



Calculating the service life of running steel wire ropes

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by Dipl.-Ing. Roland Verreet

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1. Introduction

In contrast to other parts of the machinery which can be laid out with unlimited fatigue strength running wire ropes always have a limited service life. Therefore they must be inspected and examined at regular intervals so that they are replaced well before failure.

Crane designers, however, would like to have a rough estimation of the service life of the ropes already in the early stages of conceiving the cranes, so that they can, if necessary, improve the reeving system. This is one of the reasons why for many years Casar have carried out calculations for their customers to predict the service life of wire ropes. This brochure is meant to offer in-depth information on the method of calculation and to demonstrate the potential and limitations of the forecasting procedure.

2. Calculating the number of sustainable bending cycles

Even the inventor of the wire rope, Oberbergrat Albert from Clausthal-Zellerfeld, carried out fatigue tests with wire ropes to compare the service lives of different rope designs. After him rope researchers like Benoit, Wörnle or Müller carried out a vast number of wire rope bending fatigue tests. They examined the effect of the essential factors of influence on the service life of the rope. Prof. Feyrer from the university of Stuttgart has summed up their findings in a formula which allows to predict the service life of wire ropes in reeving systems with sufficient accuracy. The Feyrer formula reads

$$\lg N = bo + (b1 + b4 \bullet \lg \frac{D}{d}) \bullet (\lg \frac{S do^2}{d^2 So} - 0.4 \bullet \lg \frac{Ro}{1770}) + b2 \bullet \lg \frac{D}{d} + b3 \bullet \lg \frac{d}{do} + \frac{1}{b5 + \lg \frac{1}{d}}$$

In this formula

- N indicates the number of bending cycles
- d the nominal rope diameter in mm
- D the diameter of the sheave in mm
- S the rope line pull in N
- 1 the length of the most heavily strained rope zone in mm and
- Ro the nominal tensile strength of the wire in N/mm^2

 $S_0 (= 1 \text{ N/mm}^2)$ and $d_0 (= 1 \text{ mm})$ are the unit line pull and the unit diameter which make the fractions dimensionless. The factors b0 to b5 are rope-specific parameters which must be determined separately in a great number of bending fatigue tests for every single rope design.

For many years Casar have carried out bending fatigue tests on their own-label test devices in order to determine those parameters for Casar Special Wire Ropes. Every single test increases the number of data corroborating the predictions of the service life of a rope.

2.1 The average number of bending cycles N

By means of statistical procedures it is possible to determine the factors b0 to b5 for different reliabilities of the predictions. For instance, the commonly quoted average number of bending cycles \tilde{N} is the number of bending cycles which -under the given circumstances- would be achieved in a great number of tests as the average value of all test results of a certain rope design.

Normally the average number of bending cycles is the value which the designer or operator of a crane is eager to know. He is interested in the number of bending cycles that he will achieve <u>on average</u>. However, he must bear in mind that <u>average</u> <u>value</u> also indicates that in a great number of tests one half of all the wire ropes will exceed that value whereas the other half will <u>not</u> reach it.

That means that a number of bending cycles defined as the <u>average value</u> of a great number of tests can under no circumstances be guaranteed for one single wire rope by the rope's or the crane's manufacturer: The term <u>average value</u> itself implies that half of all ropes do <u>not</u> achieve that value.

2.2. The number of bending cycles Ñ

There are situations in which it does not suffice to know that the wire rope will achieve the calculated number of bending cycles <u>on average</u>. It's rather a question of determining a number of bending cycles which will be achieved with a high probability by nearly all the ropes in operation. However, the statistical spread of the test results during bending fatigue tests indicates that it is virtually impossible to predict a number of bending cycles which will be achieved in any case. Therefore a number of bending cycles N10 is calculated which is achieved by 90 % of all the wire ropes tested at a probability of 95 %, and only 10 % of all the ropes tested do not

achieve that value. It is self-evident that the number of bending cycles N_{10} must always be smaller than the average number of bending cycles \tilde{N} .

3. The definition of a bending cycle

A bending cycle is defined as the change from the straight state of the rope into the bent state and back again into the straight state (symbol -) or as the change from the bent state into the straight state and back again into the bent state of the same direction (symbol \sim). Whenever a rope runs over a sheave the respective rope zone carries out a complete bending cycle (ie a change from the straight into the bent and back again into the straight state); whenever a rope runs onto a drum it carries out half a bending cycle (ie a change from the straight into the bent state).

4. The definition of a reverse bending cycle

A reverse bending cycle is defined as the change from the bent state into the straight state and again into the bent state, but of the opposite direction (symbol \searrow).

As to the definition of a reverse bending cycle not occurring in the same plane the experts' opinions differ widely. In Sheet 1 of DIN 15020, for instance, a change from the bent state into the straight state and into a bent state in a plane offset by 90° (Fig. 1a) is defined as a simple bending cycle, whereas a change from the bent state into the straight state and into a bent state in a plane offset by 120° (Fig. 1b) is defined as a reverse bending cycle.

Practice, however, shows that not only the <u>angle</u> between the bending planes decides if the damage of the rope is greater than in the case of a simple bending cycle but also the <u>distance</u> between the sheaves which have been arranged under such an angle. So, with short distances between the sheaves the damage to the wire rope is already considerably greater at an angle of about 90° than with a simple bending cycle, so that that case should be defined as a reverse bending cycle, whereas with great distances between the sheaves very often there is hardly any negative effect on the service life of the rope, even at angles of 120° and more, because the wire rope can rotate between the two sheaves round its axis for exactly that angle, so that finally it runs over both sheaves in the same bending direction.

To be on the safe side, contrary to the recommendation of DIN 15020, bending cycles with a change of the bending plane of 90° and more should always be counted as reverse bending cycles.



Fig. 1: Change of the bending plane by $90^{\circ}(a)$ *and* $120^{\circ}(b)$

5. Service life prediction

The author has written a computer program which calculates, based on Prof. Feyrer's formula, for a set of given parameters (rope design, nominal rope diameter, diameter of the sheaves, line pull, nominal wire tensile strength and length of the most heavily strained rope zone), the sustainable average number of bending cycles \tilde{N} until rope discard and rope break as well as the number of bending cycles N10 until rope discard and rope break which 90 % of all ropes achieve with a 95 % probability.

<u>Example</u>: For a given rope design with a nominal rope diameter of 30 mm, a sheave diameter of 600 mm, a line pull of 40,000 N, a tensile strength of 1770 N/mm^2 and a most heavily strained rope zone of 20,000 mm, the program calculates 400,000 bending cycles until rope discard and 900,000 bending cycles until rope break.

The program also offers graphical illustration of the results as a function of any of the factors of influence. Fig. 2 shows the illustration of the average number of



Fig. 2: Number of bending cycles as a function of the nominal rope diameter

bending cycles until rope discard (lower curve) and until rope break (upper curve) as a function of the nominal rope diameter.

Fig. 3 shows the number of bending cycles until rope discard (lower curve) and until rope break (upper curve) as a function of the sheave diameter. With increasing sheave diameter the numbers of bending cycles increase overproportionally.

In the present case the wire rope achieves an average number of bending cycles of 400,000 for a sheave diameter of 600 mm. Increasing the sheave diameter by only 25 % to 750 mm will already double the rope's service life. Fig. 4 shows the number of bending cycles until rope discard (lower curve) and until rope break (upper curve) as a function of the chosen line pull. The diagram clearly illustrates that with increasing line pull the numbers of bending cycles decrease overproportionally.



Fig. 3: Number of bending cycles as a function of the sheave diameter



Fig. 4: Number of bending cycles as a function of the line pull

6. The optimal nominal rope diameter

Whereas the numbers of sustainable bending cycles continuously increase with increasing sheave diameter (Fig. 3) and continuously decrease with increasing line pull (Fig. 4), the sustainable numbers of bending cycles at first increase with increasing nominal rope diameter, but after exceeding a maximum value they decrease again with the nominal rope diameter still increasing. The nominal rope diameter for which the numbers of bending cycles achieve their maximum is called "Optimal Nominal Rope Diameter".

In our example a wire rope with a nominal diameter of 10 mm reaches its discard state at only 50,000 bending cycles (Fig. 2). Admittedly, this rope with a sheave diameter of 600 mm operates under a very favourable D/d-ratio of 60, but it is obvious that the specific tensile stress under the influence of the chosen line pull of 40,000 N is by far too high for such a thin wire rope.

At first the service life increases with increasing nominal rope diameter. A rope of twice that diameter, ie 20 mm, reaches its discard state at about 340,000 bending cycles, which is nearly seven times that number. Of course, the D/d-ratio has been reduced to 30, but the load bearing cross section of the rope has quadrupled and can cope with the chosen line pull of 40,000 N much more easily than the rope with the nominal diameter of 10 mm.

If we double the nominal rope diameter again, we do, however, not gain further increase of the number of bending cycles: a rope with the nominal diameter of 40 mm and a number of bending cycles of 300,000 until rope discard does not achieve the same number as a rope with the nominal diameter of 20 mm. It does not fail because of the line pull - the safety factor is now 16 times as high as the one for the rope with a diameter of 10 mm. It fails because the sheave diameter of 600 mm is by far too small for the nominal rope diameter of 40 mm (D/d = 15).

Evidently the ropes shown in the left part of the graph fail because the specific line pull is too high - the bending stresses are small here. In the right part of the graph the ropes fail because the D/d-ratio is too small - the specific line pull is small here. Between the two parts the graph shows a maximum where the sum of the damaging influences of line pull and bending stresses is minimal. As mentioned above, the diameter for which the graph shows a maximum is called "Optimal Nominal Rope Diameter". In Fig. 2 the Optimal Nominal Rope Diameter can be found at about 27 mm. The average number of bending cycles achieved with this nominal rope diameter is 410,000.

7. The most economical rope diameter

A rope designer should not choose a nominal rope diameter greater than the optimal diameter. He would spend more money on a shorter service life. He should rather choose a nominal rope diameter slightly under the optimal one. As our example (Fig. 2) shows, a nominal diameter of 24 mm has nearly the same service life as a nominal diameter of 27 mm. The rope diameter, however, is more than 10 % smaller than the optimal one. This means that a much more moderately priced rope achieves nearly the same number of bending cycles. In addition, the width of the drum can be reduced considerably.

As a result, the most economical nominal rope diameter always lies slightly below the optimal rope diameter, ie at about 90 %.

8. The influence of the load collective

The number of bending cycles which can be achieved in a reeving system depends on many factors of influence. Under the same conditions a high-quality rope, for instance, will easily achieve three times the number of bending cycles of a simple rope design. Similarly, a well-lubricated and regularly relubricated wire rope will normally achieve a much higher number of bending cycles than an insufficiently lubricated rope of the same design. Another important factor of influence is, of course, the crane's mode of operation.

When the designer classifies the crane into a group of mechanism of the standard he already decides whether the rope of his crane will enjoy a long or only a very short service life, because it depends on that classification whether the reeving system will lift the same load with a thick or a thin wire rope and whether that rope will run over sheaves with a great or a small D/d-ratio. Ropes in the highest group of mechanism will achieve approximately 200 times the number of bending cycles compared to their counterparts from the lowest group of mechanism.

In the past it was suggested to state in the standard the expected number of lifting cycles of a rope as a function of the group of mechanism. However, ropes in reeving systems of the same group of mechanism do not necessarily achieve the same service life, not even if the reeving systems are identical. The reason is that reeving systems within the same group of mechanism can operate with very different load collectives.

Fig. 5 shows the groups of mechanism according to DIN 15020. It is evident that a reeving system with a line pull of, for instance, 100,000 N can be classified into

group of mechanism 4m if it operates with load collective "light" for more than 16 hours a day, but also if it operates with load collective "medium" for more than 8 to up to 16 or with load collective "heavy" for more than 4 to up to 8 hours per day.

Class	of		V ₀₀₆	v ₀₁₂	V ₀₂₅	V ₀₅	v_1	v ₂	V ₃	V4	V5	
working time		Average working time per day in h, relative to 1 year		up to 0,125	> 0,125 0,25	> 0,25 0,5	> 0,5 1	> 1 2	> 2 4	>4 8	> 8 16	> 16
	No	Desig- nation	Explanation		Group of mechanism							
Load collective	1	Light	Low frequency of maximum load	1E _m	1E _m	1D _m	1C _m	1B _m	1A _m	2 _m	3 _m	4 _m
	2	Medium	Same frequency of small, medium and highest loads	1Em	1D _m	1Cm	1B _m	1A _m	2 _m	3 _m	4 _m	5 _m
	3	Heavy	Highest loads almost every time	1D _m	1Cm	1B _m	1A _m	2 _m	3 _m	4 _m	5 _m	5 _m

Fig. 5: The groups of mechanism according to DIN 15020

Despite their different modes of operation all the three reeving systems are dimensioned according to the greatest line pull that might occur. This, however, is the same in all the three cases, resulting in the same minimal rope diameters. However, the highest line pull, which is, of course, the most negative factor of influence on the rope's service life, occurs in the three groups of mechanism at very different frequencies. Therefore the service life of the ropes in these three reeving systems will be of very different length, although they are classified in the same group of mechanism. This phenomenon is investigated in the following.

Fig. 6 shows the dimensioning of rope drives for the 9 groups of mechanism of DIN 15020 for a normal transport and a coefficient h2 = 1.12 for the sheaves. Deliberately the line pull (100,000 N) was chosen very high in order to keep the rounding errors very small when dimensioning the nominal rope diameter. As shown in Fig. 6 the nominal rope diameters for the 9 groups of mechanism range from 20 mm to 42 mm and the sheave diameters from 250 mm to 1,310 mm.

Rope design:StrFill factor:0.6Spinning factor:0.8Tensile strength:19	tratoplast .608 .87 960	t Line Typ Coe	e pull [N]: e: fficient h2	100, norn : 1.12	000 nal transpo	ort				
Group of mechanism:		1Em	1Dm	1Cm	1Bm	1Am	2m	3m	4m	5m
Layout according to DIN c-value: Min. rope diameter d min: Chosen rope diameter: Coefficient h1 sheave: Min. sheave diameter D mi	in:	0.063 19.92 20 11.2 250	0.067 21.19 22 12.5 297	0.071 22.45 23 14.0 353	0.075 23.72 24 16.0 426	0.085 26.88 27 18.0 542	0.095 30.04 31 20.0 673	0.106 33.52 34 22.4 841	0.118 37.31 38 25.0 1045	0.132 41.74 42 28.0 1310

Fig. 6: Dimensioning of rope drives for the groups of mechanism 1Em to 5m

For all the 9 groups of mechanism the theoretically sustainable average number of bending cycles of the rope in operation (in our case a Casar Stratoplast) was determined for the following conditions:

- line pull always maximal
- line pull according to load collective "heavy"
- line pull according to load collective "medium"
- line pull according to load collective "light"

The line pulls of the three load collectives and the frequencies of their occurrence were chosen according to the numerical examples of DIN 15020 (Fig. 7).

	No.	Designation	Load [%]	Frequency [%]
Load collective	1	Light	100.00 44.00 16.00	10.00 40.00 50.00
	2	Medium	100.00 77.30 54.70 32.00	16.67 16.67 16.67 50.00
	3	Heavy	100.00 63.00	50.00 50.00

Fig. 7: The numerical examples of the load collectives of DIN 15020 forming the basis of the calculations

Fig. 8 shows the sustainable number of bending cycles for the wire ropes when running over sheaves with the calculated sheave diameters under maximal load until rope discard and until rope break as well as for the line pulls according to the load collectives "heavy", "medium" and "light" (cf. Fig. 7).

Group	Only maximum load		Load collective "heavy"		Load collecti	ve "medium"	Load collective "light"	
nism	Discard [-]	Break [-]	Discard [-]	Break [-]	Discard [-]	Break [-]	Discard [-]	Break [-]
5 m 4 m 3 m 2 m 1 Am 1 Bm 1 Cm 1 Dm 1 Em	$\begin{array}{c} 1,029,100\\ 440,400\\ 203,400\\ 98,700\\ 49,900\\ 25,500\\ 16,300\\ 11,000\\ 7,600\end{array}$	2,632,500 1,065,100 466,100 212,500 101,700 48,400 28,900 18,400 12,000	1,324,300 559,300 255,400 122,200 61,200 30,800 19,400 12,900 8,800	3,407,200 1,361,700 589,600 265,300 126,000 59,200 34,800 21,900 14,100	3,048,800 1,254,500 559,300 259,100 127,200 61,800 37,200 23,700 15,700	7,931,000 3,095,500 1,311,700 573,400 267,300 121,700 68,700 41,500 26,100	6,110,200 2,472,700 1,085,000 491,600 237,700 112,700 65,400 40,300 26,100	16,001,600 6,154,500 2,571,700 1,102,400 507,100 225,800 123,700 72,600 44,800

Fig. 8: Number of bending cycles \tilde{N} until rope discard and until rope break under maximum load and under the line pulls of the three load collectives

Fig. 9 shows the number of bending cycles of the three load collectives divided by the number of bending cycles under maximal load, which were set as 100 %. Fig. 10 shows the calculated values in graphical form. It is obvious that the relationship is almost perfectly linear.

Which value should be the basis in the standard when calculating the expected number of hoisting cycles? In group of mechanism 4m under load collective "heavy" our wire rope would achieve a service life of approx. 28 % higher than under maximal load; under load collective "medium" it would achieve nearly three times the service life and under load collective "light" even nearly six times the service life!

Even in the lowest group of mechanism 1Em the wire rope would still achieve double the service life under load collective "medium", and under load collective "light" it even achieves almost four times the service life it would achieve under maximum load!

Generally speaking, the sustainable numbers of bending cycles increase with increasing group of mechanism; however, according to Fig. 8 a wire rope with load collective "light" in group of mechanism 3m with a number of bending cycles of

Group	Only maximum load		Load collective "heavy"		Load collective "medium"		Load collective "light"	
nism	Discard [%]	Break [%]	Discard [%]	Break [%]	Discard [%]	Break [%]	Discard [%]	Break [%]
5 m 4 m 3 m 2 m 1 Am 1 Bm 1 Cm 1 Dm 1 Em	100 100 100 100 100 100 100 100 100	100 100 100 100 100 100 100 100	129 127 126 124 123 121 119 117 116	129 128 126 125 124 122 120 119 118	296 285 275 263 255 242 228 215 207	301 291 281 270 263 251 238 226 217	594 561 533 498 476 442 401 366 343	608 578 552 519 499 467 428 395 373

Fig. 9: Number of bending cycles \tilde{N} until rope discard and until rope break according to Fig. 8, divided by the number of cycles under maximum load



Fig. 10: Number of bending cycles \tilde{N} until rope discard and until rope break of the load collectives, divided by the number of bending cycles under maximum load

1,085,000 achieves a much higher number of bending cycles than a wire rope in group of mechanism 4m with load collective "heavy" which only manages 559,300 cycles. It is obvious that the number of bending cycles to be expected is not only dependent on the group of mechanism but also, and rather more so, on the load collective. That is the reason why it is not very sensible to list the expected number of hoisting cycles in a standard without taking into account the load collectives.

9. The weighting of a reverse bending cycle

Early comparisons of test results of bending fatigue tests with simple bending cycles and of bending fatigue tests with reverse bending cycles led to the assumption that a reverse bending cycle would damage a wire rope about twice as much as a simple bending cycle. Therefore in DIN 15020 it was stipulated that <u>one</u> reverse bending cycle should be counted as <u>two</u> simple bending cycles. Further investigations under different conditions, however, showed that the damaging influence of the reverse bending cycles increases with improving conditions, ie the higher the expected service life of the rope, the greater the damaging effect of the reverse bending cycles. Consequently, the percental reduction of the service life of a wire caused by a reverse bending cycle increases with increasing sheave diameters and with decreasing line pulls.

According to Feyrer the number of sustainable reverse bending cycles until rope discard and until rope break can be determined as a function of the sustainable number of simple bending cycles and the D/d-ratio. The following formulas apply:

$$\tilde{N}_{A} = 3.635 \cdot \tilde{N}_{A} \cdot (D/d)^{0.499}$$

 $\tilde{N}_{n} = 9.026 \bullet \tilde{N}_{n}^{0.618} \bullet (D/d)^{0.424}$

Together with Casar engineer Dr. Briem the author has investigated the reduction of the number of bending cycles caused by a change of the bending direction for the different conditions of all the groups of mechanism in DIN 15020. This investigation was based on the reeving systems indicated in chapter 8. Fig. 8 shows the number of simple bending cycles, Fig. 11 the number of reverse bending cycles.

Fig. 12 shows the weighting factors as the relation of the number of bending cycles according to Fig. 8 and the reverse bending cycles according to Fig. 11. As for load collective "medium" the weighting factors for the numbers of cycles until rope discard are shown as a bar chart (cf. Fig. 13). It is evident that a reverse bending cycle in the most heavily strained groups of mechanism 1Em and 1Dm for load collective "medium" damages a wire rope about twice as much as a simple bending cycle. Under the conditions of group of mechanism 2m the damage caused by the reverse bending

Group	Only maximum load		Load collective "heavy"		Load collecti	ve "medium"	Load collective "light"	
nism	Discard [-]	Break [-]	Discard [-]	Break [-]	Discard [-]	Break [-]	Discard [-]	Break [-]
5 m 4 m 3 m 2 m 1 Am 1 Bm 1 Cm 1 Dm 1 Em	207,500 110,900 62,500 36,400 21,800 13,100 9,100 6,600 4,900	344,300 187,600 107,400 63,000 38,200 23,000 15,800 11,400 8,300	245,700 130,200 72,900 42,000 25,000 14,900 10,200 7,300 5,400	403,800 218,300 124,200 72,300 43,600 26,000 17,700 12,700 9,200	430,000 224,000 123,300 69,500 40,900 23,800 15,800 11,000 7,900	680,600 362,700 203,600 116,400 69,400 40,600 27,000 18,800 13,500	685,600 353,100 192,300 106,900 62,300 35,600 23,100 15,800 11,200	1,050,200 554,600 308,700 174,300 103,200 59,500 38,800 26,600 18,800

Fig. 11: Reverse bending cycles until rope discard and until rope break under maximum load and under the line pulls of the three load collectives

Group	Only maximum load		Load collective "heavy"		Load collecti	ve "medium"	Load collective "light"	
nism	Discard [-]	Break [-]	Discard [-]	Break [-]	Discard [-]	Break [-]	Discard [-]	Break [-]
5 m 4 m 3 m 2 m 1 Am 1 Bm 1 Cm 1 Dm 1 Em	4.96 3.97 3.25 2.71 2.28 1.94 1.79 1.67 1.56	7.65 5.68 4.34 3.37 2.66 2.11 1.83 1.62 1.44	5.39 4.29 3.51 2.91 2.44 2.07 1.90 1.76 1.64	8.44 6.24 4.75 3.67 2.89 2.28 1.97 1.73 1.53	7.09 5.60 4.54 3.73 3.11 2.60 2.35 2.15 1.98	11.65 8.54 6.44 4.93 3.85 3.00 2.55 2.21 1.94	8.91 7.00 5.64 4.60 3.82 3.17 2.83 2.55 2.34	15.24 11.10 8.33 6.32 4.92 3.79 3.19 2.73 2.38

Fig. 12: Weighting factors for reverse bending cycles

cycle is already nearly 4 times as great and finally, in group of mechanism 5m about seven times as great.

Naturally, the values will vary within certain limits for the different load collectives and especially for different rope designs, but the figures gained may serve as an indicator for the fact that the traditional way of counting (1 counter reverse bending cycle equals 2 simple bending cycles) does not sufficiently take into account the damaging influence of reverse bending cycles. Additionally, from these calculations one can draw the conclusion that reverse bending cycles should especially be avoided in rope drives of the higher groups of mechanism.



Fig. 13: Weighting factors for reverse bending cycles in load collective "medium"



Fig. 14: 4 lifting hoists

10. Comparing the wire rope service lives of 4 different hoists

In the following the wire rope service lives of 4 single-part hoists are compared. The drum and sheave diameters are the same in all four hoists and the ropes are of the same design and have the same nominal diameter. The mode of operation of all four hoists is also the same.

In hoist 1 (Fig. 14 a) the wire rope runs immediately onto the drum. During every lifting process the rope zones which run onto the drum carry out half a bending cycle. During every lowering process they carry out another half bending cycle, so that during every hoisting cycle one bending cycle (1 BC) is carried out (Fig. 15).



Fig. 15: Hoist 1 carries out 1 bending cycle during every hoisting cycles

In hoist 2 (Fig. 14 b) the most heavily strained rope zone first runs over a sheave where it carries out one complete bending cycle. Then it runs onto the drum and carries out another half bending cycle. When lowering the load the same piece of rope carries out another half bending cycle when spooling off the drum and yet one more bending cycle when running over the sheave, so that all in all 3 bending cycles are generated during every hoisting cycle (Fig. 16).

In hoist 3 (Fig. 14c) the most heavily strained rope zone first runs over two sheaves and carries out one complete bending cycle on each. Then it runs onto the drum carrying out another half bending cycle. When lowering the load the same piece of rope carries out another half bending cycle when spooling off the drum and two more bending cycles when running over the sheaves, so that all in all 5 bending cycles are generated during every hoisting cycle (Fig. 17).



Fig. 16: Hoist 2 carries out 3 bending cycles during every hoisting cycle.



Fig. 17: Hoist 3 carries out 5 bending cycles during every hoisting cycle.

In hoist 4 (Fig. 14 d) the sequence of the bending cycles is similar to that in hoist 3. However, the designer has arranged the drum in a way that the rope, when leaving the second sheave and running onto the drum, carries out a complete reverse bending cycle (RBC).

Consequently, during every hoisting cycle the most heavily strained rope zone carries out 3 bending cycles and 2 reverse bending cycles (Fig. 18). According to the weighting of the reverse bending cycle (depending on the working conditions 1 reverse bending cycle damages the rope just as much as 2 to 9 bending cycles; cf. Fig. 12) the most heavily strained rope zone here endures between 7 and 21 bending cycles.

So, depending on the crane design, during every complete hoisting cycle (=one lifting and one lowering process) 1, 3, 5 or 7 to 21 bending cycles are generated.



Fig. 18: Hoist 4 carries out 7 to 21 bending cycles during every hoisting cycle.



Fig. 19: The average wire rope service lives of the 4 hoists

Let us now suppose that hoist 1 achieves a service life of 24 months (Fig. 19). Under the same working conditions hoist 2 would generate three times the number of bending cycles, which would reduce the service life of the rope to 8 months. Hoist 3 would generate five times the number of bending cycles and the service life of the rope is reduced to a mere 5 months. Hoist 4 would generate 7 to 21 times the number of bending cycles reducing the rope's service life to between 3 1/2 months and 5 weeks.

It is quite clear that the service life of the rope does not only depend on the D/dratio, the line pulls, the quality of the rope design or on the crane's mode of operation; the number and arrangement of the sheaves in the reeving system are at least of the same importance.

11. Assessing the most heavily strained rope zone

Not all zones of a rope are subjected to the same number of bending cycles. For instance, the 3 dead-end turns on the drum are subjected to no bending cycles at all once the rope is installed. Other rope zones will run onto and off the drum during the lifting and lowering process, but apart from that they are not bent over any other sheaves. Other rope zones first run over several sheaves and onto the drum during the hoisting process and carry out the same bending cycles in the reverse order when the load is being lowered.



Fig. 20: Four-part hoist

After a certain working period the wire rope will most probably not fail along one of the less strained zones; the discard state will first be reached within that zone which carries out the highest number of bending cycles per hoisting cycle.

The position where this most heavily strained rope zone can be found does not only depend on the geometry of the rope drive but also on its mode of operation. Therefore it is not so easy to determine that most heavily strained rope zone and it is advisable to do this with the help of a computer. In the following this procedure is described using a four-part hoist as a simple example (cf. Fig. 20).



Fig. 21: Determining the most heavily strained rope zone

Under "State 1" the top section of Fig. 21 shows the rope length of the crane with the hook in its lowest position. Part of the rope is on the drum, three further zones can be found on sheaves S1, S2 and S3 at he beginning of the lifting process.

Under "State 2" the rope length is shown with the hook in its highest position. A large part of the rope length is now on the drum and again 3 large rope zones of equal

length can be found on sheaves S1, S2 and S3 with a considerably shorter distance between them. The calculation by the computer to determine the most heavily strained rope zone takes the following line of procedure:

The rope zone which can be found on the drum during "State 2", but was not there during "State 1", has obviously been wound onto the drum during the lifting process and has therefore carried out half a bending cycle.

All the rope zones which in "State 2" can be found <u>on the left</u> of sheave S3, but were still <u>on the right</u> of sheave S3 in "State 1", obviously ran over sheave 3 during the lifting process and have therefore carried out one complete bending cycle. Similarly, the zones which have carried out one bending cycle during the lifting process are determined for sheave 2 and sheave 1.

All the zones which can be found <u>on</u> sheave 3 in "State 2", but which were not there in "State 1", have obviously run onto sheave 3 during the lifting process but have not left it. Therefore they have carried out half a bending cycle during the lifting process.

All the zones which were <u>on</u> sheave 3 in "State 1", but are not there any more in "State 2", have obviously left sheave 3 during the lifting process carrying out half a bending cycle. This line of procedure for calculating the half bending cycles on the sheaves is applied to sheaves S2 and S1 in an analogous way.

Following these calculations all bending cycles along the rope length are totalized as shown in the lower section of Fig. 21.

When lowering the rope the same procedure occurs in reverse order, so that for a complete hoisting cycle the bending cycles will double compared to the figures shown in the lower section of Fig. 21.

If there was a rope zone which would run over all the three sheaves and then onto the drum, the maximum number of bending cycles during a lifting process would be $3 \ 1/2$ (1 for each sheave an 1/2 for the drum). As is shown, however, in the upper section of Fig. 21, there is no rope zone running over all the three sheaves. The most heavily strained rope zone is rather a zone running over sheaves 2 and 3 and then onto the drum. So the maximum number of cycles is $2 \ 1/2$ for the lifting process and 5 for the complete hoisting cycle. The most heavily strained rope zone which is subjected to that number of cycles is marked in the lower section of Fig. 21. That is the zone where the wire rope will first reach its discard state, provided there are no other relevant factors of influence.

12. The Palmgren-Miner-Rule

The damage accumulation hypothesis by Palmgren and Miner was first developed to calculate the service lives of ball bearings. Later research found out that the Palmgren-Miner-Rule can also be applied to wire ropes. Here it means that the relative numbers of bending cycles until discard or break of a wire rope (ie the numbers of cycles divided by the sustainable number of bending cycles) always add up to 1.

$$\Sigma \, \frac{ni}{Ni} = 1$$

In this formula ni is the number of bending cycles under condition i, and Ni the number of bending cycles sustainable under condition.

Two simple examples may illustrate the application of the Palmgren-Miner-Rule:

Example 1:

During every lifting process a wire rope carries out one bending cycle over a sheave under a line pull of 10 t. During every consecutive lowering process it carries out another bending cycle under a line pull of 4 t.

For a line pull of 10 t the number of cycles until discard is calculated at $N_1 = 30,000$, for a line pull of 4 t at $N_2 = 210,000$. How many complete hoisting cycles (= 1 lifting process under 10 t, 1 lowering process under 4 t) can the wire rope carry out until it reaches discard state?

Given $n_1 = n_2 = n$, the result according to Palmgren-Miner is:

N/30,000 + n/210,000 = 1, and therefore n = 26,250

The number of complete hoisting cycles until reaching discard state is 26,250.

Example 2:

During every hoisting cycle a wire rope runs over 4 sheaves with a diameter of 400 mm and over 2 sheaves with a diameter of 280 mm.

For a sheave with the diameter of 400 mm the sustainable number of bending cycles until discard state is $N_1 = 300,000$, for a sheave with the diameter of 280 mm it is $N_2 = 100,000$. How many hoisting cycles can the wire rope carry out until reaching discard state?

 $n_1 = 4n$, $n_2 = 2n$ and 4n/300,00 + 2n/100,000 = 1

and therefore n = 30,000

The wire rope can carry out 30,000 hoisting cycles until reaching discard state.

13. Factors of influence which are not taken into account

13.1 Corrosion

It goes without saying that severe corrosion can reduce the service life of a running rope considerably. Under the influence of corrosion a deviation from the predicted values must be expected.

13.2 Lubrication

On the one hand the lubricant is supposed to prevent the wire rope from corroding; on the other hand it is implemented to reduce the friction between the individual wires in order to guarantee smoother shifting of the rope elements when the rope is being bent. Insufficient lubrication will probably lead to a shorter service life of the rope.

13.3 Abrasion

As explained above the rope-specific parameters b0 to b5 were determined during bending fatigue tests on test machines. Consequently, abrasion to the extent caused by the relative motion of the rope elements and between the wire rope and the sheave, has influenced the values of the parameters. Excessive abrasion, however, which occurs for instance when wire ropes are working in abrasive environment, can reduce the service life to under the predicted value.

13.4 Groove material

The bending fatigue tests for determining the rope-specific parameters are carried out on steel sheaves. For other groove materials the rope performance can change.

13.5 Shape of the grooves

Ideally the grooves of the sheaves should have a diameter of nominal rope diameter +6% to +8%. The bending fatigue tests for determining the rope-specific parameters are carried out with a groove diameter of nominal rope diameter +6%.

If the grooves are too tight or too wide, the service life of the rope will be reduced in any case.

Based on laboratory tests some researchers have published reduction factors for the rope's service life as a function of the groove measure. The author, however, is convinced that in practice the reductions of the rope's service life caused by incorrect groove geometry are greater than those determined in the laboratory tests. During a laboratory test the wire rope can adapt to the groove by deforming along the short test zone to fit the groove. In practice, however, every time the rope travels over the sheave with the incorrect geometry, a different section of its circumference will come to lie in the groove, so that a deformation as mentioned above is not possible.

13.6 Fleet angle

When a rope runs onto a sheave under a fleet angle, it first touches the flange of the sheave and then rolls into the bottom of the groove. Normally the twist induced into the wire rope during this process has a negative effect on the rope's service life. Occasionally reduction factors for the wire rope service life dependent on the size of the fleet angle were suggested. But, as above, the author is of the opinion that those factors, gained exclusively by laboratory tests, do not provide a true picture of practical operations.

The extent of the rope's damage caused by the twist induced along one section does not only depend on the amount of the twist but also on the length of the rope which is subjected to taking that twist. For instance, the same twist of 360° is negligible if it can spread along 100 m rope length. It may, however, reduce the service life considerably, if it is induced into a rope length of only 10 m.

13.7 Tension-tension stresses

A wire rope does not only fatigue because of bending cycles running over sheaves or drums, but also because of repeated changes of line pull. Therefore even a standing rope, which never runs over a sheave, as for instance the suspension rope of a crane jib, has got a limited service life which normally is, however, several times higher than the service life of the running ropes of the same installation. The calculation of the service life of wire ropes strained by fluctuating tension will be dealt with in a separate brochure.

Before and after ropes run over sheaves, in most cases a change of line pull will occur in the wire rope caused by picking up and putting down the load. Provided the number of bending cycles per hoisting cycle is great and the damage to the rope caused by the change of line pull is at least one magnitude smaller than the damage caused by the bending cycles, the influence of the change of line pull on the service life of a running wire rope can be neglected.

Let us suppose, for example, that in a reeving system the number of sustainable bending cycles until discard was $\tilde{N}A = 100,000$ with the rope running over sheaves in a loaded condition. If during every hoisting cycle the rope runs back over 5 sheaves and forth over 5 sheaves under load (ie it carries out 10 bending cycles per hoisting cycle), in terms of figures it will achieve its discard state after 10,000 hoisting cycles, not taking into account the effect of the change of line pull on the rope's service life.

During each of those hoisting cycles the rope is, however, additionally damaged by the change of line pull from basic level to load level and back to basic level. The rope can take that change of line pull for example $\tilde{N}AZ = 1$ million times until reaching its discard state. According to Palmgren-Miner the damage adds up in the following way:

1/N = 10/100,000 + 1/100,000

and therefore N = 9,900

It is obvious that the result changes by only 1 % when taking the change of line pull into account.

13.8 Recurring motions during automatic operation

Increasing the line pull lengthens the wire rope, reducing the line pull shortens it. That change of length occurs along the greatest part of the rope length without any impediment from outside, and the damaging effect of such a change of line pull can quite easily be simulated in fluctuating tension fatigue tests. Those zones of the wire rope, however, which are just lying on a sheave or on the first winding of the drumwhen the change of line pull occurs, show a very different reaction. Those ropes zones can only lengthen or shorten if they carry out motions relative to their support, e.g. the groove. These sliding motions are always accompanied by additional stresses in the wire rope and by wear on the rope's surface as well as on the surface of the sheave or drum.

If the points of loading or unloading always change, the damaging additional stresses will spread along the rope length. If, for instance, a certain rope zone comes to lie on and needs to slip over a sheave with only every twentieth loading action, the influence on the service life is negligible.

If, however, the rope undergoes the change of line pull always in the <u>same</u> zone, as is the case in automatic operation, the change of line pull <u>cannot</u> be neglected. In this case always the same rope zones are subjected to additional stresses and greater wear. That is one of the reasons why installations with automatic operation and recurring motions achieve clearly shorter service lives than comparable machines with random operation.



Fig. 22: A drum sawn in two by the wire rope at the load pickup point

Fig. 22 shows a drum sawn in two by the wire rope at the load pickup point. It is perfectly clear that also the rope was subjected to extreme wear.

Research into the influence of the additional damage on the load pickup point does exist but, in the author's opinion, it has not been verified to an extent which would justify taking it into account for the calculations.

14. Optimizing the reeving system

The software presented here does not only allow to predict the wire rope service life under given conditions, it also makes it possible to optimize the reeving system with regard to maximum wire rope service life or, if the wire rope service life is given, to minimize costs and/or structural dimensions.

For example, the reeving system in Fig. 14b, consisting of a drum and a sheave, is supposed to carry out on average 200,000 hoisting cycles until rope discard. The drum, the gear box and the motor should be as small as possible. As Fig. 16 shows, per hoisting cycle two bending cycles are carried out on the sheave and one on the drum. By means of the software one can work out how much the sheave diameter must be increased to compensate for the loss of wire rope service life caused by the reduction of the drum diameter (Fig. 23). The first calculation determines the number of bending cycles which the wire rope can carry out on the drum with the given diameter under the given conditions. Then the Palmgren-Miner formula is applied to calculate the necessary number of bending cycles on the sheave for achieving the desired 200,000 hoisting cycles. Finally the sheave diameter required for that number is calculated.

Drum	BC	BC	Sheave	Hoisting
diameter	drum	sheave	diameter	cycles
[mm]	[-]	[-]	[mm]	[-]
500	600,000	600,000	500	200,000
475	501,600	665,300	515	200,000
450	415,000	720,100	530	200,000
425	339,600	973,100	575	200,000
400	274,600	1,472,400	645	200,000

Fig. 23: The combination of drum and sheave diameters which will achieve 200,000 hoisting cycles each

15. Final remarks

Casar and the author will always be willing to provide service life predictions either as a help for dimensioning when designing reeving systems or for evaluating existing ones. As mentioned several times before, because of the statistical nature of the predictions and because of the many additional factors of influence on wire rope service lives the calculated values can under no circumstances be guaranteed.

For calculating a wire rope service life the following information is required:

- 1. Very detailed documents about the reeving (sketch and/or design drawings), and information on the mode of operation
- 2. The rope design (e.g. Casar Stratoplast, regular lay, 1770 N/mm²)
- 3. The nominal rope diameter (e.g. 20 mm)
- 4. The sheave diameter (e.g. 500 mm)
- 5. The drum diameter (e.g. 400 mm)

6. The line pull (e.g. 20,000 N) or the load collective per line (e.g. 10,000 N in 60 % of all hoisting cycles, 25,000 N in 40 % of all hoisting cycles)

It goes without saying that all the information submitted for the calculation will be treated with the strictest confidence.

Please send any comments or suggestions for improvements to the author of this brochure.

Dipl.- Ing Roland Verreet Ingenieurbüro für Fördertechnik Grünenthaler Str. 40a, D-52072 Aachen Tel. 02 41 / 17 31 47, Fax 02 41 / 1 29 82 e-mail: R.Verreet@t-Online.de

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CASAR DRAHTSEILWERK SAAR GMBH

Casarstrasse 1 P.O. Box 187 Phone: Phone Sales Dept.: Fax Sales Dept.:

D-66459 Kirkel• Germany
D-66454 Kirkel• Germany
+ 49-6841 / 8091-0
++ 49-6841 / 8091-350
++ 49-6841 / 8091-359

E-mail: sales.export@casar.de http://www.casar.de