RESEARCH OF THE CAR BRAKING PROCESS

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Abstract

This paper describes analysis of a car braking process – the peculiarities of car wheel-to-road adhesion, the influence of distribution of braking forces on car stability between front and rear axles. Structural designs of braking systems are analyzed with respect to their meeting the EU standards.

Experimental measurements of braking force coefficients for some models of cars are presented with the analysis how these coefficients meet the EU standards.

Two kinds of stability (trajectory and course)in the paper has been analyzed. The first one characterizes the car ability of copying a presettled curvilinear trajectory of motion and the second characterizes the ability to retain the straight-line motion. Diagrams of braking process dependence between braking forces of front and rear axles are presented. Breaking diagrams for several cars are presented. Average braking force coefficient was expressed by the dependence upon the deceleration coefficient for the front axle and the rear axle.

Keywords: *transport, vehicles, safety, braking system, braking force coefficients*

1. Introduction

The present paper describes some theoretical foundations of the car braking stability and technical requirements for its achievement. On the basis of these experimental results characteristics determining the stability of car braking are evaluated.

2. Concepts and definitions

The ratio of longitudinal and perpendicular road reactions to a wheel is called the longitudinal force coefficient [4]:

$$\mu_x = \frac{F_x}{F_z}. \qquad (1)$$

The ratio of maximal longitudinal and perpendicular loading forces is called the maximum longitudinal force coefficient $\mu_{x\,max}$:

$$\mu_{x\,max} = \frac{F_{x\,max}}{F_z}. \qquad (2)$$

Potential properties of a wheel in accordance with the adhesion with the road can be characterized by the longitudinal adhesion utilization coefficient, which equals the ratio of the longitudinal force coefficient and the maximum longitudinal force coefficient:

$$\varepsilon_x = \frac{\mu_x}{\mu_{x\,max}}. \tag{3}$$

This coefficient does not exceed 1,0 (for ideal braking $\varepsilon_x = 1{,}0$).

The longitudinal force coefficient for every axle of a car which is being braked can be expressed as follows:

$$\begin{aligned} \mu_{xf} &= \frac{B_f}{F_{zf}}, \\ \mu_{xr} &= \frac{B_r}{F_{zr}}, \end{aligned} \tag{4}$$

where: B_f – braking force of front axle; B_r – braking force of rear axle; F_{zf} – perpendicular road reaction to wheels of front axle; F_{zr} – perpendicular road reaction to wheels of rear axle.

Perpendicular dynamic road reactions to car axles are defined by the following equations:

$$\begin{aligned} F_{zf} &= F_{zfstat} + mg\frac{h}{l}z = mg\left(\frac{l_r}{l} + \frac{h}{l}z\right), \\ F_{zr} &= F_{zrstat} - mg\frac{h}{l}z = mg\left(\frac{l_f}{l} - \frac{h}{l}z\right), \end{aligned} \tag{5}$$

where: F_{zfstat}, F_{zrstat} – static reactions to front and rear axles; m – mass of the car; $g = 9{,}81\,m/s^2$ – free fall acceleration; l_r, l_f – distances between car mass center and corresponding axles; l – wheelbase; h – height of mass center over the road surface; z – deceleration coefficient.

The deceleration coefficient is as follows:

$$z = \frac{a_x}{g}, \tag{6}$$

where: a_x – car deceleration m/s^2.

Accordingly the deceleration coefficient can be expressed by use of the braking forces:

$$z = \frac{B_f + B_r}{mg}. \tag{7}$$

3. Cases of the car braking stability

There are two kinds of stability: 1) trajectory and 2) course. The first one characterizes the car ability of copying a presettled curvilinear trajectory of motion and the second characterizes the ability to retain the straight-line motion.

Let us examine the cases which can occur during braking [2]:

1. The loss of trajectory stability as the result of front wheels locking. In such situation the wheels loss the opportunity of transmission of lateral forces and the car becomes uncontrollable and runs practically straight forward irrespective of the position of steering wheels. The locked front wheels in such case do not compensate the lateral forces and a sidelong slide of the front axle with the velocity V_{sl} (see Fig. 1., b) can took place. Then the front axle begins the motion around the turning center O at velocity $V = \sqrt{V_a^2 + V_{sl}^2}$. The component F_y of the centrifugal force F_c reduces the sidelong slide of the front axle, therefore the car stability is regained as the braking interrupts.

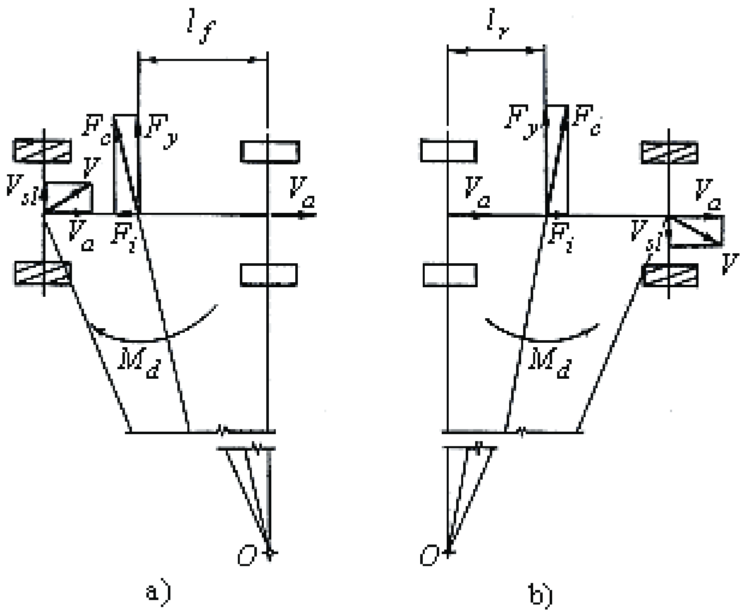

Fig. 1. Critical cases of car braking: a) locked rear wheels; b) locked front wheels

2. The loss of course stability occurs when the rear wheels become locked. In such case the force F_y do not suppresses the rear axle slide, but intensifies it (see Fig. 1., a). Therefore the destabilizing torque $M_d = F_y \cdot l_f$ increases and the slide grows.

It is known [2] that if the wheels of the rear axle lock and the car deviates about 20^o, it becomes unstable and even after the interruption of braking the car response not adequate to control actions is observed.

The perpendicular dynamic road reaction to wheels F_z (see equations (5)) depends on car mass, location of its mass center, structure of suspension, character of car motion (e.g. the braking intensity).

In braking the loading of the front axle increases and correspondingly the loading of the rear axle decreases at the same level. As it is seen from the probabilistic point of view wheels of the rear axle can be locked first because of considerable decrease of the road-to-wheel reaction F_z and condition of wheel stability can be disturbed. The condition can be expressed as follows:

$$F_\Sigma = \sqrt{F_x^2 + F_y^2} \le \mu_{max} \cdot F_z , \qquad (8)$$

where: F_y – cross directed road-to-wheel reaction.

As it seen from the last equation, the greater F_x causes the more intensive braking of the wheel and less car stability.

4. Structural principles of stability assurance

For prevention of rear wheels locking the braking system need to be so designed, that deceleration coefficient for the case $\mu_x = 0{,}2 \ldots 0{,}8$ would be as follows [1, 5]:

$$z \geq 0{,}1 + 0{,}85(\mu_{x\,max} - 0{,}2)\,. \quad (9)$$

Then the longitudinal adhesion utilization coefficient must meet the following requirement:

$$\varepsilon_x = \frac{z}{\mu_{x\,max}} > 0{,}85 - \frac{0{,}07}{\mu_{x\,max}}\,. \quad (10)$$

In graphical expressions of the presented inequalities the adhesion straight line of the front axle have to be over the corresponding line of the rear axle (see Fig. 2.).

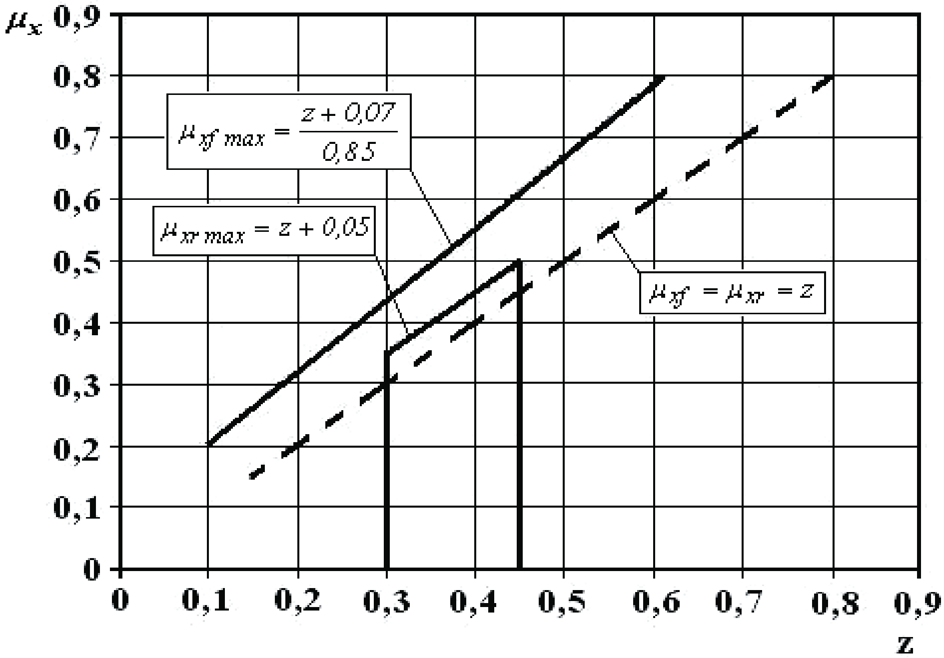

Fig. 2. Requirements for M_1-class car braking

For cars of M_1-class , when the adhesion coefficient μ_x is in the limits $0{,}15 \ldots 0{,}8$, the longitudinal force coefficient of front wheels must be equal to or higher as compared with such coefficient of rear wheels. But for such cars $\mu_{xr} > \mu_{xf}$ is allowed when $z = 0{,}3 \ldots 0{,}45$. In such a case the longitudinal force coefficient must meet the condition $\mu_{xr} \leq z + 0{,}05$ (see Fig. 2.). The conventional regulators of braking forces assured such characteristics of the examined cars.

If the car has no regulator of braking forces, then the ratio of braking forces of the front and rear axles is constant – $B_f \,/\, B_r = const.$

For such structure of the braking system graphs of the longitudinal force coefficients of the front and rear axles have the form shown in Fig. 3. [3].

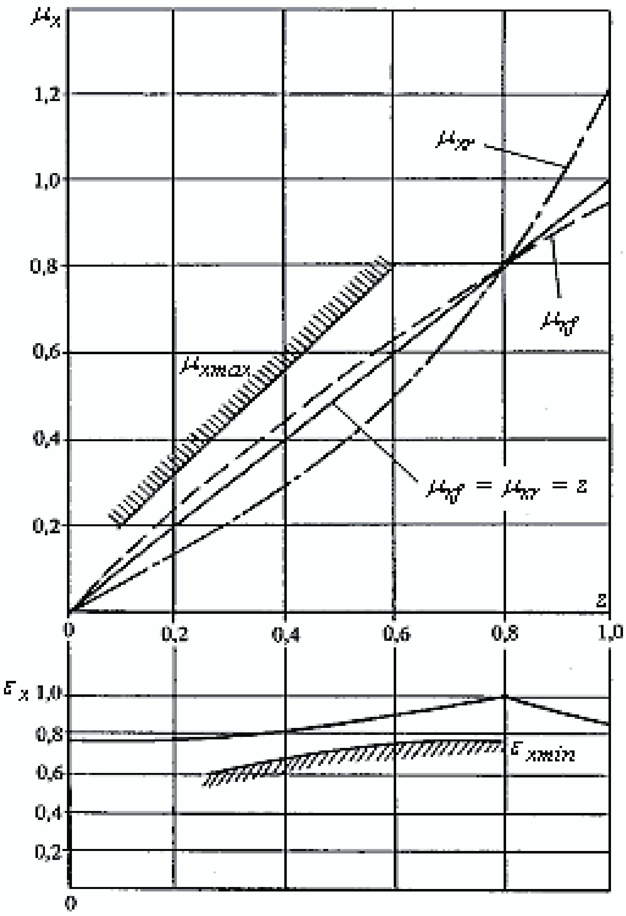

Fig. 3. Diagrams of braking process in the case of a linear dependence between braking forces of front and rear axles

Up to the point $z = 0{,}8$ $\mu_{xf} > \mu_{xr}$. If $z = 1{,}0$, then the probability of locking is greater for the rear wheels. If longitudinal force coefficients for both axles are the same and are equal to z, i.e. $\mu_{xf} = \mu_{xr} = z$, then we have the ideal braking. Really the car braking system can be designed for ideal braking only for a dynamic situation, when $\mu_x = 0{,}8$, $z = 0{,}8$; the car is fully loaded. Then $\mu_{xf} = \mu_{xr} = z = 0{,}8$; $\varepsilon_x = 1{,}0$, i.e. wheel-to-road adhesion is used at *100%* (see Fig. 3.) and both axles of the car are being braked equally.

If the wheels brake the car with the same efficiency, efforts are being made to draw nearer the graphs of coefficients μ_{xf} and μ_{xr} to the ideal one. For this reason the need of special means arises enabling a proper ratio of pressures in the braking systems of the front and rear wheels. Because of the increased danger of locking of rear wheels (if compared with the front ones) attempts are being made for retaining the lower pressure in the braking system of rear wheels, if pressure in the braking system of front wheels exceeds some limit during the emergency braking. Some kinds of braking force regulators are used for this purpose.

The use of such regulators modifies the form of diagrams of the braking process (see Fig. 4.) – in the diagram of μ_{xr} a brake is seen, which draws nearer this diagram and characteristic of ε_x to the ideal one.

Efficiency of braking force regulators is very high for small wheelbase cars with a highly located mass center and in the case of great difference between static and dynamic perpendicular loading.

The operation of such regulators is influenced by the accuracy of installation, residual deformation of suspension (e.g. springs), turning of rear axle during braking a.o. Therefore the goal of the carried out experiments was to determine the characteristics of braking force regulators for the used cars, i.e. to evaluate how much the regulation characteristics vary during the car exploitation.

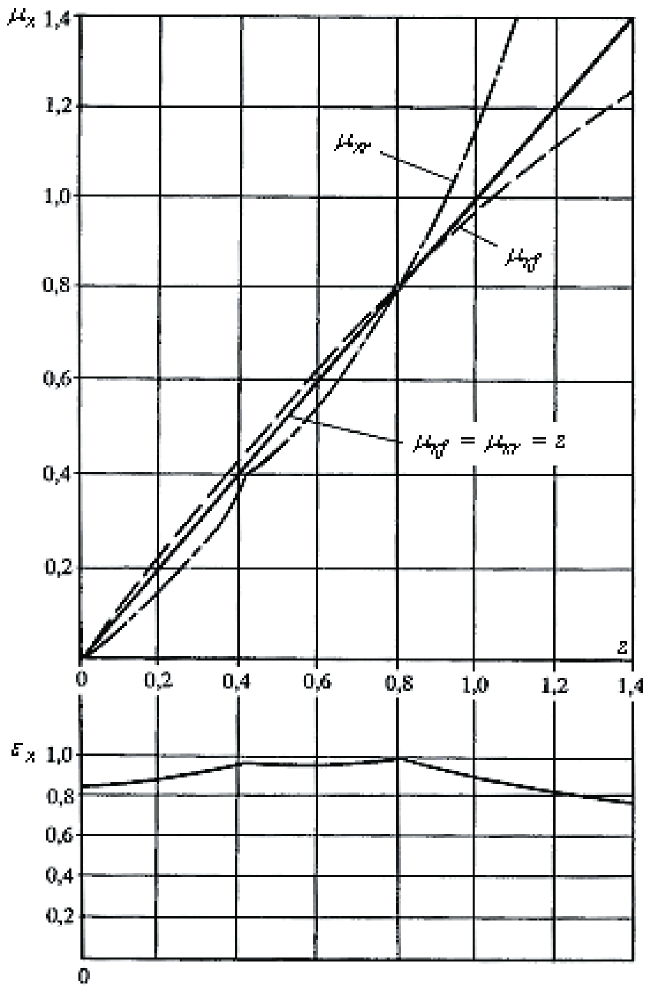

Fig. 4. Diagrams of braking process with braking force regulation

5. Experiments of car braking efficiency

With the goal of examination how the main car braking systems meet the 71/320/EEC Directive the longitudinal force coefficients μ_{xf} and μ_{xr} for front and rear axles were measured in 26 cars. The results showed which axle locked first during braking.

As braking by the main brakes goes on dynamically the dynamic experiment allows better discovering of imperfections characteristic to such braking, therefore the experiments were carried out on the plate-type test bench for brake testing HEKA UNIVERS A2. The braking deceleration was measured by the portable deceleration metering instrument.

The longitudinal force coefficients μ_{xf} and μ_{xr} for front and rear axles were calculated in accordance with formulae (4).

Expressions for the calculation of perpendicular dynamic road reactions to front and rear axles can be obtained from (5):

$$F_{zf} = m_f \cdot g + a_x \cdot \frac{h}{l} \cdot m,$$
$$F_{zr} = m_r \cdot g - a_x \cdot \frac{h}{l} \cdot m, \tag{11}$$

where: m_f and m_r – masses of car front and rear axles, kg; m – car mass, kg; h/l – ratio of height of car mass center and car wheelbase (in experiments the average statistical value of h/l for all cars was accepted equal to $0{,}225$).

The obtained values of coefficients μ_{xf} and μ_{xr} were compared with each other and decisions were made about car stability during braking and about the meeting of braking forces the EU standards.

The determined characteristics of 10 cars meet the regulations of the EU Directive very well (e.g. FORD ESCORT car – see Fig. 5.). 8 cars (three HONDA CIVIC's among them) were found having the excessive braking in the front axle (Fig. 6.). The same number of cars (8, among them three LADA's) were found having the reverse effect – rear wheels tended to lock first (Fig. 7.).

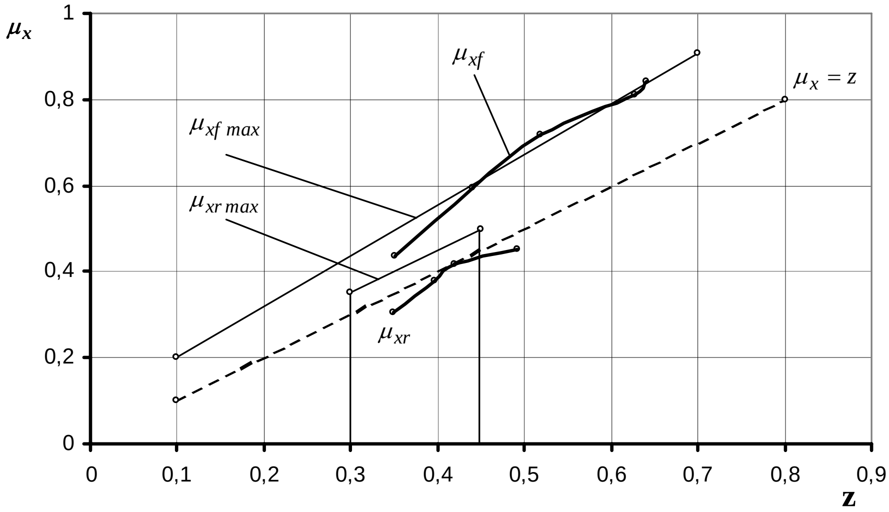

Fig. 5. Braking diagrams of a FORD ESCORT car

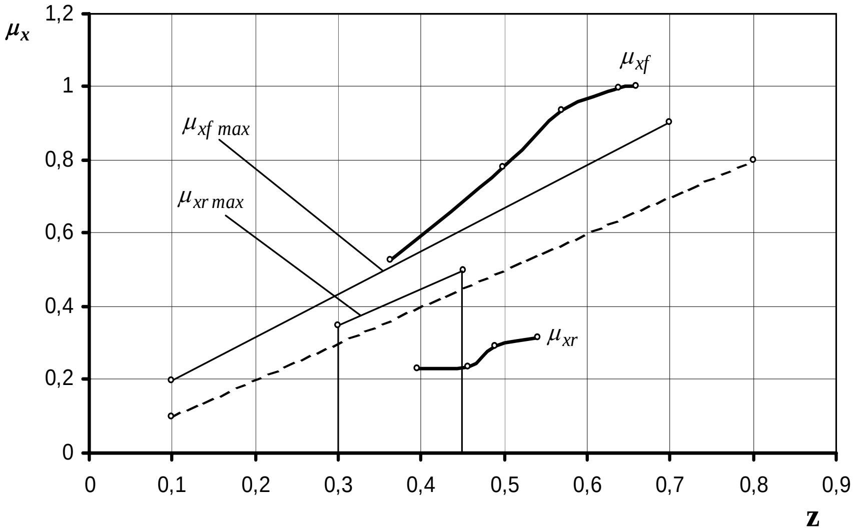

Fig. 6. Braking diagrams of a HONDA CIVIC car

Values of μ_{xf} and μ_{xr}, obtained in the experiments are shown in Fig. 8. The comparative dissipation of braking efficiency for front and rear axles is seen for these cars together with the straight line $\mu_x = z$ of ideal braking.

Based on this statistical results it was established (by the linear regression) that the average braking force coefficient can be expressed by the following dependence upon the deceleration coefficient for the front axle of the studied cars:

$$\bar{\mu}_{xf} = 1{,}4 \cdot z - 0{,}013, \tag{12}$$

and correspondingly for the rear axle:

$$\bar{\mu}_{xr} = 1{,}61 \cdot z - 0{,}22. \tag{13}$$

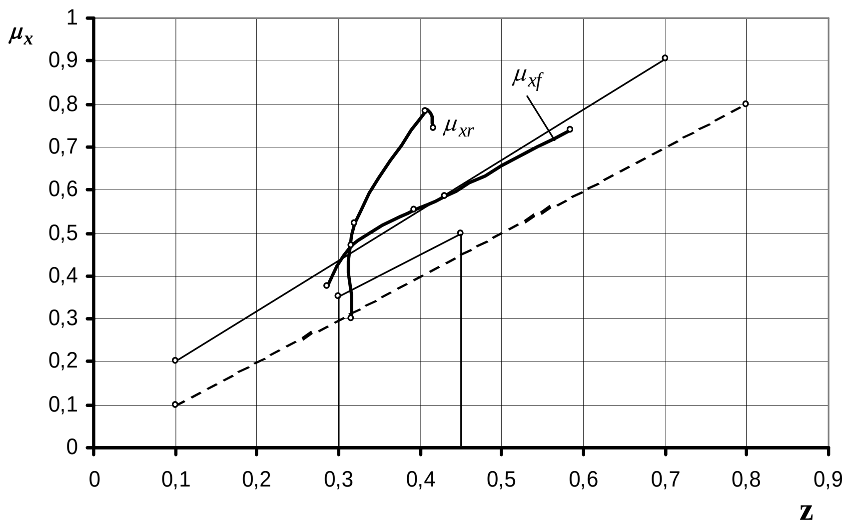

Fig. 7. Braking diagrams of a LADA 2101 car

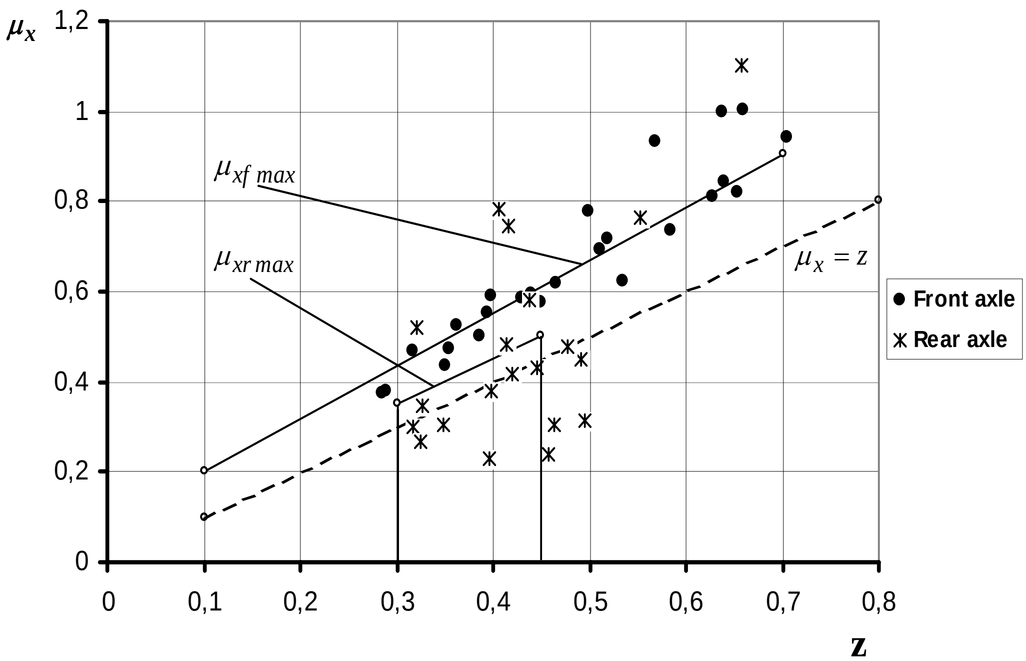

Fig. 8. Distribution of longitudinal force coefficients

6. Conclusions

1. Among the examined used cars only for 40% of them the braking forces coefficient meet the requirements of the EU Directive 71/320/EEC, while other cars were with defects of the braking system – too large longitudinal force coefficient of wheels of the front (μ_{xf}) or of the rear (μ_{xr}) axles.

2. The analysis of average values of μ_{xf} and μ_{xr} established that the front wheels braked about $1{,}5$ time more intensively than the rear ones.

References

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[5] Regulation No. 13. Uniform provisions concerning the approval of vehicles of categories M, N, O with regard to braking. Geneva, 1995.