World Applied Sciences Journal 27 (4): 428-431, 2013
ISSN 1818-4952

DOI: 10.5829/idosi.wasj.2013.27.04.13619

Azones of Static and Slip Friction in the Patch of Contact of a Vehicle Tire with Solid Bearing Surface

E.V. Balakina, N.M. Zotov and A.P. Fedin

Volgograd State Technical University (VolSTU), Volgograd, Russia

Abstract: In the contact patch of the tire of a vehicle, moving in a straight line with acceleration or braking, there are sections of both static and slip friction. These sections, together with vertical forces, cause the effort, transferred in the contact patch in longitudinal direction. The direction of this effort coincides with the central plane of the wheel. If a lateral (transversal) force, acting on the wheel axis, occurs for any reason, then a transversal effort, in addition to longitudinal one, also occurs in the contact patch. This transversal effort at initial time, in contrast to longitudinal effort, is taken only by sections of static friction in the contact patch. Based on analytical and experimental analysis, this paper defines the rules for determining the parameters of sections of static and slip friction in the patch of contact of a tire with the road. Relying on these rules, it is found that certain presently known views about where sections of beginning of a slip in the patch of contact of a vehicle tire with the solid bearing surface are unjustified in some cases. It is shown that the location of a section, where slip begins, influences the vehicle stability and steerability. It is found that problems on determining the places where slip begins in the contact patch should be solved jointly using diagrams of both tangent and normal stresses. It is also determined that a change from static friction to slip friction takes place in those sections of the contact patch where tangent efforts exceed their permissible levels; and the method for determining these thresholds is also considered in this paper.

Key words: Tires · Contact patch · Zones of friction types · Stresses · Responses

INTRODUCTION

It is well known that the contact patch of tire of a vehicle, moving over a solid flat bearing surface, has sections (zones) of static and slip friction [1, 2], differing by a number of characteristics. It is noteworthy that the section of static friction creates the static friction force and the section of slip friction creates the slip friction force in the contact patch. The static and slip friction forces differ in particular in that the direction of the vector of static friction forces is opposite to the direction of vector of sum of forces, acting on the body; and the direction of the vector of forces of resistance to slip is opposite to the direction of vector of body speed relative to motionless surface. In view of this difference, until the time when the entire surface of the contact patch of the wheel will slip over the bearing surface (i.e., until the time of occurrence of skid or wheelspin), the section of static friction in the contact patch takes not only a part or full longitudinal force in the contact patch, but also full transversal force, if exists. At the same time, i.e., until the occurrence of the full slip of the contact patch, the section of the slip friction takes only a part of longitudinal force, directed in opposite direction to the velocity of motion.

As another difference between the static and slip friction forces, we should consider the fact that the static friction force depends on the value of horizontal force, which acts on the body and which can change from zero to its maximum value. This maximum is determined by the static friction coefficient and normal load. At the same time, the slip friction force does not depend on the horizontal force, acting on the body, but depends on the slip friction coefficient and normal load. As far as the horizontal force, acting on the body, is less than the maximum value of static friction force, the body stays at the state of rest. On the other hand, if the force becomes larger than this value, the body will start moving and forces, determined by the slip friction laws, will occur in the zone of contact. At the same time, if an external

Corresponding Author: Balakina, Volgograd State Technical University (VolSTU), Lenin avenue, 28, 400005, Volgograd, Russia.

horizontal force will be larger or smaller than the slip friction force, then the body, accordingly, will speed up or slow down.

In the presence of lateral force and in the absence of lateral slip of a wheel of a moving vehicle, the zone of slip friction takes longitudinal load and realizes a part of longitudinal response of the bearing surface and the zone of static friction takes longitudinal and lateral loads and realizes both a part of longitudinal and the total lateral response of bearing surface. This affects the phenomena: withdrawal of elastic wheel and oscillations of steered wheels, which determine the stability properties of motion, steerability and brake dynamics of vehicle [1-7].

The literature sources give uncertain and sometimes even contradictory, data on locations of zones of friction in the patch of contact of a tire with bearing surface under different conditions of wheel load. At the same time, the knowledge about the locations of friction zones in the contact patch will allow more accurate calculation of the moments, turning the steered wheels around turning axes, which determines the parameters of angle oscillations of the steered wheels and their tire stabilization.

It is well known [8, 9] that, in wheel braking regime, the diagram of normal stresses in the contact patch shifts toward the rear part of the contact patch. At the same time, as shown in Fig. 1, the distance from resultant normal response of the bearing surface R_z to the geometric center of the contact patch is taken as b [8, 9].

If we decompose this general scheme of Fig. 1 into two separate schemes, shown in Fig. 2,*a*,*b*, we can identify the meanings of the two components of the total magnitude of the shift b [9]. The first component (Fig. 2,*a*) characterizes inelastic component of the shift and the second component (Fig. 2,*b*) characterizes elastic component of the shift in the direction of propulsion force.

The authors of this paper have experimentally proved that the component a is negative in braking regime. Since the component a characterizes the elastic longitudinal deformation of a tire and represents a displacement in the direction of the propulsion force, it is positive in the braking regime; therefore

$$|b| = |a| - |c|$$

Values of both components of the displacement b are estimated.

The initial X_0 and final X coordinates of the normal response of the bearing surface from the beginning of the contact patch can be calculated from the formulas:

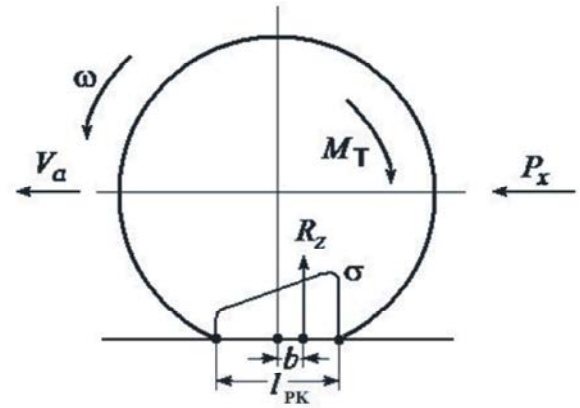
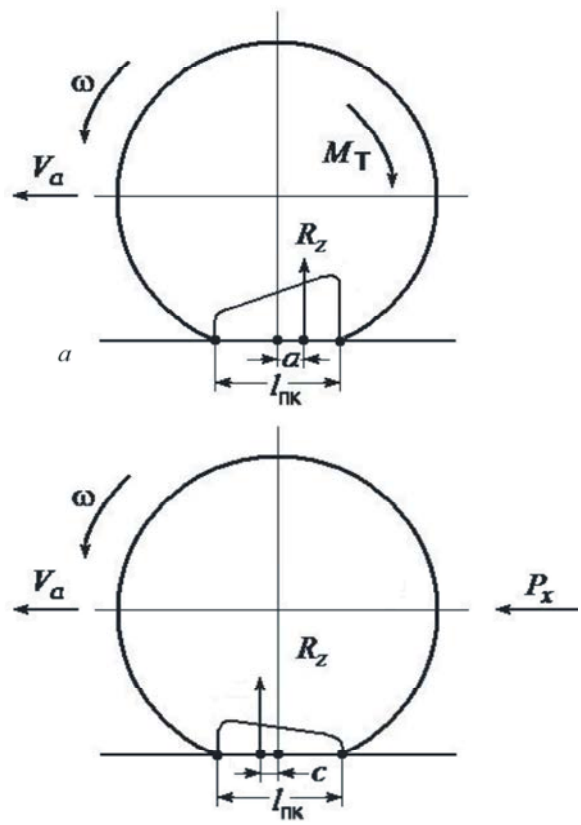


Fig. 1: General scheme of loading the braking wheel [8, 9]



ПК - PK

Fig. 2: Forms of diagrams of normal pressures in the contact patch under the impact of (*a*) only braking moment of the wheel and (*b*) only propulsion force exerted by the body (frame)

$$X = \frac{2}{3} l_{av}; \quad X_0 = \frac{1}{2} l_{av};$$

$$a_{max} = X - X_0 = \frac{1}{6} l_{av}.$$

Thus, it is found that a has the maximum value

Table 1: Calculation of the maximum value of inelastic component of displacement of resultant normal response of the bearing surface from the geometrical center of the contact patch

Automobiles	Radial stiffness coefficient of tires [1,2] C_{tz}, N/mm	Permissible normal load of a wheel $[P_z]$, H	Maximum vertical deformation of tires h_z, mm	Length of the contact patch l_{av}, cm	a_{max}, cm
Cars	150-250	3000-6500	20-25	15-26	3-4.5
Trucks	500-1100	12500-30000	25-30	30-38	5-6.5

Table 2: Calculation of the maximum elastic component of displacement of resultant normal response of the bearing surface from the geometrical center of the contact patch

Automobiles	Coupling radius of tires r_C, mm	Torsional stiffness coefficient of tires [1,2] $C_{t\beta}$, Nm/rad	Longitudinal stiffness coefficient of tires [1,2] $C_{tx} \approx \frac{C_{t\beta}}{(r_c)^2}$,N/mm	Maximum possible propulsion force, H	C_{max}, cm
Cars	152-216	17200-45800	745-1100	2800-6050	0.3-0.5
Trucks	254	86000..200500	1330..3110	11600..27900	0.8-0.9

Table 3: Components of the shift b of resultant normal response of bearing surface from geometrical center of contact patch

Automobiles	a_{max}, cm	c_{max},cm
Cars	3-4.5	0.3-0.5
Trucks	5-6.5	0.8-0.9

$$a_{max} = \frac{1}{6} l_{av},$$

where l_{av} is the length of the patch of contact of tire with the road.

After the lengths of the contact patches of tires of cars and trucks were calculated using the known dependences (from their geometric sizes, radial stiffness and radial load capacity), values a_{max} were obtained, as shown in Table 1.

After the longitudinal elastic displacement of tires of cars and trucks were calculated (from their longitudinal stiffness and longitudinal load capacity), values c_{max} were obtained, as shown in Table 2.

$$\text{Here, } c_{max} = \frac{[P_x]}{C_{tx}},$$

where $[P_x]$ is the maximum possible propulsion force; C_{tx} is the longitudinal stiffness coefficient of a tire;

$$[P_x] = [P_z].\ \varphi_{max},$$

where φ_{max} is the maximum coefficient of adhesion of a tire with dry asphalt.

The results of the calculations are summarized in Table 3.

From Table 3 it can be seen that the elastic shift b is about an order of magnitude smaller than the inelastic shift a. Hence, we can conclude that, for braking wheel, the shift b of resultant normal response of bearing surface from geometrical center of contact patch is almost always negative. An exception may be only the situation at the beginning of braking, when the braking moment had still no time to grow ($M_B \approx 0.1.\ M_{Bmax}$).

Using the diagram of normal stresses in the patch of contact of braking wheel with the road, we can compose the diagram of permissible tangent stresses in the patch of contact of the braking wheel with the road:

$$[\tau] = f_s.\ \sigma,$$

where $[\tau]$ are the permissible tangent stresses in the patch of contact of the braking wheel with the road; σ are the normal stresses in the patch of contact of the braking wheel with the road as given in diagram in Fig. 1; and f_s is the coefficient of static friction between the material of the tire and the bearing surface.

It is well known how the diagrams of tangent stresses look in the regime of a braking wheel [10]. Based only on the diagrams of tangent stresses, some authors draw conclusion about the mutual arrangement of zones of friction types in the patch of contact of elastic wheel with the solid bearing surface. This approach is incorrect, since a section with slip in the contact patch occurs not just where tangent stresses are maximal, but rather where they start to exceed the permissible stresses, determined by the static friction, i.e., when

$$\tau_i \geq [\tau_i] = f_{II}.\ \sigma_i.$$

Thus, section with slip in the patch of tire contact with the road can only be determined by comparing the diagrams of tangent and normal stresses. From this comparison we can conclude that, in the braking regime of a vehicle wheel (in contrast to driving regime), the section of the static friction lies behind; and the section with slip friction occurs in the front part of a contact patch and, as the coefficient of the longitudinal slip of a wheel grows, it increases toward the rear part of the contact patch. This phenomenon was proved by the authors on the basis of experiments, performed in Volga State Technical University (VSTU), during which the zones of static and slip friction were recorded using video instrumentation by means of optically transparent model of a road, interacting with elastic tire-protected wheel, loaded with radial force and braking moment.

CONCLUSIONS

- Diagrams of tangent stresses alone give incorrect conclusions about the place where slip begins in the patch of contact of vehicle tire with solid bearing surface.
- The problems on determining the slip sections in the patch of contact of vehicle tire with solid bearing surface should be solved using both the diagrams of tangent stresses and the diagrams of normal stresses. The contact is lost on sections where $\tau_i \geq [\tau_i] = f_s \cdot \sigma_i$.

REFERENCES

1. Balakina, E.V. and N.M. Zotov, 2011. Stable Motion of Wheel Vehicles. Volgograd. Politekhnik, pp: 464.
2. Balakina, E.V., 2012. The Improvement in Stability of Wheel Vehicle Motion through Pre-design Selection of Parameters of Chassis Elements. Saarbrucken (Germany), LAP LAMBERT Academic Publishing GmbH & Co, pp: 467.
3. Delanne, Y., G. Schaefer, D. Lechner, V. Schmitt and G. Beurier, 2001. Vehicle Dynamics and Tyre Road Friction Performance Models. The Report of the 2nd International Colloquium on Vehicle Tyre Road Interaction «Friction Potential and Safety: Prediction of Handling Behavior», Florence, pp: 11.
4. Pacejka, H.B., 2012. Tire and Vehicle Dynamics. Elsevier Ltd, USA, pp: 632.
5. El-Nashar, M., 2010. Vehicle Tire Road Forces. Deutschland: LAP LAMBERT Academic Publishing, pp: 212.
6. Jazar, R.N., 2008. Vehicle Dynamics: Theory and Application. Springer Science + Business Media, LLC, pp: 1015.
7. Koskinen, S., 2010. Sensor Data Fusion Based Estimation of Tyre-Road Friction to Enhance Collision Avoidance, M. S. thesis, Tampere University of Technology, pp: 209.
8. Moskovkin, V.V. and V.A. Petrushov, 1969. Partial Slipping Motion in a Contact as One of the Sources of Shift in Resultant Normal Response of Elastic Wheel. In the Proceedings of NAMI, no. 106. Elastic Wheel in the Series of Transmission Mechanisms. Moscow. NAMI Publishing, pp: 41-51.
9. Petrushov, V.A., 2008. Automobiles and Road Trains: New Technologies of Research into Resistance of Rolling and Air. Moscow. TORUS Press, pp: 352.
10. Klennikov, E.V., V.I. Knoroz and I.P. Petrov, 1970. Experimental Studies of Normal and Tangent Stresses in the Contact of a Tire. Moscow. NAMI Publishing, pp: 78-95.