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Influence of Teeth Manufacturing Tolerances of Cylindrical Geared Wheels on Gear Operating Characteristics Based on Spinner Stretching Apparatus

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Abstract
In the paper, issues related to the errors and inaccuracies resulting from improper operation of the drive system of the stretching apparatus of a ring spinner are discussed. The aim of the paper was an analysis of a design solution for the driving system of a ring spinner and its operating behaviour influencing the quality parameters of the fiber jet. In the paper, modelling of the involute teeth shape by means of a BEASY package was done. Based on the model prepared, an assessment of the sensivity of geometrical teeth outline on particular dynamical quantities for the gear of the stretching apparatus was performed. An improvement in the parameters of operating of the spinner can be achieved via changing of the design characteristics of elements of the driving system; in particular, modification of the teeth shape was considered. Within the scope of the work, numerical analyses of the meshing geared wheels were conducted taking into consideration the unevenness of operating of the drive system as well as load distribution acting along the line of action. In the paper, the boundary element method was utilised for evaluation of the load distribution in relation to spectral analysis connected with the unevenness of the fibers' jet mass.

Key words: *gear, machine diagnostics, manufacturing deviations, geared wheel.*

Nomenclature:

- a — *current distance between axes,*
- $A, B, C, E, F, H_z, N_w, S, V$ — *exchangeable geared wheels,*
- a_0 — *nominal distance between axes,*
- d — *shaft diameter,*
- g_m — *amplitude of tooth tip relief,*
- h_m — *length of tooth tip relief,*
- p_b — *nominal circular pitch,*
- r_a — *radius of addendum circle of geared wheel,*
- r_b — *radius of base circle,*
- r_{w0} — *rolling radius of the tool,*
- u — *local gear ratio (for a pair of cylindrical geared wheels),*
- x_R, y_R — *parametric equations of equidistant curve,*
- α — *pressure angle,*
- α_w — *roll pressure angle,*
- $\Delta\alpha$ — *deviation of pressure angle,*
- Δp — *deviation of pitch,*
- $\Delta\phi$ — *rotational angle of driving wheel (pinion),*
- λ — *wave length.*

Introduction

The constant development of new production techniques and increasing user demands cause that machines producers are forced to put into practice innovative solutions as well as to improve the operating parameters of their products. Ring spinners are machines which enable the manufacturing of high quality products, simultaneously being reliable at a satisfactory level. The application of new techniques, new design solutions and special materials leads – just within the design stage and planning diagnostics procedures – to the utilisation of modern modelling and calculation methods as well as to the consideration of unconventional design solutions. Namely the passing of a rotational moment (torque) throughout the rotating machine parts is performed via the series of gears, which cooperate directly with the stretching apparatus of the machine.

The stretching apparatus itself is an essential element of the spinner driving system which assures the uniformity and stability of operation of the device. Any defects have a direct influence on the spinning process and are considered as an essential factor influencing breaking of the spinning jets. In cases where the yarn is made (manufactured) improperly, a defect arises, revealing itself in the occurrence of unevenness of the yarn linear mass, which is connected with the random distribution of jet fibers in a cross-section of the yarn.

One of the main problems in the functioning of contemporary industrial systems still remaining is the improvement of the quality of the manufacturing process. Therefore there is an extreme necessity for the improvement of the quality of all machinery subsystems related to the manufacturing process, especially improvement of the operating of the drive systems is required. The drive systems are subjected to some occasional mechanical failures. Moreover breakdowns of the drive systems are time-consuming and result in the necessity of immobilization of the whole production line – which, in turn, essentially increases the production costs, finally reducing the factory's income. Unexpected breakdowns are especially disruptive, and could be sometimes caused by improper assembly work, as mentioned previously, resulting in unplanned stoppage, which decreases overall equipment effectiveness. Moreover this damage even influences the functioning of the factory itself – if the plant activity is based on the idea (or manufacturing strategy) called quick response manufacturing.

One of the prevention methods aiming at avoidance of unplanned stioppages of the drive systems is their permanent monitoring using special dedicated equipment. The aims of monitoring are as follows: detection of symptoms of damage to particular elements and support and/or planning of repair schemes – assuming that repairs are relevant to the current operating states of the device.

In monitoring we consider the continuous observation of diagnostic device displays. The relevant diagnostic gauges could be mounted on particular machines or the engineers can perform an analysis of other apparatuses which register online operating parameters of the production process.

Aiming at an increase in productivity and minimisation of the occurrence of damage in industrial systems, monitoring systems are widely utilized. Recently monitoring systems have evolved from simple systems for visualisation of machine operation up to expert systems which inform about threats, indicating where a threat occurs, and which measures have to be taken and about (approximately) when total damage (beak-down) will take place. It is obvious that a monitoring system cannot not be established without diagnostic methods and procedures which enable the measurement of appropriate parameters and derivation of algorithms for the detection of damage.

Gears are widely applied in versatile textile machinery e.g. ring spinner, as torque reducers. Therefore the continuous and reliable operating of these textile machines depends on the strength, durability and reliability of their incorporated gears. Whipping of yarn jets is an essential factor influencing the productivity of the spinning process. Furthermore it is even one of the most important problems of the whole technique of manufacturing yarn because it influences the production output, quality of yarn manufactured as well as the quantity of unwanted waste materials yielded.

Uneven operation of the drive system as well as its particular parts are mainly responsible for the whipping of yarn. Even damage to a single element of the drive system causes unevenness of the product e.g. boldfaces and narrower segments of yarn, observable on the surface of the final product. The stretching apparatuses of ring spinners are essential elements, therefore frequent variations in the rotational velocity have a direct influence on the dynamic character of loadings of the system. The stretching apparatus is driven via a series of cooperating gears.

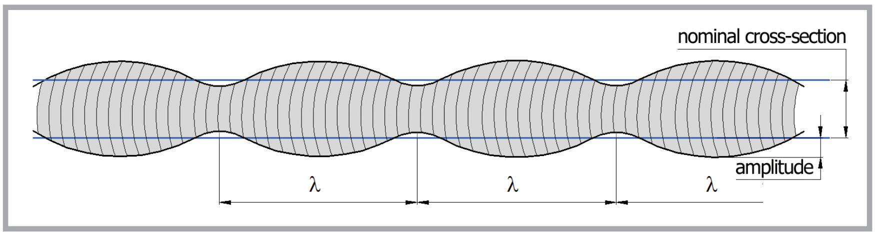

***Figure 1.** Example of the appearance of periodical damage in a fiber jet [7].*

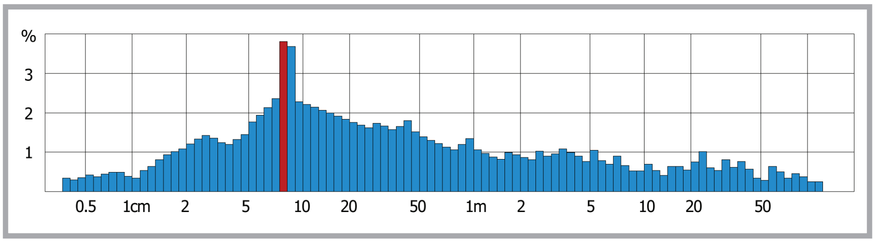

***Figure 2.** Spectrogram with exemplary marked periodical inaccuracy of distribution of the yarn jet for woollen worsted yarn of 36 tex linear mass [7].*

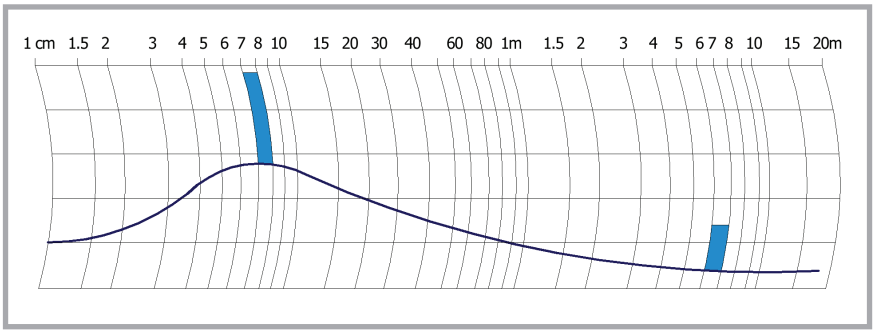

***Figure 3.** Spectrogram with marked local increased amplitudes, so called 'chimneys', which have a negative influence of product quality [7].*

In the case of seemingly properly manufactured yarn, there sometimes arises a problem connected with the appearance of unevenness of the linear mass of yarn. It is related to the random distribution of the yarn jet in the cross-section surface of yarn. In the case of randomly appearing defects, there is a possibility of determination of the size, number and intervals between boldfaces (fluctuation of linear mass). If there is a need for a more detailed description of the phenomenon observed, it is possible to make some calculations based on the statistical approach. There is a possibility of performing an analysis of periodical defects in the case of a perfectly random distribution of the yarn jet – any peaks (in spectrogram) should not be spotted i.e. any local amplifying of an amplitude (***Figure 1***). Such a type of defect relatively frequently arises not only during the manufacturing processes before the final spinning process, but also in the yarn itself. The reasons for this are as follows: The periodicity of mass distribution can be caused by versatiles factors, e.g. most frequently recognized: uneven operation of the twisting-winding device, and non-centric operating of stretching shafts inside the stretching apparatus. They can cause the appearance of the phenomenon of periodicity of the mass distribution in the yarn jet.

Within the spinning process, there are several sources of mass variability which – in practice – could be recognised as periodical in some time intervals. These defects are connected with vibrations passed from the drive system. Uneven operation occurs randomly, and can randomly disappear. In the case of particular velocities and some amounts of vibrations as well as the eccentric rotation of shafts, the probability of mass

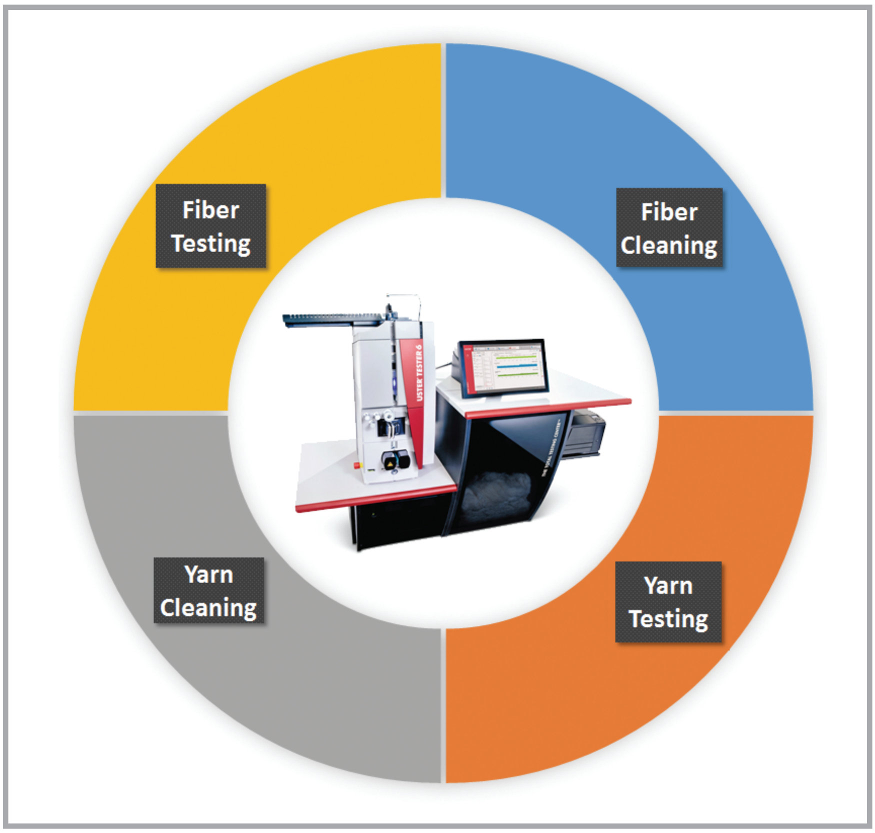

Figure 4. *'Uster Tester 6' device for monitoring of the whole technological process based on [18].*

unevenness increases. This kind of defect shows traces in visible peaks on the spectrogram. The peak's height is related to the approximated mean value of periodical variations in the linear mass in time. The type of defect described is shown graphically in the spectrogram in ***Figure 2***.

All peaks appear in some time intervals, but they are not regularly registered nor similar.

Periodical variations in the mass of the yarn jet have (not always) a statistically significant influence on the value of coefficient mass variability U or CV. These defects can – in consequence – be seen in such materials as knitwear or fabrics, however customers or consumers do not accept such products. Noticeable, unintended patterned surfaces of the final products could be even more visible (more intensive) after the tinting process, especially relating to monochrome products as well as those made of continuous chemical fibers. The level at which periodic mass variability could influence the quality and appearance of the final product depends not only on the intensity but also on the width and type of the product manufactured, type of fiber, type and linear mass of yarn as well as on the kind of yarn dyeing technique applied [19-20] (see ***Figure 3***).

Aiming at full control of quality, a new apparatus, the 'Uster Tester 6', was released by the company 'Uster Technology'. The modified device enables monitoring of the whole technological process. The measurement and monitoring set co-operates with the moduli installed directly on machines (see ***Figure 4***).

Due to such a modular solution, there is a possibility of identification of technological errors (failures) caused by damage to working elements of the drive systems.

The introductory condition – enabling localisation of defects in the drive system of the spinner analyzed – is precise knowledge of the layout of the gears and other mechanical parts inside the installation (see ***Figure 5***). In general, the following analyses should be done.

Damage to the gear or a blockage of the drive F-C (see ***Figure 5***) arising from contaminations possibly coming from materials via rotation of the front shaft of the spinner is passed throughout the system – having the following parameter: wave length: $\lambda_1 = d_{F,C} \cdot \pi \approx 8$ cm.

In the case where the gear system driving output shafts B-A and intermediate wheel

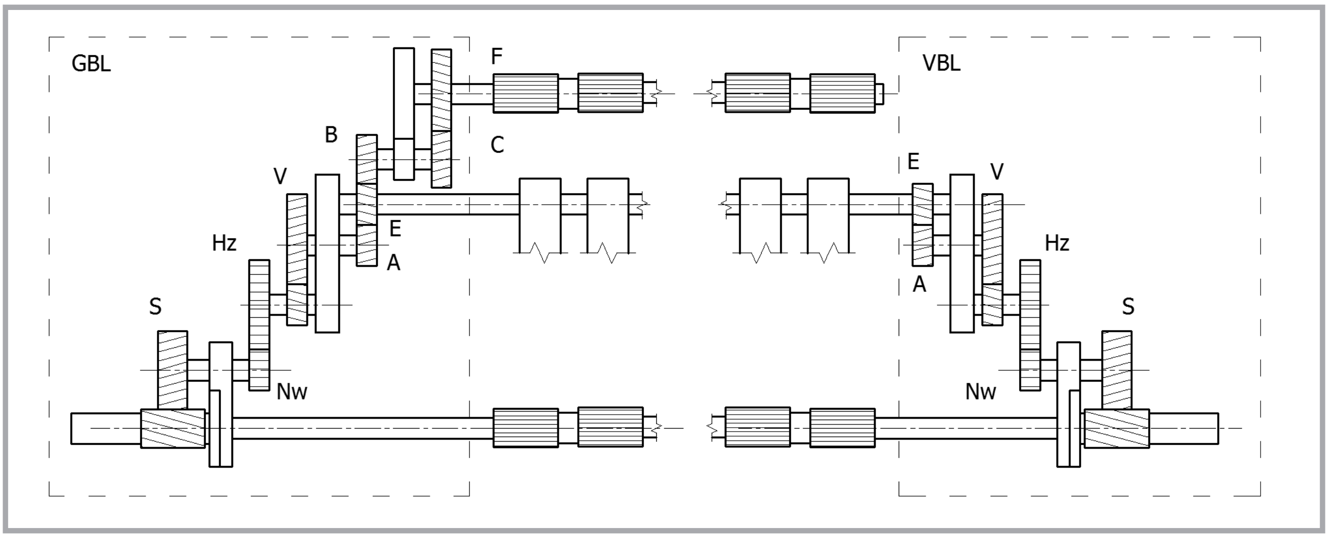

Figure 5. *Scheme of stretching apparatus of the ring spinner Fiomax 2000 [17].*

E – being damaged, then the relationship between co-operating geared wheels (number of teeth C and F, respectively) can be written as $\lambda_2 = \lambda_1 \cdot u_{F,C} = 88$ cm. Consecutive damage will occur after the performance of 11 rotations by the previous stretching shaft.

Due to damage to geared wheel A, in the yarn jet the same unevenness arises as in above-considered case related to the wheel F (***Figure 5***). It results from the fact that both wheels co-operate in the same two-stage gear. Damage to geared wheel B causes excitation of the wave whose length is related to the numbers of appropriate teeth i.e.: $\lambda_3 = \lambda_2 \cdot u_{A,B} = 2.64$ m (***Figure 6***).

Summarizing, the stretching apparatus should be considered as an essential part of the spinner drive system, therefore frequent changes in the rotational velocity have a direct influence on the dynamic character of load variability. The role of the stretching apparatus is the setting of tensions [11] via a system of co-operating geared wheels.

The aim of the paper was an analysis of a design solution for the drive system of a ring spinner and its operating behaviour influencing the quality parameters of the fiber jet.

Among the gear elements, the gear wheels are the parts which are most frequently subjected to damage [4, 10]. The damage could be of the following types: fatigue breakage or casual breakage (***Figure 7.a***), scoring or wear (***Figure 7.b***) as well as the arising of pitting holes on the surfaces of teeth faces and/or flanks (***Figure 7.c***).

Utilisation of better and better materials for elements of, for example, gear wheels, allows for the diminishing of the mass and size of the gear. However, a side effect consists in the diminishing of the stiffness of its elements, due to which elastic deformations along with manufacturing tolerances (deviations) have a direct influence on the reliability of gears and the whole stretching apparatus [21].

Determination of teeth shape parameters for cylindrical gear wheels of the drive system of spinner

Properties of toothing and teeth engagement depend on the profile and shape of teeth as well as on their dimensions. In ***Figure 8***, a sketch of the toothing of a typical gear co-operating with the stretching apparatus is presented. The shape of the transition surface and tooth profile is determined taking into account the geometrical shape of the working tool (tooth of a toothed rack) utilised in the manufacturing process.

In the drive systems, involute (also known as evolvent) profiles are utilised for the teeth of the gear wheels, therefore further considerations are fully dedicated to the toothing of these geometrical properties [8-9]. Involute, more precisely 'the involute of a circle', is defined as the trajectory of a particular point of a straight line rolling without slipping around the circle, called basic or central. In the theory of gear wheels, the circle from which the involute originates is called a base circle, and its diameter is denoted by d_b (***Figure 9***).

The transition curve at the tooth root is manufactured by the cutting edge of a machining tool. As a result of this machining process, the transition curve, with respect to its geometry, is an equidistant

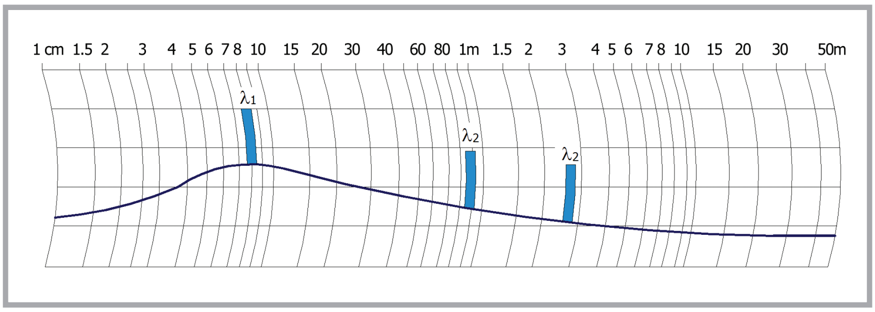

Figure 6. *Spectrogram for woollen worsted yarn of 36 tex linear mass, where the local amplification of amplitudes are depicted, illustrating the damage to driving elements of the ring spinner Fiomax 2000.*

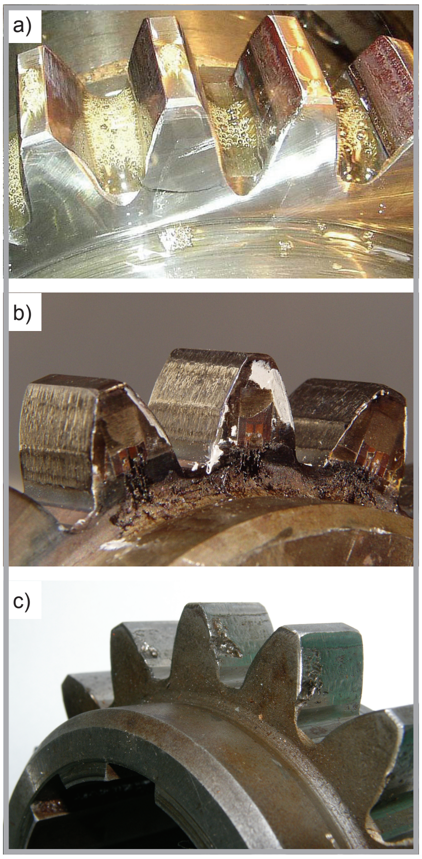

Figure 7. *Exemplary damage to teeth: a) fatigue crack, b) scoring, c) pitting.*

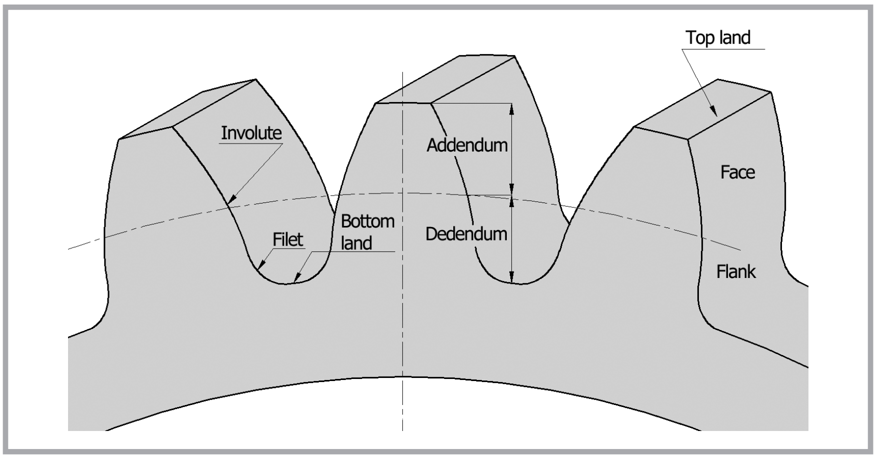

Figure 8. *Teeth shape of a typical gear belonging to the stretching apparatus sub-system.*

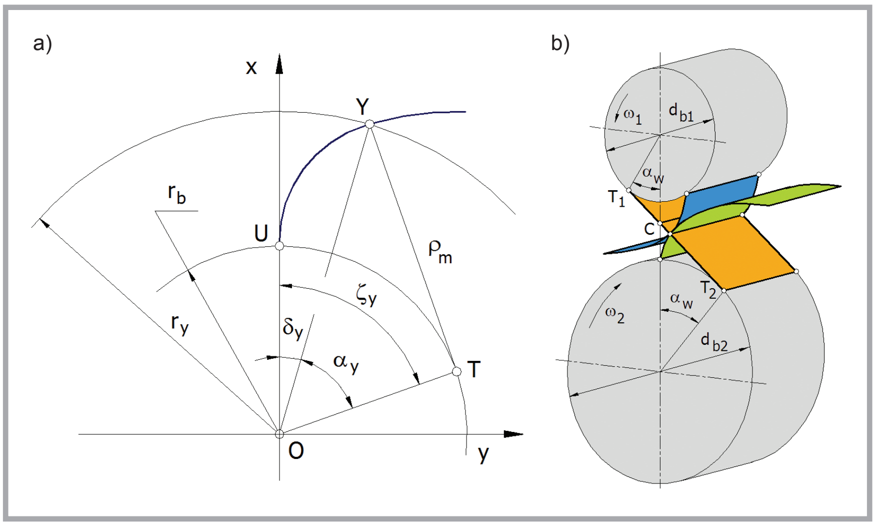

Figure 9. *a) Involute relevant to a particular circle, b) co-operation of two involute surfaces of engaged cylindrical gear wheels [8].*

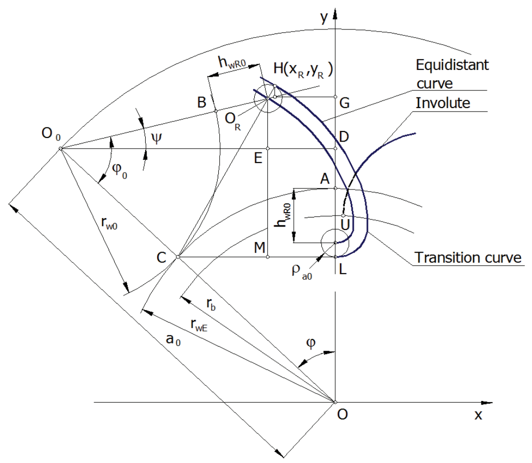

Figure 10. *Transition curve at the tooth foot for the stretching apparatus of the ring spinner [8].*

trajectory of the point of origin of the radius of the cutting edge of a particular tool (***Figure 10***) [8].

Figure 10. Transition curve at the tooth foot for the stretching apparatus of the ring spinner [8] ***Equations (1)*** and ***(2)*** where: a_0 – nominal distance between the axes $(a_0 = r_{wE} + r_{w0})$, r_{w0} – roll radius of the machining tool, $r_{R0} = r_{w0} + h_{wR0}$, $i_0 = r_{wE} / r_{w0}$.

The geometrical characteristics of gear wheels applied in the stretching drive system are relatively complex. However, it enables smooth operation of the whole drive system of the ring spinner. In ***Figure 11***, an exemplary engagement of two cylindrical gear wheels is presented – as a front view.

A properly designed gear should not cause any disturbances in the operation of the whole mechanism. In the case of ring spinner machinery, there are several gears in a row passing drive to the versatile subsystems. Therefore damage to one link in the chain causes defects in the whole drive systems.

Disturbances in the motion of the spinner driving system can occur in the following circumstances and due to the following reasons:

- errors during the manufacturing (machining) of the toothing of gear wheels – leading to the cutting of teeth tips or undercutting the near tooth foot,
- co-operation of two scored geared wheels – causing eventual blockage of gear teeth. In the case of high torque, the phenomenon leads to fatigue breakage due to crack propagation along the side surface of a particular tooth of a particular gear wheel,
- improper service and/or maintenance works – which disenables the proper (radial) position of the pinion.

Influence of teeth maufacturing deviations on the operation of the gear based on the ring spinner Fiomax 2000

Increasing demands related to the strength, durability and reliability of gears, with simultaneous diminishing of their dimensions and increasing of passing loadings, yield the necessity of application of more and more effective methods of calculations and analysis as well as experimental investigations [12-14]. Diagnostics of the drive systems of machines and devices consisting in investigations of the influence of the correction of the teeth shape of cylindrical gear wheels on their durability should be based not only on proper measurements but also on modelling and effective calculation methods. Numerical calculations can be performed e.g. by means of the finite element method (FEM) or (BEM) i.e. boundary element method.

$$x_R = -a_0 sin\varphi + r_{R0} sin(1+i_0)\varphi + \frac{[r_{R0} sin(1+i_0)\varphi - r_{w0} sin\varphi]\cdot\rho_{a0}}{\sqrt{r_{w0}^2 + r_{R0}^2 - 2r_{w0}\cdot r_{R0} cos\, i_0\varphi}} \qquad (1)$$

$$y_R = a_0 cos\varphi - r_{R0} cos(1+i_0)\varphi + \frac{[r_{w0} cos\varphi - r_{R0} cos(1+i_0)\varphi]\cdot\rho_{a0}}{\sqrt{r_{w0}^2 + r_{R0}^2 - 2r_{w0}\cdot r_{R0} cos\, i_0\varphi}} \qquad (2)$$

Equations (1) *and* ***(2).***

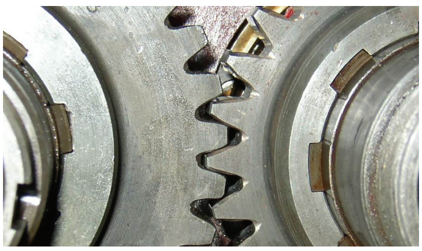

Figure 11. *Engagement of two gear cylindrical wheels.*

The boundary element method [1-3, 6] has been more and more popular recently in solving versatile engineer problems, especially for the task of analysis of static and dynamic contact. Therefore this method is especially suitable for the analysis of contact problems – due to the following reasons:

- contact phenomena related to teeth flanks i.e. in general, boundaries of the bodies are considered – however, in fact, profile lines are modelled in the method (cylindrical gear wheels),
- due to discretisation of the teeth boundaries only, diminishing of the size of the problem considered is received,
- possibility of direct consideration of boundary forces and displacements in the zone of contact,
- lower number of elements resulting from the division of the boundary (in comparison to FEM), which simultaneously means that the time of the performance of calculations is shorter.

Aiming at the solving of the problem of co-operation of cylindrical gear wheels for the gear incorporated in the stretching apparatus of the spinner Fiomax 2000, BEASY commercial software was utilized. The task consistedin, for example, determination of the maximal values of loadings. The solver of the BEASY program is effective and flexible. The program itself is a comprehensive system which enables the solving of versatile complex problems e.g. elastic, thermal, fatigue and cracking. Moreover the method of boundary elements can be applied for problems of corrosion, vibration, fluid mechanics, acoustics and some other problems [4-5].

For the purpose of analysis of modified teeth profiles, it was necessary to write our own so called preprocessor allowing the generation of arbitrary teeth profiles.

The task of this preprocessor is the precise numerical defining of shapes of teeth profiles based on a series of set (assumed) geometrical parameters. Namely the surface of the gear wheel was divided into the following fragments:

- involute segment of teeth flanks,
- tooth transition surface (undercut),
- arcs of circles of teeth roots,
- arcs of circles of teeth tips (edges of top lands),
- lines closing the profile into a continuous line.

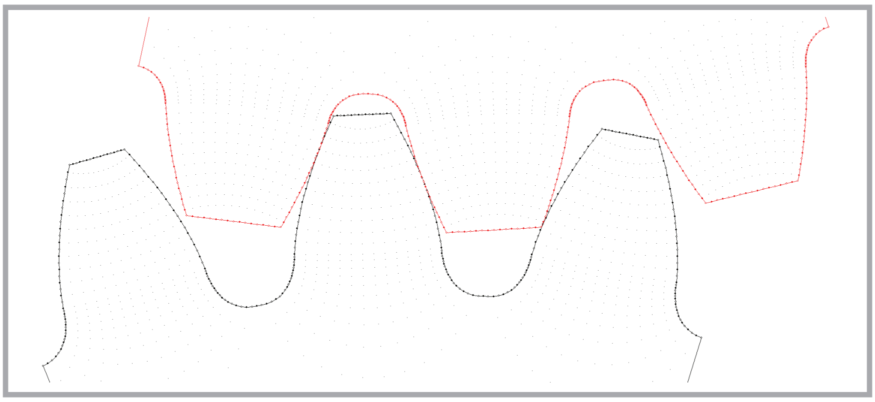

***Figure 12.** Exemplary engagement of a pair of gear wheels of the stretching apparatus – the flank tooth surface is divided in boundary elements.*

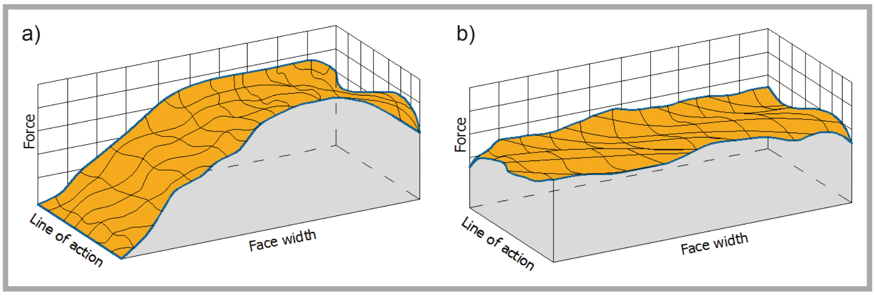

***Figure 13.** a) loading distribution in the case of the wheel manufactured without modification, b) loading distribution in the case of toothing where deviations are taken into account [8].*

Aiming at the elimination of the influence of the fixing manner on calculation results, the model comprises a particular tooth together with two neighbouring ones (***Figure 12***). After generation of a circle, the pre-processor divides the boundary into elements, each one with particular nodes. These are points at which values of the variables considered are calculated e.g.: coordinates of the boundary line, displacements, boundary forces etc. In the BEASY system, in solving planar problems (2D) and axial-symetric ones, three-node elements (of rectilinear and/or parabolic shape) are utilised, due to the assumption that the boundary shape (outline) is always modelled by means of three-node elements. During displacement, boundary forces and boundary stresses can be modelled by means of constant elements (then the medium node is used), linear elements (two outermost nodes are used) and parabolic elements (all three nodes are used). All nodes i.e. boundary points and internal points are numbered.

In ***Figure 13.a***, there is an exemplary distribution of loading in a case where the deviation was not entered. In ***Figure 13.b*** there is a scheme of loading distribution in a case where the deviations were entered.

As can be seen in ***Figure 13***, in the case of a wheel where appropriate deviations were applied, the loading distribution on the tooth flank is more even. Due to this, we can ensure the improvement of operating parameters of the stretching apparatus of the spinner.

For determination of the influence of the manufacturing deviation of the teeth of gear wheels on their cooperation (in the drive system), it was necessary to compare the cooperation parameters of the gear of the spinner Fiomax 2000 i.e. with deviation and without deviations.

Analyses were performed for the error of pressure angle and that of pitch. In calculations, it was assumed that the error is related only to the drive wheel and that the cooperating wheel is manufactured perfectly. As a matter of fact, this is a simplification of the problem, however it makes that its description is easier because the sum of errors (for a pair of wheels) is extremely important in the considerations.

Aiming at determination of the influence of geometrical deviations in the drive system of the spinner Fiomax 2000, the following characteristic values were calculated:

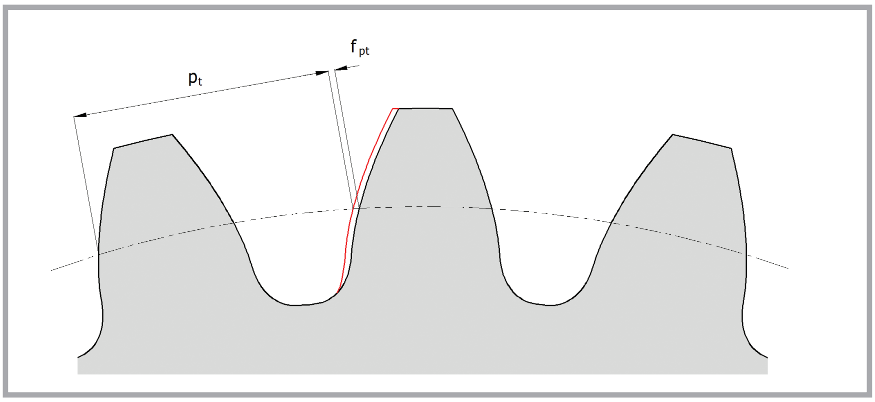

Figure 14. *Chordal pitch with deviation.*

- maximal contact stresses (for the engagement of teeth, co-operation of teeth flanks of a pair of gear wheels),
- maximal bending stress at the tooth root,
- rotational velocity of the drive wheel,
- stiffness of toothing,

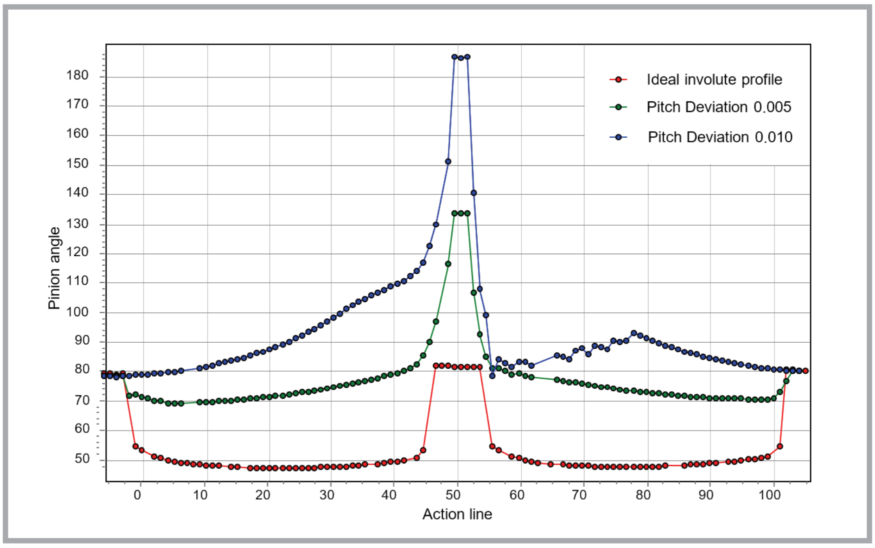

Figure 15. *Dependence of rotational angle of the pinion on pitch deviation.*

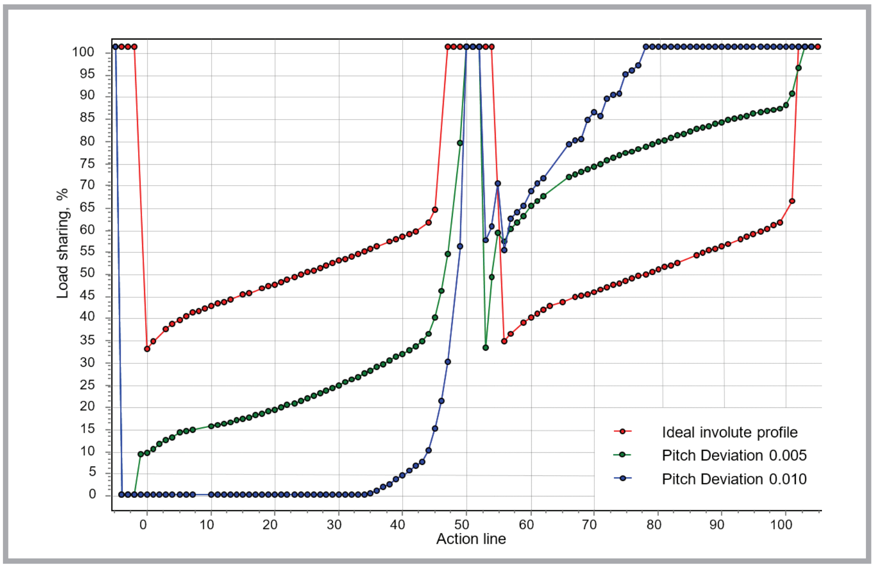

Figure 16. *Division of loadings when the pitch deviation has been taken into account.*

- load sharing between the engaged pairs of teeth (co-operating teeth).

The line of action is a geometric place of all contact points of teeth in a mesh (in engagement). In reality, gear cooperation does not have a point nature; on the contrary, engagement takes place on a particular (tooth flank) section. Therefore, in practice, the geometrical region of all contact points is a neighboring area around the theoretical line of action.

Influence of pitch deviation of cylindrical gear wheel of the gear belonging to the stretching apparatus mechanism

Pitch is a basic parameter of a cylindrical gear wheel, which is formally defined as the distance between adequate teeth flanks measured along the arc of the pitch circle (see ***Figure 14***).

In industrial practice, the unevenness of consecutive pitches causes an increase in dynamical loadings, especially in the case of high tangent velocities, as well as an increase in noise emitted by the gear.

In the analysis performed, the following pitch deviations were assumed: 5 μm, 10 μm and 15 μm, which correspond to the preciseness classes, respectively: $5(f_{pt}=\pm 5\mu m)$, $7(f_{pt}=\pm 10\mu m)$ i $9(f_{pt}=\pm 20\mu m)$ according to Polish Standard PN-ISO 1328-1.

Pitch deviations have an influence (especially) on the rotational angle of the drive wheel (pinion) $\Delta\phi$, and thus on the stiffness of toothing (mesh stiffness), which depends on the geometrical parameters of the gear wheels and physical properties of their materials. The international standard ISO 6336 [15] defines mesh stiffness as the ratio between the normal force and displacement for a pair of engaged teeth (where the tooth width is equal to 1). The displacement is determined in the direction tangent to the tooth profile, in the face cross-section.

It should be underlined that the increase in pitch deviation is related not only to an increase in the rotational angle$\Delta\phi$ but also to a special phenomenon for high deviations – the second maximum arises (in the course), corresponding to the final moment of cooperation (engagement) of the central teeth pair (i.e. a tooth of the pinion is taken back via the value of pitch deviation). This phenomenon will occur when the pitch deviation is lower than the deflection of the pair of loaded teeth i.e. for high deviations and low loadings.

Influence of the deviation of the pressure angle of a cylindrical tooth inside the stretching apparatus of the spinner Fiomax 2000

The maximal bending stresses (tangent to the transition line – according to the standard PN-ISO 6336) occur at the tooth root on the bending side in cases where the inter-teeth force is pointed towards the outer point of the one-pair engagement *(external pressure (engagement) point is define as the point in which ends cooperation of one teeth pair and initiates cooperation of two teeth pairs, simultaneously)* D (***Figure 17***). Values of the radiuses of the profile curves can be calculated by means of the following formula [8]:

$$\rho_{2B2} = \sqrt{r_{a2}^2 - r_{b2}^2} - p_b \qquad (3)$$

$$\rho_{1B2} = a \cdot sin\alpha_w - \rho_{2B2} \qquad (4)$$

where:
r_{a2} – radius of gear wheel addendum circle, r_{b2} – radius of base circle, a – current distance between the axes, α_w – operating pressure angle, p_b – transverse contact ratio (on line of action).

The points of contact of teeth during rotation are placed along the particular line (action line), where special intervals (line segments) can be distinguished. Angle α between the interval mentioned and a line tangent to the pitch circles at their common point is called a pressure angle, being simultaneously an involute parameter. The so called nominal profile, commonly utilised in machine design and recognised world-wide, is equal to α = 20°.

In the analyses performed, the following deviations of the pressure angle were assumed: ±2' and +4', which are inside the precision classes: 5($f_{H\alpha} = \pm 3{,}3 \mu m$) and 7($f_{H\alpha} = \pm 6{,}5 \mu m$) according to the standard (incorporated into Polish legal system) PN-ISO 1328-1.

The deviation of the pressure angle causes an identifiable change in all parameters of the toothing. It is also obvious that it has an influence on the pinion rotational angle $\Delta\phi$, which, in turn, has a significant influence on the dynamic force acting on the gear (***Figure 18***). Analysing the chart, it can be realised that each deviation of the pressure angle causes an increase in amplitude.

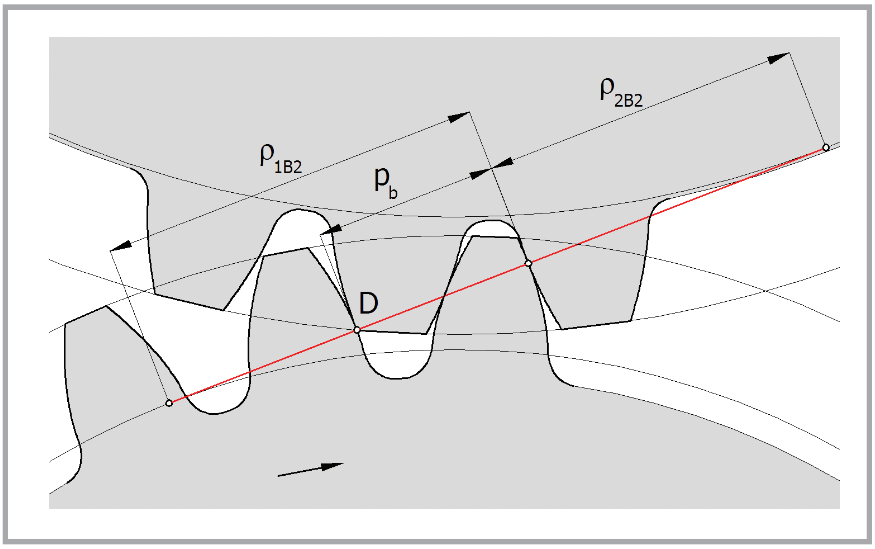

Figure 17. *Determination of the point of force application.*

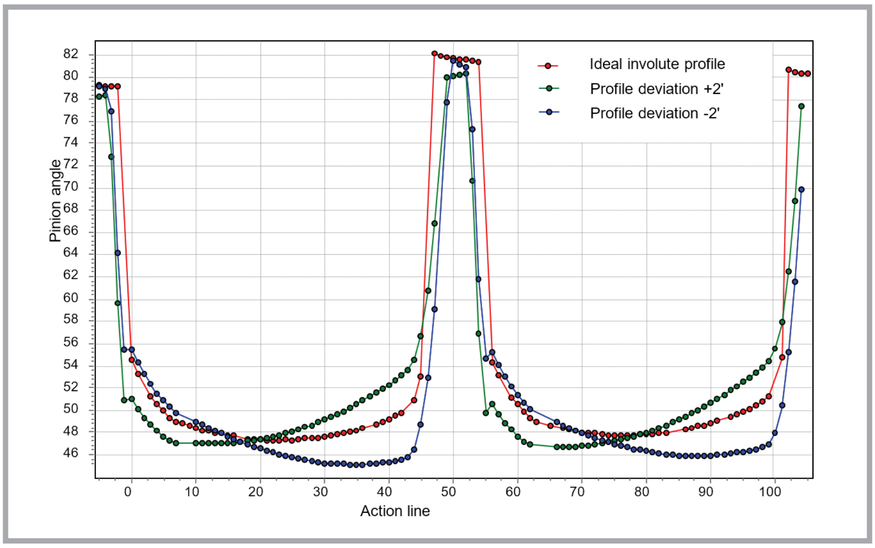

Figure 18. *Relationship between the pinion rotational angle and deviation of pressure angle.*

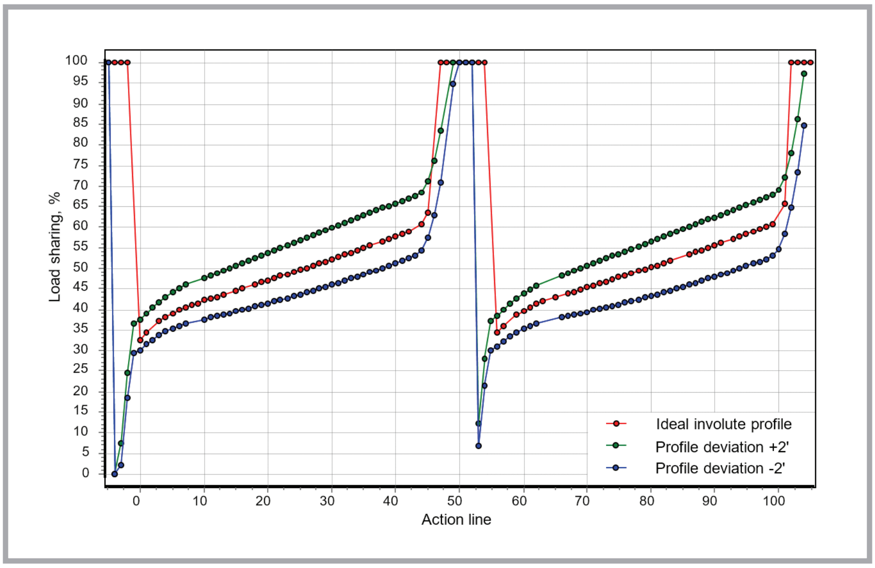

Figure 19. *Influence of pressure angle deviation on load sharing.*

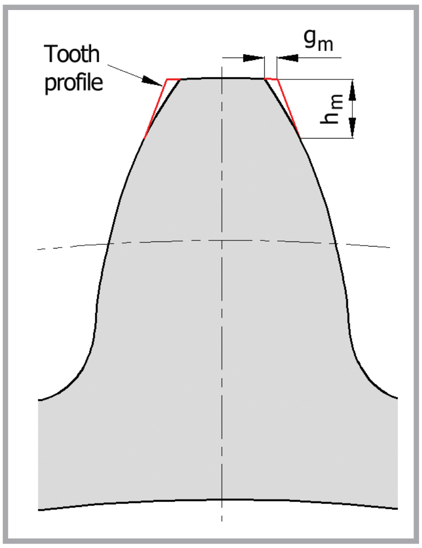

Figure 20. Modification of tooth profile at the gear tooth tip.

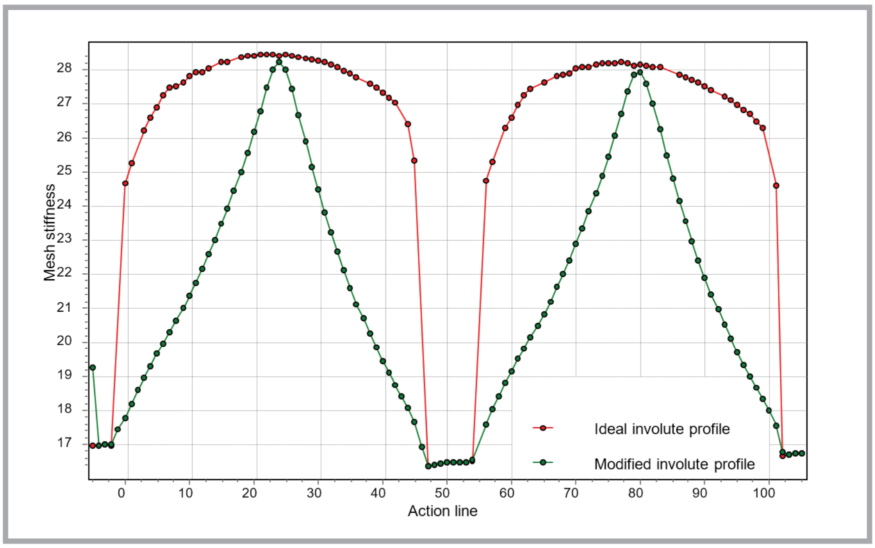

Figure 21. Course of mesh stiffness for standard and modified teeth.

It can be recognised that there is a noticeable influence of the angle deviation on load sharing between the pairs engaged (***Figure 19***). Based on the figure presented, one can see that the positive deviation of the pressure angle increases the load at the moment of getting meshed (entering into engagement).

Co-operation of modified profiles of cylindrical gear wheels of the stretching apparatus of the ring spinner

As a profile modification, we consider a purposeful change in the tooth cross-section in comparison to the theoretical one. The aim of the modification is the elimination of impact-type cooperation between particular pairs of teeth tips when they enter into engagement (in a mesh), due to bending of the cooperating teeth. The following measurements are taken: modifications of the tooth tip and root (in general, the tooth profile). Due to the fact that the tip of one wheel cooperates with the root of the second – irrespective of the type of profile modification – the same affect is achieved. The result consists in increasing clearance between the cooperating tooth tip and root. In the further part of the present paper, we perform an analysis of tip modification because its geometrical parameters are easier for description, and it allows to achieve the same goals as for modification of roots.

Modification of the tooth profile at the tooth tip can be described via two parameters: the height of the modification h_m and the depth of the modification g_m

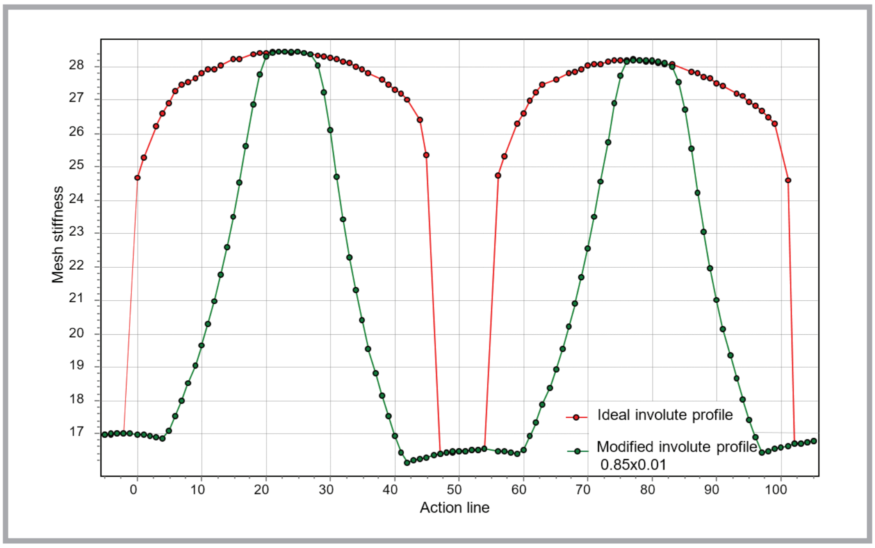

Figure 22. Course of mesh stiffness for improper modified teeth

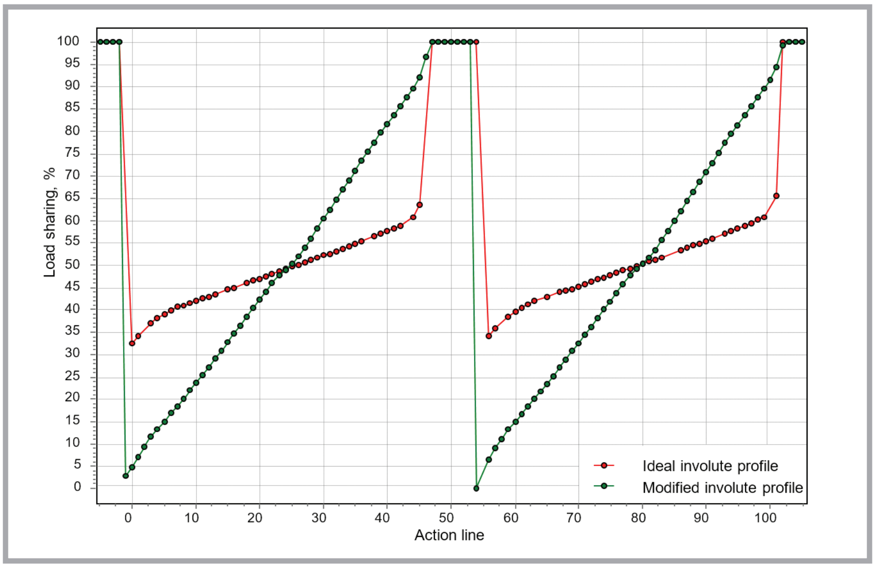

Figure 23. Load sharing between pairs of teeth in the case of the modified profile.

(***Figure 20***) [8, 16]. The height of the modification is chosen in such a manner that the remaining arc of the involute (non-modified part) should assure the proper transverse contact ratio (number of teeth in contact in a transverse plane) $\varepsilon_\alpha = 1$. It ensures the continuity of passing movement in the case of a lack of loading, which in fact does not exist. Due to an increase in loading, the bending of teeth takes place. In consequence, the modified arc of the tooth profile is subjected to engagement. Therefore the depth of modification g_m has to be established in such a manner that for the nominal loading of the gear there is a contact of teeth tips. Unfortunately there are no known precise analytical methods allowing the determination of the modification depth.

Application of the modification of teeth tips can essentially improve the operation of the gear, reducing the wear of teeth surfaces. However, the main goal of utilisation of modification is the improvement of dynamic properties of the gear and diminishing of noise emitted by it. In ***Figure 21***, we can see a comparison of courses of mesh stiffnesses for wheels with teeth of standard and modified profiles, respectively. The height of modification was equal to 1.02 mm and the depth just to 6 μm for the pinion and cooperating wheel.

In the chart, a noticeable improvement in the course of mesh stiffness can be observed, which converted from a rectangular shape into (nearly) a triangular one. The less steep course of stiffness variability results in the diminishing of dynamic forces occurring in the gear. However, it should be taken into account that such a correct course was obtained for the gear loading tested, for which the modification was selected (fitted). If the modification is not properly chosen, then its effect will be lower (***Figure 22***).

The proper choice of parameters of the modification can be easily confirmed by analysing the chart of the division of loadings between the co-operating pairs of teeth (***Figure 23***).

As can be seen, based upon the charts above, for the properly modified loaded tooth profile the loading increases linearly from zero up to the maximal value, and then diminishes linearly to zero.

Conclusions

Analysis of the co-operation between particular subsystems of the ring spinner and setting of the optimal operating parameters is a complex problem. Focusing attention even on the stretching apparatus itself, several complicated problems were recognised which are connected with maintenance of the drive system and passing of the torque phenomena. One factor consists in the fact that (due to loadings) the engaged teeth of the gear wheels as well as the housing and all elements passing forces are subjected to deformations, which, together with the manufacturing deviation and possible errors of assembly cause that the loading distribution is uneven. In consequence, these behaviours influence the operation of the ring spinner directly, automatically affecting the spinning process; in particular it causes a diminishing of the quality of the product obtained.

Due to vibrations of the masses of the geared wheels, additional inter-teeth forces are excited. The elastic teeth engaged act as spring, and therefore the vibrations are passed onto the whole machinery, and in the worst case it can cause, for example, breakage of yarn in the twisting-winding system.

The method proposed for improvement of the distribution of loadings and simultaneous improvement of operating parameters consists in the utilisation of the correction and modification of geometrical characteristics of gear wheels co-operating inside the stretching apparatus of the ring spinner.

References

1. Burczyński T. Metoda elementów brzegowych w mechanice. Wspomaganie komputerowe CAD – CAM, WNT, Warszawa 1995
2. Brebbia C A, Dominguez J. *Boundary elements. An introductory course*; Computational Mechanics Publications, Southampton 1992
3. Burczyński T. Metoda elementów brzegowych w wybranych zagadnieniach analizy i optymalizacji układów mechanicznych; Wydawnictwo Politechni Śląskiej, 1989.
4. Drewniak J and Rysiński J. Evaluation of fatigue life of cylindrical geared wheels. *Solid State Phenomena* 2013;199: 93-98.
5. Drewniak J and Rysiński J. Fatigue life and reliability of power engineering machines and their elements. *Energetyka* 2013; (107): 12, 9-16.
6. Drewniak J, Rysiński J and Praszkiewicz M. Analysis of fatigue life and dynamics of gear train by boundary element method. *Mechanik* 2013; 12: 1-8.
7. Jackowski T, Cyniak D and Czekalski J. Wpływ wybranych parametrów decydujących o jakości formowanych przędz. *Przegląd Włókienniczy + Technik Włókienniczy* 2006; (2): 53-57.
8. Jaśkiewicz Z and Wąsiewicz A. *Przekładnie walcowe cz.1*, WKiŁ, Warszawa, 1992.
9. Jaśkiewicz Z and Wąsiewicz A. *Przekładnie walcowe cz.2*, WKiŁ, Warszawa, 1995.
10. Rysiński J and Wróbel I. Diagnostics of machine parts by means of reverse engineering procedures. *Advances in Mechanical Engineering* 2015; 7(5):1–9. DOI: 10.1177/1687814015584543.
11. Rysiński J, Drobina R and Tomaszewski J. Probability of the Critical Length of a Fatigue Crack Occurring at the Tooth Foot of Cylindrical Geared Wheels of the Drive System of a Fiomax 2000 Ring Spinner. *Fibres and Textiles in Eastern Europe* 2017; 25, 1(121): 134-144. DOI: 10.5604/12303666.1227895.
12. Rysiński J and Sidzina M. Insitu diagnostic investigations of gear parameters with use of industrial automation installations and http protocol. *Measurement Automation Monitoring* 2012; 58, 11: 950-952.
13. Tomaszewski J and Rysiński J. Diagnostics of gears and compressors by means of advanced automatic system. *Acta Mechanica et Automatica* 2015; 9, 1: 19-22, DOI: 10.1515/ama-2015-0004.
14. Tomaszewski J and Rysiński J. The concept of vibro-acoustic symptohoms for diagnosis of fatigue cracks in gears. *Acta Mechanica Slovaca* 2005;9, 3-B: 193-202.
15. ISO 6336-1:2006. Calculation of load capacity of spur and helical gears. Part 1: Basic principles, introduction and general influence factors.
16. Wang Q and Zhang Y. A model for analyzing stiffness and stress in a helical gear pair with tooth profile errors. *Journal of Vibration and Control* 1-18. DOI: 10.1177/1077546315576828.
17. FIOMAX 2000 – Technical specification
18. USTER TESTER 6 – The Total Testing Center™ – brochure.
19. Lewandowski S, Drobina R and Józkowicz I. Comparative Analysis of the Ring Spinning Process, Both Classic and Compact: Theoretical Reflections. Part I: Elaboration of the Statistical Model Based on Multiple Regression. *Fibers and Textiles in Eastern Europe* 2010; 18, 4(81): 20-24.
20. Józkowicz I, Drobina R and Lewandowski S. Comparative analysis of the ring spinning process, both classic and compact: Part II: The Verification of Created Models. *Fibres and Textiles in Eastern Europe* 2010; 18, 5(82): 28-34.

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