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Impact of winding drum shell ribbing of a hoisting machine on its strength and manufacture costs

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Leszek Kowal [1a], Tomasz Sinka [2]

[1] MWM Elektro Sp. z o.o., Armii Krajowej 24, 32-540 Trzebinia, Poland

[2] KOMAG Institute of Mining Technology, Pszczyńska 37, 44-100 Gliwice, Poland

[a] e-mail: l.kowal@mwm.com.pl

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Abstract:

A strength analysis of winding drums of hoisting machines, with particular attention paid to an impact of using circumferential ribs on stresses in the drum shell, is presented in the article. The winding drum, used as rope shafting and a load–carrying element of a hoisting conveyance, hung on the rope, is a widespread, historical solution applied successfully even nowadays not only in the case of mine shaft winders but also in many other applications in the industry. Hoisting machines with a rope carrier of this type are used in single – drum and double – drum winders both in shallow shafts as well as in very deep shafts, even up to 3000 m. In the Polish mines their application has been limited significantly in mine shaft winders due to an implementation of more compact hoisting machines with a frictional transmission of motion from the rope shafting onto the rope (Koepe type systems). However, drum machines still find an application, e.g. smaller machines in auxiliary shafts, as machines for shaft inspections and repairs as well as bigger single–drum machines used for shaft–sinking a double–drum machines installed in shallow shafts operating on a few levels.

Streszczenie:

W artykule przedstawiono analizę wytrzymałości bębnów nawojowych maszyn wyciągowych ze szczególnym uwzględnieniem wpływu stosowania żeber obwodowych na naprężenia w płaszczu bębna. Bęben nawojowy stosowany jako pędnia i element nośny naczynia wyciągowego zawieszonego na linie jest rozpowszechnionym i historycznym rozwiązaniem sprawdzającym się z powodzeniem i dzisiaj nie tylko w górniczych wyciągach szybowych, ale w wielu innych zastosowaniach w przemyśle. Maszyny wyciągowe z tego typu nośnikiem liny są stosowane w maszynach wyciągowych jedno i dwubębnowych w szybach płytkich jak i bardzo głębokich, nawet do 3000 m. W polskich kopalniach ich stosowanie zostało znacznie ograniczone w górniczych wyciągach szybowych na rzecz bardziej zwartych maszyn wyciągowych z ciernym przeniesieniem ruchu z linopędni na linę (systemy typu Koepe). Maszyny bębnowe jednak nadal znajdują swoje zastosowanie np.: mniejsze maszyny w szybach pomocniczych jako maszyny do prowadzenia rewizji i naprawy szybów, jak również większe maszyny jednobębnowe stosowane przy głębieniu szybów lub maszyny dwubębnowe zabudowywane w płytkich szybach obsługujących kilka poziomów.

1. Introduction

Hoisting machines of mine shaft winders, due to the way of transmitting the rope shafting torque to the load carrying rope, on which the hoisting conveyance is hung, can be divided into winding drums (Fig.1) and those with frictional contact of the Koepe type (Fig.2).

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Fig. 1. Double-drum hoisting machine Kinga Shaft

Fig. 2. Two-rope hoisting machine of Koepe type

Single - or double-drum hoisting machines as well as double-drum ones with a possibility of hanging the conveyance on two ropes (Blair system) are used in shallow shafts and very deep ones exceeding 3000 m (Blair type machines). The winding-drum system has been used in the underground mining industry for hundreds of years (Fig.3, Fig. 4). Drum winches have very many applications in various branches of industry. Double - drum hoisting machines with re-set drums enable an adaptation of conveyances location for servicing a few mining levels. Their disadvantage consists in a relatively big weight in relation the weight of the transported run-of-mine.

Fig. 3. Horse gear Wieliczka Salt Mine [1]

Fig. 4. Steam double-drum hoisting machine Museum of Bochnia Salt Mine [2]

Winding drums of hoisting machines had different design forms over the years. Their construction was often connected with the technology level related to drives of machines. Apart from typical cylindrical drums (Fig. 5) other design solutions also appeared e.g. cylindrical-conical (Fig. 6).

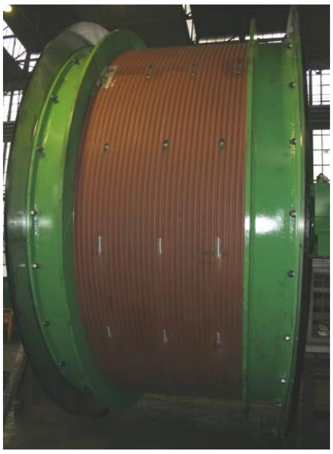
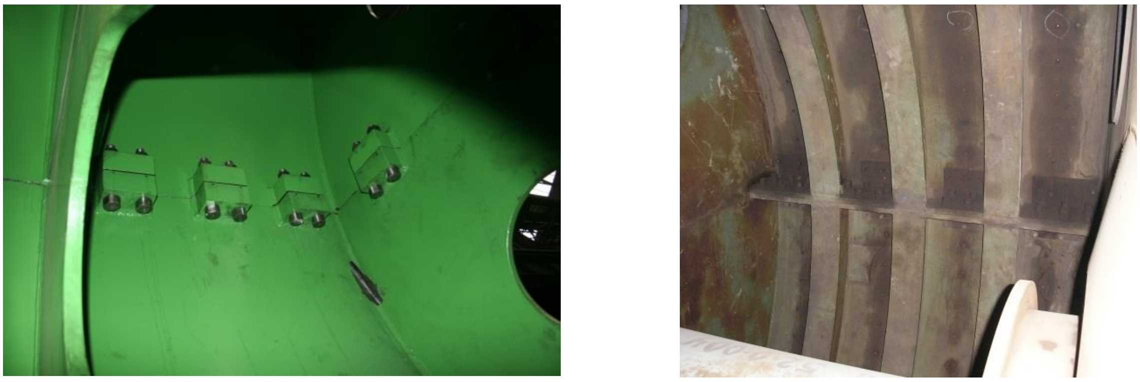


Fig. 5. Winding drum with a steel, grooved lining installed on the drum shell

Fig. 6. Cylindrical-conical winding drum with direct grooving on the drum shell

Winding drums are adapted to a single-layer or a multi-layer winding of rope. In the mines all over the world it is possible to find a series of different types of winding drums, produced several dozen years ago and operated up till the present time. Hoisting machines with winding drums have been and still are designed at the KOMAG Institute of Mining Technology.

One of design problems, related to the winding drum shell strength of the hoisting machines, is addressed in the article. It concerns an impact of circumferential ribs on the drum shell strength (Fig. 7) in the aspect of manufacture costs.



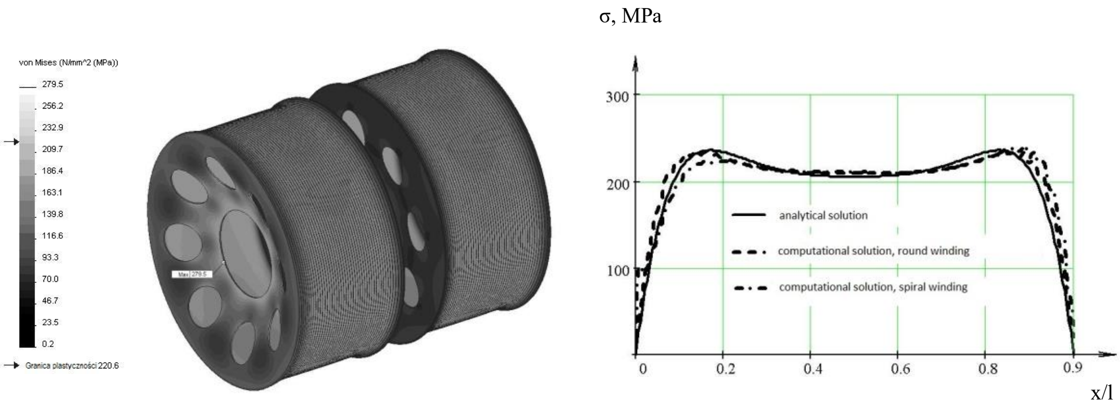
a) shell without ribs b) shell with circumferential ribs

Fig. 7. Interior of winding drum

An impact of ribbing locations and type of drum shells on the strength, in connection with manufacture costs, is discussed in this article. The discussion is concentrated on a technical justification of advantages and disadvantages of constructing winding drums shells without any ribs and with circumferential ribs.

2. Analytical model

Analytical methods (Fig. 8) [5,6,8] used to be applied for an assessment of the strength of winding drums during their designing process, but at present numerical methods, based on the Finite Element Method (Fig. 9) [3,4,7,8,9] are used.



a) FEM analysis results b) Results of analytical calculations

Fig. 8. Strength calculations of winding drum of the hoisting machine [8]

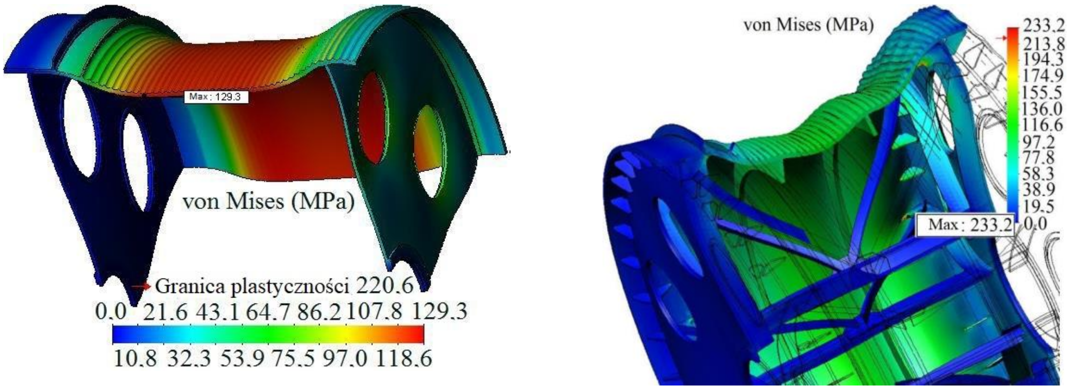
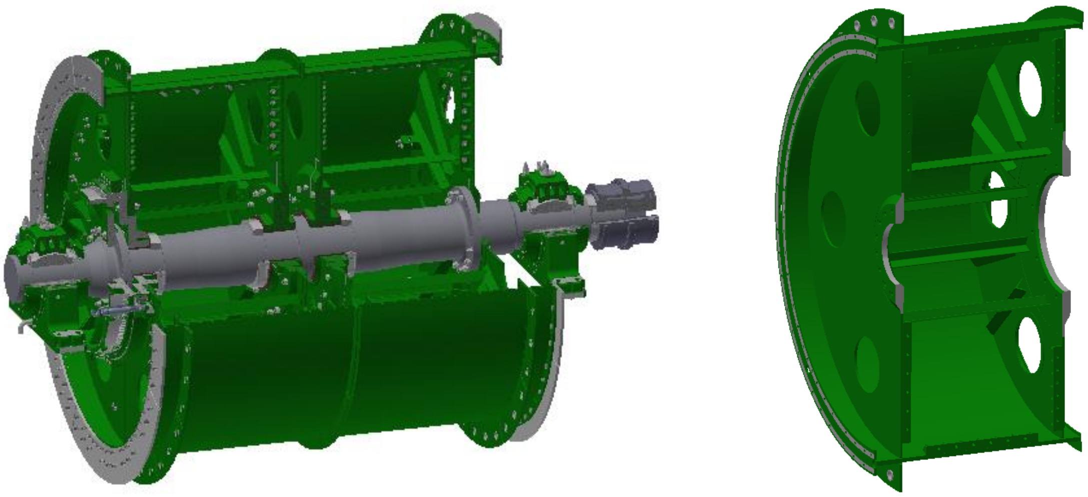


Fig. 9. Results of numerical analyses of different construction winding drums [9]

Applying the finite element methods, an analysis of the winding drum of the rope winding diameter – 7000 mm and the winding zone width – 3200 mm was constructed. The drum is adapted for a collaboration with the hoisting rope of 63mm diameter, wound in one layer. The suggested hoisting depth of the BB - 7000 machine (Fig. 10), with the drum under analysis, is about 820 m and the maximal force in the rope, loading the drum, accepted for calculations, is 580 kN.



a) main shaft assembly b) half of the winding drum

Fig. 10. Elements of BB- 7000 hoisting machine [11]

The main load of the winding drum comes from the force occurring in the hoisting rope wound onto it. Winding the rope on the drum causes a generation of an axial force, directed perpendicularly to the plane of the drum shell and expressed as the pressure exerted on the shell, determined according to the relationship (1) [6]:

$$p = \frac{S}{R \cdot s \cdot d} \tag{1}$$

where:

p – pressure exerted on the drum shell, Pa,
S – force in the rope, N,
R – radius of the drum shell, m,
s – number of layers of the wound rope,
d – rope diameter, m.

Besides the components of force in the rope, tangential to the drum shell (Fig. 11), the component of force perpendicular to the Ssr drum axis, causing its torsion and the component of force parallel to the Spr drum causing its lateral deflection, affect the drum.

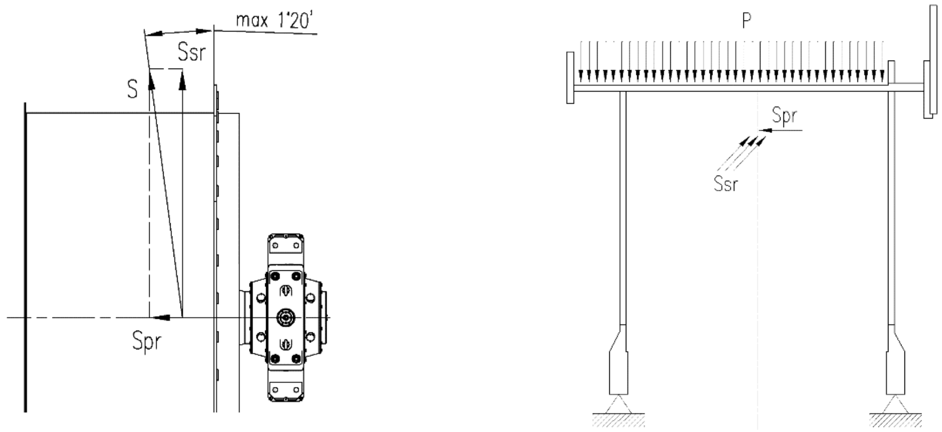


Fig. 11. Schematic diagram of the drum loading

In relation to accepted assumptions, the maximal value of the pressure exerted on the shell is about 2.5 MPa and it decreases when the rope is being wound on the drum. The weight of one-meter hoisting rope of Ø 63mm is 19 kg which causes that after winding 850 m, the force in the rope will reduce to about 420 kN, which causes a reduction of the pressure, exerted to the shell, to the value of about 1.7 MPa. To reduce the number of calculations variants, the most disadvantageous case was assumed, in which the whole drum shell is loaded with the pressure of 2.5 MPa. Strength calculations were conducted with use of the finite element method. Block elements were used for a construction of the model. The general FEM model of the drum is shown in Fig. 12.

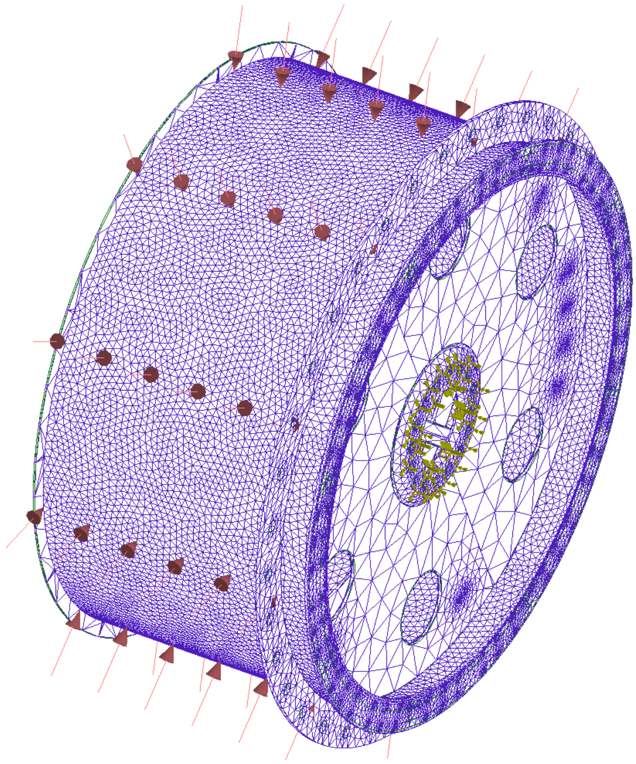
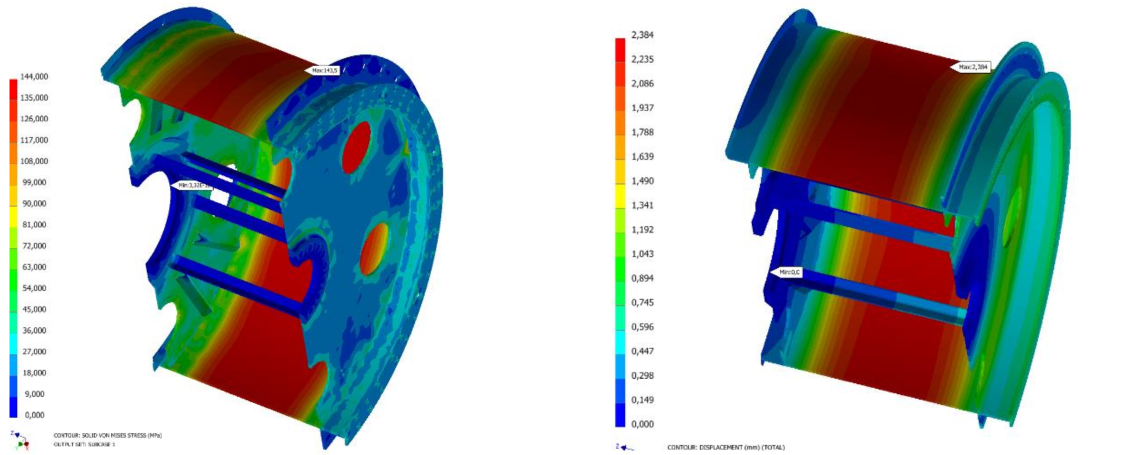


Fig. 12. Analytical model of the BB – 7000 drum

3. Result of calculations

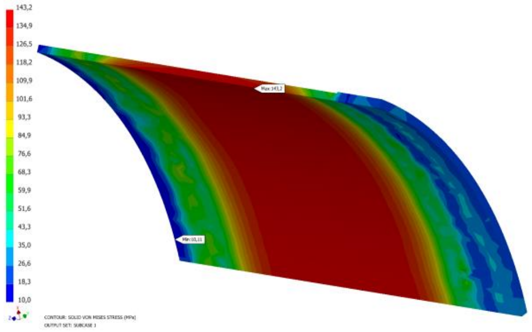
In Fig.13 the results of the drum strength calculations are presented in a form of maps of stresses and deformations. The maximal stress value in the drum shell was about 143 MPa and the value of deformations – about 2.4 mm.



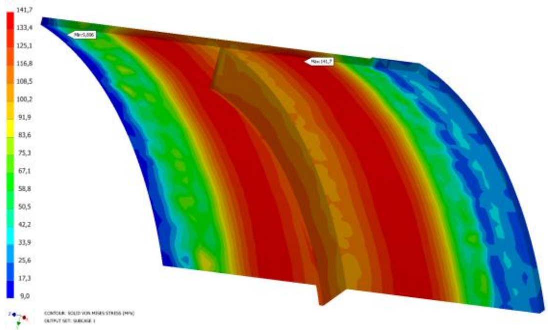
a) a map of drum reduced stresses [MPa] b) a map of drum global deformations [mm]

Fig. 13. Results of calculations

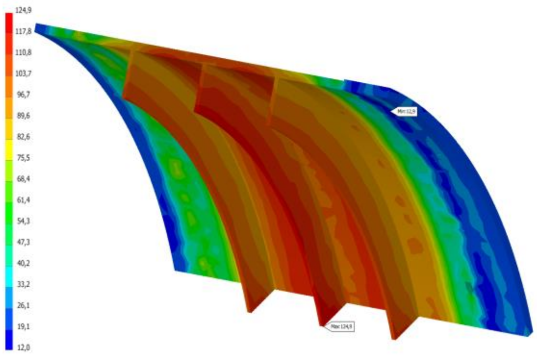
Conducting the drum strength calculations in the case of introducing circumferential ribs was the following step of analyses. It was assumed that the circumferential ribs, rings of 350 mm width and 40 mm thickness, should be analysed. Three drum variants: with one, three and five ribs, installed under the shell, were subject to an analysis. In Fig. 14 the results of calculations are presented in a form of maps of reduced stresses in the drum shell sector.



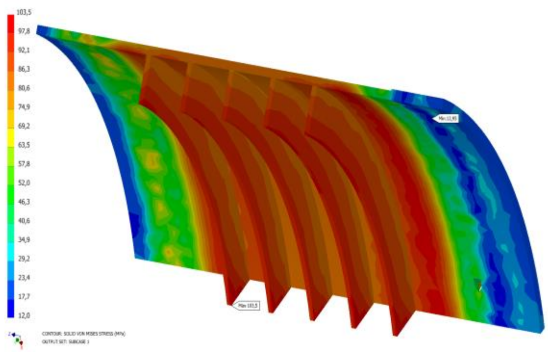
a) without ribs



b) one rib



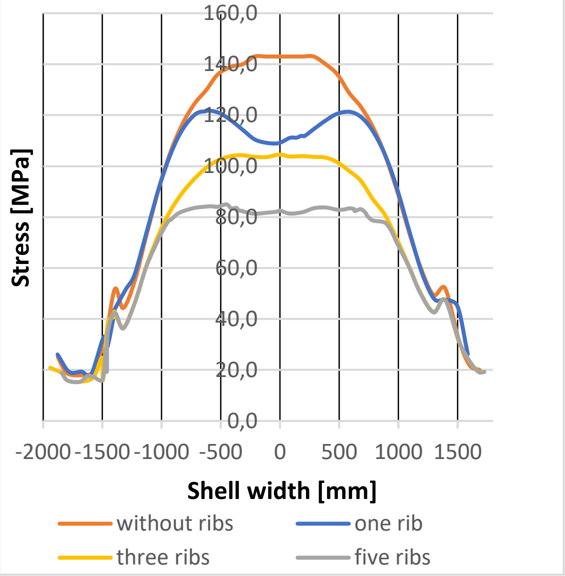
c) three ribs



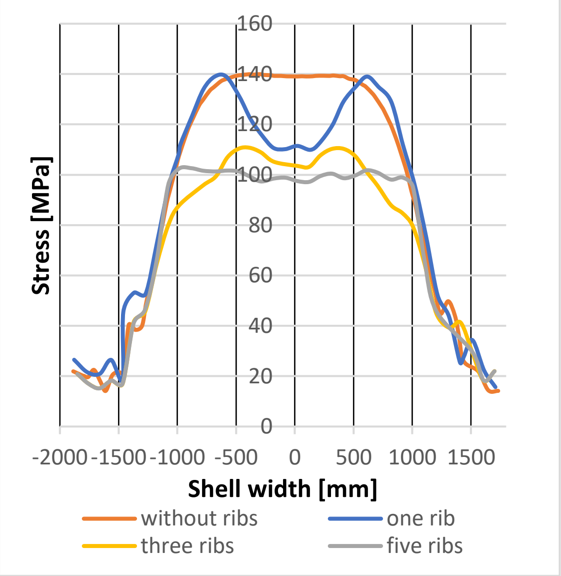
c) five ribs

Fig. 14. Reduced stresses in the drum shell and circumferential ribs

To compare the obtained results of calculations, in Fig. 15 the results are listed in a form of graphs of the stresses distribution along the shell width in relation to the number of applied circumferential ribs. An impact of the ribs, causing a reduction of stresses in relation to a smooth shell (without ribs) can be seen on the graphs. An implementation of three ribs caused a reduction of stresses in the drum shell of about 13%, but an introduction of five ribs – a reduction of stresses of about 28% in comparison with the smooth shell. An application of one rib reduced the stresses in the central part of the shell of about 16%, however it had no impact on the rest of the shell, where the stresses were similar as in the case of the shell without ribs.



a) shell external side



b) shell internal side

Fig. 15. Stresses in the drum shell in relation to a various number of circumferential ribs

An introduction of circumferential ribs caused a reduction of stresses in the drum shell at the cost of the stresses in the ribs themselves. In Fig. 16 maximal stresses in the ribs, in the case of the analysed variants with a different number of ribs, are presented. The obtained results indicate that the biggest stresses in the ribs occur for the variant with one rib, i.e. 127.6 MPa and the smallest ones when five ribs are applied, i.e. 103.5 MPa. The biggest difference of stresses between the rib and the shell occurs in relation to the variant with one rib and it reaches about 14 MPa, whereas in the other cases the stresses in the shell and in the ribs are comparable.

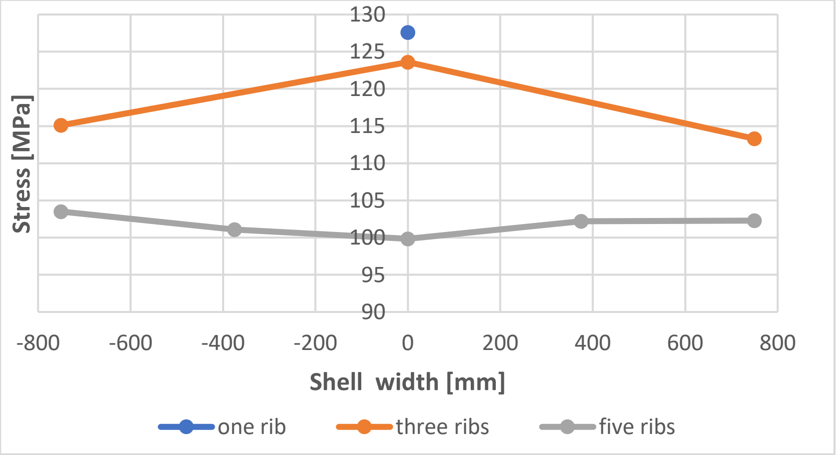


Fig. 16. Maximal stresses in ribs

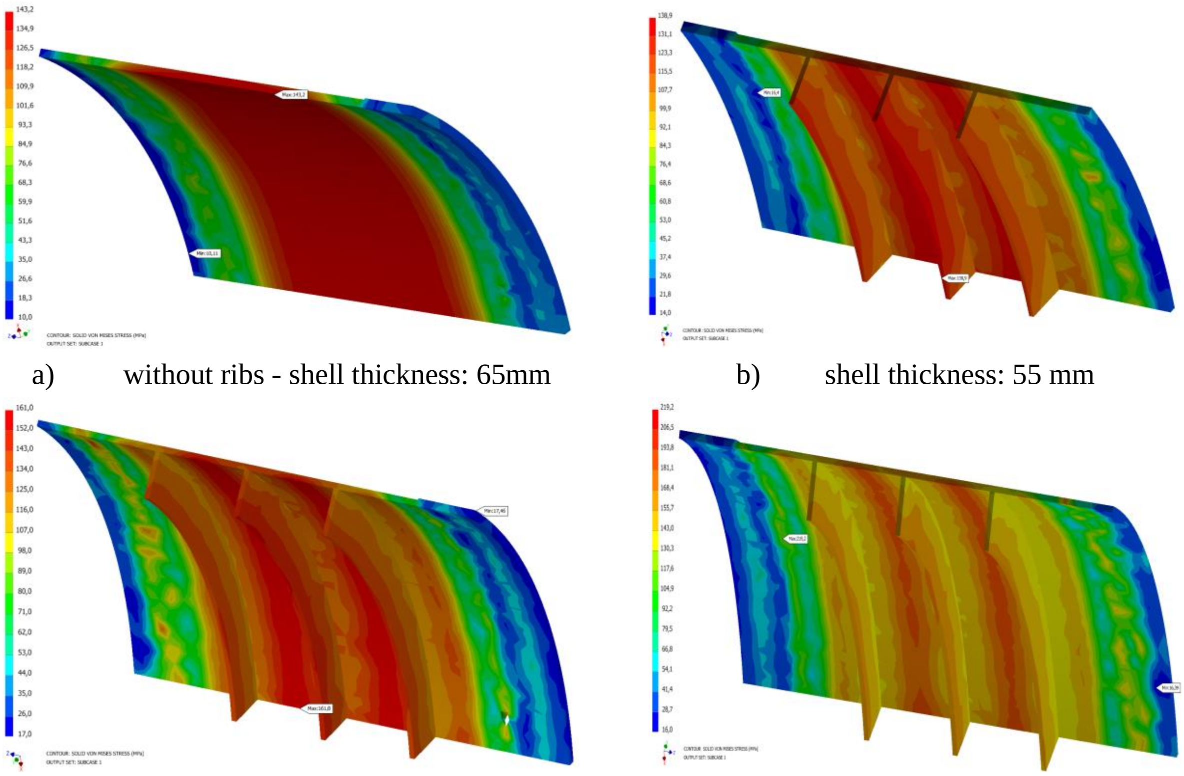
A comparison of the values of stresses, occurring in the shell of the constant thickness and in the drum ribs for a different number of ribs, is shown in Table 1.

Table 1. Values of maximal stresses in the shell of 65 mm thickness equipped with ribs in relation to the drum with a different number of ribs.

Number of ribs	0	1	3	5
Shell [MPa]	143.2	141.7	116.3	103.5
Rib [MPa]	0	127.6	124.9	103.5

An introduction of the following ribs gives a measurable effect in a form of lower and lower values of stress in the shell and in the ribs, however it generates additional costs of the drum manufacture. One of the reasons of introducing circumferential ribs includes technological limitations connected with a necessity of coiling very thick sheets or also due to the overall dimensions of the drum shell and its weight. In the other cases ribbing can be thereby unjustified economically.

In the following analyses an attempt was undertaken to show how an introduction of circumferential ribs would enable to reduce the drum shell thickness, at keeping the stresses on the level as in the case of the drum with the shell without ribs. This analysis was conducted at an assumption of introducing three circumferential ribs of geometry and parameters of the drum shell loading as before. The thicknesses of shells or calculations were as follows: 32.5 mm, 45 mm and 55 mm. In Fig. 17 the results of calculations are presented in a form of maps of reduced stresses.



a) without ribs - shell thickness: 65mm

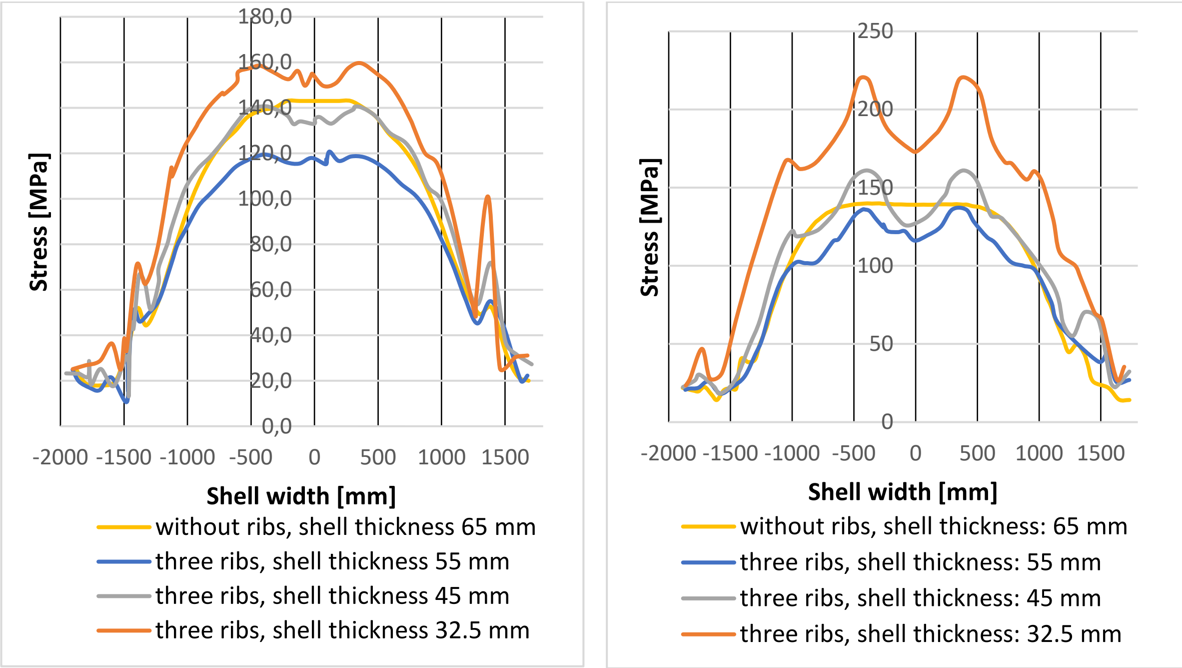
b) shell thickness: 55 mm

c) shell thickness: 45mm

d) shell thickness: 32.5 mm

Fig. 17. Reduced stresses in relation to different thicknesses of shells

The obtained results of stresses values in the shells of different thicknesses, equipped with three ribs, were compared with the stresses in the shell without ribs, of the thickness: 65 mm (Fig. 18).



a) shell external side b) shell internal side

Fig. 18. Stresses in drum shell with three circumferential ribs in relation to different thicknesses of shells

The obtained results indicated that a reduction of the shell thickness by 50% (i.e. 32.5 mm) caused an increase of stresses by 53% in relation to the smooth drum of the shell thickness 65 mm, but a reduction of the shell thickness by 30% (i.e. 45mm) caused an increase by 13%. The most optimal variant concerned the shell thickness of 55 mm. In this case a reduction of the shell thickness by 15% in relation to the basic thickness (65 mm) enabled to obtain the stresses in the shell on the level of about 137 MPa. These values are close to the stresses in the shell without ribs of the thickness 65 mm (143.2 MPa). To get a full image of the stresses level in the construction, the stresses in circumferential ribs for different thicknesses of drum shells, are presented in Fig. 19.

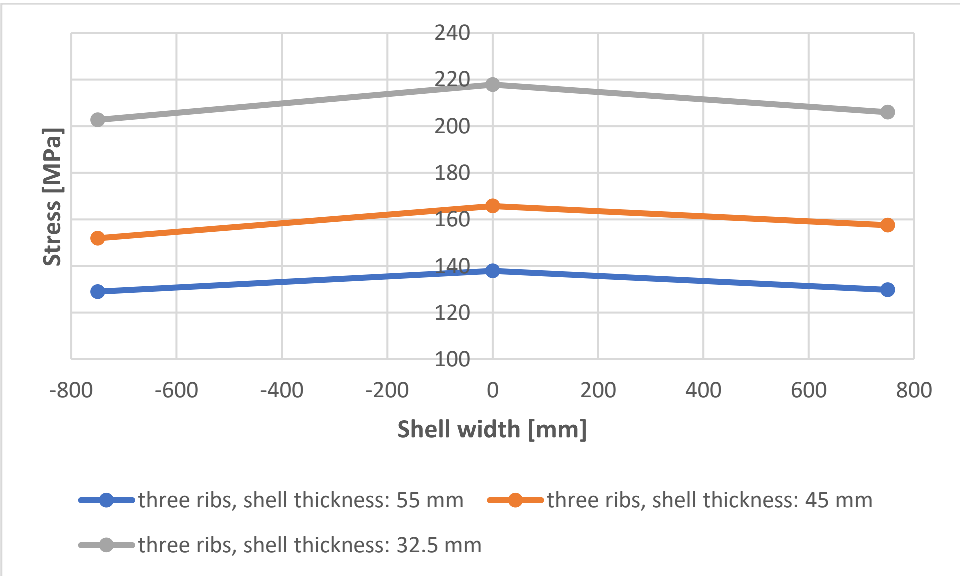


Fig. 19. Maximal stresses in ribs

The results of calculations are presented in Table 2 for drums of different shell thicknesses equipped with three ribs.

Table 2. Stresses in drum shells and in ribs in the case of different shell thicknesses and three circumferential ribs.

Shell thickness [mm]	Drum without ribs [MPa]	Drum with ribs	
		Shell [MPa]	Rib [MPa]
65	**143.2**	116.3	124.9
55	-------	**136.6**	**138.9**
45	-------	161.0	154.5
32.5	-------	219.2	172.0

4. Summary

The presented results of strength calculations of the winding drum of the hoisting machine indicated that an installation of circumferential ribs under the shell gives measurable advantages in a form of reduced stresses. It is achieved at the cost of additional technological machining i.e. a manufacture of rings and then their welding to the drum shell. An introduction of circumferential ribs enables to reduce the shell thickness. A correlation between the drum weight, in the case of the shell thickness change and a number of introduced ribs, is shown in Table 3.

Tabela 3. Drum weight for different shell thicknesses and a different number of circumferential ribs.

Shell thickness [%]	Drum without ribs [kg]	Drum with one rib [kg]	Drum with three ribs [kg]	Drum with five ribs [kg]
65.0	**76800**	**79020**	**83460**	**87900**
55.0	70650	72870	77310	81750
45.0	64500	66720	71160	75600
32.5	56800	59020	63460	67900

An approximate weight of the drum without circumferential ribs, with the shell of 65mm thickness, subject to an analysis, is about 76800 kg. It can be assumed that from the strength point of view (Table 2) a drum with the shell of 55 mm thickness and three ribs will be an equivalent of a drum with the shell of 65 mm thickness without ribs. Therefore, keeping the same level of stresses in the drum with the shell without ribs and with three ribs, it was not possible to obtain any advantages in the aspect of an essential reduction of the drum weight-an increase of weight below 1%.

An installation of circumferential ribs under the shell requires a series of additional technological operations. A comparative tabulation of operations for a manufacture of a drum with the shell without ribs and with circumferential ribs is presented in Table 4.

Special attention should be paid to the fact that an efficiency of ribs can be achieved exclusively in the case of their exact fitting to the drum shell and their good connection by welded joints.

An introduction of circumferential ribs can be justified in the case of drum shells of very big thicknesses e.g. above 150 mm, big overall dimensions and a big weight, when a reduction of these parameters turns out to be indispensable.

Table 4. A comparison of manufacturing technologies of a shell with ribs and of a shell without ribs.

Name of technology	Shell without ribs	Shell with ribs	Comments
Bending of shell sheet	**yes**	**yes**	Bending of shell sheets from 20 to 120 mm does not cause any technological problems. With an increase of thickness of the sheets subject to bending, a number of companies which can do this operation, decreases.
Machining of shell from inside	no	**yes**	A need of machining some parts of shell to obtain a round surface for fitting the ribs (rings) may occur.
Manufacture of ribs	no	**yes**	Rings made of segments, then welded with each other. Edges must be prepared for an execution of welds.
Welding of ribs to shell	no	**yes**	In the case, under analysis, about 3x21m=63m of K-weld (estimated about 40h). An additional hazard is a deformation of shell sheet rolling due to an introduction of heat during welding.
Tests of welds	no	**yes**	Magnetic-powder method or ultrasounds

The results of the FEM numerical calculations, presented in the article, are the result of research and development projects realized at the KOMAG Institute. These calculations were conducted with use of the Autodesk Inventor Nastran Software.

5. Conclusions

While designing rope shaftings of hoisting machines in the system with winding drums and with the frictional contact, ribbing of shells is avoided. An introduction of ribs, under the shell, parallel to the drum axis should be avoided firmly [10] as they are components which have no impact on the strength increase of the drum shell.

Circumferential ribs, which are correctly chosen according to the drum design as regards their number and geometry, give positive strength effects – a reduction of stresses in the construction. However, it should be borne in mind that in most cases their introduction causes an increase of the drum manufacture costs.

The conclusions, presented above, are related exclusively to the drum under analysis and they show some possibilities of a construction optimization as regards the strength and production costs. Due to a big variety of constructions and loading conditions, each winding drum is different and it should be designed individually.

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