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DESIGN CRITERIA FOR MULTILAYER WOUND WINCH DRUMS FOLLOWING LIGHTWEIGHT DESIGN PRINCIPLES

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1. Introduction

Industries efforts to plan ever heavier and more voluminous components that are assembled on the ground before lifting forces crane manufacturers to guarantee larger hook loads and reaches of their designs. This implies larger rope loads and bigger rope storage capacity as well as a reduction of available space for the design and dimensioning of the drums. Due to this the constant development of calculation methods for a design appropriate to the loads is of great interest to crane and winch manufacturers, especially as the design criteria of the present norms seem arguable.

2. Problem definition and calculation methods

Today's calculation methods to lay out multi-layer drums (e.g. *[Dietz 1971]*, *[Mupende 2001], [Henschel 2000]*) are based on rotationally symmetric load assumptions. The loads on a drum are principally the wrapping pressure p(x) on the drum jacket caused by the rope packet and the line loads $F_i(r)$ and $F_j(r)$ caused by the rope layers on the side discs (fig.1).

Load model according to Dietz *[Dietz 1971]*:
- Determining the jacket loads using the shell theory.
- Determining the side disc loads using the plate theory.
- Influence of the layer loads when multi-layered due to the elasticity of the ropes.

Statically undefined problem with statically undetermined primary structure.

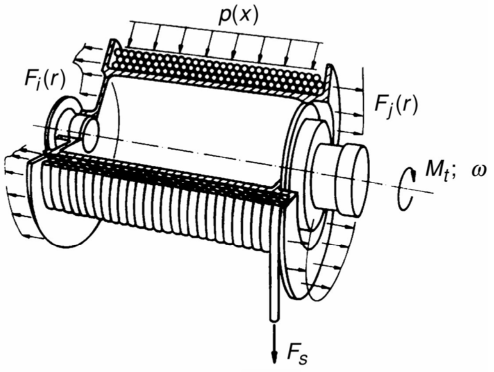

Figure 1. Rotationally symmetrical pressure- and lineload distribution as basis for the stress determination of multiply wound drums

After the first development of single layer wound drums by Ernst, in 1970 Dietz developed a calculation method for multiple layer wound drums, in which he took into account the transverse deformation of the rope and its unloading due to jacket deformation. The build up of pressure on the jacket results from a load-deformation-cupping, in which loading and relief mechanisms take place concurrently. The mechanical model in fig.2 shows how the underlying layer of rope is deformed into a smaller diameter by winding a new layer of rope upon it - this leads to a reduction of the pretensioning of the ropes in the lower layer. Fundamental influence parameters are the stiffness of the rope packets and the jacket deformation according to the model of a cylindrical shell. This results in a non linear gradient of the jacket stress relative to the number of layers.

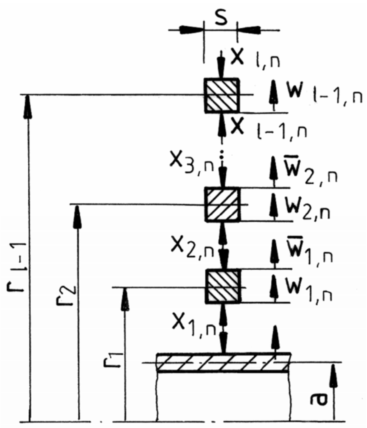

Figure 2. Abstraction of the rope package

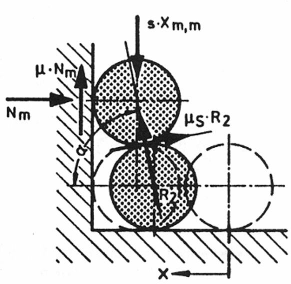

Figure 3. Climbing of the last winding

Regarding the loads on the side discs an approach using the theory of the 'climbing of the last winding' succeeds (Fig. 3). The basis of this approach rests in the fact that the last winding of a layer is forced into the converging gap between the disc and the rope under full rope load and must climb up to the diameter of the next layer. The resulting axial component stresses the drum side disc.

Henschel *[Henschel 2000]* amends this theory with the introduction of a variable transverse elasticity modulus according to the number of layers using linear and quadratic functions. These values are dependant on the distribution of the rope and can only be determined in experiments. Figure 4 shows the Institut für Maschinenwesen's test rig for measuring rope characteristics. The results (Fig.5) show, that the ratio of longitudinal to transverse stiffness, which is responsible for the unloading in fig.2, largely depends on the type of rope and the current rope load.

A further novelty in this work is the inclusion of plastic deformation, which leads to a clear theoretical increase of the load carrying capability and extends the limits of loading upwards. In the area of the jacket shell, which normally carries the largest stress amplitudes, this does not mean failure due to the prevalent compressive loads - in practice drums are 'trained' to withstand higher loads by so called 'burn in tests' in the overelastic regions of stress.

This training causes irreversible deformation, the working loads meet a geometrically changed shape of structure. The strength limit of the jacket shell is increased so far, that the design of drums must take into account completely different failure causes explained in Fig.6.

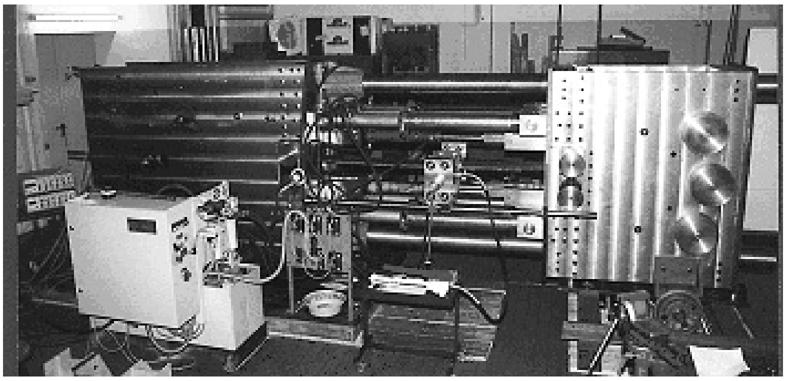

Figure 4. Test stand for measuring the longitudinal and transverse stiffness of wire ropes in different mounting positions in the drum (rope diameter from 7 to 32 mm, strain in longitudinal direction up to 1200 kN, pressure in transverse direction up to 500 kN)

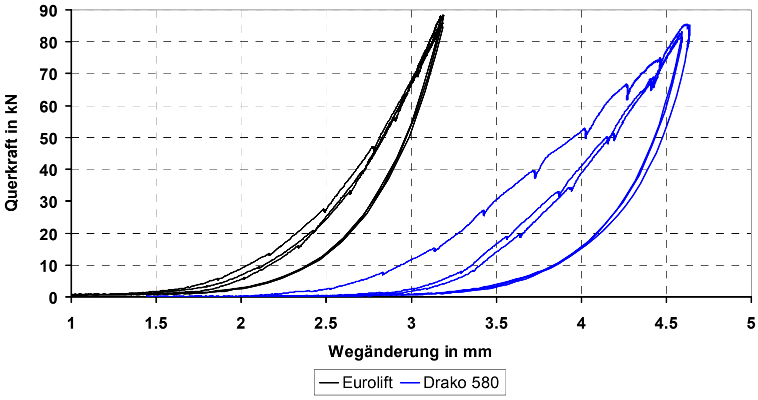

Figure 5. Hysterisis for multiple loading [KL=0,2, 4 ropes; Ø 14 mm] in transverse direction with longitudinally pre-stressed ropes as basis for the relief calculation|3|

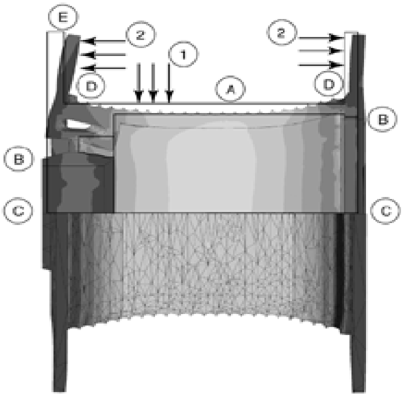

Figure 6. Winch drum, stress and deformation beahviour according to an FE calculation. Problem areas: A drum jacket; B bearing area as a result of radial distortion and misalignment; C Mounting restriction in axial direction; D Notch effects in the area of the side disc joints; E Side disc deformation with multiple windings

3. Non rotation symmetric loading and deformation

The assumption of symmetrical loading does not exactly represent the observations in practice and the experimentally proven circumferential load distribution of the jacket an side discs. This is particularly the fact for drums with drum grooves according to the principle of LeBus. The grooving principle of LeBus is characterised by four circumference sections. These are two parallel sections (PB) and two crossover sections (KB), in which the rope is diverted by half a pitch in axial direction (fig.7). The angular size of the parallel and the crossover area and the grooving pitch vary from design to design. They depend on the designated use, the rope diameter, the rope tolerances, the drum dimensions and the rope bending stiffness. Figure 7 points out the changing contact states at three chosen sections of the circumference in idealized form.

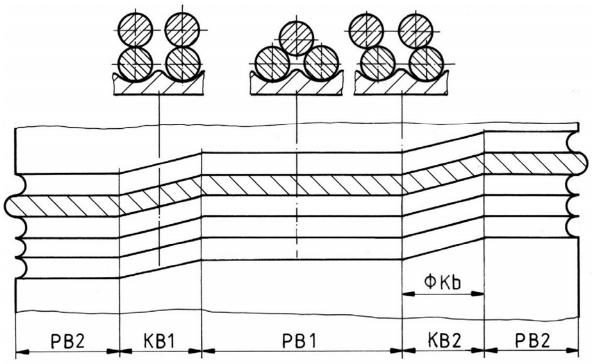

Figure 7. Arrangement of the rope windings in the parallel and crossover sections

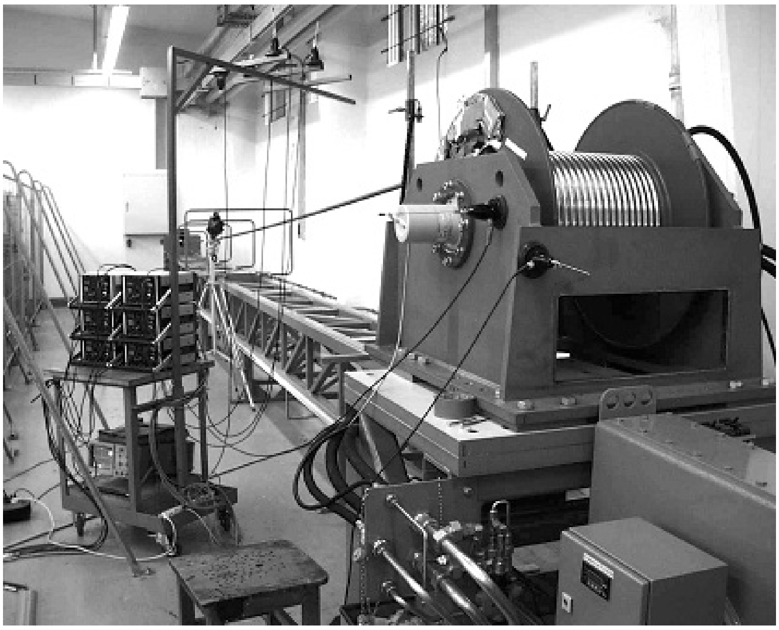

Figure 8. Winchdrum test stand: rope diameters from 12 to 36 mm, rope load of up to 300kN, rope velocity of up to 40 m/min, drum diameter of up to 800mm and 1000mm length

The institutes universal winch drum test stand (fig. 8) was used for the winding tests, the results of which are shown in fig. 9. The measurement points were placed on axially offset sections, each centrally placed in the grooves. The curve progression of one measurement point are shown as the tangential stress with a constant rope load Fs and increasing layer number. Against expectations of an almost even stress progression over the circumference of the jacket, distinct differences in the tangential stress values arise from the second layer upwards, that increase with advancing winding. When fully wound the minimal and maximal tangential stresses differ by around 45%.

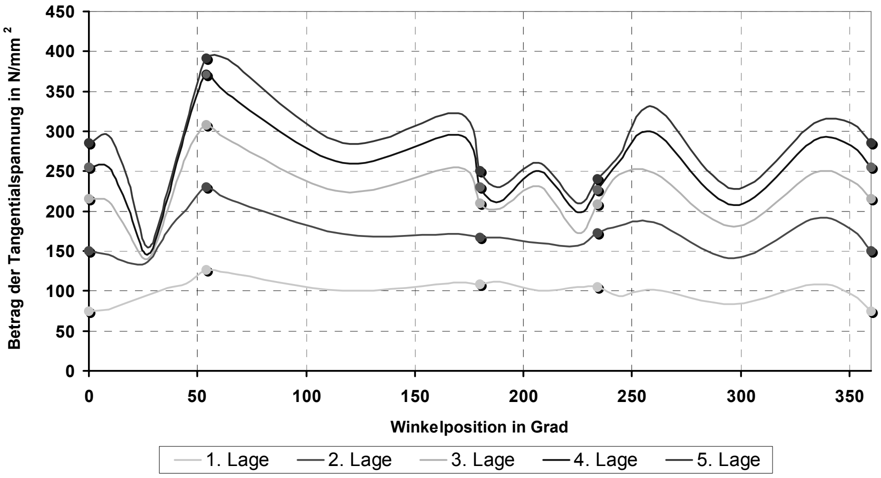

Figure 9. Non rotationaly symmetrical tangential stress distribution in the drum jacket on the inner diameter of the drum in one section layer

Within the scope of a research project *[Otto 2003]* the interrelationship between the experimentally determined stress differences and a non-rotationally symmetrical pressure loading $p(x,\Phi)$ of the jacket by the rope packet was looked into: The winding of the fist layer took place with a constant winding radius ($r_{pb,1}=r_{kb,1}$) over the whole circumference, in the crossover sections as well as in the parallel sections. From the second layer onwards the winding radius changes over the drum circumference of the drum as a result of rolling over of the lower windings within the crossover section. A winding radius difference Δr_i is created between the groove sections. This change of the winding radius $r_l(\Phi)$ causes an inversely proportional change of the radial winding pressure within each layer. With the foregone knowledge one can observe that the winding pressure is smaller in the crossover section than in the parallel section due to the larger winding radius.

The drum is not loaded in a rotationally symmetrical way due to this and experiences bending stresses and bending deformations in relation to the circumference angle. From this the optimum region of angles for the parallel and crossover sections, in which the minimal bending stresses occur, can be deducted.

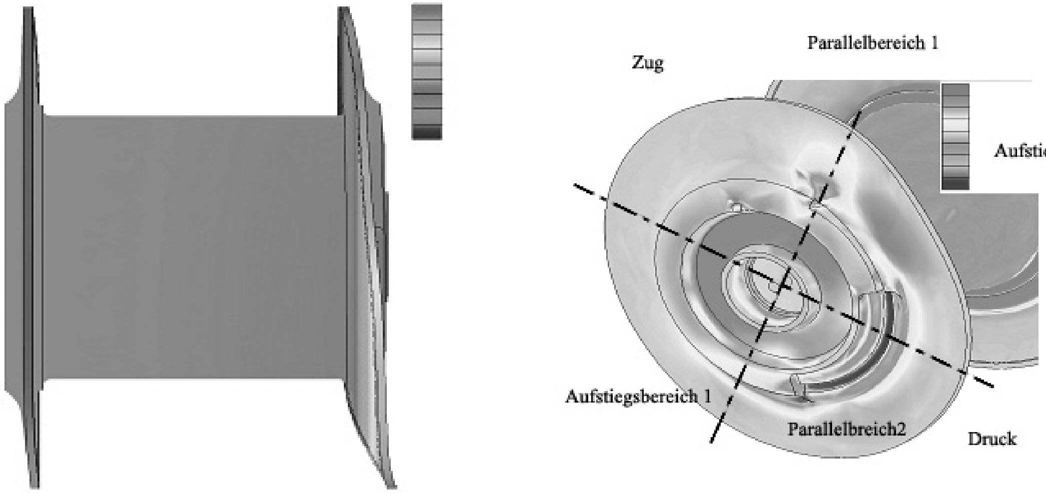

Figure 10. Stress and deformation of the side discs due to the non rotationaly symmetrical loads caused by the rope paket

As a result of the pressure differences axial displacement differences occur between the parallel and crossover sections, which cause additional axial forces in the junction area. These axial forces must be taken into account when laying out the junction geometry or the junction connections (e.g. welds, screws). Also the stress on the side discs is greatly influenced by the asymmetrical loading, that is a function of the circumferential angle after the principle of the "increasing winding", this is shown in figure 10 by means of an FE calculation: The side disc buckles unevenly and this can lead to failures due to the deflection as well as due to stress maxima in the connection to the jacket.

4. Summary

The stress of a winch drum wound with multiple layer of rope is governed by the elastic interaction of rope packet and drum structure. The assumption of a rotationally symmetrical load leads, in conjunction with the theory of the layer unloading and the stressing of the side discs, to a model, with the help of which winch drums with multiple layers of windings were developed further in the direction of light weight structures. Making use of part plastic deformation deformation in the drum jacket also increases the load capacity, but new stress limits have to be considered, which rest on the deformation of the drum jacket and the side discs.

Especially drums with LeBus winding show stresses and distortion that are not rotationally symmetrical. A theory is introduced that also includes these in the design of drums.

Reference

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