FATIGUE BEHAVIOUR OF A 24 MM LOCKED COIL WIRE ROPE

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SUMMARY

This paper gives results of trials in bend over sheave fatigue and tension fatigue for a 24 mm locked coil wire rope. General laws are then deduced that give the number of cycles to failure versus:

- average stress and diameter ratio for flexion over sheave,
- average stress and fluctuating stress for tension fatigue.

Results with and without "end effect" permit to calculate the value of that "end effect". Comparing with results of reverse bending made at the Central Mining Institute of Katowice the law of fatigue due to flexion reversal is deduced.
REFERENCES

1. Fatigue tests on hoisting wire ropes of diameters above 50 mm.

2. Calculation of rope drives.
FEYRER K. Wire N° 2. 83.

3. Damage study for a cable submitted to reverse bending over pulleys.
CLEMENT Ph. OIPEEL Bulletin N° 45.

4. Comparison of bend over sheave fatigue behavior of different ropes.
CLEMENT Ph. 150 Years Wire Rope Colloquium.

5. Fatigue of a wire rope bending on a pulley-working out a general rule giving the amount of cycles to breakage by the knowledge of damage.
CLEMENT Ph. OIPEEL Bulletin N° 40.

6. Accelerated block tension fatigue testing of wire ropes for offshore use.
YEUNG Y.C.T., WALTON J.M.
OIPEEL Round Table. Glasgow, June 1985.
NOMENCLATURE

\( N \) Number of cycles to failure
\( S \) Steel area, \( \text{mm}^2 \)
\( T \) Axial load, daN
\( \sigma \) Axial stress (\( T/S \)), daN/\( \text{mm}^2 \)
\( d \) Rope diameter, \( \text{mm} \)
\( D \) Sheave diameter, \( \text{mm} \)
\( \Delta \sigma \) Stress fluctuation (\( \sigma_{\text{max}} - \sigma_{\text{min}} \)), daN/\( \text{mm}^2 \)
\( L_e \) Equivalent load range
\( N_f \) Cycles in flexion
\( N_a \) Cycles in reverse bending
\( N_B \) Cycles for end effect
\( N_i \) Cycles for reversal
\( N_t \) Cycles in tension fatigue
\( \alpha \) Angle of contact
\( \ell \) Contact length
\( \ln \) Neperian logarithm
\( \log \) Decimal logarithm
### 1 - WIRE ROPE CHARACTERISTICS

Construction data:

<table>
<thead>
<tr>
<th>LAY</th>
<th>NUMBER OF WIRE</th>
<th>DIAMETER OF WIRE (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Central wire</td>
<td>1</td>
<td>1.7</td>
</tr>
<tr>
<td>1</td>
<td>6</td>
<td>1.57</td>
</tr>
<tr>
<td>2</td>
<td>8</td>
<td>2.8</td>
</tr>
<tr>
<td>3</td>
<td>8 + 8</td>
<td>2.47 - 1.8</td>
</tr>
<tr>
<td>4</td>
<td>13 + 13</td>
<td>2.5</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Double grooved 2.5 depth</td>
</tr>
<tr>
<td>5</td>
<td>28</td>
<td>Z wire 2.75 depth</td>
</tr>
</tbody>
</table>

Material: Galvanized steel
Breaking load: 588 000 N
Torque balanced wire rope
Spaces between wires are filled with a high viscosity grease.
2 - BEND OVER SHEAVE FATIGUE

2.1 - TESTING MACHINE

The machine used is schematized on Figure 1. Loading is applied by hydraulic actuators, and driving is achieved through a connecting rod and a crank system operated by an electric motor and gear.

The maximum load on the sample is 29 400 daN.

The machine is operated at a 0.25 Hz frequency.

The sheaves are made of E26 steel with a groove diameter 1.05 d.
2.2 - THE TESTS ACHIEVED

Three series of tests were performed, with two diameters of sheave and constant stroke of 3.1 m:

1. D - 1.5 m
   Two bends per cycle
   $\alpha = 168^\circ$
   $\ell = 2.2$ m

\[ \ell_2 = 2.1 \text{ m} \quad \ell_1 = 0.9 \text{ m} \quad \ell_2 = 2.1 \text{ m} \]

One bend Two bends One bend

2. D - 2.5 m
   Two bends per cycle
   $\alpha = 50^\circ$
   $\ell = 1.09$ m

\[ \ell_2 = 1.09 \text{ m} \quad \ell_1 = 2.01 \text{ m} \quad \ell_2 = 1.09 \text{ m} \]

One bend Two bends One bend

3. D - 2.5 m
   One bend per cycle
   $\alpha = 173^\circ$
   $\ell = 3.77$ m

\[ \ell_2 = 3.1 \text{ m} \quad \ell_3 = 1.34 \text{ m} \quad \ell_2 = 3.1 \text{ m} \]

One bend Zero bend One bend

The results are gathered in table 1.
2.3 - RESULTS

The results are gathered in Table 1 and plotted on Figure 2, in a diagram \((\sqrt{\sigma} \cdot \frac{d}{D}, \log N)\) and on Figure 3, in a diagram \((\log \sigma, \log N)\). The figures show that the parameter \(\sqrt{\sigma} \cdot \frac{d}{D}\) is not characteristic of fatigue of the locked coil rope and that the FEYRER formula fit well with results. The regression calculation on points lined up gives:

\[
\log N_f = -5.487 \log \frac{1}{\sigma} + 6.334 \log \frac{D}{d} + 3.589 \log \frac{1}{\sigma} \log \frac{D}{d} - 4.8038,
\]

with a standard deviation of 0.15.

Points \(\sigma = 59.6, D = 1.5\) and \(\sigma = 69.5, D = 2.5\) seem to be on the high gradiented part of the graph.

The lines for \(D = 1.5\) and \(2.5\) m are drawn on Figure 3.

2.4 - VALUE OF THE END EFFECT

The comparison of the results with and without "end effect" on the 2.5 m sheave, allows to calculate the value of the "end effect".

The results of bend over sheave fatigue on 2.5 m sheave with end effect, give by linear regression:

\[
N = 1.3992 \times 10^8 \sigma^{-1.881}
\]

The corresponding line and points are plotted on Figure 4.
For each value of \( \sigma \), we calculate \( N, N_F \) corresponding to a flexion without "end effect" and then applying Mink's law, the value of \( N_B \) due to the end effect by:

\[
\frac{1}{N} = \frac{1}{N_F} + \frac{1}{N_B}
\]

The results are summarized in the following table.

<table>
<thead>
<tr>
<th>( \sigma )</th>
<th>( N )</th>
<th>( N_F )</th>
<th>( N_B )</th>
</tr>
</thead>
<tbody>
<tr>
<td>29.8</td>
<td>235 926</td>
<td>244 866</td>
<td>6.46 ( \times ) 10^6</td>
</tr>
<tr>
<td>37.2</td>
<td>155 445</td>
<td>165 900</td>
<td>2.467 ( \times ) 10^6</td>
</tr>
<tr>
<td>44.7</td>
<td>110 036</td>
<td>120 190</td>
<td>1.302 ( \times ) 10^6</td>
</tr>
</tbody>
</table>

By regression:

\[ N_B = 4.3879 \times 10^{12} \sigma^{-3.9637} \]

From reference 5, COMMA I results give for \( \frac{d}{D} = 0.0096 \):

\[ N_B = 3.92 \times 10^7 \sigma^{-1.657} \]

The locked coil rope appears less sensitive to the "end effect" with greater influence of \( \sigma \).
2.5 - ESTIMATION OF FATIGUE DUE TO FLEXION REVERSAL

A comparison with results of Dr. ANKUS at the Central Mining Institute (Ref. 1) permits an estimation of the fatigue due to flexion reversal.

The results of Dr. ANKUS for a locked coil rope of 46 mm bended with two sheaves, as shown in the scheme, are gathered in the next table:

![Diagram of a coil rope with sheaves](image)

<table>
<thead>
<tr>
<th>D</th>
<th>D/d</th>
<th>T</th>
<th>σ</th>
<th>N_a</th>
</tr>
</thead>
<tbody>
<tr>
<td>1 500</td>
<td>32,6</td>
<td>31 400</td>
<td>21,6</td>
<td>42 700</td>
</tr>
<tr>
<td>1 860</td>
<td>40,4</td>
<td>33 350</td>
<td>23</td>
<td>70 500</td>
</tr>
<tr>
<td>1 860</td>
<td>40,4</td>
<td>36 300</td>
<td>25</td>
<td>61 300</td>
</tr>
</tbody>
</table>

The results are analysed as in reference 3.

Each cycle gives 2 flexions and 1 reversal so:

\[
\frac{2}{N_a} = \frac{2}{N_f} + \frac{1}{N_1}
\]

from which \(N_1\) is calculated.
The next table gives calculated data and the value of $N_i$ for COMMA I rope in the same conditions:

<table>
<thead>
<tr>
<th>$\sigma$</th>
<th>D/d</th>
<th>$\sqrt{\sigma}$ d/D</th>
<th>$N_a$</th>
<th>$N_f$</th>
<th>$N_i$</th>
<th>$N_i$ COMMA I</th>
</tr>
</thead>
<tbody>
<tr>
<td>21.6</td>
<td>32.6</td>
<td>0.814</td>
<td>42 700</td>
<td>73 460</td>
<td>50 980</td>
<td>23 190</td>
</tr>
<tr>
<td>23</td>
<td>40.4</td>
<td>0.754</td>
<td>70 500</td>
<td>99 800</td>
<td>121 840</td>
<td>29 380</td>
</tr>
<tr>
<td>25</td>
<td>40.4</td>
<td>0.787</td>
<td>61 300</td>
<td>96 970</td>
<td>83 300</td>
<td>26 520</td>
</tr>
</tbody>
</table>

There are not enough values for establishing the law $N_i (\frac{D}{d}, \sigma)$.

The effect of the reversal appears less important than on COMMA I rope.
3 - TENSION FATIGUE

3.1 - THE TESTS ACHIEVED

The tests were performed on a hydraulic machine AMSLER of 100 000 daN maximum load, with a frequency of 250 cycles per minute.

Three mean stresses $\sigma_m$ and four stress variations $\Delta \sigma$ were used, the results are gathered in table 2. It appears that the main parameter is the stress variation $\Delta \sigma$; the value of the mean stress has little influence.

By regression, for $\Delta \sigma > 24$ daN/mm$^2$

$$\ln N_t = 40,491 - 1,806 \sigma_m - 5,5 \Delta \sigma$$

with a standard deviation of 0.32.

3.2 - COMPARING WITH RESULTS OF YOUNG AND WALTON FROM BRITISH ROPEs

Results (ref. 6) are given for two different ropes:
- 38 mm  1 x 172 strand
- 51 mm  1 x 139 strand

with the Equivalent Load Range parameter:

$$Le = 100 \frac{\text{Load max} - \text{Load min}}{\text{Minimum breaking load} - \text{Load min}}$$

From these results regression gives:

- for 38 mm  1 x 172
  $$\ln N = 25,22124 - 3,48098 \ln Le$$

- for 51 mm  1 x 139
  $$\ln N = 34,31394 - 5,77274 \ln Le$$
The results and regression lines are plotted on figure 5.

The parameter $Le$ computed for the locked coil rope is given on table 2, the regression gives:

$$\ln N = 35,047 - 6.2616 \ln Le$$

results and regression line are plotted on figure 6.

The regression on all results for the three ropes gives the mean line:

$$\ln N = 30,2136 - 4.8761 \ln Le$$

which is plotted on figures 5 and 6.

The 51 mm $1 \times 139$ rope is the more resistant, followed by locked coil and 38 mm $1 \times 172$.

The numbers of cycles for $Le = 30$ are as follows:

<table>
<thead>
<tr>
<th>51 mm</th>
<th>$1 \times 139$</th>
<th>$2.37 \times 10^6$</th>
</tr>
</thead>
<tbody>
<tr>
<td>24 mm</td>
<td>locked coil</td>
<td>$9.36 \times 10^5$</td>
</tr>
<tr>
<td>38 mm</td>
<td>$1 \times 172$</td>
<td>$6.48 \times 10^5$</td>
</tr>
<tr>
<td>D</td>
<td>TENSION daN</td>
<td>(\sigma) daN/mm(^2)</td>
</tr>
<tr>
<td>----</td>
<td>-------------</td>
<td>-----------------</td>
</tr>
<tr>
<td>1.5</td>
<td>1960</td>
<td>5.0</td>
</tr>
<tr>
<td></td>
<td>6870</td>
<td>17.4</td>
</tr>
<tr>
<td></td>
<td>11770</td>
<td>29.8</td>
</tr>
<tr>
<td></td>
<td>17660</td>
<td>44.7</td>
</tr>
<tr>
<td></td>
<td>23540</td>
<td>59.6</td>
</tr>
<tr>
<td>WITHOUT END EFFECT</td>
<td>5890</td>
<td>14.0</td>
</tr>
<tr>
<td></td>
<td>11770</td>
<td>29.8</td>
</tr>
<tr>
<td></td>
<td>15700</td>
<td>39.7</td>
</tr>
<tr>
<td></td>
<td>19620</td>
<td>49.7</td>
</tr>
<tr>
<td></td>
<td>23540</td>
<td>59.6</td>
</tr>
<tr>
<td></td>
<td>27470</td>
<td>69.5</td>
</tr>
<tr>
<td>END EFFECT</td>
<td>11770</td>
<td>29.8</td>
</tr>
<tr>
<td></td>
<td>14700</td>
<td>37.2</td>
</tr>
<tr>
<td></td>
<td>17660</td>
<td>44.7</td>
</tr>
</tbody>
</table>

The number of broken wires before failure is very small in all trials.

**TABLE 1 - BEND OVEN SHEAVE FATIGUE OF A 24 mm LOCKED COIL WIRE ROPE**
<table>
<thead>
<tr>
<th>$\sigma_m$</th>
<th>$\Delta \sigma$</th>
<th>$L_e$</th>
<th>$N$</th>
</tr>
</thead>
<tbody>
<tr>
<td>29.8</td>
<td>24.8</td>
<td>18.9</td>
<td>No failure at $10^7$</td>
</tr>
<tr>
<td></td>
<td>34.7</td>
<td>25</td>
<td>$2.767 \times 10^6 - 2.855 \times 10^6$</td>
</tr>
<tr>
<td></td>
<td>42.2</td>
<td>30</td>
<td>$9.538 \times 10^5$</td>
</tr>
<tr>
<td>37.2</td>
<td>24.8</td>
<td>20</td>
<td>No failure at $10^7$</td>
</tr>
<tr>
<td></td>
<td>34.7</td>
<td>26.9</td>
<td>$2.067 \times 10^6 - 1.86 \times 10^6$</td>
</tr>
<tr>
<td></td>
<td>42.2</td>
<td>31.7</td>
<td>$4.3 \times 10^5 - 9 \times 10^5$</td>
</tr>
<tr>
<td>44.7</td>
<td>24.8</td>
<td>21.3</td>
<td>No failure at $10^7$</td>
</tr>
<tr>
<td></td>
<td>34.7</td>
<td>28.5</td>
<td>$7.54 \times 10^5 - 1.972 \times 10^6$</td>
</tr>
<tr>
<td></td>
<td>42.2</td>
<td>33.7</td>
<td>$8.15 \times 10^5 - 3.72 \times 10^5$</td>
</tr>
<tr>
<td></td>
<td>49.6</td>
<td>38.5</td>
<td>$1.62 \times 10^5$</td>
</tr>
</tbody>
</table>

**TABLE 2 - TENSION FATIGUE OF 24 mm LOCKED COIL ROPE**
Figure 2 - BEND OVER SHEAVE FATIGUE OF A 24 mm LOCKED COIL WIRE ROPE

D = 1.5 m  o
D = 2.5 m  x
Figure 3 - BEND OVER SHEAVE FATIGUE OF 24 mm LOCKED COIL WIRE ROPE

Sheave 1,5 m o

2,5 m x
Figure 5 - TENSION FATIGUE OF 38 mm AND 51 mm SPIRAL STRAND