WIRE ROPE
TECHNOLOGY AACHEN

How to optimize the rope force distribution in a capstan

How to optimize the rope force distribution in a capstan

by Dipl.-Ing. Roland Verreet

Content

Title cartoon: Rolf Bunse
Layout and typesetting: Benedikt Dolzer, Aachen

1. Introduction

A capstan can be seen as a black box which acts as a force amplifier or a force reducer for a rope or any other elongated body (referred to as rope from now on).

The rope might, for example, enter the black box on the high tension side with a line pull S1 and leave it on the low tension side with a line pull of S2 (Fig. 1) when lifting a load on the high tension side.

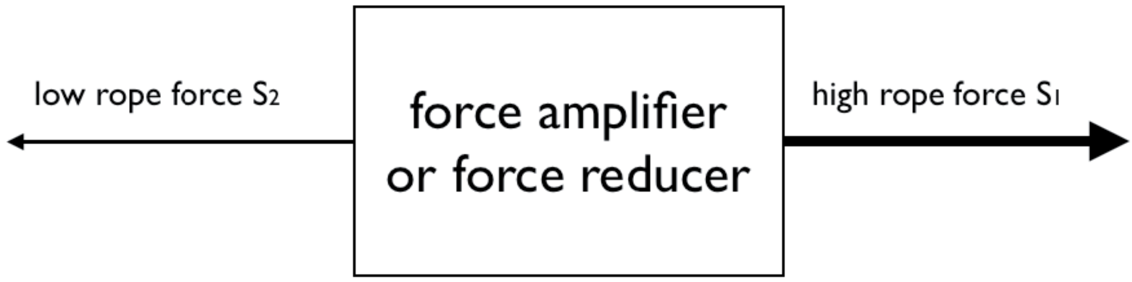

Fig. 1

Or it might enter the black box on the low tension side with a line pull S2 and leave it on the high tension side with a line pull of S1 (Fig. 1) when lowering a load on the high tension side.

The force amplification or force reduction will follow the Euler and Eytelwein formula

$$S_1 / S_2 \leq e^{\mu\alpha} \qquad \text{(Formula 1)}$$

with

S1 rope force on the high tension side
S2 rope force on the low tension side
μ coefficient of friction between rope and sheave
α angel of wrap around the traction sheave

The maximum rope force amplification or rope force reduction will be reached when

$$S_1 / S_2 = e^{\mu\alpha} \qquad \text{(Formula 2)}$$

and the rope will start sliding. In the following, I will call this factor S1/S2 the amplification factor or the reduction factor, depending on the direction of travel of the rope.

In the following, a coefficient of friction of 0,125 is used for the examples.

Because sliding of the rope in the black box is not desired, capstan systems (the black boxes) are typically overdesigned. If, e.g. for the maximum load case an amplification or reduction factor of 23 is needed (which for a coefficient of friction of 0,125 would require 4 full turns around a double drum capstan, see Fig. 2), the double drum capstan might be designed so that the rope will go around the drums with 5 full turns (Fig. 3), leading to an amplification or reduction factor of 50, which is more than twice of what is needed.

This will create two more bends for the rope travelling through the capstan system, but this seems a small price for having gained so much more safety against sliding.

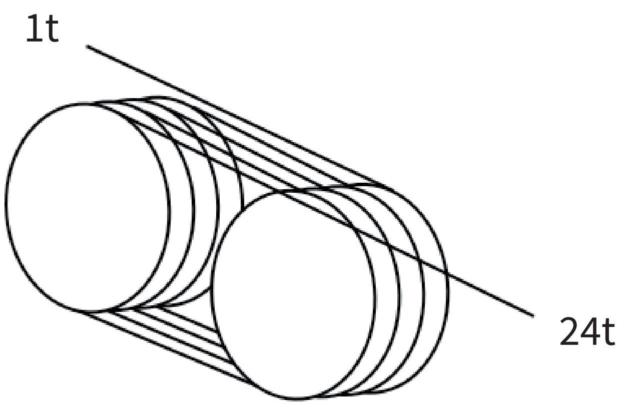

Fig. 2

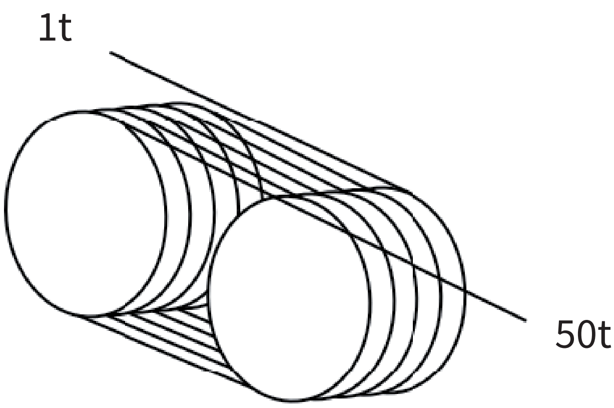

Fig. 3

For smaller loads than the maximum load, the system will be overdesigned even further. The black box (e.g. a double drum capstan system) might need an amplification factor or reduction factor of 10 (requiring 3 full turns around the double drum capstan), but it will travel around the drums 5 times, doing four unnecessary bending cycles in the process.

This, hower, is usually also seen as a small price to pay for then having a safety factor against sliding five times as high as would be necessary.

Overdesigning a capstan system, however, creates a number of problems, some of which are not understood by most of the capstan designers, and it might even reduce the safety of the system.

2. The first problem: Bending fatigue

In capstan system typically many sheaves are arranged relatively close to one another. They might be connected (such as in a double drum capstan) or individually driven, or they might be a combination of these arrangements. If the motion of the rope is often reversed (such as in a heave compensated operation where the rope travels back and forth through the capstan in order to compensate for the heave motion on the vessel it is on), the rope will travel over a greater number of sheaves under high line pull and be subjected to a greater number of bending cycles than necessary. This might lead to a premature failure of the rope.

3. The second problem: Heat generation

When travelling back and forth through a capstan in order to compensate for the heave motion, the affected rope sections will be bent every time they enter a sheave and they will be straightened again when they leave the sheave. Due to the friction between the rope elements and between the rope and the sheaves, the affected rope sections and the capstan sheaves will heat up. This might lead to a loss of lubricant and, as a consequence, to an accelerated degradation of the rope. Heat might also start a strain ageing process in the rope wires. Both these effects become more relevant with increasing rope diameters.

4. The third problem: Line pull peaks in the system

Fig. 4 shows the rope force for a double drum capstan system in which the rope travels around the two drums seven times (or 14 times 180°). The rope force on the low tension side is set to 1t.

The rope force on the high tension side could be 244t before sliding occurs (amplification factor 244).

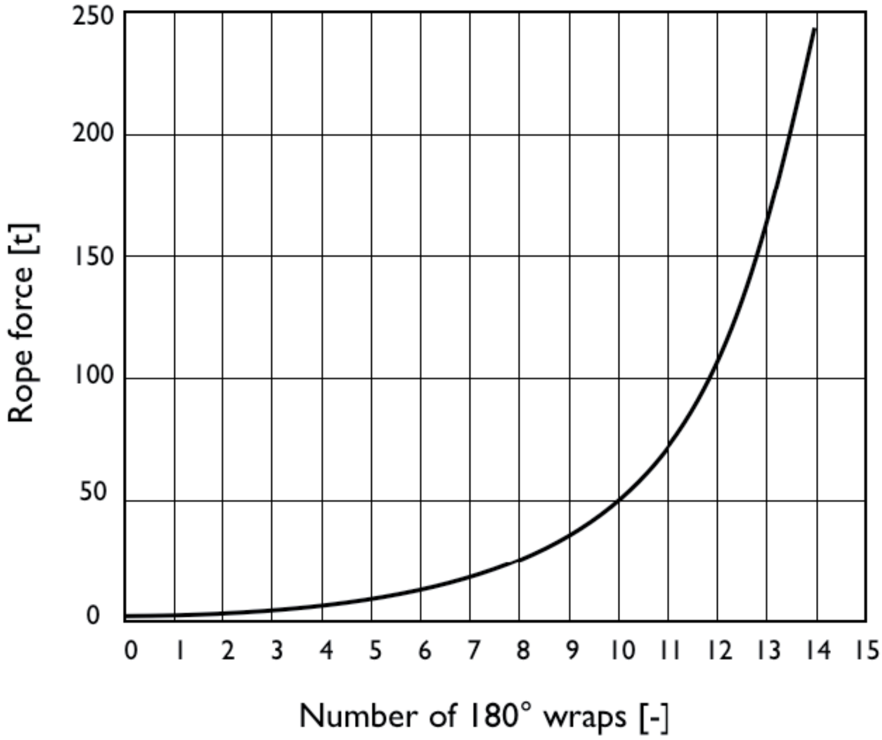

Fig. 4

The system is designed for lifting loads with a line pull of 50t on the high tension side (amplification factor 50) which would require 5 turns (or 10 x 180°) only. Again, the extra safety against sliding achieved by adding two more turns around the drums is considered to be a bonus.

Fig. 5 shows the theoretical force reduction along the rope arcs around the drum from the low tension side to the high tension side. The rope rope enters on the right hand side and travels over sheaves 14, 13, 12 and 11 with the high force S1 (which is 50t). Then, according to Leonard Euler´s and Johann Albert Eytelwein´s theory, the rope force will reduce according to the e- function shown in Formula 1 until it reaches the lower force S2 (which is 1t).

So according to the established theory which is the basis for all capstan designs the maximum rope force in the system will always be found on the entrance (when lifting) or the exit (when lowering) of the last sheave on the high tension side.

In practice, however, sometimes rope force distributions as in Fig. 6 can be found: An e-function builds up from both sides of the capstan, creating a force peak inside the capstan. In the case of our example, the force peak on the 12th sheave will be more than twice as high as the rope force on the high tension side.

The rope travelling through the capstan will not only do 4 bending cycles which are not required to prevent the rope from sliding, these bending cycles will even be done under loads higher than what is considered the maximum line pull in the system.

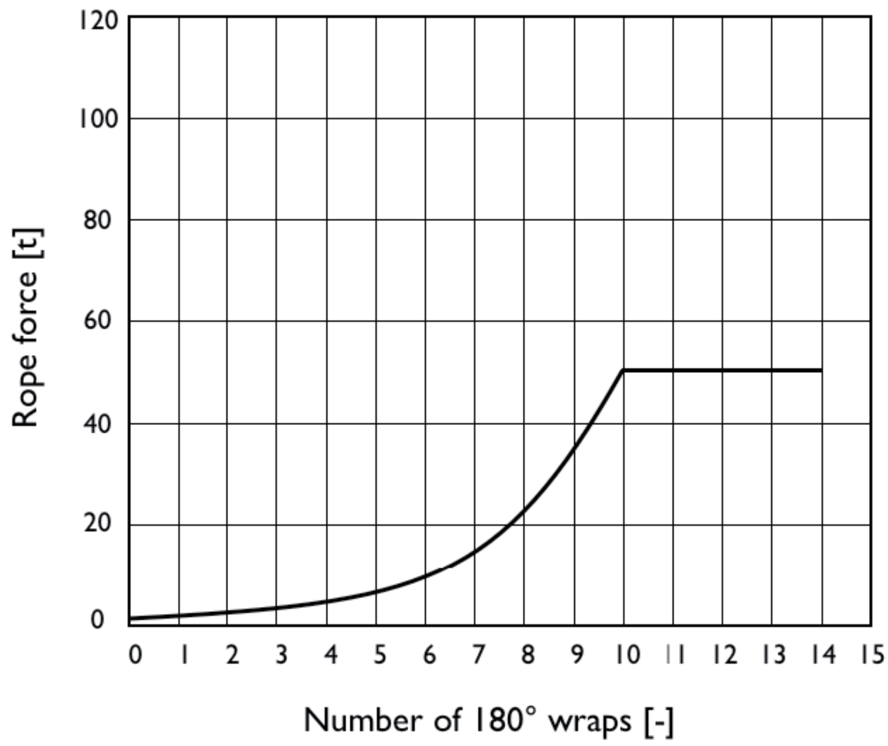
Rope force [t]
120
100
80
60
40
20
0
0 1 2 3 4 5 6 7 8 9 10 11 12 13 14 15
Number of 180° wraps [-]

Fig. 5

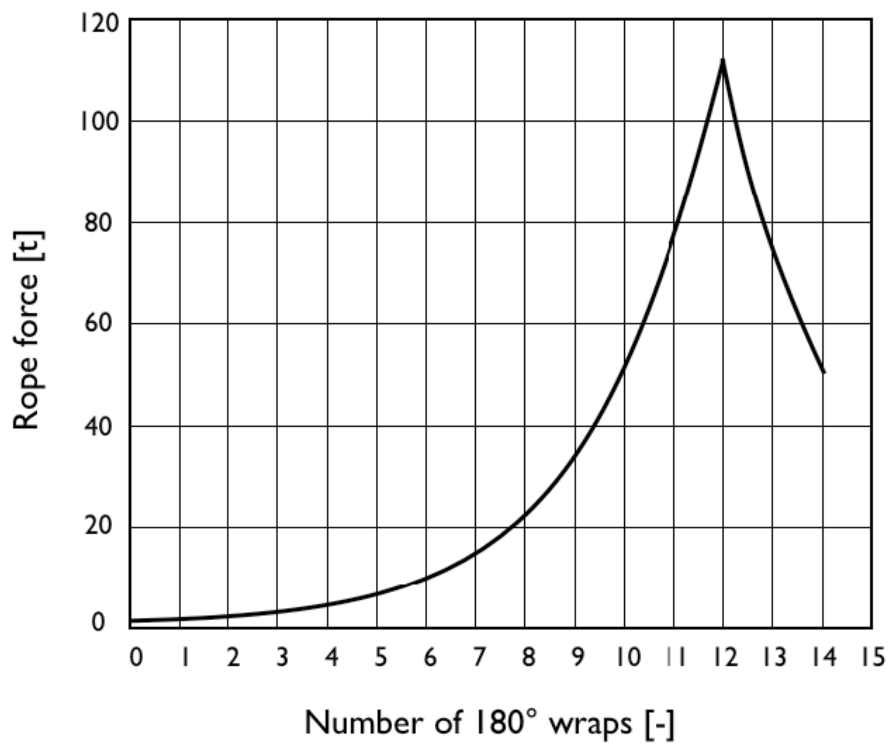
Rope force [t]
120
100
80
60
40
20
0
0 1 2 3 4 5 6 7 8 9 10 11 12 13 14 15
Number of 180° wraps [-]

Fig. 6

The combination of performing unnecessary bending cycles in the system and the fact that these bending cycles will be performed under much higher line pulls than necessary will always lead to much higher bending fatigue and, as a consequence, to a considerably reduced rope life.

In several cases the line pulls have increased to a level higher than the breaking strengths of the ropes which led to overload failures of the ropes within the capstan system.

Some failures of capstan axes might also be a consequence of the fact that the rope forces in the system were much higher than what the designer had anticipated.

5. The fourth problem: Shock waves and impacts

Fig. 7 again shows the rope force as a function of the arc around the capstan for a double drum capstan with 7 full rope turns around the drum. The rope force on the high tension side is again 50 t, the rope force of the low tension side is 1t. The capstan works in heave compensation mode, continuously lifting and lowering a load in order to compensate for the ship´s motion.

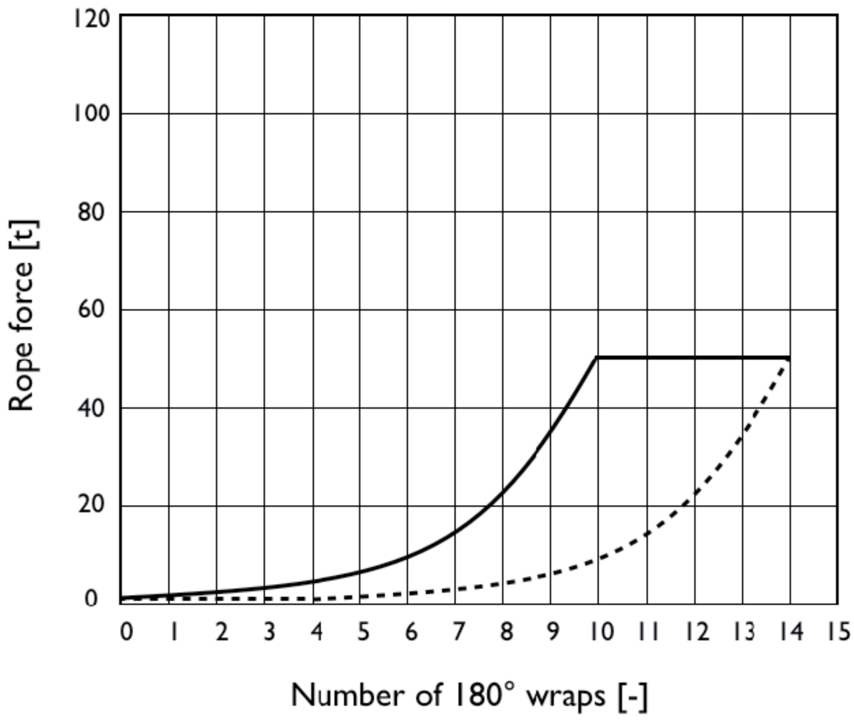

Fig. 7

When lifting the load, the rope will come in on sheave number 14 (on the right hand side of the diagram) and travel with the same line pull over sheaves 14, 13, 12 and 11 until the rope force will go down according to an e-function over sheaves 10 to 1 (solid line).

When lowering the load, the rope will come in on sheave number 1 (on the left hand side of the diagram) and travel over the first, second, third and forth sheave with a constant line pull of 1t until the rope force will go up according to an e-function, reaching a line pull of 50t at the exit of the last sheave (dotted line).

When lifting, the rope force will follow the solid line, and a rope section on the tenth sheave will have a line pull of 50t (point A in Fig. 8).

When lowering, the rope force will follow the dotted line, and a rope section will now have a line pull of only 10.4t (point B in Fig. 8). The rope force in point A is 4.8 times as high as in point B.

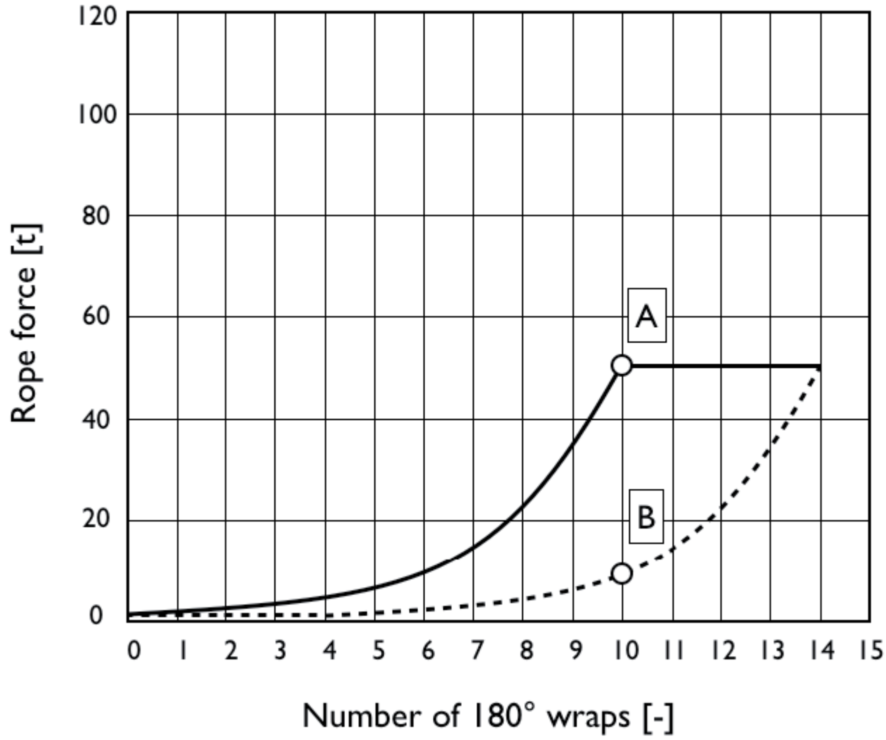

Fig. 8

When the rope is repetitively lifting and lowering, e.g. in heave compensation mode, the rope section must alternatively be in point A and point B. This means that every time the motion of the rope is reversed, the rope force in this section will either have to be increased to almost five times the initial load level or lowered to about a fifth of the load level.

This will lead to a great amount of tension-tension fatigue which will add up to the unnecessary bending fatigue (caused by the superflous sheaves explained above). In addition the rope will continuously change its length while lying on a sheave in order to adapt to the changing line pulls, causing abrasion both on the rope and on the capstan sheaves.

One possible solution to overcome the above mentioned problems is to disconnct the sheaves which are not required from the capstan and to connect them again to the capstan whenever they are needed. It is important in this case to disconnect the sheaves on the low tension side.

Another possibility would be to control all sheaves independently. Such a solution, however, requires an intelligent control system with information about all rope forces in the system, and it is very costly.

Another new proposal, which will be discussed here in more detail, is to control the rope force distribution in the capstan system by increasing or reducing the line pull on the low tension side.

Fig. 9 shows the level of the rope forces in the capstan described above for a rope force on the high tension side of 50t and for a rope force on the low tension side reduced to 0,4t. The points A and B again show the rope force on the 10th sheave for lifting (point A) and lowering (point B).

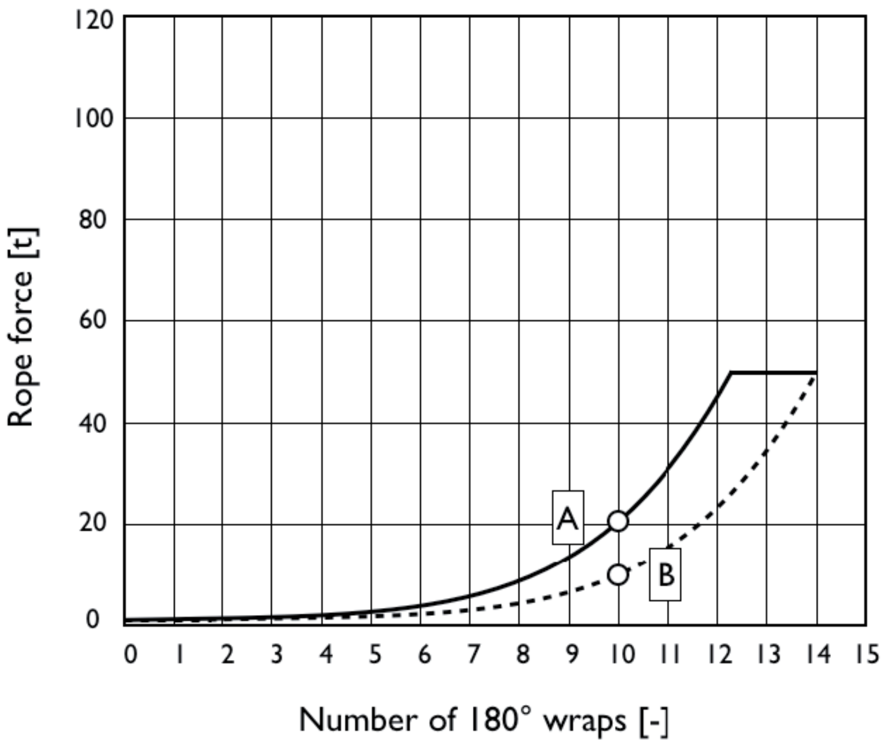

Fig. 9

As can be seen, the distance between point A and point B has become much smaller. The rope force when lifting (point A) is now reduced from 50t to 20.3t, but it is still higher than the force when lowering which is still 10.4t.

If the force on the low tension side is reduced to 0,205t, the curves for lifting and lowering and also points A and B become the same (Fig. 10).

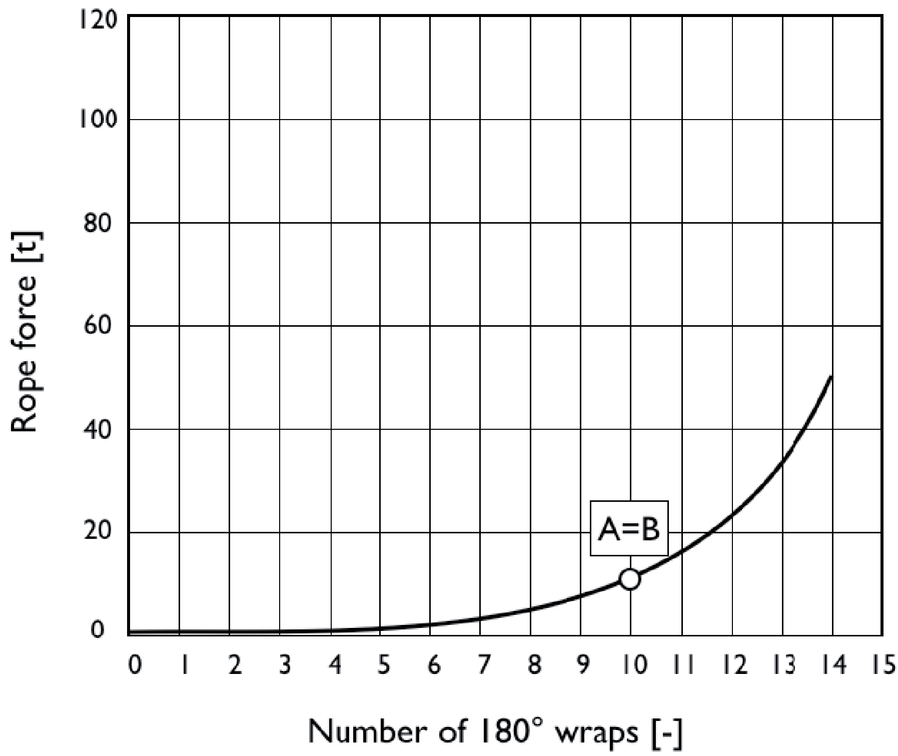

Fig. 10

And once the curves for lifting and lowering become the same, no major changes in line pull will occur when changing from lifting to lowering.

The two curves become identical when the rope force on the low tension side is equal to the rope force on the high tension side divided by the amplification or reduction factor of the black box (the capstan system).

$$S_1 / S_2 = e^{\mu\alpha} \qquad \text{(Formula 2)}$$

This means that the system must always be operated at or near the force at the low tension side for which the rope starts sliding.

It must be noted here that sheave liner materials with a higher coefficient of friction (such as Becorit which is widely used on friction winders in the mining industry) will reduce the number of necessary sheaves tremendously.

So controlling the rope force on the low tension side as a function of the loads applied on the high tension side and at the same time using sheaves with such liner material might reduce the number of sheaves and therefore also the amount of bending fatigue caused by the capstan tremendously.

Another favourable execution of a capstan system which is controlled by varying the line pull on the low tension side is that this tension can be adjusted when the motion of the rope is reversed.

As an example, the system is working as shown in Fig. 9. When the system is lifting, the rope enters sheave 14 and travels towards sheave 1, establishing a force distribution along the rope as shown by the solid line. The line pull reduces from 50t to 1t in the process.

When the motion of the rope is reversed because the mode of operation changes from lifting to lowering, the rope will come in on sheave 1 and travel towards sheave 14. The line pull will change from 1t to 50t in the process, and the distribution must follow the dotted lines because intermediate load levels between the solid and the dotted line will not be stable. This is because according to the law of Euler and Eytelwein the force along the rope is either constant or follows an e-function.

When the motion is reversed, rope sections which are in the system, e.g. on sheave 10, must change their line pull from 20,3t (point A) to 10,4t (point B). This will sometimes happen very suddenly, subjecting the rope and the whole lifting system to dynamic loads. When the changes are small, the elasticity of the rope will reduce the effect.

If, however, the line pull on the low tension side is increased at the moment when the motion is reversed, the outward motion will have a stable force distribution as as shown by the dotted line in Fig. 11.

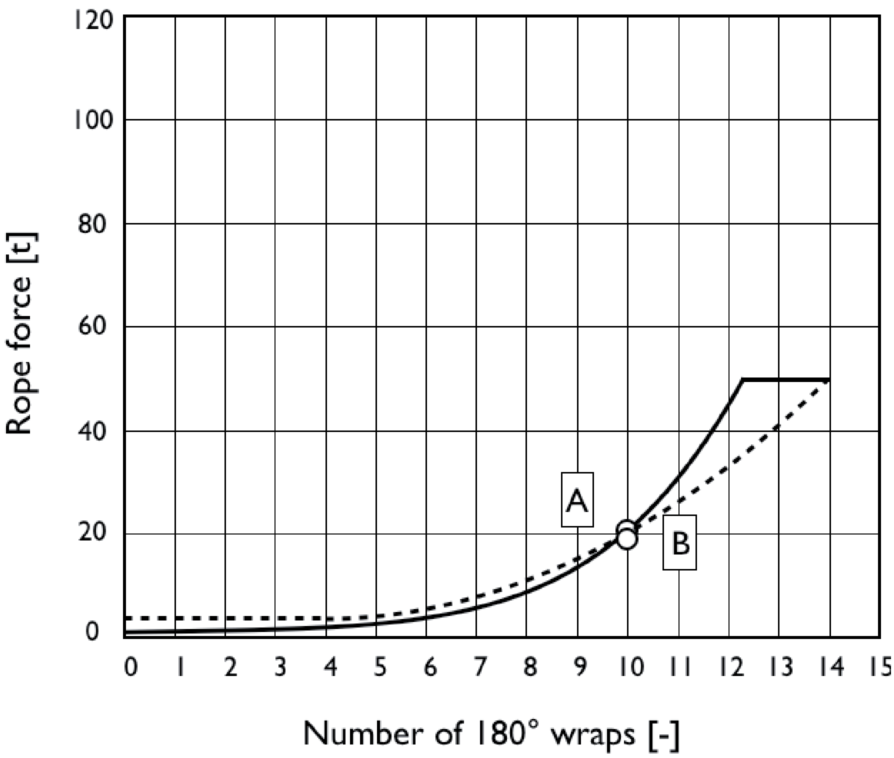

Fig. 11

Compared to the dotted line in Fig. 9, the load *level* is increased, which will increase the amount of bending fatigue, but the *changes* in load level when reversing the motion of the rope will be reduced tremendously.

So if the load level on the low tension side is reduced to a minimum when lifting and then raised again when lowering, the effects discussed above can be combined: the general load level will be smaller than in a system without these control mechanisms, and the changes in load level when reversing the motion will be reduced as well.

Operating near the load level which creates the same load distribution for lifting and for lowering (see Fig. 10) and slightly increasing the load level on the low tension side when lowering will bring both the load level and the load changes to a minimum and still guarantee a safe operation.

Fig. 12 shows a lowering system with a load, a load cell in the sheave axis, a sheave, a capstan, a guide sheave with a load cell in the sheave axis and a storage drum.

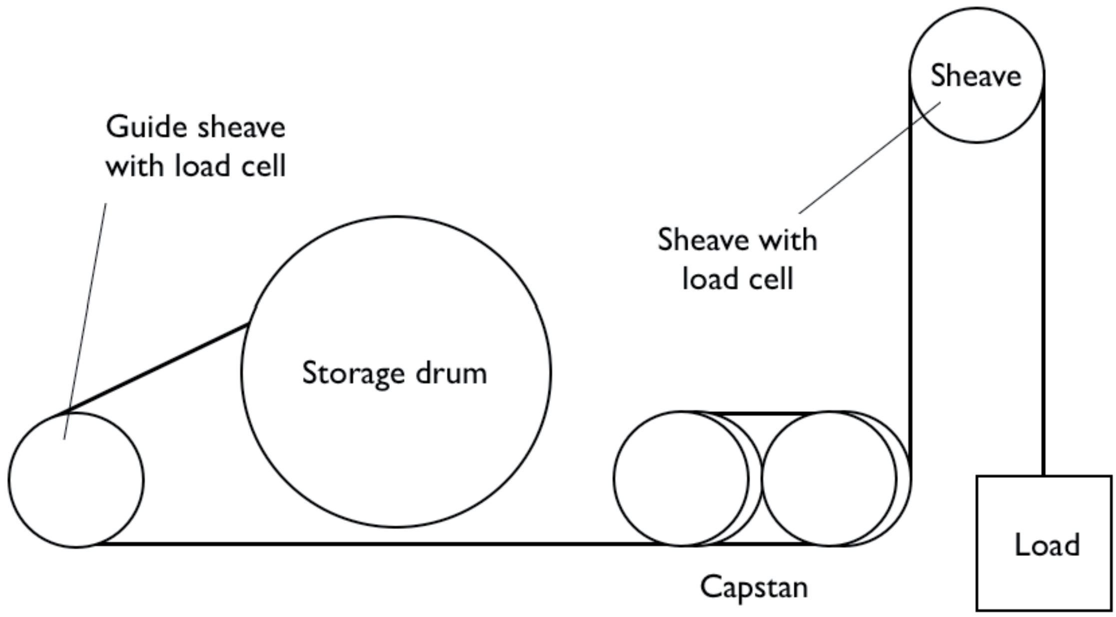

Fig. 12

The rope force on the low tension side could be controlled by the storage drum directly. For lifting and lowering operations, this would be sufficient. In heave compensation mode, however, this would require the storage drum to continuously rotate back and forth which might be a problem because of the great inertias involved.

It is therefore advisable to have a tensioning system between the capstan and the storage drum (Fig. 13) which again could be equipped with a load measuring system.

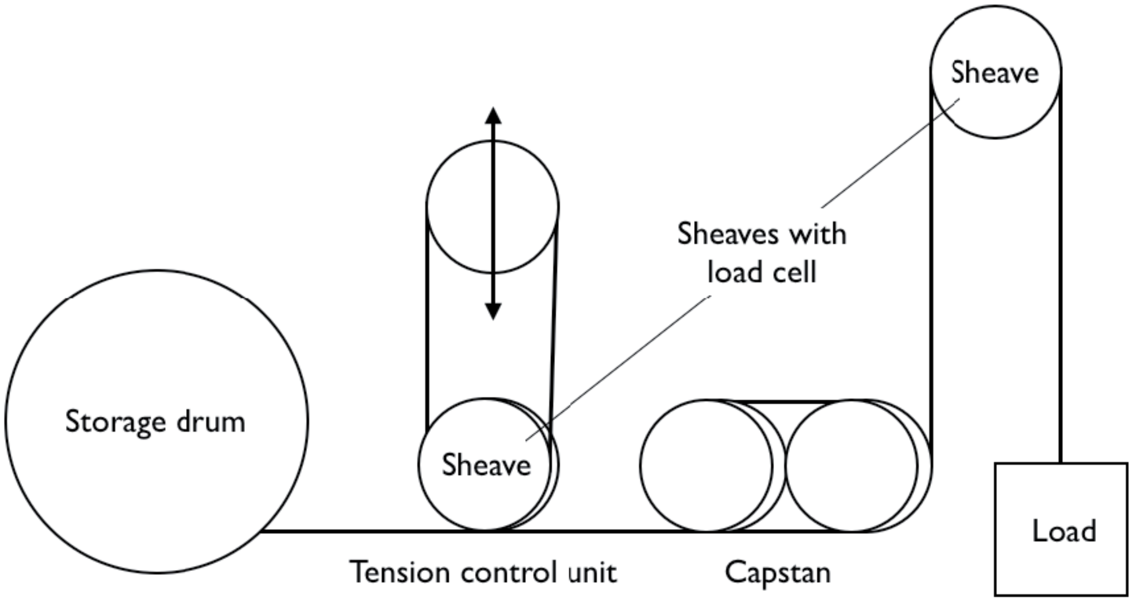

Fig. 13

In heave compensation mode, only the capstan and the tensioning system would move while the big masses of the payload and the drum with the rope on it will not.

After a few lifts or test lifts with the system, an intelligent control mechanism will know what the amplification or reduction factor of the system is, maybe even how the factor will change when using a dry rope or a wet rope. It could then just measure the line pull at the high tension side, divide it by this factor in order to determine the force at which the rope will start sliding and keep the line pull on the low tension side slightly above this level.

Alternatively, a starting value for the amplification factor can be manually set and reduced until the rope starts sliding. Increasing the rope force slightly again will then secure the load and ensure that the system is operated with the optimum rope force distribution.

It must be noted in this context that in a heave compensation mode the rope forces will not change to a great extend because the load will be kept in the same location and only ship motions relative to the load are compensated.

A great advantage of the solution proposed here is that every overdesigned existing capstan system could be fitted with this control system and rendered capable of reverse motions. The number of wraps around the capstan could be reduced, and the mechanism controlling the back tension would ensure a safe operation.

Once the existing capstan is equipped with such a control mechanism on the low tension side it will operate with much lower rope forces than before. This will lead to much greater rope life and reduced abrasion both on the rope and on the capstan sheaves.

Peak loads in the system will no longer occur, which will greatly improve the safety of the capstan operation.

The rope forces will easily be controlled by controlling the forces on the low tension side: the system reacts in a a more or less linear way.

It would be much more difficult, however, to control a capstan system with varying coefficients of friction: the system will react in an exponential way.

If, for example, the coefficient of friction between the rope and the sheave changes when the rope gets wet, the ideal rope force S2 might change in a disproportionate way. If the system cannot react fast enough on such changes, it might be a solution to always operate the system with the lower coefficient of friction (which is typically when the rope is wet).

So in an offshore application it might be advisable to keep the rope section operating in the capstan in heave compensation mode wet at all times. This might also help to reduce the rope temperature, although operating at a minimum level of line pull will already reduce the amount of heat generated during heave compensation.

The control system could be “learning” from previous operations so that after some operations the system will automatically adjust the rope forces to the new conditions.

If the double drum capstan is used for heave compensation, the method of operation explained in this paper will reduce the amount of heat generation tremendously:

In an overdesigned system, the average line pull (and therefore the amount of fatigue and the amount of heat generated) will be very high (Fig. 14).

If the same system is operated close to the point of slippage, the line pulls (and therefore the amount of fatigue and the amount of heat generated) will be much lower (Fig. 15).

In addition, the amount of heat generated will decrease at every single 180° wrap and therefore create a temperature gradient. This will allow the heat generated to flow to cooler areas (Fig. 15).

Such a gradient will not develop, however, in conventional overdesigned systems. If the neighbouring 180° wraps generate the same amount of heat, there will be no temperature gradient and the system will heat up tremendously (Fig. 14).

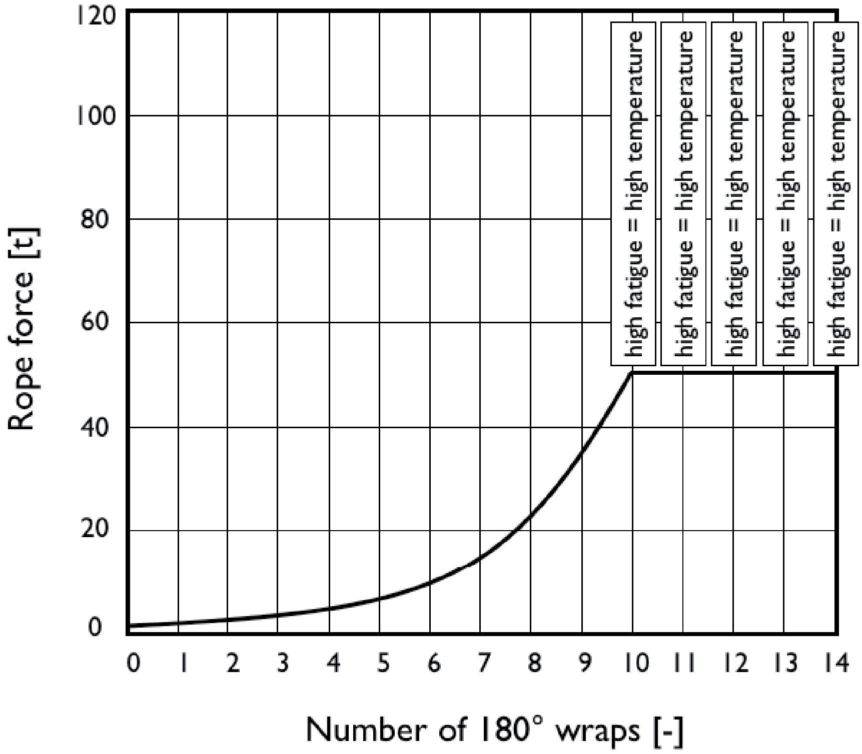
120
100
80
60
40
20
0
Rope force [t]
0 1 2 3 4 5 6 7 8 9 10 11 12 13 14
Number of 180° wraps [-]
high fatigue = high temperature
high fatigue = high temperature
high fatigue = high temperature
high fatigue = high temperature
high fatigue = high temperature

Fig. 14

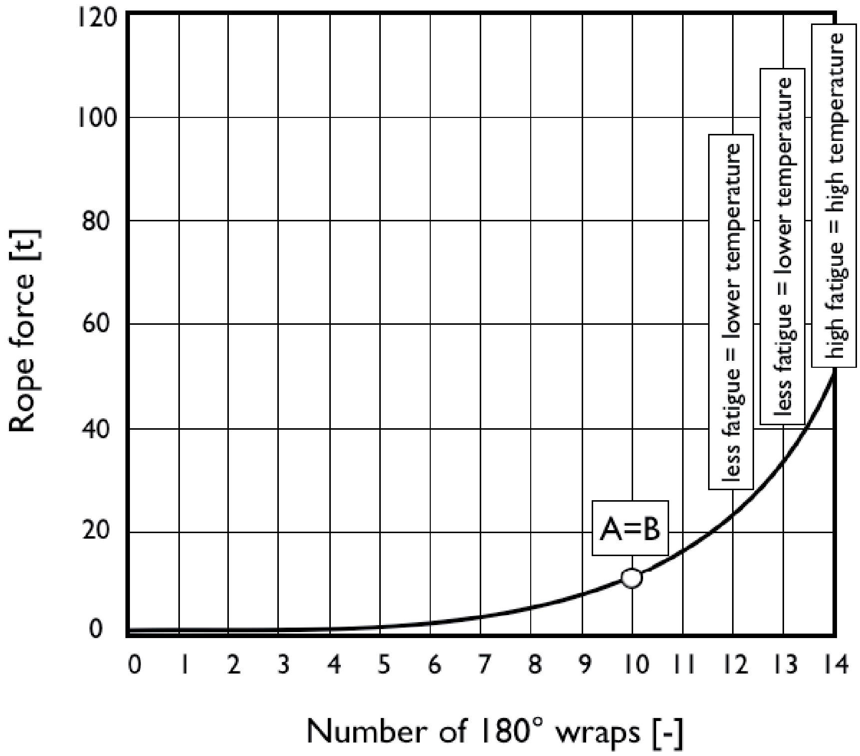
120
100
80
60
40
20
0
Rope force [t]
0 1 2 3 4 5 6 7 8 9 10 11 12 13 14
Number of 180° wraps [-]
A=B
less fatigue = lower temperature
less fatigue = lower temperature
high fatigue = high temperature

Fig. 15

The author welcomes any comments on this paper. Please contact:

Wire Rope Technology Aachen
Dipl. - Ing. Roland Verreet
Grünenthaler Str. 40a
52072 Aachen
Germany

r.verreet@t-online.de
www.ropetechnology.com

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