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RECONSTRUCTION OF THE MAIN CYLINDER OF CARDING MACHINE–OPTIMIZATION OF DIMENSIONS WITH THE USE OF THE FINITE ELEMENT METHOD

The following paper presents the solution to the problem of searching the best shape – structural form of the bottoms and optimal dimensions of the main cylinder of the carding machine with consideration to the criterion of minimal deflection amplitude. The ANSYS package of the Finite Element Method has been used for the analysis. Polak-Ribery conjugate gradient method has been applied for searching the optimal solution, basing on the parametric model of the cylinder written with the use of *Ansys Parametric Design Language.* As a result of the performed analyses, reduction of maximum deflection value at approximately 80% has been obtained. Optimal cylinder dimensions enable application of a new textile technology – microfibre carding and improvement in the quality of traditional carding technology of woollen and wool-like fibres.

1. Introduction. The object of the analysis

Carding – one of the most important textile technological processes whose aim is intermixing loose fibres, removing rubbish and short fibres as well as straightening and parallel arrangement of the fibres left. Next, they are to be formed into a semi-finished product in the form of a thin layer of fibres with the width of the machine – web or sliver which is formed by condensing the web. Woollen and wool-like fibres (chemical fibres with similar properties) are reworked with the use of roller carding machines. Their scheme is shown in Fig. 1.

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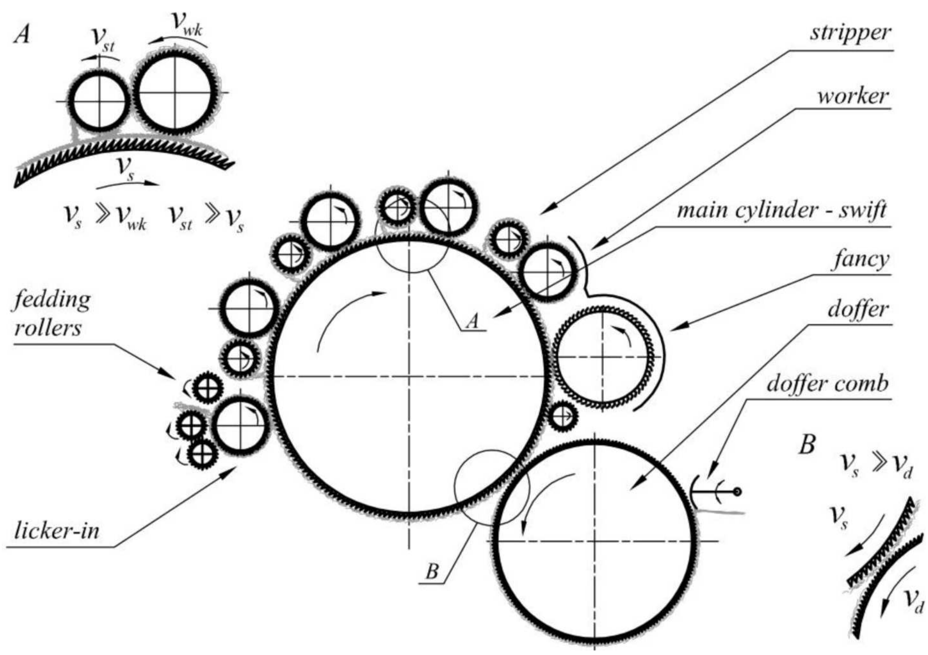

Fig. 1. Carding machine scheme

On the surface of the main cylinder – swift, the doffer and working rollers (workers and strippers) there are teeth of a saw metallic wire (Fig.3) or card clothing needles, which are reeled coil to coil with appropriate tension. Fed by feeding rollers and licker-in (the first carding machine in carding set) and the common roller (the following carding machines in carding set), the fibres are taken over by the swift teeth and carried to the other pairs of rollers worker-stripper. Thanks to the large difference of tangential velocities of the swift v_s and the worker $v_{wk}(v_s \gg v_{wk})$ and the small distance between them, their teeth separate, comb and straighten the carried fibres.

The stripper, whose tangential velocity v_{st} is greater than the tangential velocity of the worker ($v_{st} > v_{wk}$), feeds back a part of fibres for another carding. The process is repeated between the cylinder and the following pairs of rollers worker-stripper. However, the distance between the swift and next workers decreases. Finally, a roller rotating at high speed – fancy ($v_f > v_s$) with long bent needles lifts the fibres above the teeth of the swift which carries them further and condensing them on the teeth of the slowly rotating doffer (v_d) because $v_s \gg v_d$. If the carding machine is the last machine in the carding set, the oscillating blade (doffer comb) combs and condenses the fibres after carding, forming the web.

Carding quality depends to a large degree on the height and shape of the gap between the swift and worker and between the swift and the doffer.

When carding thin fibres, the gap height is between 0.3 and 0.15mm, and considering desirable uniformity of the web, its shape should be close to the shape of a rectangle. Taking into account the durability of the metallic card wire teeth (hardness of a tooth is decreasing from top to base), the cylinder should not be ground after the wire has been reeled. It is worth to add that textile industry more frequently uses super-thin fibres (microfibers), whose carding requires smaller distances between working rollers and the swift and between the swift and the doffer than those mentioned above.

The swift should be designed in such a way so that after reeling the metallic card wire on it at tension, the deflection of its shell does not exceed several hundredths of a millimeter. Taking into account its dimensions (diameter × length: the main cylinder Ø 1500mm × 2500mm, the doffer Ø 1270mm × 2500mm) and the wall thickness usually between 10 and 14mm, the requirements as to the construction stiffness are very high.

The cylinders of modern carding machines are almost solely welded constructions (Fig. 2). They consist of the shell reeled of metal sheet welded along its edges, bottoms with hubs stiffened with ribs, the shaft and the reinforcement rings. Such constructions are used by leading carding machine manufacturers, and particular solutions are different as to the side hubs construction and reinforcement rings.

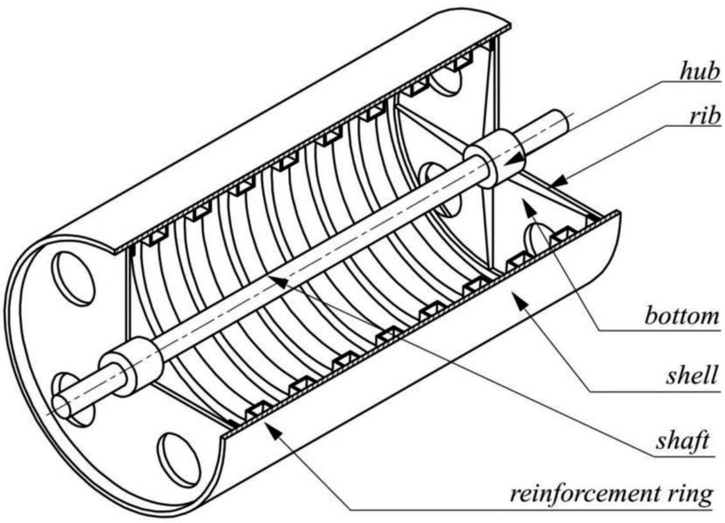

Fig. 2. Cylinder construction

2. Loads acting on the main cylinder of carding machine

The loads acting on the cylinder result from:

– the influence of fibres on the teeth of card wire during the carding process; the forces are small [2] and are omitted when calculating deflection of the cylinder shell,
– the construction deadweight and the centrifugal force; their influence on the cylinder deflection may also be neglected due to small rotational speed of the cylinder (circa 100÷200rpm) [5],
– reeling at tension of the metallic card wire.

Reeling at tension S and the reeling pitch t of the card wire on the cylinder with the radius R (Fig. 3) exerts pressure on the cylinder shell directed radially inward with the value of

$$p_r = \frac{S}{tR} \tag{1}$$

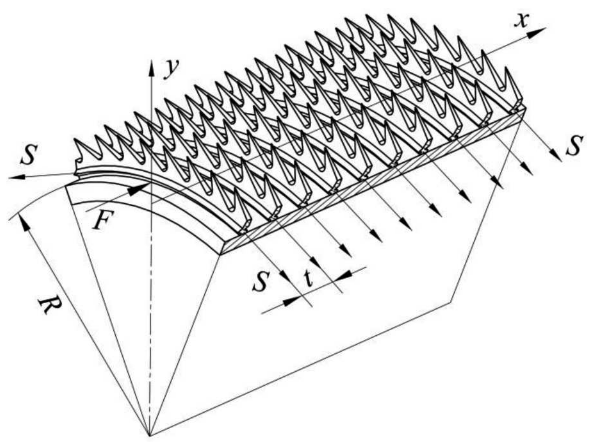

Fig. 3. Tension forces acting on the metallic card wire coils

As shown in paper [6], taking into account only radial pressure p_r in calculations of the cylinder shell deflection does not give sufficient compatibility of calculation results and experimental measurements. When reeling the metallic card wire at tension S, the coils are pressed against each other with the force F acting in the direction of cylinder axis and the so-called side wire which is the stopper ring. After reeling all coils, the last coil rests against the side wire on the opposite side of the cylinder. One may say that the cylinder shell is reeled with 'foil of metallic card wire' which is initially tensioned and acts on the cylinder in radial direction with the pressure p_r and in axial direction with surface force p_x which is the result of residual friction forces between the metallic card wire and the cylinder shell. The value of the surface force p_x is expressed by the following equation:

$$p_x = \mu p_r \tag{2}$$

where: μ – coefficient of static friction between the saw wire flange and the cylinder surface.

The authors of the paper [6] proved that taking into account both the pressure p_r as well as the surface force p_x when calculating the deflection of the cylinder shell, gives sufficient compatibility of calculations with experimental tests.

3. Optimization of cylinder dimensions – summary of the previous work results

The deflection $y(x)$ of the cylinder shell with the radius of the middle geometric cylinder R and the thickness h with stiff bottoms under the influence of the radial pressure p_r may be determined by means of the following differential equation

$$\frac{d^4y}{dx^4} + 4k^4y = \frac{12p_r\left(1-\nu^2\right)}{Eh^3} \tag{3}$$

where: $k^4 = \dfrac{3\left(1-\nu^2\right)}{h^2R^2}$, E – Young's modulus, ν – Poisson's ratio

The deflection amplitude is then equal to:

$$\Delta = y_{\max} = -\frac{p_rR^2}{Eh} \tag{4}$$

Therefore, an obvious conclusion can be drawn. Decreasing the deflection amplitude of the cylinder with a given radius and with a given load applied requires increasing the wall thickness h. However, the increase in thickness h increases the cost and the mass of the cylinder. Moreover, it makes the mass moment of inertia bigger, which in turn causes problems with starting the machine and imposes the necessity of using special brakes for stopping the cylinder in the required time in the case of a break-down. Therefore, in the 1990s carding machines were equipped with cylinders with internal ring-like reinforcements (Fig. 2). The rings stiffen the shell without significantly increasing its mass moment of inertia. Optimization task of dimensions of the cylinder with reinforcement rings was subject to the analysis in paper [5]. Decision variables of the task were the shell thickness and the ring cross-section area. The task was solved by minimizing deflection amplitude at the same time keeping the mass of the cylinder without rings and minimizing the mass at the same time keeping the permissible deflection amplitude. The task was solved for a different number of rings determining their optimal number. The task used the continuous calculation model and analytical solution of the problem of cylinder shell deflection under the influence of radial pressure p_r taking into account the flexibility of rings and bottoms. The nongradient Powell's method of improving direction has been applied to solve the optimization task. The results were implemented by a local carding machine manufacturer. Fig. 4 shows the comparison of the deflection line of cylinder shells with the same mass without rings and with reinforcement rings.

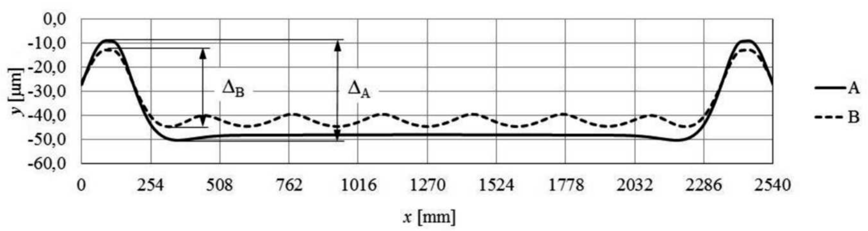

Fig. 4. The deflection line of the cylinder shell: A – without rings, B – with reinforcement rings

The value of the deflection amplitude in the case of A is $\Delta_A \cong 41\mu$m. Introducing reinforcement rings enables the decrease in the amplitude of almost 22% to the value of $\Delta_B \cong 32\mu$m in the case of variant B.

Further work on the development of the construction of carding machine cylinders and growing requirements as to its stiffness showed the necessity to develop a discrete model of the cylinder which would be more adequate for the real construction. Such a calculation model developed with the use of the FEM method was described in paper [6]. The model took into account the pressure p_r and the surface force p_x, flexibility of all construction elements of the cylinder and additionally local flexibilities of welds and the deviations in wall thickness of the cylinder shell after machining. The model was experimentally verified. It is worth to add that the cylinder analysed with the help of the model instead of bottoms welded of metal plates was equipped with cast iron wheel hubs with five arms.

The analysis of the deflection line of the cylinder shell with optimal thickness with rings with optimal dimensions [4] proves that further decrease of the deflection amplitude requires applying special bottoms welded with steel plates (Fig. 5) which would be flexible in axial and radial direction and at the same time would ensure the required stiffness of the whole cylinder both during work and during its manufacturing (grinding the surface with the disk-type grinding wheel before reeling the metallic card wire).

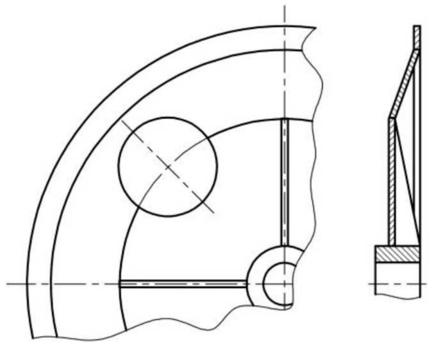

Fig. 5. Changed construction of the bottom with conical ring

4. Optimization of cylinder and bottoms

In order to determine optimal form of cylinder construction and its bottoms with conical ring, parametric FEM model has been prepared and appropriate optimization task has been formulated and solved.

4.1. FEM Model of the cylinder

The discrete calculation model of the cylinder (Fig. 3a) with a flat side hub (compare Fig. 2) was made in ANSYS package [1] using *Shell 63* elements (shell and bottoms and its ribs) and *Beam 188* elements (rings and shaft). *Shell 63* is a four-node shell element with six degrees of freedom in each node – three translations and three rotations, which takes into account the membrane and bending state of the shell. *Beam 188* is a two-node beam element which is consistent with Timoshenko beam theory. During the process of shaft discretization, it was ensured that the nodes were located in places where there are self-aligning rolling bearings supporting the cylinder. Hinged support on the right side (the side where the belt transmission driving the cylinder is placed) and the roller support on the left side were applied in the nodes.

Due to the fact that in real constructions the side hub circumference is permanently fixed to the shell, the shell nodes which lie on the side hub surface and the adjacent nodes on the left and right side were coupled, ensuring the compatibility of node values (translations and rotations).

Axisymmetrical radial surface pressure p_r and pressure acting in the direction of the cylinder axis p_x (Fig. 6b) were applied to the cylinder shell. The pressure values were assumed on the basis of [7] as equal to p_r =220 kPa and p_x =33 kPa.

All analyses were performed with the use of input files for ANSYS programme. Using the *Ansys Parametric Design Language,* a parametric FEM model of the analysed construction has been written and the optimization task formulated. It is worth to emphasize that the input file prepared in this way enables fast analysis and optimization of cylinder constructions for various working widths and constructional forms (e.g. with any number of reinforcement rings).

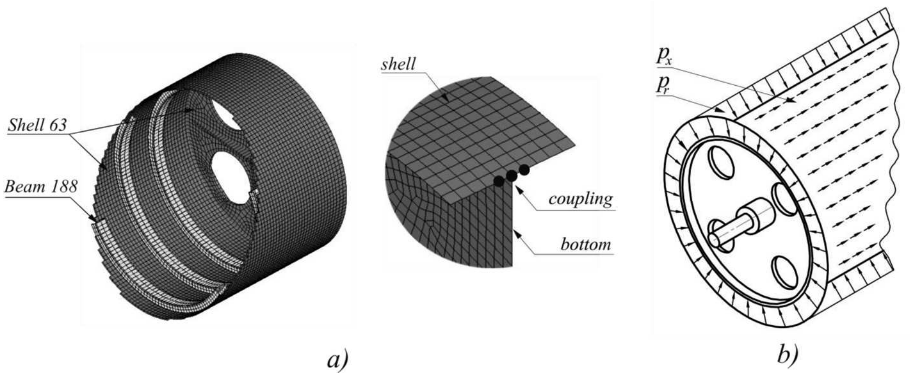

Fig. 6. Discrete cylinder model: a) FEM model – fragment, b) loads scheme

Discrete models for cylinders having bottoms with conical ring (Fig. 5) have been developed in a similar way.

4.2. Calculation results

As a result of the discrete model analysis of the cylinder with flat bottoms, a graph of deflection line has been obtained (Fig. 7). The value of deflection amplitude is $\Delta \cong 36\mu$m. It is worth stressing that taking into account the pressure in the axial direction p_x causes the increase of deflection amplitude at approximately 12% (compare section 3). Moreover, it is worth mentioning that taking into consideration pressure p_x results in the line of shell deflection not being symmetric.

In order to examine the influence of structural form of the bottoms on the value of deflection amplitude, a series of numerical tests for various configurations of cylinder constructions have been performed. The two of them, most interesting from the point of view of the following paper, are presented below. In the case of configuration I (Fig. 8) one may assume that the bottoms bending under the pressure p_r cause 'straightening' of the shell and significant improvement of the present state. In reality, deflection amplitude is decreased to $\Delta \cong 25\mu$m.

Analysing the deflection line in Fig. 8, one may notice that on the right side the deflection amplitude decreases. Basing on the observation, the discrete calculation model was modified in order to correspond to configura-

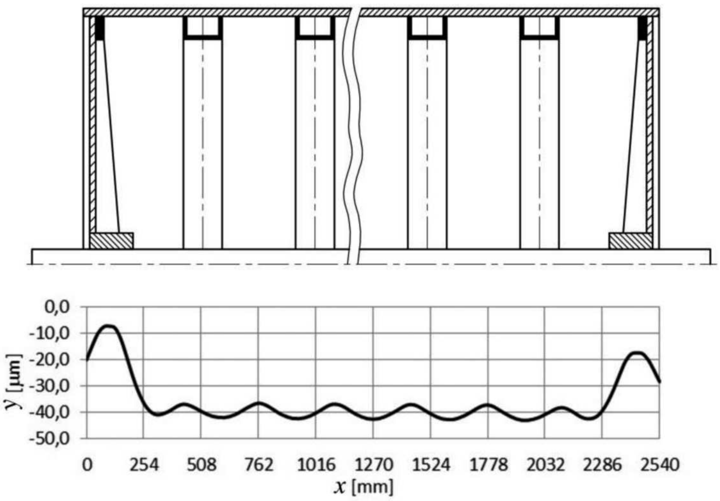

Fig. 7. The scheme of the cylinder and the deflection line with flat bottoms

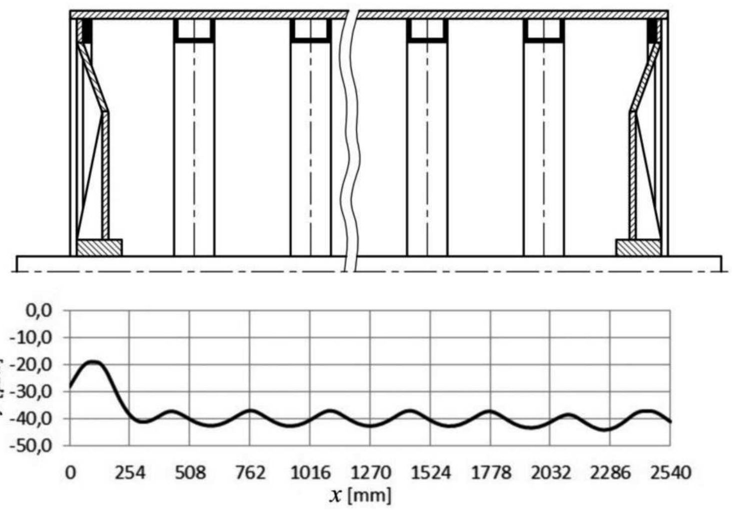

Fig. 8. The scheme of the cylinder and the deflection line for configuration I

tion II in Fig. 9. In this case, the deflection amplitude of the shell is $\Delta \cong 8\mu$m. For this type of constructional form of the cylinder, one may practically eliminate the influence of axial pressure p_x on the deflection of the shell (warping effect of the shell disappears).

4.3. Optimization task

In order to determine optimal dimensions of the cylinder as presented in Fig. 9, the following optimization task has been solved [3]:

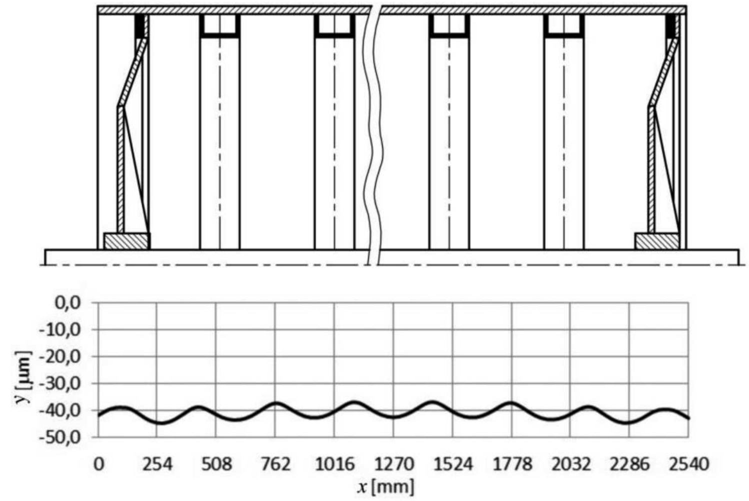

Fig. 9. The scheme of the cylinder and the deflection line for configuration II

1. Decision variables (Fig. 10)
   – thicknesses: the shell g_1 and the bottoms g_2,
   – the distance between the bottom and the first reinforcement ring b,
   – dimension c which is characteristic for the bottom form i.e. for the angle of the conical ring,
   – dimensions of the reinforcement cross-section: h,s, t,
2. Objective function:
   – deflection amplitude of the cylinder shell $\Delta \rightarrow \min$ (compare Fig. 4),
3. Limitations
   – the mass of the cylinder with optimal dimensions should not exceed the mass of the cylinder before optimization: $\hat{m} \leq m_0$,
   – maximum reduced stress according to Huber-Mises hypothesis in the construction cannot be greater than permissible stresses for constructional steel S235JR:k_r =120MPa.

Moreover, appropriate variation ranges for decision variables have been introduced for the task. The ranges resulted from among other things technological conditions such as: difficulties in proper manufacturing of the shell by reeling a flat sheet of steel with the width of over 14mm, as well as boundary dimensions of a section of reinforcement rings resulting from the technology of reeling the ring and the availability of the semi-finished product.

In order to solve the task, a Polak-Ribery conjugate gradient method has been used together with the interior penalty method.

As a result of solving the above-mentioned optimization task, optimal values of decision variables $\hat{g}_1/\hat{g}_2$ = 14/11 mm, $\hat{b}$ = 298 mm, $\hat{h}/\hat{s}/\hat{t}$ = 35/24/4.4 mm have been obtained. The mass of the optimal con-

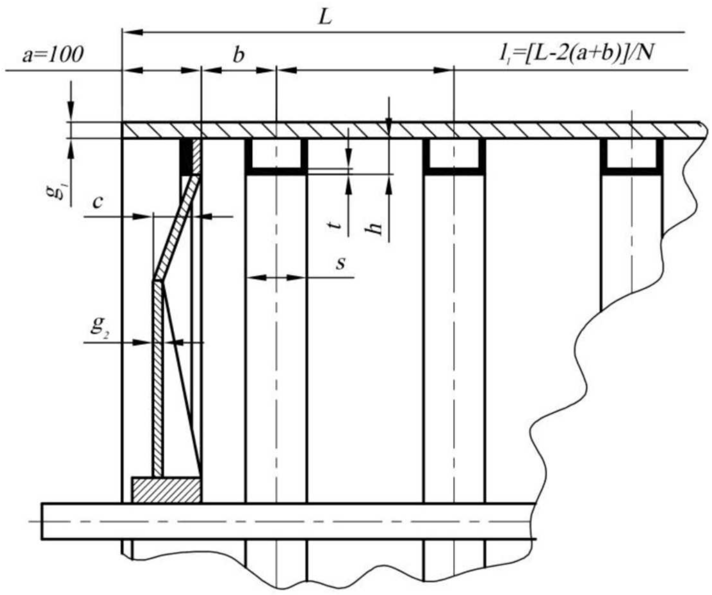

Fig. 10. Denotations of decision variables in optimization task

struction is equal to its mass before optimization and is $\hat{m}$=1635 kg, the deflection amplitude is $\Delta = 6.2\mu$m (Fig. 11). In relation to the deflection before optimization, it decreased by another 20%.

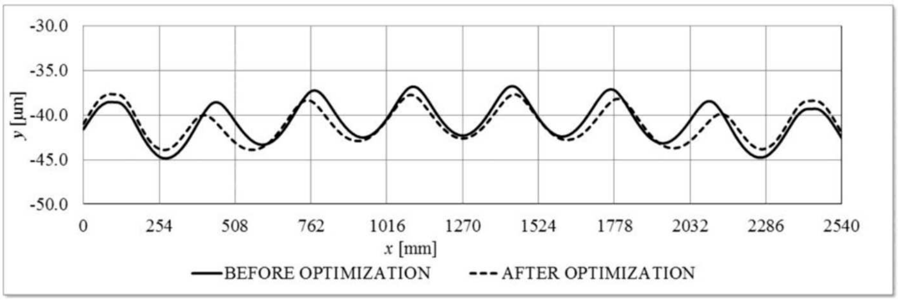

Fig. 11. Deflection lines of the cylinder shell before and after optimization

5. Conclusion

The following article presents development work aiming at improving the stiffness of the construction of axisymmetric, thin-walled cylinder shell with bottoms and internal rings under the influence of axisymmetric radial pressure and the pressure acting along the shell axis. Such constructions are used, among other things, in textile machines – carding machines. The aim of the work was to decrease the deflection of the cylinder shell without significantly increasing its mass especially the mass moment of inertia in relation to the axis of rotation.

All numerical simulations have been performed on discrete models developed with the use of the FEM method. Analysing the results of the previous work, an improved construction of bottoms with a conical ring has been suggested (Fig.5). The best arrangement in the cylinder (Fig.9) has been determined for the bottoms, obtaining the decrease in the deflection amplitude of the cylinder shell from 36 μm (Fig.7) to 8 μm (Fig.9) that is at about 71%. Further decrease in the cylinder shell deflection to the value of 6.2 μm i.e. at another 20% was achieved by formulating and solving the appropriate optimization task of cylinder dimensions.

Considering the aim of the work, one may conclude that the suggested constructional form of the cylinder, with optimal dimensions according to the criterion of minimal deflection amplitude, ensures stiffness of the cylinder appropriate for carding very thin fibres including micro-fibres.

In a real cylinder, one must take into account bigger shell deflections caused by: cylinder manufacturing tolerance (radial run-out of the cylinder shell is ± 10 μm), metallic card wire manufacturing tolerance (on the average ± 10 μm) and lack of uniformity of the cylinder wall thickness and possible flexibility of the welds [6].

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Rekonstrukcja bębna głównego zgrzeblarki wałkowej – optymalizacja wymiarów z wykorzystaniem metody elementów skończonych

S t r e s z c z e n i e

W pracy przedstawiono rozwiązanie zadania poszukiwania najlepszego kształtu – postaci konstrukcyjnej dennicy oraz optymalnych wymiarów bębna głównego zgrzeblarki wałkowej z uwagi na kryterium minimalnej amplitudy ugięcia. Do analiz wykorzystano pakiet ANSYS metody elementów skończonych. Rozwiązania optymalnego poszukiwano metodą gradientu sprzężonego w wersji Polaka-Ribery, bazując na parametrycznym modelu bębna zapisanego z wykorzystaniem języka *Ansys Parametric Design Language*. W wyniku przeprowadzonych analiz uzyskano zmniejszenie wartości maksymalnego ugięcia o około 80%. Optymalne wymiary bębna umożliwiają realizację nowej technologii włókienniczej – zgrzeblenia mikrowłókien oraz poprawę jakości w tradycyjnej technologii zgrzeblenia włókien wełnianych i wełnopodobnych.