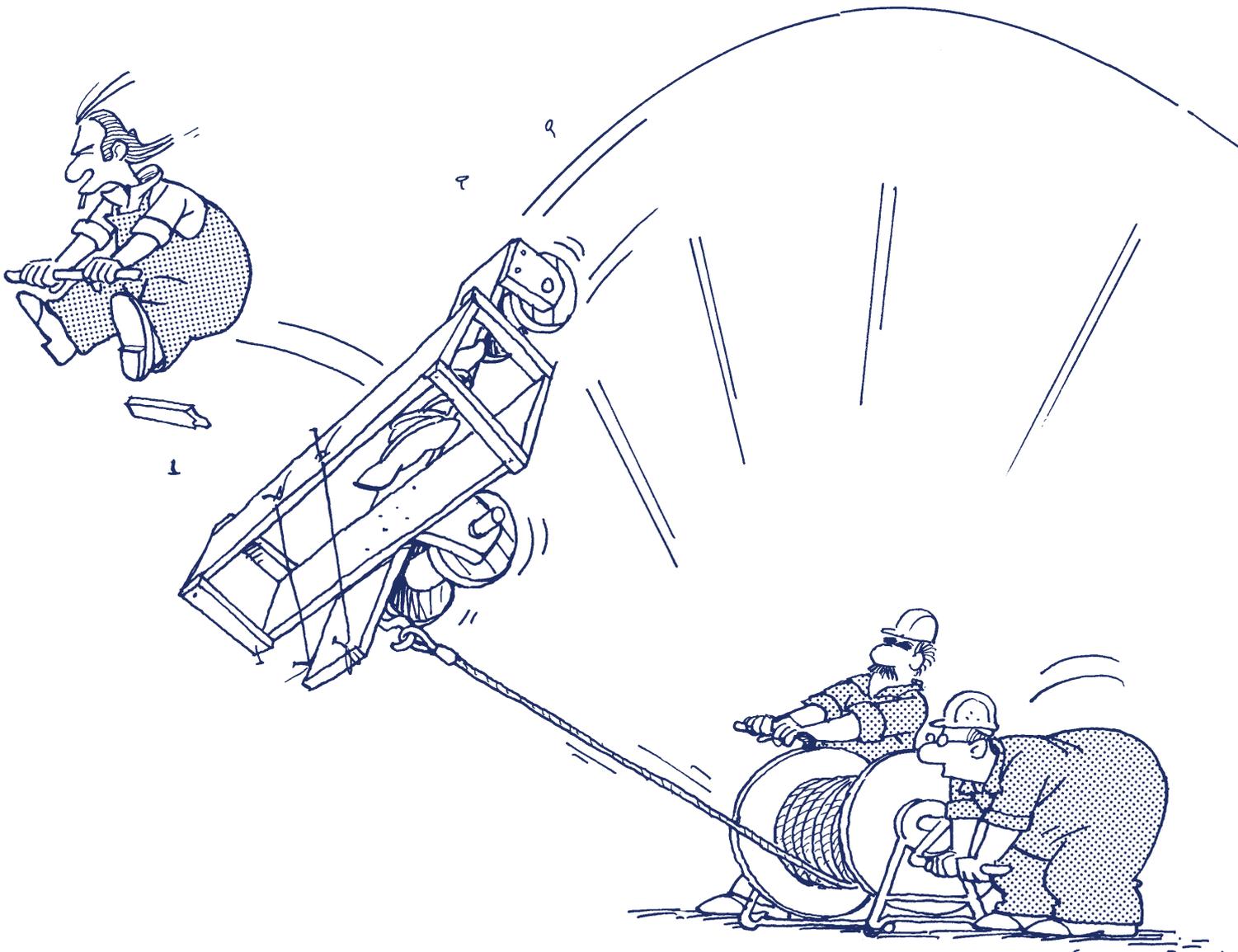


WIRE ROPE

TECHNOLOGY AACHEN



Ropes and reeving systems
The last 40 years

Ropes and reeving systems

The last 40 years

by Dipl.-Ing. Roland Verreet

Content

1. Ropes	5
2. Sheaves.....	16
3. Rope testing	25
4. Drums	33
5. Inspection.....	41
6. End connections.....	47
7. Conclusion.....	52

© 2015, 2018 Ingenieurbüro für Drahtseiltechnik Wire Rope Technology Aachen GmbH

Cartoons: Rolf Bunse

Layout and typesetting: Benedikt Dolzer, Aachen

Reproduction, in whole or in part, only with written permission of the author.

I was born in June 1950, and in June 2015 I reached my discard state, as rope engineers would call it. But I did not retire. The crane and the rope business are too interesting.

I started school at the age of 5, one year earlier than normal. The effect of that was that wherever I went, I was always the youngest. I was the youngest in school, I was the youngest at college and then I was the youngest at the university.

And when I started as an engineer in the wire rope industry, I was the youngest again.

You get used to a situation like that. But one day you look around and you find that something has changed: your former colleagues have all retired and now you are the oldest. That is shocking!

Then you start to realize that all these youngsters around you have no idea how their industry looked like 40 years ago. Many of them were not even born at that time.

When I was asked to give a presentation at the 21st International Offshore Crane & Lifting Conference I decided to present a very personal view of the last 40 years of the development of ropes and reeving systems in which I have always taken an active part. So please excuse me if I present the subject from my own, very subjective point of view.

1. Ropes

1.1. Computer assisted rope design

Today we take it for granted that sophisticated computers are used to design steel wire ropes. These computers did not exist when I started in the wire rope industry as a young engineer in 1975. At that time steel wire ropes were designed using empirical factors for strand diameters, wire diameters and for rope and strand lay lengths. These factors were found by trial and error, and the customers were the guinea pigs.

The cross section of the center wire of a strand is a circle, but the cross section of a wire wrapped around this center wire is neither a circle nor (as still many people believe) an ellipse but a kidney-shaped section. Fig. 1 shows the circular wire and the kidney-shape of a wire wrapped around it with different lay lengths.

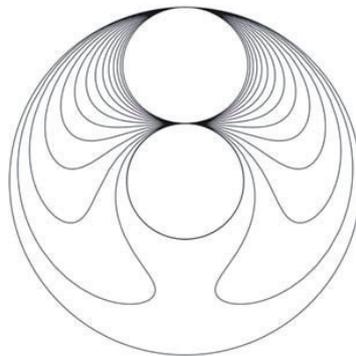


Fig. 1

In 1974, still studying engineering in Aachen, I started designing strands using the revolutionary HP 65 pocket computer which allowed to program 100 key commands (Fig. 2). I could type in an x-value, run a magnetic card through the machine, and after a few seconds the computer would give me the corresponding y-result. This way within a few minutes I could design the true cross section of a wire in the strand.



Fig. 2

The first Commodore PET was introduced in 1977, and it had a stunning RAM of 4K (Fig. 3). A typical Laptop today would have 250.000 to 4 Mio. times this capacity.

I used a Commodore PET to design and even plot the first strands by computer. It was still a time-consuming process, but it allowed for the first time to calculate the complete true cross section of a strand.



Fig. 3

When my older son started school his teacher asked me if I was an artist. I told her: “No, I am an engineer.” She was surprised: “Your son told me you sit in your office all day long and draw flowers.”

I quickly found out where the misunderstanding came from: The first plotters were not only extremely expensive, they were also very loud and a pen was moving at high speed in one direction while the paper was moving underneath it in another. The plotter was working all day long, and it made a lasting impression on my children. When I did not need the printouts any more, I gave them to my children who then coloured and finished the “flowers” (Fig. 4).

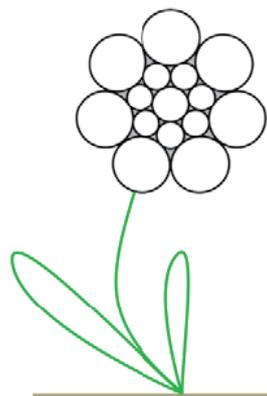


Fig. 4

When recalculating existing rope designs I found a great number of mistakes in the designs, and I established a set of design rules which are still valid 40 years later.

Recently I inspected a hoist rope of a reactor crane in a nuclear power plant. The rope was in service since 1969. I was not worried by the age of the wire rope, I was worried by the fact that this rope (which was working in a safety critical application) dated from a time when ropes were not yet designed using computers.

1.2. Ropes with plastic infill

In 1834 wire rope was invented by Oberbergrat Albert. His design avoided wire crossovers within the rope. The need for higher breaking strengths, however, quickly led to the development of strands with several layers of wires, introducing dangerous wire crossovers within the strands. This problem again was solved by Tom Seale who invented the parallel lay strand in 1884.

But the industry asked for even higher breaking strengths, and for many applications the fibre core was replaced by an independent wire rope core (IWRC, Fig. 5).

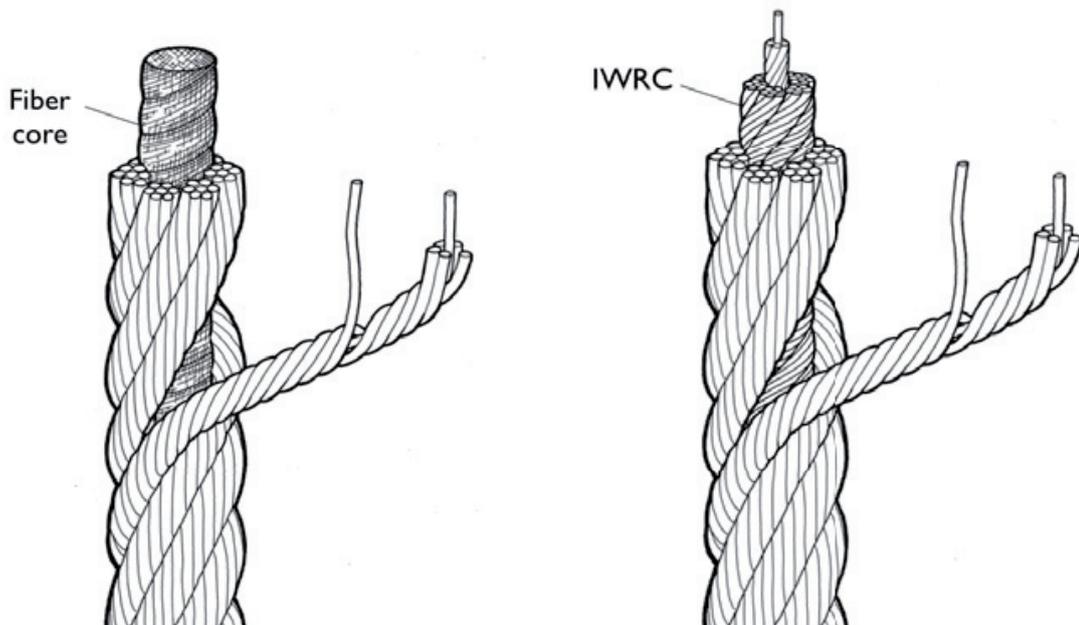
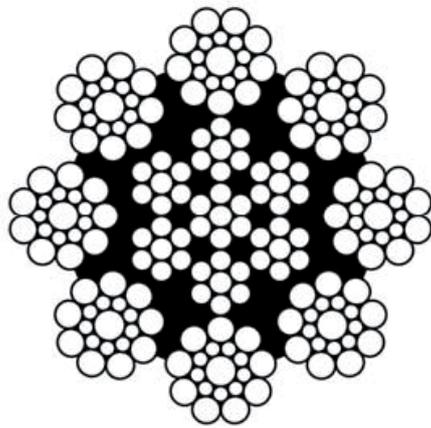
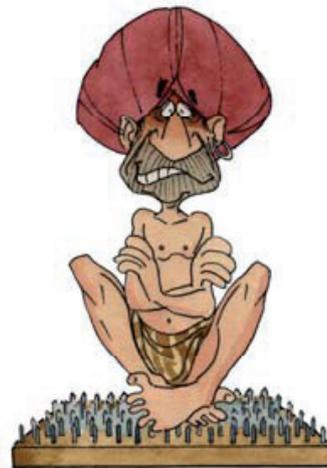


Fig. 5

This, however, introduced new problems: While before the outer wires of the outer strands were lying on a soft center element, they were now crossing over the outer wires of the IWRC, leading to internal wire breaks which could not be detected during a visual inspection (Fig. 6).



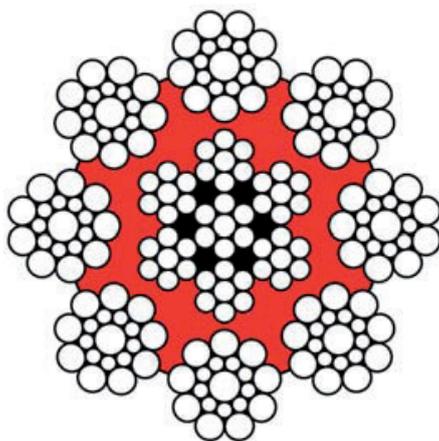
Without plastic infill



Crossovers

Fig. 6

In 1975 the company I worked for developed and patented the plastic infill of steel wire ropes. A plastic coating now protected the IWRC against loss of lubricant and ingress of moisture, but most importantly it avoided the crossovers inside the rope and provided a soft cushion for the outer strands (Fig. 7).



With plastic infill



No Crossovers

Fig. 7

1.3. 8 strand ropes versus 6 strand ropes

40 years ago 8 strand ropes were made by a few specialists only. 6 strand ropes were dominating.

Bending fatigue tests, however, e.g. at the University of Stuttgart, showed that the fatigue life of 8 strand ropes was considerably higher than that of 6 strand ropes. The main reason for this is the greater number of contact points and the greater contact area between the outer strands of the rope and the sheave, leading to much lower bearing pressure (Fig. 8).

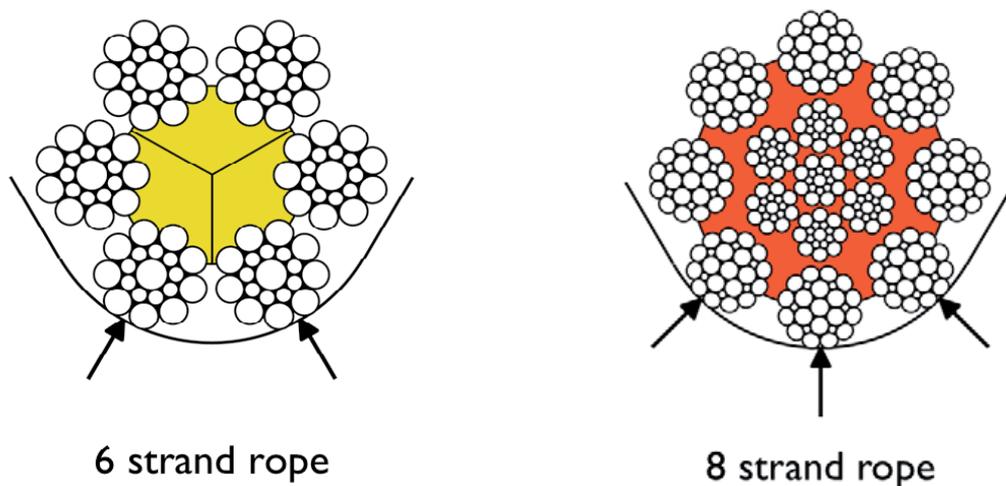


Fig. 8

I could watch the same scenario many times: the salesman of a rope company would argue against 8 strand ropes for years, and then suddenly he would start to praise them. Then I knew that his company had finally bought an 8 bobin closer.

1.4. Ropes with compacted strands

The crane industry was hungry for higher and higher breaking strengths: when using a stronger rope crane designers could reduce the rope diameter while still maintaining the same design factor.

And because the minimum drum and sheave diameters were defined as a multiple of the rope diameter (the so-called D/d -ratio), a smaller rope diameter led to smaller sheave and drum diameters. Smaller drum diameters in turn led to lower torque requirements, and therefore to smaller motors and gearboxes. So reducing the rope diameter by 1mm could reduce the overall costs of a crane trendously.

In 1975 compacted strands were used to prestress concrete, and I wondered why nobody made ropes out of these strands. I was told that because of the higher steel content these strands were very stiff, and that the rope made out of such strands would not be flexible enough. I was not convinced, and I made a first rope out of compacted strands. It turned out to be surprisingly flexible.



Fig. 9

When a rope gets bent, its strands must move relative to each other. We found that because of their smooth surfaces (Fig. 9) compacted strands would move much more easily than conventional strands.

We continued developing and testing the new technology, and as early as 1978 we launched the first series of ropes made out of compacted strands (Fig. 10).

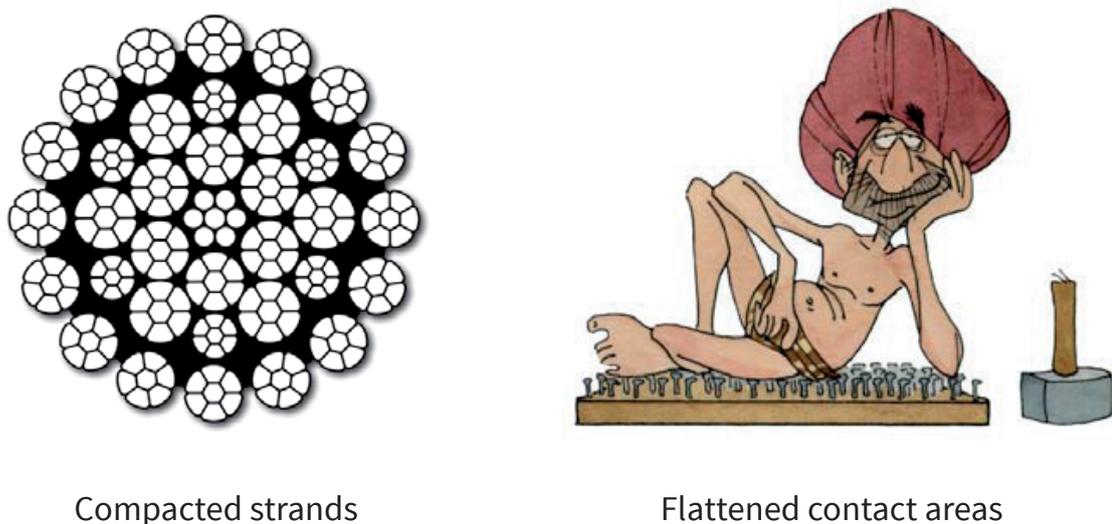


Fig. 10

Not only did these ropes provide much higher breaking strengths, they also had much better contact conditions against the sheave and drum grooves (Fig. 11) and within the ropes.

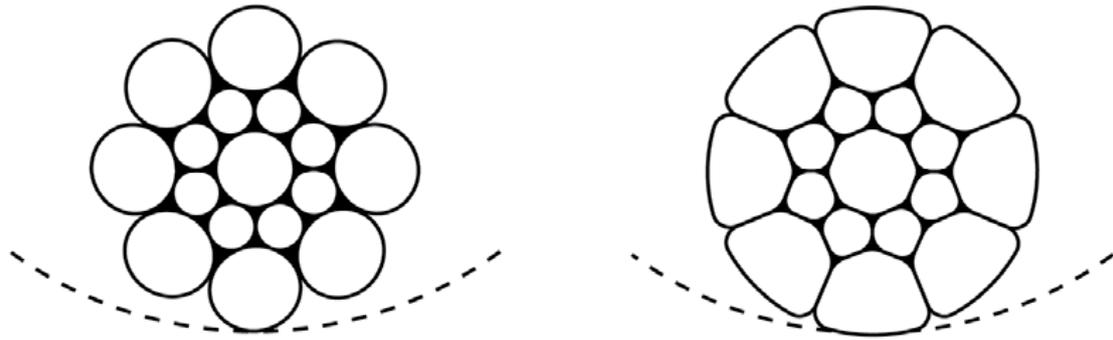
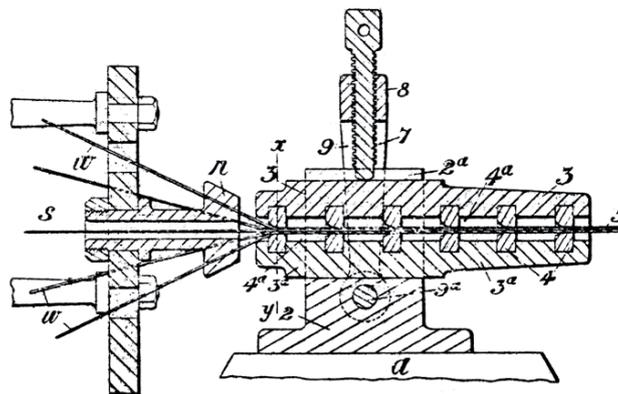


Fig. 11

The new ropes were very successful on the market, but the manufacturer of the compacted prestressing strands claimed he had a patent on the compacting process from the late 1950s which was still valid. He admitted he had never thought of making ropes out of these strands, but now he would start doing so. And he forbade us to continue to violate the patent by producing compacted strands.

Fortunately while working on the process I had found a patent predating theirs by almost 70 years (!), and it was obvious that I could have brought down their patent in a lawsuit easily (Fig. 12).



Charles John Banks 1891

Fig. 12

So we made an agreement that our two companies would now make ropes out of compacted strands, and they sued everybody else who violated their (invalid) patent. This strategy worked for quite a number of years.

Another major step in the evolution of steel wire rope was the development of swaged ropes with improved multi layer spooling performance and increased radial stiffness (Fig. 13).

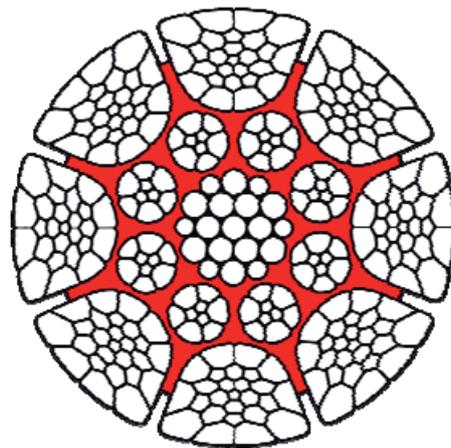


Fig. 13

With increasing rope length in deep sea or mining applications the influence of the self weight of the rope becomes more and more important. This opened the doors for lighter ropes made of high strength fibres. But no discard criteria were available for these ropes.

Together with a team of engineers from the wire rope and fibre industry I developed a hybrid rope in which the IWRC was replaced by a high strength fibre rope core (Fig. 14).



Fig. 14

The problem of the different elasticities of the materials was used by pretensioning the fibres. For these ropes, the discard criteria of steel wire ropes apply.

1.5. Ropes with variable lay lengths

Fig. 15 shows a rope application with a great height difference. This could be a deep shaft mine or an offshore application. The rope section at the bottom carries the payload (10t), and rope section at the top carries the payload (10t) plus the weight of the rope itself (another 10t).

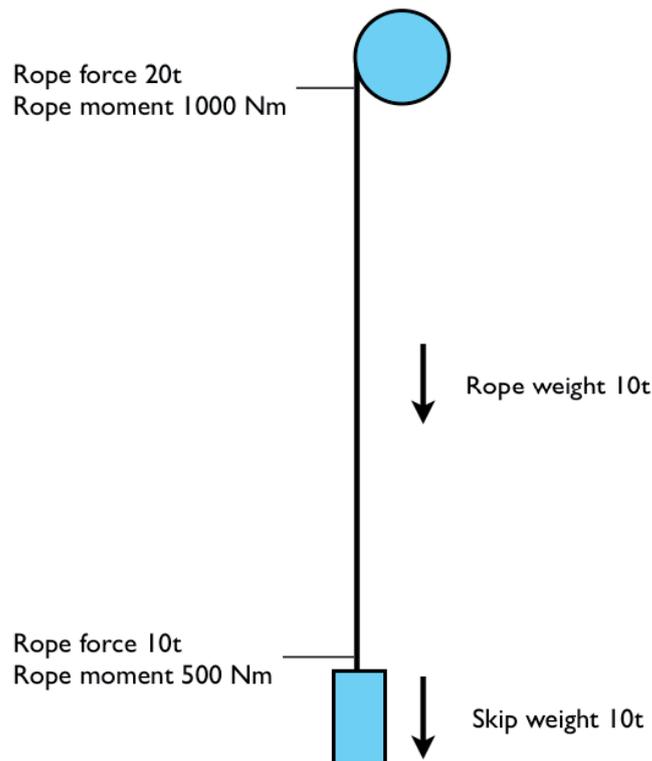


Fig. 15

This means that at the top the rope is subjected to twice the rope force it is subjected to at the bottom. If the rope is not rotation resistant, it will therefore develop a moment at the top about twice as big as at the bottom. As a consequence, the rope will change its lay length: It will open up at the top, lengthening the rope lay, and it will close at the bottom, shortening the lay.

From then on it will operate in a severely twisted condition and it will only achieve a very poor fatigue life.

But operating with twisted ropes is also dangerous: as soon as the twisted rope gets slack (e.g. because the payload touches the sea bed), it will immediately form hockles.

In deep shaft mines this is a matter of life and death: If the skip of a deep shaft mine travels upwards and the rope suddenly stops due to an emergency break, the skip will continue to move upwards due to its inertia and it will “overtake” the rope. The rope above the conveyance will immediately get slack and form a kink. Shortly afterwards the skip will fall back, and the kinked rope is likely to break. The skip will then fall down a few hundred meters. Many miners have lost their lives in such accidents .

30 years ago I sat in an airplane and thought about how to overcome these problems. It occurred to me that when you install a rope with a great height difference it will rotate around its own axis until it has an equilibrium of moments in every position along the rope length. So why not produce the rope with this continuously changing lay length?

If you install such a rope it will not have to rotate in order to come to an equilibrium of moments: it already has the correct shape!

I patented the idea, and then I tried to persuade both rope makers and users to try the concept. With no success. Finally I talked about the concept at a conference in the UK, and fortunately a famous Ukrainian wire rope engineer was in the audience. He immediately understood the implications of the idea, and he went home, manufactured ropes with variable lay lengths and put them into service on drum winders. The ropes performed much better than any design they had been used before.

This was the breakthrough, and today ropes with variable lay lengths gain an increasing market share, especially in shafts deeper than 1500m.

Ropes with variable lay lengths are not yet used in the offshore industry, most offshore engineers might not even have heard of them. But I am convinced that this will change in the next few years.

1.6. Dimensions

Over the last 40 years, rope lengths, rope diameters and total rope weights have gone up tremendously. 50 mm was considered a large rope diameter, and rope weights were limited to about 50t. Today rope diameters of more than 120mm are not uncommon. The world record at the moment is a rope with a diameter of 175mm produced in Malaysia (Fig. 16).

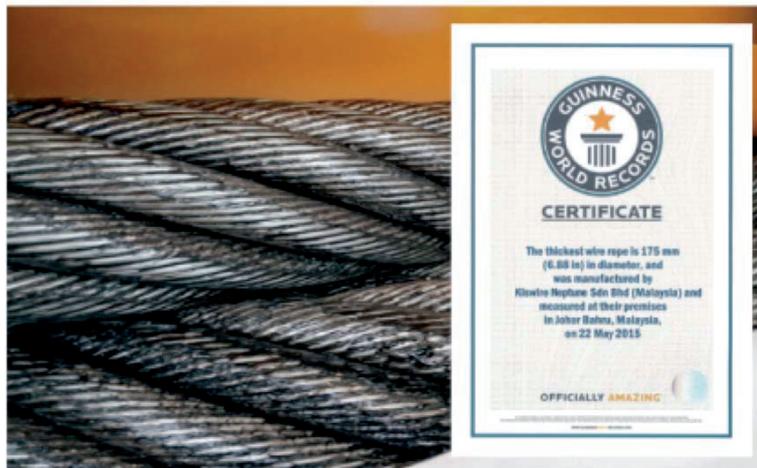


Fig. 16

Crane capacities have grown, too.

In 1975, when I started to work in the wire rope industry, Krupp built a 900t portal crane, the largest of its time. Today the lifting capacities of the largest onshore cranes are around 4000t, and the capacities of offshore cranes are even much higher.

Fig. 17 shows an illustration of the “Pioneering Spirit” owned by Allseas. The vessel has a topside lift capacity of 48.000t (using hydraulic rams) and a jacket lift capacity of 25.000t (using steel wire ropes).



Fig. 17 (Source: Allseas, Internet)

Fig. 18 shows two of the four A&R ropes for the Pioneering spirit. With unit weights of more than 400t each these ropes were the heaviest ever produced in one length.



Fig. 18

2. Sheaves

2.1. Groove angles

40 years ago sheaves in reeving systems were made of steel only. In most European countries the groove angle was 45° , in the UK it was 52° .

One day in the early 80s a German crane maker called me to discuss a problem. He had supplied some ship-to-shore container cranes to the US, and they were birdcaging the hoist ropes within a few days. He had already used up all the spare ropes and could not get ropes in the US fast enough. He had sold the same type of crane more than 20 times in Europe and had never seen this problem.

At that time ISO 4309 said that a birdcage was a result of a shock-load, but by then all the birdcages I had seen were the result of twisting the rope. But where in the reeving system had the rope been twisted? And why did it not happen on the cranes operating in Europe?

We looked at all his drawings, but everything seemed OK. Then I asked him for the drawings of the sheaves. He said he did not have them: He had sold the cranes without sheaves. In order to reduce the number of spare parts his customer had insisted to have the same sheaves on the German crane as on his other American made cranes, and he had installed American made sheaves.

It was immediately clear to me that this was the reason for the problems: In the US the groove angle of the sheaves was 30°! This meant that a rope entering the sheave with a fleet angle would touch the flanges of the grooves much higher than in a European made sheave, and then it would roll down much longer distances into the bottom of the groove. As a result, the rope would be twisted much more than on a sheave with a groove angle of 45°! We installed 45° sheaves, and the problem disappeared.

Later I could solve a greater number of similar problems in reeving systems with excessive fleet angles in Europe by installing sheaves with a groove angle of 60°.

Many crane makers insisted they were not allowed to increase the groove angle: DIN 15061 specifically asked for 45°! But that was not correct: The Standard says 45° min (Fig. 19)!

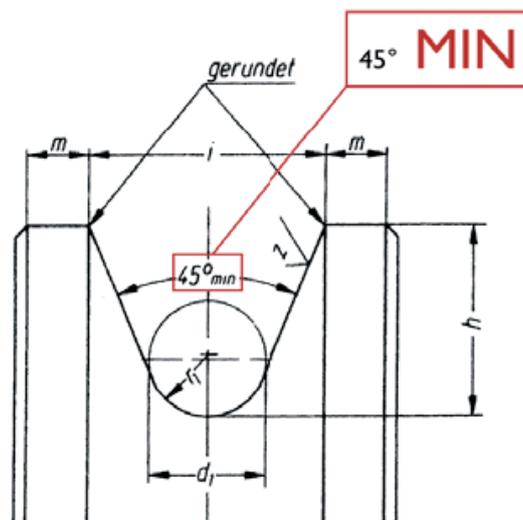


Fig. 19

The crane designers also feared that due to the smaller angle of support in the groove (120° instead of 135°) the wire rope fatigue life would be reduced. But the difference in bearing pressure between 45° degree sheaves and 60° degree sheaves is not very significant (Fig. 20), and bending fatigue tests showed that there was no measurable difference bending fatigue performance.

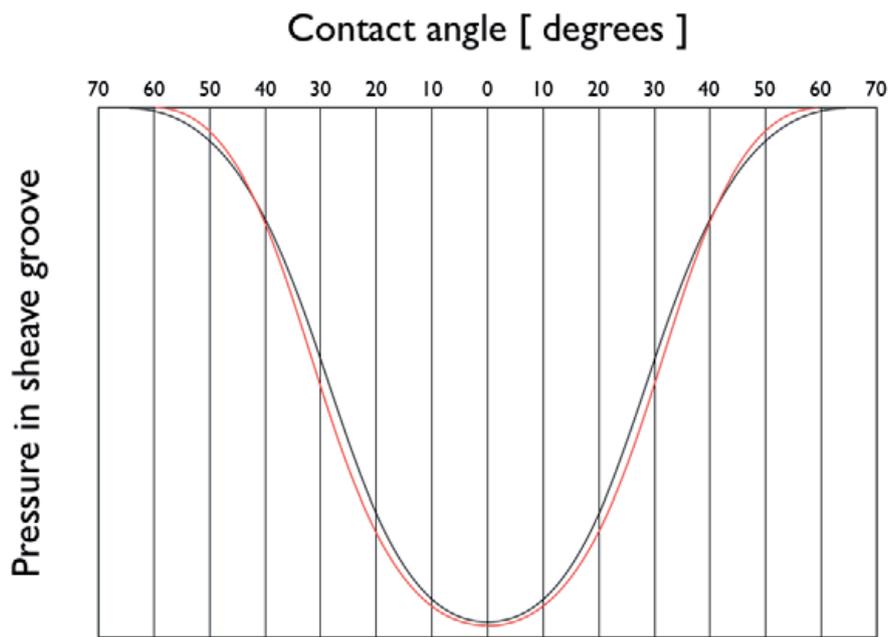


Fig. 20

The strongest resistance, however, came from those who thought it was obvious that increasing the groove angle would increase the danger of a rope “jumping the sheave”.

So we built a simple test hoist where the fleet angle could easily be varied by tilting the only sheave in the system (Fig. 21).

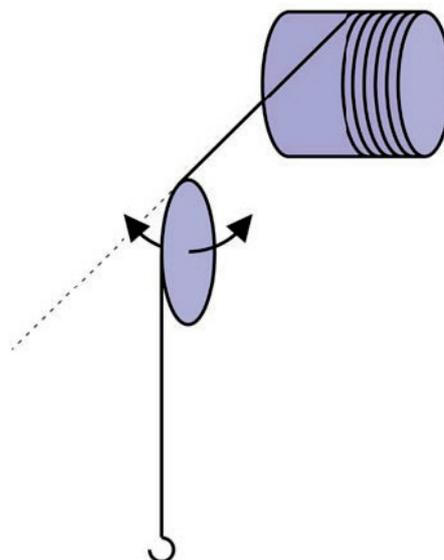


Fig. 21

We tested different types of rope on sheaves with groove angles of 30°, 45° and 60° (Fig. 22), and in all tests the grooves with 60° groove angle tolerated the greatest fleet angles.

It became obvious that a small groove angle did not prevent the rope from “jumping out of the sheave”, it did prevent the rope from *entering* the sheave!

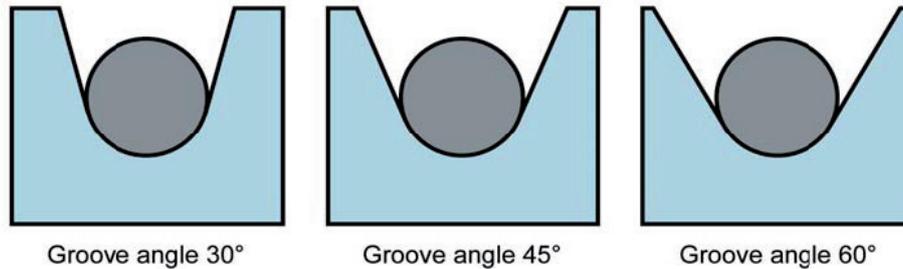


Fig. 22

Very slowly crane makers accepted the advantages of larger groove angles, and today the Standards such as ISO 4309 or EN 13135 allow for groove angles of 30° (for Americans who want to continue having trouble) to 60° (for those of us who don't) (Fig. 23).

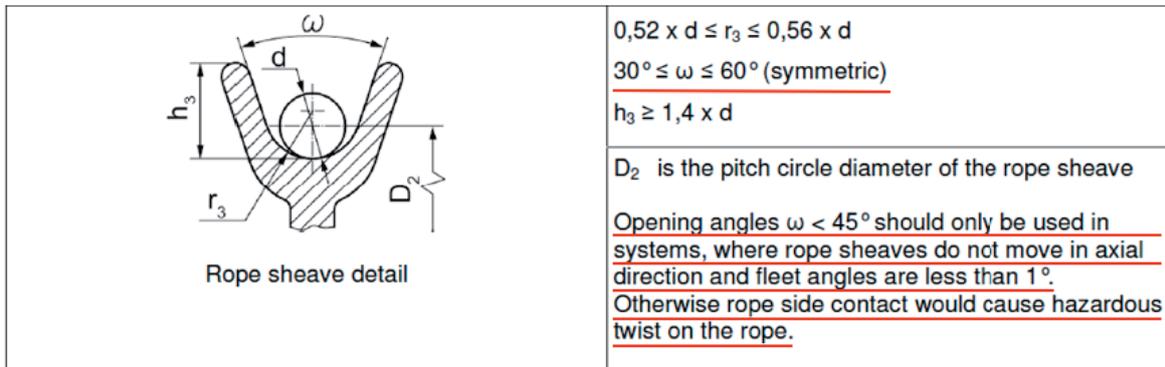


Fig. 23

A few years went by, and then the chief engineer of a large crane company called me. “I still owe you an information.” “Go ahead, I am listening”.

He told me that initially he had resisted to change to sheaves with groove angles of 60° because then he would have the same type of crane on the market with two different kinds of sheaves.

But then he realized that on their large lattice boom cranes their new and very successful end stopper (Fig. 24) could not be used in reeving system with 45° sheaves, but that it was possible to use it with 60° sheaves. So he decided to change to the wider grooves.

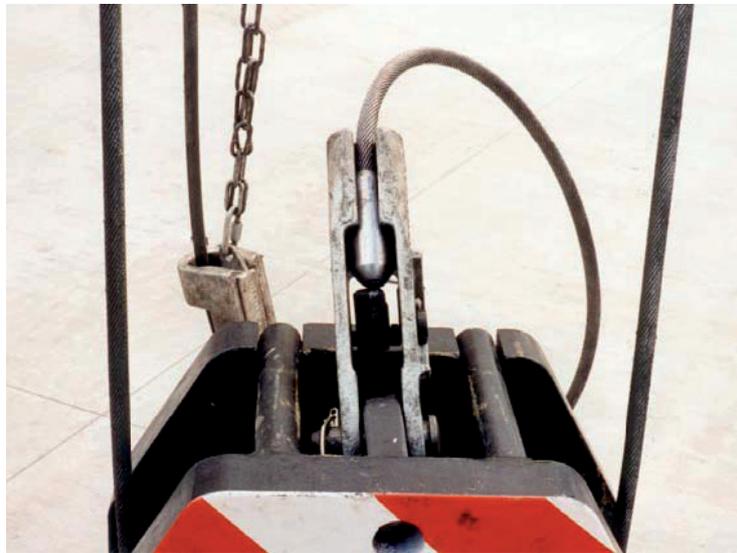


Fig. 24

“And what is the information you owe me?” “This was two years ago, and after we have changed to the 60° grooves we have not had a single complaint about block twist on these cranes!” The cranes with the 45° sheaves continued to have this problem.

2.2. Too many sheaves

When I started in the industry 40 years ago, a crane designer had to draw every sheave by hand (!) and in three different perspectives (!). This was a good reason to keep the number of sheaves to a minimum.

With modern CAD systems a sheave is drawn within seconds, and this makes it too easy to install unnecessary sheaves. I often see cranes which could have done much better with less sheaves!

I lecture on ropes and reeving system at the university of Clausthal (the city where wire rope was invented), and I use the following example to explain the effect of adding a sheave to the reeving system:

Crane 1 spools the hoist rope directly onto a drum. Crane 2 spools the hoist rope over a sheave and onto the drum (Fig. 25). The difference in fatigue life between the hoist ropes of the two cranes is tremendous.

2.3. Plastic sheaves

Plastic sheaves first appeared on the market at the end of the 1970s. They soon became very popular because they were cheaper, and for the smaller diameters also much lighter, than steel sheaves. And there were rumours that steel wire ropes lasted much longer on plastic sheaves than on steel sheaves.

But not everybody was convinced. The chief engineer of a larger crane company listened to the sales pitch of the sheave salesman, then opened the window of his 3rd floor office and threw the sheave in a wide arc over the parking lot below. When hitting the ground, the sheave burst into thousands of pieces.

“This would not have happened with a steel sheave”, he told the salesman. “You can come back to me when you have fixed this problem.”

Around that time a large crane maker asked me to perform bending fatigue tests with ropes on plastic sheaves under the same conditions we were testing ropes on steel sheaves. I installed a plastic sheave in the bending fatigue machine and started the test. About half an hour later I heard a loud bang: The sheave had burst.

When travelling over a sheave, the rope condition changes from straight to bent to straight, and the rope elements move relative to each other. The heat generated during this process can easily dissipate via a steel sheave, but not so easily via a plastic sheave: the plastic material is a thermal insulator. As a result, the rope gets much warmer on a plastic sheave than on a steel sheave, and finally the sheave material starts to yield and the sheave starts to wobble. Under the force of twice the line pull of the rope the sheave will finally buckle and break.

One day a tower crane manufacturer calls me: “What have you changed on your hoist rope?” “Nothing, why do you ask?” “You must have changed something. From a certain date on all our new cranes have problems with block twist”.

We found that the crane maker had changed something: exactly from that date on the crane was sold with plastic sheaves. Going back to steel solved the problems: The coefficient of friction between the rope and the sheave flanges is higher with plastic sheaves than with steel sheaves. Therefore under a fleet angle a plastic sheave will generate much more rope twist, leading to block twist under conditions where with a steel sheave the block was stable.

In the following years, several large cranes standing in the hot sun collapsed because of a combination of fleet angle and heat (Fig. 27).

In the 1980s plastic sheaves became very popular because apparently the ropes lasted much longer on plastic than on steel sheaves.

But then it became apparent that the number of rope failures increased, and in many cases plastic sheaves were involved. The ropes would fail without showing any external wire breaks.



Fig. 27

If a rope travels 180° over a steel sheave with a line pull of e.g. 5t, the sheave presses against the rope with a reaction force of 10t (Fig. 28, left). The forces act against the outer wires of the rope which are in direct contact with the groove surface, creating very high local bearing pressures in the wires (Fig. 28, right).

Over time, fatigue cracks will develop at those points of contact (Fig. 29). With every additional bend, the crack will propagate further until finally the wire will break completely.

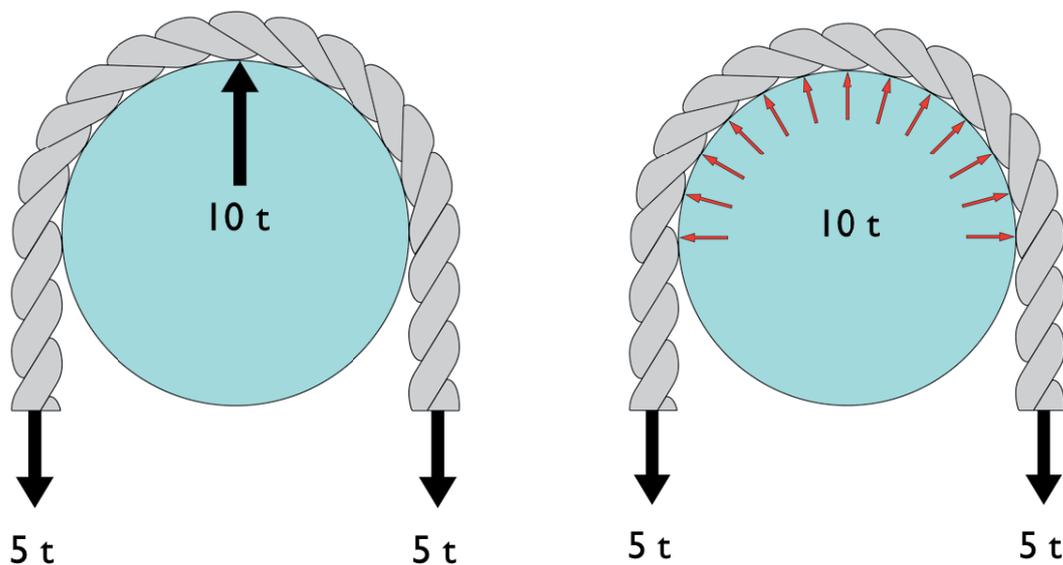


Fig. 28

The increasing number of wire breaks will weaken the rope (that is the bad news), but these wire breaks will occur at the rope surface and will therefore be visible during a rope inspection (that is the good news).



Fig. 29

Engineers started to understand that the plastic sheave would provide a soft bed for the outer wires of the rope so that the wires would no longer break at the points of contact with the sheave grooves (that was the good news).

But because no wire breaks were visible during visual rope inspections the ropes were now left in service for much longer and started to fail from the inside out until the whole rope parted without ever showing any signs of deterioration (that was the bad news).

Plastic sheaves have never been forbidden, but they have survived on cranes only in applications where other mechanisms (e.g. the drum crushing on multi layer drums) provided enough damage on the outside before any internal damage could lead to a rope failure.

3. Rope testing

3.1. Break tests

40 years ago all a rope maker had to know about his wire rope was its diameter and its breaking strength.

Very few crane designers asked for the wire rope modulus of elasticity or the radial stiffness, and most rope makers would not know what that was.

This has changed a bit, but not very much. Still today many rope makers do not know that the rope modulus is not constant but that it changes with the amount of preloading (Fig. 30) or, even under a constant mean load, as a function of the number of bending or tension-tension cycles.

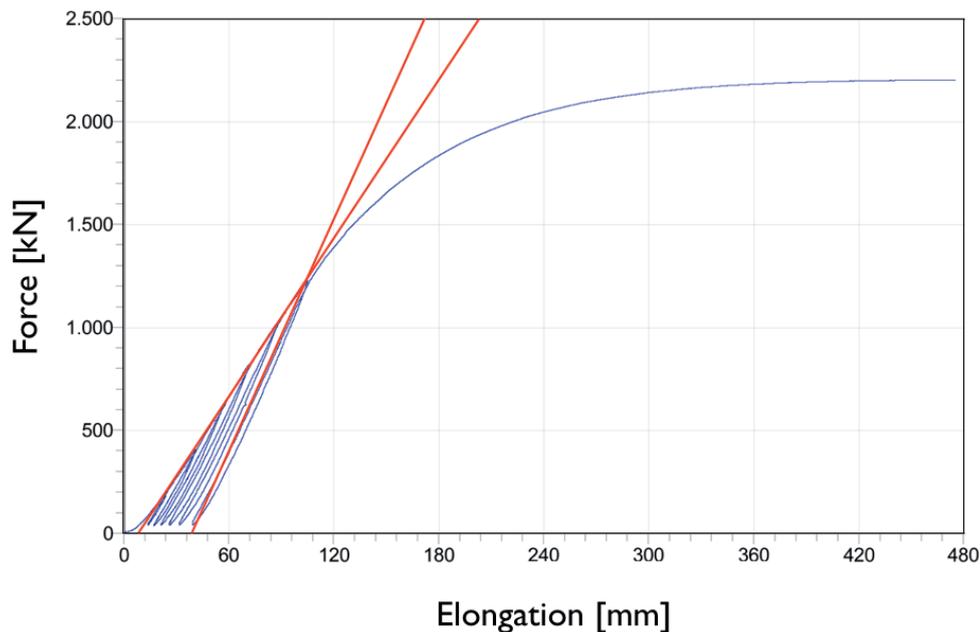


Fig. 30

Wire ropes are still valued by their strength, although other factors might be more important for the application.

As an example, Fig. 31 and Fig. 32 show force- elongation charts for two ropes with almost identical breaking strengths. The energy absorption capability, however, which is represented by the area under the chart and which is proportional to the product force • elongation, is about twice as big for the first rope with the much higher elongation at break (Fig. 31) than for the second rope (Fig. 32).

So in spite of comparable breaking strengths the rope in Fig. 32 is more likely to fail under a high dynamic load.

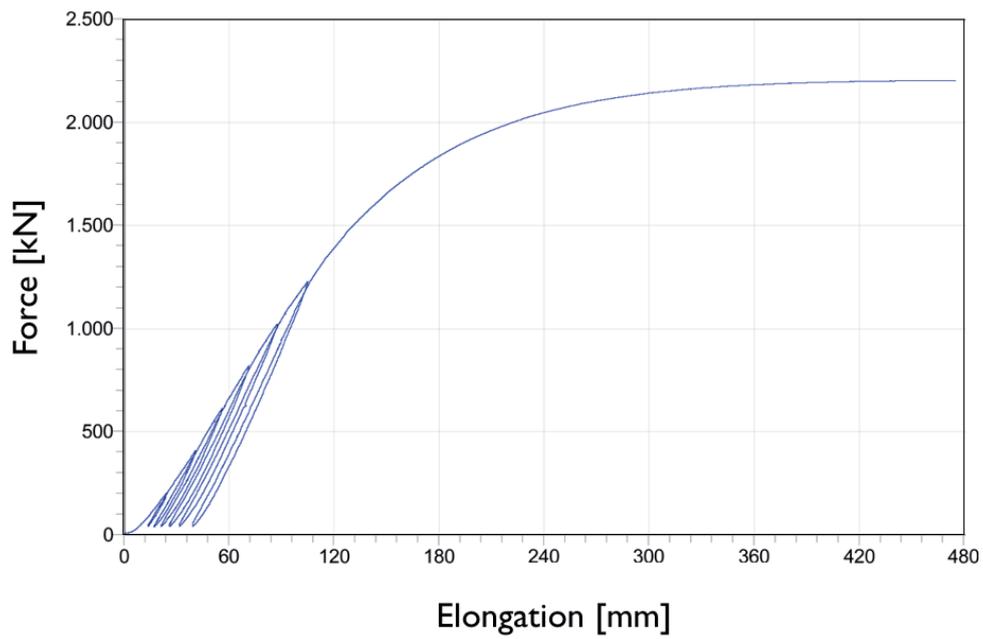


Fig. 31

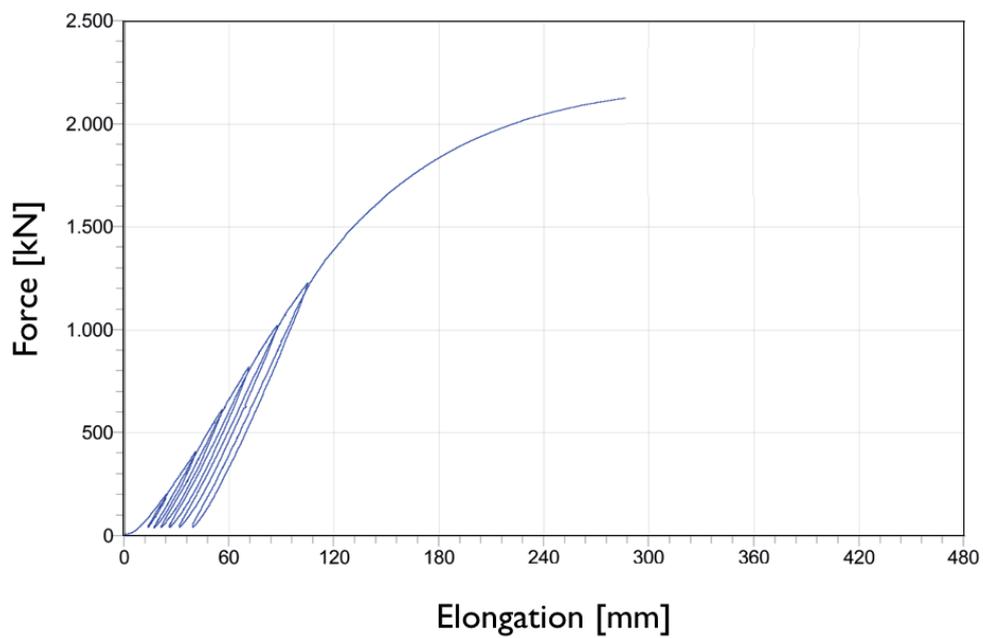


Fig. 32

3.2. Bending fatigue tests

For the last decades, bending fatigue tests with steel wire ropes have been made on test machines with a drive sheave and a test sheave (Fig. 33) or on machines with two oscillating test sheaves (Fig. 34).

The rope section to be tested moved back and forth over one sheave only. Is that a realistic simulation of your rope application? I guess not.

The test results are often very dubious. As an example: The rope section travelling over the sheave completely, making two bends back and forth, often have less wire breaks that the sections only spooling onto the sheave and not leaving it on the other side (therefore making only one bend back and forth). This is because the sheave would “milk” loosnesses out of the test zone, tightening the sections with 2 bends per machine cycle and loosening even further the sections making only half that number.

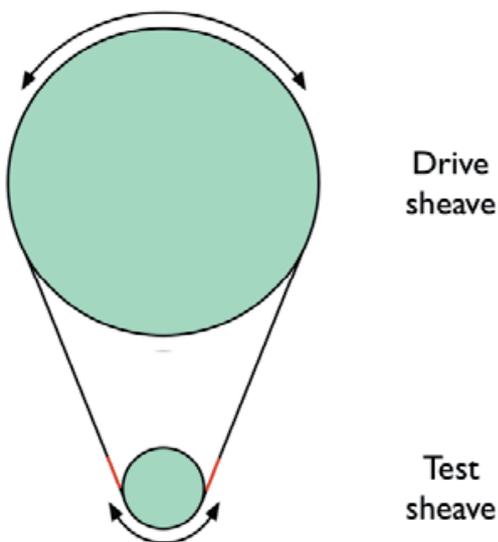


Fig. 33

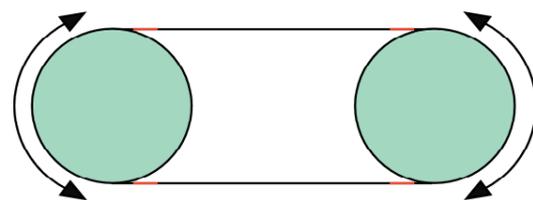


Fig. 34

I thought about how a realistic test machine would have to look like. It was clear that it would have to have more than one test sheave.

At first sight the test machine according to Fig. 34 looks like it fulfills this requirement. It has two test sheaves. But in this machine not one rope section travels over both test sheaves! We have the same unrealistic arrangement as in the first machine, only two times!

That makes it even worse: Heat generated on the test sheave in Fig. 33 can travel via the rope to cooler areas (on the drive sheave where much less heat is generated). But heat generated on one of the test sheaves in Fig. 34 cannot travel via the rope to the other test sheave because there the same amount of heat is generated.

So it was clear that a more realistic test machine would have to have a sheave arrangement which provided a heat gradient.

One day I sat down to put my vision of a test machine on paper. Together with Jean Marc Teissier (dep, France) I built a prototype, and one of the first tests we made on the prototype machine were for NASA.

Today Mr. Teissier sells machines according to this concept for a large range of rope diameters.

The machine has an arrangement of typically 5 sheaves (the number can be varied). The sheaves are arranged in a way that the rope travels through the system without fleet angles (Fig. 35).

This is because we want to test the influence of the line pull and the D/d ratio on the fatigue life of the rope, and not the influence of a fleet angle.

Fig. 36 shows a general view of the machine.

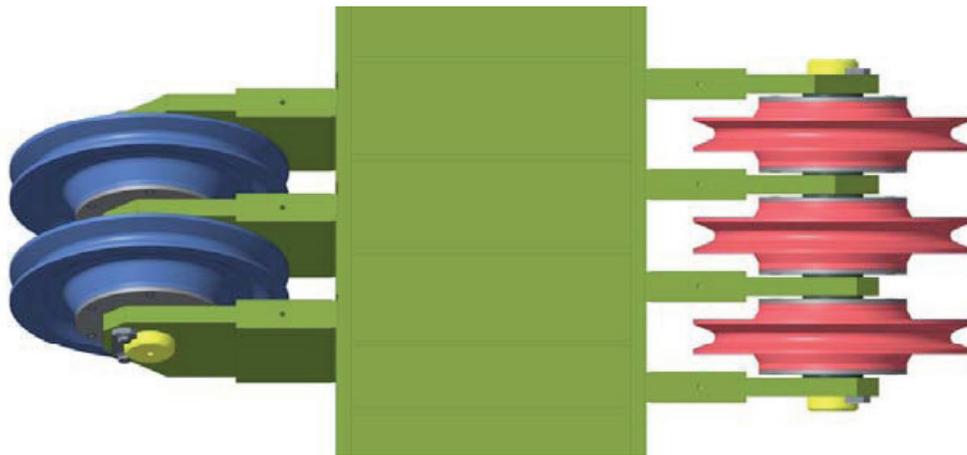


Fig. 35

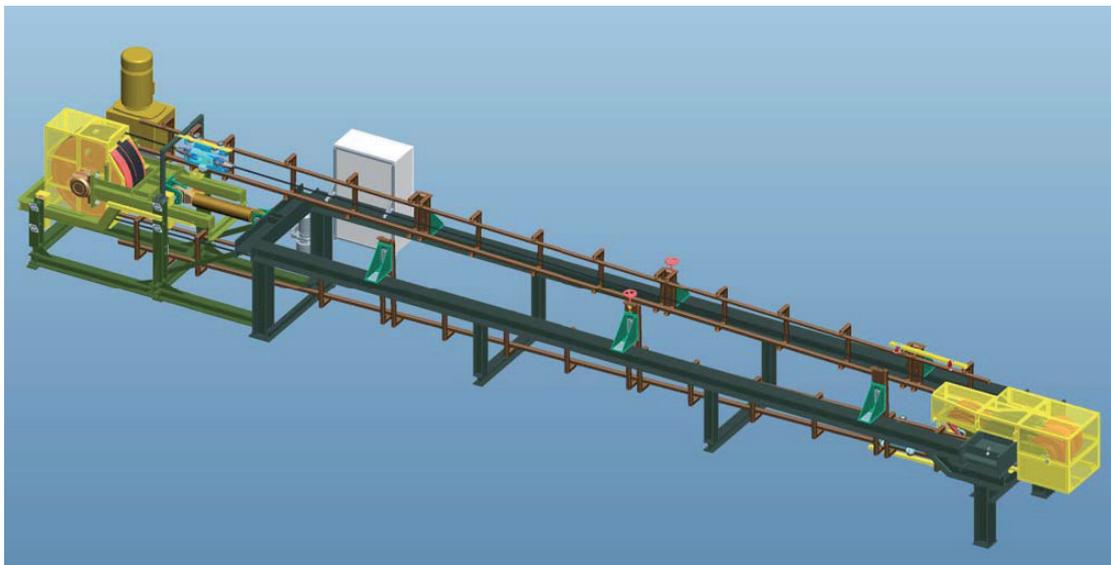


Fig. 36

After installing the test rope the line pull is set by activating a hydraulic cylinder (on the left). Then you press the “start” button, and the test rope starts running back and forth from the drum (on the left) through the 5 sheaves and then back through the sheaves to the drum. The test is never stopped until the rope fails with a loud bang.

Conventional rope test machines are stopped very frequently in order to measure rope diameter changes and to count wire breaks. We do not do this. But then how do we get these data? Do not worry, we will get these data and many more after the test. We even get the number of wire breaks inside the rope over the lifetime of the rope without ever stopping the machine.

And here is the trick: Only the middle section of the test rope will travel over *all five* sheaves, just the middle section. On the left and on the right hand side of the middle section we find two rope sections which travel over *four* sheaves only and do not make it to the fifth (Fig. 37).

So after the test is finished (and the middle section of the rope is broken) we can analyze these two sections which have made exactly 80% of the number of cycles of the middle section. So no matter how many cycles the rope will do under the chosen conditions, these two sections will represent the condition of the rope after 80% of the cycles to failure.

To the left and to the right of those two sections we find two rope sections which travel over three sheaves only and do not make it to the fourth. So these two sections represent the rope condition after 60% of the cycles to failure. And of course we have rope sections travelling over 2, 1 or 0 sheaves only, representing the rope condition after 40%, 20% and 0% of the number of cycles to failure (Fig. 38).

So after the test we can measure lay lengths and rope diameters and count wire breaks on the rope surface on these different rope sections representing the different stages of the rope life.

Then we can plot the curve of the number of wire breaks over the life time (in % of the cycles to failure) and check at which point in the rope life we had reached the discard number of wire breaks as shown in ISO 4309 (Fig. 39). So we can determine when we reach the discard state without ever stopping the test and without ever counting wire breaks during the test. We do the counting when the next test is already running.

Then we can dismantle the sections and measure IWRC diameters and lay lengths and count the wire breaks on the underside of the outer strands or on the IWRC. Imagine you would do that with the conventional test machine: once you open your rope the test will definitely be over. On this machine, however, you can dismantle the rope after 20% of the rope life, after 40%, after 60%, after 80% and after failure although you only have made *one* fatigue test!!!

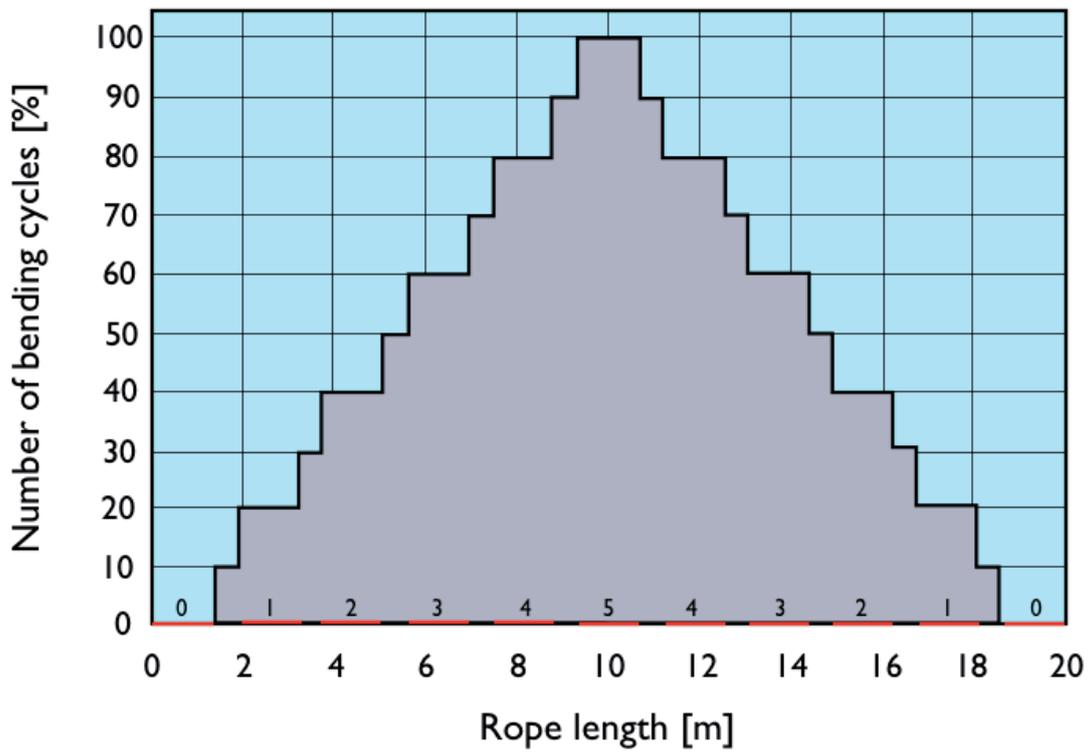


Fig. 37

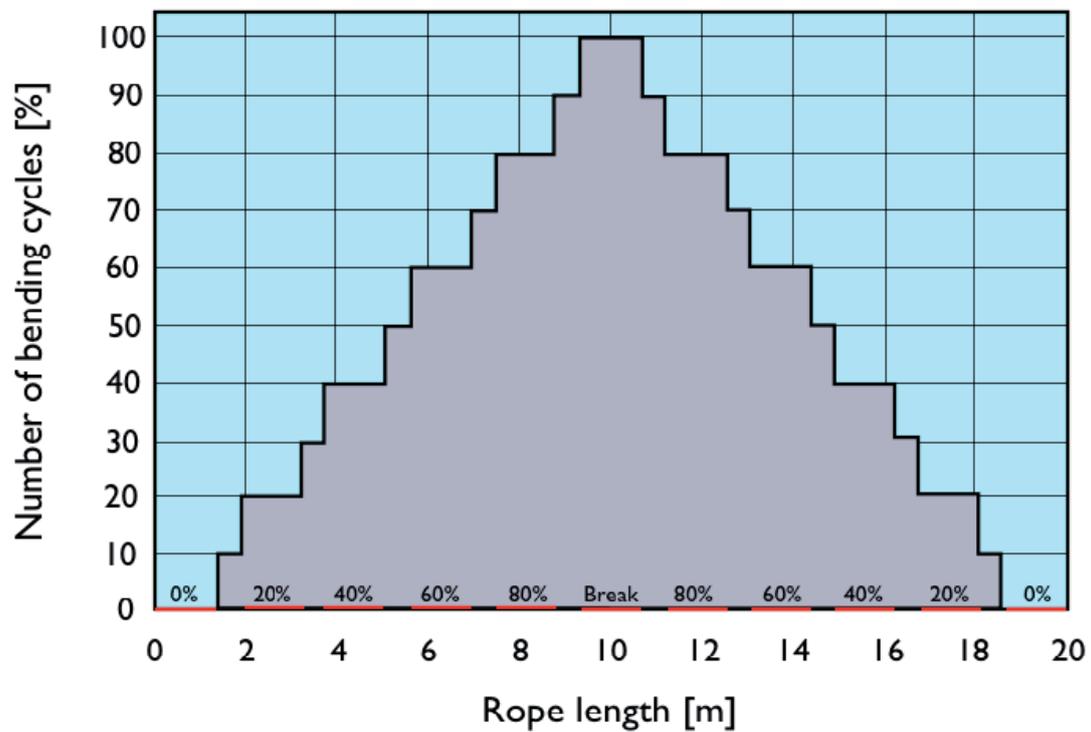


Fig. 38

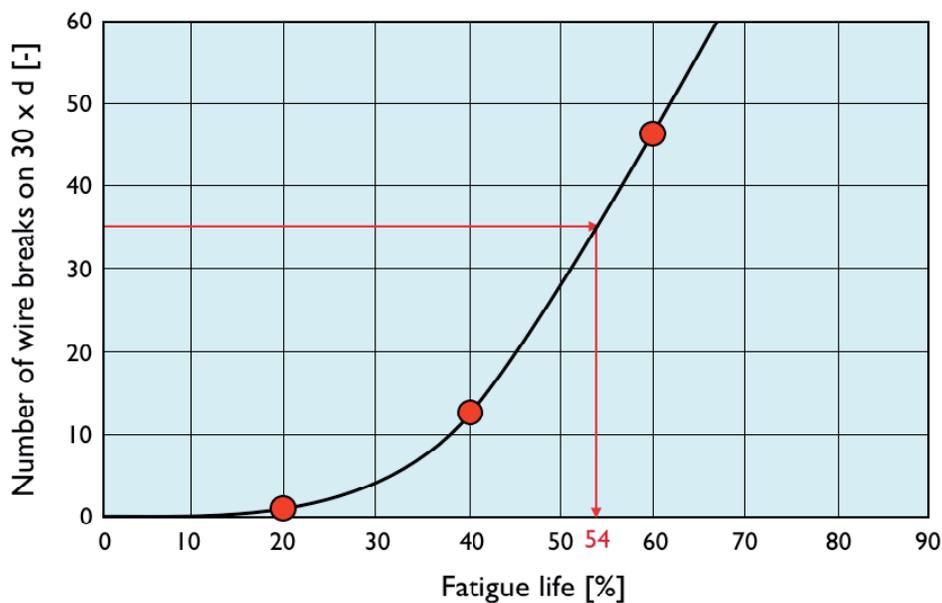


Fig. 39

But that is not all: If you want to know what the remaining breaking strength or the modulus of elasticity of the rope is after 60% of the cycles to failure, just make a pull test with the 60% sample of the other side!

You can check at which stage in rope life the different components of the rope start to deteriorate. And you can modify the rope design if one component deteriorates too early. This will speed up steel wire rope development tremendously.

I had designed the first rope ever with compacted strands and a plastic infill in 1978. The rope named Turboplast is still sold today and became the most copied special wire rope in the world. More than 30 years later I thought I knew everything you can know about this rope design. How I was wrong! One test on this innovative test machine taught me a lot more about the product.

Analyzing the different rope sections will also give you a good idea how your rope will look like in certain stages of its life. Fig. 40 shows how the rope looks after 0%, 20%, 40%, 60%, 80% of the cycles to failure and the 100% sections near the break.

The 40% section has wire breaks, but the breaking strength of this section is higher than that of the new rope. The 60% section has an even greater number of wire breaks, but it still has 97% of the strength of the new rope.

It is obvious that this bending fatigue machine simulates real rope applications much better than the conventional designs discussed above.

With only one fatigue test the machine provides a wealth of information on the rope (most of which I have not even mentioned here).

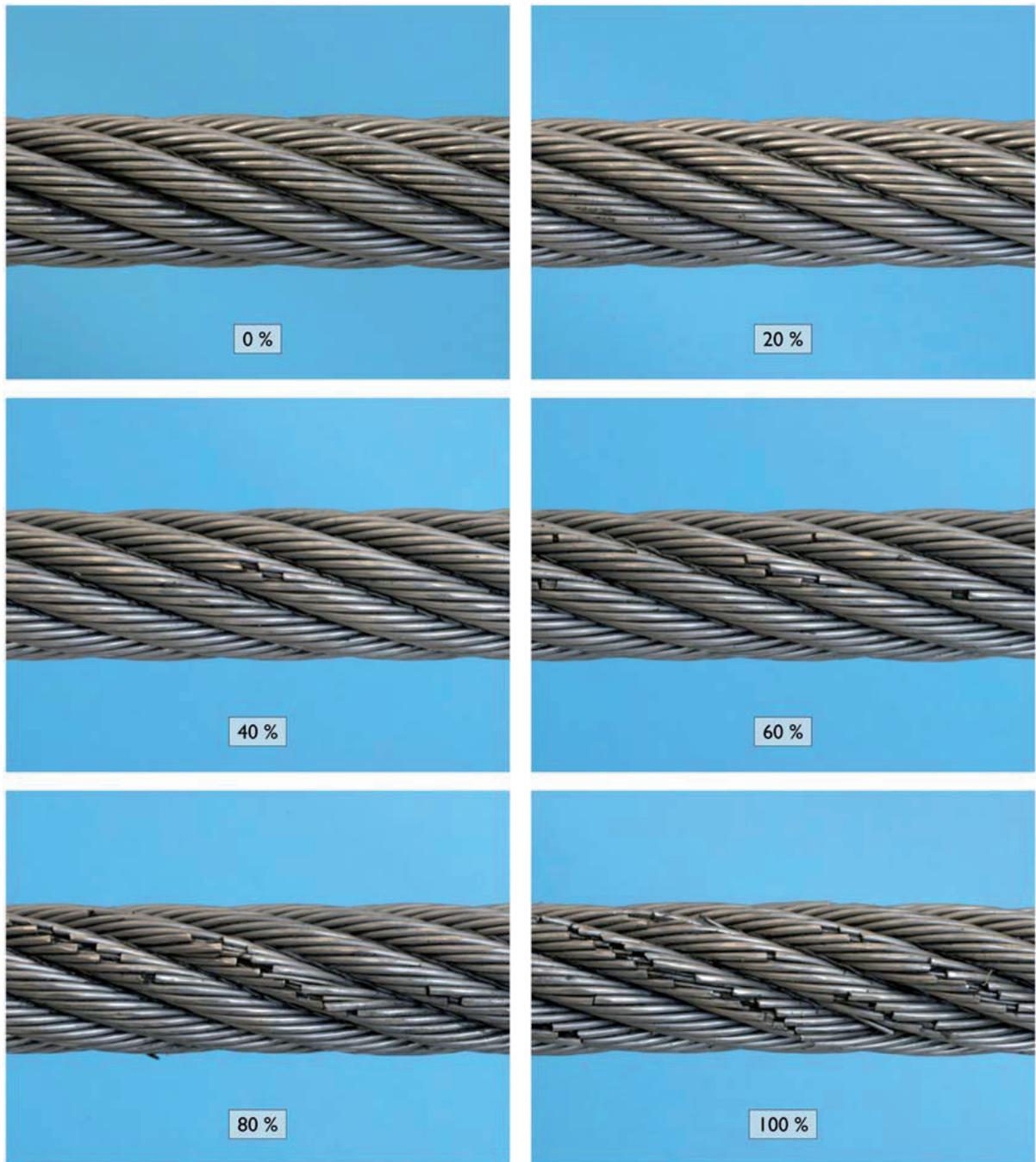


Fig. 40

4. Drums

Today we are used to see drums with multiple layer spooling and large D/d ratios (Fig. 41).



Fig. 41

40 years ago a typical crane drum had a small D/d ratio in order to reduce the torque requirement. In order to accommodate a great rope length it had to have a great number of wraps. So it had to be very long (Fig. 42).

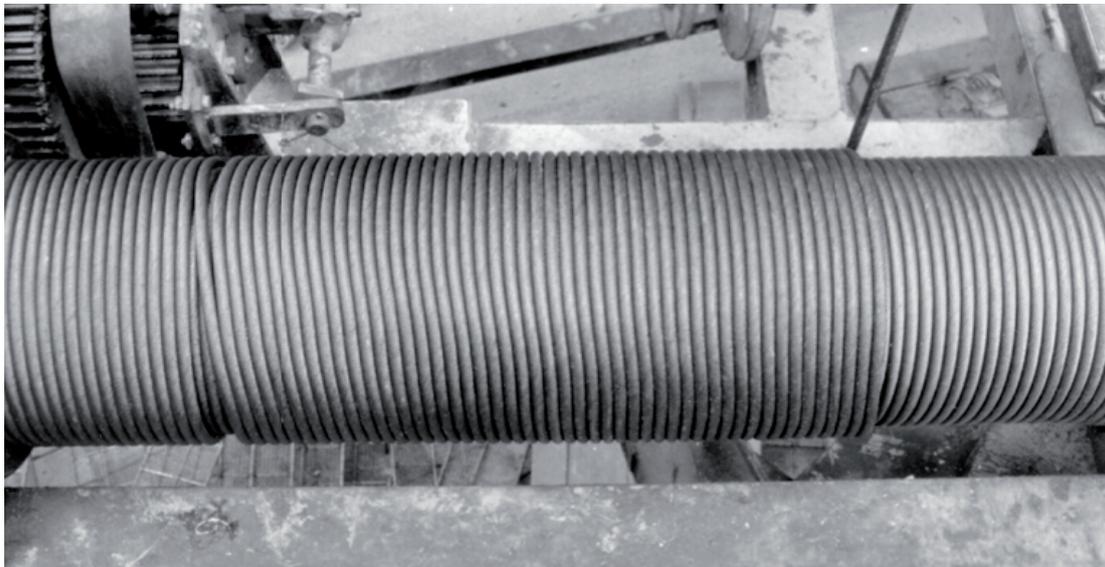


Fig. 42

The fleet angles both on the drum and on the first sheave became excessive, leading to spooling problems (Fig. 42) as well as rope damage (Fig. 43) and drum damage (Fig. 44).



Fig. 43

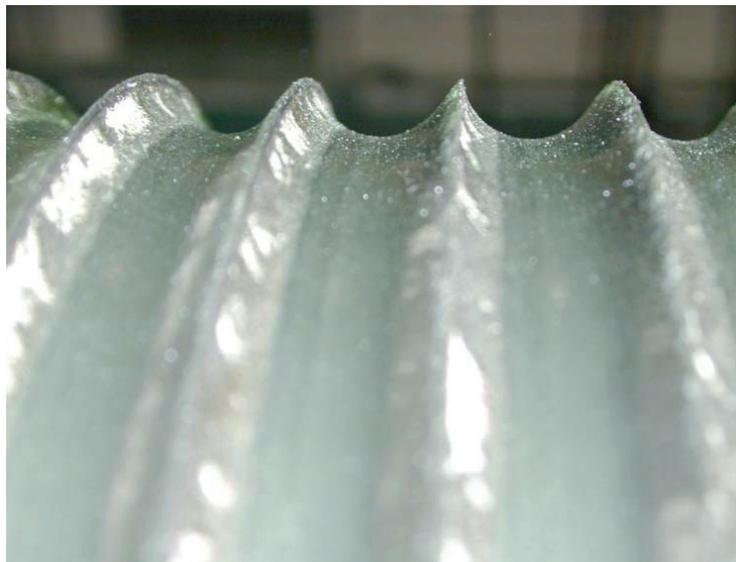


Fig. 44

At that time I developed a program to calculate if and where the rope entering or leaving the drum would contact the neighbouring wrap or the rim of the drum grooves based on the drum geometry and on the fleet angles. The software would calculate a large number of cuts through the drum body (Fig. 45) and then run the sections as a movie.

Over the years I could identify and correct hundreds of faulty drum designs using this software.

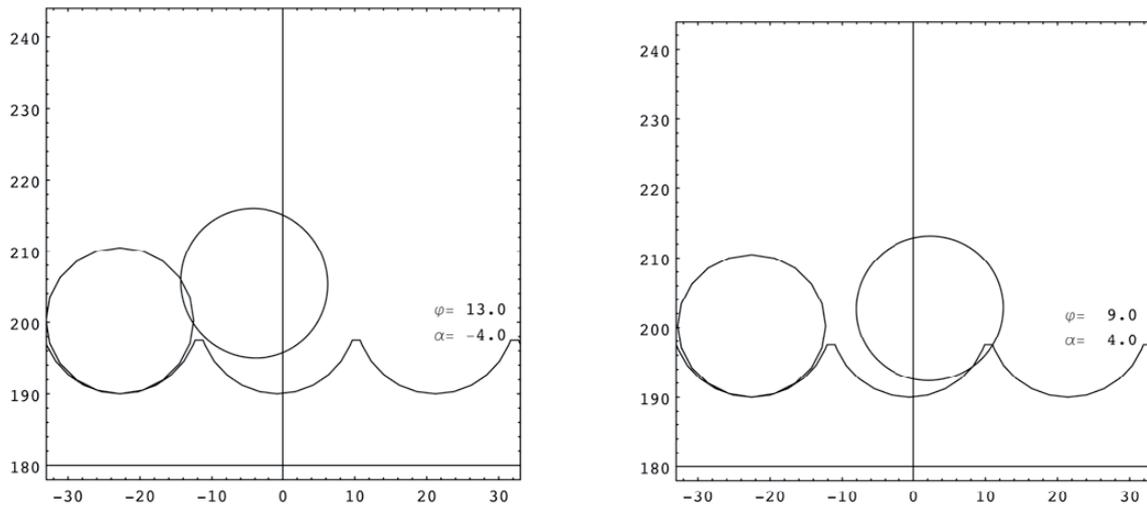


Fig. 45

It also became apparent that the number of outer strands of the rope plays an important role. The drums in Fig. 46 and Fig. 47 have the same pitch and the ropes have the same diameter, but on average on the drum there will always be a larger gap between two neighbouring 6 strand ropes (Fig. 46) than between the neighbouring 18 strand ropes (Fig. 47).

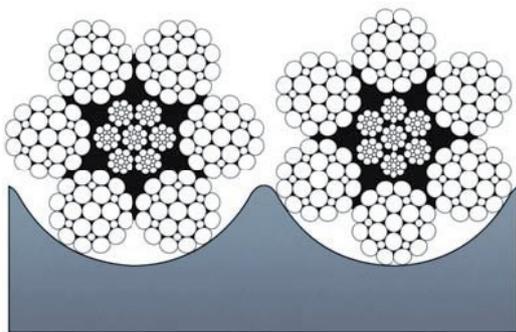


Fig. 46

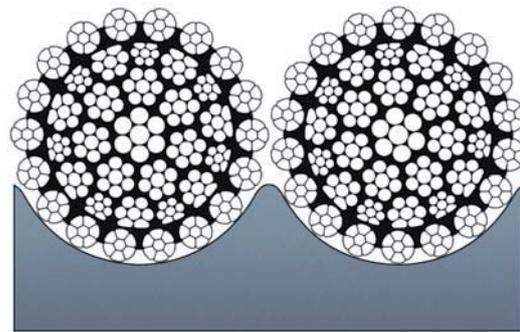


Fig. 47

4.1. Multi layer drums

When tower cranes got higher, a second rope layer on the drum was introduced (Fig. 38). That was when the problems started: In the second layer the rope was no longer guided by a smooth groove. It followed the bed of the first layer (spooling in the wrong direction) until it hit against the neighbouring wrap.

Then it had to “cross over”: It had to leave the bed created by two wraps of the first layer and had to climb on top of one wrap, thereby increasing the contact force by about 73% (Fig. 48).

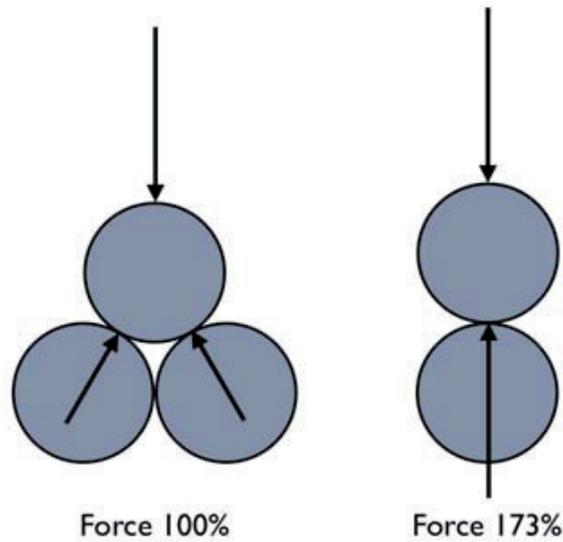


Fig. 48

Quite often, and especially when in combination with changes in line pull, this increased contact forces led to severe rope damages in the crossover zones (Fig. 49).



Fig. 49

Today drums spooling 8 to 10 layers are not uncommon. Spooling the rope onto the drum under tension from the bottom layer up normally works fine. But spooling a few layers of rope onto the drum without tension and then picking up a load under high line pull is dangerous: The rope has a tendency to pull in between two wraps of the previous layer. As a consequence, the rope forces against the drum flanges increase, and the flanges might pop off.

Another problem might occur when the rope is spooled off the drum again when load is lowered: the pulled-in section of rope might not come off the drum, and as a consequence the load will suddenly be lifted again (Fig. 50).

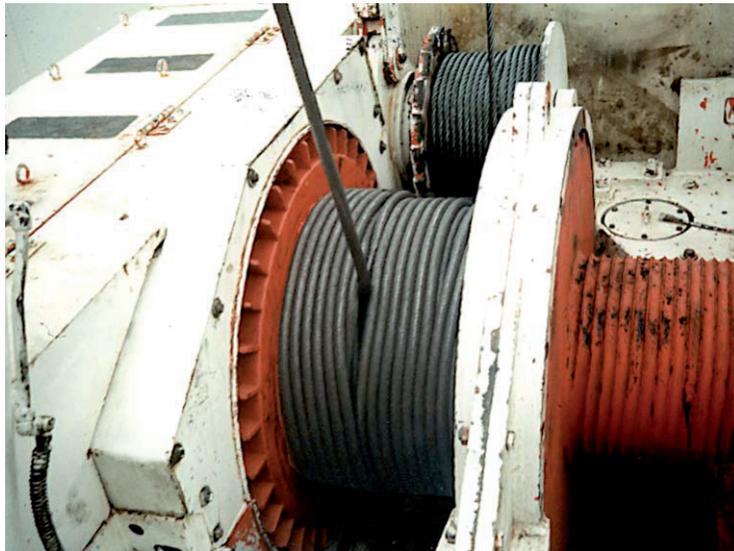


Fig. 50

In such a situation the the hoist rope might break, or the crane might tip over.

The industry has not yet found a good system to pretension the first layers on the drum under tension to avoid such a problem.

4.2. Lang's lay ropes

When I started in the wire rope industry 40 years ago steel wire ropes were typically made in regular lay, and only occasionally a Lang's lay rope was produced. That has changed: You will hardly find a regular lay rope on a large tower crane or mobile crane today. This change has been caused by the multi layer drum.

When two neighbouring wraps of regular lay rope come in contact during multi layer spooling, their outer wires (which are pointing in the same direction) form indentations. The next wrap coming in will press them together even further. When the rope wants to spool off again it will be locked to its neighbour, and the outer wires will get severely damaged (Fig. 51).



Fig. 51

Fig. 52 and Fig. 53 show the difference: The outer wires of two neighbouring wraps of regular lay rope form indentations (Fig. 52) while those of Lang's lay ropes do not (Fig. 53).

Compacting the outer strands reduces the danger of indentations, and the best remedy is to use Lang's lay ropes with compacted outer strands.

Recently a leading crane manufacturer has introduced a new concept which might solve the problem of the mutual damage of neighbouring wraps in multiple layer spooling: The drum “knows” at which point the rope spooling onto the drum will hit the neighbouring wrap, and exactly when that is going to happen the drum moves to the side by half a drum pitch so that the rope will be able to spool into the new position without even touching the neighbouring wrap.

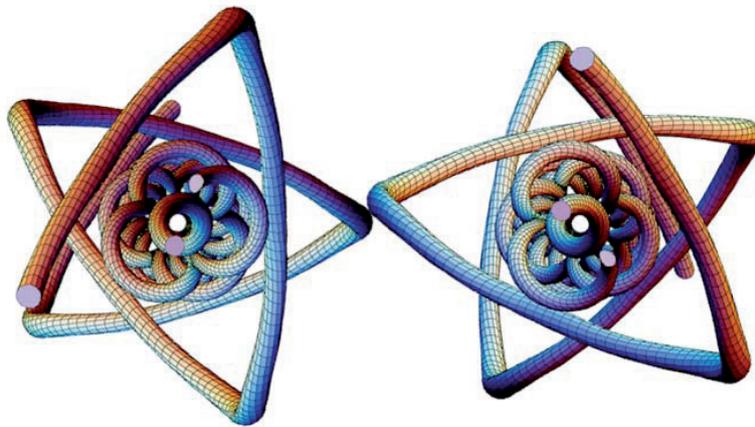


Fig. 52

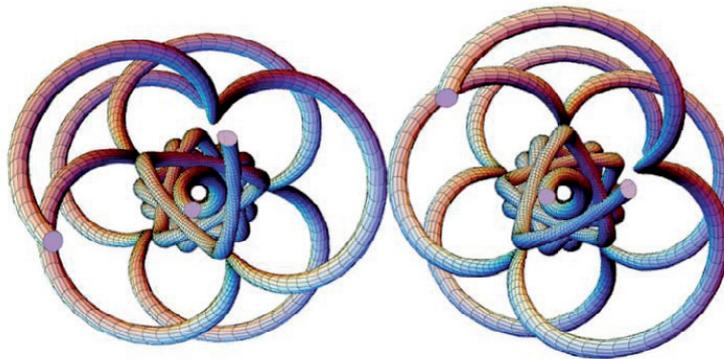


Fig. 53

4.3. Drum stability

Over the years the number of rope layers on the drum increased, and we saw the first drum barrels collapsing.

If we spool 9 rope layers onto the drum, the rope forces trying to compress the drum barrel will not be 9 times as high as with one layer, but (dependent on the radial stiffness of the rope used) between 2 and 5 times as high (Fig. 54, © Prof. Lohrengel, University of Clausthal, modified).

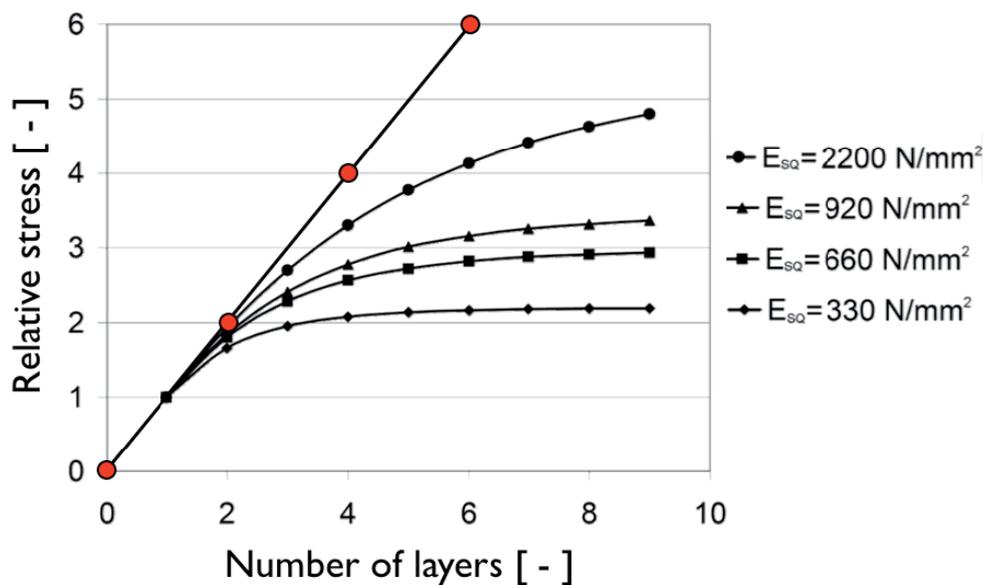


Fig. 54

Fig. 55 shows a drum deformed by several layers of rope.

Unfortunately some widely used drum design rules were faulty, and from time to time a drum barrel collapsed.



Fig. 55

“Softer” (less radially stiff) ropes deformed more and caused less stress to the drum barrel, but more stress on the flanges. This meant that when using a radially stiff rope under maximum load your drum barrel could collapse, and when using a less radially stiff rope your flanges could pop off. Which one do you prefer?

Why did we see a greater number of drum barrel collapses offshore than onshore although for both applications drums have been designed using the same faulty rules?

This can easily be explained by the fact that many offshore winches are used up to their maximum permissible line pull while land based cranes will probably never experience their maximum permissible line pull. For these cranes in most rigging configurations the permissible tipping moment is the limiting factor. It will in most cases be reached at much lower line pulls.

4.4. Fibre ropes on multi layer drums

For the same line pull fibre ropes spooling in multiple layers apply much lower compression forces onto the drum body than steel ropes. But they deform too much when the next layer of rope tries to cut in and therefore show poor spooling behaviour in multiple layers. A new drum design with conical flanges might be able to solve this problem (Fig. 56, © Deep Tek).



Fig. 56

5. Inspection

40 years ago most rope designs were not very sophisticated. If an outer wire broke the two wire ends would stand up. So rope inspection was easy: your colleague ran the rope through his hand, and every time he cried you noted a wire break 😊.

Today strands might be preformed and/or compacted, and if an outer wire breaks, the wire ends might just pull apart by a fraction of a millimeter and otherwise stay in their position. Especially when the rope is well lubricated it will almost be impossible to detect the wire break.

Fig. 57 shows a rope made of compacted strands. At the top you see the lubricated rope in service. This is what an inspector would see. In the middle you see the same section of rope after cleaning in an ultrasonic cleaner. At the bottom you see the same rope after bending it by hand.

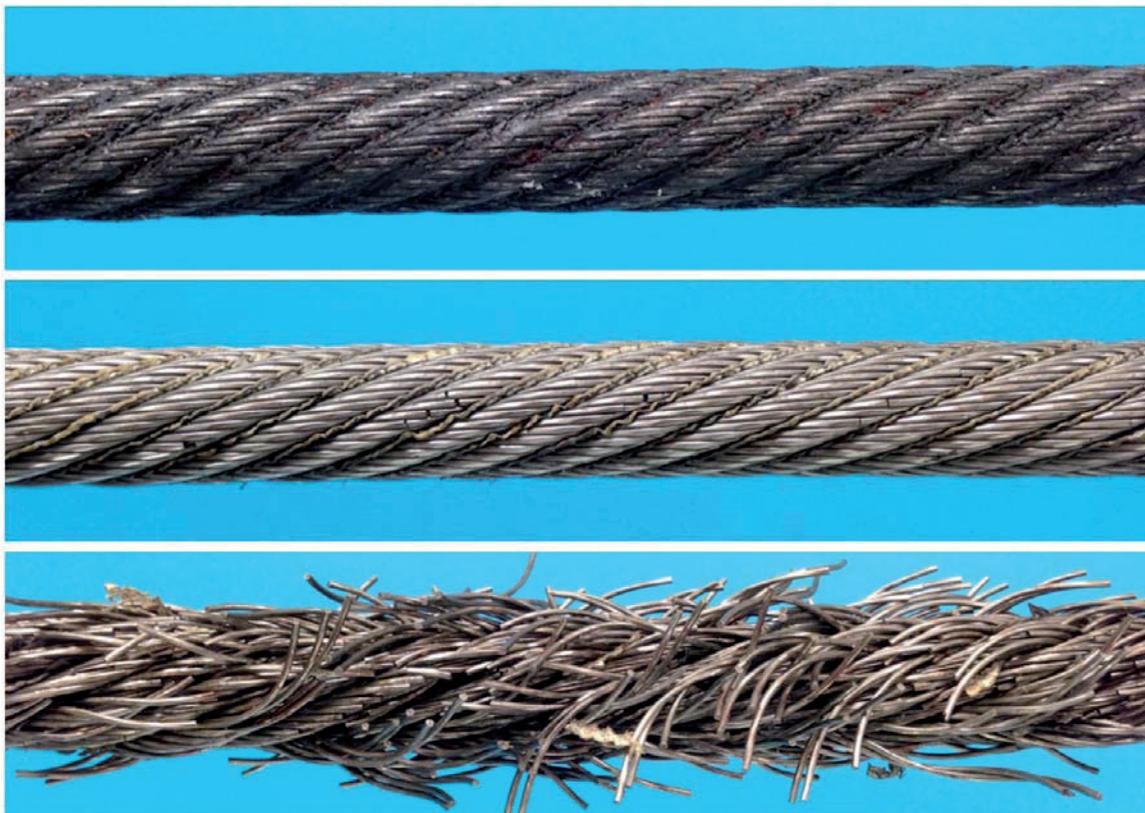


Fig. 57

Finding external wire breaks on a lubricated rope with preformed and compacted outer strands is difficult, and finding the (in this case numerous) internal wire breaks is almost impossible.

To make matters worse, the percentage of the load bearing cross section of the rope we can see during a visual inspection has tremendously reduced over time.

In Mr. Albert's first wire rope, manufactured in 1834, every single wire was visible from the outside, even when it was lying inside the rope (Fig. 58, left). So at that time 100% of the load bearing cross section of the rope could be visually inspected. This made the first wire rope in history relatively safe.

With increasing number of rope elements this percentage, however, came down tremendously: In modern ropes, only about 20% of the load bearing cross section is visible from the outside (Fig. 58 middle and right). Today, visual inspection means 20% evidence and 80% hope.

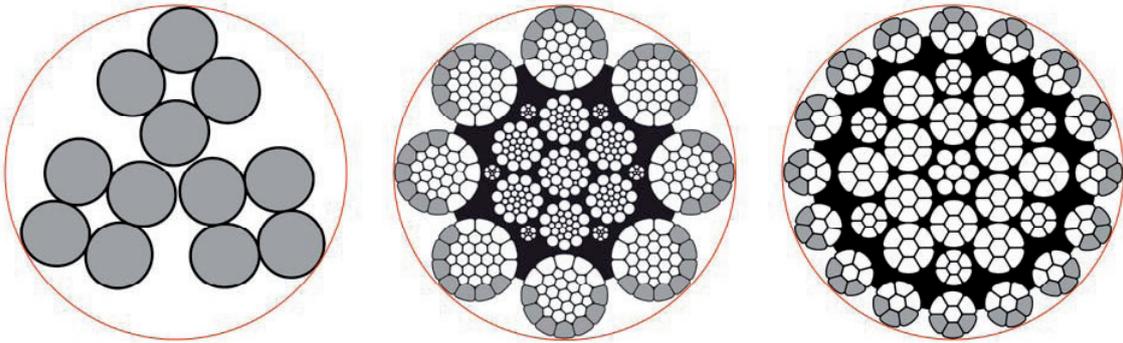


Fig. 58

In large diameter ropes (e.g. $\geq 80\text{mm}$) often the total number of rope wires increases even further and as a result the percentage of the cross section visible from the outside becomes even smaller.

5.1. Magnetic testing

During the second world war, gun barrels were tested by running them through a magnetic field. Distortions of the magnetic field lines indicated defects in the barrel. After the war, this method was modified to detect wire breaks outside and inside steel wire ropes. Fig. 59 shows a wire break in a strand and the resulting signal in the measuring chart.

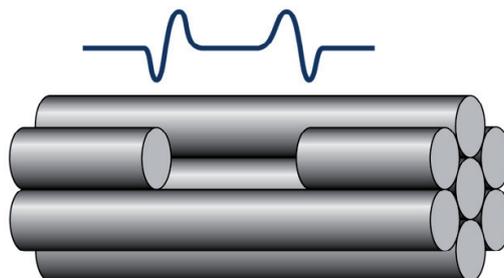


Fig. 59

The magnetic test machines have been improved over the years, but the interpretation of the signals still requires a lot of expertise. In addition, the reliability of the signal gets worse with an increasing number of rope wires.

Ten years ago I patented the idea to use the regularity and the self-similarity of a wire rope for computer-aided visual rope inspection: Fig. 60 shows a Figure of the patent.

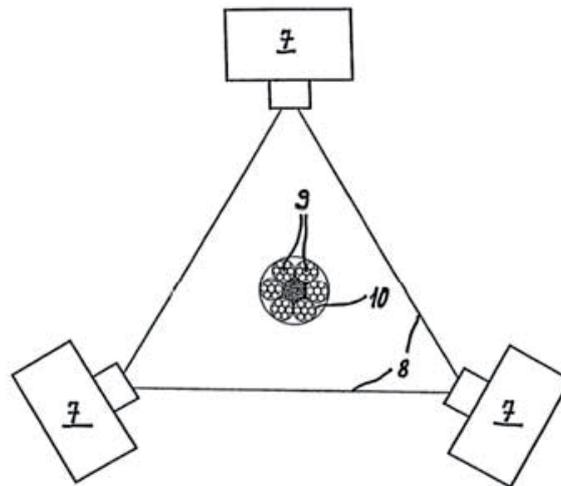


Fig. 60

In order to demonstrate the feasibility of the idea I triggered a strobe light by the Kope drum rotation of a deep mine shaft in Australia. I pointed the strobe light against the hoist rope which was spooling at very high speed.

Under the flashes of the strobe light, the movement of the rope was immediately frozen. Instead of seeing a “gray tube” moving with high speed we saw a static rope: the first flash showed one lay length, the second flash showed the next lay length, etc., and because all lay lengths looked the same the rope looked like it was not moving at all (Fig. 61).



Fig. 61

But if the strobe flash illuminated one lay length which was NOT like the others, it jumped into your eye. So wire breaks were easily detected.

If the lay length got longer along the rope length, the static picture started stretching, and if the diameter got smaller along the rope length, the static picture started “breathing”: In the static picture the rope did not move, but it slowly got thinner.

It was as if you looked at a photo of your long dead grandmother, and suddenly she starts smiling!



Fig. 62

I used special cameras to continuously record the rope surface for later analysis. My prototypes were still using film (Fig. 62), but it was obvious that for practical applications digital cameras and powerful data storage systems would be required.

Based on this idea Stuttgart University has in the meantime built a number of units which scan the whole surface of the rope using 4 cameras (Fig. 63) and then use image recognition software to trace every individual wire (Fig. 64).

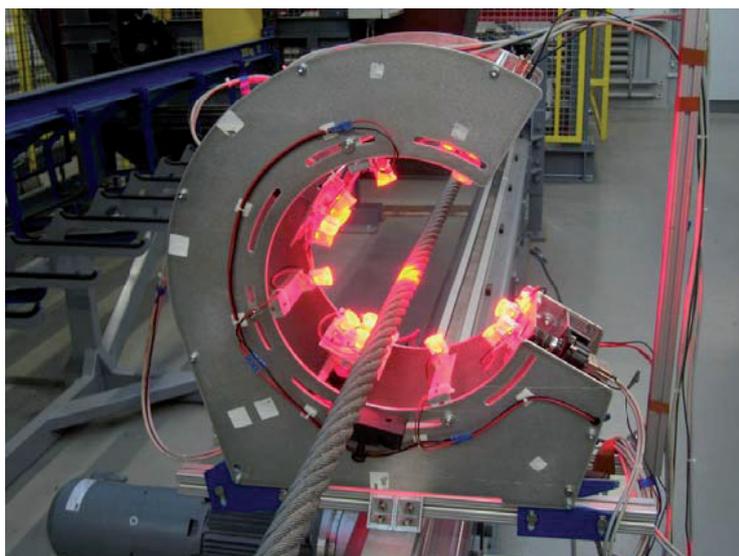


Fig. 63 (© IFF University of Stuttgart)

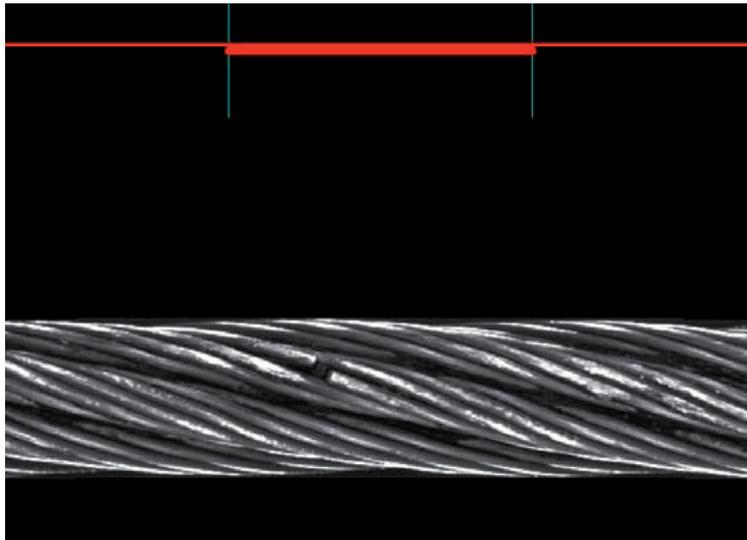


Fig. 64 (© IFF University of Stuttgart)

Next to wire breaks and diameter changes, the lay length of the rope is measured over the whole rope length (Fig. 65).

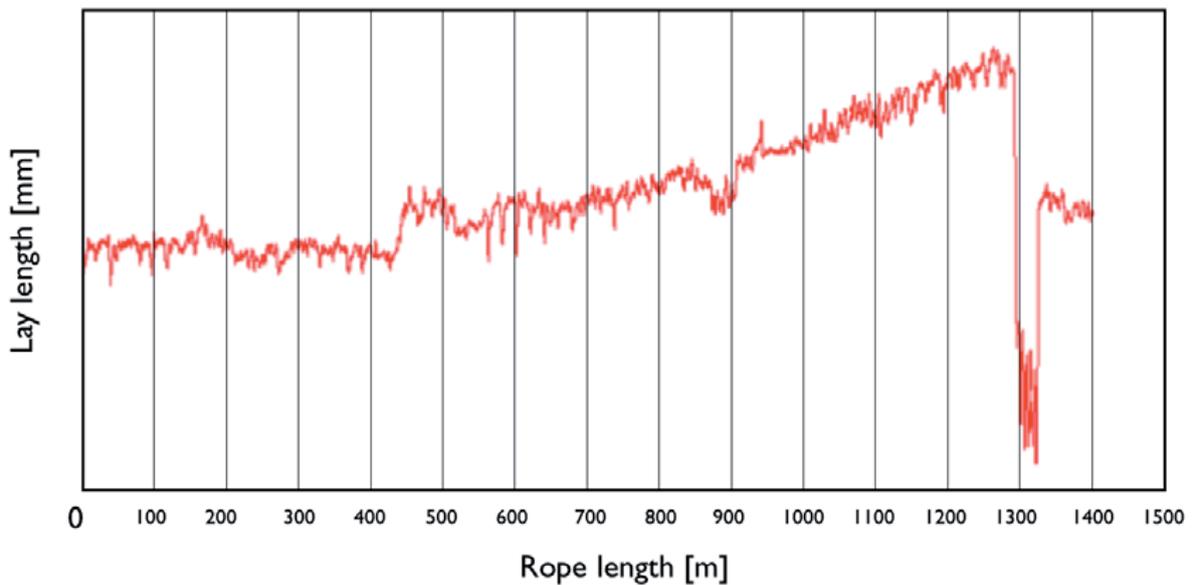


Fig. 65 (© IFF University of Stuttgart)

So far these machines are mainly used to inspect ropes of aerial tramways and deep shaft mines, but ropes in offshore applications might follow.

6. End connections

Most end connections we use today were already used 40 years ago. But some of them had a restricted use because of some faulty assumptions.

6.1. The turnback loop

The turnback loop with an aluminium ferrule is a widespread and reliable end connection. During the pressing process the wire rope does not get damaged where it is in contact with the aluminium sleeve because if the stresses become too high the aluminium “gives” and yields. But where the live end of the rope is pressed against the dead end, the rope is pressed against an object as hard as itself, and none of the two will “give”.

In this compression zone the outer wires of the neighbouring rope falls are almost parallel for regular lay ropes, but they will cross at an angle for Lang's lay ropes. It was therefore assumed that the outer wires of Lang's lay ropes would get severely damaged during the pressing process, and it was forbidden to use this end connection with Lang's lay ropes.

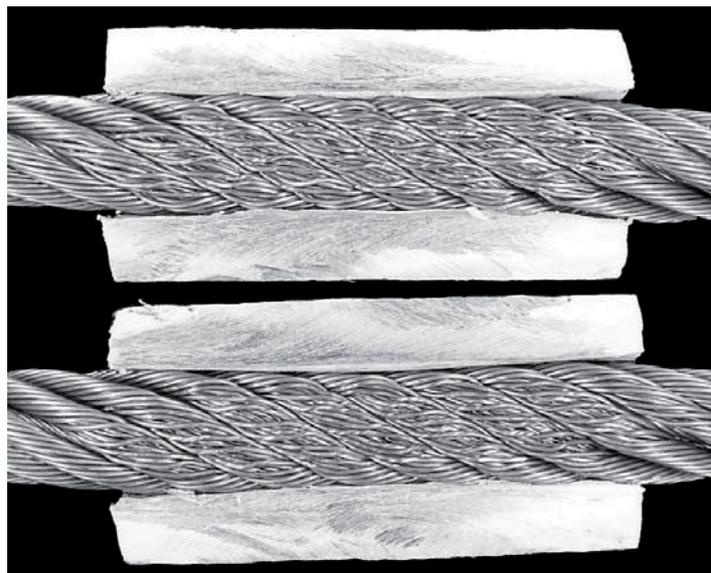


Fig. 66

40 years ago Lang's lay ropes became popular because of the multi layer spooling problems with regular ropes, but they could not be terminated with the most popular end connection for tower crane and mobile crane ropes. So I started to compare the performance of both rope types pressed with aluminium sleeves. The damage in the contact area between the live and the dead end of the rope turned out to be as great for regular lay ropes (Fig. 66) as it was for Lang's lay ropes (Fig. 67), and the breaking strengths were absolutely comparable.

The tension-tension fatigue performances of Lang's lay ropes in aluminium sleeves were slightly inferior, but the hoist ropes would fatigue more by bending fatigue than by tension-tension fatigue.

With these results and the pressure of the crane industry to use Lang's lay ropes with this termination the ban quickly disappeared.

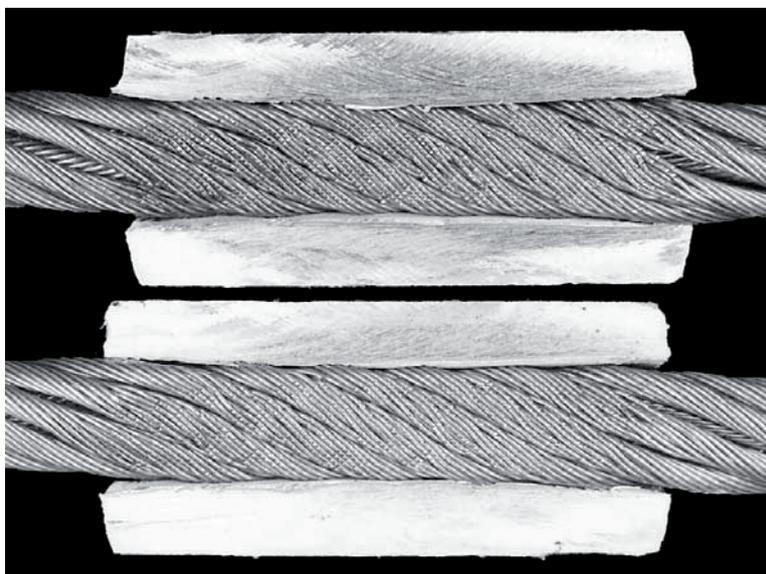


Fig. 67

6.2. The US Federal specification for spelter sockets

According to the US Federal Specification RR-S-5500 (Fig. 68), a spelter socket must have 1, 2 or 3 grooves in its pocket to prevent the metal cone from coming out of the socket if the rope gets suddenly unloaded.

TABLE III. Number and the size of the grooves in wire rope sockets.

Socket size	Grooves	
	Number	Depth (approximately)
Inches		Inch
1/4 to 3/4, inclusive	1	1/16
7/8 to 1-1/2, inclusive	2	1/8
1-5/8 and over	3	3/16

Fig. 68

What the makers of the Standard have missed is that these grooves not only prevent the socket from coming *out* when it gets unloaded, it also prevents the cone from coming *in* when the rope gets loaded.

The cone of a spelter socket can only hold the rope safely if under load it is pulled into the pocket so that it gets wedged, so that with increasing line pull the rope will get wedged more and more and will get held with greater and greater force.

If the cone can not pull into the socket the holding force is limited to the bonding forces between the rope wires and the socketing material, and the rope might pull out of the socketing material.

More than 20 years ago I wrote a brochure about end connections, and there I described the problem with the grooves of the Federal Specification. A few months later the largest producer of these sockets with grooves contacted me and complained that since I had published my brochure their sales for spelter sockets had dropped severely because customers now doubted their safety.

They agreed with my arguments, however. So I proposed they should simply change the Federal Specification and get rid of the grooves in their sockets. They answered that in the US nobody, not even the president of the United States, could change a Federal Specification.

But then these engineers had a brilliant idea: The Specification only defines the number and the required depth of the grooves, but not the thickness (Fig. 68). So they decided they would make the grooves so thin that they would shear off at the first loading. Then the cone would have no rings penetrating into the socket, and it would pull into the socket under load like in any other socket without grooves. And this is what they did (Fig. 69).

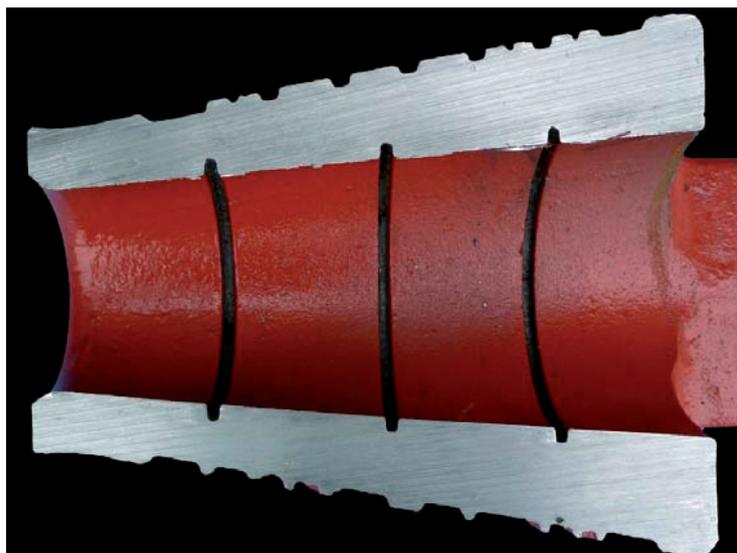


Fig. 69

But there are many other sockets on the market with thick grooves, and for those end connections the concern remains.

The manufacturer of the resin socketing material Wirelock© addresses this problem in their instructions for use. They require that large grooves in sockets must be filled with putty before pouring resin into the socket (Fig. 70).

The idea behind both concepts is the same: We cannot get rid of the grooves, but we can make sure they don't do any harm.

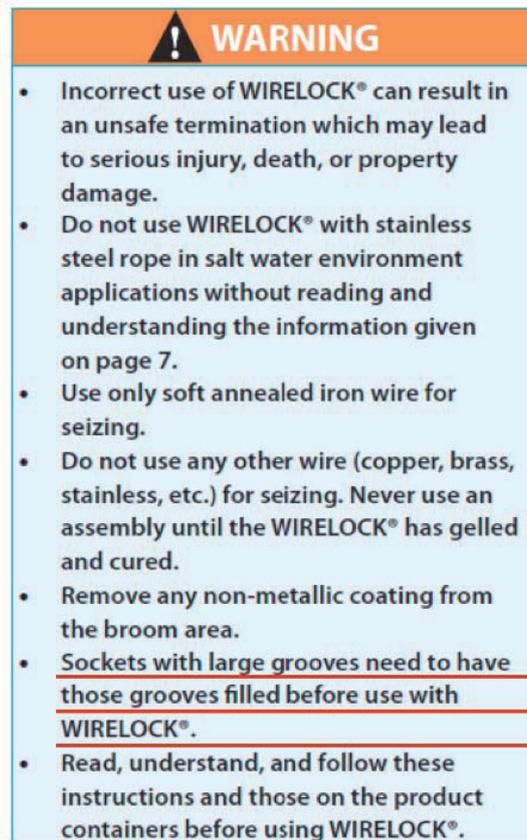


Fig. 70

6.3. The wedge socket

Murphy's law says that if something can go wrong it will go wrong.

Something did go wrong a number of years ago when a worker took the pocket of a wedge socket out of a box, then got distracted by a colleague, and finally he took the wedge out of a box. What he did not realize was that he had taken the wedge out of the wrong box. The wedge was for a different rope diameter than the pocket. A few days later a man got killed when the hoist rope of a crane pulled out of the socket because the wedge was too small.

This can not happen with the assembly shown in Fig. 71: The pocket, the wedge and the rope clip fit the same rope diameter and are not delivered individually but as one assembled piece.



Fig. 71

Another thing you could do wrong with an asymmetric wedge socket (the name asymmetric already indicates that) is that you install your dead end on the wrong side of the pocket. The assembly in Fig. 71 only allows you to clamp the dead on one side (which is the correct side).

It is probably not true for the offshore industry, but on land today workers are less and less educated, so we need more of these fool-proof solutions.

7. Conclusion

The last 40 years of the rope and crane industry have been very interesting, and I am pleased that I could take an active part in it.

We have solved many of the problems surrounding our products, but don't worry, we have left enough challenges for the coming generation of wire rope and crane engineers.

Wire Rope Technology Aachen

Dipl.-Ing. Roland Verreet

Grünenthaler Str. 40a • 52072 Aachen • Germany

Phone: +49 241- 173147 • Fax: +49 241- 12982

Mail: R.Verreet@t-online.de

www.ropetechnology.com