

The thermal loading estimation of the friction pairs of a vehicle automated brake system

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Abstract. The paper describes the approach for the evaluation of predesign-thermal load of the braking mechanism for vehicles with ABS.

Methodology: A method is proposed for determining the energy quenched in a braking mechanism with ABS using three approaches.

Research implications: The study was the basis for creating a computer model of the temperature field of the braking mechanism, which, on the whole, makes it possible to talk about a system for calculating the thermal loading of braking mechanisms with ABS.

Conclusions: Due to the phenomenon of non-synchronous rotation of the car's wheels during braking, due to various factors, the brake mechanisms of even one axle of the car can have different performance indicators along the friction path

The increasing availability of automated braking systems, such as antilock braking systems (ABS), in addition to the obvious advantages in terms of active safety, creates a number of problems caused by changes in the working process. In particular, this applies to the popular in recent years, high-speed cars J – class-crossovers and SUVs, which, as a rule, while maintaining high-speed mode have an increased weight compared to conventional cars. This increases the amount of kinetic energy that must be extinguished during braking.

However, in contrast to the traditional way of stopping a skid, the main part of the kinetic energy of a car with ABS is extinguished by the operation of friction in the brake mechanism, which inevitably leads to an increase in thermal loading, especially when using by the manufacturers of brake systems of cars, the traditional elements of basic models.

At the same time, it is known from literature sources that the overheating of the brake friction pairs contributes to development of a critical feding, accompanied by a sharp decrease (up to 50%) in the coefficient of friction of the brake linings, as well as increased wear of the countertops, with the formation of macroscoles [1]. Therefore, it is necessary to assess the thermal load of friction pairs of brake mechanisms of cars equipped with automated systems.

The analysis of the effect of the principal change in the working process of braking the car wheel with ABS on the distribution of the work necessary to extinguish the kinetic energy of the car as an example is shown in figure 1. The graph shows: the change in the kinetic energy of the car with ABS when braking from an initial speed of 60 km / h on dry asphalt during braking (W_k), the work spent on friction in contact with the road tire (ATR) and hysteresis losses in the tire (AFF), as well as absorbed in the brake mechanism (ATM).



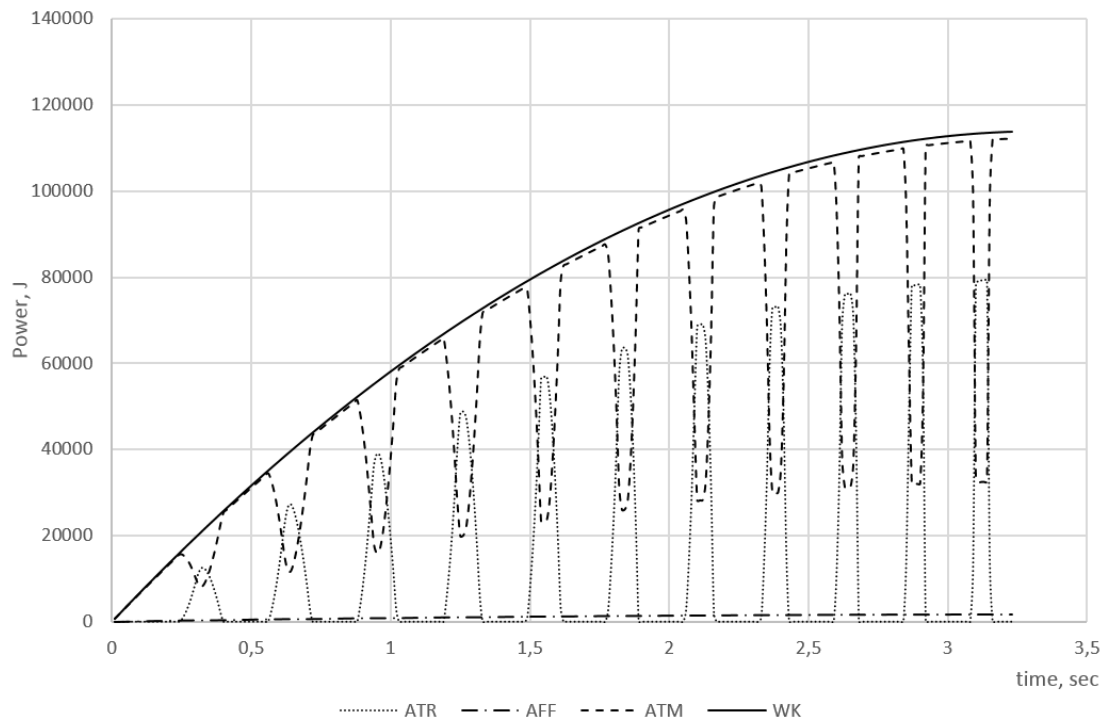


Figure 1. Distribution of work spent on damping the kinetic energy of the vehicle with ABS during braking.

The graph shows that the braking of the car with ABS, in contrast to the skid braking, the main part of the kinetic energy is extinguished by the friction work in the friction pairs of the brake mechanism. Therefore, despite the development of design and calculation technologies, the assessment of the probability of possible failure of the vehicle brakes during operation due to **feding** is an urgent task. It is important that in the design of braking vehicles with ABS manufacturers have the opportunity to assess the effectiveness of their work at the stage of pre-design calculation. The calculation of the average temperature of friction elements in the braking process is of great interest in assessing the temperature of the brake and can be particularly useful in cases where the direct measurement of temperatures at real points of contact and the friction surface as a whole is difficult.

To solve this problem, it is necessary to have computational methods that allow to analyze such performance characteristics as the change in the braking moment of friction, the temperature of the surface of the brake discs, the speed and duration of braking, as well as the work performed depending on the physical, mechanical and thermal properties of the materials of the friction pair, as.

The average temperature of friction pairs, taking into account the short duration of braking, without taking into account heat transfer to the environment, can be determined by the dependence obtained by Professor A.V. Chichinadze [2]

$$g^* = \frac{W_{T.П.}(1 - \alpha_{T.П.}'')b_1}{\lambda_1 A_{a1} t_T} \left[\frac{1}{3} \tau_N + \tau_W F_{01} \right], \quad (1)$$

where: $W_{T.П.}$ - full braking operation; $\alpha_{T.П.}''$ - the distribution coefficient of heat flow; A_{a1} - the nominal area of friction;

t_T - duration of braking; $\lambda_{1,2}$ - coefficient of thermal conductivity; τ_N , τ_W - time characteristics of power and operation; F_{01} - Fourier number of friction elements.

In turn, the coefficient of distribution of heat fluxes required for the calculation of thermal fields can be found from the expression [2]

$$\alpha''_{T.H.} = 1 / \left[1 + \frac{b_1 \lambda_2}{b_2 \lambda_1} \left(\frac{\frac{1}{3} \tau_N + F_{O2} \tau_W}{\frac{1}{3} \tau_N + F_{O1} \tau_W} \right) \right], \quad (2)$$

where $b_{1,2}$ is the thickness of friction elements.

The given dependencies allow to calculate the average temperature of brake discs with the help of finite element software [6] with the known full braking operation. Initial parameters in them are parts and Assembly of brake mechanisms, values of heat flows, heat transfer coefficients. Thus, as an example, figure 2 shows the model of temperature distribution of the brake disc of the car with ABS at a coefficient of convective heat transfer of 35 W / m²•K and ambient temperature of 294 K, obtained by using a finite element software..

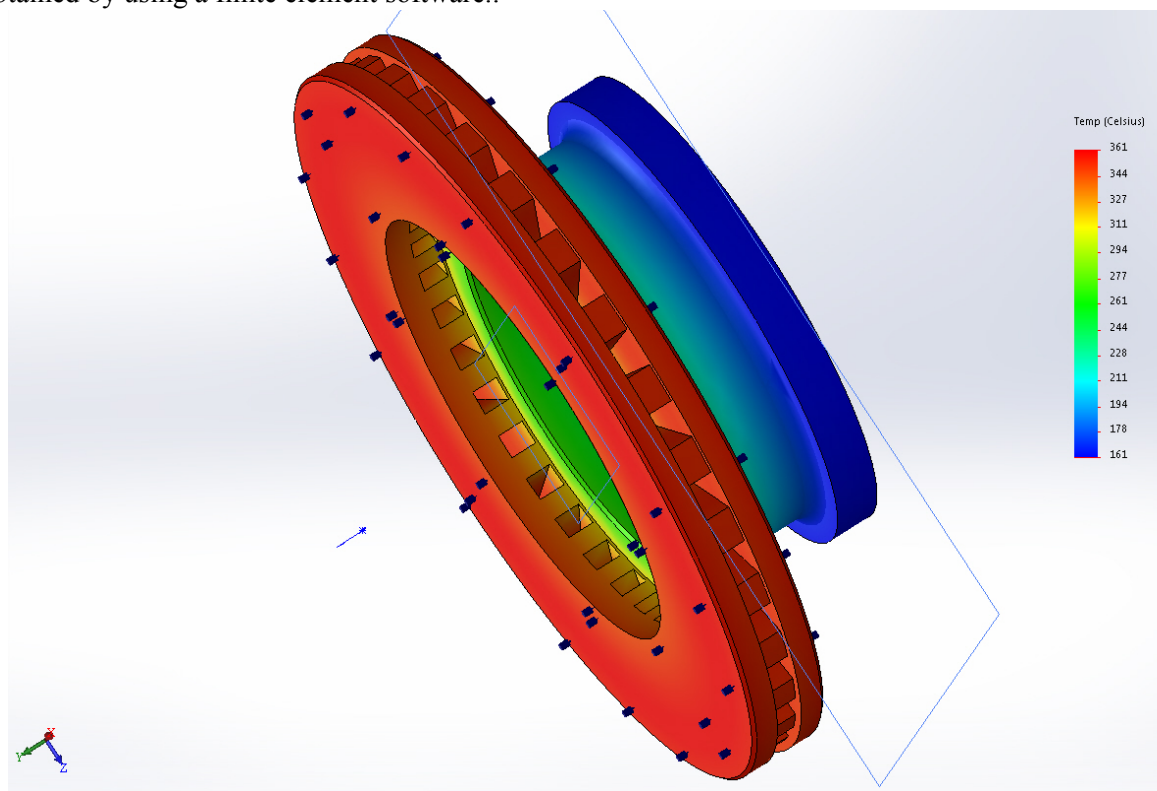


Figure 2. The model of the temperature distribution of the brake disc vehicle with ABS.

As mentioned above, for the application of the dependencies obtained by A.V. Chichinadze [2], it is necessary to find the full braking operation based on the braking force on the disc and the friction path of the brake disc. At the stage of pre-design analysis to determine the braking force, several approaches are possible. The most promising approach is based on the determination of the braking performance brought to the brake disc, which can be found with sufficient accuracy through the ratio of the longitudinal reaction and the braking torque. To do this, we associate them with the value of the dynamic radius. Despite the conventionality of this ratio, this is enough to estimate the amount of work needed. Brake torque on the wheel is determined by the known dependence

$$M_{m\partial} = R_x \cdot r_{\partial}. \quad (3)$$

To determine $\xi_{a\partial c}$ the value of the longitudinal reaction, we will use the degree of using the maximum coefficient of adhesion $\xi_{a\partial c}$ in the operation of the anti-lock braking system [3]. Then the expression for the longitudinal reaction of the wheel will take the form

$$R_x = R_z \cdot \varphi_{\max} \cdot \xi_{a\partial c}. \quad (4)$$

The value is determined from the requirements of standards for the braking performance of the car with ABS. So, for dry asphalt, it should not be below the level of the braking skid $\xi_{a\partial c} = 0,9$, wet and compact snow $\xi_{a\partial c} = 0,85$. Introduction to the calculation of this indicator greatly simplifies the calculation process and eliminates the consideration of the dependencies of the algorithm of a particular anti-lock system, since the algorithm and the matrix of values of the control unit of the anti-lock system are often a trade secret manufacturers of brake systems.

The value of the braking force on the average circumference of the brake disc is found from the expression:

$$P_m = 2 \frac{M_{m\partial}}{r_{\partial uc}} \quad (5)$$

where $r_{\partial uc}$ is the radius of the brake disc.

Taking into account the dependence (4), we obtain an expression for the braking force in the form

$$P_m = 2 \frac{R_z \cdot \varphi_{\max} \cdot \xi_{a\partial c} \cdot r_{\partial}}{r_{\partial uc}} \quad (6)$$

To find the full operation of friction in the brake mechanism, it is necessary to know the friction path in the "brake disc – pad" pair. Theoretical background to determine the path of friction of individual wheels of the vehicle in braking mode is given in [5].

The rotation of each individual wheel of the vehicle with ABS when braking is characterized by a different combination of speed modes due to the operation of the brake system and the contact of the tire with the road surface, which is shown in Fig. 3.

At the same time, the following characteristic periods can be distinguished: braking, disinhibition, blocking or use of the wheel, free rolling and quasi-uniform movement in the area $\dot{\omega}_{\kappa} \approx 0$.

Each period is characterized by a different combination of speed and load performance. Taking into account the short duration of the ABS cycle, we assume that each period is characterized by a steady slowdown (acceleration) j_{ycm} and an average speed.

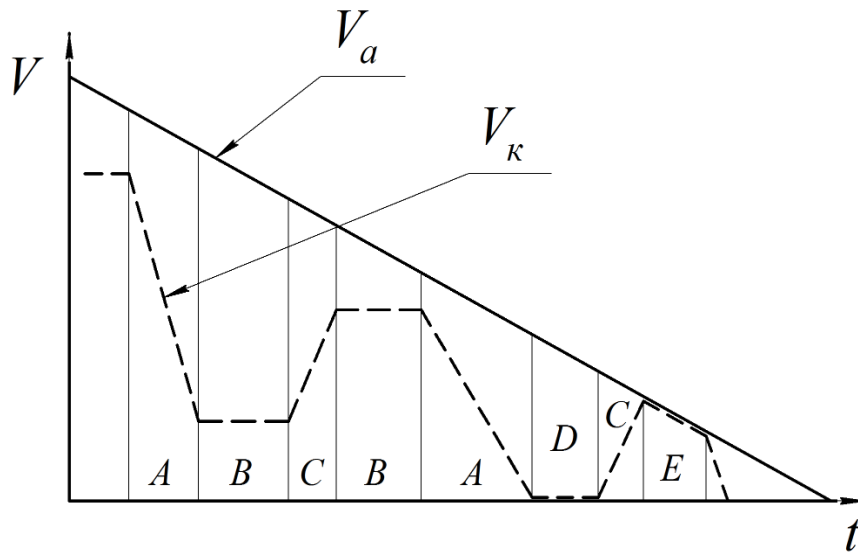


Figure 3. Period options during braking:

A - braking B - quasi-uniform rotation, C - disinhibition, D - Yuz, E - free rotation (full release), V_a – vehicle speed, V_k – given radial rotation speed of the wheel.

With the same values of the braking distance, cars with ABS and without it can be implemented different characteristics for friction pairs "drum (disc)-pad" and "bus-road". To calculate the friction path in each pair of brake mechanisms "disc-pad" use the dependence

$$L_{Tp} = (V_{H,K} + V_{K,K})(t_{K,K} - t_{H,K})(r_{\bar{o}} / r_K) / 2, \quad (7)$$

$$\text{or } L_{Tp} = (V_{H,K}^2 - V_{K,K}^2) / 2j_{ycm,K};$$

where: $V_{H,K}$, $V_{K,K}$ - the radial speed of the wheel at the beginning and end of the time interval;
 $t_{H,K}$, $t_{K,K}$ - the moments of time at the beginning and end of the time interval; $r_{\bar{o}}$ - the radius of the brake drum; r_K - the radius of the wheel.

The friction paths for each period are determined from the following dependencies.
 The period of inhibition and disinhibition

$$L_{Tp} = (V_{H,K}^2 - V_{K,K}^2) / 2j_{ycm,K}, \quad (8)$$

where: $V_{H,K}$, $V_{K,K}$ - wheel speed at the beginning and at the end of the period, $j_{ycm,K}$ - the steady-state acceleration of the wheel, r_K - wheel radius, $r_{\bar{o}}$ - the average radius of the brake disc.

Free wheel rolling $L_{Tp} = 0$,

Wheel lock $L_{Tp} = 0$

Constant (quasi-uniform) rotation of the wheel

$$L_{Tp} = V_{H,K} \cdot t_{\Pi} \quad (9)$$

where: t_{Π} - the duration of the period of uniform rotation of the wheel.

To find the General friction path in all areas, let's sum the above dependencies

$$L_{Tp,mM} = \sum_{i=0}^I L_{TpTi} + \sum_{j=0}^J L_{TpPj} + \sum_{n=0}^N L_{Tp\Pi n}, \quad (10)$$

where: $L_{mp, mm}$ - the friction path of the lining of the drum; I - the number of braking areas; J - the number of areas of disinhibition; N - the number of areas of constant rolling of the wheel;

L_{TpTi} - the friction path for a pair of "disc-pad" on the braking area;

L_{TpPj} - way friction pair "disc pad" on the j-th section of the release;

$L_{Tp\Pi n}$ - the friction path for the pair "disc-pad" on the n-th section of the wheel constant rolling braking.

Finally, the friction work brought to the brake disc is determined from the expression:

$$W_{T.П.} = L_{Tp} P_m, \quad (11)$$

Due to the phenomenon of non-synchronous rotation of the wheels of the car during braking, due to various factors, both in the presence of ABS and without it, the brake mechanisms of even one axis of the car can have different indicators of work on the friction path.

It should be noted that the definition of the full braking performance and, as a consequence, the determination of the energy quenched in the brake mechanism was the basis for the creation of a computer model of the temperature field of the brake mechanism, which generally makes it possible to talk about the system for calculating the thermal loading of the brake mechanisms with ABS.

References

- [1] Bezyazychnyy V.F., Lyubimov R.V., Timofeyev. 2000. Eksperimentalnoye izucheniye protsessov razrusheniya poverkhnostnykh sloev metalla pri ustanovivshemsya protsesse fretting-iznashivaniya. *Sb. nauchn. trudov Tverskogo gos. tekhn. un-ta «Mekhanika i fizika friktsionnykh kontaktov»*. vyp. 7. p.24-28.
- [2] Chichinadze A.V. 1970. Teplovaya dinamika treniya. 171 p.
- [3] Revin A.A., Dygalo V.G., 2014. Formirovaniye osnovnykh ekspluatatsionnykh svoystv avtotransportnykh sredstv v rezhime tormozheniya. *Avtomobilnaya promyshlennost*. **11**. p. 3-5.
- [4] Revin A.A., Poluektov M.V., Radchenko M.G., Zabolotnyy R.V. 2013. Vliyaniye rabocheho protsessa ABS na dolgovechnost elementov shassi avtomobilya: monografiya. 224 p.
- [5] Tumasov A.V., Groshev A.M., Kostin S.Yu., Saunin M.I., Trusov Yu.P., Dygalo V.G., 2011. Issledovaniye svoystv aktivnoy bezopasnosti transportnykh sredstv metodom imitatsionnogo modelirovaniya. *Zhurnal avtomobilnykh inzhenerov*. **2**. p. 34-37.
- [6] Turbin I.V., Epishkin V.E., Solomatin N.S. 2014. Vliyaniye koeffitsiyenta treniya na tribotekhnicheskiye kharakteristiki par treniya diskovogo tormoza. *Materialy konferentsii: «Perspektivnyye napravleniya razvitiya avtotransportnogo kompleksa : sbornik statey VIII Mezhdunarodnoy nauchno-proizvodstvennoy konferentsii.»* p. 124-128.
- [7] S. Voloaca. Concerns regarding temperature distribution obtained by experiments and finite element analyses for types of brake discs/ S. Voloaca, G. Fratila/ U.P.B. Sci, Bull., Series D, Vol. 74, Iss. 3, 2012. – p. 33