On friction for traction of elevators.

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ABSTRACT.

If the amount of friction is not critical the use of the formula of Eytelwijn/Euler to describe the relation between friction and traction give a workable base for the design.
Under more severe demands the used simplifications of the rope surface, the contact forces and the friction coefficient can lead to less successful installations due to lack of friction, excessive traction and wear.

The main subjects of this study, which is carried out under the auspitien of the Dutch LIFT INSTITUUT, are the relation of the friction coefficient, the creep velocity and the contact forces at the wires of a wire rope.

1. INTRODUCTION.

Friction may be simply clarified with theoretical solutions like $W = \mu * N$, as soon as the shape of the bodies in contact are not simple, unexpected surprises may occur.

Two centuries ago, about thirty years before Albert used the first wire rope in mines, Eytelwein and Euler developed their theoretical solution for the friction between a driving belt and a driving wheel.

$$\frac{T_1}{T_2} = e^{\mu \cdot \alpha} \qquad (1)$$

with belt forces $T_1 > T_2$, μ = friction coefficient, α = creep $\angle$.

It lasted until 1927 before Donandt [1], Hymans and Helleborn [2] adapted the formula (1) to the traction with wire ropes.

The friction coefficient μ became the apparent friction coefficient f, which also took in consideration the influence of the different directions of the contact pressure. In fact the apparent friction coefficient f was the product of the friction coefficient μ and a groove factor g.

The derivations of the pressure on the surface of the groove and the groove factors f were based on the idea, that the reaction of the contact pressure $p(\varphi)$ would give a uniform surface pressure on the bottom of the groove like in figure 1.

$$\frac{p(\varphi)}{\cos\varphi} = \text{Constant.} \qquad (2)$$

This assumption let the vertical displacement of the wire rope be equal along the groove, or:

Pressure $p(\varphi)$ can be derived from the vertical equilibrium with $dN = Td\theta$.

$$dN = \int p(\varphi) \cdot \cos\varphi \cdot dA \qquad (3)$$

With D for the sheave and d for the rope diameter, dA becomes:

$$dA = \frac{D}{2} \cdot d\theta \cdot \frac{d}{2} \cdot d\varphi \qquad (4)$$

$$dW = f \cdot dN = \mu \cdot \int p(\varphi) \cdot dA \qquad (5)$$

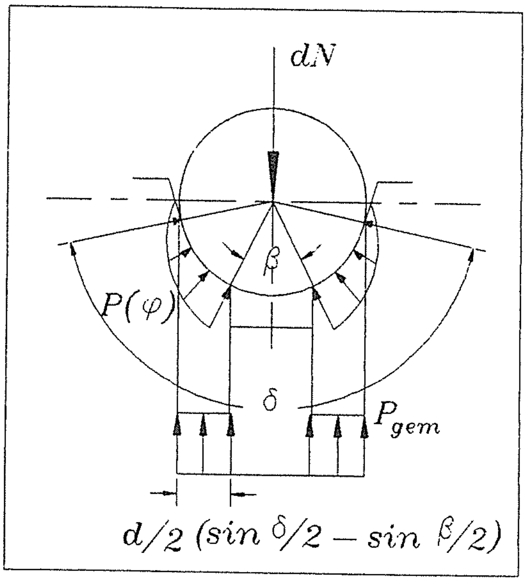

Fig.1. Contact pressure at the groove.

The friction dW in tangential direction of the sheave, which causes the increase of the rope force dF, will be:

$$g = \frac{f}{\mu} = \frac{\int p(\varphi) \cdot dA}{dN} \qquad (6)$$

This means :

$$Td\theta = \frac{D \cdot d}{4} \cdot d\theta \cdot \int_{*}^{*} C \cdot \cos^2 \cdot d\varphi. \qquad (7)$$

The formulas (2) and (3) give the next integral. The equation (7) leads after the substitution of the correct limits and some conversions to the solutions for the various kinds of grooves.

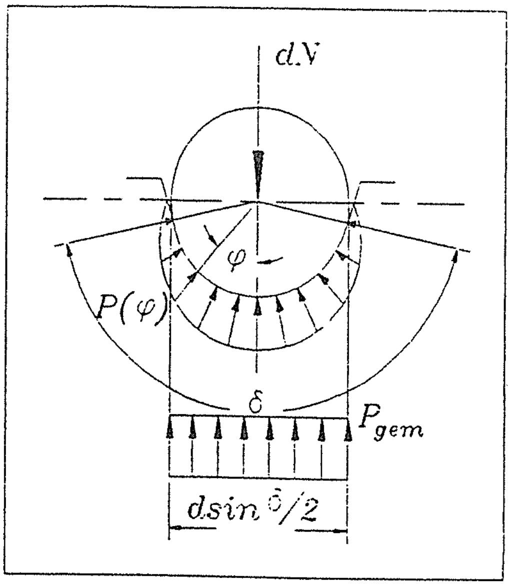

Fig. 2. The under cut groove.

For the round groove like figure 1 the groove factor is:

$$g = 4 \cdot \frac{\sin\frac{\delta}{2}}{\delta + \sin\delta} \qquad (8)$$

If the limits of figure 2 are used in the formula (7), the expression for the groove factor of the under cut grove is found.

$$g = 4 \cdot \frac{\sin\frac{\delta}{2} - \sin\frac{\beta}{2}}{\delta + \sin\delta - \beta - \sin\beta} \qquad (9)$$

In the literature some times the groove factors differ from the formulas (8) and (9) and have the shape:

$$g = \frac{4}{\pi}, \text{ or } g = 4 \cdot \frac{1 - \sin\frac{\beta}{2}}{\pi - \beta - \sin\beta} \qquad (10)$$

In this case was supposed that the groove has been worn in such a way, that the contact angle of the wire rope should be 180°.

After the twenties many authors made research into this subject. Among these Kuhn [3], Molkof [4] and Aberkrom [5] made very valuable contributions to the knowledge of traction with wire ropes. Never the less some questions were left.

It is too easy to say that the cause of this shortage lies with the approach of their experiments, but it has to be taken in mind that the possibilities for measurements and the working up developed very fast the last ten years.

The tendency to perform experiments which simulated the reality, had the advantage of including almost all the influences. But this made the estimation of the influence of each very difficult and some times impossible.

Now with the knowledge of the past we can evaluate the separated questions and try to solve these as good as possible, in making good use of the nowadays possibilities.

2. THE AIM OF THIS RESEARCH.

If the above mentioned theories are used for the design of traction sheaves of elevators this often will not give much difficulties. But if there are special circumstances, specific problems can rise.

The capacity of the traction for example can be too much if occasionally one of the rope parts become slack.

The intention to prevent the slip of the wire rope makes that large reserves with regard to the creep angle are built in.

It was the aim of this study to investigate how much unknown traction capacity may be reserved, if the traction is calculated in the traditional way.

The formulas (5) and (6) indicate that the real friction coefficient μ as well as the groove factor g could be the cause for unexpected large or small values of the creep angle and so for too much or too less traction capacity.

Therefore this paper written under the auspitien of the Dutch LIFT INSTITUUT deals with experiments on friction and on the value of the groove factor.

3. THE FRICTION COEFFICIENT.

The European Standard EN 81-1 recommends for all cases a friction coefficient of $\mu = 0.09$ and neglect the following influences on the friction:

The relative velocity of the wire rope on the traction sheave. The amount of tension force at the wires.
The amount of pressure.
The material of the sheave.
The wire material of the rope.
The condition of the surfaces in contact.
The lubrication.

To investigate all these possible influences at an elevator is hardly impossible. Therefore tests on friction coefficients are performed at single wires, which were suspended at a dynamometer and loaded with different weights.(Figure 3)

The wire is pressed in such a way by a steel roller against the round surface of a turning sheave, that the wire remains straight. The pressure force is caused by weights, which acts via an angle lever. This lever can be moved very accurately towards and from the sheave. In this way the load at the wire and the changes of it caused by the friction can be measured with the dynamometer and directly be read out, without further changes in trimming of the test bench.

The difference of both the read outs is the friction force, which easily can be converted into a friction coefficient, in using the well known pressure force.

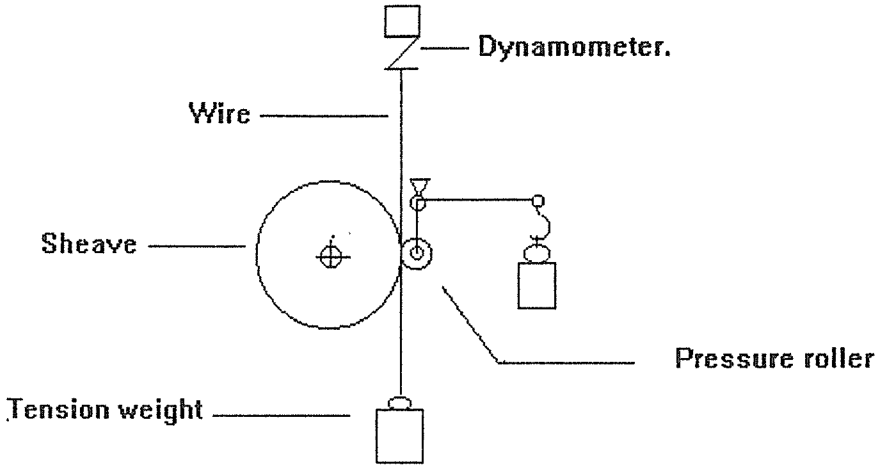

Figure 3. Scheme of the test bench.

The velocity of the sheave can be widely varied. Relative velocities between 0.022 mm/s and 4.294 mm/s are possible due to the application of a frequency controlled motor.

This makes the tests easy to perform and also more insensible for unknown influences. To investigate also the influence of the sheave materials and the lubrication, four sheave were mounted on the driving shaft.

An earlier research, which was mainly focused on the amount of wear of different material combinations, revealed that the combination of cast iron GG35 with wire of the class 1360 N/mm^2 gave rather good resistance against wear. Also during this tests was discovered that the friction of the combination nodular cast iron GGG60 and wire of the class 1770 N/mm^2 showed larger friction forces under comparable conditions. Therefore these two combinations were selected for this research.

Two sheaves of each material were mounted to measure the influence of the lubrication. One sheave of each material was lubricated with BEL RAY and the other was degreased.

The diameter of the sheaves was a compromise between the velocities wanted and the reduced curvature, calculated with the theory of Hertz. A sheave diameter of 125 mm together with a wire diameter of 0.8 mm was a good compromise. This 0.8 mm is the diameter of the outerwires of a 13 mm elevator rope in 8X19 S.

The tests were performed with eight relative velocities. The choice of these was based on experimental results of the creep velocity measurements and the amount of creep of one wire in a wire rope during one passage of the traction sheave. The results of these tests were reported at the ELEVCON congress in Amsterdam 1992 [6]. Traction ratios of 1.15 to 1.35 gave with low velocities of the rope a relative velocity of 20 to 35 μm/s. Supposing that this creep velocity will increase more or less linear with the velocity of the traction sheave, the maximum relative velocity during the tests must be about 4 mm/s to be comparable with an elevator velocities of 4 m/s.

The mentioned material combinations were tested with every selected velocity and each under four different tensile forces and four contact forces.

For the choice of the tensile forces was based on the safety factors 12 and 16. But it was also taken into the account that the traction force can only be spread on all the wires after this is introduced by the wires in contact with the sheave. Further the natural deviation of the tensile load in the different wires was taken in the calculation. This gave the following tensile forces: 67 N, 100 N, 134 N and 150 N.

On the basis of the reaction force at the sheave generated by the lower force F_2 on the rest angle, the pressure forces were calculated between the critical wire and the surface of a round groove and of an under cut groove. These calculations resulted in the choice of the following contact forces 100 N, 125 N, 175 N and 200 N.

During the preparatory tests the reproducibility appeared to be very good. But the condition of the material surfaces in contact gave a relatively large influence on the test results. Therefore the 1024 tests were performed with new wires as well as worn wires. The latter were flattened up till 0.1 mm. At the surface of the sheave appeared a clearly visible groove. The over two thousand measuring results were stored in a number of spreadsheets. This enabled the production of a large number of graphics. These were a great help for the analyses of the data obtained.

TABLE 1.

Average values of friction coefficients with creep velocities between 1 mm/s and 4 mm/s.				
Material combination	67 N	100 N	134 N	150 N
Material combinations both with new wires, dry and lubricated.	0.13	0.12	0.11	0.09
GGG60/1770 worn, dry	0.16	0.18	0.41	0.52
GGG60/1770 worn, lub.	0.22	0.19	0.17	0.16
GG35/1360 worn, dry	0.14	0.11	0.10	0.10
GG35/1360 as well as GGG60/1360 worn, lub.	0.19	0.18	0.15	0.13

To show all the measured friction coefficients and graphs out of the concerning report [7] is not efficient. Therefore a selection is made of the most interesting part of the average values and two examples of the graphs. The friction data are given in the table 1 and the graphs are shown in the figures 4 and 5.
The table is restricted to the coefficients which are measured with creep velocities as these happen with rope velocities of 1 up to 4 m/s. The influence of the contact forces are left because these appeared to be minor.

The large values of the combination GGG60/1770 with worn wires and without lubrication are unwished because of vibrations .

The figure nr. 4 shows the course of the friction coefficient as a function of the relative velocity.

It also appears that the friction coefficient decreased with increasing tensile force in the wire.

Contrary to the mentioned influence of the tensile forces, it was found that the pressure force had only minor influence.

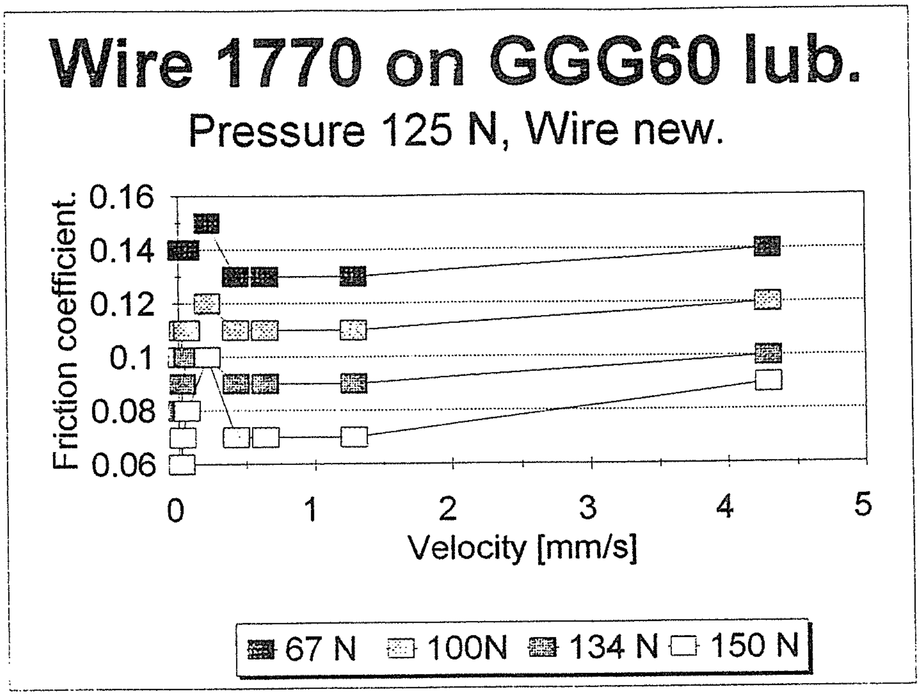

Fig. 4. Friction coefficients as a function of the relative velocity.

If cast iron GG35 is used, the lubrication did not decrease the friction coefficients and with nodular cast iron GGG60 it degreased not so far as was expected.

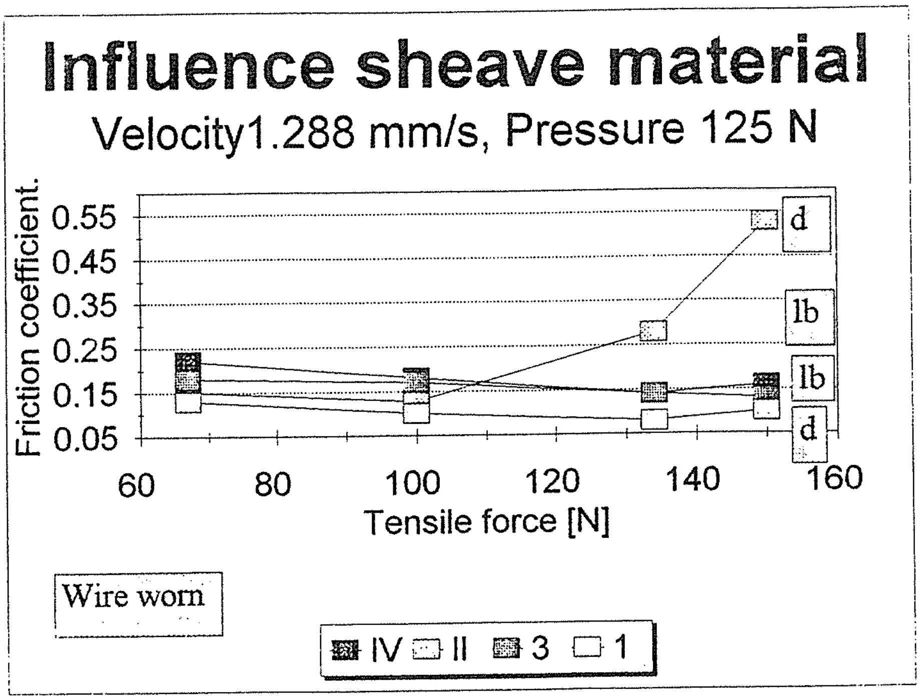

Fig. 5. The friction coëfficiënt as a function of the tensile force.

Figure nr.5. shows the influence of the sheave materials GGG60 (IV) and GG35 (3) if the point of contact is lubricated (lb), as well as the materials GGG60 (II) and GG35 (1) if the point of contact was degreased (d).

Further it appeared, that nodular cast iron produced larger friction coefficients than the cast iron GG35.

Worn wires have larger friction coefficients than new wires and because new wires are quickly run in, it is necessary to calculate also with friction coefficients of worn wires.in.

The wire material had only minor influence on the friction.

From the above mentioned friction behaviour can be concluded, that a standard friction coefficient for calculations can cause unwished large over capacity of the traction ratio. So if he has sufficient experiments available it may be good to give the designer of traction sheaves the possibility to use the friction coefficient which belongs to the applied sheave material.

4. THE VALUE OF THE GROOVE FACTOR.

In looking at the picture of a steel wire rope and at figure 1, it will cause the feeling that this model does not meet the reality. To verify the value of groove factors calculated in this way, there are performed a lot of measurements with the test bench of the figure 6.

A 32 mm wire rope 8x19 S with steel core and outerwires of 2 mm is tied into a horizontal tensile tester, while the rope is rigged into a carriage with two guide sheaves and a measuring sheave.

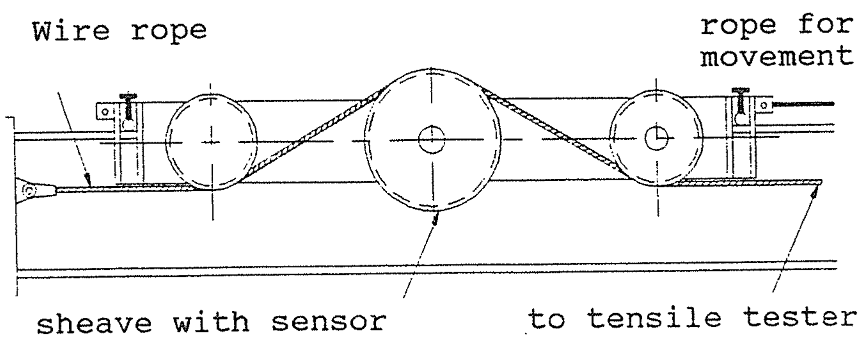

Fig.6. Groove factor measurements.

If the carriage is moved on the frame of the tensile tester, the sheave rolls on the rope.

The wires which come in contact with the groove of the sheave will bear contact forces in proportion to the tensile force and the position of these in the groove.These contact forces are measured with a piëzo-electric sensor of 6 mm in diameter and 8.5 mm in length, which is mounted in a holder and becomes into contact with the wire by way of a pin through an accurately drilled hole in the measuring sheave. (See figure 7.)

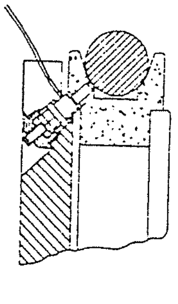

Fig.7. The piëzo-electric sensor.

During the movement the measuring signals of this dynamometer pass successively a charge amplifier, a low-pass filter, a signal converter and a computer to work up the data.

The measurements result in graphs, with vertical the pressure force at the wire and horizontal the number of samples from the convertor. The latter can be translated into the covered angle of the sheave. The measurements are performed at various points of contact at a round groove and at an under cut groove.

Figure 8 gives an example of a measurement with an under cut groove under 94 degrees.

Small eccentricities of the measuring sheave makes the shape of the graphs somewhat different if was moved there or back. Therefore an average of the corresponding samples of both graphs was made.

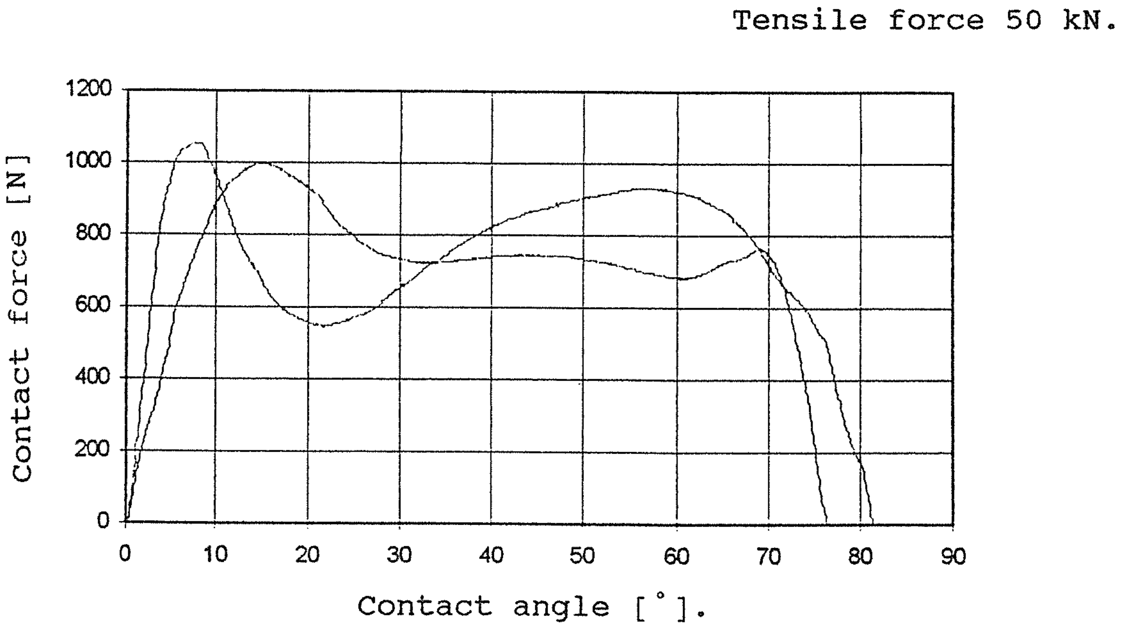

Fig.8. Measurement of contactforce with an under cut groove.

Also the average value of all the samples of such an average graph was calculated. In this manner a value of the contact force was got which represents the possible average friction force at the particular point of contact during one passage of the rope over the sheave.

Series of these data of measurements under comparable rope forces at all the wires in contact with an under cut groove, gave the capability of traction of this type of groove under the particular load. This procedure was performed for different rope forces in use with traction sheaves. This resulted in a factor which in the table 2 is indicated with the real groove factor.

The round groove has too many wire contacts to measure all of these. Therefore the above mentioned procedure was performed with 6 of these wires.

From these the ratios a_i between all the contact forces under particular angles φ_i and the force i=1 at the bottom of the groove were determined.

$$g = \frac{1}{\sum_{i=1}^{i=i_{gr.}} a_i} \cdot \sum_{i=1}^{i=gr.} \frac{a_i}{\cos\varphi_i} \qquad (11).$$

These ratios together with the cosines of the angles φ_i are put in the following discontinuous model of the formula (6).The values of the groove factor g, which were found with this model, can be found in the table 2 under the heading discontinuous distribution.

In table 2 the results of various calculations were compared with the groove factors as these were found with the traditional calculation.

The last column of this table contains values which are calculated with the traditional formulas (8) and (9), but with an adapted value of the angle δ. In stead of the angle δ shown in figure 1, the angle between the last visible points of contact at the groove is used for the calculation. The results of this calculation appeared to be very close to the real value. Of course the result depends on the radius of the groove.

5. COMPARISON OF TRACTION RATIOS FROM THE RESULTS.

If it is supposed that the tensile force in the wires with groove contact are between the values 100 and 134 N/mm^2 and that the traction ratio is 1.35, the averages of the coefficients from the 2nd and 3rd column of table 1 can be used for calculations of the relevant creep angles α.

TABLE 2

Comparison of the groove factors calculated in the traditional way and with the measured contact forces.				
	Traditional continuous model.	discontinuous model	real groove factor	adapted continuous model
Round groove	1.26	1.07	-	1.09
Under cut groove	2.08	1.66	1.61	1.62

Table 3 gives the results of these calculations for the under cut grooves and the researched material combinations.

The creep angles from table 3 must be compared with $\alpha = 91.8°$ which is calculated with the usual friction coefficient μ=0.09 and the traditional groove factor 2.08.

Only with the material GG35, being run in and not lubricated, the creep angle has a deficit of 9.3° with respect to α=91.8°.

TABLE 3

Creep angle α in degrees for traction ratio 1.35, under cut grooves, different materials and various models of groove factors					
Material combination	μ	continuous	dis-continuous	real	adapted continues
GG35/1360 GGG60/1770 new, dry, lubricated	0.115	71.9	90.1	92.9	92.3
GGG60/1770 worn, dry, lubricated	0.18	45.9	57.5	59.3	59
GG35/1360 worn,dry	0.105	78.7	98.7	101.7	101.1
GG35/1360 worn and lubricated	0.165	50.1	62.8	64.7	64.3

The largest reserve of the creep angle is 45.9°. This reserve, however, has entirely been caused by the influence of the two times larger friction coefficient. A more realistic value of the reserve is 34.3°. There the influence of the friction coefficient is damped by the 23 % lower groove factor.

Taking in account that with the traditional method of calculation also two safety factors must be used, the examples show that occasionally too low traction ratios are calculated.

6. LITERATURE:

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