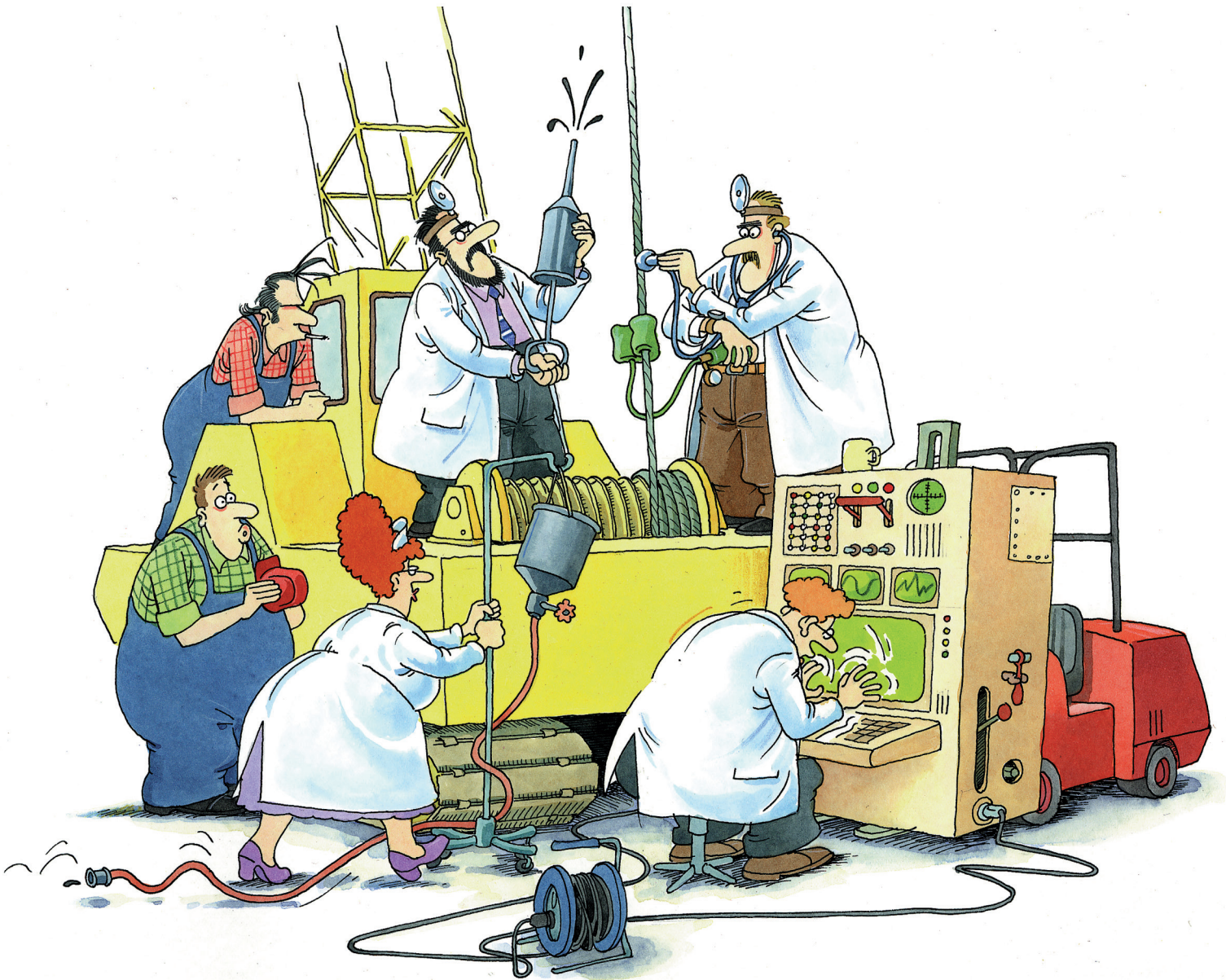


WIRE ROPE

TECHNOLOGY AACHEN



**Calculating the service life
of running steel wire ropes**

Calculating the service life of running steel wire ropes

by Dipl.-Ing. Roland Verreet

Content

1. Introduction	4
2. Calculating the number of achievable bending cycles	4
3. The definition of a bending cycle	6
4. The definition of a reverse bending cycle	6
5. Service life prediction	7
6. The optimal nominal rope diameter	10
7. The most economical rope diameter	11
8. The influence of the load collective	11
9. The weighting of a reverse bending cycle	15
10. Comparing the wire rope service lives of 4 different hoists	18
11. Assessing the most heavily strained rope zone	21
12. The Palmgren-Miner-Rule	24
13. Factors of influence which are not taken into account	25
14. Optimizing the reeving system	28
15. Final remarks.....	30

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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 Caesar 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 achievable bending cycles

Even the inventor of the wire rope, Oberbergat 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

$$\lg N = b_0 + (b_1 + b_3 \cdot \lg \frac{D}{d}) \cdot (\lg \frac{S}{d^2} - 0.4 \cdot \lg \frac{R_o}{1770}) + b_2 \cdot \lg \frac{D}{d} + \lg f_d + \lg f_L + \lg f_E$$

f_d takes into account the scale effect, f_L the length of the most stressed rope zone and f_E the type of rope core.

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
- Ro the nominal tensile strength of the wire in N/mm²

The factors b_0 to b_3 are rope-specific parameters which must be determined separately in a great number of bending fatigue tests for every single rope design.

Many rope makers run their own tests on bending fatigue machines to determine those parameters for their own products. Every single test increases the number of data corroborating the predictions of the service life of these ropes.

2.1. The average number of bending cycles \tilde{N}

By means of statistical procedures it is possible to determine the factors b_0 to b_3 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 on average. However, he must bear in mind that average value 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 not reach it.

That means that a number of bending cycles defined as the average value 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 average value itself implies that half of all ropes do not achieve that value.


2.2. The number of bending cycles N_{10}

There are situations in which it does not suffice to know that the wire rope will achieve the calculated number of bending cycles on average. 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 N_{10} 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 ). 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 ).

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 angle 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 distance 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.

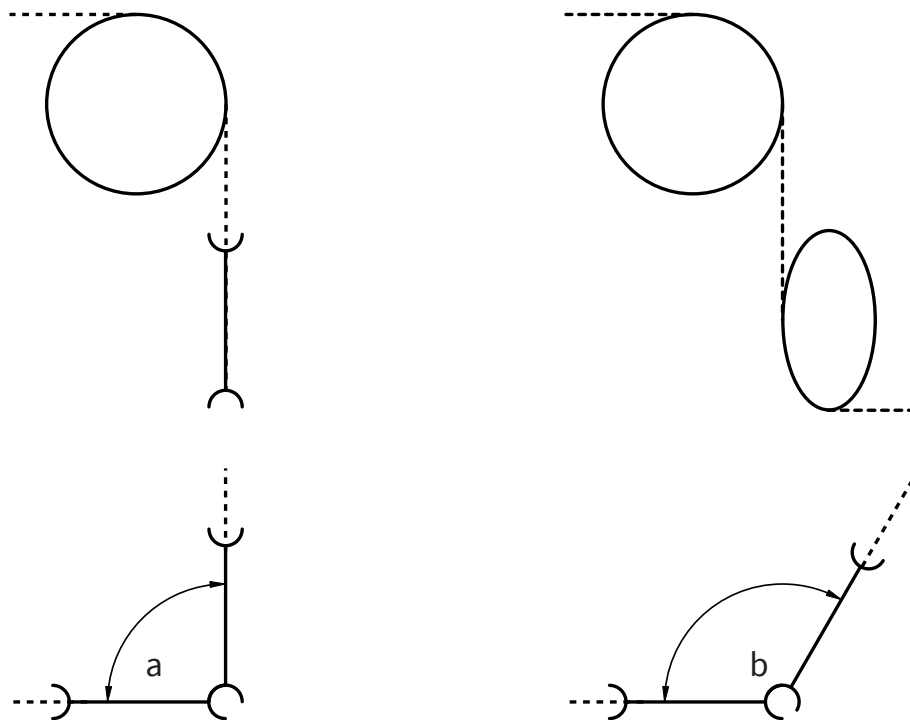


Fig. 1: Change of the bending plane by 90° (a) and 120° (b)

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 achievable average number of bending cycles \tilde{N} until rope discard and rope break as well as the number of bending cycles N_{10} until rope discard and rope break which 90 % of all ropes achieve with a 95 % probability.

Example: 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/mm² 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.

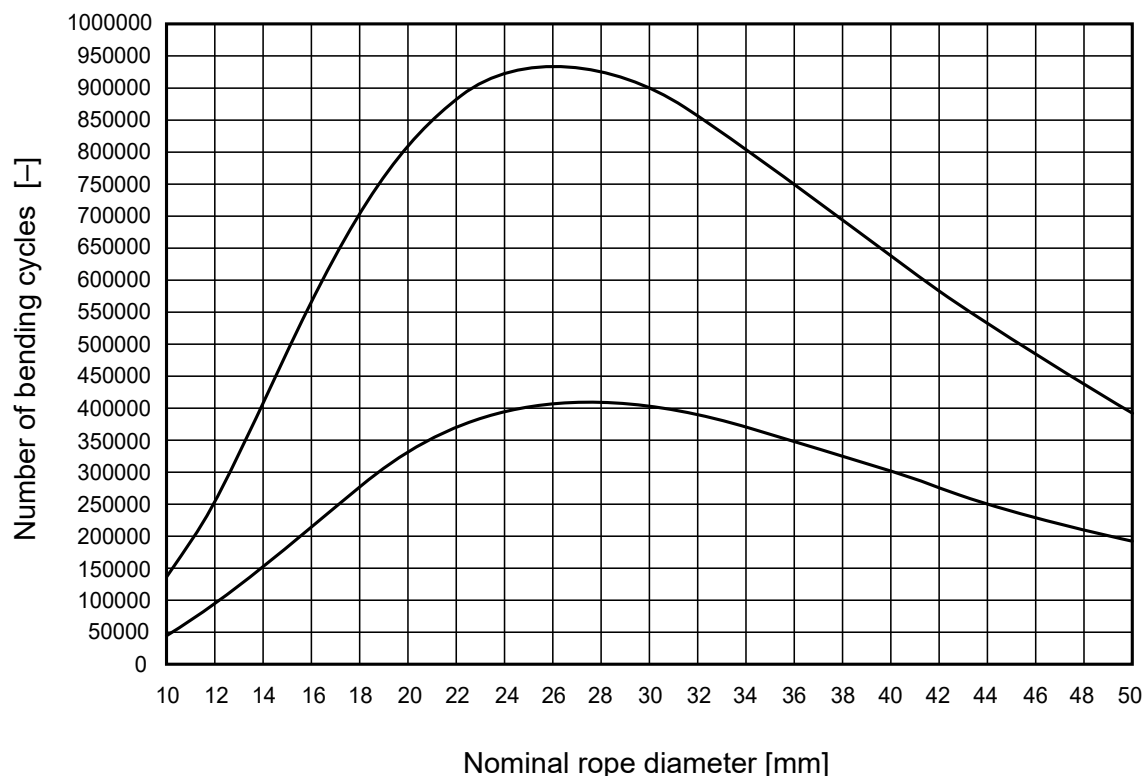


Fig. 2: Number of bending cycles as a function of the nominal rope diameter

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 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.

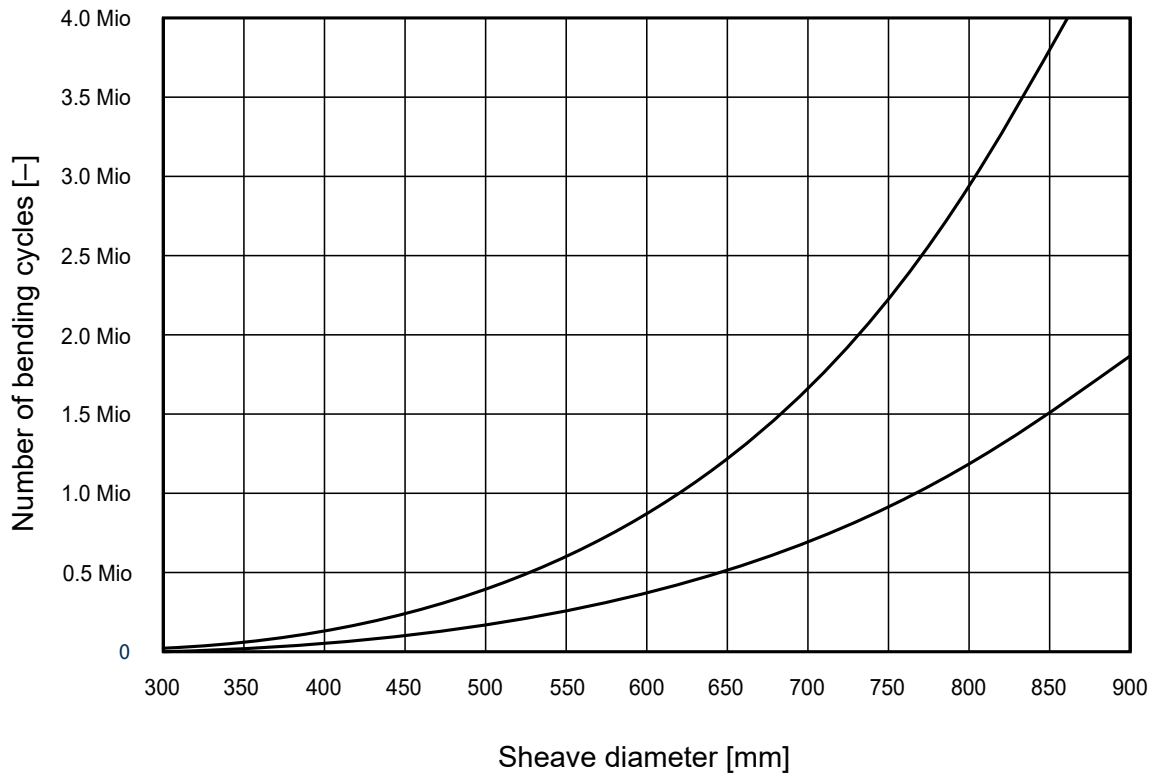


Fig. 3: Number of bending cycles as a function of the sheave diameter

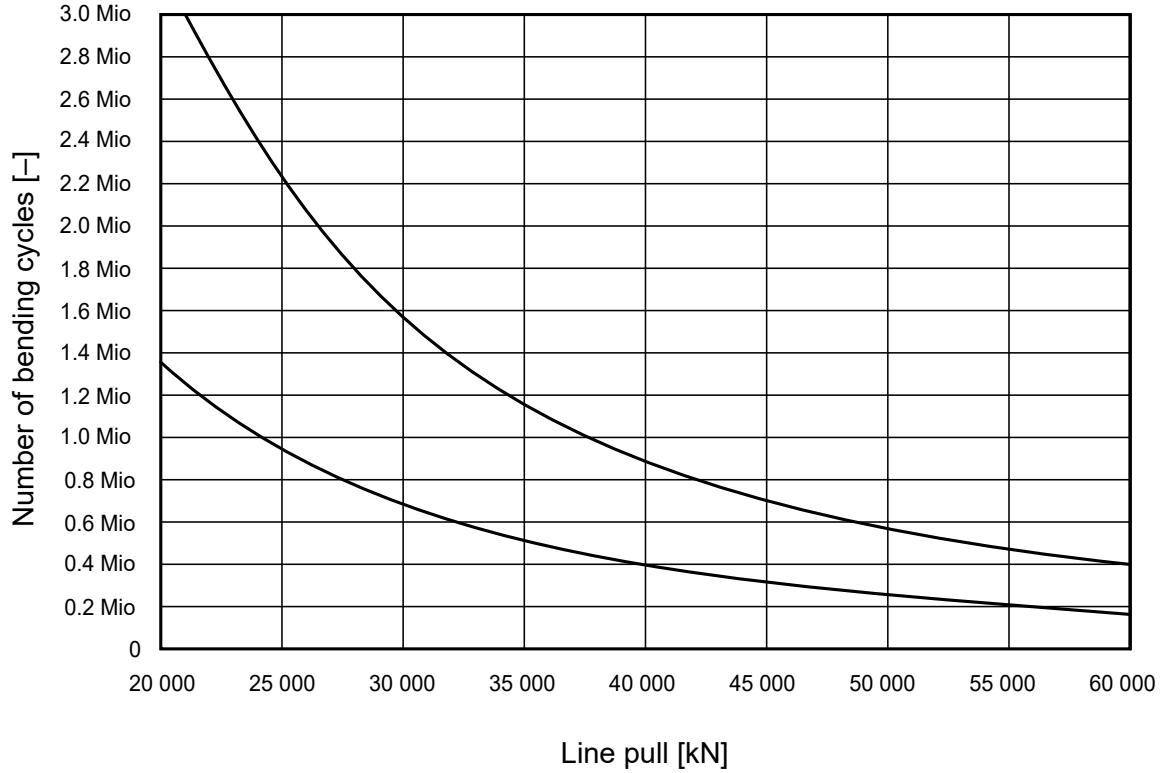


Fig. 4: Number of bending cycles as a function of the line pull

6. The optimal nominal rope diameter

Whereas the numbers of achievable bending cycles continuously increase with increasing sheave diameter (Fig. 3) and continuously decrease with increasing line pull (Fig. 4), the achievable 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 working time	Symbol		V006	V012	V025	V05	V1	V2	V3	V4	V5	
	Average working time per day in h, relative to 1 year		≤ 0,125	> 0,125 ≤ 0,25	> 0,25 ≤ 0,5	> 0,5 ≤ 1	> 1 ≤ 2	> 2 ≤ 4	> 4 ≤ 8	> 8 ≤ 16	> 16	
Load collective	No	Designation	Explanation		Group of mechanism							
	1	Light	Low frequency of maximum load	1Em	1Em	1Dm	1Cm	1Bm	1Am	2m	3m	4m
	2	Medium	Same frequency of small, medium and highest loads	1Em	1Dm	1Cm	1Bm	1Am	2m	3m	4m	5m
	3	Heavy	Highest loads almost every time	1Dm	1Cm	1Bm	1Am	2m	3m	4m	5m	5m

Fig. 5: The groups of mechanism according to DIN 15 020

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.

Rope design:	Stratoplast									
Fill factor:	0.608		Line pull [N]:	100.000						
Spinning factor:	0.87		Type:	übl. Transport						
Tensile strength:	1960		Coefficient h2:	1.12						
Group of mechanism:	1Em	1Dm	1Cm	1Bm	1Am	2m	3m	4m	5m	
Layout according to DIN										
c-value:	0.063	0.067	0.071	0.075	0.085	0.095	0.106	0.118	0.132	
Min. rope diameter d min:	19.92	21.19	22.45	23.72	26.88	30.04	33.52	37.31	41.74	
Chosen rope diameter:	20	22	23	24	27	31	34	38	42	
Coefficient h1 sheave:	11.2	12.5	14.0	16.0	18.0	20.0	22.4	25.0	28.0	
Min. sheave diameter D min:	250	297	353	426	542	673	841	1045	1310	

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

Fig. 6 shows the dimensioning of rope drives for the 9 groups of mechanism of DIN 15020 for a normal transport and a coefficient $h_2 = 1.12$ 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.

For all the 9 groups of mechanism the theoretically achievable average number of bending cycles of the rope in operation (in our case a rope 8 x 25F - EPWRC) 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).

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

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

Fig. 8 shows the achievable 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“ (see Fig. 7).

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.

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

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

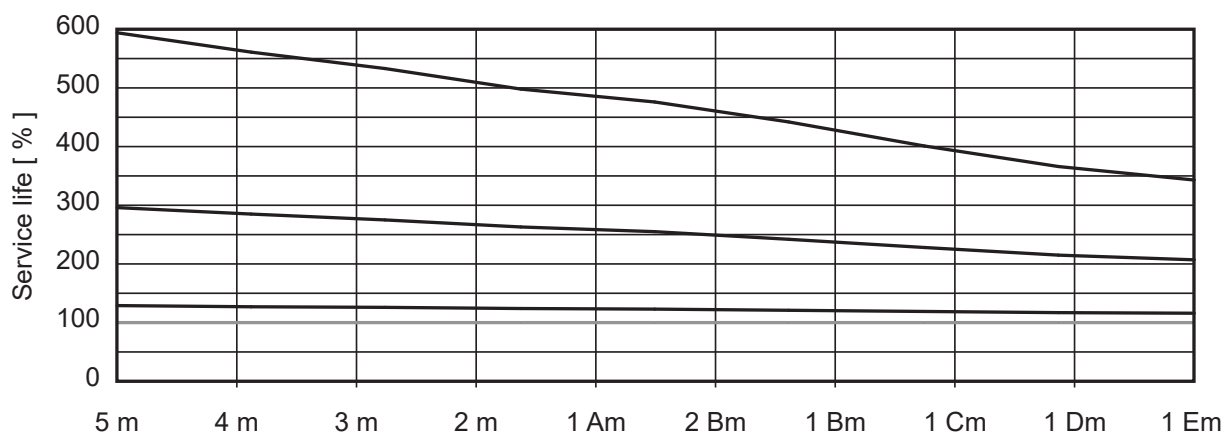


Fig. 10: Number of bending cycles \tilde{N} until rope discard and until rope break of the load collectives, divided by the number of bending cycles under maximum load

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 achievable 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 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 one reverse bending cycle should be counted as two 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 achievable reverse bending cycles until rope discard and until rope break can be determined as a function of the achievable number of simple bending cycles and the D/d-ratio. The following formulas apply:

$$\tilde{N}_{A\sim} = 3.635 \cdot \tilde{N}_{A\smile}^{0.671} \cdot (D/d)^{0.499}$$

$$\tilde{N}_{\sim} = 9.026 \cdot \tilde{N}_{\smile}^{0.618} \cdot (D/d)^{0.424}$$

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.

Group of mechanism	Only maximum load		Load collective "heavy"		Load collective "medium"		Load collective "light"	
	Discard [-]	Break [-]	Discard [-]	Break [-]	Discard [-]	Break [-]	Discard [-]	Break [-]
5 m	207,500	344,300	245,700	403,800	430,000	680,600	685,600	1,050,200
4 m	110,900	187,600	130,200	218,300	224,000	362,700	353,100	554,600
3 m	62,500	107,400	72,900	124,200	123,300	203,600	192,300	308,700
2 m	36,400	63,000	42,000	72,300	69,500	116,400	106,900	174,300
1 Am	21,800	38,200	25,000	43,600	40,900	69,400	62,300	103,200
1 Bm	13,100	23,000	14,900	26,000	23,800	40,600	35,600	59,500
1 Cm	9,100	15,800	10,200	17,700	15,800	27,000	23,100	38,800
1 Dm	6,600	11,400	7,300	12,700	11,000	18,800	15,800	26,600
1 Em	4,900	8,300	5,400	9,200	7,900	13,500	11,200	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 of mechanism	Only maximum load		Load collective "heavy"		Load collective "medium"		Load collective "light"	
	Discard [-]	Break [-]	Discard [-]	Break [-]	Discard [-]	Break [-]	Discard [-]	Break [-]
5 m	4.96	7.65	5.39	8.44	7.09	11.65	8.91	15.24
4 m	3.97	5.68	4.29	6.24	5.60	8.54	7.00	11.10
3 m	3.25	4.34	3.51	4.75	4.54	6.44	5.64	8.33
2 m	2.71	3.37	2.91	3.67	3.73	4.93	4.60	6.32
1 Am	2.28	2.66	2.44	2.89	3.11	3.85	3.82	4.92
1 Bm	1.94	2.11	2.07	2.28	2.60	3.00	3.17	3.79
1 Cm	1.79	1.83	1.90	1.97	2.35	2.55	2.83	3.19
1 Dm	1.67	1.62	1.76	1.73	2.15	2.21	2.55	2.73
1 Em	1.56	1.44	1.64	1.53	1.98	1.94	2.34	2.38

Fig. 12: Weighting factors for 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 (see 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 cycle is already nearly 4 times as great and finally, in group of mechanism 5m about seven times as great.

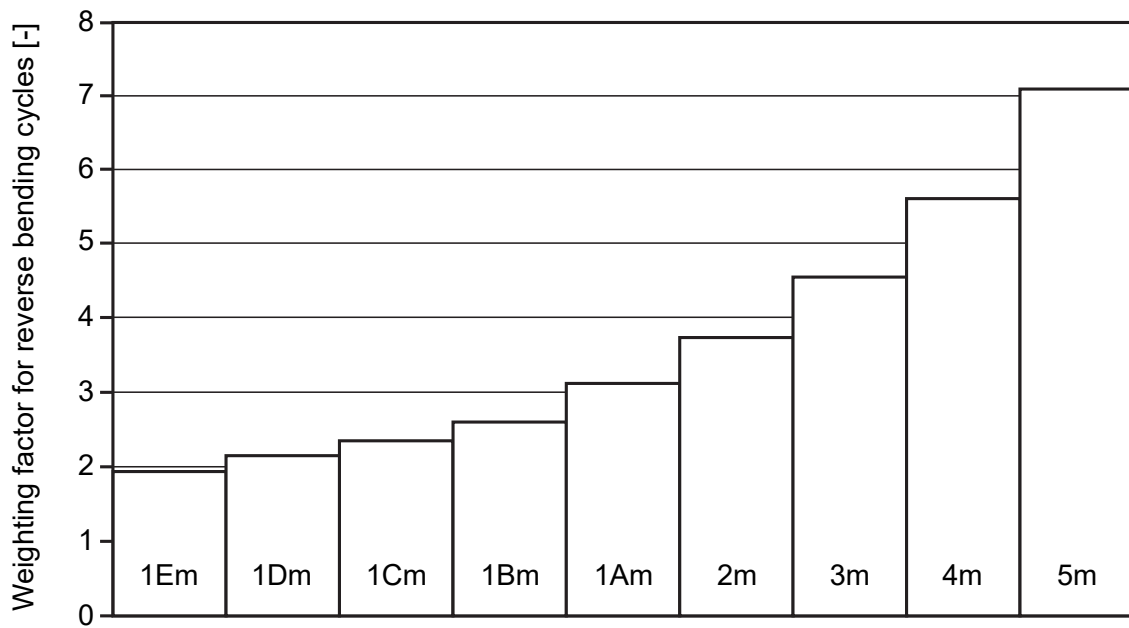


Fig. 13: Weighting factors for reverse bending cycles in load collective „medium“

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.

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.

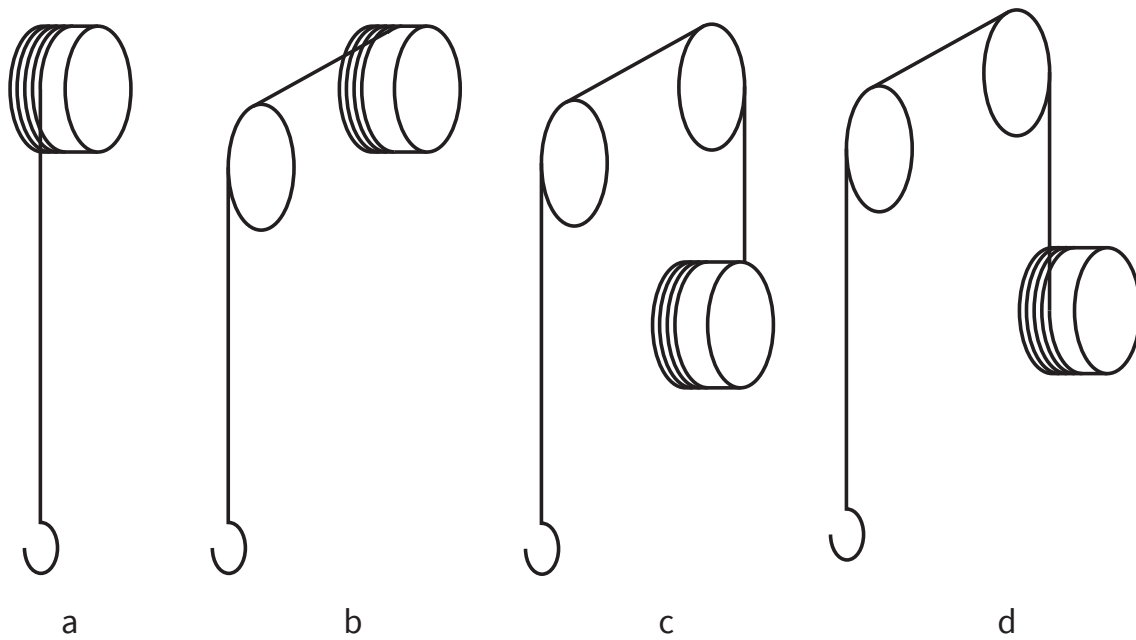


Fig. 14: 4 lifting hoists

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).

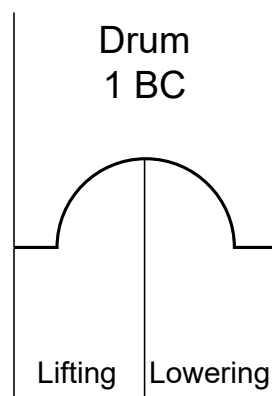


Fig. 15: Hoist 1 carries out 1 bending cycle during every hoisting cycles

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).

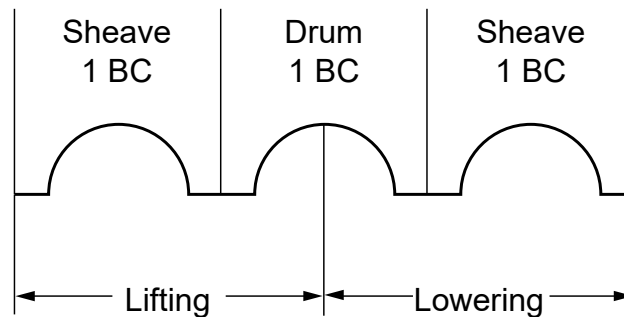


Fig. 16: Hoist 2 carries out 3 bending cycles during every hoisting cycle.

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).

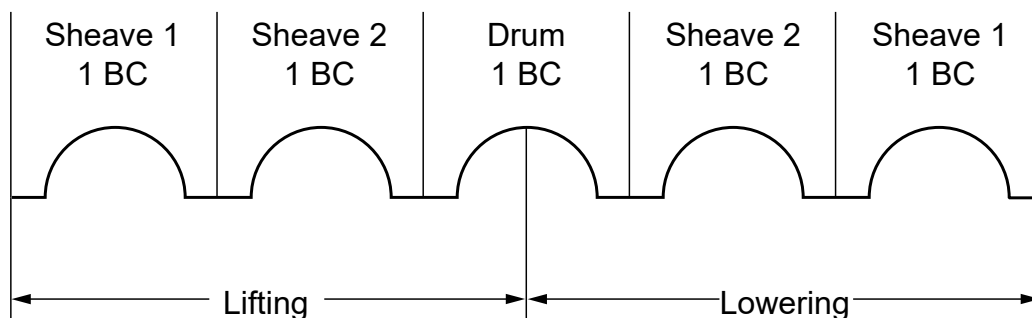


Fig. 17: Hoist 3 carries out 5a bending cycles during every hoisting cycle.

In hoist 4 (Fig. 14d) 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).

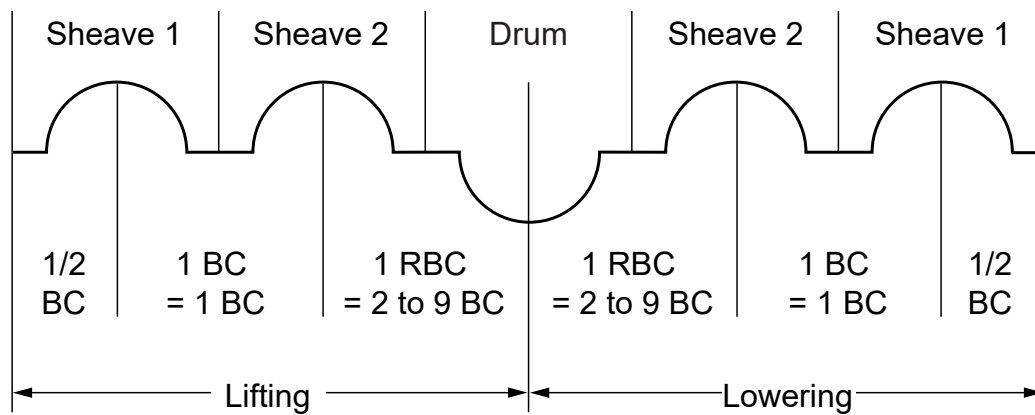


Fig. 18: Hoist 4 carries out 7 to 21 bending cycles during every hoisting cycle.

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; see 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.

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/d-ratio, 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.

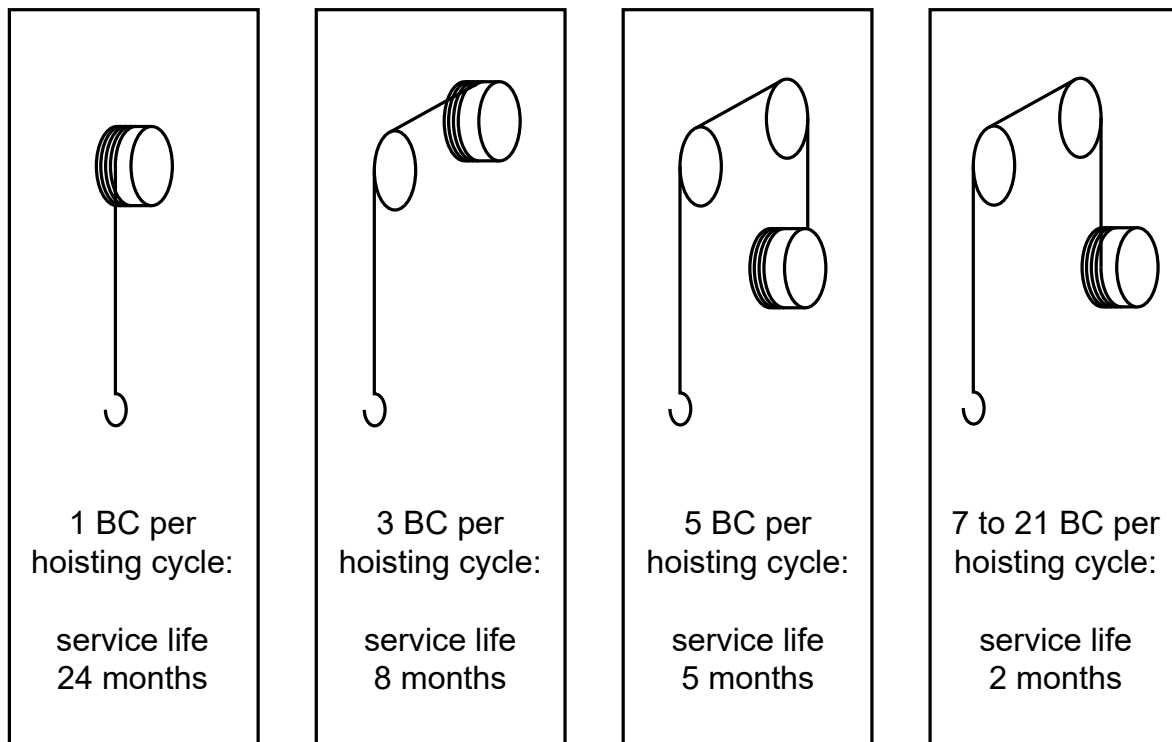


Fig. 19: The average wire rope service lives of the 4 hoists

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.

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 (see Fig. 20).

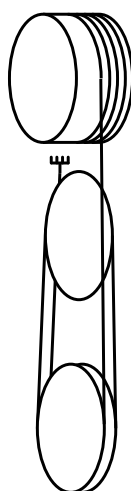


Fig. 20: Four-part hoist

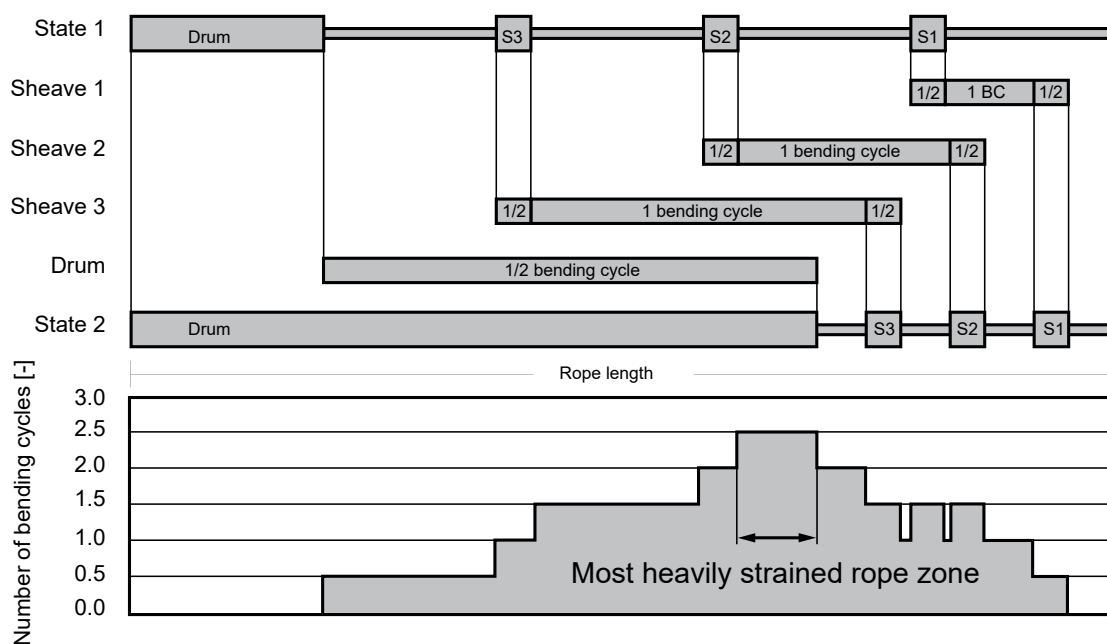


Fig. 21: Determining the most heavily strained rope zone

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 S3, S2 and S1 at the 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 S3, S2 and S1 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 on the left of sheave S3, but were still on the right 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 on 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 on 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 $3 \frac{1}{2}$ (1 for each sheave and $\frac{1}{2}$ 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 $2 \frac{1}{2}$ 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 achievable number of bending cycles) always add up to 1.

$$\sum \frac{n_i}{N_i} = 1$$

In this formula n_i is the number of bending cycles under condition i , and N_i the number of bending cycles achievable under this condition.

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

12.1. 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:

$$n/30,000 + n/210,000 = 1, \text{ and therefore } n = 26,250$$

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

12.2. 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 achievable 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?

$$n_1 = 4n, n_2 = 2n \text{ and } 4n/300,000 + 2n/100,000 = 1 \text{ and therefore } n = 30,000$$

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 b_0 to b_3 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 achievable 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:

$$\begin{aligned}1/N &= 10/100,000 + 1/1,000,000 \text{ and therefore} \\ N &= 9,900\end{aligned}$$

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 drum when 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 same zone, as is the case in automatic operation, the change of line pull cannot 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 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.

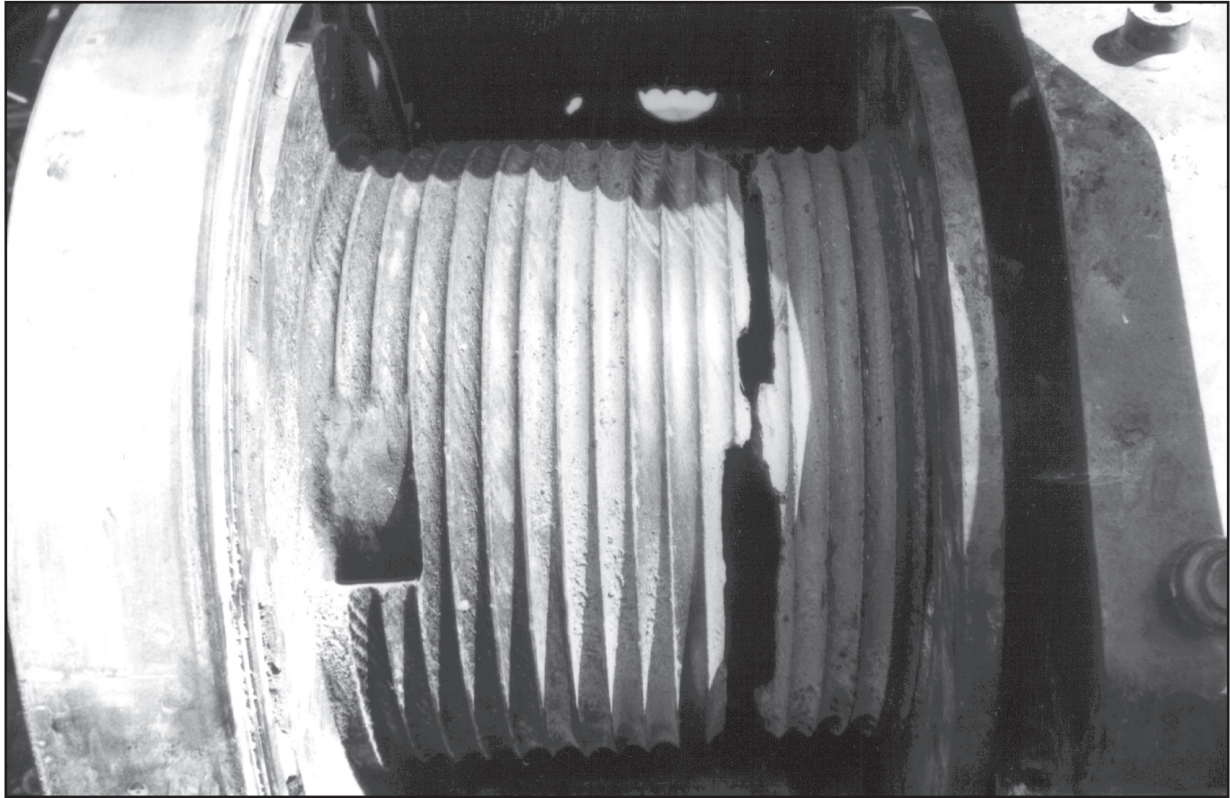


Fig. 22: A drum sawn in two by the wire rope at the load pickup point

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).

Drum diameter [mm]	BC drum [-]	BC sheave [-]	Sheave diameter [mm]
500	600.000	600.000	500
475	501.600	665.300	515
450	415.000	720100	530
425	339.600	973100	575
400	274.600	1.472.400	645

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

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.

15. Final remarks

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. 8 x 25F - EPWRC, regular lay, 1770N/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.

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