The Calculation of Stress Distribution of Big Rope Sheaves

Thomas Kuczera

ThyssenKrupp Elevator Innovation GmbH, Research and Innovation Center Pliezhausen, Robert-Bosch-Str. 5, 72124 Pliezhausen, Germany, thomas.kuczera@thyssenkrupp.com

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Abstract The aim of this paper is to analyse the load conditions (pressure between rope and sheave) and the stress distribution of large sheave systems. Both an analytical model for calculation of contact forces between rope and sheave was developed as well as extensive theoretical and experimental tests were realized to determine the load and stress situation of big sheaves.

A practical numerical model based on a parameter assisted multi-body simulation model for the simulation of the dynamic run of a rope over a sheave was developed. The outcomes of this are the following findings.

By small wrap angles (smaller than 60°) the peaks of the line contact pressure go partially or completely together when the rope enters onto and runs of the sheave and this results in a noticeable higher contact force, which was unknown so far. Within the analysed rope constructions there are maximum forces of 6 times of the constant (so far considered for the dimensioning of sheaves) part of the line contact pressure by small wrap angles.

Also the rope forces and pressure load acting on sheaves due to acceleration forces during start-up process or rundown of conveyor systems, and the pressure due to a fleet angle between rope and sheave, were looked at with this multi-body-system. Finally, in the research study a method for the calculation of the load of sheaves was developed based on the finite-element-method for the calculation of the deformations and stresses of the sheave.

The verification of the developed calculation and simulation methods could be done successfully by a sheave of a rope way used in practice. Consequently, the results of this research study provide fundamental guidelines for the design of big sheaves in lifting applications.

1 STATE OF THE ART AND RESEARCH

Up to now, it was assumed that for strength calculation and dimensioning of sheaves (Fig. 1) the line contact pressure (normal force) between rope and sheave is constant [1]. But when the rope enters onto or runs off the sheave the line contact pressure increases because of the bending stiffness of the rope [2]. These peaks can reach up to 4 times of the average line contact pressure, which depends on rope construction, rope force and ratio of diameters between rope and sheave. Furthermore there are more influencing factors unconsidered as for example lateral forces because of fleet angles or dynamic forces because of start-up processes and rundowns of conveyor systems.

So far, due to low computing power, the strength calculation (deformation and mechanical stress) of sheaves could only be done by highly simplified models of the sheave and with the help of frame analysis programs. With this method a detailed model of the sheave e.g. containing welds and screw couplings is impossible. So the internal stress situation of sheaves could not be calculated completely up to now.

Figure 1 Sheaves of a ship lift

2 PRESSURE BETWEEN ROPE AND SHEAVE - ANALYTICAL MODEL

Both, an analytical model for calculation of contact forces between rope and sheave was developed, as well as extensive theoretical and experimental tests were realized to determine the load and stress situation of big sheaves.

The basis for calculation of constant part of the line contact pressure q was an existing analytical model (Eq. 1) with the rope force F and the radius of the sheave R_0 [3].

$$q = \frac{F}{R_0} \tag{1}$$

This calculation method was extended with the mass and the velocity of the rope (Fig. 2).

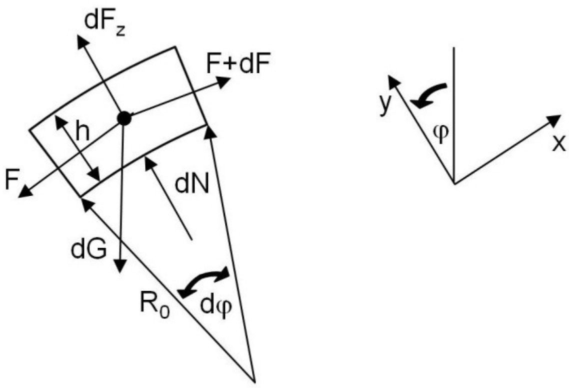

Figure 2 Forces on a rope element including mass and velocity

With the weight of a small element (Eq. 2) (A_B rope cross section and $R=R_0+h/2$)

$$dG = g \cdot dm = g \cdot \rho \cdot dV = g \cdot \rho \cdot A_B \cdot R \cdot d\varphi \tag{2}$$

and the centrifugal force (Eq. 3)

$$dF_z = \frac{{v_0}^2}{R} \cdot dm = \rho \cdot A_B \cdot {v_0}^2 \cdot d\varphi \tag{3}$$

the line contact pressure is calculated including mass and velocity of the rope (Eq. 4)

$$q = \frac{F - \rho \cdot A_B \cdot v_0^2}{R_0} \tag{4}$$

In the next step there is also friction F_R between rope and sheave considered (Fig. 3).

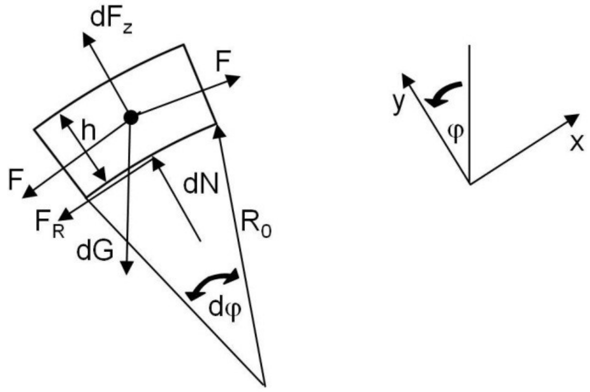

Figure 3 Forces on a rope element including mass, velocity and friction

The inequality of all forces provides the condition for adherence (Eq. 5) (μ coefficient of friction).

$$F \geq \frac{\rho \cdot A_B}{\mu} \cdot \left(g \cdot R \cdot (\sin\varphi - \mu \cdot \cos\varphi) + \mu \cdot v_0^2\right) \tag{5}$$

With this equitation it is possible to calculate the limit between adherence and sliding of the rope (Eq. 6).

$$\varphi_{\lim it} = \arccos\left(\frac{\alpha \cdot \mu \pm \sqrt{\mu^2 + 1 - \alpha^2}}{\mu^2 + 1}\right) \tag{6}$$

with

$$\alpha = \frac{\mu}{g \cdot R}\left(\frac{F}{\rho \cdot A_B} - v_0^2\right) \tag{7}$$

The existing calculation method of the line contact pressure between rope and sheave was extended with the mass and the velocity of the rope. So the line contact pressure depends now on the rope force, the bending radius, the mass of the rope and the velocity of the rope. Furthermore the new calculation method allows getting the limit angle between adherence and sliding of the rope with the sheave.

3 PRESSURE BETWEEN ROPE AND SHEAVE - MULTI-BODY-SIMULATION

Due to essential enormous processing power, it is currently not possible to simulate dynamically a completely detailed steal wire rope with all wires and strands by the finite-element-method, when the rope runs over the sheave. Therefore a practical numerical model based on a parameter assisted multi-body simulation model for the simulation of the dynamic run of a rope over a sheave was developed. This multi-body simulation model enables the calculation of contact forces between rope and sheave (Fig. 4).

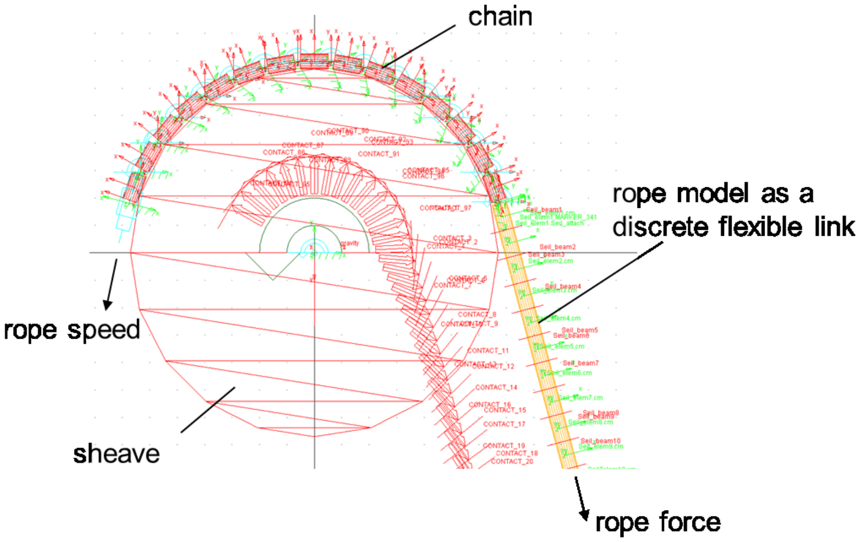

Figure 4 Multi-Body-Model of rope and sheave

Because of modelling the rope as a "discrete flexible link", the developed model of the rope enables simulation periods of few minutes. The discrete flexible link consists of fixed cylinder elements linked with beam-elements. The cylinder elements simulate both the external geometric form and the weight of the rope. The beam-elements represent the elastic part of the rope, which can be used for the assignment of Young´s modulus, bending resistance and damping coefficients of the rope.

In this model there is also a chain necessary, which consists of fixed cylinder elements linked with rotation joints. This chain moves the rope over the sheave. Because the starting position of the rope (the discrete flexible link) is only possible in a straight line.

Because it is an approximate model of the rope, it has to be calibrated with the help of results of experimental measurements. Therefore the results of measurement of the research study of Häberle [2] could be used and were prepared as the basics for the development of the multi-body-system. Furthermore the database was extended by own measurements on sheaves of a ship lift and a rope way used in practice. After the calibration of the analogous model, the model could be used for analyses which have not been possible so far. This resulted in the following findings.

3.1 Small wrap angles

By small wrap angles (smaller than 60°) the peaks of the line contact pressure go partially or completely together when the rope enters onto and runs off the sheave and this results in a noticeable higher contact force, which was unknown so far. With the developed numerical model for the first time it is possible to calculate these forces qualitatively and quantitatively. Within the analysed rope constructions there are maximum forces of 6 times of the constant (so far considered for the dimensioning of sheaves) part of the line contact pressure by small wrap angles (Fig. 5).

Therefore a calculation of the line contact pressure at small wrap angles by the given calculation method is no more acceptable.

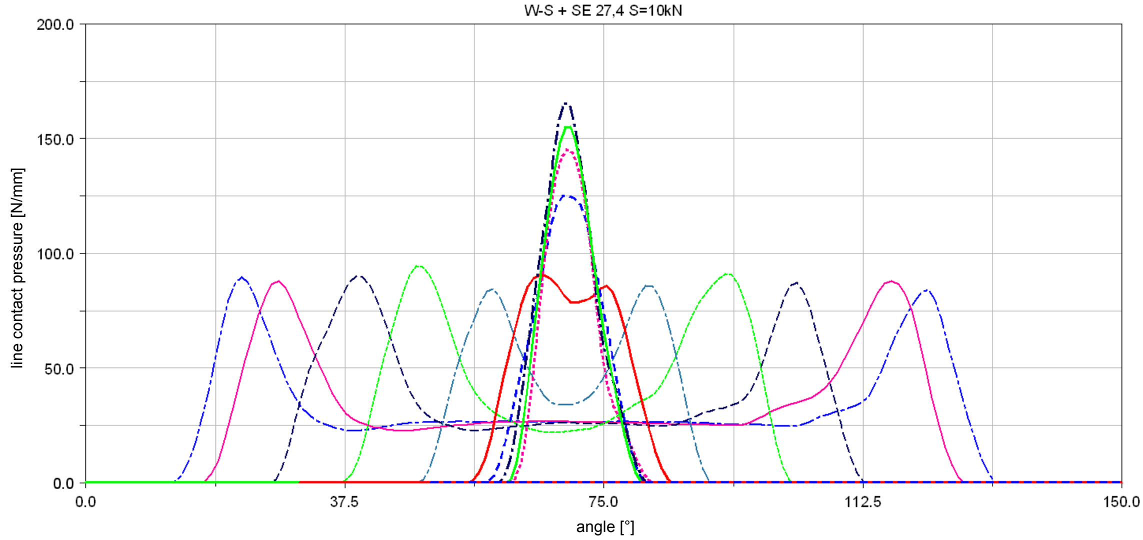

Figure 5 Influence of small wrap angles

3.2 Acceleration forces

At the start-up process or rundown of conveyor systems the rope force changes due to acceleration forces. This means that the contact forces between rope and sheave are variable and even higher during acceleration. For the first time these contact forces can be calculated by the help of the developed numerical model for a complete dynamic drive of the conveyor system with acceleration phase, phase with constant velocity and deceleration phase (Fig. 6). The rope forces (10, 30, 50, 70 and 100kN) are modelled as solid spheres with weight in this simulation to have the mass inertia represented. The results show that the forces between rope and sheave are directly proportional to the acceleration. Furthermore it is possible to analyse the effects of vibrations in longitudinal direction of the rope on the contact forces, because of the lurch at the acceleration. Furthermore this results in temporary higher contact forces.

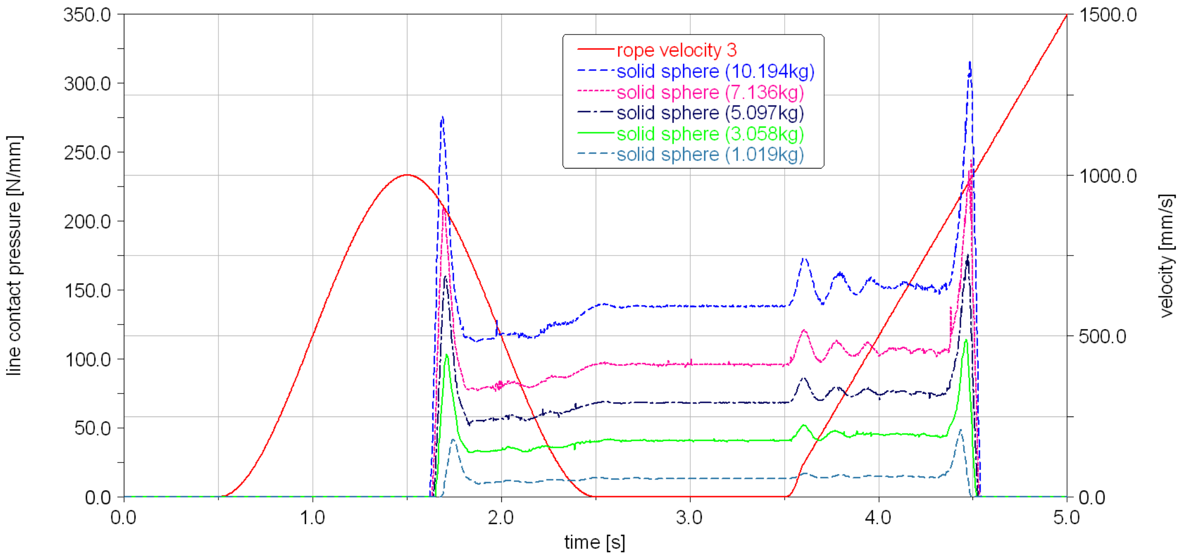

Figure 6 Influence of rope force during dynamic drive

3.3 Fleet angle

If there is a fleet angle between rope and sheave, which often cannot be avoided in wire rope drive systems, there are the same effects as if the rope entered onto or ran off the sheave. So because of the bending resistance of the rope, there are higher contact forces between rope and sheave, especially cross to the internal groove sidewall of the sheave. Up to now this could only be calculated with an extremely simplified analytic model. With the developed numerical model it is possible to calculate the amplitude and the course of the contact force for the first time (Fig. 7). The amplitude of the contact force is mainly influenced by the fleet angle and the rope force. Furthermore the geometry especially the angle of the internal groove sidewall has also an effect on the contact forces because of the fleet angle.

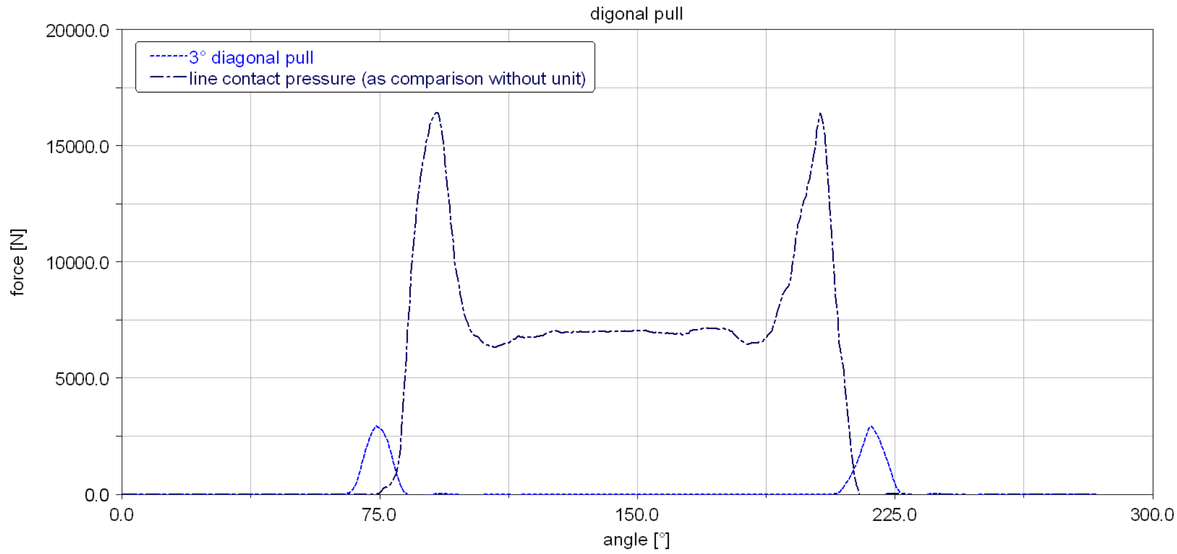

Figure 7 Influence of fleet angle

4 STRESS DISTRIBUTION OF BIG ROPE SHEAVES

Finally, in the research study a method for the calculation of the load of sheaves was developed, based on the finite-element-method for the calculation of the deformations and stresses of the sheave.

At sheaves with a spoke design or a similar spoke design, the critical position of the sheave is in general on the spokes (Fig. 8). These parts are alternately stressed with a combination of nominal tensile / compression stress and a superposed bending stress during rotation of the sheave. So the spokes are deformed in s-shape. The alternate bending stress of the spoke is relevant for the dynamic safety dimensioning of the sheave.

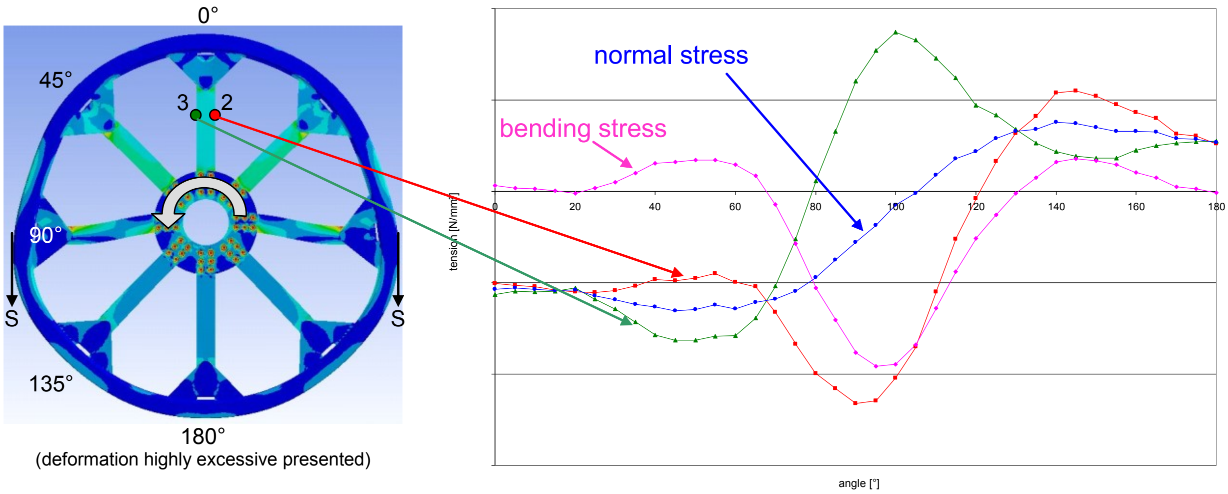

Figure 8 Stress distribution of loaded sheave

5 VERIFICATION

The verification of the developed calculation and simulation methods could be done by a sheave of a rope way used in practice (Fig. 9).

For the measurement of deformation of the sheave, strain gauges were positioned at the critical point (with the highest tension) on the sheave. All measurement equipment was located on the sheave, so that it was possible to measure the deformations during rotation of the sheave. The measurement included different load situations of the rope way system and also emergency stops.

Figure 9 Sheave of a rope way

These experimental measurements of this rope way sheave were successfully compared with the calculated results for that sheave (Fig. 10).

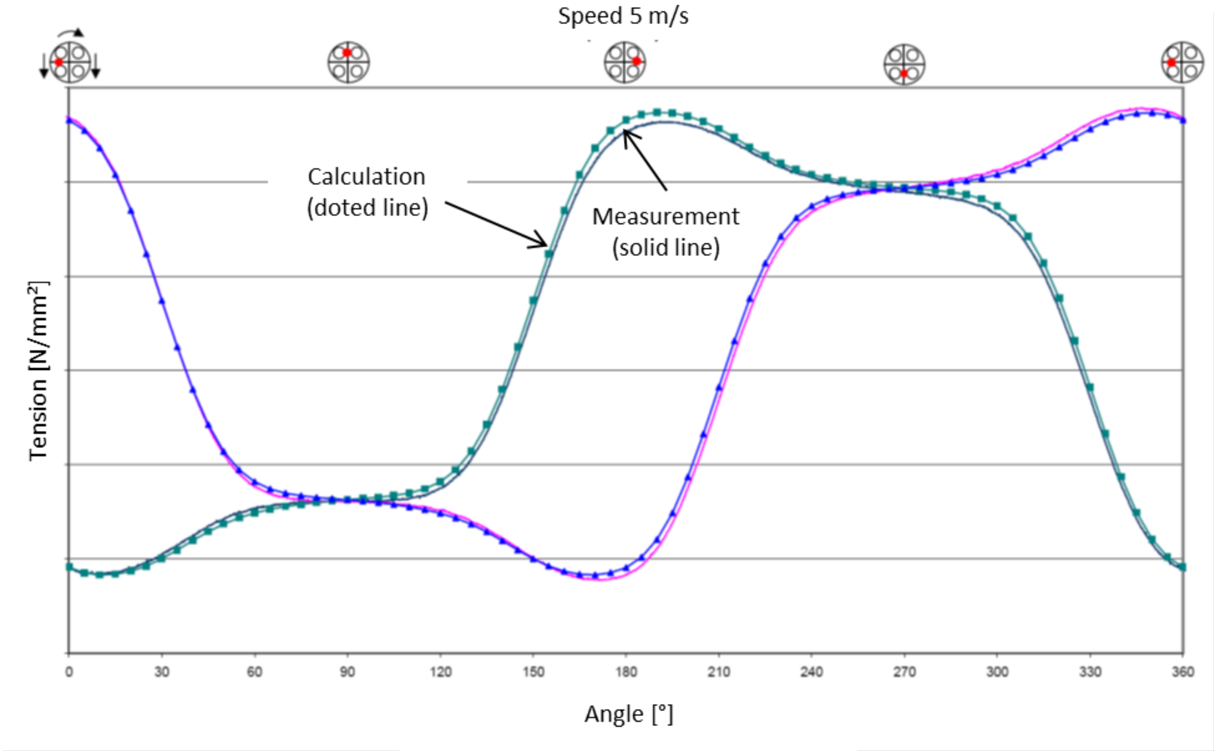

Figure 10 Comparison between calculation and measurement

6 SUMMARY

Within this research study the pressure between rope and sheave was analysed related to small wrap angles, dynamic forces and fleet angles. Therefore both an analytical model and a multi-body-model were generated to calculate these forces. With these results it was possible to calculate the stress distribution of sheaves with the help of finite-element-analyses. The experimental verification of the models has successfully been completed using a practical sheave – rope installation.

With this research study [4] an input for the systematically and safety dimensioning of big sheaves was contributed.

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BIOGRAPHICAL DETAILS

Name: Thomas Kuczera

Date of birth: 5.8.1981

Place of birth: Filderstadt-Bonlanden (Germany)

Family status: married (1 child)

Nationality: German

2002 – 2006 – Study of mechanical handling at University of Stuttgart

2007 – 2012 – Research assistant and PhD student at University of Stuttgart,
Institute for mechanical and handling and logistics (IFT).
Topic: *Calculation of the stress situation of big rope sheaves*

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