Model for estimation of the friction coefficient in automotive brakes under extremely high temperatures

Aleksandar Grkic, Slavko Muzdeka, Zivan Arsenic, Cedomir Duboka

***Abstract*— Extremely high temperature achieved in the contact surface between the disc and the pad during braking has a large impact on the brake performance. Unintended consequence of extremely high temperature can cause complete brake malfunction, so called fading. There are a number of mathematical models describing the main tribological properties of various materials, including friction materials used as brake linings or pads. The application of these models for prediction or estimation of friction properties under determined working conditions is, however generally very limited. It is demonstrated in the paper by an example which has rather high practical importance, in order to shorten the experimental test.**

***Index Terms*—braking, temperature of friction surface, coefficient of friction, mathematical modeling.**

I. INTRODUCTION

Numerous research showsthat the temperature achieved during braking on the contact surface between the disc and the pad has a large impact on the coefficient of friction [1-5]. Extremely high temperature, finally, can cause complete brake malfunction, so called fading [6]. A large number of authors used different mathematical methods in trying to describe a coefficient of friction in the contact zone of the friction pair, but a great majority of them are often very complex and contain a number of limitations [7-10]. The functional characteristics of friction materials for motor vehicle brake linings determine the brake performance as well as the behavior of motor vehicle during braking. It is also well known, that the best way to determine operational characteristics of friction materials is purely empirical that is to say experimentally. However, one knows that this method in the specific research area is lengthy and expensive.

Therefore, theoretical and mathematical determination of the fundamental tribological characteristic of friction materials in function tribological parameters, is of prime importance, due to fact that time and auxiliary expenses are radically reduced. In hitherto published works [11-14], sufficient space is given to the application of the mathematical modeling, predicting the operational characteristics of friction materials used for motor vehicle brake linings. This work has no intention to show explicitly the method of the actual mathematical modeling, but to direct itself on explaining how to develop the specific tribological model based on short term experimental results obtained from laboratory test of motor vehicle friction pads, on inertia dynamometer, as well as to explain the potential exploitation of the resulting tribological model in predicting the nature of friction under extremely high operating temperatures.

Manuscript received November 22, 2014.

Aleksandar Grkic, Department of Military Mechanical Engineering, Military Academy University of Defence, Belgrade, Serbia, +381658445135.

Slavko Muzdeka, Department of Military Mechanical Engineering, Military Academy University of Defence, Belgrade, Serbia, +381658445135.

Zivan Arsenic, Automotive Department, Faculty of Mechanical Engineering, University of Belgrade, Belgrade, Serbia, +381695937109, **Cedomir Duboka**, Belgrade, Serbia, +381659871070,

II. EXPERIMENTAL RESEARCH

Experimental research was undertaken using frictional brake pad lining of motor vehicle, at Single-end full-scale inertial dynamometer PSK-20, developed at the Faculty for Mechanical Engineering, University of Belgrade. The outlook of the inertia dynamometer is shown in Fig.1. This particular inertia dynamometer is universal, i.e. it could be applied in testing the whole range of different brakes built in motor vehicles-cars, vans pick-ups etc. Single-end full-scale inertial dynamometer PSK-20 consists of three basic groups:

1. the propulsory group,
2. combination of different flywheels and
3. the part in which the brake under the test is mounted.

As shown in Figure 2 single-end full-scale inertial dynamometer PSK-20 is arranged ‘in-line'. A system of six flywheels (3) is mounted onto a common central shaft (4) to simulate the mass of the test vehicle. Variable inertia is obtained by engaging the appropriate flywheels, 10-200 kgm^2. Precise inertia simulation is obtained by ‘trimming' using the DC electric motor (1), with continuous regulation of angular speed which drives flywheels via a periflex clutch (2). The brake disc (8) is with flange (5) firmly attached to the shaft (4), while the brake calliper (9) is mounted for fixed flange (6) which is firmly connected to the foundation (7). The full-scale inertial dynamometer has been used for strictly controlling brake’s operating conditions namely application pressure, initial speed, and temperature at the contact surface. Whole system is equipped with a PC-based automatic control and data.

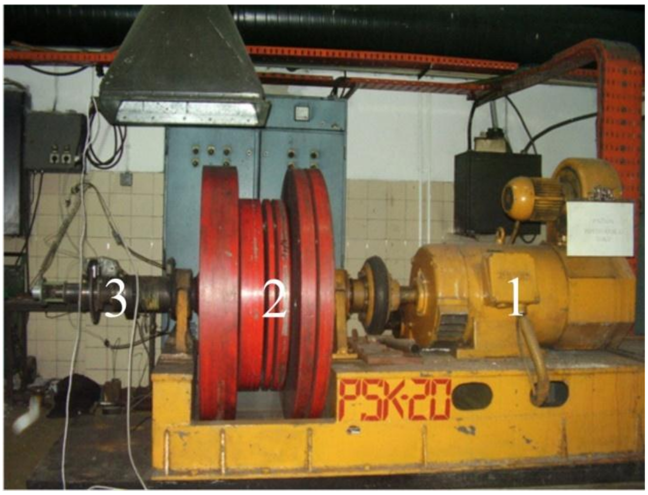


Figure 1. Single-end full-scale inertial dynamometer

Control unit manages angular speed and control line pressure, while data acquisition system measures: application pressure, speed, friction surface temperature, and braking torque. Brake torque is measured by strain gages, control line pressure by pressure transducers, speed by an optical encoder and temperature using thermocouple.

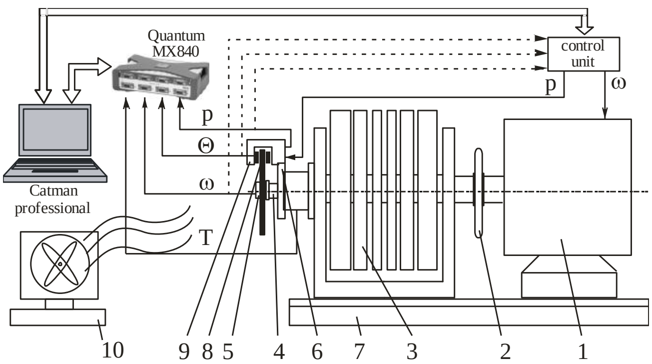


Figure 2. Principal scheme of the measuring system for brake testing

In this way repeatability of brake's operation conditions for different types of friction material have been provided. The experimental research programmed consisted of two parts. The first part of the research comprehended the evaluation of the relationship between the friction coefficient (μ) and activating pressure (p), speed (v) and temperature (θ), so that the experimental results for the friction coefficient could be obtained, and used in the development of the specific friction model. In experiment were used 3 different samples brake pads (3 different producers) of same brake. Within the frame of experiments, 27 brake applications were performed, using different values of the influential parameters namely activating pressure, speed and temperature. The magnitudes of those parameters correspondent to the real braking conditions, experienced by the friction pad lining on a motor vehicle. The results are shown in table 1. As it can be seen in Table 1, measurements were carried out during 27 consecutive (or repeated) full-stop brake applications with an initial brake speed of 30, 60 and 90 km/h, control line pressure of 20, 40 and 100 bar and initial brake temperature of 50, 100 and 150 °C. It is done in order to provide relatively average braking regimes under different brake's friction surface temperature regimes. In each brake application, brake disc was first accelerated until reaching the predetermined initial brake speed and then braked to the full stop. After completing the braking, brake disc remains stand still. Speeding up of the disc for the next brake application starts few instances before the brake reaches predetermined initial temperature.

Table 1. Results on friction tests

Line pressure	Initial brake speed	Mean interface temperature	Coefficient of friction EXPERIMENTAL (-)		
bar	km/h	°C	Pad 1.	Pad 2.	Pad 3.
20	30	50	0,413	0,425	0,395
40	60	50	0,405	0,431	0,415
100	90	50	0,452	0,441	0,424
20	30	100	0,405	0,420	0,392
40	60	100	0,425	0,435	0,418
100	90	100	0,453	0,438	0,422
20	30	150	0,375	0,421	0,375
40	60	150	0,425	0,436	0,407
100	90	150	0,435	0,445	0,421

In the second part of the experimental research, 63 brake applications were performed with the same activating pressures and speed, in the first part, while the operating temperatures were considerably above those normally developed in the process of braking on a vehicle under typical conditions (from 200 °C to 500 °C). The results from this section of the research are shown in table 2.

Table 2. Results on friction tests

Line pressure	Initial brake speed	Mean interface temperature	Coefficient of friction EXPERIMENTAL (-)		
bar	km/h	°C	Pad 1.	Pad 2.	Pad 3.
20	30	200	0,392	0,415	0,388
40	60	200	0,425	0,419	0,408
100	90	200	0,442	0,444	0,435
20	30	250	0,391	0,405	0,392
40	60	250	0,406	0,420	0,395
100	90	250	0,452	0,429	0,414
20	30	300	0,399	0,402	0,365
40	60	300	0,406	0,422	0,385
100	90	300	0,428	0,435	0,421
20	30	350	0,389	0,386	0,385
40	60	350	0,411	0,415	0,406
100	90	350	0,435	0,445	0,411
20	30	400	0,392	0,403	0,385
40	60	400	0,415	0,418	0,399
100	90	400	0,425	0,425	0,415
20	30	450	0,371	0,376	0,376
40	60	450	0,405	0,413	0,395
100	90	450	0,427	0,441	0,425
20	30	500	0,369	0,404	0,375
40	60	500	0,395	0,411	0,401
100	90	500	0,422	0,438	0,416

III. MATHEMATICAL MODELLING OF FRICTION

The aim of the first part of the mathematical modeling of friction was to find out the relationship between the friction coefficient (μ) and activating pressure (p), speed (v) and temperature (θ). To achieve this purpose the theory of experimental tests was used. The modern theory of experimental tests consists of the statistical multifactor analysis plans. These multifactor plans can be used for:

- the mathematical modeling of phenomena, processes and systems in space and time,
- the study of the nature and mechanisms of internal processes, and
- the optimization and optimal control processes in technical systems.

Multifactor experimental design is characterized by two main features:

- minimum number of experimental points, resulting in much lower costs and shorter duration of experimental investigations of processes and systems, and
- maximum of information on the effects of a mathematical model of the process.

Mathematical model of friction is realized using a statistical method known as regression analysis. Regression analysis deals with setting up a stochastic model of the object of research, which explains the condition and behavior of the given object, in a sufficiently reliable manner, within the covered experimental space. Regression analysis shows a form of connection between two variables using the regression line, which is determined by this condition: the sum of squares of the vertical distances of the points to the direction is the smallest, the so called method of the least squares. The least square method is an arithmetic procedure which allows adjusting the pre selected shape of functions to empirical distribution of the data and defining its parameters minimizing the sum of squared deviations of the functions of the empirical data sets. This method, however, does not reveal the shape of the function which is best suited to empirical data, but allows a choice between the functions of a given shape which suits best. Results obtained in any experimental research allow to determine interrelation between the test results in mathematical form. In planning of the experiment, there are two cases:

- mathematical model of tests phenomenon or system is known and based on previous knowledge of the limited experimental area, or
- mathematical model is unknown

The actual analytical form of response function is actually unknown, and the mathematical form is a more or less accurate model of it. During the research, a numerous mathematical models of friction are formed. The coefficient of friction during braking of motor vehicles, is variable and depends on many factors, which is well known. Basically, influential factors on the functional characteristics of friction materials for brakes of motor vehicles can be classified into two groups. The first group consists of factors that depend on the operating conditions or methods of use of the brake system (pressure, temperature and speed), and the second group consists of construction impacts, friction material and metal elements. With this in mind it is assumed that the appropriate mathematical model of the friction coefficient applied can be a stepped function of the third order in these forms:

$$\mu_1 = k_1 \cdot p^{A_1} \cdot v^{B_1} \cdot \theta^{C_1} \qquad (1)$$

$$\mu_2 = k_2 \cdot p^{A_2} \cdot v^{B_2} \cdot \theta^{C_2} \qquad (2)$$

$$\mu_3 = k_3 \cdot p^{A_3} \cdot v^{B_3} \cdot \theta^{C_3} \qquad (3)$$

Where:

- p - activating pressure,
- v – speed,
- θ – temperature,
- k_1, k_2, i k_3 – constants that depend on the structure of the applied friction material and applied metal element and
- A_1, A_2, A_3, B_1, B_2, B_3, C_1, C_2 i C_3 coefficients to be determined experimentally.

The assumed function of the coefficient of friction must be expressed logarithmically

$$\ln \mu_{1,2,3} = \ln k_{1,2,3} + A_{1,2,3} \ln p + B_{1,2,3} \ln v + C_{1,2,3} \ln \theta \quad (4)$$

and apply a method of the least squares to obtain constants $k_{1,2,3}$, and coefficients $A_{1,2,3}$, $B_{1,2,3}$ and $C_{1,2,3}$. Using the experimental result for the friction coefficient obtained in the first part of the research, the specific method of mathematical modeling was conducted, with the following resulting expression:

Pad 1.: $\mu_1 = 0{,}372 \cdot p^{0{,}068} \cdot v^{-0{,}017} \cdot \theta^{-0{,}0202}$ (5)

Pad 2.: $\mu_2 = 0{,}384 \cdot p^{0{,}07} \cdot v^{-0{,}02} \cdot \theta^{-0{,}195}$ (6)

Pad 3.: $\mu_3 = 0{,}365 \cdot p^{0{,}063} \cdot v^{-0{,}016} \cdot \theta^{-0{,}0178}$ (7)

IV. ANALYSIS OF OBTAINED RESULTS

The fidelity of such friction model is illustrated in fig.3, where one can clearly see that the differences between the friction coefficients obtained theoretically and experimentally differ quite negligibly. Figure 3 shows the results comparison of experimental and theoretical values of the coefficient of friction on the first sample of pads. In all 27 brake applications differences were below 3%. It is also important to note that the friction model shown by the expression (1, 2, 3) characterizes the behavior on the friction coefficient in function of the influential parameters (p, v, θ).

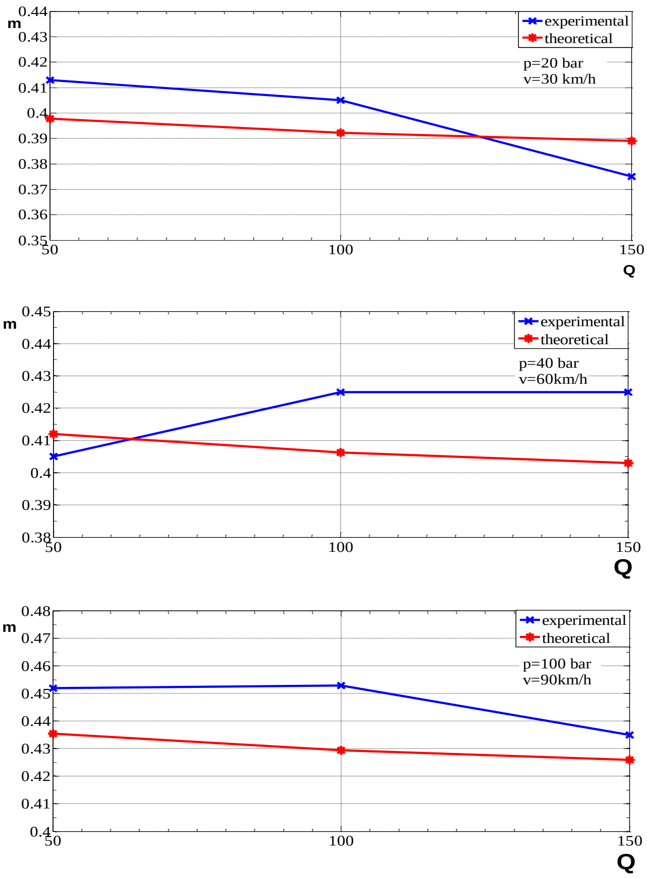
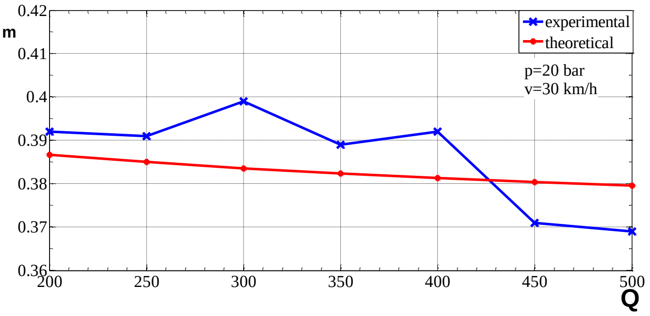
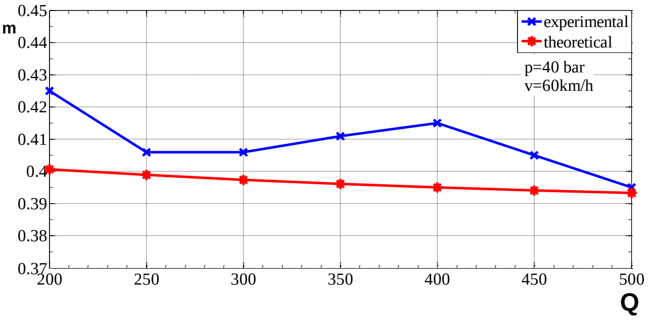


Figure 3. Comparison of experimental and theoretical values of the coefficient of friction

After verification of the mathematical model of friction by comparing theoretical and experimental results carried out is estimation the value of the coefficient of friction under extremely high temperature. By comparing the results obtained in the second part of the experimental research, shown in table 2, and the results using expression (1, 2, 3), it can be seen that the actual error is less than 3%, which is satisfactory. This can be clearly seen on fig. 4.





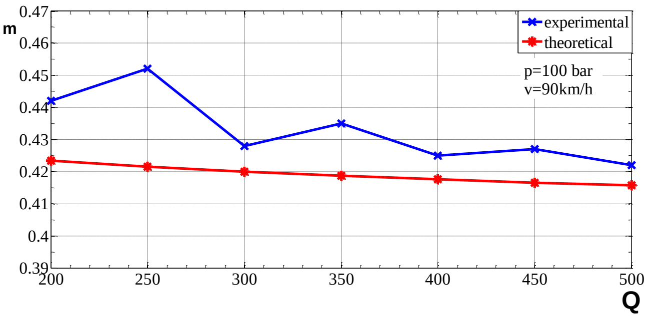


Figure 4 shows the comparative values obtained by experimental tests and the results obtained by mathematical model for the first brake pad. At this stage, it is quite easy to realize that the values of the friction coefficient, resulting from equations 5, 6 and 7 and experimentally (at increased temperatures), are, by its character, in function of the influential parameters (p, v, θ) equivalent, which suggests that the friction model resulted from short term laboratory testing on inertia dynamometer, is virtually applicable for predicting the friction of the applied friction material, even at extremely high temperatures, on the same vehicle, under quite unfavorable operating conditions.

V. CONCLUSION

In the first part of this paper results of the experimental tests were used to develop a mathematical model of friction. Using the mathematical model of friction, value of the coefficient of friction under conditions of extremely high temperatures it was estimated and that was confirmed by experimental results in the same conditions. On the basis of conducted experimental research, obtained results and their analysis, the following conclusions can be drawn:

- The results from 27 brake applications enable the developing of the friction model for the given brake pad lining.
- Reliability of the results presented model is confirmed by experimental results which show the deviation is in frame 3%.
- The resulting friction model can serve in predicting the friction and in that way the efficiency of the brake pad lining material, even at increased temperatures. In this way estimation of the actual behavior of the automotive brake is ensured in exploatation, even under extremely difficult operating conditions.

REFERENCES

[1] Talati F., Jalalifar S. (2009). Analysis of heat conduction in a disk brake system, Heat Mass Transfer 45, p.1047–1059,
[2] Litos P., Honner M., Lang V., Bartik J., Hynek M., (2008), A measuring system for experimental research on the termomechanical coupling of disc brakes, Proc. IMechE Vol. 222 Part D: J. Automobile Engineering, , 1247-1257.
[3] Bansal D. G., Streator L. J., (2009), A method for obtaining the temperature distribution at the interface of sliding bodies, Wear 266, 721–732.
[4] Cho H-J, Cho C-D, (2008), A study of thermal and mechanical behaviour for the optimal design of automotive disc brakes, Proc. IMechE Vol. 222 Part D: J. Automobile Engineering.
[5] Ostermeyer G. P., Graf M., (2013), Influence of wear on thermoelastic instabilities in automotive brakes, Wear 308, 113–120.
[6] Duboka Č., Glasner E.C., Todorović J., Arsenić Ž.: Tribo-mutations in tribo-mechanical systems, Balkantrib '96 2nd Intl. Conf. on Tribology, Thessalonniki, 5-7 June '96
[7] Ludema K.C., Mechanism-based modeling of friction and wear, Wear 200, pp. 1-7, 1996.
[8] Dmitriev A.I., Osterleb W., Klob H., Numerical simulation of typical contact situations of brake friction materials. Tribology International 41 (2008) 1–8
[9] Popov V.L., Psakhie S.G., Physical nature and properties of dynamic surface layers in friction, Tribology International 39, pp. 426–430, 2006.
[10] Rice S.L., Moslehy F.A., Modeling friction and wear phenomena, Wear 206, pp. 136–146, 1997.
[11] Arsenić Ž.: Modeling of the friction and wear characteristics of semi-metallic friction materials for commercial vehicle disk brakes, International Journal for Vehicle Mechanics, Engines and Transportation Systems, Volume 21, Number 3, September 1995, YU ISSN 0350 – 1027.
[12] Todorović J., Duboka Č., Arsenić Ž.: Influence of the Brake Interface Temperature on the Brake Force Distribution under Service Conditions, SAE Paper 900006
[13] Arsenić Ž.: Investigation on prediction possibilities for functional characteristics of motor vehicle brake friction materials, Doctor Thesis, Fac. Mech. Eng. Belgrade, 1986.

Aleksandar Grkic Military Academy Belgrade, Department of Military Mechanical Engineering, Section for combat vehicles, teaching assistant. Co-author of 1 monograph, 5 papers published in national journals, 15 papers presented at national and international conferences, co-author of 2 laboratory equipmentes and testing sets and participant in the implementation of the 2 scientific projects. Member of JUMV.

Slavko Muzdeka Associate Professor at Military Academy Belgrade, Department of Military Mechanical Engineering. Author / co-author of 4 monographs and / or parts of monographs, books or textbooks,10 papers published in national journals, 20 papers presented at national and international conferences, co-author of 2 laboratory equipmentes and testing sets and participant in the implementation of the 4 scientific projects. Member of JUMV.

Zivan Arsenic full professor at Faculty of Mechanical Engineering, University of Belgrade Automotive Department. Author / co-author of 8 monographs and / or parts of monographs, books or textbooks, 20 papers presented at international congresses, 8 papers published in international and 17 national journals, 63 papers presented at national and international conferences, 3 Head and participant in the implementation of the 22 scientific project, co-author of 2 and 5 protected patents, co-author of more than 20 laboratory facilities, equipment and testing sets, co-author of more than 40 software necessary for operating the testing sets, the collection, processing and presentation of test results, as well as in the field of maintenance and reliability. Manuel Hunoy Prize 1986, FISITA. Member of JUMV.

Cedomir Duboka full professor at Faculty of Mechanical Engineering, University of Belgrade Automotive Department. Author/co-author of 28 monographs or their parts, books and teaching books, 28 papers presented at international congresses, 22 papers published in international and 29 papers published in national scientific and professional journals, 111 papers presented at both international and national conferences. He was invited author at 7 international and several national conferences, manager of 15 and a team member in 25 research projects, co-author of 2 pending and 5 announced patents, co-author of more than 20 realized patents and laboratory facilities or equipment, more then 40 software for automatic control of test stands, data acquisition and processing, data analyses and presentation, including statistical data processing and analyses, effectiveness and reliability analyses, more than 10 documented test methods, and more than 400 works of limited circulation (calculations, tests, analyses etc.). He is a co-author of the development of Frimeks Laboratory at the Faculty of Mechanical Engineering in Belgrade (1979 – 1982), co-author of the development of 10 units of clutch and brake inertia dynamometers built for own purposes of the Faculty of Mechanical Engineering in Belgrade, or customers in Serbia and abroad. He is a founder of JUMV – Yugoslav Society of Automotive Engineeris, where he is President since 1992, and also a Council Member at both FISITA and EAEC. Since 1982 he is a member of SAE – the most prominent society of automotive engineers, and since 2007 is in charge of establishing the EVU Coutry Group Serbia. Member of the Board of Automotive Clubs of Yugoslavia and Serbia for years serving as a vice president. Member of GRRF Group of experts within WP.29/ ECE OUN, Geneva, for more then 25 years; member of ISO/TC22/WG6-WG10 (Braking performance tests and Braking terminology). He is co-editor of the MVM Journal published in Kragujevac (Serbia) and a Member of the Editorial Board of the Internet Journal "International Journal of Vehicle Technologies". At FISITA Congress Belgrade 1986. he was granted with Manuel Hunoy Prize for the best paper presented by young authors. He is principal organizer of highly recognized international conferences "Science and Motor Vehicles.