STUDY ON DYNAMICS OF CATERPILLAR TRACK BODIES

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Abstract: In this paper, we report results of the study we carried out to determine the amount of contact stress developing in various operating states between a roller and the caterpillar track body by the special machine of surface mining.

Key-words: Bucket-wheel excavator, bucket, tooth, energy, lignite.

1. INTRODUCTION

Due to the high mass of machinery used and the soil conditions prevailing in opencast mining, change of place of these machines can only be effected by caterpillar mechanisms. (Figure 1) With vehicles of this type, gravitational power and other operational forces of the excavator are transmitted to the caterpillar tracks by moving rollers. High weight of the machines and limitations on the number of rollers in caterpillar tracks result in build-up of a significant surface tension at the stressed surfaces of the moving rollers that make covering the rolling surface with some flexible material impossible. Thus, power transmission between the moving rollers and the caterpillar track bodies takes place at a metal-to-metal contact.

During manufacturing, surfaces of the moving rollers exposed to high loads are subjected to heat treatment – usually temper hardening –, in order to be capable of bearing high surface stresses. In this paper, we report results of the study we carried out to determine the amount of contact stress developing in various operating states (linear forward moving, rolling from one caterpillar track element to another) between a roller and the caterpillar track body providing the rolling surface.

The 3D model of the caterpillar track body and the roller was set up for both states. The state corresponding to linear moving is shown in Figure 2. Applying the fi-

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nite element (FE) calculation to this model we studied forces developing in the particular elements.

Fig. 1. View of the caterpillar track body and the roller

Fig. 2. 3D model of the caterpillar track body-roller system

2. PRINCIPLES OF CALCULATING CONTACT STRESS

Scientific basis of the subject was established by H. Hertz, who, in the late 1800's, originally studied *elastic* deformations arising in elastic bodies under the influence of stress at a point or along a line. Later, the stress state in the environment of the contact was mostly studied by A.N. Dinnyik and N.M. Beljajev who thereby made possible a theoretically sound calculation of the contact strength. Relationships coming from their theory can be found at several places in the technical literature.

According to this theory, curved surfaces of the connecting bodies are initially touching each other in a single point, then they are elastically deformed as a result of the force *F*. Parameters of the contact interface thus formed – which is, being the intersection of surfaces of different curvature, an ellipse – are the half small axis *b* and the half big axes *a* with a distributed system of acting forces.

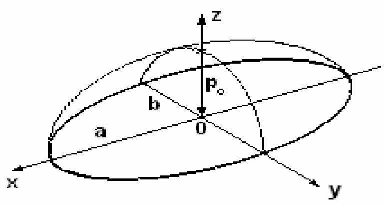

Fig. 3. Parameters of the contact interface

Maximum of the distributed system of forces, p_0, is at the center of the ellipse, ceasing to zero as approaching the border line. (See Figure 3) From among the main stress values specifying stress state of the contacting bodies' materials, the highest one is that falling in direction of the z axis (σ_z).

Relationship necessary for its calculation is given here according to ref [1], without a proof.

$$\sigma_z = p_0 \frac{ab}{\sqrt{a^2 + z^2}\sqrt{b^2 + z^2}} \tag{1}$$

If $z = 0$, i.e. at the contact point, its value equals to p_0 and gradually decreases towards the interior of the material.

3. MATERIAL CHARACTERISTICS USED AND SCOPE OF THE CALCULATIONS

In our calculations – in lack of more precise information – we considered the base as well as the roller as elements built up of homogeneous, isotropic material with elastic parameters characteristic of an average kind of steel. Finite element program has built up the 3D model from 10 mm size, tetrahedron-shaped elements. A finer resolution, due to the excessive size of the bodies, would cause a prohibitively long running time on the average performance computer we used. *Resulting stress values are average values calculated for such an element.* Thus, in spite of the point-like nature of the contact, the *maximum stress value* obtained in the run is an *average stress* of the element in the state of highest stress.

We would like to emphasize that our calculations are valid for an *elastic* deformation of the material. As it will be seen later, calculated stress values are greater than 900 N/mm^2 in the majority of cases studied. According to data provided by the manufacturer of the excavator, hardness of the rolling surfaces' material after temper hardening should reach this value. Thus, at stresses exceeding this value the material already suffers permanent deformation and *the stress values themselves only inform about the extent to which the limit was exceeded, but they by no means represent the true value of stress in the material.*

4. LINEAR ROLLING ON A SINGLE BASE ELEMENT IN HORIZONTAL PLANE OF MOVEMENT

4.1. Determination of the maximum stress value by theoretical calculation.

This is the most general case. Symmetry plane of the roller is vertical and its line of intersection with the caterpillar track body is the trace of the rolling at the same time. This arrangement is shown in Figure 4. Data relevant to rollers of the excavator studied and to elements of the caterpillar track body: R_{11} = 275mm, R_{12} = 250 mm, R_{21} = ∞, R_{22} = 275 mm. Further parameters of the machine: total weight: m = 970 t; number of rollers n = 32.

Assuming an even weight distribution among the rollers, we calculate with a force of F = 297 kN for each roller for the machine studied.

Results of the calculations are given in Tables 1, 2, and 3. Interpretation of values present in each row is the following: F - force experienced by a single roller, kN; p_0 - maximum value of the contact strength, N/mm^2; p_{avg} - average value of the contact strength, N/mm^2; T - area of contact, mm^2.

Calculations were performed in the 200 kN ≤ F ≥ 400 kN force range, while changing diameter of the roller ($D = 2R_{11}$) and internal curvature of the caterpillar track body (R_{22}). These results can be seen in Tables 1 to 3.

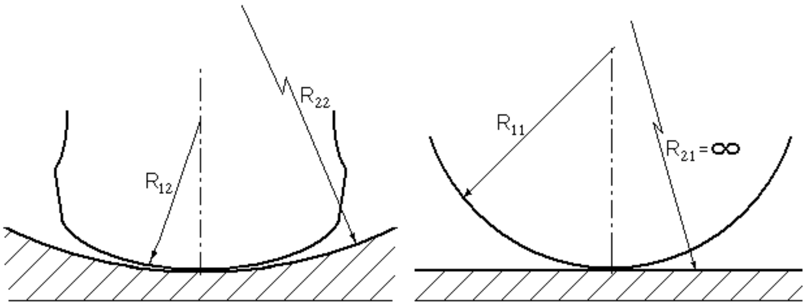

Fig. 4. Curvature radii of the contact zone. Cross section(left) longitudinal profile(right)

Table 1. **Roller diameter: *D* = 0.5 m**

R_{11} = 0.25 m; R_{12} = 0.25 m										
R_{22}	R_{22} = 0.25 m			R_{22} = 0.275 m			R_{22} = 0.3 m			
F	F = 200	F = 297	F = 400	F = 200	F = 297	F = 400	F = 200	F = 297	F = 400	kN
p_0	543	619	684	1044	1191	1315	1203	1373	1516	N/mm^2
p_{avg}	362	413	456	696	794	877	802	915	1010	N/mm^2
T	552	719	877	304	374	456	249	324	395	mm^2

Table 2. **Roller diameter: D = 0.55 m**

R_{11} = 0.275 m; R_{12} = 0.25 m										
R_{22}	R_{22} = 0.25 m			R_{22} = 0.275 m			R_{22} = 0.3 m			
F	F = 200	F = 297	F = 400	F = 200	F = 297	F = 400	F = 200	F = 297	F = 400	kN
p_0	509	581	641	998	**1140**	1258	1152	1315	1451	N/mm^2
p_{avg}	339	387	428	666	**760**	839	768	877	968	N/mm^2
T	588	767	934	300	**391**	476	260	339	413	mm^2

In Table 2, results obtained for the studied machine are set in bold. As it can be seen the p_0=1140N/mm^2 maximum intensity exceeds value of the permissible stress, $б_z$=900 N/mm^2. As it can be inferred from content of the cell in the same position of Table 3, by increasing the roller diameter only a slight improvement can be achieved (theoretical case). By increasing the diameter from D = 550mm to D = 600mm the decrease in insignificant. (1140 N/mm^2 →1122 N/mm^2) Reduction of the diameter can normally be conceived only to a slight extent, as space available does not allow greater size increase.

Table 3. **Roller diameter: D = 0.6 m**

R_{11} = 0.3 m; R_{12} = 0.25 m										
R_{22}	R_{22} = 0.25 m			R_{22} = 0.275 m			R_{22} = 0.3 m			
F	F = 200	F = 297	F = 400	F = 200	F = 297	F = 400	F = 200	F = 297	F = 400	kN
p_0	480	548	605	983	1122	1238	1117	1275	1407	N/mm^2
p_{avg}	320	366	404	655	748	825	745	850	938	N/mm^2
T	624	812	990	305	397	484	268	349	426	mm^2

Bringing radii of curvature closer to each other seems to be a better solution, furthermore, a feasible one. Reducing radius of curvature of the caterpillar track body's internal arc (R_{22}) results in a significant decrease to the maximum value of the contact stress. In the extreme case R_{22} can be assumed to be equal to R_{12}, the radius of rounding of the roller. In this case the initial contact can not be regarded as point-like, it takes place on an r = 250 mm diameter arc. Thus the higher surface permitting the deformation to remain in the elastic range is already created at a lower value of deformation. Maximum value of stress at these parameters: p_0 = 581 N/mm^2. (See shaded cells of Table 2) It is to be noted, that it is practical to keep the $R_{22} > R_{12}$ relationship valid, in order to have the rollers remain in a straight line while rolling.

Results represented by the number series shown in the Tables will be clearer if we also present them in form of Diagrams. These are shown in Diagrams 1 and 2.

Another possibility to bring radii of curvatures closer to each other is to increase rounding radius of the roller (R_{12}), while keeping diameter of the roller (R22) and curvature of the caterpillar track body (R12) unchanged. This version has only been calculated for the R_{12} = R_{22} = 275 mm case. The results are given in Table 4. Thus, column of bold numbers corresponds to a hypothetic state in which rollers of the machine studied would be manufactured with a rounding of R_{12} = 275 mm, while keeping the diameter unchanged. It is to be noted here, too, that it is practical to keep the $R_{22} > R_{12}$ relationship valid and select R_{12} to be slightly less than 275 mm, in order to have the rollers remain in a straight line while rolling. In Diagram 2 results belonging to variables of the studied machine are shown together with the two approaches for the reduction possibility.

Tabel 4. **Roller diameter: D = 0.55 m**

R	$R_{12} = R_{22}$ = 0.275 m			
F	F = 200	F = 297	F = 400	kN
p_0	509	**581**	641	N/mm^2
p_{avg}	339	**387**	428	N/mm^2
T	588	**767**	934	mm^2

As also revealed by the number series, the latter approaches gave equivalent results. This is why the corresponding two curves are congruent. As regards their effect, they reduce value of the contact intensity to approx. half of its original size.

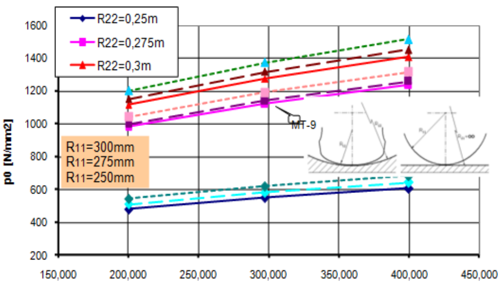

Diagram 2 The average contact pressure variation wit the load for different values of R22 radius

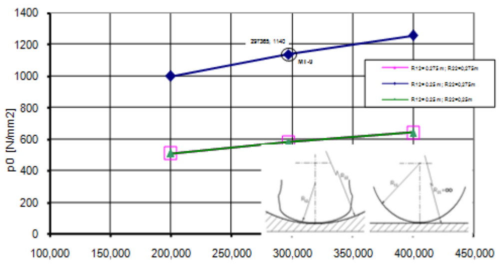

Diagram 1 The average contact pressure variation wit the load for different values of R12 and R22 radii

To represent stresses developing in the vicinity of the contact point, it is necessary to break the model up. Elements thus obtained can be shown in Figure 5.

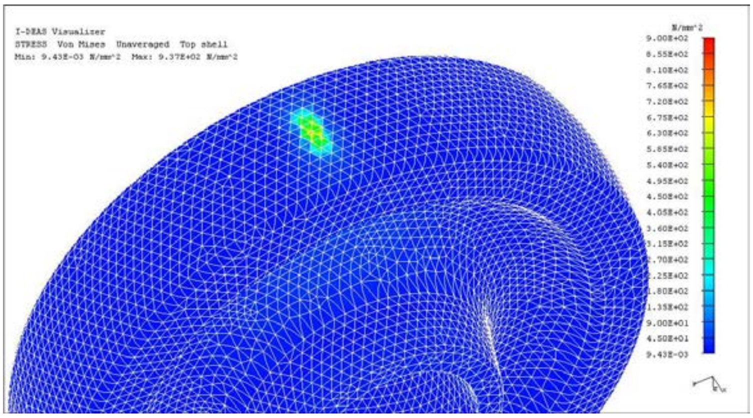

Fig. 5. FEM discretization of the roller

From the results it can be seen that stresses developing in materials of both the moving roller and the caterpillar track body exceed the 900 N/mm^2 value, thus both suffer a plastic deformation.

Comparing results obtained by the theoretical calculation vs. the VE analysis we can see that they are close to each other. Maximum value obtained by the calculation (p_0 = 1140 N/mm^2) is higher than that provided by the VE calculation (σ_z = 937 N/mm^2). Presumably, the cause is that – while theoretical calculation gives the maximal intensity of the distributed force system – the VE calculation provides the stress value developed in the material. Of course, stress value at the surface (z=0) should equal to the intensity at the same location, i.e. $p_0 = \sigma_z(x = 0; y = 0; z = 0)$. (They have opposite directions!) As it has already been indicated, however, the VE calculation was performed using a 10mm mesh and the result is an average stress valid for the cell exhibiting the highest stress.

5. ROLL-OVER FROM ONE BASE ELEMENT TO THE OTHER IN HORIZONTAL PLANE OF MOVEMENT

5.1. Determination of the maximum stress by theoretical calculation.

Figure 6 shows relative locations of the roller and the bases, in an intermediate position of the roll-over. As forked end of one base surrounds narrowed end of the other, the plane of rolling formed by the two bases has discontinuity in the vicinity of the pin. At the rolling-over, for a moment, roller is entering a “hole” formed in between the two bases. For this reason, during the roll-over process, with respect to the base, curvature conditions of the contacting surfaces compared to rolling on the same base change. In this situation, roller is supported by both bases simultaneously, consequently about half of the weight is exerted in each contact points. For calculations of the surface loads, dimensions providing the higher loading state have been taken into account. Accordingly, the rollers' characteristic radii: R_{11} = 275 mm; R_{12} = 250 mm. R_{22} = 275 mm remains at the base, however, R_{21} is changed significantly – according to the production documentation R_{21} = 20 mm.

Even without a calculation, it can be guessed that the surface formed during contact is highly narrow, in such case of a linear kind. This leads to a significant increase – one might as well say a jump-up – of the calculated amount of contact stress. Calculated values characteristic to the studied machine are shown in Table 5. Due to the high curvature of the contact environment, values of the developing stress exceed 3000 N/mm^2, a very high quantity. The numerical value itself has hardly a significance any more. However, it calls attention to the fact that contacting surfaces suffer significant permanent deformation at each roll-over.

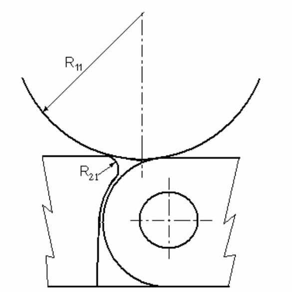

Fig.6. Relative locations of the roller and the bases, in an intermediate position

Table 5. **Roller diameter: *D* = 0.55 m**

	R_{12} = 20mm	
F	F = 297	kN
p_0	3119	N/mm^2
p_{avg}	2079	N/mm^2
T	71	mm^2

Results of the calculations harmonize with our information on the ways of manufacture of excavators, namely, manufacturers seek a solution different from the above, just to reduce the significant permanent deformations caused by the high stress peaks.

5.2. Study of the stress state by calculation of FE.

To study this state a model shown in Figure 7 was designed. As side guiding tubes do not play any role at formation of the contact stress, they have been omitted. Relevant distances and round-off values are otherwise values to be found in documentation of the studied machine.

Stress state obtained in this loading situation is illustrated in Figure 8. To represent stresses developing in the vicinity of the contact point, here it is also necessary to break the model up.

As it also follows from the approach, of the two contact points of the roller the higher value of stress (σ_z = 1910 N/mm^2) is obtained at the site with lower radius.

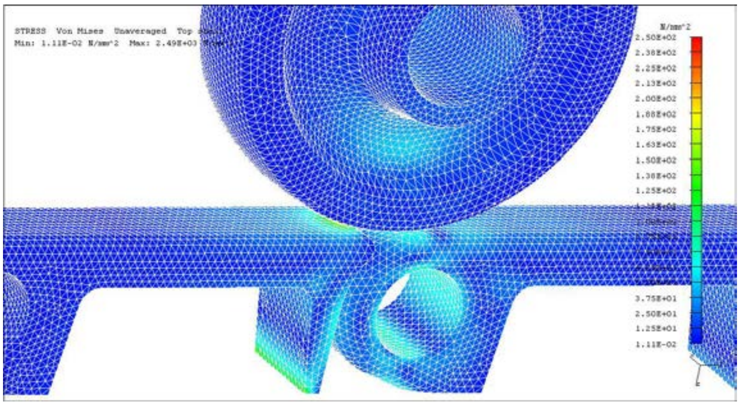

Fig. 7. 3 D model of the situation depicted in fig. 6

Similarly, to the linear rolling case, theoretical calculation gave higher surface intensity than the stress value obtained from VE analysis, in this case, too. By refining the grid, result of the model calculation would approach the value obtained in the theoretical calculation assuming a pointwise contact.

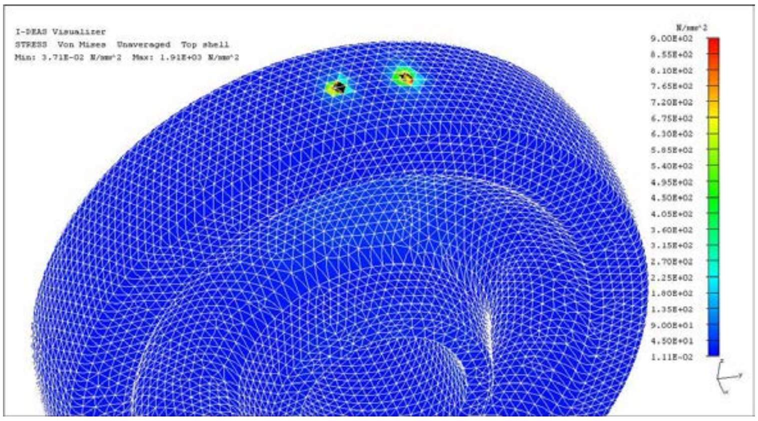

Fig. 8. Stress state resulted from FEM modeling showing the 2 stress peak zones

6. CONCLUSION

The results of the study carried out in order to determine the values of contact stress developed in various operating states between a roller and the caterpillar track body of a special machine of surface mining are reported.

The displacement of these machines can only be effected by caterpillar mechanisms, in which gravitational and otheroperational forces of the excavator are transmitted to the caterpillar tracks by moving rollers.

High weight of the machines and limitations on the number of rollers incaterpillar tracks result in occurrence of a significant surface contact stresses at the surfaces of the moving rollers, that make covering the rolling surface with some flexible material impossible, and the force transmission between the moving rollers and thecaterpillar track bodies takes place at a metal-to-metal contact.

Based on a 3D model of the roller- caterpillar track body realized in a CAD software, and considering the baseas well as the roller as elements built up of homogeneous, isotropic material with elastic parameters characteristic of an average kind of steel, we determined using the classical Hertz theory the stresses and FEM analysis in two operating cases.

Linear rolling on a single base element in horizontal plane of movement and roll-over from one base element to the other in horizontal plane ofMovement

Following the calculi, it results a maximum intensity of stress which exceeds the value of the permissible stress. By increasing the roller diameter only a slight improvement can be achieved (theoretical case).

Bringing radii of curvature closer to each other seems to be a better solution,furthermore, a feasible one. Reducing radius of curvature of the caterpillar track body'sinternal arc results in a significant decrease to the maximum value of the contactstress. Thus the higher contact surface allows to the deformation to remain in the elastic range.

Another possibility to bring radii of curvatures closer to each other is to increaserounding radius of the roller, while keeping diameter of the roller andcurvature of the caterpillar track body unchanged.

The next step was to perform a FEM analysis. When generating the finiteelement model, we used simplified caterpillar track elements and only a model of thebody.

From the results it can be seen that stresses developing in materials of both the movingroller and the caterpillar track body exceed the elastic limit value, thus both suffer aplastic deformation.

Comparing results obtained by the theoretical calculation vs. the FEM analysis wecan see that they are close to each other. Maximum value obtained by the calculationis higher than that provided by the FEM calculation.

Presumably, the cause is that – while theoretical calculation gives the maximal intensityof the distributed force system – the FEM calculation provides the average stress value developedin the material.

For the second situation , at the rolling-over, for a moment, roller is entering a "hole" formed betweenthe two bases. For this reason, during the roll-over process, with respect to the base,curvature conditions of the contacting surfaces compared to rolling on the same basechange. In this situation, roller is supported by both bases simultaneously, consequentlyabout half of the weight is exerted in each contact points. For calculations of the surface by theoretic method, loads, dimensions providing the higher loading state have been taken into account.

This leads to a significant increase – onemight as well say a jump-up – of the calculated value of contact stress. Calculatedvalues of the developed stress exceed 3000N/mm2, a very high value . The numerical value itself has hardly a significance anymore. However, it calls attention to the fact that contacting surfaces suffer significantpermanent deformation at each roll-over.

The FEM analysis for this second situation provideshigher value of stress (σz = 1910N/mm2) which is obtained at the site with lower radius.

Similarly to the linear rolling case, theoretical calculation gave higher surface intensitythan the stress value obtained from FEM analysis, in this case, too. By refining the grid,result of the model calculation would approach the value obtained in the theoretical calculation assuming a pointwise contact.

Based on results, we seek together with the manufacturer another constructive solution of the articulation of base elements, in view to increase the radii of contact zones.

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