ABOUT MODELING OF DRY FRICTION BETWEEN A WHEEL AND A ROAD SURFACE

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The friction between the wheel and the road, its measurement and the relationship with the risks of road accidents are important issues. Thousands of road engineers around the world are doing this. In many countries, there are certain thresholds for road adhesion that determine the least acceptable road friction. If the friction level is below this value, the risk of accidents may increase. These thresholds are the result of a study of the relationship between the speed of the center of the wheel and its angular velocity. The purpose of the work is to investigate the methods used for measuring dry friction and the quantitative relationships between the coefficients of friction with the road, the risk of accidents and the theoretical approach to their estimates. The connection of friction between the roadbed and the wheel of the car with the risk is obvious. If the road is very slippery, then the risk of skidding increases significantly, but the exact determination of the threshold values of the coefficient of friction, which should set the parameters of the road surface, is not an easy task. This problem can be divided into two parts – methods of measuring friction and the relationship between friction and the risk of an accident.

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1. Introduction

When moving any wheeled vehicle, the main and only element in contact with the road (road surface) is the wheel. Forces resulting from (due to) such contacts are external to the system of material bodies forming the structure of any vehicle. These forces significantly affect the form of the basic equations of dynamics of the system under consideration (the equations of motion of the center of mass and the equations of kinetic moment relative to the center of mass of the vehicle). Without taking into account these equations, no mathematical modeling of controlled or uncontrolled movement of a vehicle (car, wheeled robot) is possible. Therefore, the study (both qualitative and quantitative) of the nature and models of forces arising in the area of wheel-road contact is an important and necessary task in the mathematical modeling of these vehicle movements.

When a deformable wheel is rolling, tangential and normal forces arise in the area of its contact with the road, and, secondly, moments of these forces relative to the center of the wheel. Tangential forces have the character of dry friction forces, which significantly depend on normal forces, and are also determined by the properties of the wheel and the pavement (coefficient of friction). The normal forces and their distribution over the contact area are determined by the properties of the deformable wheel, as well as its kinematics, i.e., depend on the dynamics of the movement of the vehicle itself. For example, with the rectilinear movement of the vehicle, the resulting normal reaction is shifted forward or backward, depending on whether its movement is accelerated or slowed down (Andronov and Zhuravlev, Rozenblat). When calculating the moments of these forces, it is necessary, in addition to the properties of the contacting surfaces, to take into account also the kinematics of the vehicle movement. For example, when turning a vehicle, it is necessary to take into account the angular velocity of its rotation, which leads to the appearance of torsion friction moments in the area of contact of the wheels with the road.

One of the simplest and most popular (classical) models of wheel contact with the road is the point contact model. As a consequence of this assumption, a classical Coulomb model appears for the forces of dry sliding friction and rest (Coulomb, Jellett, Penleve, Appel). This model attracts with its simplicity and ability (sometimes) to obtain good analytical and reliable (mostly qualitative) results. As a rule, this model is not suitable for obtaining acceptable quantitative results. In this case, it is necessary to complicate the friction models used, taking into account both the "inaccuracy" of the contact spot and the dependence of the distribution of normal and tangential forces in the contact area on the kinematics of vehicle movement (for example, the Contensu-Zhuravlev model). In this case, it is necessary to perform rather complex integrations over the contact area (the shape of which, as a rule, is unknown). The expressions obtained for the final main forces and moments of friction and normal reactions should be used in the study of vehicle dynamics. In some cases, it is also possible to obtain acceptable analytical results here (Rozenblat). However, for practically important cases it is necessary to turn to numerical methods (Zhuravlev, Klimov, Borisov, Rozenblat). The Contensu-Zhuravlev model of two-dimensional friction is of great importance when studying, for example, the problem of shimming the wheels of an airplane landing gear during landing (see the article by Zhuravlev, Klimov).

It should be noted that a similar problem exists for airfields, but in but this problem is outside the scope of this study and, therefore, is not considered.

2. Influence of tire condition on accident rate

As noted in [20], the technical condition of the car and its pneumatic tires significantly affect road safety. It is necessary to take into account the defects of the tread, the breaker, the sidewall and the side of the tire.

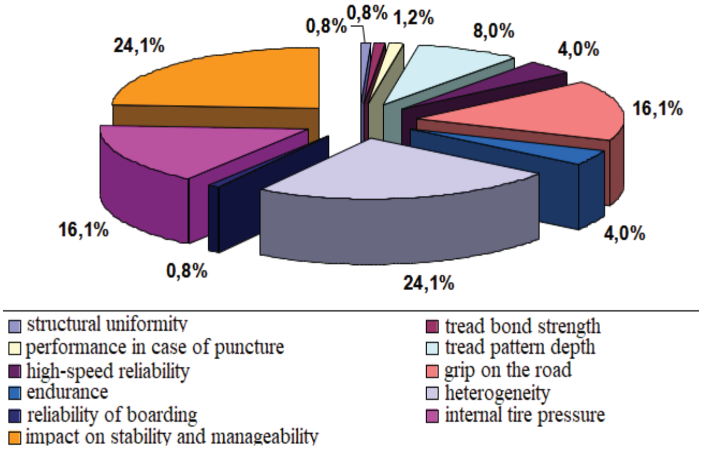

Figure. 1. Causes of accidents related to the operation of tires

According to statistics, in particular, accidents for technical reasons account for 1.8% to 5.3% of all accidents. Of these, 15% are due to tire malfunctions of the busiest, 97.3% of the total intensity is created by passenger cars and 2.7% by trucks. Thus, mathematical models describing the contact interaction of the tire with the road surface are relevant.

3. Experimental studies

Friction is an important road parameter, which, unfortunately, is very difficult to measure. The devices used to measure friction are not very sophisticated, but the friction forces they are trying to measure are very sensitive to a number of parameters that are difficult to control. In Meyer et al. the study of several parameters affecting the amount of friction of a locked wheel, friction measurements in accordance with ASTM1 E274, is summarized. In the UK, SCRIM (Side force Equivalent Road Inventory Machine) is used. The factors influencing the value of SCRIM friction were determined in a series of studies [1-15]. The difference in friction of about 5% between two consecutive measurements of the same road surface using the same device is not something surprising. This difference may increase when using a different friction device or with an increase in the time between measurements.

When measuring friction, three types of measurement are usually involved: the tire being measured, the road surface, and any contamination that interacts with both the tire and the road, for example, water (wet friction), dust or wear waste. The obtained friction values largely depend on all three bodies – their material properties, local contact pressures, relative velocities, etc.

Of the road parameters, texture is the most important, and the influence of macrotexture has also aroused the greatest interest in road friction studies. A qualitative and detailed overview of pavement texture measurements can be found in the Sandberg report. The differences between micro, macro and megatexture are defined in ISO 13473-1.

When assessing friction on the road, a typical approach is to keep all influencing parameters constant, except for the road surface. The surface is usually wetted with a certain amount of water, and a standardized measuring tire is used. A smooth tire similar to the one specified in ASTM E524 or PIARC2 a smooth test tire, ribbed tires such as ASTM E501, or a tire with a pattern in ASTM E1136 are examples of conventional tires for evaluating the friction of the road surface (Fig. 2). ASTM E524 and E501 have an outer diameter of 703 mm, ASTM E1136 has an outer diameter of 648 mm, and PIARC smooth and ribbed test tires have an outer diameter of 646 mm.

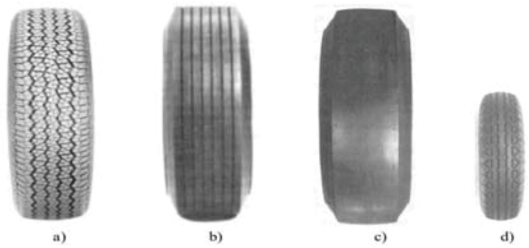

Figure 2. 4 standard types of tires for testing the properties of the roadway. a) ASTM E1136 b) ASTM E501 c) ASTM E524 d) T49

Figure 2 also shows the Swedish standard tire for measuring friction on the road, Trelleborg T49. With an outer diameter of 420 mm, it is smaller than the ASTM and PIARC test tires. Another smaller standard test tire is the ASTM E1844-96 smooth test tire for GripTester. This tire has an outer diameter of 258 mm.

The friction values are very sensitive to the tire used, and even externally identical samples from two different batches of standardized tires can give different measurement results, not very different in absolute values, but significantly different. The reasons for this are, for example, small differences in the composition of tire rubber, as well as possibly small differences in tire geometry.

According to research by Henry and Bachmann, ribbed and smooth standard test tires (ASTM and PIARC) have a lower coefficient of friction on a conventional wet road than a commercial passenger car tire. As the water depth increases, the performance of the rib test tire increases, and it has been found that it provides higher friction values than a slightly worn conventional passenger car tire.

The friction of the road surface is usually estimated in the summer. There are seasonal fluctuations that need to be taken into account. The wet friction of a particular road surface is usually higher in spring than in autumn due to the effect of depolishing from snow removal and studded tires.

In winter, there is a risk that the water spreading during the wet friction measurement will freeze. The road surface can also be covered with snow or ice. Therefore, friction measurements in winter are usually aimed at assessing maintenance in winter.

It is also possible to measure the difference between tires of different brands, the study measured the wet grip of about 82 different tires. All measurements were carried out on the same asphalt road surface. The best tire has an optimal wet friction coefficient of about 1.0, and the worst tire is about 0.7. Nordström and Gustavsson found a range of optimal friction coefficient between 0.6 and 0.85 in a similar study using a different measurement principle and about 250 different tires for passenger cars.

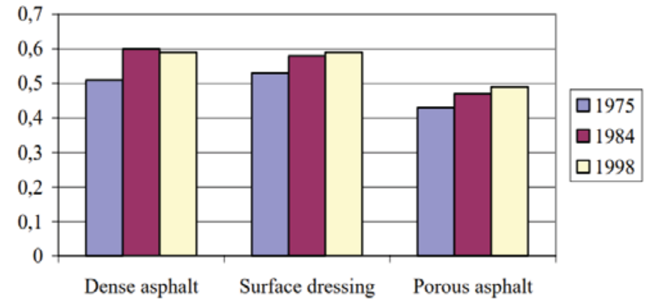

Figure 3. Friction coefficient of 3 different PIARC test tire supplies

This clearly indicates the importance of the tire in assessing road friction, and also clearly show that individual cars may experience different levels of friction, even if the friction of the pavement is constant, as estimated using a single standard tire.

Road friction can be measured in one of 5 different conditions: locked wheel (100% slip), constant slip (usually 10 to 20% slip), variable slip (0 to 100% slip), constant slip angle (usually 20 degrees) and consideration of the delay when the ABS is triggered. The effect of the sliding angle on the lateral force (lateral force) is shown in Figure 4.

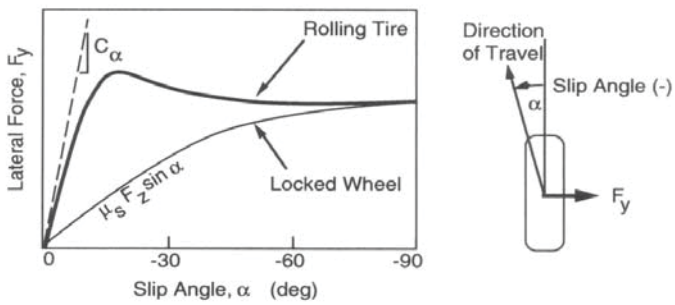

Figure 4. Lateral friction force depending on the sliding angle (Gillespie [17])

Schulze, Garibaldi and Chauvet [14] present the results of the analysis from the Netherlands, Germany and France. In the Netherlands, all traffic accidents on state roads for two years, 1965 and 1966, were used in regression analysis. The accident rate was calculated based on the number of accidents for a certain period on the selected road section and the total number of vehicle kilometers for the same period and on the same section.

The friction values for each road section were measured using the standard Dutch test method (86% slip). Wet friction values were used for all accidents that occurred in wet weather, and dry friction values for those accidents that occurred in dry conditions. Figure 5 shows the relationship between the level of friction and the frequency of accidents.

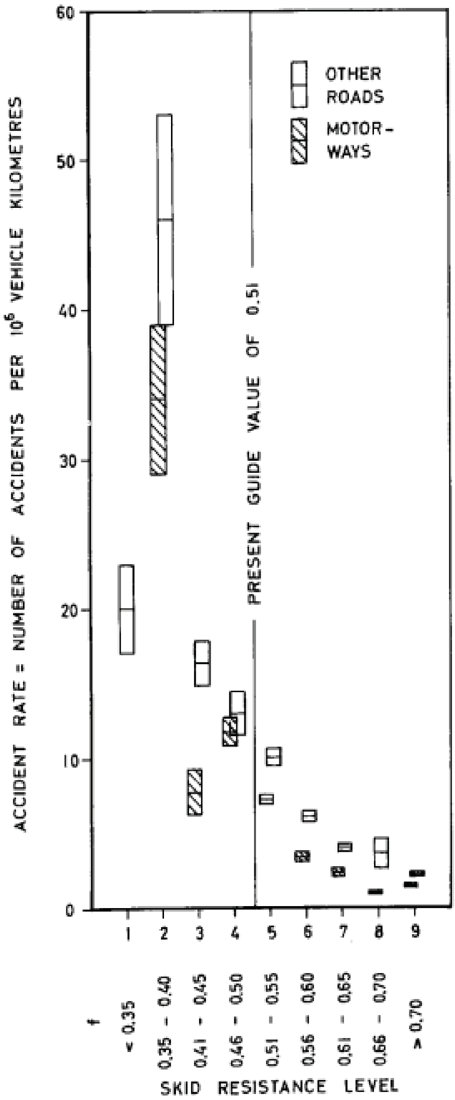

Figure 5. The frequency of accidents (mean and standard deviation) depending on the coefficient of friction

4. One-dimensional model of dry friction motion using a Frode pendulum

For systems with oscillations, dry friction manifests itself as a dissipative factor. The action was reduced to the suppression of the emerging oscillatory movements – the suppression of free oscillations up to their complete cessation (in a finite time!) and the weakening of the intensity of forced oscillations. However, in nature and technology, there are oscillatory processes in which dry friction plays the opposite role, being the cause of undamped oscillations in systems with dry friction. Examples of such movements, called frictional self-oscillations, are the vibrations of the violin string, the creaking of the brakes of subway cars, the intermittent nature of the sliding of the driven links in the mechanisms of devices and much more. A peculiar kind of frictional self-oscillation is also represented by such formidable natural phenomena as earthquakes.

There are several physical mechanisms of excitation of self-oscillations by dry friction forces. One of them, which is most often found in works on the study of frictional self-oscillations, consists in the dependence of the friction force on the sliding speed, more precisely, in the decreasing nature of this dependence at low sliding speeds. This is exactly the behavior of the coefficient of friction f it is found in the experiment for most materials – after starting from a state of rest, the friction force first drops, at a certain value $u = u$ the sliding speed reaches a minimum, then increases again (Fig. 6, a solid branch of the curve $f(u)$).

Such a dependence can be approximated by a branch of a cubic parabola with a center of symmetry on the ordinate axis, assuming:

$$cx + mg\, f(0) = 0, \quad -f_0 \le f(0) \le f_0 \qquad (1)$$

Combining the minimum point of the parabola with the corresponding point of experimental dependence, we obtain:

$$A = f_0\,, \qquad B = -\frac{3}{2}\frac{f_0 - f_*}{u_*}\,, \qquad C = \frac{f_0 - f_*}{2u_*^3}$$

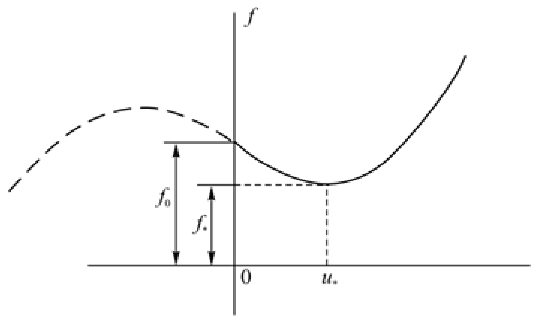

Figure 6. Change in the friction force

As a result, we come to the following equation for the approximating curve:

$$F(u) = f_0 - \frac{f_0 - f_*}{2}\frac{u}{u_*}\left(3 - \frac{u^2}{u_*^2}\right). \qquad (2)$$

Let us study the phenomenon of frictional self-oscillations using the mechanical model "load on an infinite moving belt" (Fig. 7) and approximation (2) to describe the dry friction acting between the load and the belt. Let V – constant tape speed ($V > 0$), m – cargo weight, c – spring stiffness, x – displacement of the load from its position with an undeformed spring, f – coefficient of friction, depending on the module of the relative velocity of the load $u = \dot{x} - V$.

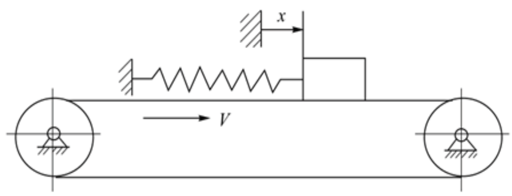

Figure 7. The "load on an infinite moving belt" model

Then the differential equation of the motion of the load will be written as follows:

$$m\ddot{x} = -cx - mgf_0 sgn(\dot{x} - V) + \\ +mg\,\frac{f_0 - f_*}{2}\,\frac{\dot{x} - V}{u_*}\left[3 - \frac{(\dot{x} - V)^2}{u_*^2}\right]. \qquad (3)$$

The cargo can also be in a state of equilibrium — relative and absolute. At relative equilibrium ($\ddot{x} = 0, \dot{x} = V$) the load moves evenly along with the tape. The equation of relative equilibrium:

$$-cx + mg\,f(0) = 0, \quad -f_0 \le f(0) \le f_0 \qquad (4)$$

defines the stagnation area – the set of load positions in which the elastic force of the spring remains modulo less than the maximum friction force of rest.

The equation of absolute equilibrium ($\dot{x} = 0, \ddot{x} = 0, x = x^{\circ} = const$)

$$-cx^{\circ} + mgf_0 - mg\,\frac{f_0 - f_*}{2}\,\frac{V}{u_*}\left(3 - \frac{V^2}{u_*^2}\right) = 0 \qquad (5)$$

determines the positions of the load in which it can remain stationary in the surrounding space. There can be either two or one absolute equilibrium positions (Fig. 8). The equilibrium positions located on the increasing branch of the friction characteristic are stable, on the declining one they are unstable. For the future, only the latter are of interest, since the phenomenon of self-excitation is associated precisely with the presence of instability.

Let $x = x^{\circ}$ – any equilibrium position on the declining branch of the friction characteristic, $\tilde{x} = x - x^{\circ}$ – deviation of the load from this position during self-oscillation. To determine $\tilde{x}$ we get:

$$m\ddot{\tilde{x}} = -c\tilde{x} - mf_0 sgn(\dot{\tilde{x}} - V) \\ +\frac{1}{2}mg(f_0 - f)\frac{\dot{\tilde{x}} - V}{u_*}\left[3 - \frac{(\dot{\tilde{x}} - V)^2}{u_*^2}\right] - cx^{\circ} \qquad (6)$$

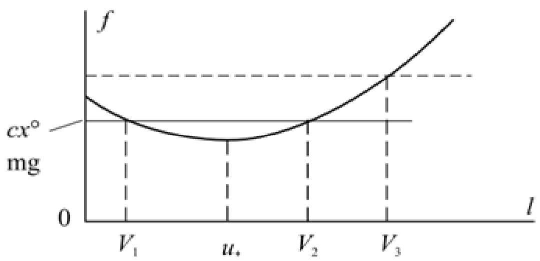

Figure 8. Positions of absolute equilibrium

We introduce dimensionless quantities and write this equation in the following dimensionless form:

$$\ddot{\xi} = -\xi - h\,sgn(\dot{\xi} - v) + \frac{1}{2}hr(\dot{\xi} - v)\left[3 - (\dot{\xi} - v)^2\right] - \xi,$$

$$\xi = \frac{k\tilde{x}}{u_*}, k = \sqrt{\frac{c}{m}}, v = \frac{V}{u_*}, h = \frac{gf_0}{ku_*}, r = \frac{f_0 - f}{f_0}$$

$$(0 \le r \le 1), \xi^{\circ} = \frac{kx^{\circ}}{u_*}. \qquad (7)$$

Denote $\xi_1 = \xi,\ \xi_2 = \dot{\xi}_1$ and we will rewrite (7) in the normal form of Cauchy:

$$\dot{\xi}_2 = \xi_1$$

$$\dot{\xi}_2 = -\xi_1\,h\,sgn(\xi_2 - v) + \frac{1}{2}hr(\xi_2 - v)[3 - (\xi_2 - v)^2] - \xi^{\circ}. \qquad (8)$$

Assuming h small and carrying out the replacement

$$(\xi_1, \xi_2) \to (a, \varphi)\text{: } \xi_1 = a \sin\sin\varphi - x^{\circ},\ \xi_2 = a\cos\cos\varphi, \qquad (9)$$

we bring the system (7) to the form prepared for averaging

$$\dot{a} = -hsgn\,(a\cos\cos\varphi\ - v)\cos\cos\varphi \qquad (10)$$
$$+\frac{1}{2}hr(a\cos\cos\varphi - v)[3 - (a\cos\cos\varphi - v)^{\,2}] \\ \cos\cos\varphi\,,$$

$$\dot{\varphi} = 1 + \frac{h}{a}sgn\,(a\ \cos\cos\varphi - v)\ \sin\sin\varphi\ -$$
$$-\frac{1}{2a}hr(a\cos\cos\varphi - v)\,[3 - (a\cos\cos\varphi - v)^{\,2}]\sin\sin\varphi.$$

After averaging, we get when $a < v$

$$\dot{a} = -\frac{3}{4}hra + \frac{3}{16}hra(4v^2 + a^2), \qquad (11)$$
$$\dot{\varphi} = 1,$$

when $a > v$

$$\dot{a} = \frac{2h}{\pi a}\sqrt{a^2 - v^2} - \frac{3}{4}hra + \frac{3}{16}hra(4v^2 + a^2), \qquad (12)$$
$$\dot{\varphi} = 1.$$

The first case refers to self-oscillations without crossing the break point of the friction characteristic — all events occur on the negative branch of the function $F(\dot{x})$, expressing the friction force as a function of absolute velocity $\dot{x}$ (Fig. 9).

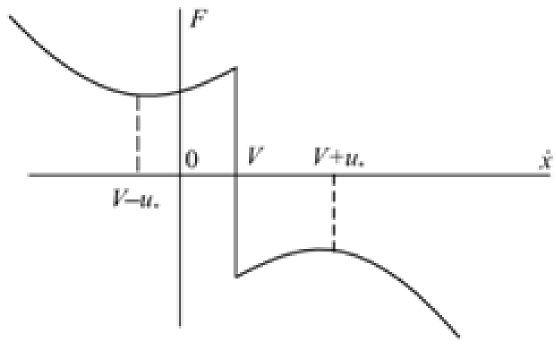

Figure 9. Self-oscillations without crossing the point of rupture of the friction characteristic

The amplitude of the corresponding limit cycle is obtained from the first equation (10) when $\dot{a} = 0$:

$$a = 2\sqrt{1 - v^2} \qquad (13)$$

It is this type of self-oscillation that is given in the literature as an example of quasi-harmonic (Thomson) self-oscillations in dry friction systems.

The second case, corresponding to quasi-harmonic self-oscillations with the intersection of the breaking point of the friction characteristic, was discovered and investigated relatively recently.

Here, the equation for determining the amplitude has a more complex form (the equation of the 4th degree with respect to a^2):

$$\sqrt{a^2 - v^2} = \lambda a^2[4(1 - v^2) - a^2],\ \lambda = \frac{3r\pi}{32}. \quad (14)$$

The equation is defined in the domain

$$v < a < 2\sqrt{1 - v^2},\ \ v > 0. \quad (15)$$

Let's introduce the function

$$\chi(a, v) = \frac{\sqrt{a^2 - v^2}}{a^2[4(1 - v^2) - a^2]} \quad (16)$$

and write the equation in the form

$$\chi(a, v) = \lambda,\ \ a > v,\ \ 0 \le \lambda < \frac{3r\pi}{32}. \quad (17)$$

Here are some properties of the function $\chi(a, v)$ with a fixed $v = v_i = const$. At $v_i < \frac{3-\sqrt{3}}{6} = 0{,}4597$ derivative $d\chi/da$ is equal to zero at two points where

$$a^2 = \frac{2}{3}\left(1 \pm \sqrt{1 - 6v_i^2(1 - v_i^2)}\right). \quad (18)$$

The lower value corresponds to the maximum of the function $\chi(a, v_i)$ for more — a minimum. By $v = v_i > 0{,}4597$ function $\chi(a, v_i)$ monotonously increases with growth a, going to infinity at $a \to 2\sqrt{1 - v^2}$. Function Graphs $\chi(a, v_i)$ for some values v_i shown in Figure 5.

The smallest of the minima of the function $\chi(a, v_i)$ occurs when $v_i \to 0$ and is equal to 0,3247. At the same time, the maximum value of the parameter λ in the equation (1.17) compose $\lambda_{max} = \lambda|_{r=1} = \frac{3\pi}{32} = 0{,}2945$. Hence, each of the graphs of the function $\chi(a, v_i)$ intersects a straight line $\chi = \lambda, \lambda < 0{,}2945$ only at one point, and therefore there is a single mode of self-oscillation of the type in question. Because the condition is met for it $v \ll a \ll 2/\sqrt{5} < 1$, then higher degrees a, v in equation (16), the amplitude of the limiting cycle can be neglected and approximately determined from the equation

$$\sqrt{a^2 - v^2} = \frac{3}{8} r\pi a^2. \quad (19)$$

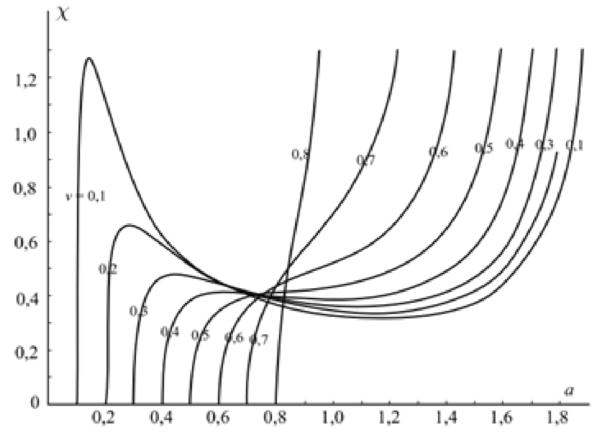

Figure 10. Graphs of the function $\chi(a, v_i)$ for some values v_i

From it we immediately find

$$a^2 = \frac{32}{9r^2\pi^2}\left(1 \pm \sqrt{1 - \frac{9}{16} r^2\pi^2 v^2}\right). \quad (20)$$

The root with the smallest value satisfies the condition of the problem. Comparing the two considered quasi-harmonic modes of self-oscillations, we note the following. In the first case, the balance of incoming and dissipated energy necessary for the existence of self-oscillations is provided due to cubic terms in the characteristic of the friction force. For frictional self-oscillations of the second type, the presence of cubic terms in the friction characteristic is not mandatory – a jump in the friction force and linear terms is sufficient. Initially, it was possible to limit ourselves to a local description of the friction force in the vicinity of the breaking point. It also follows from the above that the mode with the intersection of the break point is more rough and therefore easier to implement in practice. The system "remembers" about the limit cycle even if the condition for the existence of the cycle is violated (negative values of the root expression in formula (20) – a pronounced thickening of the turns of the phase trajectory in the vicinity of the former limit cycle is observed on the phase plane.

At $a \ge 1$, undamped oscillations of a completely different nature can also be established in the system, containing intervals of relative rest of the load on the belt (so-called relaxation (discontinuous) self-oscillations). This happens as follows. The load, sliding along the belt, at some point $t = t_*$ appears simultaneously in the stagnation area and at the point of rupture of the friction characteristic:

$$-x_1 \le x = x(t_*) \le x_1,\ \ x_1 = \frac{f_0 mg}{c},\ \ \dot{x} = \dot{x}(t_*) = V. \quad (21)$$

Then the sliding stops, and the load moves further, moving along with the tape. Upon reaching the boundary of the stagnation area $x = x_1$, a new sliding stage begins, which ends with the same state of the load as at the moment t_*, etc. A periodic movement is established in which the load slides along the tape part of the period, and its other part moves uniformly with the tape (rests on the tape).

Figure 11 (a) shows an approximate view of the corresponding limit cycle on the phase plane. The section P_2P_1 corresponds to the joint movement of the load with the tape. At $x = x_1$ (point P_1), the load breaks from the resting state on the tape and then slides along the tape until it stops on it at some $x = x_2$ $(-x_1 < x_2 < x_1)$ (point P_2). Then the cycle repeats. An exact analytical description of the curved section of the limit cycle is not possible due to the non-integrability of equation (3).

Relaxation self-oscillations are very different from the harmonic law, therefore, for their study, the replacement of the form (9) turns out to be of little use.

Another reference mechanical system, often used to explain the conditions of occurrence and properties of frictional self-oscillations, is the Frode pendulum. This is a physical pendulum suspended on a shaft of radius r, which rotates uniformly with a certain angular velocity ω (Fig. 11 (b)). Coulon's dry friction of the same type acts between the shaft and the pendulum (2).

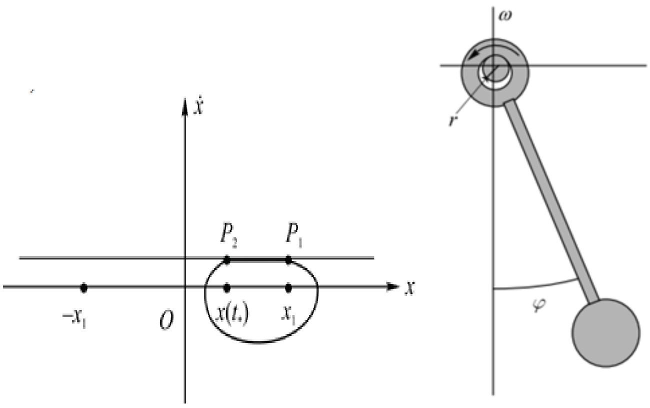

Figure 11. a) Type of limit cycle b) Physical pendulum

Small oscillations of such a pendulum lead to an equation quite similar to equation (3). Therefore, everything that was set above for the load on the belt will also take place in the Frode pendulum.

Conclusion

The significance of the condition of tires on the accident rate of road traffic is investigated. The most promising methods for estimating the dry friction coefficient have been identified. A model of the physical mechanism of excitation of self-oscillations by dry friction forces is proposed and numerical estimates of its parameters are obtained.

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О МОДЕЛИРОВАНИИ СУХОГО ТРЕНИЯ МЕЖДУ КОЛЕСОМ И ДОРОЖНЫМ ПОКРЫТИЕМ

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Аннотация

Трение между колесом и дорогой, его измерение и связь с рисками дорожно-транспортных происшествий являются важными проблемами. Этим занимаются тысячи дорожных инженеров по всему миру. Во многих странах существуют определенные пороговые значения дорожного сцепления, которые определяют наименьшее приемлемое трение дороги. Если уровень трения ниже этого значения, то риск несчастных случаев может возрасти. Эти пороговые значения являются результатом исследования связи между скоростью центра колеса и его угловой скоростью. Цель работы – исследовать используемые методы измерений сухого трения и количественных соотношений между коэффициентами трения с дорогой, риском аварий и теоретическим подходом к их оценкам. Связь трения между дорожным полотном и колесом автомобиля с риском очевидна. Если дорога очень скользкая, то риск заноса значительно повышается, но точное определение пороговых значений коэффициента трения, которое должно задавать параметры дорожного покрытия, является непростой задачей. Данную проблему можно разделить на две части – методы измерения трения и связь между трением и риском аварии.

Ключевые слова: *сухое трение, автомобильные потоки, механика автомобильных колес.*

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