MECHANICAL BEHAVIOR OF HIGH PERFORMANCE FIBER ROPES IN TECHNICAL APPLICATIONS

Prof. Dr.-Ing. Markus Michael[1], Dr.-Ing. Thorsten Heinze[1], Dr.-Ing. Jens Mammitzsch[1]

[1] Technische Universität Chemnitz, Endowed Professorship of Technical Textiles & Textile Mechanical Components, Reichenhainer Str. 70, D-09126 Chemnitz, Germany

Email: markus.michael@mb.tu-chemnitz.de, thorsten.heinze@mb.tu-chemnitz.de, jens.mammitzsch@mb.tu-chemnitz.de, web page: http://www.tu-chemnitz.de

Keywords: fibre ropes, cyclic bend, winding test, CBOS test machine, abrasion

ABSTRACT

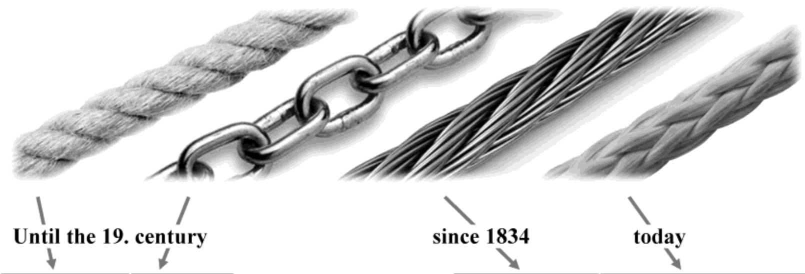
During the last years, fiber ropes have proven to be an option to replace steel wire ropes in many technical applications. Especially for hoisting applications or winch-based applications, fiber ropes (e.g. made from high-modulus polyethylene, TLCP fibers or aramid fibers) do offer a lot of advantages. Due to their low material density paired with high strength, these fiber ropes can ease handling and installation of ropes in technical applications at comparable performance. For applying a fiber rope in a technical application, e.g. hoisting gears, knowledge on the behavior under mechanical stress is essential. Especially the behavior under tensile load, tension-tension and cyclic bending has to be known to find an adequate fiber rope for special applications. This publication is showing the mechanisms to correctly analyze the application-related mechanical properties of fiber ropes and the testing machinery, and describes special applications where fiber ropes can already be used to replace steel wire ropes.

1 INTRODUCTION

Steel wire ropes and round steel chains are a common and well investigated mechanical component in engineering. Due to their high strength and the good predictability of their lifetime ([2]), there are used in hoisting gears, cranes and other lifting applications. The main disadvantage of steel chains is their non-redundancy. If one link is breaking, the whole chain is failing and can no longer be used in the application (

Figure 1). With the development of wire ropes in 1834, this problem has been solved by twisting iron wires to form a wire rope (0). These ropes now showed redundancy, because the break of one or more wires (nowadays steel wires to be used instead of iron wires) did no more cause the immediate failure of the whole tension member.

A disadvantage of steel wire ropes is the high density of steel, which causes a high weight of the mechanical component, especially in bigger rope diameters and high rope lengths. This leads to high energy consumption just for moving the mechanical component, the steel wire rope. Regarding high discharge heads (e.g. deep mining or ultra-high cranes), this also leads to the fact, that a big amount of the strength of the rope is needed to bear the load of the steel wire rope's weight, what results in lower payload.



Hemp rope	Chain	Requirements	Wire rope	HM-HT-fiber rope
✓	×	High operational safety	✓	✓
×	✓	High tenacity	✓	✓
×	✓	Low elongation	✓	✓
✓	×	Low mass	×	✓

Figure 1: Tension members in vertical conveying applications. [4]

Since the late 1970s, a number of polymer fibers are available on the market. Ropes made from such polymer fibers do offer a comparable or even better tensile strength as steel wire ropes. During the last years, ropes made from these polymer fibers have found their ways into technical applications because of the lower density of the fibers, what leads to a better strength-payload-ratio, compared to steel wire ropes ([5]).

The use of steel wire ropes is limited by the accessible hoisting depth L. The main reason for this is the high rope weight, which is resulting in a reduction of payload and unfavourable twist behaviour (change in the lay length). In simplified term, the payload m_F can be described as the difference between maximum permissible mass m_{zul} and rope mass m_s as follows:

$$m_\mathrm{F}(L) = m_\mathrm{zul} - m_\mathrm{s} = \{ F_\mathrm{min} / (s \cdot g) \} - T_\mathrm{t} \cdot L \quad (1)$$

As the tensile strength of steel wire ropes does not increase significantly, and the rope weight cannot be reduced, increased hoisting lengths can only realized by reducing payload m_F or by smaller safety factors s. However, economical transport of goods rules out the option of reducing the payload. For this reason, greater hoisting depths can currently only be implemented by the length-related reduction of safety factors, for instance in compliance with SABS 0294, by

$$s(L) \geq 2500 / (4000 + L) \quad (2)$$

or in compliance with TAS (cargo transport), by

$$s(L) \geq 7.2 - 0.005 \cdot L \quad (3)$$

These measures do negatively affect the rope's service life and consequently the operating safety. Future endeavors to increase hoisting depths to more than 3000 m and increase payload at the same time, are requiring new tension members – such as HM-HT fiber ropes – with lower weight at comparable tensile strength.

Table 1 is giving a comparison of HM-HT fiber ropes which provide higher breaking force (factor 1.2), lower rope mass (factor 4.9) and consequently a substantially higher breaking length (factor 6) compared to steel wire ropes (Seale).

Rope type	**d_N[mm]**	**F_{min} [kN]**	**Tt [kg/m]**
Fiber rope (Technora® T221)	48	1600	1.76
Steel wire rope (6x19S-FC 1770 U sZ)	48	1349	8,58

Table 1: Rope parameters.

Assuming ideal conditions, the maximum payload corresponds to the mass m_{Ref} which can be achieved using a hoisting rope when fully utilizing the rope's minimum break load ($\lim_{s\to 1} m_{zul}(s)$) and neglecting the rope mass ($\lim_{L\to 0} m_s(L)$).

If the payload m_F is, resulting from the relevant safety factors and hoisting depths, placed in relation to the reference mass m_{Ref} using the values from table 1, the relative payload (Figure 2) can be calculated for both, the steel wire rope and the HM-HT fiber rope, depending on the relevant safety factors in accordance with equation (2) and equation (3) for different hoisting depths, which helps to characterize the technical potential.

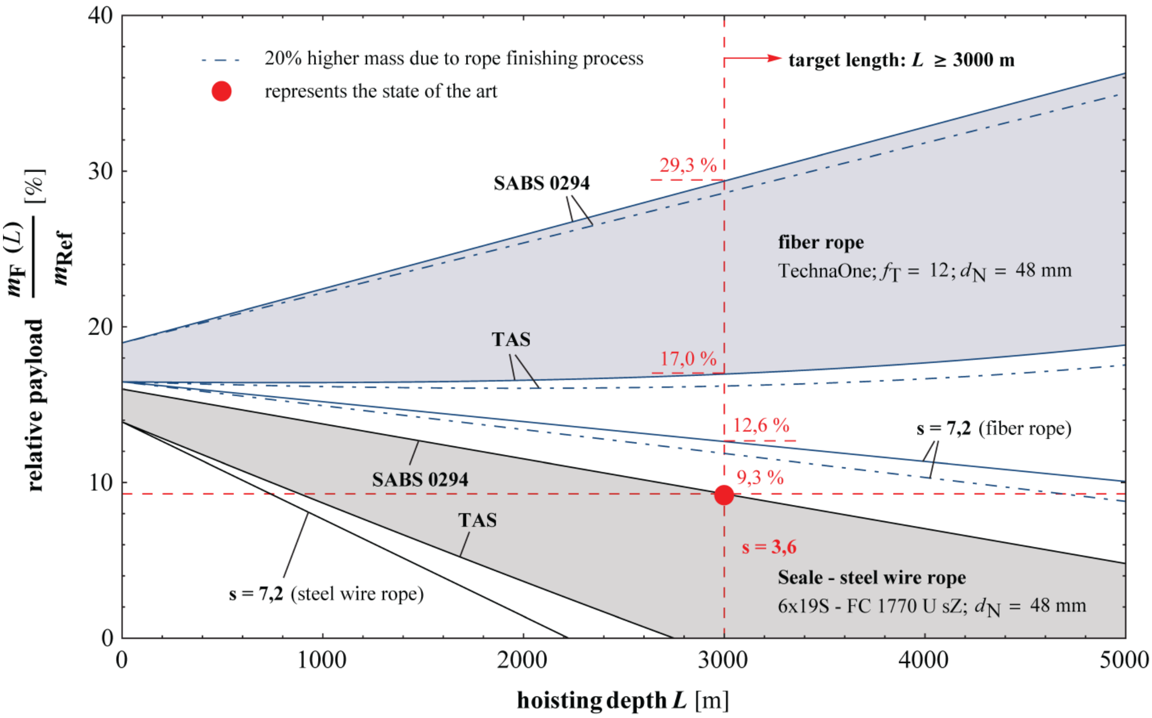


Figure 2: Influence of the hoisting depth on the payload. [4]

Figure 2 indicates that using steel wire ropes at hoisting depths of more than 3000 m, which are already achieved in African mines ([7]), is technically only feasible by working at substantially reduced safety factors, e.g. in accordance with SABS 0294. At a length of L = 3000 m the safety factor is extremely small (s = 3.6), which causes a need for very high D/d-ratios ($D/d \geq 75$; relation between sheave diameter D and rope diameter d) in order to still achieve an adequate service life. By comparison, the selected HM-HT fiber rope is showing exceptionally high potential in terms of the payload, calculated in accordance with SABS and TAS, which is increasing despite a concurrent increase in rope mass with increased hoisting depth L. This is due to the fact that the safety factors decrease more markedly with rising hoisting depth than the rope weight can increase over the same length. In this way, the payload is higher (factor 3), when using the selected fiber rope and applying safety factors in accordance with SABS 0294, already at a conveying depth of L= 3000 m.

Despite their many advantages compared to steel wire ropes, fiber ropes are still no common mechanical component for mechanical engineering. This is due to the fact that running fiber ropes and their failure mechanisms are not as well investigated as this is the case for steel wire ropes and round steel chains (Figure 3). Especially discard criteria for fiber ropes are not too well investigated, but this is an important factor to be known, when designing tension members for safety-related applications like elevators, cranes and hoisting gears, where a break of a tension member can lead to heavy machinery damage and personal injury.

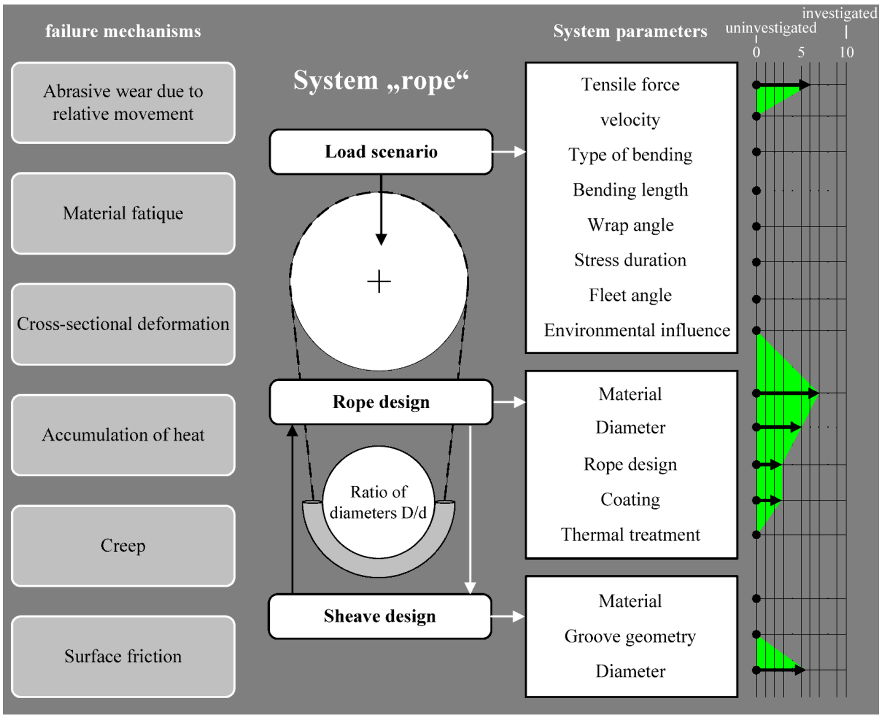


Figure 3: Failure mechanisms and state of investigations on fiber ropes [3]

Due to the facts named above, the Endowed Professorship of Technical Textiles & Textile Mechanical Components at Technische Universität Chemnitz is working on gaining knowledge on the service life of fiber ropes in technical applications.

For many technical, winch-based applications, like cranes and hoisting gears, but also for elevators, it is important to exactly know about the performance of fiber ropes when running over sheaves and/or being wound on drums. Due to the bad state of investigation on fiber ropes, there are no valid international standards for application-related tests, like cyclic bend over sheave (CBOS) or winding/spooling tests for fiber ropes, nor are there any internationally valid standards that are describing design and accuracy of testing machinery for performing such tests on fiber ropes.

The guideline “VDI 2358 – wire ropes for materials handling equipment” has been established by the Association of German Engineers (short form: VDI). This guideline is, beside other topics, giving recommendations on how to set up the testing sheaves for testing the performance of steel wire ropes in cyclic single bending and cyclic reverse bending tests, and the groove geometry to preferably use for such tests ([8]). This guideline is authoritative in Germany and is mostly taken ‘as if authoritative’ in German-speaking countries, though it is still not regulating how to apply the rope load during the CBOS tests and it is also not specifying appropriate testing parameters.

This report is addressed to operators and professionals in the field of hoisting applications. It is giving samples of machinery setups that can be used for performing CBOS test and winding/spooling tests on fiber ropes, showing some test parameters and results, and some influences on the testing results, which have to be taken into account.

2 EXPERIMENTATION

Before starting the investigations on the behavior of fiber ropes under cyclic bending and in winch-based applications, machinery for performing CBOS tests (single bending and reverse bending) on fiber ropes and a test bench for winding an spooling ropes on rope drums have been established. For designing the machinery for CBOS test, a research on the state of the art has been done, based on guideline VDI 2358 and already existing setups of machinery for performing CBOS tests on steel wire ropes and fiber ropes. Further, testing machinery for simulating the behavior of fiber ropes wound on winch drums has been established.

2. 1 CBOS TESTS

As shown in guideline VDI 2358, the recommended operation principle for performing CBOS tests is the principle of "active rope and passive sheave". This means that the rope is driven by a deflection sheave and is guided over one test sheave for cyclic single bending (
Figure 4). For cyclic reverse bending, the rope is to be guided over several test sheaves, while none of these test sheaves is driving the rope.

Caused by the fact that guideline VDI 2358 is not specifying how to load the rope (e.g. hydraulics system or weights or spindle drive), a research in literature and publications has been performed to see, how this has been managed in existing systems and to estimate advantages and disadvantages of these systems.

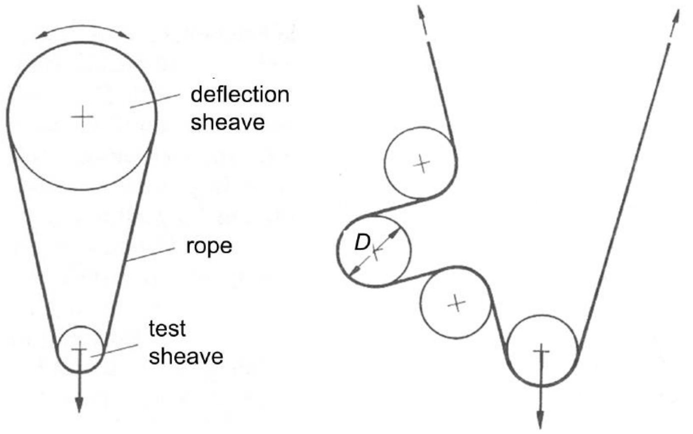


Figure 4: Test setup for CBOS tests [8]

2.1.1 MACHINERY FOR CYCLIC SINGLE BENDING

For designing the equipment for the cyclic single bending tests on fiber ropes, the system of "active rope and passive sheave" has been overtaken. Regarding the fact that a range of the D/d-ratio between $12.5 \leq D/d \leq 63$ is named to be an appropriate range for testing of steel wire ropes ([2], [4]), it was necessary to create machines for special ranges of rope diameters.

The smaller CBOS test machine, shown in Fig. 5, is appropriate for testing ropes with a diameter up to 8 mm under cyclic single bending. The maximum rope load is about 12.5 kN and is applied by a spindle drive. The test sheaves can be exchanged, so the full range of D/d-ratios can be tested. The drive sheave is driven by a servo motor, what enables to perform 25 full bending cycles per minute and a continuously adjusted stroke length. A contact-free measurement of the temperature at the rope's surface during the CBOS test is included.

Figure 5: CBOS test machine 12.5 kN (left) and 90 kN (right).

The bigger CBOS test machine, also shown in Figure 5, is designed for testing ropes with a diameter up to 16 mm under cyclic single bending. The maximum rope load is about 90 kN and is also applied by a spindle drive. The test sheaves can be exchanged for testing various D/d-ratios. The drive sheave is driven by a servo motor, what enables to perform 30 full bending cycles per minute, in this case. Contact-free measurement of the temperature of the rope's surface during CBOS test is included, too.

As know from previously performed tests concerning drivability and friction behavior of fiber ropes on drive sheaves for elevators ([6]), the groove geometry and the flank angle of the groove do have an not negligible influence on the rope's service life. For accurately controlling the bending zone at the CBOS machines without fixing the rope to the drive sheave, an adhesive metallic tape is fixed on the rope's surface. An electromagnetic sensor is taking the position of the metallic tape during CBOS process and the control unit is, if needed, adjusting the rope's position, which helps to avoid errors in determining the rope's lifetime, which might occur when the rope is slipping on the sheaves. In addition, the CBOS test can be interrupt for detaching the test rope and having samples for other tests such as recording the stress-strain behavior over the number of bending cycles. Main advantages of these machines are: freely programmable load characteristics, continuous, constant loading (control unit with load cell), continuously adjusted stroke length, accurate surveillance of the bending zone and real-time monitoring of rope's surface temperature.

2.1.2 MACHINERY FOR CYCLIC REVERSE BENDING

For designing the machinery for cyclic reverse bending, the operating principle of 'active rope and passive sheave' has been disregarded and the opposite, a system with passive rope and active sheave, has been created, as shown in Figure 6.

Figure 6: Test machine for cyclic reverse bending.

As to be seen in Figure 6, the rope is fixed at the bottom and at the top with a splice, while the top fixing point is vertically movable to apply the rope load. The bending cycles are applied via a vertically oriented and vertically movable, chain driven machine slide. The deflection sheaves and test sheave are mounted on this machine slide. The position of the test sheaves or other test set-ups can be adjusted and varied to generate flexible set-ups beside CBOS tests, as shown in Figure 7. This means that the machine enables various set-ups, e.g. for performing CBOS tests with single and reverse bending, but also for creep tests and abrasion tests. The operating principle has a lot of advantages, like:

- almost constant and directly measurable rope force

- small drive units for force transmission (until now: FS ≈ FP/2 ; now: FS = FP)
- no need for adjustment of bending zone
- constant wrap angles on the sheaves
- recording of rope damage

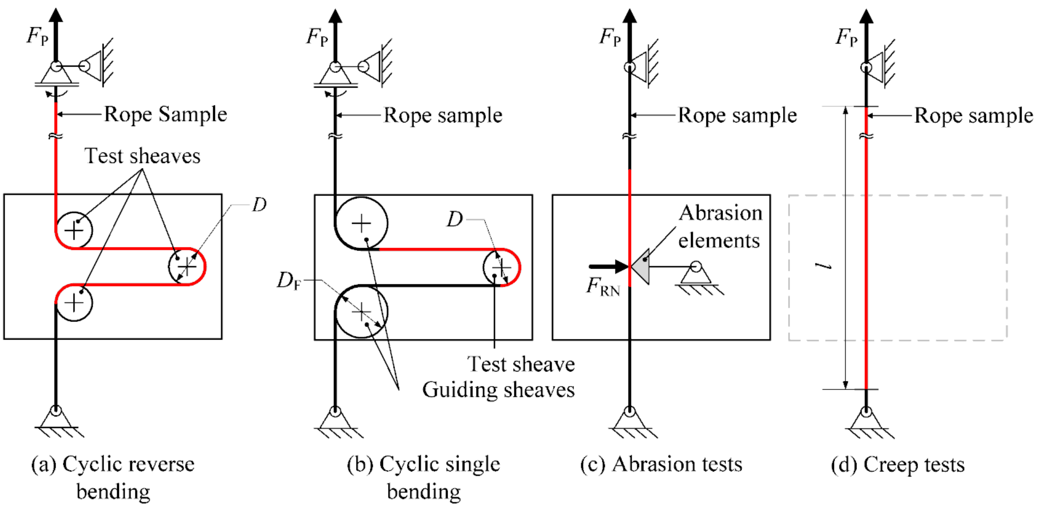


Figure 7: Variety of test set-ups [4]

In
Figure 8, a lot of CBOS test set-ups are shown, for investigating several issues, e.g. the influence of varying wrap angles, CBOS tests with tensile load varying during the test, for comparison of Woehler- and fatigue curves, and CBOS tests with introduced rope twist for simulation the rope's behavior e.g. in a shaft hoisting system.

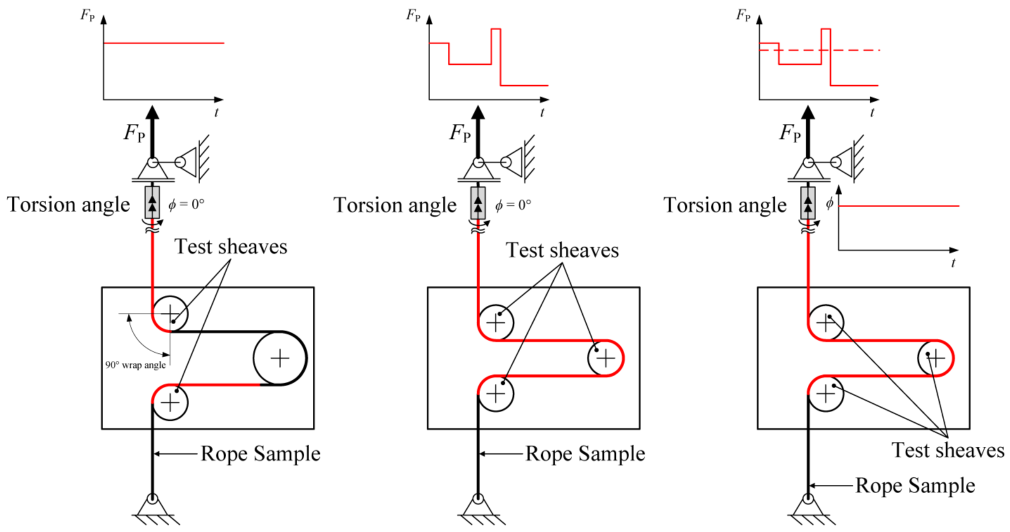


Figure 8: Variety of CBOS test issues.

2.2 MACHINERY FOR WINDING/SPOOLING TESTS

As named in chapter 2, a test bench for simulation the behavior of fiber ropes wound on rope drums in multiple layers has been established. A system was created, where the rope is wound from one rope drum onto another rope drum. The rope is guided over a deflection sheave and loaded with a certain load, as shown in
Figure 9.

Figure 9: Test machine for winding tests.

The rope load is applied by a chain drive, which is moving the deflection sheave vertically. The engine control is realizing the compensation of the varying length of the rope layers by permanently controlling the rope load and re-stressing the rope. In this way, the test setup allows to wind/spool large rope lengths without using a high test tower. The two rope drums, with a diameter of the drum shell of 205 mm, are driven by one chain drive with one driveshaft, to make sure that they have the same rotational speed. The diameter of the deflection sheave is 630 mm (D/d = 40), to make sure that the main damage of the rope is caused by the process of winding and unwinding on the rope drums and not by the bending cycles initiated by the deflection sheave ([4]). The maximum rope load is 60 kN, the maximum rotational speed of the rope drums is 35/min, what results in a maximum rope speed of about 55 m/min, depending on the number of layers, the rope diameter and the number of the actually spooled layer.

3 TEST RESULTS

3.1 RESULTS OF CBOS TESTS – SINGLE BENDING

Within the tribological system 'fiber rope', internal and external interactions occur. Internal interactions are due to the fibers, their arrangement in the rope and their changes in length due to the test set-up ([4]). Relative movements are predominantly desirable for internal contact between fibers. This is the reason, why low coefficients of friction are required for the internal interactions. External interactions occur between rope and drum and between the stacked rope layers. In these cases, relative movements are predominantly undesirable, what is resulting in a requirement for high coefficients of friction. As a result of the facts named above, is a conflict between the requirements for internal interactions and external interactions. For lifting and hoisting applications, a favorable rope design is, to separate the rope into two components with different main functions, as shown in Figure 10. The first component is a load bearing braided rope core, consisting high-modulus (HM), high-tenacity (HT) polymer fibers, and the second component is a supporting braided cover, consisting of polymer fibers, which do not implicitly need to be of HM-HT fibers.

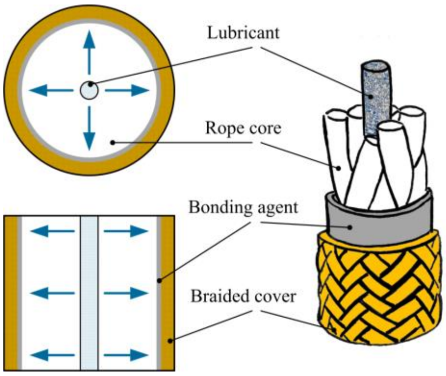


Figure 10: Tension member for hoisting applications, based on high-strength synthetic fibers.

In this case, the functions bearing the load, low coefficient of friction and flexure can be attributed to the rope core, while the functions of dimensional stability and drivability are to be attributed to the rope cover.

This separation of functions enables an optimal adjustment of lubrication of the core on one hand, and a high coefficient of friction between rope surface and drive sheave or rope termination, on the other hand. Additionally, the cover is helping to protect the rope from environmental influences, e.g. dust, UV radiation and water, and ensures the necessary dimensional stability and compensates the relative movement, and thereby the force transmission, of the load bearing strands of the rope core. A monolithic design of the rope core further helps to avoid lubricant emerging from the rope core.

In general, a self-stabilization of the rope core is only possible with hard coating polymers (e.g. polyurethane with Shore 70A or more). Theses coatings are suppressing kinking effects in the fibers and are establishing a wear-resistant protection layer. But such coatings do mostly provide high coefficients of friction, what leads to the effect that they are working as an abrasive in the rope, when the protection layer is damaged. In previously performed investigations ([4]), it has been shown that the fiber finished do mostly have a positive influence on the service life of fiber ropes. Therefore, the separation of functions (as shown above) has been developed under the aspect of increasing the service life of the ropes with fluid coatings (lubricants). Such fluid coatings can be oils and waxes. Oils do show two disadvantages. The can penetrate the rope cover, due to capillary actions, and thereby reduce the traction, e.g. between rope cover and drive sheave. Further, the compression of the rope can lead to oil emerging out of the rope and thereby cause a lack of lubricant in the rope core, what might lead to a shorter service life of the rope. These effects are narrowed when using waxes. Due to this, a wax with added polytetrafluorethylene has been investigated, in pilot CBOS tests. As shown in

Figure 11, the number of bending cycles to break increased by a factor 3.7 in comparison to the same Technora® rope without the wax, but with just the fiber finish T221.

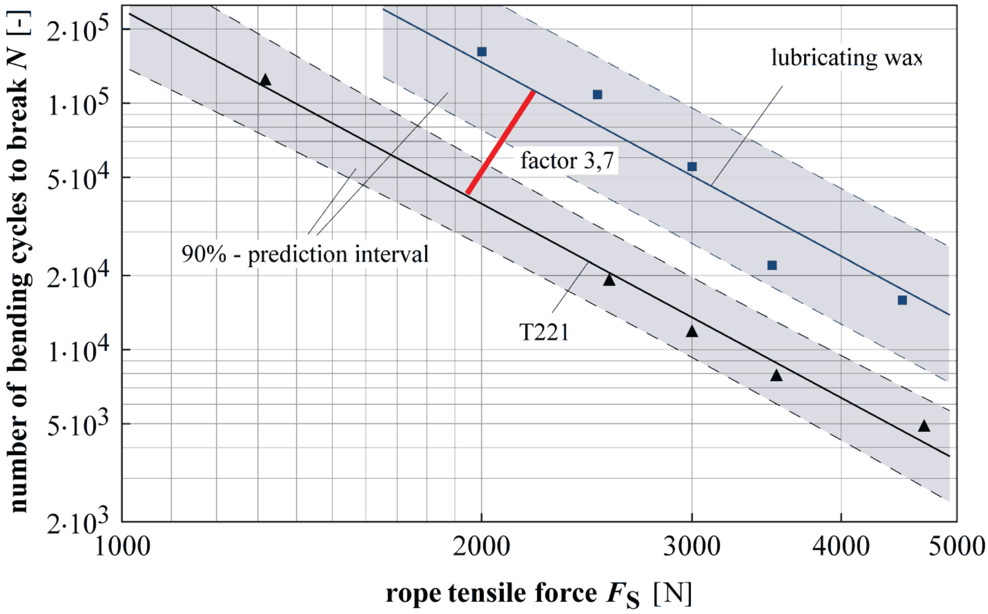


Figure 11: Endurance test (CBOS) with lubricating wax (D/d = 12.5; groove radius r ≈ 2.2 dN).

Due to the fact that the rope seems to be dimensionally very stable when covered with a tight braided cover, it is assumed that the cover is inhibiting kinks in the HM-HT fibers and is reducing the flank pressure. Proof of the positive influence of braided covers has been made based on a 32-strand braided cover, made from polyester fibers. For proving this thesis core ropes with and without cover have been investigated in single bending CBOS tests. In

Figure 12, it is shown that service lifetime has been increased by a factor 2, just by applying the braided cover on a rope made from Technora® T221.

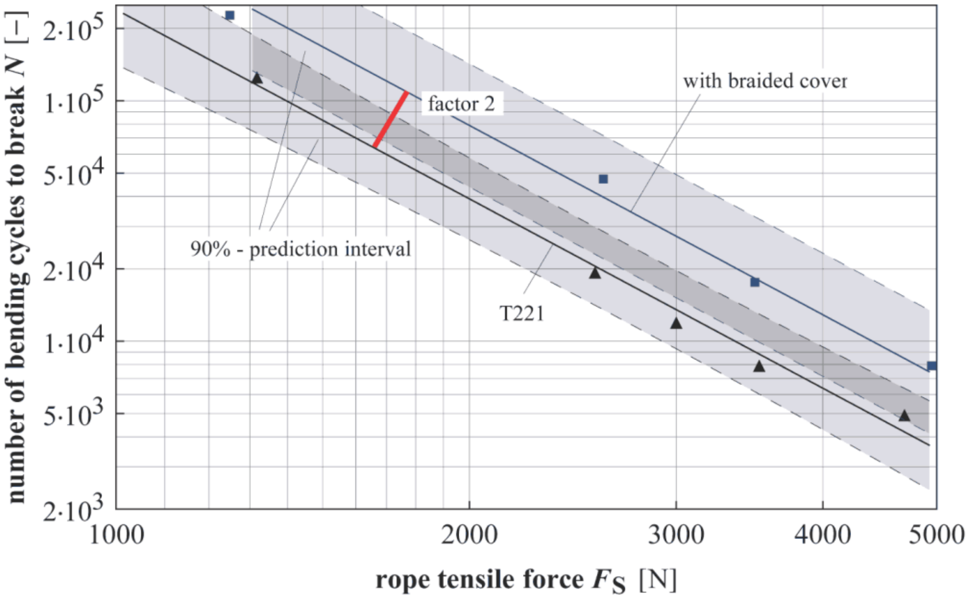


Figure 12: Endurance test (CBOS); core – cover design (D/d = 12.5; groove radius r ≈ 2.2 dN).

Within these tests, it has been found that the selected cover shows a high abrasion resistance. It is assumed that the cover can bear much more bending cycles, what lead to the further assumption that the function of the cover is ensured for the whole service life of the covered rope structure. The much higher elasticity of the polyester fibers in addition with the transverse pressure, initiated by the short braid pitch of the cover, result in a good fitting of the cover onto the shape of the rope core. This leads to a tight fit and an optimal support of the rope's shape, even in the bending zone.

3.2 RESULTS OF WINDING TESTS

For testing the rope performance of ropes on a rope drum, ropes with a diameter of 14 mm, made from DSM's ultra-high molecular weight polyethylene fiber Dyneema® SK75, have been used. The ropes have been loaded at about 15% of their mean break load (15% MBL) and have been wound at a rope speed of about 10 m/min (13 cycles per minute) on non-grooved rope drums, as shown in

Figure 13.

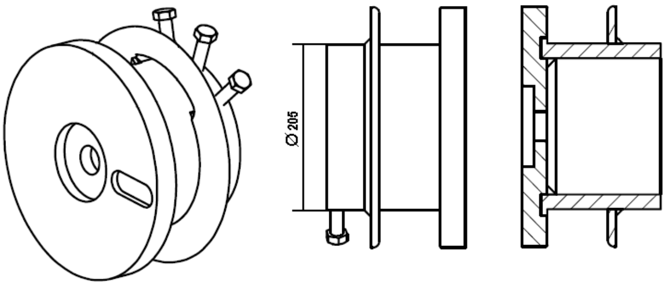


Figure 13: Rope drum used for the winding tests.

On the left-sided rope drum (Figure 13), three layers of rope have been mounted; on the right-sided rope drum, one layer has been mounted. The two top layers of rope have been moved from left side to right side in reversing operation, what means that one full cycle is simulating two times of winding onto the drum and two times of unwinding from the drum. The ropes have been cycled until break and the remaining performance of the base layers, which have not been wound, has been determined in tensile tests and compared to the rope's state of delivery, to investigate influence of the rope stack on the properties of the base layers.

It was found that the rope was always failing in one of the wound layers, never in the base layer, after range of number of cycles from 3817 to 4532 cycles. The break load of the base layer ropes is lower, though these layers have not been coiled and uncoiled, during the test. In Table 2, the break load of the base layers is shown in comparison to the rope in state of delivery.

Rope sample	Mean Break Load [kN]	Remaining tensile Strength [%]
State of delivery	177	100
After winding test	126 - 134	71 - 75

Table 2: Break load of base layers of fiber ropes after winding test

As can be seen, all the rope samples show a lower break load of about 70% to 75%, compared to state of delivery. Having a look on the rope segments of the base layer, it was found that the lost there mostly round shape and have been deformed.

Figure 14 is showing a sample of rope, taken out of the base layer after having performed the winding cycles.

Figure 14: Deformation of base layer rope.

Figure 15: Abrasive damage at base layer (left) and upper layer (right).

As can be seen in

Figure 15, visible abrasion occurs on the rope surface. This is due to slippage between the layers, when a layer is coiled or uncoiled. Comparing this abrasive damage to a rope sample taken out of a coiled and uncoiled layer (

Figure 15), which still is at 50% of the break load of the rope in state of delivery, it is assumed that abrasive damage is not the main reason for the low remaining break strength. So, the loss in tensile strength is mostly caused by the compressive forces that are applied to the base layer by the top layers.

4 CONCLUSIONS

The most properties of running fiber ropes can be investigated with the test equipment, presented in this paper. Among other things, the advantage to separate a fiber rope's functions into several rope layers has been verified with tests, performed on these test machines. It was shown, that the number of bending cycles to break is higher by a factor 3.7 in the rope with a core conditioned with lubricating wax, compared to a rope without the wax. Furthermore, it was shown that a braided cover is reducing any kink effects in the pressure-sensitive HM-HT-fibers, what has been verified by an increased service life (factor of around 2).

For winding tests of fiber ropes, it can be concluded that winding on non-grooved rope drums is not appropriate for fiber ropes. Due to the compressive forces of the rope stack (stack wound at a high rope load), the base layers are showing a high, non-reversible deformation, resulting in a loss of break strength, though these rope segments have never been spooled. Further tests are needed to investigate how this behavior can be compensated with different grooves. First tests have shown that a groove geometry as used for steel wire ropes according to DIN 15061 (r = 0.525 dN) is not helping to improve. It is assumed that special groove geometry has to be designed.

Figure 16 is showing an actual approach of rope drum, which is meant to better support rope's geometry in future winding and spooling tests.

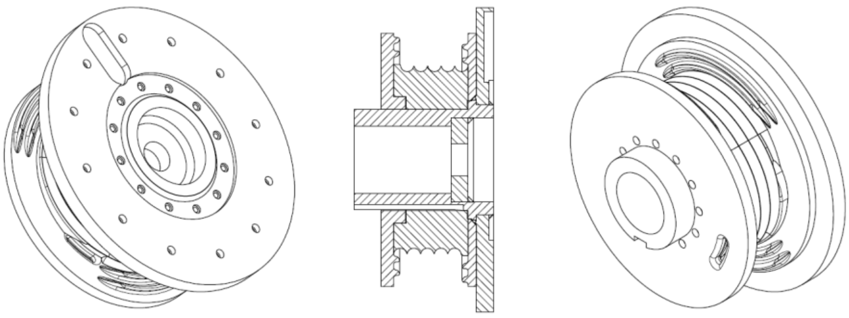


Figure 16: New design of rope drum and groove.

REFERENCES

[1] W.A.J. Albert, Die Anfertigung von Treibseilen aus geflochtenem Eisendraht. Bergbau und Hüttenkunde. Vol. 8. Berlin/Germany: Archiv für Mineralogie, Geognosie, Bergbau und Hüttenkunde, 1835.

[2] K. Feyrer, Drahtseile. Bemessung, Betrieb, Sicherheit. Berlin/Germany, Springer-Verlag GmbH, 2000.

[3] T. Heinze, Dimensionieren je nach Einsatzfall. Hebezeuge Fördermittel 6/2011: Pages 366-369.

[4] T. Heinze, Doctoral thesis. Zug- und biegewechselbeanspruchte Seilgeflechte aus hochfesten Polymerfasern. Technische Universität Chemnitz, Chemnitz/Germany. 05 Nov 2013. <https://katalog.bibliothek.tu-chemnitz.de/Record/0008988209>

[5] H.A. McKenna, J.W.S. Hearle and N. O'Hear, Handbook of fiber rope technology. Cambridge/UK, Woodhead Publishing Limited, 2004.

[6] M. Michael, Doctoral thesis. Beitrag zur Treibfähigkeit von hochfesten synthetischen Faserseilen. Technische Universität Chemnitz, Chemnitz/Germany. 30 Mar 2011. <https://katalog.bibliothek.tu-chemnitz.de/Record/0001147716>

[7] Siemag M-Tec, Blair-Doppeltrommel-Fördermaschine für South Gold Mines. Techreport Siemag, 2006.

[8] VDI 2358, Wire ropes for materials-handling equipment Verein Deutscher Ingenieure e.V., Düsseldorf/Germany, 2012, www.vdi.de.