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APPENDIX A Anchorheads

An anchor head is drilled with a number of tapered holes suitable for accepting jack temporary wedges. An anchor head of the 830 SSL strand jack unit for example contains 54 tapered holes (Figure 3). In Figure 1 a characteristic example of an anchor head is presented. The quantity of tapered holes of the anchor heads is scaled to the required capacity.

The main specifications of the anchor heads are presented in Table 1.

Table 1: Specifications Anchor heads. Source: [Mammoet]

<table>
<thead>
<tr>
<th>Specifications</th>
<th>SSL830</th>
<th>SSL550</th>
<th>SSL300</th>
<th>SSL100</th>
</tr>
</thead>
<tbody>
<tr>
<td>Main dimensions(mm)</td>
<td>∅520x190</td>
<td>∅450x190</td>
<td>∅354x100</td>
<td>∅200x150</td>
</tr>
<tr>
<td>Weight (kg)</td>
<td>271</td>
<td>207</td>
<td>60</td>
<td>40</td>
</tr>
<tr>
<td>Material</td>
<td>34CrNiMo6</td>
<td>34CrNiMo6</td>
<td>34CrNiMo6</td>
<td>34CrNiMo6</td>
</tr>
<tr>
<td>Wedges / holes (Qty)</td>
<td>54</td>
<td>36</td>
<td>18</td>
<td>7</td>
</tr>
<tr>
<td>Layer thickness surface treatment (mm)</td>
<td>0.1-1.6</td>
<td>0.1-1.6</td>
<td>0.1-1.6</td>
<td>0.1-1.6</td>
</tr>
<tr>
<td>Solid lubricant</td>
<td>Molycote D321 R</td>
<td>Molycote D321 R</td>
<td>Molycote D321 R</td>
<td>Molycote D321 R</td>
</tr>
</tbody>
</table>

Note that the surface treatment of the anchor heads is nitro carburizing.

Figure 1: Anchor head of 830 SSL strand jack unit

Figure 2: 830 SSL anchor head with 54 wedges

Figure 3: Tapered wedge seatings of an anchor head
In order to reduce friction, the wedge seats of the anchor head are pre-treated with Molykote D321 R, a dry lubricant (see Figure 4).

Figure 4: Molykote D321 R Pre-treated Anchor head
APPENDIX B  WEDGE PROPERTIES

The wedges used in the strand jack units contain three wedge parts. The wedge is equipped with a friction or grip profile on the inside of each part. The main dimensions of the wedge and wedge parts are presented in Figure 5 en Figure 6.

![Figure 5: Assembled wedge and wedge part](image)

The three wedge parts are assembled with an elastic rubber ring. This ring retains the three wedge segments/parts together during operation (Figure 7).

![Figure 7: Assembled wedge with rubber ring](image)

Table 2: Specifications TT 18/44 wedge (sources : Mammoet,KANIGEN )

<table>
<thead>
<tr>
<th>Main specifications</th>
<th>TT 18/44 Wedge</th>
</tr>
</thead>
<tbody>
<tr>
<td>Main dimensions(mm) :</td>
<td>0.25x0.44x0.18x0.8</td>
</tr>
<tr>
<td>Total weight (kg) :</td>
<td>0.450</td>
</tr>
<tr>
<td>Material :</td>
<td>DIN 1.6523 (SAE 8620)</td>
</tr>
<tr>
<td>Wedge parts (Qty) :</td>
<td>3</td>
</tr>
<tr>
<td>Weight wedge part (kg) :</td>
<td>0.150</td>
</tr>
<tr>
<td>Hardening (Carburizing) temperature (°C) :</td>
<td>900-1000</td>
</tr>
<tr>
<td>Tempering temperature (°C) :</td>
<td>190-200</td>
</tr>
<tr>
<td>Hardness of substrate after tempering (indication*) (HV) :</td>
<td>700-760</td>
</tr>
<tr>
<td>Surface treatment :</td>
<td>Electroless Nickel plated (KANIGEN method)</td>
</tr>
<tr>
<td>Proces temperature surface treatment (°C) :</td>
<td>200-280</td>
</tr>
<tr>
<td>Hardness of surface layer (indication*) (HV) :</td>
<td>500-550</td>
</tr>
<tr>
<td>Hardness substrate after surface treatment (indication*) (HV) :</td>
<td>560-620</td>
</tr>
<tr>
<td>Layer thickness surface treatment (µm) :</td>
<td>25-50</td>
</tr>
<tr>
<td>Solid lubricant :</td>
<td>Molykote D321 R</td>
</tr>
<tr>
<td>Optimal layer thickness lubricant (µm) :</td>
<td>5-20</td>
</tr>
<tr>
<td>Total cost (euro) :</td>
<td>20</td>
</tr>
<tr>
<td>Manufacturer :</td>
<td>TT Fijnmechanica</td>
</tr>
</tbody>
</table>

* Values for reference only, exact values have to be determined by material tests

For main specifications of the currently used TT 18/44 wedge is referred to Table 2(In Table 2 several specifications are noted with “indication”, these values need to be reviewed by hardness tests). In order to reduce friction, the wedges are pre-treated before operation with Molykote D321 R, a dry lubricant.
APPENDIX C Ø18 mm DYFORM strands

The strand bundle contains several Dyform strands with a maximum operation capacity of 167 kN/strand (specified by Mammoet).

The quantity of strands depends on the used unit type. The Dyform strand contains 7 twisted, high capacity, steel wires with a flattened side (see Figure 8). They are specially developed for heavy lifting and its length can be up to 1500 meters to meet the job requirements.

The main specifications of the Dyform strand are presented in Table 3.

Table 3: Main specifications Dyform strand. Source: [Bridon wire Ltd]

<table>
<thead>
<tr>
<th>Main specifications</th>
<th>BSS896 (DYFORM)</th>
<th>Nominal values</th>
<th>Tolerances</th>
</tr>
</thead>
<tbody>
<tr>
<td>Nominal diameter (mm) :</td>
<td>Ø18</td>
<td>+0.4, -0.2</td>
<td></td>
</tr>
<tr>
<td>Mass (kg/m) :</td>
<td>1.75</td>
<td>+0.4%, -2%</td>
<td></td>
</tr>
<tr>
<td>Tensile strength (Rm) (N/mm²)</td>
<td>1700</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Surface hardness of wires (HV)</td>
<td>430-480</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Steel area (mm²)</td>
<td>223</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Breaking load (Fm) (kN)</td>
<td>380</td>
<td></td>
<td></td>
</tr>
<tr>
<td>0.1% proof load (Fp 0.1) (kN)</td>
<td>323</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Load at 1% elongation (Ft 1.0) (kN)</td>
<td>334</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Wires (Qty)</td>
<td>7</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Manufacturer :</td>
<td>Carrington Wire Ltd / Bridon Wire Ltd</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
APPENDIX D Operation cycles Strandjack unit

In this appendix a review is presented on two main operation cycles of the strand jack unit SSL to familiarize the reader with the general concepts of the Strand jack-unit SSL:

- Jack up cycle (lifting the load)
- Jack down cycle (lowering the load)

D.1 Jack up cycle (lifting the load)

In the next paragraph the Jack up cycle of the hydraulic Strand jack units is described. This principle concerns all types.

The start position (see Figure 9):
The 18 mm dyform compact strands are installed through the unit and the load is connected. Both upper and lower anchor heads are "locked" (red in Figure 9).

Step 1 (see Figure 9):
The piston of the jack extends and raises the upper anchor head including the locked strands and connected load. During this movement the wedges in the lower anchor head are pulled up slightly by the movement of the strands; the lower anchor head is still closed. In case of failure of the upper anchor head, the wedges in the lower anchor head secure the load.
Step 2 (see Figure 10):
In top position the load is transferred from the upper anchor head to the lower anchor head by slightly retracting the piston, approx. 15 mm. During this load transfer both anchor heads remain locked.

Step 3 (see Figure 10):
After the transfer, the upper anchor head is opened hydraulically. The upper release cylinder lifts the release plate with the release tubes and shifts the wedges/grips up and out of their seatings (Note the difference between left and right detail picture). The upper anchor head is now “unlocked” (blue in Figure 10) allowing free passage of the strands.
Step 4 (see Figure 11):
With the upper anchor head “unlocked”, the piston of the jack retracts and returns to the starting position. The strands slide through the wedges in the upper anchor head.

Step 5 (see Figure 11):
At the end of step 4, the upper anchor head is “locked” again by the upper release cylinder retracting the release plate and release tubes.

The jack up cycle of step 1 to 5 is repeated till the required jack up distance is accomplished.
D.2 Jack down cycle (lowering the load)

In the next paragraph the Jack down cycle of the hydraulic Strand jack units is described. This principle concerns all types.

**START POSITION**

**STEP 1**

**The start position (see Figure 12):**
The 18 mm Dyform compact strands are installed through the unit and the load is connected. Both upper and lower anchor heads are "locked" (red in Figure 12).

**Step 1 (see Figure 12):**
The upper anchor head is opened hydraulically. The upper release cylinder lifts the release plate with the release tubes and shifts the wedges/grips up and out of their seatings (Note the difference between detail picture of the upper anchor head in **Start position** and **Step 1**). The upper anchor head is now "unlocked" (blue in Figure 12) allowing free passage of the strands.

Figure 12: Start position and Step 1 of the Jack down cycle
Step 2 (see Figure 13):
With the upper anchor head “unlocked”, the piston of the jack extends till approx. 15 mm before end of the outward stroke (compare the position of the main hydraulic cylinder in step 1 and step 2). The strands slide through the wedges in the upper anchor head during the stroke.

Step 3 (see Figure 13):
In the position “15 mm before end of the outward stroke” the upper anchor head is “locked” by the upper release cylinder retracting the release plate and release tubes (red in detail upper anchor head in see Figure 13 step 3).
Figure 14: Step 4 and Step 5 of the Jack down cycle

**Step 4 (see Figure 14):**
The load is transferred from the lower anchor head to the upper anchor head by slightly extending the piston of the main cylinder further (approx. +15 mm) to the end of the stroke. During this load transfer both anchor heads remain “locked” (both red in Figure 14 step 4). During this movement the wedges in the lower anchor head are pulled up slightly by the movement of the strands; the lower anchor head is still closed. In case of failure of the upper anchor head, the wedges in the lower anchor head secure the load.

**Step 5 (see Figure 14):**
After the transfer, the lower anchor head is opened hydraulically (“unlocked”; blue in detail of lower anchor head step 5). The lower release cylinder lifts the release plate with the release tubes and shifts the wedges/grips up and out of their seatings (Note the difference between left and right detail picture step 4 and step 5). The lower anchor head is now “unlocked” (blue in Figure 14) allowing free passage of the strands.
**Figure 15: Step 6 and Step 7 of the Jack down cycle**

**Step 6 (see Figure 15):**
The piston of the jack retracts and lowers the closed upper anchor head including the strands and load (movement is stroke) until approx. 15 mm before end of the inward stroke. The lower anchor head is still “unlocked” (blue in Figure 15) during this movement and allows free passage of the strands.

**Step 7 (see Figure 15):**
In the position “15 mm before end of inward stroke”, the lower anchor head is “locked” hydraulically by the lower release cylinder retracting the release plate and release tubes (red in detail lower anchor head in Figure 15 step 7).
Step 8 (see Figure 16):
After “locking” the lower anchor head, the load is transferred from the upper anchor head to the lower anchor head by slightly retracting the piston of the main cylinder further to the end of the inward stroke, approx. -15 mm. During this load transfer both anchor heads remain locked.

The jack down cycle of step 1 to 8 is repeated until the required jack down distance is accomplished.
APPENDIX E  Calculations wedge model

In this appendix a mathematical model of the strand locking system is presented. In the following considerations a single wedge part is examined and the spring force of the pre-stress spring is neglected.

\[ F_L = \frac{F_A \cdot L}{A_{\text{cable}}} \]

\[ \mu_{\text{kout}} = \text{Kinetic coefficient of friction between outer wedge surface and anchor head surface} \]
\[ \mu_{\text{kin}} = \text{Kinetic coefficient of friction on inner wedge surface and cable surface} \]
\[ N_{\text{out}} = \text{Normal force on outer wedge surface} \]
\[ N_{\text{in}} = \text{Normal force on inner wedge surface} \]
\[ \sigma_{\text{cw}} = \text{Compressive stress between wedge and cable} \]
\[ \sigma_{\text{Nout}} = \text{Compressive stress perpendicular wedge surface and anchor head} \]
\[ \sigma_c = \text{Tensile stress in cable as result of } F_L \]
\[ F_L = \text{Load force} \]
\[ \alpha = \text{Wedge angle} \]
\[ \tau_{\text{out}} = \text{Friction shear stress between outer wedge surface and anchor head} \]
\[ \tau_{\text{cw}} = \text{Friction shear stress between wedge and cable} \]
\[ S = \text{Position of slip front (for detailed information is referred to Appendix G2.2)} \]
\[ R_i = \text{Inner radius of wedge and therefore outer radius of strand/cable} \]
\[ A_{\text{cable}} = \text{Cross section area of cable} \]

From Figure 17, the following equations can be obtained:
\[ \tau_{\text{out}} = \mu_{\text{kout}} \cdot \sigma_{\text{nout}} \]  \hspace{1cm} (a) \quad \text{and} \quad \tau_{\text{cw}} = \mu_{\text{kin}} \cdot \sigma_{\text{cw}} \]  \hspace{1cm} (b)

**E.1 Areas and projected wedge areas** \( A_{\text{in}} \), \( A_{\text{out}} \), \( A_{\text{ineff}} \) en \( A_{\text{outeff}} \)

The estimation of the theoretical outer and inner surface of a single wedge part (assuming cylindrical), \( A_{\text{in}} \) and \( A_{\text{out}} \), is formulated as:

\[ A_{\text{in}} = \frac{2}{3} \pi \cdot R_{\text{i}} \cdot S \]  \hspace{1cm} (c)

\[ A_{\text{out}} = \frac{2}{3} \pi \cdot R_{\text{out}} \cdot \frac{S}{\cos(\alpha)} \]  \hspace{1cm} (d)

This according to Figure 18.

**TOP VIEW WEDGE PART**  \hspace{1cm} **SIDE VIEW WEDGE PART**

\[ \begin{align*}
A_{\text{ineff}} &= R_{\text{i}} \cdot S \cdot \sqrt{3} \quad (e) \quad \text{and} \quad A_{\text{outeff}} &= R_{\text{out}} \cdot \frac{S}{\cos(\alpha)} \cdot \sqrt{3} \quad (f)
\end{align*} \]

These equations can be derived from Figure 19.
Figure 19: Top view and side view wedge part in relation to $A_{\text{eff}}$ and $A_{\text{out}}$

The total normal force $N_{\text{out}}$, acting perpendicular on the effective surface of the wedge part becomes (see also Figure 17):

$$N_{\text{out}} = \sigma_{\text{Nout}} \cdot A_{\text{out}} = \sigma_{\text{Nout}} \cdot R_{\text{out}} \cdot \frac{S}{\cos(\alpha)} \cdot \sqrt{3} \quad (g)$$

The total normal force $N_{\text{in}}$, acting perpendicular on the inner surface of the wedge part becomes (see also Figure 17):

$$N_{\text{in}} = \sigma_{\text{cw}} \cdot A_{\text{ineff}} = \sigma_{\text{cw}} \cdot R_{\text{i}} \cdot S \cdot \sqrt{3} \quad (h)$$

### E.2 Horizontal wedge equilibrium

The total equilibrium of the horizontal forces on the wedge part results in the following equation (according to Figure 17):

$$N_{\text{out}} \cdot \cos(\alpha) = N_{\text{in}} - \tau_{\text{out}} \cdot \sin(\alpha) \cdot A_{\text{out}} \quad (i)$$

After substitution of equations a, g, h and f in i

$$\sigma_{\text{Nout}} \cdot R_{\text{out}} \cdot \frac{S}{\cos(\alpha)} \cdot \sqrt{3} \cdot \cos(\alpha) = \sigma_{\text{cw}} \cdot R_{\text{i}} \cdot S \cdot \sqrt{3} - \mu_{\text{kout}} \cdot \sigma_{\text{Nout}} \cdot \sin(\alpha) \cdot R_{\text{out}} \cdot \frac{S}{\cos(\alpha)} \cdot \sqrt{3}$$

After solving and rearranging, we get

$$\sigma_{\text{Nout}} \cdot R_{\text{out}} \cdot (1 - \mu_{\text{kout}} \cdot \tan(\alpha)) = \sigma_{\text{cw}} \cdot R_{\text{i}} \quad \text{or} \quad \sigma_{\text{Nout}} \cdot \frac{R_{\text{out}}}{R_{\text{i}}} \cdot (1 - \mu_{\text{kout}} \cdot \tan(\alpha)) = \sigma_{\text{cw}} \quad (j)$$
E.3 Vertical wedge equilibrium

The total equilibrium of the vertical forces on the wedge part results in the following equation (according to Figure 17):

\[ A_{in} \cdot \tau_{cw} - \tau_{out} \cdot \cos(\alpha) \cdot A_{out} - \sigma_{Nout} \cdot \sin(\alpha) \cdot A_{out} = 0 \]  

\[ + \]  

\[ (k) \]

After substitution of equations b, c, a, d in k, we get

\[ \frac{2}{3} \pi \cdot R_{i} \cdot S \cdot \mu_{kin} \cdot \sigma_{cw} - \mu_{kout} \cdot \sigma_{Nout} \cdot \cos(\alpha) \cdot \frac{2}{3} \pi \cdot R_{out} \cdot \frac{S}{\cos(\alpha)} - \]

\[ \sigma_{Nout} \cdot \sin(\alpha) \cdot \frac{2}{3} \pi \cdot R_{out} \cdot \frac{S}{\cos(\alpha)} = 0 \]

After solving and rearranging, this results in

\[ \frac{R_{i} \cdot \mu_{kin} \cdot \sigma_{cw} = \sigma_{Nout} \cdot R_{out} \cdot (\mu_{kout} + \tan(\alpha))}{or} \]

\[ \mu_{kin} \cdot \sigma_{cw} = \sigma_{Nout} \cdot \frac{R_{out}}{R_{i}} \cdot (\mu_{kout} + \tan(\alpha)) \]

}\[ (l) \]

E.4 Wedge operating range

Substituting equation j in l results in

\[ \mu_{kin} \cdot \sigma_{Nout} \cdot \frac{R_{out}}{R_{i}} \cdot (1 - \mu_{kout} \cdot \tan(\alpha)) = \sigma_{Nout} \cdot \frac{R_{out}}{R_{i}} \cdot (\mu_{kout} + \tan(\alpha)) \]

After rearranging

\[ \mu_{kout} = \frac{\mu_{kin} - \tan(\alpha)}{1 + \mu_{kin} \cdot \tan(\alpha)} \quad \text{and so} \quad \mu_{kin} = \frac{\mu_{kout} + \tan(\alpha)}{1 - \mu_{kout} \cdot \tan(\alpha)} \]

\[ (m) \]

These two equations represent the “borderline” of the wedge operating range.
E.5 Vertical cable equilibrium

When the equilibrium of the vertical forces on the cable/strand in Figure 17 is considered, the following equation can be acquired:

\[ F_L = 2 \cdot \pi \cdot R_i \cdot \tau_{cw} \cdot S \]  

(n)

With

\[ F_L = \sigma_c \cdot A_{cable} = \sigma_c \cdot \pi \cdot R_i^2 \]  

(o) and \[ \tau_{cw} = \mu_{kin} \cdot \sigma_{cw} \]  

(b)

After substitution of (o) and (b) in equation (n), the result is:

\[ \sigma_c \cdot \pi \cdot R_i^2 = 2 \cdot \pi \cdot R_i \cdot \mu_{kin} \cdot \sigma_{cw} \cdot S \]  

(p)

After rearranging:

\[ \sigma_c = \frac{2 \cdot S}{R_i} \cdot \mu_{kin} \cdot \sigma_{cw} \]  

(q)
APPENDIX F  Malfunction of the wedge

For the benefit of the analysis, three possible cases are described to clarify the malfunction of the wedge, according to the mechanical wedge model:

1. Wedge operation with : $\mu_{\text{kin}}$ Constant, increasing $\mu_{\text{kout}}$
2. Wedge operation with : $\mu_{\text{kout}}$ Constant, decreasing $\mu_{\text{kin}}$
3. Wedge operation with : Increasing $\mu_{\text{kout}}$, decreasing $\mu_{\text{kin}}$

In the following paragraphs these load cases are described in detail.

F.1 Wedge operation with : $\mu_{\text{kin}}$ Constant, increasing $\mu_{\text{kout}}$

When the wedge operation is started, a proper and normal operation of the wedge is assumed. This implies that the friction conditions are below the “good/bad” borderline in Figure 20. For example, fictive estimated start values of $\mu_{\text{kout}}$ and $\mu_{\text{kin}}$ in operation are 0.2 and 0.5 (see start operation point A in Figure 20).

![Figure 20: Wedge operation with increasing $\mu_{\text{kout}}$](image)

With increased $\mu_{\text{kout}}$, the operation point B is positioned in the red “Malfunction Area”. The friction force between wedge outer surface and tapered hole surface is too high and consequently gripping action is less; the strand slips through the wedge.
**F.2 Wedge operation with: \( \mu_{kout} \) Constant, decreasing \( \mu_{kin} \)**

When the wedge operation is started, a proper and normal operation of the wedge is assumed. For example, fictive estimated start values of \( \mu_{kout} \) and \( \mu_{kin} \) in operation are 0,2 and 0,5 (see start operation point A in Figure 21).

![Friction coefficients diagram](image)

*Figure 21: Wedge operation with decreasing \( \mu_{kin} \)*

When \( \mu_{kin} \) decreases during operation below the value \( \leq 0,35 \) and the value \( \mu_{kout} \) is constant, the operation point of the wedge is moved from A to point C (see Figure 21). Operation point C is positioned in the red “Malfunction Area”. The friction force between wedge inner surface and cable surface is too low, consequently the gripping action is less; the strand slips through the wedge.

**F.3 Wedge operation with: Increasing \( \mu_{kout} \), decreasing \( \mu_{kin} \)**

In most practical cases the values of the friction coefficients will change simultaneously, influenced by environmental circumstances (temperature, relative humidity etc.).
A combination of increasing $\mu_{\text{kout}}$ and decreasing $\mu_{\text{kin}}$ will lead to malfunction of the wedge conform Figure 22, the operation point of the wedge is moved from A to point D. Malfunction of the wedge appears.

Figure 22: Wedge operation with decreasing $\mu_{\text{kin}}$ and increasing $\mu_{\text{kout}}$
APPENDIX G  Micro slip during jack up

In every load cycle (jack up and jack down) micro slip will occur in the wedges of the upper and lower anchor head. This is a result of the difference in axial strain (as a result of the load force \( F_L \)) between the cable and wedge in consequence of the difference in stiffness. The axial strain in the cable is larger than in the wedge. Hence the strain in the loaded cable can not be “followed” by the wedge what results in a relative displacement between the cable outer surface and wedge inner surface. Micro slip in combination with a load force results in wear of the inner friction surface of the wedge and eventually in a reduction of the inner friction coefficient \( \mu_{\text{kin}} \). Finally this will cause malfunction of the wedge.

When the piston of the jack extends, the closed upper anchor head is raised including the strands. The load force \( F_L \) and related strain in the strand increase during this movement until the anchor head bears the total present load force \( F_L \). The situation of the full loaded wedge in the anchor head is described schematic in Figure 23.

\[ F_L = \text{Load force} \]
\[ V = \text{Spring force (pre-stress)} \]

![Figure 23: Loaded wedge during raising of upper anchor head](image-url)
\( N_{in} \) = Normal force on inner wedge surface  
\( \mu_{\text{kin}} \) = Kinetic coefficient of friction on inner wedge surface  
\( Q \) = Length arc of wedge part  
\( \alpha \) = Wedge angle  
\( \sigma_{cw} \) = Compressive stress between wedge and cable  
\( \sigma_{c} \) = Tensile stress in cable as result of \( F_L \)  
\( \varepsilon_{c} \) = Strain in cable  
\( S \) = Position of slip front  
\( L \) = Length of friction profile  
\( A_{\text{cable}} \) = Cross section area of cable  
\( E_{c} \) = Modulus elasticity of cable  
\( \tau_{cw} \) = Friction shear stress  

At the end of the trajectory of the increasing load force \( F_L \)(the upper anchor head bears the total load force \( F_L \)), the slip front has propelled to distance \( S \). In the cross section of the cable on position \( S \), the tensile stress \( \sigma_c = 0 \). Along the inner friction surface of the wedge part, width \( Q \) and length \( L \), exists a compressive stress \( \sigma_{cw} \) between wedge and cable as a result of the normal force \( N_{in} \). The (pre-stress) spring force \( V \) is neglected and the stiffness of the wedge is considered infinite in the following considerations.

This compressive stress \( \sigma_{cw} \) present on the 3 wedge parts is formulated as:

\[
\sigma_{cw} = \frac{N_{in}}{3 \cdot Q \cdot L}
\]

The compressive stress \( \sigma_{cw} \) is able to transmit an average friction shear stress \( \tau \) on the contact area:

\[
\tau_{cw} = \mu_{\text{kin}} \cdot \sigma_{cw}
\]

The load force \( F_L \) is compensated by three inner friction surfaces, with an area of \( Q \cdot s \) and thus is derived for \( F_L \):

\[
F_L = 3 \cdot Q \cdot s \cdot \tau_{cw} = 3 \cdot Q \cdot s \cdot \mu_{\text{kin}} \cdot \sigma_{cw} = \frac{3 \cdot Q \cdot s \cdot \mu_{\text{kin}} \cdot N_{in}}{3 \cdot Q \cdot L} = \frac{s \cdot \mu_{\text{kin}} \cdot N_{in}}{L}
\]

A increasing force \( F_L \) results in a increasing value of \( s \), as soon as \( s > L \) total slip and \( F_L = F_{\text{max}} = \mu_{\text{kin}} \cdot N_{in} \) occurs.

- If \( s > L \) : macroslip occurs, the strand slips through the wedge  
- If \( 0 < F_L < F_{\text{max}} \) : microslip acts along length \( s \).

For the tensile stress \( \sigma_c \) and strain \( \varepsilon_c \) in the cable we can write

\[
\sigma_c = \frac{F_L}{A_{\text{cable}}} \quad \varepsilon_c = \frac{\sigma_c}{E_{c}} = \frac{F_L}{E_{c} \cdot A_{\text{cable}}}
\]

The length \( s \) can be calculated from the relation

\[
s = \frac{F_L}{\mu_{\text{kin}} \cdot N_{in}} \cdot L \quad (\text{Eq. G.1})
\]
The tensile stress $\sigma_c$ diminishes linear with values on positions $x = 0$ to $x = s$ (see Figure 23) according to

$$\sigma_{c(x=0)} = 0 \quad \text{and} \quad \sigma_{c(x=s)} = \frac{F_L}{A_{\text{cable}}}$$

And analogically with the equations for the tensile stress $\sigma_c$, equations for strain $\varepsilon_c$ are

$$\varepsilon_{c(x=0)} = \frac{\sigma_c}{E_c} = \frac{0}{E_c \cdot A_{\text{cable}}} = 0 \quad \text{and} \quad \varepsilon_{c(x=s)} = \frac{\sigma_c}{E_c} = \frac{F_L}{E_c \cdot A_{\text{cable}}}$$

A graph of the $x$-position against strain $\varepsilon_c$ is provided in Figure 24.

![Figure 24 Graph of position x, strain and tensile stress in cable wedge](image)

With usage of Figure 24 the total cable elongation $dl$ in the wedge is derived. The elongation $dl$ of the cable in the wedge corresponds with area $P$

$$dl = P = \frac{1}{2} \cdot \varepsilon_{c(x=s)} \cdot s$$

With

$$\varepsilon_{c(x=s)} = \frac{\sigma_c}{E_c} = \frac{F_L}{E_c \cdot A_{\text{cable}}} \quad \text{and} \quad s = \frac{F_L \cdot L}{\mu_{\text{kin}} \cdot N_{\text{in}}}$$

After substitution, $dl$ becomes
\[ dl = \frac{1}{2} \cdot \varepsilon_c(x,s) \cdot s = \frac{F_L \cdot L}{2 \cdot E_c \cdot A_{cable} \cdot \mu_{\text{kin}} \cdot N_{in}} \]

Function \( \varepsilon_c(x) \) can be described as (see Figure 24):

\[ \varepsilon_c(x) = \frac{F_L \cdot x}{E_c \cdot A_{cable}} \quad \text{for} \quad 0 \leq x \leq s \]

With the definition \( \frac{du}{dx} = \varepsilon_c(x) \) follows:

\[ \frac{du}{dx} = \frac{F_L \cdot x}{E_c \cdot A_{cable}} \]

To obtain the local relative displacement function \( u(x) \)

\[ u(x) = \int \left[ \frac{du}{dx} \right] \cdot dx = \int \left[ \frac{F_L \cdot x}{E_c \cdot A_{cable}} \right] \cdot dx \quad \text{for} \quad 0 \leq x \leq s \]

This leads to the relative displacement function \( u(x) \)

\[ u(x) = \frac{F_L \cdot x^2}{2 \cdot E_c \cdot A_{cable}} + C \quad \text{for} \quad 0 \leq x \leq s \]

With \( u(0) = 0 \) consequently \( C = 0 \), relative displacement function \( u(x) \) becomes

\[ u(x) = \frac{F_L \cdot x^2}{2 \cdot E_c \cdot A_{cable}} \quad \text{for} \quad 0 \leq x \leq s \quad (\text{Eq. G.2}) \]
Equation G-2 is presented graphically in Figure 25.

The magnitude of relative displacement $u(x)$ (and thus wear) of the outer cable surface against the friction profile depends mainly on the location of the slip front $S$, assuming $F_L$, $A_{cable}$, $E_c$ constant. Maximum slip is located at the loaded “outlet” of the cable at position $S$. 
Abstract

The company Mammoet BV meets difficulties with cable wedges/clamps of their strand jack units. Slipping of strand through the wedges and high maintenance costs of the strand jack units hence, are the main aspects. In this report general operation fundamentals of the current strand jack unit are explained and detailed information of essential components is provided. It also provides a straightforward theoretical/mathematical approach on the probable causes of malfunction and unreliability of the strand jack wedge. To improve the service life, reliability and handling properties of the wedge and strand, an alternative wedge design is presented.
Summary

Mammoet is a leading company specialized in solving extreme heavy lifting and transport challenges. With offices in 30 countries and over more than 1500 employees around the world Mammoet provides its customers with knowledge, skills and equipment at onshore or offshore location. The solutions that Mammoet provide are used in economic sectors like the Petro Chemical industry, Civil works, Power plants and Offshore. The best project example of the latter is of course the salvation of the Russian nuclear submarine the Kursk.

In various projects Mammoet applies a compact Strand jack unit placed on a gantry or mast for precise lifting, lowering or pulling of heavy loads. This unit is, due to its compact design, very suitable in areas where conventional equipment as cranes cannot be placed. This Strand jack unit is generally based on a hydraulic cylinder with a wedge principle to lock the strands in each stroke. Two wedges, situated in the lower and upper anchor head in the cylinder, lock each strand in the unit when the load is lifted or lowered.

Mammoet meets difficulties especially with these wedge clamps of the strand jack units. Slipping of strand through the wedges and high maintenance costs of the strand jack units hence, are the main aspects. In the specialized and prestigious market of heavy lifting and transport safety and operating reliability are important topics, decisive in maintaining a good reputation. It’s obvious that research on the used wedge became more than desirable.

According to the calculations performed in Appendix E the proper operation area of the wedge can be characterized as:

\[
\mu_{\text{kout}} < \frac{\mu_{\text{kin}} - \tan(\alpha)}{1 + \mu_{\text{kin}} \cdot \tan(\alpha)} \quad \text{and so:} \quad \mu_{\text{kin}} > \frac{\mu_{\text{kout}} + \tan(\alpha)}{1 - \mu_{\text{kout}} \cdot \tan(\alpha)}
\]

With:
- \( \mu_{\text{kout}} \): Kinetic coefficient of friction between outer wedge surface and anchor head surface
- \( \mu_{\text{kin}} \): Kinetic coefficient of friction on inner wedge surface and cable surface
- \( \alpha \): Wedge angle
- \( F_L \): Load force

The wedge gripping quality depends on inner-and outer friction coefficients \( \mu_{\text{in}}, \mu_{\text{out}} \) and the wedge angle \( \alpha \). The characteristic graph of Figure 13 provides the proper operation area and malfunction area of the wedge, as a function of \( \mu_{\text{in}}, \mu_{\text{out}} \) (with \( \alpha=8.19^\circ \)).

Several environmental circumstances and principles can have an influence on the value of \( \mu_{\text{kout}} \):

- **Adhesion**: between wedge outer surface and tapered hole surface
- **Plowing /third body effects**: between wedge outer surface and tapered hole surface

For \( \mu_{\text{kin}} \):

- **Accumulation of dirt**: between strand outer surface and inner surface of the wedge
- **Wear on inner friction profile of wedge during load cycles**
During operation relative displacement occurs between the inner friction profile and strand outer surface. The relative displacement $u(x)$ can be characterized as:

$$u(x) = \frac{F_L \cdot x^2}{2 \cdot E_c \cdot A_{cable}}$$

- $F_L$ = Load force
- $A_{cable}$ = Cross section area of cable
- $E_c$ = E-modulus of cable material

This relative displacement which initiates wear is called micro slip.

The micro slip reducing design, presented in this report, can increase service life of the wedge.

Applying 190 high capacity solid strands with diameter of 9 mm can realize the following objectives:

- No internal relative displacements between wires during operation: less elastic power is lost by friction/wear. Therefore no internal wear is initiated. In general a strand with a diameter of > 9 mm contains multiple wires, and the internal wear and elastic energy loss will increase drastically by an increasing number of strand wires.

- The 9 mm strand also improves handling due to a weight reduction of 1.25 kg/m compared to the original 18 mm strand. In a proper 900-ton configuration of the strand jack unit SSL, 190 wires of 9 mm are used.

In this 9 mm configuration, 380 small 9 mm “low cost” wedges are used.
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1 Introduction

Mammoet is a leading company specialized in solving extreme heavy lifting and transport challenges. With offices in 30 countries and over more than 1500 employees around the world, Mammoet provides its customers with knowledge, skills and equipment at onshore or offshore location. The solutions that Mammoet provide are used in economic sectors like the Petro Chemical industry, Civil works, Power plants and Offshore. The best project example of the latter is of course the salvation of the Russian nuclear submarine the Kursk.

In various projects Mammoet applies a compact Strand jack unit (see Figure 1 and Figure 2) placed on a gantry or mast for precise lifting, lowering or pulling of heavy loads. This unit is, due to its compact design, very suitable in areas where conventional equipment as cranes cannot be placed.

![Figure 1: Lifting 6.700 mTonnes deck with 12 strand jack units](image1)

![Figure 2: Two strand jack units on top of a mast](image2)

This Strand jack unit is generally based on a hydraulic cylinder with a wedge principle to lock the strands in each stroke. Two wedges, situated in the lower and upper anchor head in the cylinder, lock each strand in the unit when the load is lifted or lowered.

Mammoet meets difficulties especially with these wedge clamps of the strand jack units. Slipping of strand through the wedges and high maintenance costs of the strand jack units hence, are the main aspects. In the specialized and prestigious market of heavy lifting and transport safety and operating reliability are important topics, decisive in maintaining a good reputation. It’s obvious that research on the used wedge became more than desirable.

In this research report chapter two provides an overall view on Mammoet, its clients, mission statement and scope of work. In chapter three a profound and extensive examination is performed on the existing strand jack system and its components. General operation fundamentals of the current strand jack unit are explained and detailed information of essential components is provided. Chapter four provides a theoretical approach on the probable causes of mal-function and unreliability of the strand jack wedge. Mathematical equations, which
characterize the operation of the wedge, are derived and a graphical presentation of the “wedge malfunction area” is presented. In chapter five wear by micro slip is discussed and a micro slip/wear reducing wedge design is presented in chapter six.

The general conclusion and future perspectives/recommendations are carried out in chapter seven.
2 Mammoet

Mammoet is a specialist in heavy lifting and transport; it has subsidiaries and agencies throughout the world. Besides contracting on a turnkey basis, Mammoet offers crane rental and related services.

2.1 Merger

The new heavy lift combination was formed on 12 July 2000, when the shares of Mammoet Transport BV were taken over by Van Seumeren Holland BV from Royal Ned Lloyd. Van Seumeren Group has acquired Mammoet's entire business, including its activities in the United States, Asia, the Middle East and Europe. The merger of the two companies provided the possibility to offer a total package at the top of the market by scaling up to customers. Synergy results in:

- Mobilisation/demobilisation savings
- Capacity utilisation of equipment
- International commercial effectiveness
- A more efficient organisation
- Purchasing power

Both Dutch, heavy lifting and special transportation companies Mammoet and Van Seumeren Holland BV were nearly equally sized when they joined forces. Mammoet was slightly stronger in horizontal transportation. Van Seumeren was slightly stronger in vertical transportation. The combined expertise, the experience and efforts of the 1500 employees of Mammoet Holding BV form a crucial factor for the success and further growth opportunities of the company in the international project market. The new company is positioned worldwide under the name and logo “Mammoet”, with the mention “Van Seumeren Group”.

2.2 Clients and scope of work

Mammoet provides clients with tailor-made solutions and services in all scopes of work for engineered heavy lifting and multimodal transport worldwide. In Table 1 Mammoet’s clients and scope of work are indicated.

<table>
<thead>
<tr>
<th>Mammoet’s clients:</th>
<th>Forwarders</th>
<th>EPC contractors</th>
<th>Construction companies</th>
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<td>Power generating industry</td>
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<td>Chemical industry</td>
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<td>Petrochemical Industry</td>
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<tr>
<th>Mammoet’s scope of work:</th>
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<th>Jacking</th>
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<td>Lifting</td>
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<td>Transporting</td>
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<tr>
<td>Site moves</td>
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<td>Skidding</td>
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<tr>
<td>Placing</td>
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<td>Ballasting</td>
<td>From factory to foundation</td>
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<tr>
<td>Shifting</td>
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<td>Weighing</td>
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</tbody>
</table>

Table 1: Mammoet’s clients

Table 2: Mammoet’s scope of work
Figure 3 shows a gantry-lifting system, lifting an offshore module. Figure 4 is an example of a project in the chemical industry.

*Figure 3: Lifting a offshore module 600 mTonnes with 4 strand jack units on a gantry

*Figure 4: Lifting a 830 mTonnes vessel in the power chemical industry*
3 The Mammoet strand jack system

This chapter gives a broad view on the Mammoet strand jack system with the Strand Jack Units SSL and description of the working mechanism with occurring defects in the strand-locking principle. Also a detailed description of the used wedges is given.

Mammoet has designed the Mammoet strand jack system to be used for installation of heavy equipment, such as offshore-structures, equipment for petrochemical and nuclear power plants and bridges. The system has a modular set-up, designed for safe a precise lifting, lowering or pulling of heavy loads.

3.1 Hydraulic Strand jack-unit SSL

The (multi) Strand Jack Unit SSL is designed for usage in the MSG (Mammoet Sliding Gantry) and in the transport of heavy loads, horizontally or vertically. Mammoet has several Strand jack units available with different capacities and dimensions. The capacity is (of course) coupled with the quantity of strands and the diameter of the main cylinder. The main dimensions and technical data of the available Mammoet equipment are presented in Figure 5.

![Figure 5: Technical data and main dimensions of the strand jack equipment](image-url)
3.2 Components Hydraulic strand jack-unit SSL

In Figure 6 shows a general cross-section of a strand jack unit. The strand jack unit consists of the following main components:

- **Base plate:** this plate is fixed on the foundation
- **Lower anchor head (see Appendix A)**
- **Wedges (see Appendix B)**
- **Lower wedge-release-cylinder:** with the lower wedge-release-cylinder the lower anchor head opens and closes.
- **Hydraulic cylinder:** the hydraulic cylinder extends and retracts the piston. This way the strand can be moved over a stroke of 400 or 480 mm.
- **Upper wedge-release-cylinder:** with the upper wedge-release-cylinder the upper anchor head opens and closes.
- **Upper anchor head (see Appendix A)**
- **Strand guide tube (or pipe):** the guide tubes to ensure that strands passing not deflect and apply side loadings onto the top anchor assembly.
- **Ø18 mm compact strands (DYFORM) (see Appendix C)**

*Figure 6: Cross-section Strand jack unit*
3.3 Strand locking principle

The Hydraulic strand jack unit SSL contains a characteristic strand (or cable) locking principle. This locking principle is a decisive factor in the quality of performance of the strand jack unit SSL. In this paragraph a detailed description is provided. The locking principle comprises two main positions; “locked” and “unlocked” (examine Figure 7). For a detailed description of the operation cycles (jack-up and jack down) of the Hydraulic Strand jack-unit SSL is referred to Appendix D.

3.3.1 “Locked” principle

In the "locked" position, the release plate (or kick out plate) as well as the connected release tubes are fully retracted; allowing the wedges in their tapered seatings, held down by the pre-stress springs (see Figure 7). A special washer of either spigot or socket type is used between the spring and the wedge, to ensure that the spring pressure is applied equally to all three parts of the wedge (Figure 8).

The initial spring force \( \pm 400 \text{ N/spring} \) provokes an essential “pre-bite” of the wedge on the strand (Figure 8). The bite depth \( b \) of the “pre-bite” is negligible, however not to be overlooked for a proper operation of the gripping action of the wedge.

When the load force on the strand (and thus on the wedge) increases, the wedge drops over a distance \( D \) downward in the tapered seating and the bite depth \( b \) (or distance inwards) enlarges pro rata of the load force (examine Figure 9). The strand is now “locked” by the “bite” of the grip profile of the wedge.
The bite depth $b$ and distance $D$ have values of roughly 0.15-0.22 mm and 1.5-2 mm respectively (depending on circumstances in operation and differences in hardness of grip profile and strand surface). These values relate to the maximum Mammoet operation load of 167 kN /strand.

The radial movement (distance inward $b$) of the three wedge parts during an increase of the load force (conform Figure 8 and Figure 9) is presented schematically in Figure 10.
3.3.2 “Unlocked” principle

Before the “Unlocking” procedure is actuated, the load force on the wedges is reduced to zero (to exclude any damage on release plate and tubes).

Subsequently, by means of the additional hydraulic release cylinders (Figure 7) the release plate (or kick out plate with the connected release tubes) is lifted into the anchor head and shifts the wedges out of their tapered seatings (Figure 7 and Figure 11). The strand is now “Unlocked”; note the difference between Figure 9 and Figure 11.

By unlocking, free passage of the strands through the wedges is allowed; a negligible force is applied on the friction profile and no “bite” is generated (see detail Figure 11).

Figure 11: “Unlocked” position; the wedge shifted upwards
4 Theoretical problem analysis

In this chapter a mechanical wedge model is presented. With this model the theoretical situations
of mal-function of the wedge “strand locking” system are explained.

In Figure 12 the mechanical wedge model is presented with relevant parameters.

\[ \mu_{kout} = \text{Kinetic coefficient of friction between outer wedge surface and anchor head surface} \]
\[ \mu_{kin} = \text{Kinetic coefficient of friction on inner wedge surface and cable surface} \]
\[ \alpha = \text{Wedge angle} \]
\[ F_L = \text{Load force} \]

According to the calculations performed in Appendix E the “borderline” of operation of the wedge can be characterized as:

\[ \mu_{kout} = \frac{\mu_{kin} - \tan(\alpha)}{1 + \mu_{kin} \cdot \tan(\alpha)} \quad \text{and so} \quad \mu_{kin} = \frac{\mu_{kout} + \tan(\alpha)}{1 - \mu_{kout} \cdot \tan(\alpha)} \quad \text{(4a)} \]

In the different load cases of the wedge, the following two situations can occur:

**Situation 1** The strand jack wedge operates properly.

And thus

\[ \mu_{kout} < \frac{\mu_{kin} - \tan(\alpha)}{1 + \mu_{kin} \cdot \tan(\alpha)} \quad \text{and so} \quad \mu_{kin} > \frac{\mu_{kout} + \tan(\alpha)}{1 - \mu_{kout} \cdot \tan(\alpha)} \quad \text{(4b)} \]

**Situation 2** The strand jack wedge doesn’t operate properly.

And thus
The equations 4a, 4b and 4c combined in graph below for the wedge angle $\alpha = 8.19^\circ$.

The combinations of friction coefficients $\mu_{\text{kin}}$ and $\mu_{\text{kout}}$, positioned in red area of Figure 13, will cause malfunction of the “strand locking” action of the wedge (values of $\mu_{\text{kout}}$ in Figure 13 below zero are of no practical use).

Note: With a wedge angle of $8.19^\circ$, the inner friction coefficient has to be $\mu_{\text{kin}} > 0.14$ for a proper wedge operation!

For the interested reader, Appendix F clarifies several cases of wedge malfunction.

Several environmental circumstances and principles can have an influence on the value of $\mu_{\text{kout}}$:

- Adhesion: between wedge outer surface and tapered hole surface
- Plowing / third body effects: between wedge outer surface and tapered hole surface

For $\mu_{\text{kin}}$:

- Accumulation of dirt: between strand outer surface and inner surface of the wedge
- Wear on inner friction profile of wedge during load cycles

This last wear principle is considered in detail in the following chapter.
5 Wear by micro slip

In every load cycle (jack up and jack down) micro slip will occur between the wedges of the upper and lower anchor head and the strands. This is a result of the difference in axial strain (as a result of the load force \( F_L \)) between the cable and wedge in consequence of the difference in stiffness. The axial strain in the cable is larger than in the wedge. Hence the strain in the loaded cable can not be “followed” by the wedge what results in a relative displacement between the cable outer surface and wedge inner surface. Micro slip in combination with a load force results in wear of the inner friction surface of the wedge and eventually in a reduction of the inner friction coefficient \( \mu_{\text{kin}} \). Finally this will cause malfunction of the wedge (according to paragraph 4 and Appendix F).

The load force \( F_L \) and related strain in the strand increase during this movement until the anchor head bears the total present load force \( F_L \). At the end of the trajectory of the increasing load force \( F_L \) (the upper anchor head bears the total load force \( F_L \) ), the slip front has propelled to distance \( S \). In the cross section of the cable on position \( S \), the tensile stress \( \sigma_c = 0 \)

According to the calculations performed in Appendix G the position of slipfront \( S \) can be characterized as:

\[
S = \frac{F_L}{\mu_{\text{kin}} \cdot N_{\text{in}}} \cdot L \quad (5a)
\]

\( N_{\text{in}} \) = Normal force on inner wedge surface  
\( \mu_{\text{kin}} \) = Kinetic coefficient of friction on inner wedge surface  
\( F_L \) = Load force  
\( S \) = Position of slip front  
\( L \) = Total length of friction profile

An increasing force \( F_L \) results in a increasing value of \( s \), as soon as \( s > L \) total slip and \( F_L = F_{\text{max}} = \mu_{\text{kin}} \cdot N_{\text{in}} \) occurs.

- If \( s > L \) : macroslip occurs, the strand slips through the wedge  
- If \( 0 < F_L \leq F_{\text{max}} \) : microslip acts along length \( s \).

According to Appendix G the relative displacement \( u(x) \) between strand and wedge becomes:

\[
u(x) = \frac{F_L \cdot x^2}{2 \cdot E_c \cdot A_{\text{cable}}} \quad (5b)
\]
This equation is presented graphically in Figure 15.

\[ \frac{F \cdot S^2}{2 \cdot A_{cable} \cdot E_c} \]

Figure 15: Graph of relative displacement \(u(x)\) between the outer cable surface and inner friction profile of the wedge

\( F \) = Load force  
\( A_{cable} \) = Cross section area of cable  
\( E_c \) = E-modulus of cable material

The magnitude of relative displacement \(u(x)\) (and thus wear) of the outer cable surface against the friction profile depends on the location. Maximum slip (relative displacement) is located at the loaded “outlet” of the cable at \( x = S \) (see Figure 15).

Throughout the load cycles, wear takes place on the friction profile of the wedge due to the load force and relative motion (micro slip). The friction profile is reduced and flattened during operation (see Figure 16). Consequently the “strand gripping action” of the wedge is less.

Figure 16: Flattened friction profile of a wedge, mal-functioning after 501 strokes at nominal load force of 120 kN
6 Micro slip / wear reducing design

As indicated in Appendix G and chapter 5 the relative displacement \( u(x) \) between wedge friction profile and cable outer surface is characterized as (see Figure 17):

\[
u(x) = \frac{F_L \cdot x^2}{2 \cdot E_c \cdot A_{\text{cable}}} \tag{6a}
\]

Figure 17: The parabolic characterisation of the relative displacement \( u(x) \) between friction profile and cable outer surface.

\( F_L = \) Load force
\( A_{\text{cable}} = \) Cross section area of cable
\( E_c = \) E-modulus of cable material

According to Ref [3], the design of a micro slip/hysteresis reducing grip construction of a plate is equipped with “fingers” which can “absorb” the relative displacement \( u(x) \). The lengths of the “fingers” are arranged a parabolic pattern (conform the equation of \( u(x) \)). The length \( F \) of each finger can be designed proportional to the relative displacement \( u(x) \). The stiffness \( C \) of each finger is proportional to finger length \( F^{-3} \) (see Figure 18). The micro slip (and relative displacement \( u(x) \)) and thus friction is reduced by dimensioning the finger length locally in relation to \( u(x) \). Locally the stiffness of wedge is adapted; the wedge friction surface can now “follow” the displacement (or strain) of the outer cable surface and reduce the micro slip and thus wear of the grip profile. This results in the micro slip (and wear) reducing wedge designs of Figure 18.

Figure 18: Micro slip and wear reducing wedge designs A and B
Figure 18a is theoretically the best alternative, the length of the fingers is conform the parabolic equation of \( u(x) \). Practically the alternative B of Figure 18 can also reduce the wear on the inner grip profile and is very straightforward to produce. The exact amount, dimensions and pattern of the “fingers” must be determined by practical results.

### 6.1 Wear reduction on the base of vertical cable equilibrium

From the vertical cable equilibrium of Appendix E, equation 6.1a is obtained:

\[
\sigma_c = \frac{2}{R_i} \cdot \mu_{\text{kin}} \cdot \sigma_{\text{cw}} \quad (6.1a)
\]

With:

- \( \mu_{\text{kin}} \): Kinetic coefficient of friction on inner wedge surface and cable surface
- \( \sigma_{\text{cw}} \): Compressive stress between wedge and cable
- \( \sigma_c \): Tensile stress in cable
- \( S \): Position of slip front (for detailed information is referred to Appendix G2.2)
- \( R_i \): Inner radius of wedge and therefore outer radius of strand/cable

The values of the parameters \( \mu_{\text{kin}}, \sigma_{\text{cw}}, \sigma_c \) of equation 6.1a can be considered as:

- \( \mu_{\text{kin}} \): Constant, assuming a value between 0.05 and 1.
- \( \sigma_{\text{cw}} \): Constant, being a maximum design or material stress value
- \( \sigma_c \): Constant, being a maximum design or material stress value

As a result of the assumptions of the above, the ratio \( S/R_i \) of equation 6.1a must also have a constant value. Nevertheless numerous combinations of \( S \) and \( R_i \) can satisfy this condition.

Ergo, if we design a wedge configuration with the assumed values of \( \mu_{\text{kin}} = 0.2 \), \( \sigma_{\text{cw}} = 300 \text{ N/mm}^2 \), \( \sigma_c = 745 \text{ N/mm}^2 \) and \( \alpha = 8.19^\circ \), the ratio will be:

\[
\frac{S}{R_i} = 2 \cdot \frac{\sigma_c}{\mu_{\text{kin}} \cdot \sigma_{\text{cw}}} = 6.2 \quad (6.1b)
\]

and with equation 5a the ratio \( S/L \) becomes:

\[
\frac{S}{L} = \frac{F_L}{\mu_{\text{kin}} \cdot N_{\text{in}}} \quad (6.1c)
\]

and with \( N_{\text{in}} \) (see Figure 14) being roughly:

\[
N_{\text{in}} = \frac{F_L}{\tan \alpha} \quad (6.1d)
\]

Equation (6.1c) becomes:
If we combine equations 6.1b, 6.1c, 6.1d and 6.1e, Table 3 can be comprised:

![Equation Image]

If we combine equations 6.1b, 6.1c, 6.1d and 6.1e, Table 3 can be comprised:

### Calculated properties of different strand/wedge combinations

<table>
<thead>
<tr>
<th>Dimensions</th>
<th>Strand Capacity of one strand Position slip front Length of friction profile Weight/m</th>
<th>n strands needed for 900 ton</th>
</tr>
</thead>
<tbody>
<tr>
<td>Radius Ri (mm)</td>
<td>Steel Area (mm²)</td>
<td>Ratio S/Ri</td>
</tr>
<tr>
<td>9**</td>
<td>223**</td>
<td>6.2</td>
</tr>
<tr>
<td>6</td>
<td>113*</td>
<td>6.2</td>
</tr>
<tr>
<td>5</td>
<td>78.5</td>
<td>6.2</td>
</tr>
<tr>
<td>4.5</td>
<td>63.6</td>
<td>6.2</td>
</tr>
<tr>
<td>4</td>
<td>50.3</td>
<td>6.2</td>
</tr>
<tr>
<td>3</td>
<td>28.3</td>
<td>6.2</td>
</tr>
</tbody>
</table>

* Calculated with $n \cdot R_i^2$, ** Reference value of strand (Appendix C)

Table 3: Calculated wedge/strand configurations

When a strand with a diameter of 9 mm ($R_i=4.5$) is chosen for example, a solid single high capacity wire can be used. This solid 9 mm wire has better wear/handling properties than the original 18 mm strand;

- **No internal relative displacements between wires during operation: less elastic power is lost by friction/wear. Therefore no internal wear is initiated. In general a strand with a diameter of > 9 mm contains multiple wires, and the internal wear and elastic energy loss will increase drastically by an increasing number of strand wires. (See Figure 19)**

- **The 9 mm strand also improves handling due to a reduced weight (Table 3; a weight reduction of 1.25 kg/m!). In a 900-ton configuration of the strand jack unit SSL, 190 wires of 9 mm are used.**

![Figure 19](image-url)
6.2 “9 mm” wedge / anchor head configuration

When the 9 mm wire of paragraph 6.1 is applied, the anchor head /wedge configuration is modified. This configuration is shown in Figure 20.

Figure 20: Wedge / anchor head configuration with 190 “9 mm” wedges

The geometry of the “9 mm” wedge is presented in Figure 21.

Figure 21: The geometries of the “9 mm” wedge and the original 18 mm wedge
7 General conclusion

- According to the calculations performed in Appendix E the proper operation area of the wedge can be characterized as:

\[
\mu_{\text{kout}} < \frac{\mu_{\text{kin}} - \tan(\alpha)}{1 + \mu_{\text{kin}} \cdot \tan(\alpha)} \quad \text{and so:} \quad \mu_{\text{kin}} > \frac{\mu_{\text{kout}} + \tan(\alpha)}{1 - \mu_{\text{kout}} \cdot \tan(\alpha)}
\]

With:
- \( \mu_{\text{kout}} \) = Kinetic coefficient of friction between outer wedge surface and anchor head surface
- \( \mu_{\text{kin}} \) = Kinetic coefficient of friction on inner wedge surface and cable surface
- \( \alpha \) = Wedge angle
- \( F_L \) = Load force

The wedge gripping quality depends on inner-and outer friction coefficients \( \mu_{\text{in}}, \mu_{\text{out}} \) and the wedge angle \( \alpha \). The characteristic graph of Figure 13 provides the proper operation area and malfunction area of the wedge, as a function of \( \mu_{\text{in}}, \mu_{\text{out}} \) (with \( \alpha = 8.19^\circ \)).

Several environmental circumstances and principles can have an influence on the value of \( \mu_{\text{kout}} \):

- **Adhesion**: between wedge outer surface and tapered hole surface
- **Plowing /third body effects**: between wedge outer surface and tapered hole surface

For \( \mu_{\text{kin}} \):
- **Accumulation of dirt**: between strand outer surface and inner surface of the wedge
- **Wear on inner friction profile of wedge during load cycles**

- During operation relative displacement occurs between the inner friction profile and strand outer surface. The relative displacement \( u(x) \) can be characterized as:

\[
u(x) = \frac{F_L \cdot x^2}{2 \cdot E_c \cdot A_{\text{cable}}}
\]

\( F_L \) = Load force
\( A_{\text{cable}} \) = Cross section area of cable
\( E_c \) = E-modulus of cable material

This relative displacement which initiates wear is called *micro slip*.

- The micro slip reducing design, presented in this report, can increase service life of the wedge.

- By applying 190 high capacity solid strands with diameter of 9 mm the following objectives can be realized:
No internal relative displacements between wires during operation: less elastic power is lost by friction/wear. Therefore no internal wear is initiated. In general a strand with a diameter of > 9 mm contains multiple wires, and the internal wear and elastic energy loss will increase drastically by an increasing number of strand wires.

The 9 mm strand also improves handling due to a reduction of 1.25 kg/m compared to the original 18 mm strand. In a proper 900-ton configuration of the strand jack unit SSL, 190 wires of 9 mm are used.

- In this 9 mm configuration, 380 small 9 mm “low cost” wedges are used.

### 7.1 Future perspectives/ Recommendations

- Cost / service life ratio in case of the 9 mm strand is an important issue to be studied.

- The wear and reliability tests on the alternative wedge designs have to be evaluated (conform standard procedures) with the reference properties attained from a reference test.

- **Less influence of dirt, corrosion and environment can be accomplished with a totally new strand jack design.** A new construction can be investigated with a cable “locking” system not dominated by friction (in case of the current wedge principle, operation quality is dominated by friction coefficients and thus by the operation environment), resulting in a very reliable lifting system not influenced by the environment.
References

APPENDICES
STRANDJACK WEDGES
Friction coefficients, micro slip and handling

Report nr. DCT 2005-78

By: H.G.M.R. van Hoof
Idnr: 501326

20 November, 2005
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APPENDIX A Anchorheads

An anchor head is drilled with a number of tapered holes suitable for accepting jack temporary wedges. An anchor head of the 830 SSL strand jack unit for example contains 54 tapered holes (Figure 3). In Figure 1 a characteristic example of an anchor head is presented. The quantity of tapered holes of the anchor heads is scaled to the required capacity.

The main specifications of the anchor heads are presented in Table 1.

Table 1: Specifications Anchor heads. Source: [Mammoet]

<table>
<thead>
<tr>
<th>Specifications</th>
<th>SSL830</th>
<th>SSL550</th>
<th>SSL300</th>
<th>SSL100</th>
</tr>
</thead>
<tbody>
<tr>
<td>Main dimensions (mm)</td>
<td>Ø520x190</td>
<td>Ø450x190</td>
<td>Ø354x100</td>
<td>Ø200x150</td>
</tr>
<tr>
<td>Weight (kg)</td>
<td>271</td>
<td>207</td>
<td>60</td>
<td>40</td>
</tr>
<tr>
<td>Material</td>
<td>34CrNiMo6</td>
<td>34CrNiMo6</td>
<td>34CrNiMo6</td>
<td>34CrNiMo6</td>
</tr>
<tr>
<td>Wedges / holes (Qty.)</td>
<td>54</td>
<td>36</td>
<td>18</td>
<td>7</td>
</tr>
<tr>
<td>Layer thickness surface treatment (mm)</td>
<td>0.1-1.6</td>
<td>0.1-1.6</td>
<td>0.1-1.6</td>
<td>0.1-1.6</td>
</tr>
<tr>
<td>Solid lubricant</td>
<td>Molycote D321 R</td>
<td>Molycote D321 R</td>
<td>Molycote D321 R</td>
<td>Molycote D321 R</td>
</tr>
</tbody>
</table>

Note that the surface treatment of the anchor heads is nitro carburizing.
In order to reduce friction, the wedge seats of the anchor head are pre-treated with Molykote D321 R, a dry lubricant (see Figure 4).

Figure 4: Molykote D321 R Pre-treated Anchor head
APPENDIX B  WEDGE PROPERTIES

The wedges used in the strand jack units contain three wedge parts. The wedge is equipped with a friction or grip profile on the inside of each part.

The main dimensions of the wedge and wedge parts are presented in Figure 5 and Figure 6.

The three wedge parts are assembled with an elastic rubber ring. This ring retains the three wedge segments/parts together during operation (Figure 7).

For main specifications of the currently used TT 18/44 wedge is referred to Table 2 (In Table 2 several specifications are noted with "indication", these values need to be reviewed by hardness tests). In order to reduce friction, the wedges are pre-treated before operation with Molykote D321 R, a dry lubricant.
APPENDIX C Ø18 mm DYFORM strands

The strand bundle contains several Dyform strands with a maximum operation capacity of 167 kN/strand (specified by Mammoet). The quantity of strands depends on the used unit type. The Dyform strand contains 7 twisted, high capacity, steel wires with a flattened side (see Figure 8). They are specially developed for heavy lifting and its length can be up to 1500 meters to meet the job requirements.

The main specifications of the Dyform strand are presented in Table 3.

Table 3: Main specifications Dyform strand. Source: Bridon wire Ltd

<table>
<thead>
<tr>
<th>Strand</th>
<th>BSS896 (DYFORM)</th>
<th>Nominal values</th>
<th>Tolerances</th>
</tr>
</thead>
<tbody>
<tr>
<td>Nominal diameter (mm) :</td>
<td>Ø18</td>
<td>+0.4 -0.2</td>
<td></td>
</tr>
<tr>
<td>Mass (kg/m) :</td>
<td>1.75</td>
<td>+0.4% -2%</td>
<td></td>
</tr>
<tr>
<td>Tensile strength (Rm) (N/mm²)</td>
<td>1700</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Surface hardness of wires (HV)</td>
<td>430-480</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Steel area (mm²)</td>
<td>223</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Breaking load (Fm) (kN)</td>
<td>380</td>
<td></td>
<td></td>
</tr>
<tr>
<td>0.1% proof load (Fp 0.1) (kN)</td>
<td>323</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Load at 1% elongation (Ft 1.0) (kN)</td>
<td>334</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Wires (Qty)</td>
<td>7</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Manufacturer :</td>
<td>Carrington Wire Ltd / Bridon Wire Ltd</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
APPENDIX D Operation cycles Strandjack unit

In this appendix a review is presented on two main operation cycles of the strand jack unit SSL to familiarize the reader with the general concepts of the Strand jack-unit SSL:

- Jack up cycle (lifting the load)
- Jack down cycle (lowering the load)

D.1 Jack up cycle (lifting the load)

In the next paragraph the Jack up cycle of the hydraulic Strand jack units is described. This principle concerns all types.

The start position (see Figure 9):
The 18 mm dyform compact strands are installed through the unit and the load is connected. Both upper and lower anchor heads are “locked” (red in Figure 9).

Step 1 (see Figure 9): The piston of the jack extends and raises the upper anchor head including the locked strands and connected load. During this movement the wedges in the lower anchor head are pulled up slightly by the movement of the strands; the lower anchor head is still closed. In case of failure of the upper anchor head, the wedges in the lower anchor head secure the load.
Figure 10: Step 2 and Step 3 of the Jack up cycle

**Step 2 (see Figure 10):**
In top position the load is transferred from the upper anchor head to the lower anchor head by slightly retracting the piston, approx. 15 mm. During this load transfer both anchor heads remain locked.

**Step 3 (see Figure 10):**
After the transfer, the upper anchor head is opened hydraulically. The upper release cylinder lifts the release plate with the release tubes and shifts the wedges/grips up and out of their seatings (Note the difference between left and right detail picture). The upper anchor head is now “unlocked” (blue in Figure 10) allowing free passage of the strands.
Figure 11: Step 4 and Step 5 of the Jack up cycle

Step 4 (see Figure 11):
With the upper anchor head “unlocked”, the piston of the jack retracts and returns to the starting position. The strands slide through the wedges in the upper anchor head.

Step 5 (see Figure 11):
At the end of step 4, the upper anchor head is “locked” again by the upper release cylinder retracting the release plate and release tubes.

The jack up cycle of step 1 to 5 is repeated till the required jack up distance is accomplished.
D.2 Jack down cycle (lowering the load)

In the next paragraph the Jack down cycle of the hydraulic Strand jack units is described. This principle concerns all types.

The start position (see Figure 12):
The 18 mm dyform compact strands are installed through the unit and the load is connected. Both upper and lower anchor heads are “locked” (red in Figure 12).

Step 1 (see Figure 12):
The upper anchor head is opened hydraulically. The upper release cylinder lifts the release plate with the release tubes and shifts the wedges/grips up and out of their seatings (Note the difference between detail picture of the upper anchor head in Start position and Step1). The upper anchor head is now “unlocked” (blue in Figure 12) allowing free passage of the strands.

Figure 12: Start position and Step 1 of the Jack down cycle
Figure 13: Step 2 and Step 3 of the Jack down cycle

Step 2 (see Figure 13):
With the upper anchor head “unlocked”, the piston of the jack extends till approx. 15 mm before end of the outward stroke (compare the position of the main hydraulic cylinder in step 1 and step 2). The strands slide through the wedges in the upper anchor head during the stroke.

Step 3 (see Figure 13):
In the position “15 mm before end of the outward stroke” the upper anchor head is “locked” by the upper release cylinder retracting the release plate and release tubes (red in detail upper anchor head in see Figure 13 step 3).
Step 4 (see Figure 14):
The load is transferred from the lower anchor head to the upper anchor head by slightly extending
the piston of the main cylinder further (approx. +15 mm) to the end of the stroke. During this load
transfer both anchor heads remain “locked” (both red in Figure 14 step 4).
During this movement the wedges in the lower anchor head are pulled up slightly by the
movement of the strands; the lower anchor head is still closed. In case of failure of the upper
anchor head, the wedges in the lower anchor head secure the load.

Step 5 (see Figure 14):
After the transfer, the lower anchor head is opened hydraulically (“unlocked”; blue in detail of
lower anchor head step 5). The lower release cylinder lifts the release plate with the release
tubes and shifts the wedges/grips up and out of their seatings (Note the difference between left
and right detail picture step 4 and step 5). The lower anchor head is now “unlocked” (blue in
Figure 14) allowing free passage of the strands.
**Step 6 (see Figure 15):**
The piston of the jack retracts and lowers the closed upper anchor head including the strands and load (movement is stroke) until approx. 15 mm before end of the inward stroke. The lower anchor head is still “unlocked” (blue in Figure 15) during this movement and allows free passage of the strands.

**Step 7 (see Figure 15):**
In the position “15 mm before end of inward stroke”, the lower anchor head is “locked” hydraulically by the lower release cylinder retracting the release plate and release tubes (red in detail lower anchor head in Figure 15 step 7).
Step 8 (see Figure 16):

After “locking” the lower anchor head, the load is transferred from the upper anchor head to the lower anchor head by slightly retracting the piston of the main cylinder further to the end of the inward stroke, approx. -15 mm. During this load transfer both anchor heads remain locked.

The jack down cycle of step 1 to 8 is repeated until the required jack down distance is accomplished.
APPENDIX E  Calculations wedge model

In this appendix a mathematical model of the strand locking system is presented. In the following considerations a single wedge part is examined and the spring force of the pre-stress spring is neglected.

Figure 17: Schematic presentation of mathematical wedge model

- $\mu_{\text{out}}$: Kinetic coefficient of friction between outer wedge surface and anchor head surface
- $\mu_{\text{kin}}$: Kinetic coefficient of friction on inner wedge surface and cable surface
- $N_{\text{out}}$: Normal force on outer wedge surface
- $N_{\text{in}}$: Normal force on inner wedge surface
- $\sigma_{\text{cw}}$: Compressive stress between wedge and cable
- $\sigma_{\text{Nout}}$: Compressive stress perpendicular wedge surface and anchor head
- $\sigma_{\text{c}}$: Tensile stress in cable as result of $F_L$
- $F_L$: Load force
- $\alpha$: Wedge angle
- $\tau_{\text{out}}$: Friction shear stress between outer wedge surface and anchor head
- $\tau_{\text{cw}}$: Friction shear stress between wedge and cable
- $S$: Position of slip front (for detailed information is referred to Appendix G2.2)
- $R_i$: Inner radius of wedge and therefore outer radius of strand/cable
- $A_{\text{cable}}$: Cross section area of cable

From Figure 17, the following equations can be obtained:
The estimation of the theoretical outer and inner surface of a single wedge part (assuming cylindrical), $A_{in}$ and $A_{out}$, is formulated as:

$$A_{in} = \frac{2}{3} \pi R_1 S$$  \hspace{1cm} (c)

$$A_{out} = \frac{2}{3} \pi R_out \cdot \frac{S}{\cos(\alpha)}$$  \hspace{1cm} (d)

This according to Figure 18.

The estimated projected effective outer and inner surface of a single wedge part, $A_{ineff}$ and $A_{outeff}$, can formulated as:

$$A_{ineff} = R_1 \cdot S \cdot \sqrt{3}$$  \hspace{1cm} (e)  and  \hspace{1cm} $$A_{outeff} = R_out \cdot \frac{S}{\cos(\alpha)} \cdot \sqrt{3}$$  \hspace{1cm} (f)

These equations can be derived from Figure 19.
The total normal force $N_{\text{out}}$ acting perpendicular on the effective surface of the wedge part becomes (see also Figure 17):

$$N_{\text{out}} = \sigma_{\text{Nout}} \cdot A_{\text{outeff}} = \sigma_{\text{Nout}} \cdot R_{\text{out}} \cdot \frac{S}{\cos(\alpha)} \cdot \sqrt{3} \quad (g)$$

The total normal force $N_{\text{in}}$ acting perpendicular on the inner surface of the wedge part becomes (see also Figure 17):

$$N_{\text{in}} = \sigma_{\text{cw}} \cdot A_{\text{ineff}} = \sigma_{\text{cw}} \cdot R_{i} \cdot S \cdot \sqrt{3} \quad (h)$$

### E.2 Horizontal wedge equilibrium

The total equilibrium of the horizontal forces on the wedge part results in the following equation (according to Figure 17):

$$N_{\text{out}} \cdot \cos(\alpha) = N_{\text{in}} - \tau_{\text{out}} \cdot \sin(\alpha) \cdot A_{\text{outeff}} \quad (i)$$

After substitution of equations $a$, $g$, $h$ and $f$ in $i$

$$\sigma_{\text{Nout}} \cdot R_{\text{out}} \cdot \frac{S}{\cos(\alpha)} \cdot \sqrt{3} \cdot \cos(\alpha) = \sigma_{\text{cw}} \cdot R_{i} \cdot S \cdot \sqrt{3} - \mu_{\text{kout}} \cdot \sigma_{\text{Nout}} \cdot \sin(\alpha) \cdot R_{\text{out}} \cdot \frac{S}{\cos(\alpha)} \cdot \sqrt{3}$$

After solving and rearranging, we get

$$\sigma_{\text{Nout}} \cdot R_{\text{out}} \cdot (1 - \mu_{\text{kout}} \cdot \tan(\alpha)) = \sigma_{\text{cw}} \cdot R_{i} \quad \text{or} \quad \sigma_{\text{Nout}} \cdot \frac{R_{\text{out}}}{R_{i}} \cdot (1 - \mu_{\text{kout}} \cdot \tan(\alpha)) = \sigma_{\text{cw}} \quad (j)$$
**E.3 Vertical wedge equilibrium**

The total equilibrium of the vertical forces on the wedge part results in the following equation (according to Figure 17):

\[ A_{in} \cdot \tau_{cw} - \tau_{out} \cdot \cos(\alpha) \cdot A_{out} - \sigma_{Nout} \cdot \sin(\alpha) \cdot A_{out} = 0 \]  

\[ + \]  

\[ (k) \]

After substitution of equations b, c, a, d in k, we get

\[
\frac{2}{3} \pi \cdot R_{i} \cdot S \cdot \mu_{kin} \cdot \sigma_{cw} = \mu_{kout} \cdot \sigma_{Nout} \cdot \cos(\alpha) \cdot \frac{2}{3} \pi \cdot R_{out} \cdot \frac{S}{\cos(\alpha)} - \\
\sigma_{Nout} \cdot \sin(\alpha) \cdot \frac{2}{3} \pi \cdot R_{out} \cdot \frac{S}{\cos(\alpha)} = 0
\]

After solving and rearranging, this results in

\[
R_{i} \cdot \mu_{kin} \cdot \sigma_{cw} = \sigma_{Nout} \cdot R_{out} \cdot (\mu_{kout} + \tan(\alpha))
\]

or

\[
\mu_{kin} \cdot \sigma_{cw} = \sigma_{Nout} \cdot \frac{R_{out}}{R_{i}} \cdot (\mu_{kout} + \tan(\alpha))
\]

\[ (l) \]

**E.4 Wedge operating range**

Substituting equation j in l results in

\[
\mu_{kin} \cdot \sigma_{Nout} \cdot \frac{R_{out}}{R_{i}} \cdot (1 - \mu_{kout} \cdot \tan(\alpha)) = \sigma_{Nout} \cdot \frac{R_{out}}{R_{i}} \cdot (\mu_{kout} + \tan(\alpha))
\]

After rearranging

\[
\mu_{kout} = \frac{\mu_{kin} - \tan(\alpha)}{1 + \mu_{kin} \cdot \tan(\alpha)} \quad \text{and so} \quad \mu_{kin} = \frac{\mu_{kout} + \tan(\alpha)}{1 - \mu_{kout} \cdot \tan(\alpha)}
\]

\[ (m) \]

These two equations represent the “borderline” of the wedge operating range.
E.5 Vertical cable equilibrium

When the equilibrium of the vertical forces on the cable/strand in Figure 17 is considered, the following equation can be acquired:

\[ F_L = 2 \cdot \pi \cdot R_i \cdot \tau_{cw} \cdot S \]  \hspace{1cm} (n)

With

\[ F_L = \sigma_c \cdot A_{cable} = \sigma_c \cdot \pi \cdot R_i^2 \]  \hspace{1cm} (o)

and

\[ \tau_{cw} = \mu_{kin} \cdot \sigma_{cw} \]  \hspace{1cm} (b)

After substitution of \( o \) and \( b \) in equation \( n \), the result is:

\[ \sigma_c \cdot \pi \cdot R_i^2 = 2 \cdot \pi \cdot R_i \cdot \mu_{kin} \cdot \sigma_{cw} \cdot S \]  \hspace{1cm} (p)

After rearranging:

\[ \sigma_c = 2 \cdot \frac{S}{R_i} \cdot \mu_{kin} \cdot \sigma_{cw} \]  \hspace{1cm} (q)
APPENDIX F  Malfunction of the wedge

For the benefit of the analysis, three possible cases are described to clarify the malfunction of the wedge, according to the mechanical wedge model:

1. Wedge operation with : $\mu_{\text{kin}}$ Constant , increasing $\mu_{\text{kout}}$
2. Wedge operation with : $\mu_{\text{kout}}$ Constant , decreasing $\mu_{\text{kin}}$
3. Wedge operation with : Increasing $\mu_{\text{kout}}$ , decreasing $\mu_{\text{kin}}$

In the following paragraphs these load cases are described in detail.

**F.1 Wedge operation with : $\mu_{\text{kin}}$ Constant , increasing $\mu_{\text{kout}}$**

When the wedge operation is started, a proper and normal operation of the wedge is assumed. This implies that the friction conditions are below the "good/bad" borderline in Figure 20. For example, fictive estimated start values of $\mu_{\text{kout}}$ and $\mu_{\text{kin}}$ in operation are 0.2 and 0.5 (see start operation point A in Figure 20).

![Figure 20: Wedge operation with increasing $\mu_{\text{kout}}$](image)

With increased $\mu_{\text{kout}}$, the operation point B is positioned in the red "Malfunction Area". The friction force between wedge outer surface and tapered hole surface is too high and consequently gripping action is less; the strand slips through the wedge.
F.2 Wedge operation with: $\mu_{kout}$ Constant, decreasing $\mu_{kin}$

When the wedge operation is started, a proper and normal operation of the wedge is assumed. For example, fictive estimated start values of $\mu_{kout}$ and $\mu_{kin}$ in operation are 0.2 and 0.5 (see start operation point A in Figure 21).

![Figure 21: Wedge operation with decreasing $\mu_{kin}$](attachment:figure_21.png)

When $\mu_{kin}$ decreases during operation below the value ≤ 0.35 and the value $\mu_{kout}$ is constant, the operation point of the wedge is moved from A to point C (see Figure 21). Operation point C is positioned in the red “Malfunction Area”. The friction force between wedge inner surface and cable surface is too low, consequently the gripping action is less; the strand slips through the wedge.

F.3 Wedge operation with: Increasing $\mu_{kout}$, decreasing $\mu_{kin}$

In most practical cases the values of the friction coefficients will change simultaneously, influenced by environmental circumstances (temperature, relative humidity etc.).
A combination of increasing $\mu_{\text{out}}$ and decreasing $\mu_{\text{in}}$ will lead to malfunction of the wedge conform Figure 22, the operation point of the wedge is moved from A to point D. Malfunction of the wedge appears.

Figure 22: Wedge operation with decreasing $\mu_{\text{in}}$ and increasing $\mu_{\text{out}}$
APPENDIX G  Micro slip during jack up

In every load cycle (jack up and jack down) micro slip will occur in the wedges of the upper and lower anchor head. This is a result of the difference in axial strain (as a result of the load force $F_L$) between the cable and wedge in consequence of the difference in stiffness. The axial strain in the cable is larger than in the wedge. Hence the strain in the loaded cable can not be “followed” by the wedge what results in a relative displacement between the cable outer surface and wedge inner surface. Micro slip in combination with a load force results in wear of the inner friction surface of the wedge and eventually in a reduction of the inner friction coefficient $\mu_{kin}$. Finally this will cause malfunction of the wedge.

When the piston of the jack extends, the closed upper anchor head is raised including the strands. The load force $F_L$ and related strain in the strand increase during this movement until the anchor head bears the total present load force $F_L$. The situation of the full loaded wedge in the anchor head is described schematic in Figure 23.

\[F_L = \text{Load force}\]
\[V = \text{Spring force (pre-stress)}\]
N_{in} =\text{Normal force on inner wedge surface} \\
\mu_{\text{kin}} =\text{Kinetic coefficient of friction on inner wedge surface} \\
Q =\text{Length arc of wedge part} \\
\alpha =\text{Wedge angle} \\
\sigma_{cw} =\text{Compressive stress between wedge and cable} \\
\sigma_{c} =\text{Tensile stress in cable as result of } F_{L} \\
\varepsilon_{c} =\text{Strain in cable} \\
S =\text{Position of slip front} \\
L =\text{Length of friction profile} \\
A_{\text{cable}} =\text{Cross section area of cable} \\
E_{c} =\text{Modulus elasticity of cable} \\
\tau_{cw} =\text{Friction shear stress} \\

At the end of the trajectory of the increasing load force $F_{L}$ (the upper anchor head bears the total load force $F_{L}$), the slip front has propelled to distance $S$. In the cross section of the cable on position $S$, the tensile stress $\sigma_{c} = 0$. Along the inner friction surface of the wedge part, width $Q$ and length $L$, exists a compressive stress $\sigma_{cw}$ between wedge and cable as a result of the normal force $N_{in}$. The (pre-stress) spring force $V$ is neglected and the stiffness of the wedge is considered infinite in the following considerations.

This compressive stress $\sigma_{cw}$ present on the 3 wedge parts is formulated as:

$$\sigma_{cw} = \frac{N_{in}}{3 \cdot Q \cdot L}$$

The compressive stress $\sigma_{cw}$ is able to transmit an average friction shear stress $\tau$ on the contact area:

$$\tau_{cw} = \mu_{\text{kin}} \cdot \sigma_{cw}$$

The load force $F_{L}$ is compensated by three inner friction surfaces, with an area of $Q \cdot s$ and thus is derived for $F_{L}$:

$$F_{L} = 3 \cdot Q \cdot s \cdot \tau_{cw} = 3 \cdot Q \cdot s \cdot \mu_{\text{kin}} \cdot \sigma_{cw} = \frac{3 \cdot Q \cdot s \cdot \mu_{\text{kin}} \cdot N_{in}}{3 \cdot Q \cdot L} = \frac{s \cdot \mu_{\text{kin}} \cdot N_{in}}{L}$$

A increasing force $F_{L}$ results in a increasing value of $s$, as soon as $s > L$ total slip and $F_{L} = F_{\text{max}} = \mu_{\text{kin}} \cdot N_{in}$ occurs.

- If $s > L$ : macroslip occurs, the strand slips through the wedge
- If $0 < F_{L} < F_{\text{max}}$ : microslip acts along length $s$.

For the tensile stress $\sigma_{c}$ and strain $\varepsilon_{c}$ in the cable we can write

$$\sigma_{c} = \frac{F_{L}}{A_{\text{cable}}} \quad \varepsilon_{c} = \frac{\sigma_{c}}{E_{c}} = \frac{F_{L}}{E_{c} \cdot A_{\text{cable}}}$$

The length $s$ can be calculated from the relation

$$s = \frac{F_{L}}{\mu_{\text{kin}} \cdot N_{in}} \cdot L \quad (\text{Eq. G.1})$$
The tensile stress $\sigma_c$ diminishes linear with values on positions $x = 0$ to $x = s$ (see Figure 23) according to

$$\sigma_{c(x=0)} = 0 \quad \text{and} \quad \sigma_{c(x=s)} = \frac{F_L}{A_{\text{cable}}}$$

And analogically with the equations for the tensile stress $\sigma_c$, equations for strain $\varepsilon_c$ are

$$\varepsilon_{c(x=0)} = \frac{\sigma_c}{E_c} = \frac{0}{E_c \cdot A_{\text{cable}}} = 0 \quad \text{and} \quad \varepsilon_{c(x=s)} = \frac{\sigma_c}{E_c} = \frac{F_L}{E_c \cdot A_{\text{cable}}}$$

A graph of the x-position against strain $\varepsilon_c$ is provided in Figure 24.

With usage of Figure 24 the total cable elongation $dl$ in the wedge is derived. The elongation $dl$ of the cable in the wedge corresponds with area $P$

$$dl = P = \frac{1}{2} \cdot \varepsilon_{c(x=s)} \cdot s$$

With

$$\varepsilon_{c(x=s)} = \frac{\sigma_c}{E_c} = \frac{F_L}{E_c \cdot A_{\text{cable}}} \quad \text{and} \quad s = \frac{F_L \cdot L}{\mu_{\text{kin}} \cdot N_{\text{in}}}$$

After substitution, $dl$ becomes
\[
\frac{dl}{2} = \frac{\varepsilon_{c(x=0)} \cdot s}{2 \cdot E_c \cdot A_{cable} \cdot \mu_{\text{kin}} \cdot N_{in}} 
\]

Function \( \varepsilon_c(x) \) can be described as (see Figure 24)

\[
\varepsilon_c(x) = \frac{F_L \cdot x}{E_c \cdot A_{cable}} \quad \text{for} \quad 0 \leq x \leq s
\]

With the definition \( \frac{du}{dx} = \varepsilon_c(x) \) follows

\[
\frac{du}{dx} = \frac{F_L \cdot x}{E_c \cdot A_{cable}}
\]

To obtain the local relative displacement function \( u(x) \)

\[
\int \left[ \frac{du}{dx} \right] \cdot dx = \int \left[ \frac{F_L \cdot x}{E_c \cdot A_{cable}} \right] \cdot dx \quad \text{for} \quad 0 \leq x \leq s
\]

This leads to the relative displacement function \( u(x) \)

\[
u(x) = \frac{F_L \cdot x^2}{2 \cdot E_c \cdot A_{cable}} + C \quad \text{for} \quad 0 \leq x \leq s
\]

With \( u(0) = 0 \) consequently \( C = 0 \), relative displacement function \( u(x) \) becomes

\[
u(x) = \frac{F_L \cdot x^2}{2 \cdot E_c \cdot A_{cable}} \quad \text{for} \quad 0 \leq x \leq s \quad (\text{Eq. G.2})
\]
Equation G-2 is presented graphically in Figure 25.

The magnitude of relative displacement $u(x)$ (and thus wear) of the outer cable surface against the friction profile depends mainly on the location of the slip front $S$, assuming $F_L$, $A_{cable}$, $E_c$ constant. Maximum slip is located at the loaded “outlet” of the cable at position $S$. 

Figure 25: Graph of relative displacement $u(x)$ of the outer cable surface against inner friction profile