ON INCREASE OF AXIAL LOAD

Korolkov, Evgeny P., *Moscow State University of Railway Engineering (MIIT), Moscow, Russia.*
Koturanov, Vladimir N., *Moscow State University of Railway Engineering (MIIT), Moscow, Russia.*
Korzhin, Sergey N., *Moscow State University of Railway Engineering (MIIT), Moscow, Russia.*

ABSTRACT

Studies show that the stress state in the contact area is achieved by pressure in the contact area, which influences the stress. Thus, considering the Hertz contact of a spherical indenter with elastic halfspace, Huber [1] determined stresses in the cartesian reference system by a set of expressions [2]. It is easy to see that the state of stress for each point is determined by the average pressure and physical characteristics of contacting materials and coordinates of the point in question. The objective of the authors was to investigate influence of increasing axial load on the wheel-rail interaction, stress state in the area of contact and wear rate of contacting surfaces, using mathematical tools and engineering methods.

As the result of detailed calculations the authors note that there is an increase of principal stresses and pure shear stress is outside the contact area. As wear is proportional to the applied force, the wear rate of wheel and rail surfaces increases. The authors emphasize the need to study the effect of increasing the load on other elements of the rolling stock, track superstructure, roadbed. It is known that productivity of railways can be improved by increasing crossing capacity and carrying capacity. In turn, carrying capacity at constant geometrical parameters of rolling stock units can be increased with increasing axial load. And that again entails changing the state of stress in the contact area of wheel and rail. The article deals with some of the problems arising in the elements of the rolling stock and track superstructure, but does not consider the effect of increasing the load on the roadbed, artificial structures, longitudinal forces in the train, its stability, etc. Hence the authors declare a need for a thorough and comprehensive study to present all the inevitable interdependences in the complex.

Keywords: axial load, railway, productivity, stress state, contact patch, area and rate of wear, cracks, fatigue, loading and unloading cycles, shear area.

Background. *It is known that productivity of railways can be improved by increasing crossing capacity and carrying capacity. In turn, carrying capacity at constant geometrical parameters of rolling stock units can be increased with increasing axial load. And that again entails changing the state of stress in the contact area of wheel and rail.*

Studies show that the stress state in the contact area is achieved by pressure in the contact area, which influences the stress. Thus, considering the Hertz contact of a spherical indenter with elastic half-space, Huber [1] determined stresses in the cartesian reference system by following expressions [2]:

$$\sigma_y = q_0\psi_y(r, z);\quad \sigma_z = q_0\psi_z(r, z);\quad \tau_{yz} = q_0\psi_{yz}(r, z). \qquad (1)$$

It is easy to see that the state of stress for each point is determined by the average pressure q_0, where

$$q_0 = P/S. \qquad (2)$$

In our case, P is the force of pressure of the wheel on the rail, S is area of contact. The second factor on the right side of formula (1) depends on physical characteristics of contacting materials and coordinates of the point in question. If we turn to the formulas given in [3], which deals with the state of stress in the contact area of wheel and rail, then it can be seen that the stress at a particular point depends on the pressure, which is equal to

$$q_0 = \frac{3P}{2\pi ab}, \qquad (3)$$

where q_0 is pressure in the center of the contact patch, a, b are axle shafts of the ellipse of contact.

Objective. *The objective of the authors is to investigate influence of increasing axial load on the wheel-rail interaction, stress state in the area of contact and wear rate of contacting surfaces.*

Methods. *The authors use analysis, mathematical tools and engineering methods.*

Results. *Increasing the axial load will increase stresses on the contact surface and in the area adjacent to the contact patch and an increase in stresses proportionally to the axial load.*

If we now use formulas for calculating maximum stress [4], taking into account the statutory parameters of a car's wheel and a rail, then we obtain the following calculated values of pressure and stresses in the rail:

$$q_0 = m_q \sqrt[3]{\frac{PE^2}{R^2}} = 114{,}3 \text{ MPa}. \qquad (4)$$

Coordinate of the most dangerous point lies on the axis perpendicular to the contact patch, and passes through the center of the contact area. The value of the maximum difference between principal stresses at this point is

$$\sigma_z = 0{,}63\, q_0 = 72{,}0 \text{ MPa}, \qquad (5)$$

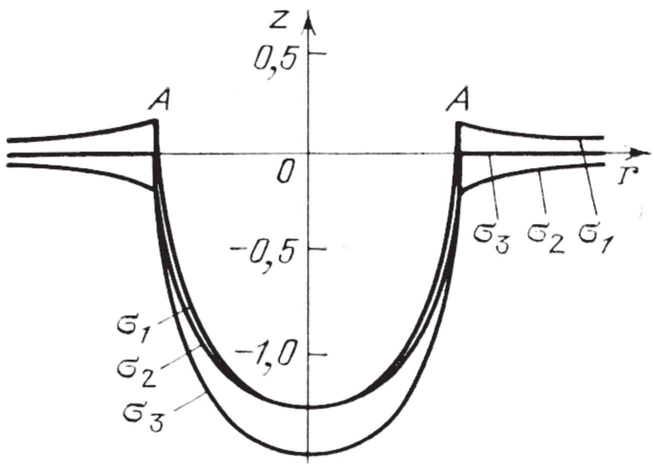
and the value of the greatest difference between the contact area is located at the end of the large axle shaft and is

$$\sigma_x = 0{,}25\, q_0 = 28{,}6 \text{ MPa}. \qquad (6)$$

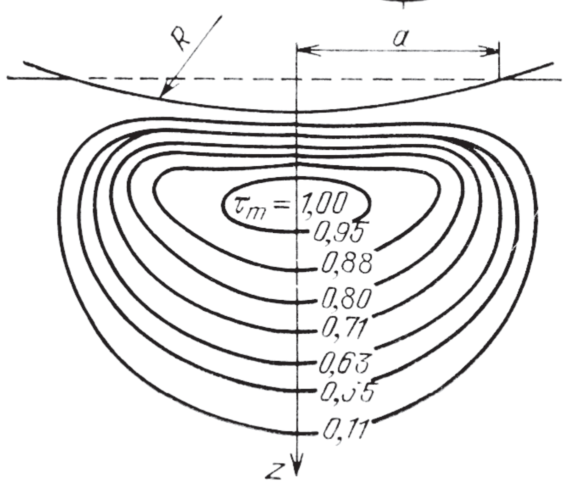
Table 1

Permissible stresses of wheel and rail steel

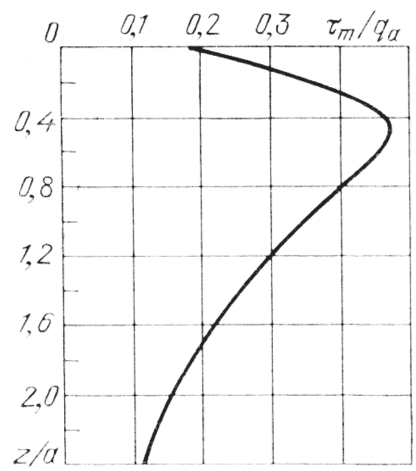
Brand of steel	MPa				
	$[\sigma_р]$	$[\sigma_и]$	$[\sigma_{кр}]$	$[\sigma_{сд}]$	$[\sigma_{см}]$
C2	80	100	65	50	120
M76B	90	120	75	65	145



Pic. 1. Principal stresses on half-space for Hertz field.



Pic. 2. Lines of equal maximum shear stresses in in depth.



Pic. 3. Change in maximum shear stresses axial plane.

Currently under static load of 125 kN and a conical profile of wheel tread bearing stress in the contact area is 114,3 MPa.

The increase in the axial load for 20% will result in an increase in the corresponding variables, the values of which will be:

q_0= 121,5 MPa, σ_z = 78,6 MPa, σ_x = 30,4 MPa. (7)

Permissible stress values of the wheel steel (C2) and the rail steel (M76B) are shown in Table 1.

Comparing the values of (7) with permissible values, we see that the stress on the surface of the contact patch of wheel exceeds the allowable bearing stress, and calculated stress on the vertical axis through the center of the contact patch is almost equal to permissible tension stress (78,6<80).

If we take into account the dynamics coefficient [5] κ=1,4, then these indicators indicated in expressions (7) will increase their value to

q_0= 135,9 MPa, σ_z = 87,9 MPa, σ_x = 34,0 MPa.

For wheels design stresses are significantly higher than permissible, and for rails they are slightly less.

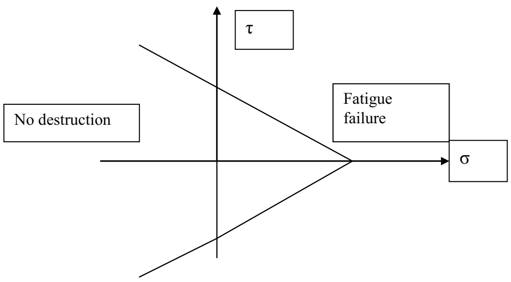
Contact interaction to which belongs the interaction between wheel and rail, is accompanied by destruction of the contacting bodies. It is noted [6] that, main shear stresses contribute to the destruction (8):

$\tau_{xy} = \tau_{12} = (\sigma_1 - \sigma_2)/2$, $\tau_{xz} = \tau_{13} = (\sigma_1 - \sigma_3)/2$, $\tau_{yz} = \tau_{23} = (\sigma_2 - \sigma_3)/2$.

The inclusion of a spherical indenter into an elastic half –space shows [3] that, under the contact area in it, main stresses are compressive and are similar in magnitude. Within the area the particulates of material are under hydrostatic compression, which eliminates the occurrence of surface failure under static compression (Pic. 1). At the same time, radial contact stresses outside the contact area become expanding ones, tangential stresses remain compressible, vertical stresses become zero. Thus, outside the contact patch, stress state is a pure shear, which contributes to the destruction of the contacting bodies.

Pic. 2 shows the lines of equal largest semidifferences of principal stresses, which are formed at a certain depth from the contact area.

The maximum shear stress occurs at a depth of z = 0,49 r (Pic. 3) when including a spherical indenter (r is a radius of the contact patch) in an elastic half-space. When wheel and rail contact each other in a static position ellipse of the contact patch has an eccentricity of 0,95, and according to [4], the maximum shear stress is τ_{yz} = 74,3 MPa and it is observed at a depth of z = 1,65 mm. With increasing pressure due to the dynamics the principal stresses at a depth of the contact area and outside the contact patch increase, and if for some reason in these areas microcracks arise, the large stresses contribute to the development of cracks in the main cracks, and ultimately – to the destruction of material.



Pic.4. Diagram of Dang Van for cyclic fatigue under multiaxial loading.

The emergence of large shear stresses across the axis of the track and the finite size of the wheel explain the occurrence of flowed metal on the tread surface towards the outer side of the wheel and the formation of a false crest. And if the number of turning of wheel sets therefore was 4,7% at some moment, then with increasing axial load this percentage increase.

It should be noted that we have only examined the impact of vertical forces. At movement of a wheel set and its transverse displacement one of wheels slips [6], tangential forces occur in the contact area of wheel and rail resulting in, shear stresses within the contact area and at the marked depth from the surface of the contact area.

Given the tangential stress criterion, based on the criteria of Dang Van [7, 8], fatigue failure may occur at low shear stresses (Pic. 4), if principal stresses are high.

Analyzing the diagram, it is necessary to remember that fatigue failures can occur at small values of the principal stresses, if the shear stresses are such that the point falls in the zone of destruction.

Fatigue failure occurs after a specific number of cycles of loading and unloading, and therefore at one rotation of the wheel deep particles pass one cycle, and surface particles along the track pass two cycles. This is because in the shear area ride patches are formed at the front and at the rear.

The increase in axial load will increase the wear rate of the rolling surface of the contacting bodies, because the wear rate [8] is directly proportional to the normal pressure:

$$\gamma = k^{12} p_n v^{12}, \qquad (9)$$

where γ is wear rate of contacting bodies;

k^{12} is aggregate coefficient of material wear of friction surfaces ($k^{12} = k^1 + k^2$, where k^1 is wear coefficient of the first body, k^2 is wear coefficient of the second body);

p_n is normal contact pressure;

v^{12} is relative sliding speed of bodies relative to each other.

The increase in the axial load by the 20% will increase the wear rate also by 20%.

Without analyzing in detail the effect of increasing axial load on the strength of the car body, we note only a proportional increase in the static deflection of the spring kit with consequent degradation of the vertical and horizontal dynamics.

As discussed in [6], in connection with the transverse displacement of wheel sets one wheel starts to slip. As a result, there are forces of resistance to movement in the form of friction forces, which are proportional to the axial load. Therefore, the increase in axial load will increase the motion resistance forces in straight and curved sections of track.

Conclusion. *The article deals with some of the problems arising in the elements of the rolling stock and track superstructure, but does not consider the effect of increasing the load on the roadbed, artificial structures, longitudinal forces in the train, its stability, etc. Hence there is a need for a thorough and comprehensive study to present all the inevitable interdependences in the complex. Without such an approach the scientific analysis turns into a naked theorization and may be a simple abstraction.*

REFERENCES

1. Huber, M. T. Zur Theorie der Berührung fester Elastischer Körper // Ann, Phys., Leipzig, 1903, Bd.14, H.1, N 6, S.153–163.

2. Kolesnikov, Yu.V., Kozhevnikov, E. M. Mechanics of contact destruction [*Mehanika kontaktnogo razrushenija*]. Moscow, Nauka publ., 1989, 224 p.

3. Belyaev, N. M. Local stresses at compression of elastic bodies [*Mestnye naprjazhenija pri szhatii uprugih tel*] In: Proceedings of the theory of elasticity and plasticity. Moscow, Gostekhizdat publ., 1957, pp. 31–146.

4. Belyaev, N. M. Calculation of maximum design stresses [*Vychislenie naibolshih raschetnyh naprjazhenij*] *Proceedings of the theory of elasticity and plasticity*. Moscow, Gostekhizdat publ., 1957, pp. 231–260.

5. Vershinsky, S.V., Khusidov, V.D., Danilov, V. N. The dynamics of the car [*Dinamika vagona*]. Moscow, Transport publ., 1991, 360 p.

6. Korolkov, E. P. Reduction of wear of the wheels of railway rolling stock with design changes in running gear. D. Sc. (Eng.) thesis [*Snizhenie iznosa koles zheleznodorozhnogo podvizhnogo sostava pri konstruktivnyh izmenenijah hodovyh chastej / Dis... dok. tehn. nauk*]. Moscow, 1997, 229 p.

7. Ekberg, A. Rolling contact fatigue of railway wheels – computer modeling and in-field data. Proceedings of 2nd mini conf. Contact mechanics and wear of railway systems, 1996, pp.154–163.

8. Socalo, A. V. Improvement of tread surface of a car's wheel on the basis of contact fatigue. Ph. D. (Eng.) thesis [*Sovershenstvovanie profilja poverhnosti katanija kolesa vagona na osnove kontaktnoj ustalosti /Dis... kand. tehn. nauk*]. Bryansk, 2011, 142 p.

9. Schultz, V. V. Form of natural wear of machine parts and tools [*Forma estestvennogo iznosa detalej mashin i instrumentov*]. Leningrad, Mashinostroenie publ., 1990, 208 p.

Information about the authors:
Korolkov, Evgeny P. – D.Sc. (Eng.), professor of Moscow State University of Railway Engineering (MIIT), Moscow, Russia, epkorolk37@rambler.ru.
Koturanov, Vladimir N. – D.Sc. (Eng.), professor of Moscow State University of Railway Engineering (MIIT), Moscow, Russia, +7(495) 684–22–10.
Korzhin, Sergey N. – Ph.D. (Eng.), associate professor of Moscow State University of Railway Engineering (MIIT), Moscow, Russia, korjin@miit.ru.
Article received 31.10.2014, accepted 19.01.2015.

The article is based on the papers, presented by the authors at the International scientific and practical conference «Rolling stock's Design, Dynamics and Strength», dedicated to the 75th anniversary of V. D. Husidov, held in MIIT University (March, 20–21, 2014).