before selecting a rope for a particular drum winder installation. Different rope constructions will have varying resistance to high radial pressure loading at the drum cross-over points.

## INFLUENCE OF CYCLIC ROPE LOAD COMBINED WITH RADIAL CRUSHING

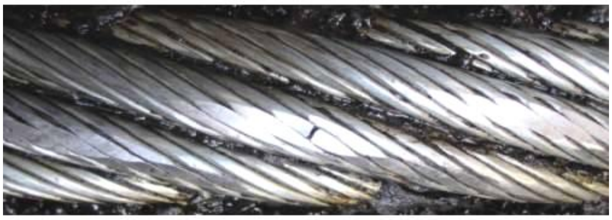
Previous studies have shown that it is the combination of radial pressure on the drum and relative axial movement of the rope cross-sections that leads to the plastic wear of the outer wires and subsequent wire failures at the half turn and layer crossover areas on the drum (Chaplin 1993). The concept of backslip is particularly relevant to drum rock winders where the ropes are always wound on to the drums under high tension and unwound under a lower tension i.e. hoisting a full skip up the shaft and then lowering an empty skip down the shaft. This results in the ropes slipping back on themselves as they leave the drum under a lower tension than initially wound on.

At the half turn crossover areas, the rope cross-sections are directly above one another as shown in Figure 4 (see also Martin and Hein 2007). This represents the worst possible contact conditions between adjacent rope layers / cross-sections. The dead turns on the drum are in a fixed rotational position and are therefore particularly prone to damage at the half turn crossovers, even more so if they have not been properly tensioned by doubling down with full skips or cages. Figure 6 shows examples of typical rope damage of the dead turns from winders 1, 2 and 7 (from Table 1 and Figure 5). In contrast, the live turns which exert this damage do not suffer in the same way due to the torsional response of the ropes in deep mine shafts (Rebel 1997). This torsional behavior leads to continuous rotation of the live cross-sections and an equal distribution of plastic wear around the live rope circumferences. The dead turns do not rotate so they are always impacted and worn in the same position. It is estimated that the damage to the dead turns is 24 times more concentrated than on the live rope sections (i.e. 15 degree versus 360 degree distribution of the damage).

Winding system No. 1



Winding system No. 2



Winding system No. 7

**Figure 6. Typical dead turn damage at the half turn crossovers on deep shaft double drum rock winders**

Figure 7 gives the change in winder conveyance payload per hoisting rope, *M*, from the empty to fully loaded condition as a percentage of the individual rope minimum breaking loads (*MBL*). This is also the difference in load between winding of the rope on to the drum (under high load) and unwinding (under lower load).

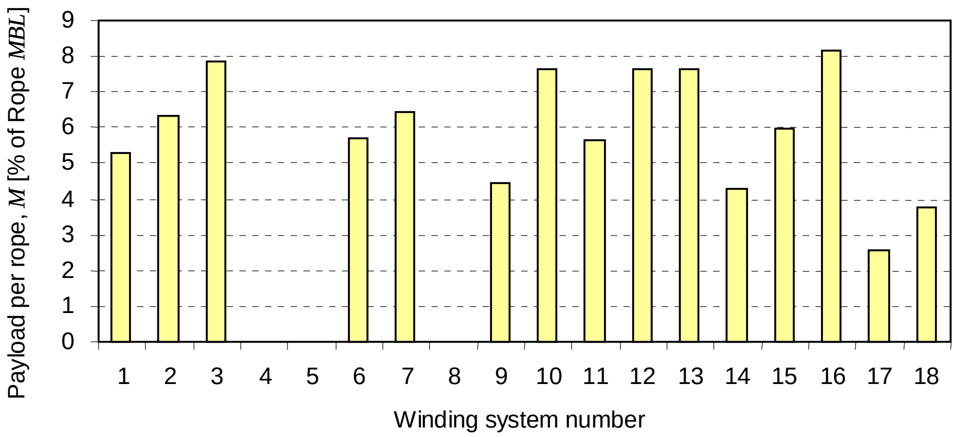



**Figure 7. Payload as a percentage of rope minimum breaking load for 18 operational winding systems as per Table 1**

It can be seen from the data in Figure 7 that some of the machines have zero payload. Numbers 4 and 5 represent a double-drum test stand facility specially designed for developing ropes for multi-layer applications but without the capability for variation in rope line pull during cycling. Winding system number 8 is a counterweight drum on a deep shaft double drum man winder. Here the counterweight load is constant at all times hence zero payload for that rope. It has been observed that the rope on the counterweight drum (No. 8) does not suffer from any of the dead turn damage shown in Figure 6 even though it has a nominal radial drum pressure of a similar magnitude to winders 1, 2 and 7, see Figure 5. The rope constructions on all four winders are the same.

Having established that the in-service damage to ropes is dependant on both the nominal radial pressure on the drum and the simultaneous changes in rope loading that lead to backslip, it is possible to define a nominal damage factor, $K_{NOMINAL}$, for the half turn and layer crossover regions which is the product of the data in Figures 5 and 7, i.e. Equation (4), as shown in Figure 8. The nominal radial pressure, $P_{TOTAL}$, is divided by the rope wire tensile grade to remove the units. The compassion of the relative magnitudes of the drum damage factors in Figure 8 is more important than the absolute values.

$$K_{NOMINAL} = P_{TOTAL} / T \; * 100 * M \qquad (4)$$

where :

$T$ = tensile grade of rope wires [MPa] = 1770 MPa for the all ropes in this paper

Note that $K_{NOMINAL}$ is simply the product of two percentage values so for example for winding system number 1 (see Table 1 and Figures 5 and 7) :

$$K_{NOMINAL} = ( 13.48 / 1770 * 100 ) * 5.28 = 4.02$$

It can also be shown (see APPENDIX) that if the rope self weight, $w$, is ignored and the spooling on the drum is assumed to take place at the average diameter of all the layers then the nominal damage factor, $K_{NOMINAL}$, can be approximated as follows :

$$K_{NOMINAL} = \left[ \frac{20 * n * Payload}{FoS * \left[ (D:d) + n - 1 \right] * d^2 * T} \right] \qquad \text{(from A4a)}$$

where :

*Payload* = conveyance payload [kN]

*FoS* = minimum rope factor of safety for the winding system, see Table 1

*D:d* = drum to rope diameter ratio for the bottom rope layer on the drum

This gives a method of calculating the damage factor directly from the basic winding system parameters and the results are within 1% of the values shown in Figure 8.

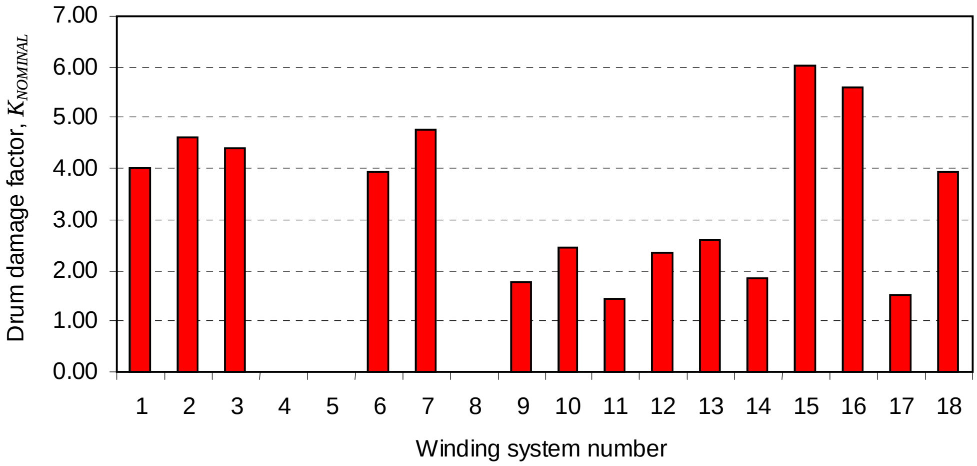



**Figure 8. Damage factor (product of normalized nominal radial pressure on the drum and percentage payload) for 18 operational winding systems as per Equation (4)**

On all winders 1, 2, 3, 6 and 7 the damage to the dead turns has been consistent and as per the examples in Figure 6. The solution to this problem has been to limit the cycles between backend cuts or drum crops to 10000 cycles so as not to break the rope wires. The experience with winders 9 to 14 and 17 has been quite different in that the same rope constructions have shown far less damage on the drum. Two cases stand out and are dealt with in detail elsewhere in these proceedings (Smith and Verreet 2005; Willemse and Schmitz, 2010). For winding system No. 10 it has been possible to achieve 30000 cycles between backends with a rope life of circa 400000 cycles and for winding system No. 11, 50000 cycles between backends with a rope life of circa 550000 cycles. In contrast winder No. 6 required strict backend cuts every 10000 cycles and only achieved a rope life of 170000 cycles. This operating experience has shown a direct correlation between the magnitude of the damage factors and the rate of rope deterioration and required maintenance frequency.

Typical triangular strand rope life on double drum rock winders is 100000 cycles with backend cuts recommended every 10000 cycles (van Zyl 2000).

For winders 15, 16 and 18 an alternative rope design, which is more tolerant to drum crushing and backslip, has been used and no notable damage is apparent on any of the dead turn crossover regions.

The CASAR double drum test stand facility (system numbers 4 and 5) had a very high nominal radial pressure (Figure 5) yet the damage seen in Figure 6 was not apparent. This is explained by the fact that the ropes were always spooled on and off the test drums under constant tension.

Rope damage has also occurred in multi-layer mobile cranes with varying payloads during a lift. An extreme example is the case where a mobile crane was lifting a series of water weights out of water during the crane testing procedures. This resulted in the rope loads increasing while spooling on to the drums. In one day the ropes were destroyed on the crane drums yet the same machine lifting constant loads had acceptable rope life. The relative motion of the ropes on the drums with combined high radial crushing pressures caused the severe damage.

Another practical example is in ropes for crane boom hoists. The experience here is that in boom hoist reeving systems where the rope load is not affected by the crane payload, rope lives are very good. However, where the boom hoist rope load fluctuates with crane payload, rope life can be reduced dramatically because of the resulting slip conditions with spooling on the boom hoist drum.

The conclusion of these observations on both mining winders and multi-layer crane applications is that damage to ropes on drums is significantly more likely where the ropes spool on an off the drums under differing tensions with the resulting axial rope movement combined with high radial crushing pressures.

## EFFECT OF INCREASED ROPE SURFACE AREA ON ROPE DAMAGE

Thus far, the total nominal pressure $P_{TOTAL}$ [Pa] seen by the bottom rope layer (and the drum) was determined using the projected area of 1 meter length multiplied by the rope diameter. This approach does not take into account the different actual contact conditions that would apply for different rope constructions. A detailed study on the surface condition and fatigue of wire ropes showed graphically that the strand contact areas vary significantly for different rope constructions and also between new and worn ropes of the same construction (Nishioka 1966).

Using principles from solid mechanics of elastic cylinders in contact and the radial contact load per unit length, like $R_{TOTAL}$ from Equation (1b), it is possible to calculate more accurate contact stresses between rope cross-sections on a multi-layer drum at the half turn cross-overs.

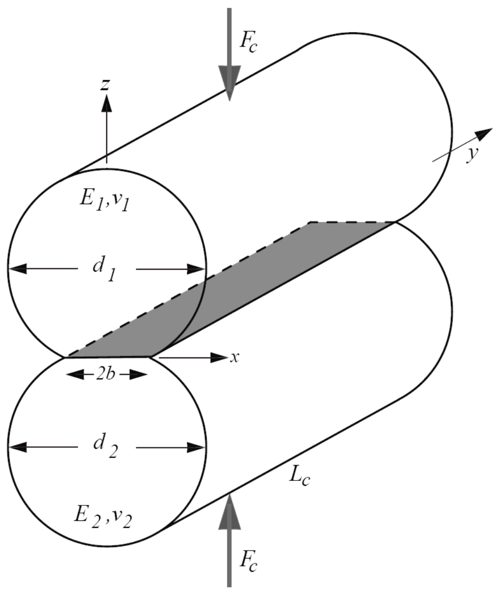



**Figure 9. General arrangement of elastic cylinders in contact**

For cylinders in contact as shown in Figure 9, the rectangular area of contact is ***2*b*L*** (Shigley 1986) where ***b*** is the half width of the contact area:

$$b = \sqrt{\frac{2 * F_C * d_C * (1 - v^2)}{p * L_C * E}} \tag{5}$$

and :

$v = v_1 = v_2 = 0.3$ = Poisson’s ratio

$E = E_1 = E_2 = 150$ GPa = assumed modulus of elasticity (i.e. less than 207 GPa for steel)

$d_C = d_1 = d_2$ = rope contacting cylinder diameter, dependant on rope construction [m]

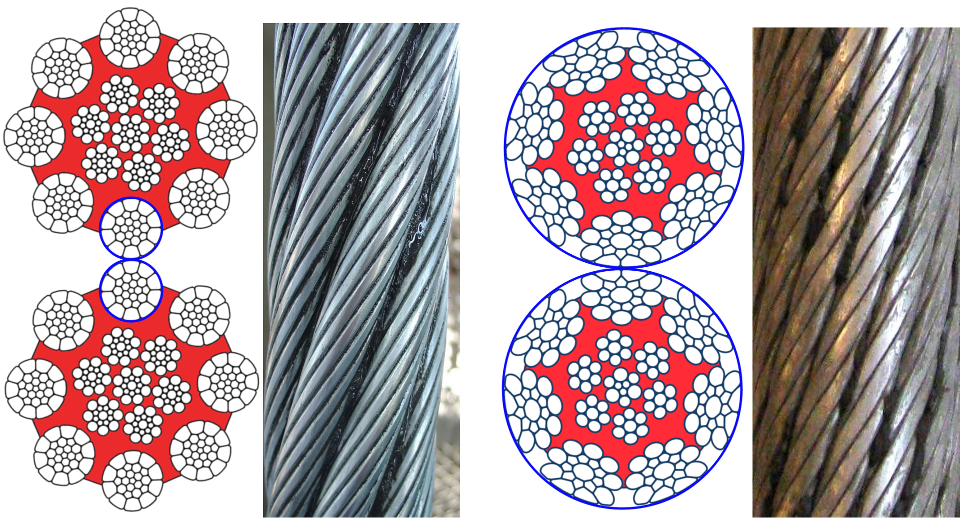
$L_C$ = contacting cylinder length [m]

$F_C$ = compressive load applied to the contacting cylinders [N]

The maximum contact stress is defined as (Shigley 1986) :

$$P_{MAX} = 2 * F_C / (\pi * b * L_C) \tag{6}$$

It is now necessary to consider what the contacting cylinder diameters, $d_C$, would be for the two rope constructions used on the 18 winding systems from Figures 5, 7 and 8. Figure 10 shows the arrangement of both constructions as they would appear in the half turn cross-over areas on the drums.



Winders 1-14 and 17 Winders 15, 16 and 18

**Figure 10. Rope constructions used on the 18 winding systems and their arrangement at the half turn crossover regions on the drums**

Winders 1 to 14 and 17 use the standard eight strand CASAR TURBOPLAST construction shown earlier in Figure 3 where winders 15, 16 and 18 use a special seven strand design with flattened outer stands, ULTRAFIT. It is apparent from the circles drawn on the cross-sections in Figure 10 that the effective contacting cylinder diameter, $d_C$, for TURBOPLAST is the strand diameter when viewed over a short rope length of say one rope diameter at the half turn cross-over point. In the case of the special seven strand design, the effective contacting cylinder diameter, $d_C$, is much larger and can be approximated by the rope diameter.

Assuming that the contacting cylinder length from Figure 9, $L_C$, is equal to the rope diameter, $d$, it is possible to calculated the maximum contact stress, $P_{MAX}$, at the interface between the first and second rope layer using Equations (5) and (6) where :

$$F_C = ( R_{TOTAL} - ( R_1 + w) ) * d = (R_{TOTAL} - ( F/r + w )) * d \tag{7}$$

where $R_{TOTAL}$ is from Equation (1b) and for winders 1 to 14 and 17 :

$d_C = 0.26 * d$ i.e. the outer stand diameter for Equation (5)

for winders 15, 16 and 18 :

$d_C = d$ i.e. the rope diameter for Equation (5)

Figure 11 shows the results of the maximum contact stress, $P_{MAX}$, at the interface between the first and second rope layer for the 18 systems. The yield stress for 1770 MPa wire is circa 1400 MPa shown by the horizontal line in Figure 11. It is clear that a number of the contact stresses are beyond the yield point of the wires and hence the typical damage pattern shown earlier in Figure 6. Note also that for winders 15, 16 and 18 the contact stresses are half of what they would have been due to the significant increase in contact area because of the flattened outer stands of those ropes. The effect is that the maximum contact stress reduces from well above the yield point to below it. This is confirmed by the observation in service that it is difficult to identify the cross-over regions on the operating ropes on winders 15, 16 and 18 as there are no indentations and wear points like in Figure 6.

Note also that there are two orders of magnitude difference between the nominal contact stresses on the drum from Figure 5 (based on projected areas) and the maximum contact stresses in Figure 11 (based on the actual rope to rope contact conditions at the half turn cross-overs).

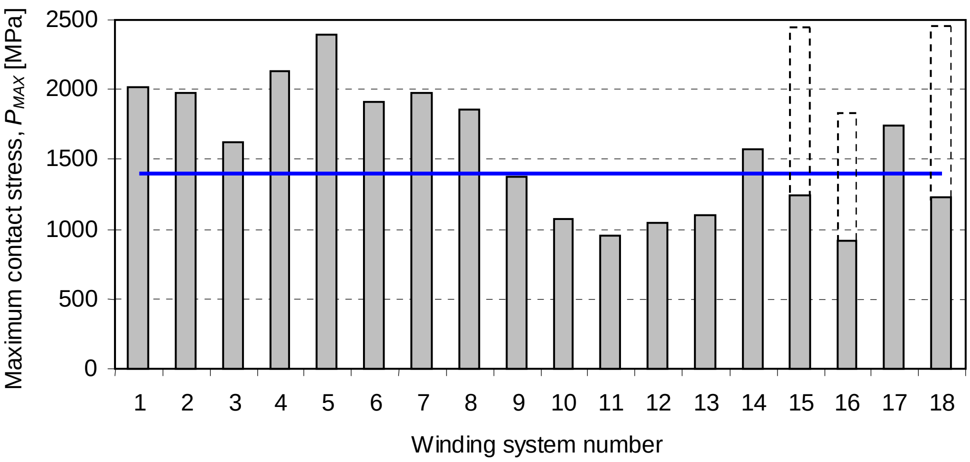



**Figure 11. Maximum contact stress between the first and second rope layer for 18 winding systems at the half turn crossover regions on the drums as per Equation (6), horizontal line is approximate yield stress for 1770 MPa wire**

As in Figure 8, the maximum contact stress in Figure 11 can also be normalized and multiplied by the percentage payload (Figure 7) to determine an arbitrary damage factor $K_{MAX}$, Equation (8) where again only the relative magnitudes are of real interest. Figure 12 shows such a calculation for the 18 winders. Here the maximum contact stress damage factors for winders 15, 16 and 18 are circa half of what they would have been for TURBOPLAST ropes with the smaller contact areas.

$$K_{MAX} = P_{MAX} \; / \; T * 100 * M \qquad (8)$$

where :

$T$ = tensile grade of rope wires [MPa] = 1770 MPa for the all ropes in this paper

Note that $K_{MAX}$ is also simply the product of two percentage values so for example for winding system number 1 (see Table 1 and Figures 7 and 11) :

$K_{MAX}$ = (2,010 / 1770 * 100 ) * 5.28 = 600

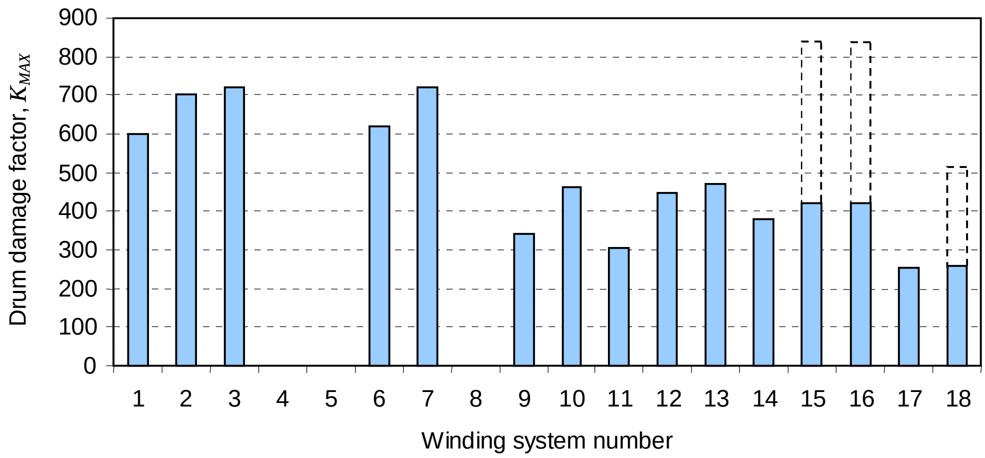



**Figure 12. Damage factor (product of normalized maximum contact stress and percentage payload) for 18 operational winding systems as per Equation (8)**

By changing the rope construction for winders 15, 16 and 18 to the special seven strand design with flattened outer stands, the deterioration effects on the drum due to radial crushing and backslip have been reduced by a factor of two compared with using the standard TURBOLAST ropes. Although the original Figure 8. damage factors for winders 15, 16 and 18 were very high, the analysis here has shown how the construction change can mitigate the effects and give the same exceptional rope performance with reduced maintenance as achieved with winders 10 and 11 (Smith and Verreet 2005; Willemse and Schmitz, 2010).

## RECOMMENDATIONS ON OPTIMUM ROPE DESIGNS FOR MULTILAYER DRUM WINDERS

In addition to the ULTRAFIT design, the TURBOFIT and STARFIT type constructions are shown in Figure 13. These ropes are also being used on high radial crushing load double drum rock winders where different torsional characteristics are required (i.e. shaft depth dependant). With both constructions a smooth outer rope surface (with larger bearing areas) is achieved which minimizes the negative effects of drum crushing and backslip. The ropes also have very high breaking loads for a given diameter due to their high fill factors.

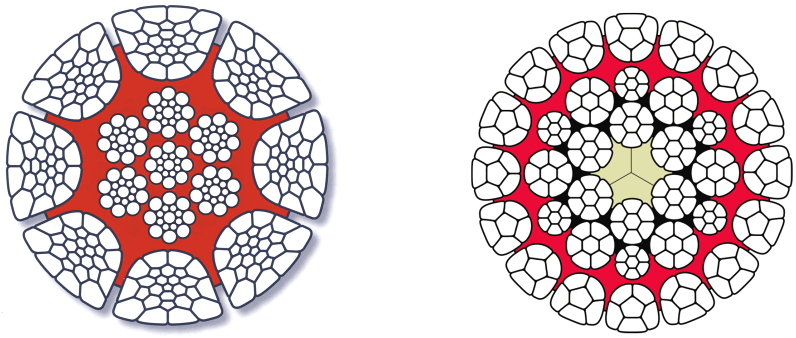


**Figure 13. CASAR TURBOFIT and STARFIT type constructions**

## CONCLUSIONS AND FURTHER INVESTIGATIONS

The analysis in this paper has shown that a nominal drum damage factor can be expressed in terms of basic winding system parameters and that there is a correlation between the damage factor and observed rope deterioration in service. The primary causes of rope damage at the parallel grooved drum cross-over points are radial pressure due to multi-layer spooling and rope payload changes leading to backslip during spooling. The damage factor analysis takes these causes into account.

Investigation of 18 operational winding systems showed that the nominal drum damage factor varies significantly between systems that have very good rope performance at the half turn cross-overs versus those where strand damage and wire breaks occur within a few months of rope operation. A factor of 3 to 4 difference in magnitude of the damage factor would exist in such comparisons. Rope maintenance frequencies are also dependant on the nominal damage factor, higher damage factor installations require more frequent drum end maintenance operations.

Further development of the drum damage factor concept showed that contact stresses between adjacent rope cross-sections can exceed the yield stress of the wire material. These calculations took in to account the actual contact conditions between ropes based on the analysis of elastic cylinders in transverse compression. By changing the rope construction from a round strand to a flattened strand design it was possible to halve the contact stresses and thereby half the maximum drum damage factor for high crushing load winders. The practical result has been that rope life and maintenance frequencies can be maintained even for the winders with nominally high damage factors by using the alternate flattened strand rope designs.

The authors are continuing to gather data from other operating winding systems to further calibrate the exact relationship between the nominal damage factor for each system and the observed rope life and required maintenance practices. This data will allow for the optimum rope design selection for every system and will also aid in the design of new winding systems due to a better understanding of the key factors that influence rope performance on multi-layer drums.

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## APPENDIX - ALTERNATE EQUATION FOR THE NOMINAL DAMAGE FACTOR

It is possible to simplify the calculation of the nominal damage factor given by Equation (4) as shown in Figure 8 by making changes to the calculation of the total radial rope load per unit length, $R_{TOTAL}$, Equation (1b). This allows for the damage factor to be determined knowing only a few winding system parameters without any summations required.

First the summation in Equation (1b) needs to be simplified. This can be done by assuming an average spooling diameter for all rope layers i.e. $r + (n\text{-}1) / 2 * d$, see Figure 4.

If $n * w$ is assumed to be zero (i.e. rope weight is negligible) and $r$ is rewritten as $(D{:}d) * d / 2$ then from Equation (1b) the following results for $R_{TOTAL}$ [kN] :

$$R_{TOTAL} = n * \left[\frac{2*F}{[(D:d)+n-1]*d}\right] = n * \left[\frac{2*MBL}{FoS*[(D:d)+n-1]*d}\right] \quad \text{(A1)}$$

where :

$D{:}d$ = drum to rope diameter ratio for the bottom rope layer on the drum

$MBL$ = minimum rope breaking load from the rope catalogue [kN]

$F$ = maximum axial rope load [kN]

$FoS$ = minimum rope factor of safety for the winding system

$n$ = number of rope layers on the drum

$d$ = rope diameter [m]

It follows that the total nominal radial pressure on the drum $P_{TOTAL}$ [MPa] from Equation (3) can be rewritten as :

$$P_{TOTAL} = \left[\frac{2*MBL*n}{1000*FoS*[(D:d)+n-1]*d^2}\right] \quad \text{(A3)}$$

Substituting Equation (A3) into Equation (4) gives a simplified form for the nominal damage factor $K_{NOMINAL}$ :

$$K_{NOMINAL} = \left[\frac{2*MBL*n*M}{10*FoS*[(D:d)+n-1]*d^2*T}\right] \quad \text{(A4)}$$

where :

$M$ = *Payload* [kN] / *MBL* [kN] * 100, as per Table 1

$T$ = tensile grade of rope wires [MPa] = 1770 MPa for the all ropes in this paper

Written alternatively in terms of conveyance ***Payload*** [kN]:

$$K_{NOMINAL} = \left[\frac{20 * n * Payload}{FoS * [(D:d) + n - 1] * d^2 * T}\right] \tag{A4a}$$

For winding system number 1, which has a ***Payload*** = 84.61 kN (see also Table 1 and Figure 7) :

$$K_{NOMINAL} = \left[\frac{20 * 4 * 84.61}{4.81 * [(95) + 4 - 1] * 0.045^2 * 1770}\right] = 4.01$$

This nominal damage factor of 4.01 compares accurately to 4.02 calculated using Equations (1b), (3) and (4).