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Obiective şi scop: Revista ştiinţifică „Ingineria Mediului şi Antreprenoriatul Dezvoltării Durabile" este o publicaţie interdisciplinară care urmăreşte o analiză ştiinţifică în scopul realizării unor dezbateri asupra ingineriei mediului şi antreprenoriatul dezvoltării durabile pe plan local, naţional sau mondial. La nivel concret sub auspiciile antreprenoriatului şi dezvoltării durabile revista va include contribuţii ştiinţifice din domeniile ingineriei mediului, managementul întreprinderii şi antreprenoriatului, prezentând tendinţele şi provocările secolului XXI în problematica dezvoltării durabile şi protecţiei mediului. Contribuţiile vor avea scopul de a oferi cititorilor materiale originale şi de înaltă calitate.

Cititori: Revista ştiinţifică este elaborată pentru a oferi o sursă de referinţe ştiinţifice la îndemâna oricărei persoane care are activitatea de cercetare în domeniul problemelor globale cu privire la protecţia mediului, antreprenoriat sau dezvoltarea durabilă. Revista oferă cadrelor didactice universitare, cercetătorilor, managerilor, profesioniştilor, antreprenorilor, reprezentanţilor ai societăţii civile şi personalităţilor din politică, un instrument de lucru pentru a dezvolta astfel o afacere durabilă protejând mediul înconjurător.

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Romanian Society of Automotive Engineers annual congresses has always succeeded to awake the desire of us, the academic members from various fields - research, design, production, exploitation, to mention only a few, to meet again, all of us who dream toward a more human environment, a quitter life, an honest friendship, beautifulness, serenity …

This year the Alma Mater Napocensis, Technical University of Cluj-Napoca, gladly organizes The International Congress of Society of Automotive Engineers of Romania – SIAR "Automotive Motor Mobility Ambient – AMMA 2013", along with a series of manifestations meant to drive attention of both Romanian and abroad specialists in the fields of automotive, transportation and environment.

This Congress will be held during 17-19th of October 2013, under the high patronage of FISITA (International Federation of Automotive Engineering Societies) having the purpose to reunite paper works comprising scientifically research, inventions and new ideas in the fields of automotive, environment and transportation, under a good quality scientific programme.

The mentioned event offers the opportunity for all the specialists involved in durable development to positively exchange opinions and contribute to a better education.

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CONTENT

CUPRINS

STUDIES OF CAN COMMUNICATION NETWORK ERRORS DESIGNED FOR FREIGHT TRANSPORT VEHICLES

STUDIUL REȚELELOR DE COMUNICARE DIN CADRUL AUTOVEHICULELOR DESTINATE TRANSPORTULUI DE MĂRFURI șI PERSOANE

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Abstract: *Nowadays, in order to ensure an efficient functioning as well as the maximum degree of safety and comfort, it is necessary to connect systems that surround us, thus the necessity of communication networks. Although these networks differ depending on the applications for which they were created, the main problems to overcome remain roughly the same: the concepts of network access, network reliability, security of transmitted data, network topology, length and bit rate, physical environment, etc. This paper contains an analysis regarding the types of errors that occur and their occurrence frequency. As a result, an evaluation of vehicle communication networks in terms of safety and reliability in operation is possible. The study focused on heavy duty vehicles and considered a predetermined number of workshop entries. The causes of errors and the resolutions for several of these errors were analyzed.*

Keywords: *network, data, CAN, bit, error, interference.*

Rezumat: *În contextul actual, pentru a asigura o funcționare cât mai eficientă, cât mai puțin poluantă și gradul maxim de siguranță și confort, este necesar ca sistemele care ne înconjoară să interrelaționeze. Soluția o reprezintă rețelele de comunicare. Cu toate că aceste rețele diferă între ele în funcție de aplicațiile pentru care au fost create, principalele probleme, care trebuie depășite rămân, în mare parte, aceleași: conceptele de acces pe rețea, elasticitatea rețelei, securitatea datelor transmise, topologie, lungime și rata de biți, mediul fizic etc. Lucrarea de față conține o analiză a tipurilor de erori care apar și frecvența lor de apariție în vederea evaluării, din punct de vedere al siguranței și al fiabilității în funcționare, rețelele de comunicare din autovehicule. Studiile au vizat autovehiculele de mare tonaj și s-au efectuat pentru un număr prestabilit de intrări în service. S-au analizat cauzele apariției erorilor și soluțiile de rezolvare pentru câteva dintre acestea.*

Cuvinte cheie: *rețea, date CAN, bit, eroare, interferență.*

1. Introduction

When a new vehicle is developed, a lot of effort must be put into testing, in order to eliminate design flaws. Regardless of this, some defects are found only after the market launch. In addition to known phenomenons that affect a certain system there can always intervene the unexpected. This paper deals with the human interference for a specific type of vehicles and failures. In these sense the effects of the “unforeseen” human factor on the CAN bus network are pointed out. Controller Area Network (or CAN), is a serial bus protocol that supports distributed real-time operation with a high level of security. Electronic control units (ECU’s) are connected within the vehicle without the need for a host computer because the protocol is based on the “broadcast diffusion” mechanism (Figure 1). This means that a message is transmitted to “everyone”. Message filtering is carried out at every station based on the message identifier (ID). When multiple stations try to send simultaneously the message with the highest priority is transmitted. The structure of a CAN message is represented in figure 2. To check whether the correct message was sent and received a cyclic redundancy code (or CRC) is used. CRC is generated by the sender in relation to the content of the message. If the message was received correctly, the receiver sends a positive acknowledgement. Otherwise a negative acknowledgement is sent and the message must be retransmitted (the message must win a new arbitration process). The control field indicates the number of bytes contained in the data field. The start of frame and end of frame define message

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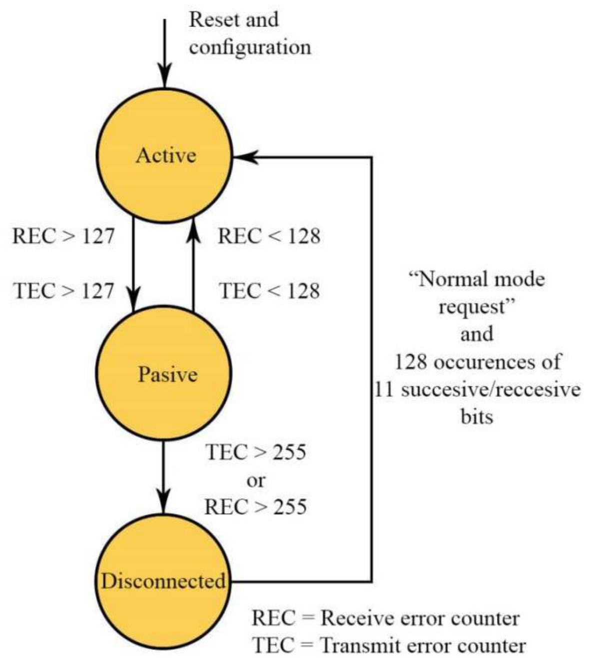

Figure 3. Error processing mechanism
Modified after [2], pg. 57

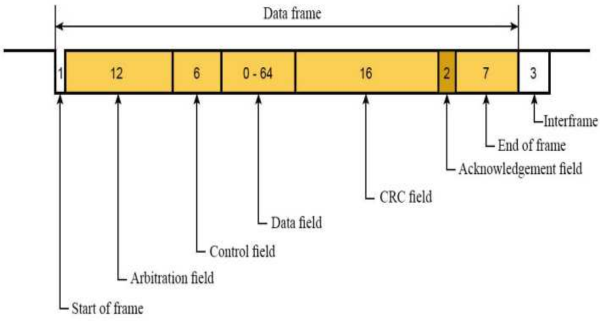

Figure 3. CAN message format
Modified after [2], pg. 40

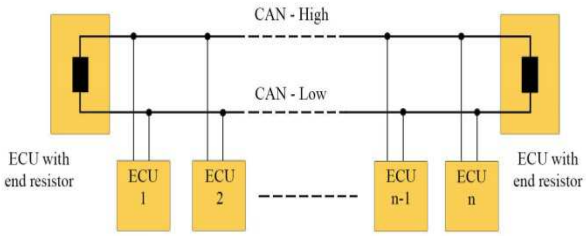

Figure 3. CAN network

boundaries. The interframe field is the minimum idle period between two messages. When it comes to the CAN protocol it is important to mention one of its most important features: the error processing mechanism. The purpose of this mechanism is to detect and localize errors and faults for a precise intervention. Microcontrollers closest to the fault source must react immediately and with the highest priority. Every microcontroller must incorporate the following two counters:

- Transmit error counter – TEC;
- Receive error counter – REC.

For every transmission/reception error detected the value stored in the counter is incremented by 8. On the other hand, for every correct message the value is decremented by 1. Therefor it is possible for a control unit to accumulate points despite of it having transmitted/received more correct than erroneous messages. In this case, the mechanism provides information about the frequency of errors. Depending on the stored values in the counter, one network node can be (Figure 3):

0 – 127: Active (Error active state) – The node can send and receive messages normally. Moreover it can send active error flags (they interrupt the current transmission). It is recommended to take control measures when one value reaches 96 points;

- *128 – 255: Passive (Error passive state)* – The node can send and receive messages normally but it can only send passive error flags (don't affect the current transmission) and only during the error frame;
- *255: Disconnected ("Bus off" state)* – In this situation, the node is no longer permitted to perform any intervention on the bus. It can reconnect after it had seen on the bus 128 consecutive error-free occurrences of 11 recessive bits.

The performances of the error processing mechanism are very important taking in consideration the fact that it influenced the choosing of the CRC type and its number of bits. There are two classes of errors:

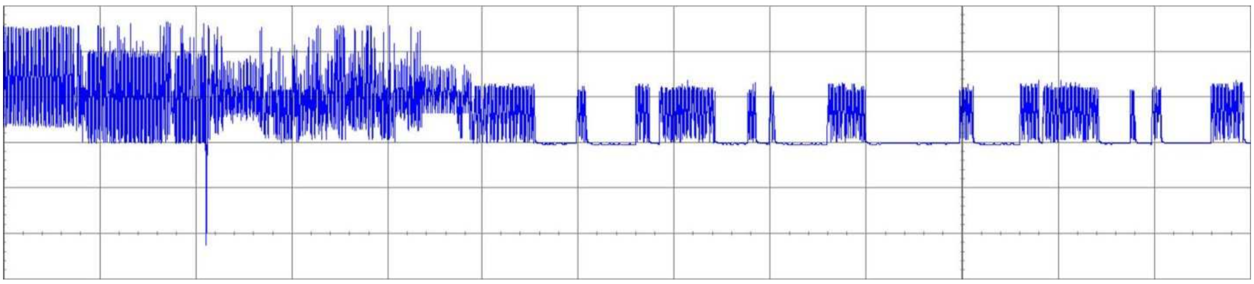

Figure 4 Recovery CAN-High signal of a control unit when reconneted to the bus
(Screenshot of StarDiagnosis Compact 4 and HMS 990)

- Errors that do not alter the frame length – in this case the only fields that can be affected by bit errors are the Identifier field and the Data field;
- Errors that alter the anticipated frame length – if bit errors affect the following fields: Start of frame, Data length code, Remote transmission request or Identifier extension, the receiver interprets the message differently than it should have.

In addition to previously mentioned residual errors the message content or the bit stuffing rule can be affected by parasites.

2. CAN in real operating conditions

The following results were obtained after studying more than 1000 workshop entries containing CAN errors of a specific freight transport vehicle model. Using specialized diagnosis equipment and software a test log was printed for every workshop entry. Only logs containing CAN bus errors were selected. For every ECU, the type of error and its corresponding number of occurrences were taken into account. After seeing the resulting numbers it was considered necessary to outline a major issue that concerns this type of vehicles as described below.

In order to achieve the desired levels of safety and comfort it is necessary to reduce/eliminate user interferences that can cause errors. A good example for this situation is the deluding of the tachograph (a device used to record speed, driving times, as well as traveled distance for 1 or 2 drivers.) recordings. In order to avoid penalties due to failure to comply with rest periods regulations, some users attach magnets to the transmission housing near the tachograph sensor. The magnetic field interferes with the sensor and prevents it from registering the real movement of the vehicle. This translates into error codes set in the ECU's which use the information provided by the sensor.

Apart from increasing the load on ECU's error memory unjustified, the intervention can lead in some cases to malfunctions of the transmission (although it doesn't set an error code it has been proven experimentally). Moreover, transmission malfunctions can lead to increased fuel consumption.

This kind of human interference could be discouraged by implementing a redundant information system. The following two methods are considered to be a good choice:

a) Installing a second sensor inaccessible to the user. The information would be stored by the tachograph and accessible only to authorized personnel. The drawback of this method is the increased cost: sensor, wiring etc.
b) Using existing sensors such as the wheel speed sensors. Because the information already exists only a small amount of reprogramming it's needed.

Table 1. *Powertrain CAN errors*

ECU	Error	Number of occurrences	Percentage
Drive control unit	CAN bus Transmission in single wire mode.	7	38,9
	Engine CAN bus between Drive control and Engine control in single wire mode.	5	27,8
	Transmission CAN fail to supply data.	5	27,8
	High-speed CAN inactive.	1	5,5
	TOTAL	**18**	**100%**
Transmission control unit	CAN bus Transmission in single wire mode.	38	50
	Communication fault on Vehicle CAN bus.	38	50
	TOTAL	**76**	**100%**
Engine control unit	CAN bus connection to Drive control faulty.	35	46,7
	CAN-High connection to Drive control faulty.	2	2,7
	Malfunction of CAN bus connection between Engine control and Exhaust aftertreatment control.	11	14,6
	Missing key recognition information on Engine CAN bus.	27	36
	TOTAL	**75**	**100%**
Retarder control unit	Faulty or missing CAN message from Transmission control.	12	57,1
	Faulty or missing CAN message from Traction control.	5	23,8
	Faulty or missing CAN message from Drive control.	3	14,3
	Faulty CAN bus communication.	1	4,8
	TOTAL	**21**	**100%**

A third, but rather extreme option, would be to limit the power output of the engine.

Another important issue is the differentiation between faulty messages sent by the data acquisition unit and CAN bus faults (i.e. “Malfunction of CAN bus connection between Engine Control and Exhaust aftertreatment control”). The majority of causes that led to this error were faulty data acquisition units. Others were due to wiring problems, while only a few were stored as result of faulty control units. It is necessary to better detection of faulty messages in order to achieve lower repair times.

Table 2. *Brake control unit CAN errors*

ECU	Error	Number of occurrences	Percentage
Brake control unit	Data transfer to brake CAN bus is faulty.	10	2,8
	Data transfer to vehicle CAN bus is faulty.	2	0,6
	Communication between CAN bus controllers faulty.	1	0,3
	Data on vehicle CAN bus missing or faulty.	108	30
	CAN message „movement-signal” from tachograph faulty.	188	52,2
	Trailer CAN signal faulty.	2	0,6
	Trailer CAN-Low signal faulty.	11	3
	Trailer CAN-High signal faulty.	12	3,3
	Trailer CAN signal has quiescent current.	26	7.2
	TOTAL	**360**	**100%**

3. Conclusions

This paper presented the CAN bus behavior in real operating conditions by studying errors stored in t he ECU’s connected to the network. After reviewing more than 1000 vehicle test logs, the following were concluded:

a) The CAN protocol is a safe environment for data transmission due to its good error processing mechanism.
b) In some cases, the user can cause faults by disturbing the data acquisition process. This kind of intervention can lead to malfunctions of powertrain components.
c) It is necessary to distinguish between faulty messages (sent by faulty units) and other electrical problems (wiring defects, lack of power supply etc.)

Reports regarding stored errors in the memory of the control units aid in deciding for the path to follow in order to eliminate the cause or causes of the faults. Moreover these reports are important for improving CAN bus quality in real operating environments.

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DAMPING ANALYSIS OF WIRE ROPE ISOLATORS, HYBRID ISOLATORS AND RUBBER ISOLATORS

ANALIZA ELEMENTELOR AMORTIZOARE CU CABLURI, IZOLATOARE HIBRIDE ŞI DIN CAUCIUC

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Abstract: The vibration attenuation represents a major objective in automotive industry. Special rubber elastic elements are identified as attenuation solutions, helping in solving this objective. At the present the wire rope isolators (WRI) represent a good solution for vibration attenuation. The aim of this paper is to present a comparison from a vibration attenuation point of view between three types of vibration isolators (WRI, hybrid wire rope and rubber elastic). It is well known that the challenge nowadays in the automotive industry consists in having the best isolators from vibration attenuation, time to market strategy, and cost-efficiency point of view. The analysis is done for a specific application for attenuate vibration of the exhaust line and the results presented here appear to be interesting for the NVH community working on this area

Keywords: *Wire rope isolators; Rubber isolators; Hybrid wire rope isolators, Damping, Hysteresis curve.*

1. **Introduction**

In order to satisfy the current customer requirements, the automotive industry focuses more on reducing the level of noise and vibrations produced in modern vehicles. Various isolators are designed for engines as mount system components. Isolators are commercially available in many different resilient materials, in countless shapes and sizes, and with widely diverse characteristics (fig.1) [11].

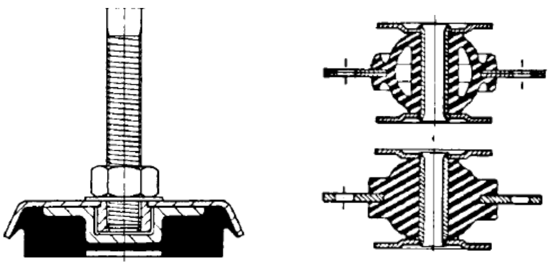

Figure1. Rubber isolators types

An important cause for interior vehicle noise is the structure-borne sound from the engine. The vibrations of the source (engine) are transmitted to the receiver structure (the vehicle) causing interior noise in the vehicle. For this reason the engine mounts must have good filtration properties for passive isolation [6]

The properties of a given isolator are dependent not only on the material of which it is fabricated, but also on its configuration and overall construction with respect to the structural material used within the body of the isolator [11].

The function of an isolator is to reduce the magnitude of motion transmitted from a vibrating system to the equipment or to reduce the magnitude of force transmitted from the equipment to its bracket.

Rubber is a unique material that is both elastic and viscous. Rubber parts can therefore function as shock and vibration isolators and/or as dampers. The isolation behavior of rubber isolators strongly depends on the excitation frequency and the pre-deformation of the mount as consequence of the weight of the source to be isolated [1] [2].

Wire rope isolators (WRI) have different response characteristics depending on the diameter of wire rope, number of strands, cable

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length, cables twist, number of cables per section and on direction of the applied force [1] [2].

Wire ropes [7] [9] can be grouped into two broad categories by the type of central core used. Independent wire rope core (IWRC) ropes are the stronger of the two and offer the greater resistance to crushing and high temperatures. Fibre core (FC) wire ropes while weaker, offer advantages in terms of flexibility, weight and of course price.

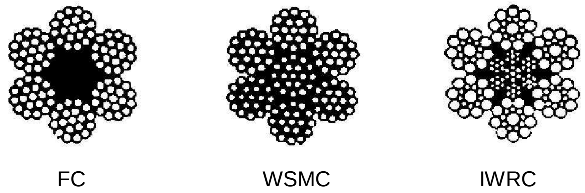

Figure 2. Wire rope sections

The function of the core in a steel wire rope is to serve as a foundation for the strands, providing support and keeping them in their proper position throughout the life of the rope. Fibre cores are generally used, when impregnated with grease, for providing internal lubrication as well as contributing to flexibility.

The construction of wire rope isolators (fig. 3) is ingenious, but still based on relatively simple design [1] [2] [3]. Stainless steel wires are twisted into a cable, which is mounted between two bars.

Figure 3. Wire rope isolators

In comparison with rubber elements, the wire rope isolators are full metal structure, maintenance-free and are not subject to aging due to external factors like oil, salt water, chemicals and temperature variation. Most applications of wire rope isolators are found in situations where equipment needs to be mounted against shock or vibration, but where sound isolation is of minor importance [1] [2] [8].

The other advantages of a wire rope isolator lie in the ability to combine a high level of both shock and vibration isolation, in combination with relatively small dimensions. Wire rope isolators are limited by their own construction and may for this reason be loaded in any direction without the risk of malfunctioning [2].

The solid structure of wire rope isolators might be a disadvantage in terms of vibration attenuation and transmissibility. The wire rope isolator's and rubbers advantages have brought into discussion the development and analysis of a new elastic element called hybrid wire rope. The hybrid wire rope isolators represent different combinations between rubber and wire rope isolators in order to obtain a new product with better properties.

2. Mathematical model

Damping is the phenomenon by which mechanical energy is dissipated (usually converted into internal thermal energy) in dynamic systems. A knowledge of the level of damping in a dynamic system is important in utilization, analysis, and testing of the system. [4] [5]

In characterizing damping in a dynamic system, it is first important to understand the major mechanisms associated with mechanical-energy dissipation in the system. Then, a suitable damping model should be chosen to represent the associated energy dissipation. Finally, damping values (model parameters) are determined, for example, by testing the system or a representative physical model, by monitoring system response under transient conditions during normal operation, or by employing already available data [5].

There is some form of mechanical-energy dissipation in any dynamic system. Several types of damping are inherently present in a mechanical system. Three primary mechanisms of damping are important in the study of mechanical systems. They are:

- Internal damping (of material)
- Structural damping (at joints and interfaces)
- Fluid damping (through fluid-structure interactions).

Internal damping of materials originates from the energy dissipation associated with microstructure defects, such as grain boundaries and impurities; thermoelastic effects caused by local temperature gradients resulting from non-uniform stresses, as in vibrating beams; eddy-current effects in ferromagnetic materials; dislocation motion in metals; and chain motion in polymers. Several models have been employed to represent energy dissipation caused by internal damping [5].

It has been noted that, for hysteretic damping (h), the damping force (or damping stress) is independent of the frequency (ω) in harmonic motion. It follows that a hysteretic damper (h) can be represented by an equivalent damping constant *(c)* of [5]:

$$c = \frac{h}{\omega} \tag{1}$$

which is valid for a harmonic motion of frequency ω . This situation is shown in figure 4 [5].

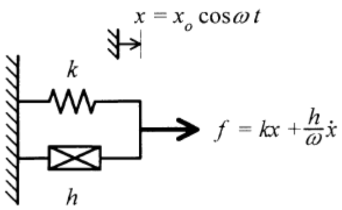

Figure 4. Systems with hysteric damping

For a damped system, the force versus displacement cycle produces a hysteresis loop. Depending on the inertial and elastic characteristics and other conservative loading conditions (e.g., gravity) in the system, the shape of the hysteresis loop will change; but the work done by conservative forces (e.g., intertial, elastic, and gravitational) in a complete cycle of motion will be zero. Consequently, the net work done will be equal to the energy dissipated due to damping only. Accordingly, the area of the displacement-force hysteresis loop will give the damping capacity ΔU. The energy dissipation per hysteresis loop of hysteretic damping ΔU_h is [5]:

$$\Delta U_h = \pi \chi_0^2 h \,. \tag{2}$$

Note that the stiffness (*k)* can be measured as the average slope of the displacement-force hysteresis loop measured at low speed. The loss factor for hysteretic damping (η) is given by:

$$\eta = \frac{h}{k}, \tag{3}$$

$$\eta = 2\xi, \tag{4}$$

Then, from equation (3), the equivalent damping ratio (ξ) for hysteretic damping is

$$\xi = \frac{h}{2k} \,. \tag{5}$$

A WRI type KR 3,5 7-02 (according Sebert Tehnologie supplier) type of elastic element was tested by applying a low-speed loading cycle and measuring the corresponding deflection. The load vs. deflection curve that was obtained in this experiment is shown in figure 5.

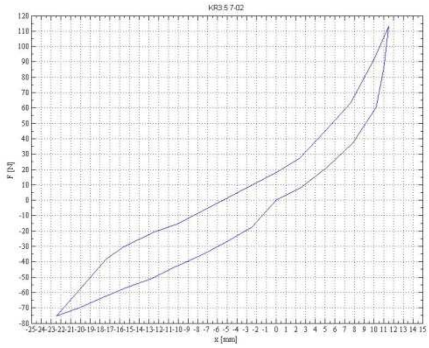

Figure 5. Hysteresis loop for KR 3,5 7-02 element

The area of the loop presented in figure 5 is:

$\Delta U_h = 811{,}59\ N\cdot mm$.

Alternatively, one can obtain this result by counting the squares within the hysteresis loop. The deflection amplitude is

$\chi_0 = 11{,}5 mm$.

Hence, from equation (2),

$$h = \frac{\Delta U_h}{\pi \kappa_0^2} = 1{,}95 \frac{N}{mm} \,.$$

The stiffness of the damping element is estimated as the average slope of the hysteresis loop; thus

$$k = 6{,}7 \frac{N}{mm} \,.$$

The equivalent damping ratio is

$$\xi = \frac{h}{2k} = 0{,}14 \,.$$

After processing the test results for the elastic element KR 3.5 7-02 resulted following quantities stiffness $k = 6{,}7\ N/mm$,, damping ratio $\xi = 0{,}14$

3. Experimental setup

Experiments were performed in the engine test bench where the exhaust line was installed on the three types of isolators: 1 rubber isolator (fig. 6), 2 wire rope isolator (fig. 7) and 3 hybrid wire rope isolators (fig. 8). The characteristic of hybrid wire rope isolator analyzed (fig.8) is ends of wire coated. These elastic elements have a coat of cable that ends with a thin layer of rubber. Rubber coated cable ends are fixed between the two plates of the elastic element. The purpose of this cable end coating was to isolate the vibration transmitted through metallic ways

The testing was performed by running the engine starting from 950 rpm to 4500rpm at full load, to cover all excitations induced in the mounting brackets.

Figure 6. Rubber isolator

Figure 7. WRI

Figure 8. Hybrid WRI

Comparative measurements were made between a type of rubber elastic element, used for the evaluated exhaust line, isolators KR type for WRI elements and hybrid WRI, with wire thickness of 3.5 mm.

As measurement points we chose the positions where the exhaust line is mounted on the bracket (fig.9, fig.10):

- ECH: 01 X/Y/Z and ECH: 02 X/Y/Z : points that evaluate vibration signal on exhaust line, in fact is the input signal which goes in these elastic elements;
- RH: 01X/Y/Z and RH: 02 X/Y/Z points that measure the signal at the exit of the elastic element. It is the signal which is filtrated by the elastic element.

The exhaust line is installed on isolators in 2 points (fig.9 and fig.10). It's mentioned the fact that the mounting points are subjected differently to the exhaust line weight, so, the RH:01 point is subjected to a smaller exhaust line weight load and the RH:02 point is subjected to a bigger weight load.

Triaxial accelerometers mounted in measuring points had sensitivity of 1 mV/(m/s^2) and mounted so as to respect coordinates:

- X – transversal on exhaust line, transversal on engine crankshaft,
- Y – longitudinal on exhaust line, longitudinal on engine crankshaft,
- Z – vertical direction.

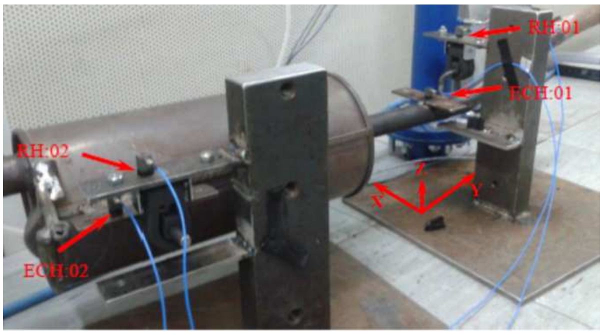

Figure 9. Exhaust line on rubber elements

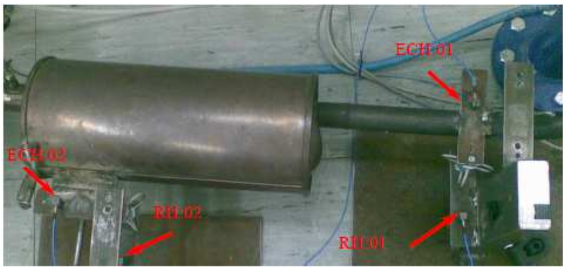

Figure 10. Exhaust line on wire rope elements

Signal acquisition was performed with a complete LMS testing system for which it was considered a signal acquisition bandwidth of 2560 Hz at 5 Hz resolution.

The graphics extracted from the signal analysis are presented in comparison in order to highlight the vibration behavior of each isolator type. To properly describe the behavior for each elastic element type, the maximum acceleration energy curves were extracted depending on the frequency (peak hold) and the maximum acceleration energy depending on the engine speed (overall level). The analysis is done for each point and the graphical representation is:

1. Black curve–rubber isolator
2. Red curve– wire rope isolators.

4. Results

A Fast Fourier Transformation (FFT) was applied to the vibration time signal in order to have the results in frequency spectra. The frequency spectrum shows the vibration amplitude as a function of frequency. When the environment is not constant in time it may be necessary to measure the peak hold (called "maximum rms") vibration levels.

To make sure that the same input signal is used, the signal measured at the points ECH: 01 and ECH: 02 on the vertical direction are represented in fig.11. It can be noticed that the system engine-exhaust line, responsible for the vibration transmitted and induced in elastic brackets, have the same amplitude level of acceleration and the same vibration behavior for all three analyzed cases.

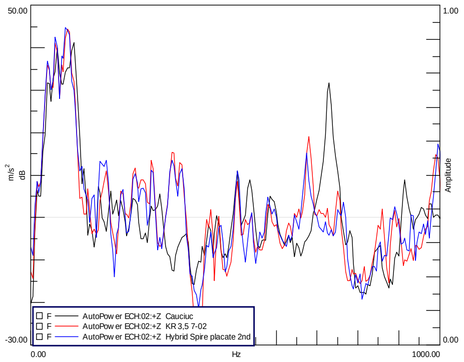

Figure 11. Peak-hold for input points

The results of the measurements held on the 3 isolators types are presented as graphics in which are presented the maximum values of the acceleration depending on frequency and the maximum energy of acceleration depending of the engine speed. So, for each measurement point (RH:01 X/Y/Z and RH:02 X/Y/Z) on the three directions, the acceleration amplitudes depending on frequency (peak-hold) and depending on engine speed (overall level) are presented. In order to have a quantitative estimation of the attenuation degree for each type of elastic element analyzed, a root mean square value was extracted in the table for each curves type.

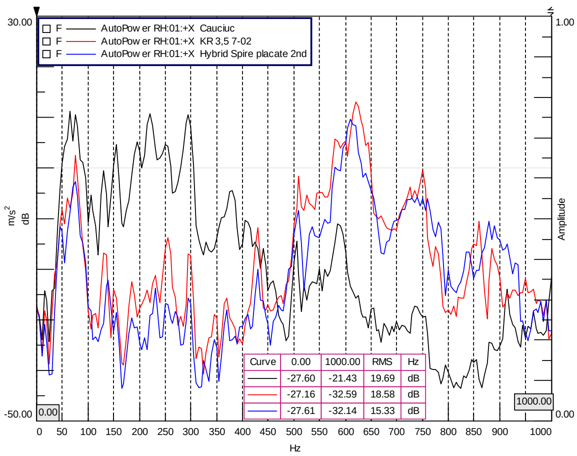

Figure 12. Peak hold for RH:01 X

In figure 12, figure 13 and figure 14 the measurement analysis results for measurement point RH:01 X/Y/Z depending of frequency are presented.

In all three directions is observed that up until the 500 Hz frequency, the low and mid frequencies, elastic elements KR and KR hybrid type shows much better attenuation of vibration than rubber element. In the higher frequencies over 500 Hz, rubber elastic element shows a higher degree of vibration attenuation.

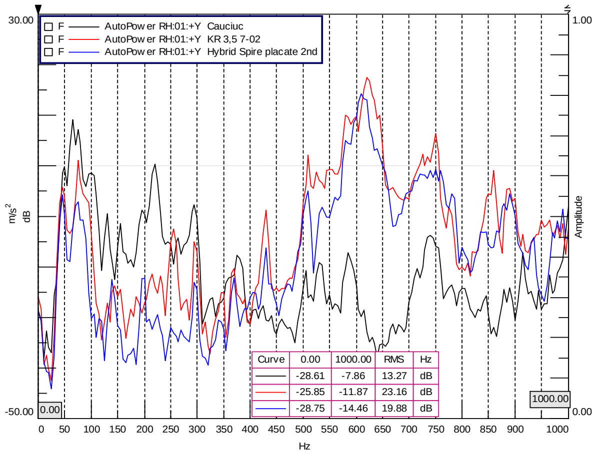

Figure 13. Peak hold for RH:01 Y

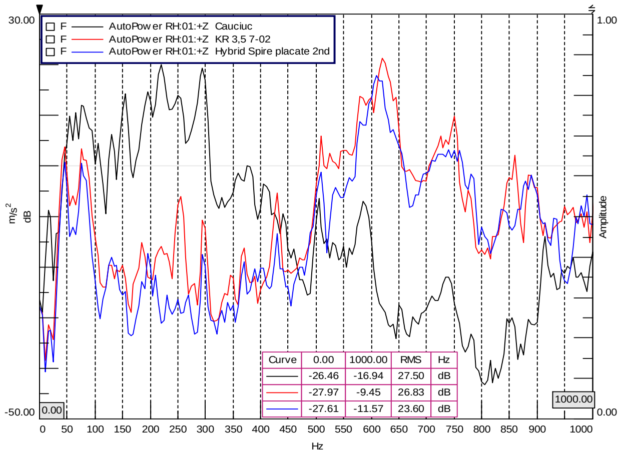

Figure 14. Peak hold for RH:01 Z

In order to have a quantity estimate of the attenuation level, in table 1 the root mean square values of all curves which define the vibration behavior depending on frequency and engine speed which correspond to measurement point RH.01 are presented. Through extracted data analysis in table 1, the fact that element KR-hybrid presents the best attenuation in the transmitted acceleration energy level with 1-4 dB RMS on X and Z directions compared to the other two isolators types, can be observed. In direction Y, longitudinal direction on exhaust line, the rubber isolators has an attenuation degree with 4 dB RMS better than KR-hybrid and 6 dB RMS better than the KR

By analyzing the root mean square values of the acceleration depending on engine speed are it is observed the fact that element KR-hybrid presents the best attenuation degree for all directions, exception is Y direction where the rubber element has better isolation properties

Table 1

Isolator type	RH:01 **Peak-Hold** m/s^2-dB RMS			RH:01 **Overall level** m/s^2-dB RMS		
	X	Y	Z	X	Y	Z
Rubber	19,7	13,3	27,5	26,6	20,7	34,4
KR 3,5 7-02	18,6	23,2	26,8	30,0	32,2	35,7
KR 3,5 7-02 hybrid	15,3	19,9	23,6	25,0	29,3	32,9

In figure 15, figure 16 and figure 17 the curves corresponding to the acceleration transmitted through the elastic elements depending on the engine speed in measurement point RH:02 X/Y/Z are presented. By analyzing these graphics on X direction, can be observed that the rubber element has a good vibration attenuation comparing with other two. On this direction transversal on exhaust line, X direction, can be identified several critical speeds determine the presence of amplitude peaks of the elastic elements KR and KR hybrid type.

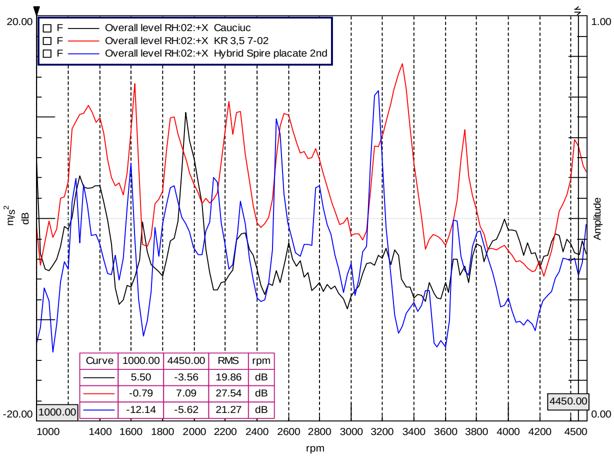

Figure 15. Overall level for RH:02 X

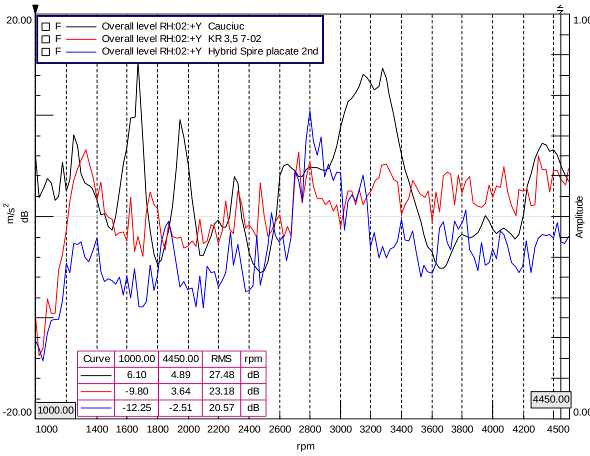

Figure 16. Overall level for RH:02 Y

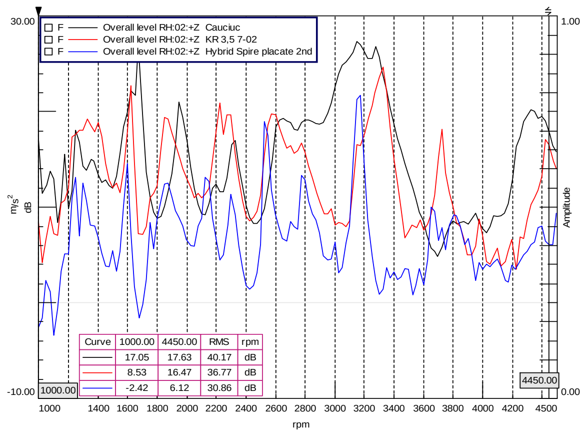

Figure 17. Overall level for RH:02 Z

Table 2

Isolators type	RH:02 **Peak-Hold** m/s^2-dB RMS			RH:02 **Overall level** m/s^2-dB RMS		
	X	Y	Z	X	Y	Z
Rubber	12,3	19,7	32,1	19,9	27,5	40,2
KR 3,5 7-02	20,2	8,7	29,3	27,5	23,2	36,8
KR 3,5 7-02 hybrid	14,8	6,9	24,1	21,3	20,6	30,9

In order to have a quantity estimate of the attenuation level, in table 2 the root mean square values of all curves which define the vibration behavior depending on frequency and engine speed which correspond to measurement point RH.02 are presented

In this table it can be seen that the elastic KR hybrid presents the best vibration damping compared to the other two elements analyzed. Both in amplitude analysis depending on frequency and depending on speed the element KR hybrid attenuates acceleration amplitude by 6-10 dB RMS better than rubber and 3-6 dB RMS better than the elastic element type KR. Also in this measured point RH: 02 is present an exceptional situation in which on the direction transverse to the exhaust line, X direction, the rubber element attenuate by 2-8 dB RMS amplitude accelerations compared to the other two elements

Through a global analysis of the vibrational behavior of elastic elements at this point we can say that KR hybrid the element has the best damping compared to the other two types analyzed

5. Conclusions

In this paper is presented an analysis on the type of isolators influence in vibration attenuation and presenting the mathematical model of isolators with hysteretic damping and damping coefficient calculation method. For this purpose, 3 types of elastic elements were tested, regarding vibration attenuation for an exhaust line mounted in engine test bench. This study is based on results obtained exclusively from experiments in which is evaluated the maximum acceleration energy related to frequency and related to the engine speed. By using these two evaluation criteria in the vibration behavior analysis, it was aimed to obtain real and correct information regarding the influence of the isolators' types in vibration attenuation.

Since the mounting points RH:01 and RH:02, where the measurements have done, were under different exhaust line weight loads, the analysis was performed individually for each point. The graphics corresponding to the maximum acceleration energy in relation to frequency and engine speed were extracted, but it was also accomplished a quantitative evaluation of the vibration attenuation degree for each WRI type. For reasons of size of the paper was decided to present just the proper graphics for maximum level of acceleration related to frequency for point RH:01 and for RH:02 were present just graphs corresponding for maximum level of acceleration related to speed

In a global analysis of all the results we can say that the elastic element KR Hybrid shows best vibration attenuation properties compared to the other two elements analyzed. However should be emphasized that as an the elastic element with metal construction and the rubber layer is not very high, at high frequencies over 500 Hz KR hybrid the elastic element does not attenuate vibrations transmitted as well as rubber. Understanding this phenomenon comes from WRI construction. Damping phenomenon of these elements occurs due to friction between the cable wires. Thus these elements shows maximum efficiency if are used in large amplitude displacements attenuation and low frequency. In the case of the exhaust line, its large displacements at low and mid frequencies have highlighted attenuation qualities of these types of elastic elements. These elements are metallic elements and small amplitude vibrations of high frequency were transmitted by these isolates to the receiver. Even if the element hybrid KR had a layer of rubber layer that was not enough to match the attenuation qualities of pure rubber isolators, but it was enough to present attenuation qualities better than KR pure metallic element.

So is highlighted in the applications presented in this article that up to frequencies of 500 Hz elastic elements KR and KR hybrid type has a much better damping than rubber. Of these two elements, KR hybrid shows the vibration attenuation by 2-6 dB RMS on all three directions and in all points as to the element KR

Introducing the mathematical model for analysis of these types of elements with hysteretic damping and the procedure for calculate the damping ratio from hysteresis curve help us achieve simulations for vibrational behavior of these elements in various applications. The presentation of these results based on simulations and mathematical model will be presented in the following articles.

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THE ANALYSIS OF THE INFLUENCE OF THE CLEARANCE ON THE IMPACT STRESSES AT GROOVES ASSEMBLIES

ANALIZA INFLUENȚEI JOICULUI ASUPRA FORȚEI DE IMPACT A ASAMBLĂRILOR CANELATE

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Abstract: *This paper takes into account two technological methods of punching of brake disks in automotive industry: conventional punching and smooth punching. The experimental tests prove that in the first situation – conventional punching, the clearance between the brake disk teeth and the central pinion grows more rapidly than in the second situation of smooth punching. Based on these results the authors developed a finite element model to study the stress level for several ascending clearance values. The model was created, pre- and post-processed with Patran program and solved with MSC Nastran SOL 700 program, both developed by the MSC Software Corporation. The resulted stress field was stored in order to perform a further durability (fatigue) analysis.*

Keywords: *analysis, unfluence, stress, impact, grooves..*

1. Introduction

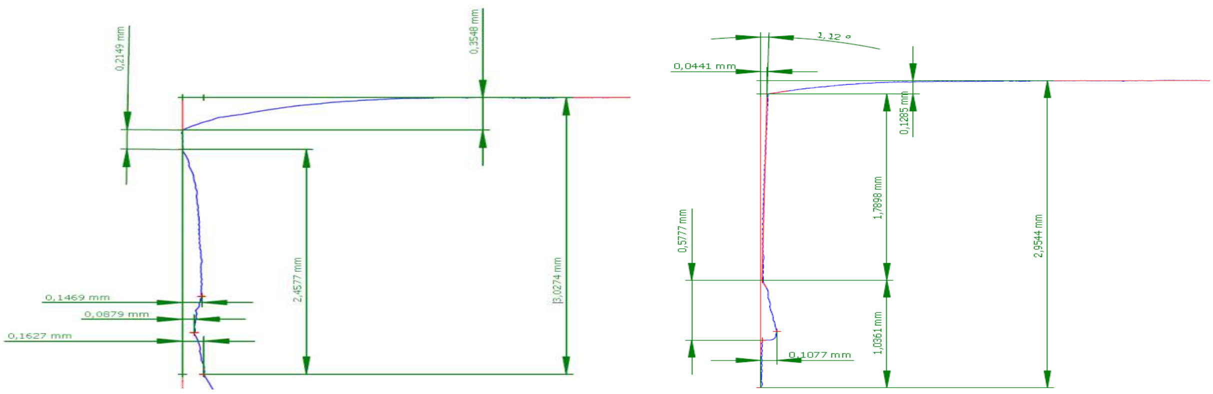
This study intends to put in evidence the influence of the clearance between the break disk teeth and the central pinion on the impact stress that occurs when breaking. For this reason the authors developed two models:

- An experimental model where was determined the clearance growth versus the number of impacts
- A finite element model where was determined for certain values of clearance the impact stress distribution versus time.

In order to study the influence of the clearance on the impact stress two technological methods were taken into consideration:

- Conventional punching - the resulted cutting section is characterized by a higher harshness and a more accentuated taper.
- Smooth punching – the resulted cutting section is smooth with lower taper.

Figure 1 presents the cutting section of the two technological methods taken into account.



Conventional punching   Smooth punching

Figure 1. Comparison between the conventional punching and smooth punching

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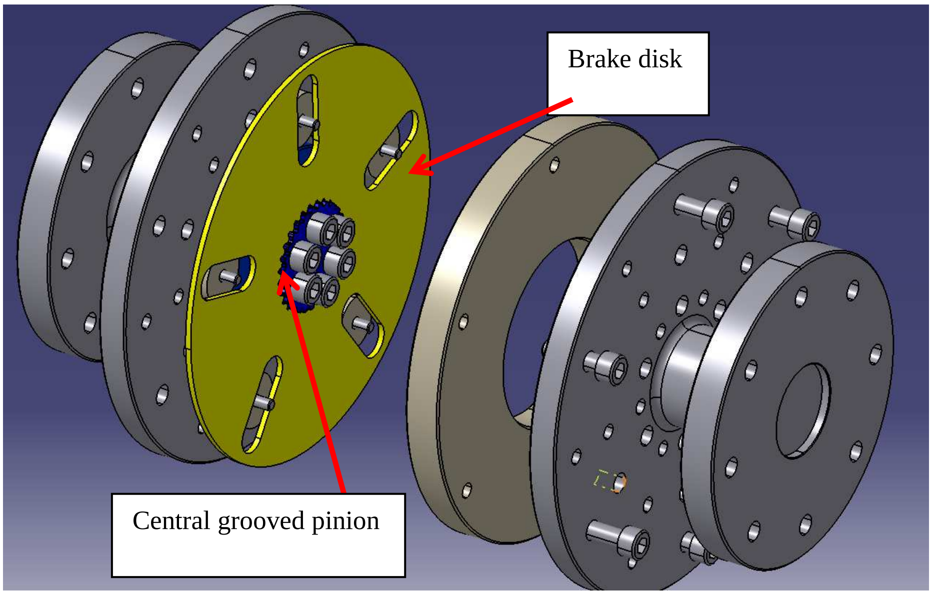

Figure 2. Experimental device to measure the clearance value versus the number of impacts

In order to measure the variation of the clearance versus the number of impacts between the brake disk and the central pinion an experimental device was designed and realized. Figure 2 presents this experimental device.

2. Measuring results

The measuring results were obtained with the above presented experimental device. The measuring results confirm the fact that, for the first technological method – conventional punching - the clearance growth more rapidly than for the second technological method – smooth punching. Figure 3 presents the variation of the clearance.

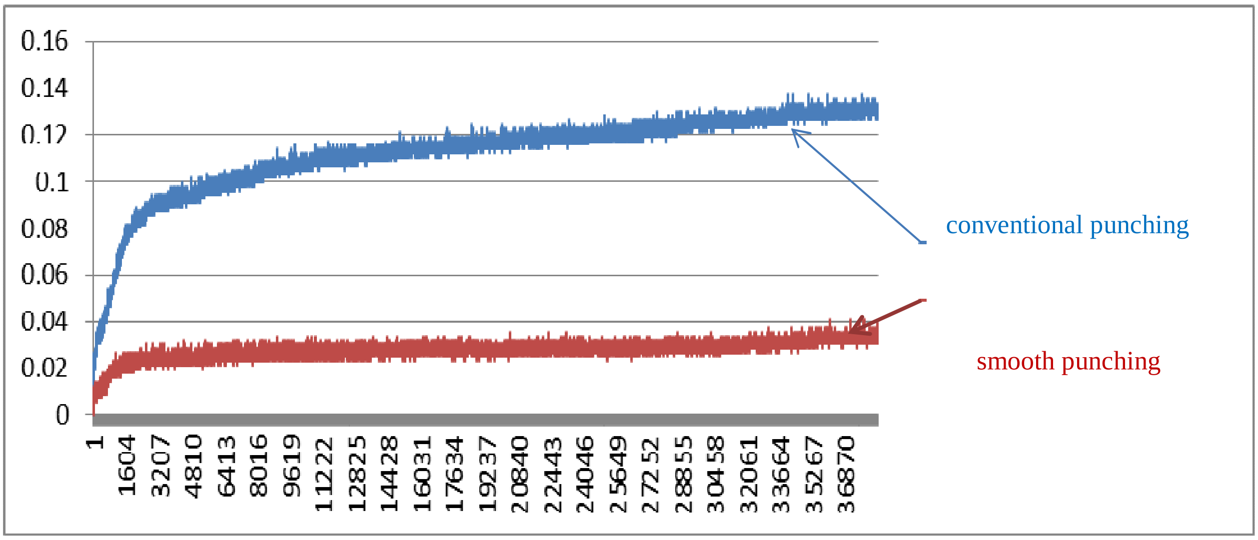

Figure 3. The variation of the clearance versus the impact number

It is easy to see that the measuring data is affected by a “noise”. In order to use this data in a finite element analysis it is necessary to filter the noise.

The noise filtering was realized with a simple MSC Software Easy 5 model that use a numerical first order lag method:

$$\tau \frac{dy}{dt} + y = x$$

In this way the filtered variation becomes for the conventional punching method:

Filtered clearances for conventional punching method

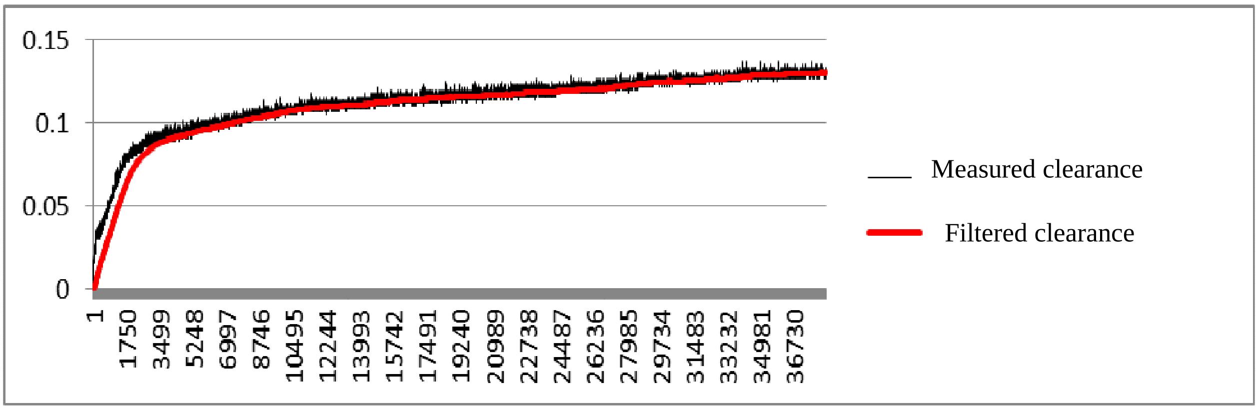

Figure 4. Measured and filtered clearance

Also, for smooth punching method the filtered results are:

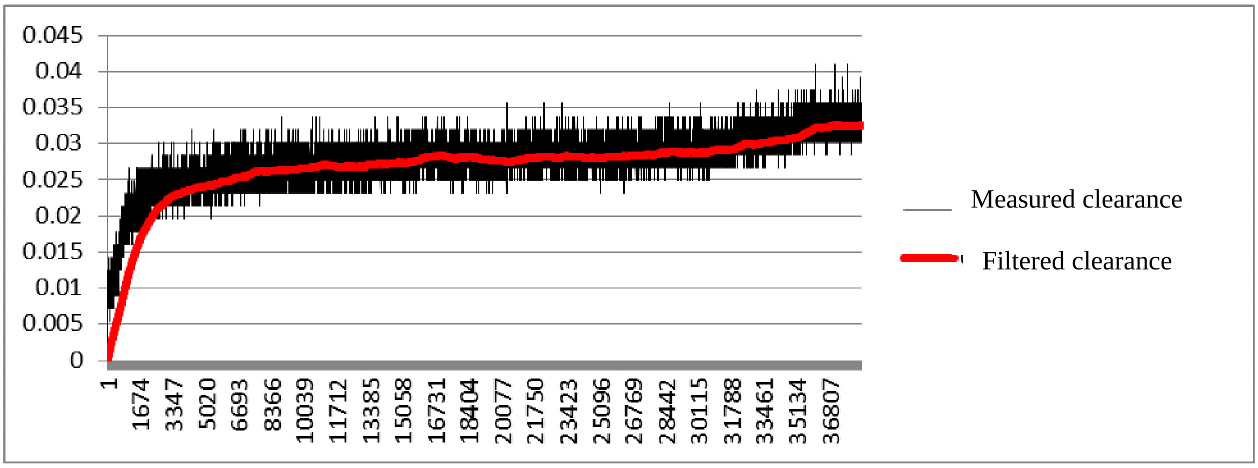

Figure 5. Filtered clearances for smooth punching method

3. Finite element model and displacements and stress results

As already presented the goal of this study is to obtain the stress and displacement distribution variation versus time for different values of clearance between the grove teeth of the brake disk and central pinion. The Patran model is presented in Figure 6.

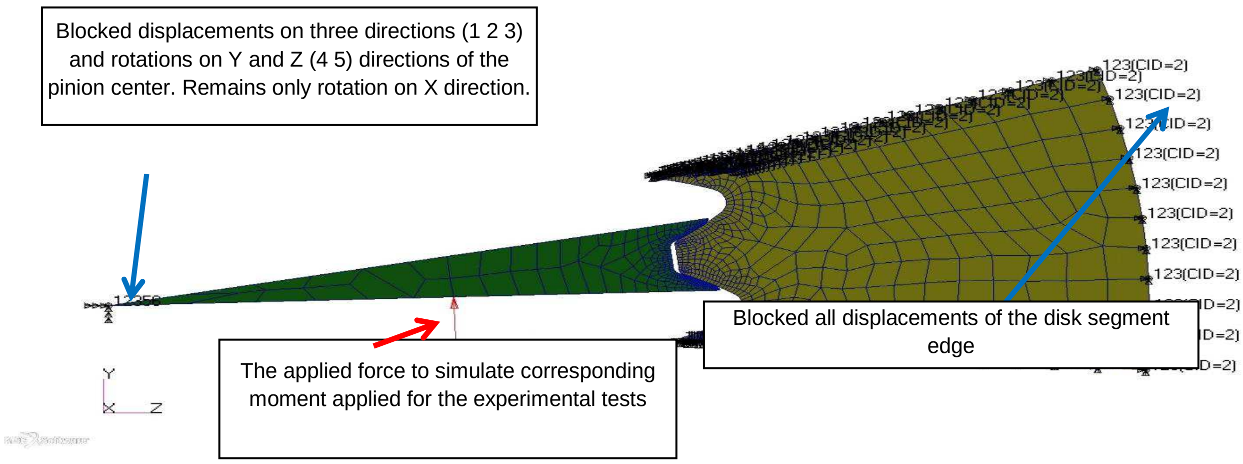

Figure 6. Patran finite element model to simulate the impact between the brake disk and central pinion

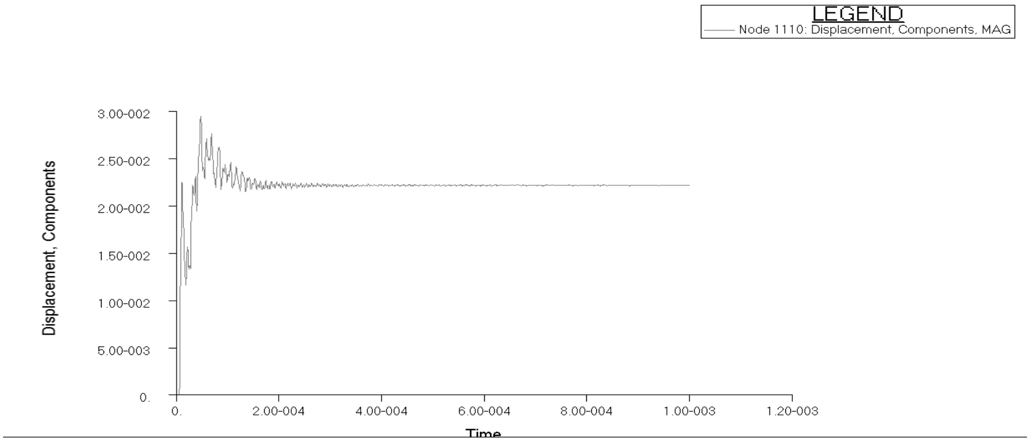

Figure 7. Displacement micro-vibration in impact area

The MSC Nastran that was run to obtain the results was an explicit transitory analysis based on SOL 700 algorithm that includes LS Dyna modules. For the impact between the central pinion and the break disk a 3% material damping was considered.

The specific clearances used for this suite of analysis are (the values are expressed in mm):

0.02450 0.02950 0.03450 0.03946 0.04450 0.04947 0.05459 0.05952 0.06470 0.06961 0.07448 0.07937 0.08460 0.08947.

For all of these values a similar response was observed: micro-vibrations that attenuates after 1 millisecond.

The typical structure response is presented in figure 7 for displacement.

4. Conclusions

The impact between the brake disk and central pinion produces significant micro-vibrations in the impact area that can affect the durability of the system. This study will be made in a further analysis.

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THE ANALYSIS OF THE CLEARANCE ON THE DURABILITY OF THE GROOVES ASSEMBLIES

ANALIZA INFLUENȚEI JOICULUI ASUPRA DURABILITĂȚII ASAMBLĂRILOR CANELATE

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Abstract: *This paper is a continuation of the paper entitled "The analysis of the influence of the clearance on the impact stresses at grooves assemblies" elaborated by the same authors. Based on the experimental results regarding the impact clearance evolution at brake disks in automotive industry and on impact stress distribution computed with a preliminary finite element model, the authors developed a fatigue and a durability analysis for the assembly taken into consideration. Two technological methods were considered for the brake disks: conventional punching and smooth punching. Based on the fatigue and durability analysis made with MSC Fatigue program the authors prove that the smooth punching procedure conducts to a better durability of the considered break disk.*

Keywords: *analysis, influence, groove, assembly, durability.*

1. Introduction

This paper is a continuation of the paper entitled *"The analysis of the influence of the clearance on the impact stresses at grooves assemblies"* elaborated by the same authors. Using the finite elements results obtained from the impact analysis of the impact between a brake disk and the central pinion, the authors demonstrate the influence of the clearance in the durability of the brake disk.

Similar as in the previous analysis two technological methods are taken into consideration:

- The conventional punching where the resulted cutting section is characterized by a higher harshness and a more accentuated taper.
- The smooth punching where the resulted cutting section is smooth with lower taper

As proved in the above mentioned paper, because the clearance the impact between the brake disk and the central pinion, the impact is accompanied by displacement and stress micro-vibration. This effect is a short time process damped in about 1 millisecond.

Using the MSC Fatigue program developed by MSC Software corporation, the authors associated this micro-vibration phenomena to a fatigue phenomena.

Studying the fatigue and the damage associated to the impact micro-vibration it was possible to calculate the damage associated to one impact.

It is also important to emphasize the fact that the authors put into evidence a correlation between the clearance and the fatigue damage associated to one impact. In this way was possible to calculate a durability for the brake disk and pinion assembly for the two technological methods of punching – conventional and smooth

2. Fatigue model and theoretical considerations

In Patran was created a suite of models with same properties but with different clearance. The considered clearance values are (expressed in mm):

0.02450	0.02950	0.03450	0.03946	0.04450
0.04947	0.05459	0.05952	0.06470	0.06961
0.07448	0.07937	0.08460	0_08947.	

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For each of these values was run a transient SOL 700 based MSC Nastran analysis. The time step for these analysis was $1x10^{-6}$ seconds. For each time step the solver generated a stress and displacement distribution in the brake disk tooth.

An example od such distribution is presented in figure 1.

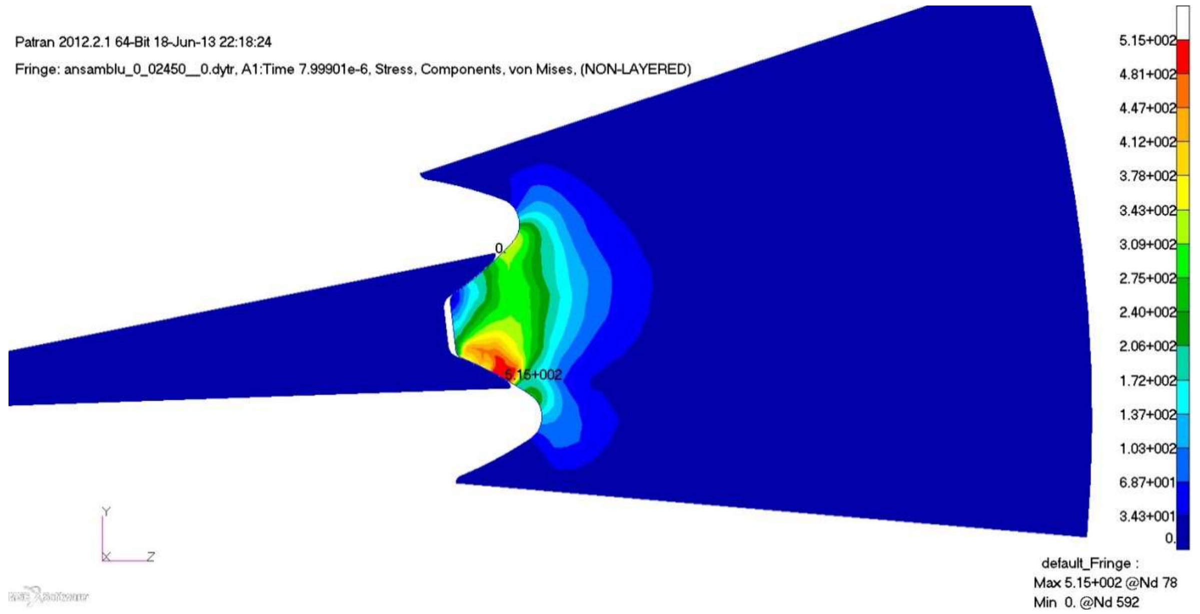

Figure 1. Sample of stress distribution in disk tooth. Equivalent von Misses stress [N/mm^2].
Clearance = 0.0246 mm, time = $7.99901 \cdot 10^{-6}$ sec

MSC Fatigue has implemented semi-empirical relation in order to obtain cyclic fatigue properties from monotonic material properties. In this way was possible to obtain a stress-life diagram associated to the brake disk material

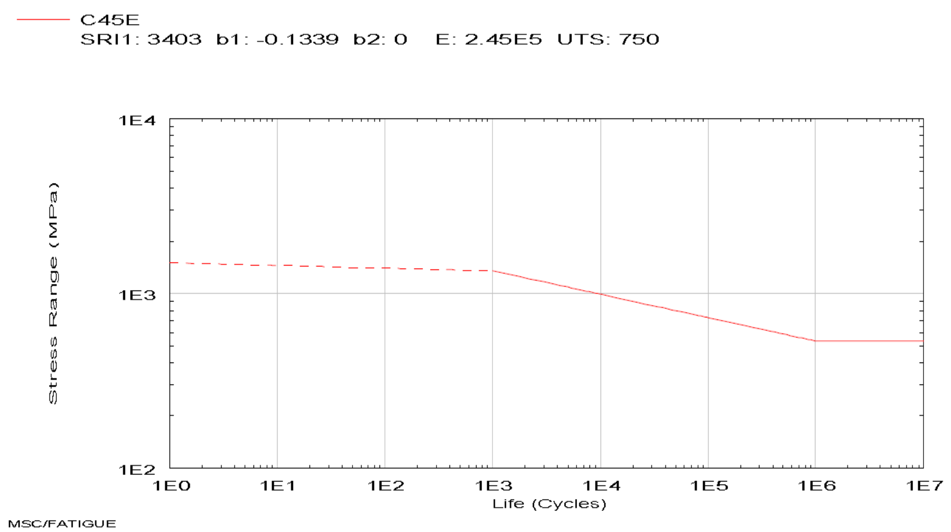

Figure 2. Stress-life diagram for C45E material obtained in MSC fatigue.

Based for these consideration for each impact/clearance case studied was calculated a fatigue damage distribution.

An example of such damage distribution is presented in figure 3 and 4.

În 1924 A. Palmgren propose the following rule:

The fatigue crack occures when the sume of the all the fatigue damages that correspond to all the ciclic loads becomes unitary.

Figure 3. Damage distribution for the minimal clearance. Maxim damage 7.81x10^{-8}.

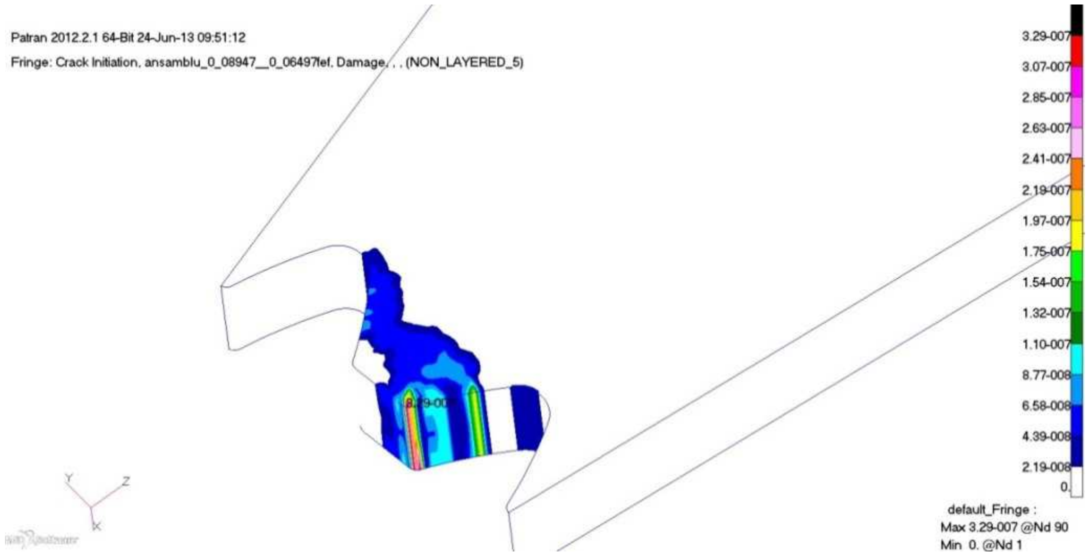

Figure 4. Damage distribution for the maxim clearance. Maxim damage 3.29x10^{-7}.

This rule was taken and popularized in 1945 by M. A. Miner. From here cmes the rule name: Palmgren-Miner.

The mathematical expression of the rule is: the crack occur when

$$\sum_i D_i^{n_i} = 1 \qquad (1)$$

According to Palmgren-Miner rule the damage field that correspond to a specific clearance was amplified with the number of impact cycles between the current clearance and the next considered clearance.

This way, in Patran, based on the each damage distribution, amplified with the corresponding number of impact cycles was possible to obtain a cumulative damage for each of the considered technological method to punch the disk brake.

Considering the inverse value of the cumulative fatigue damage – fatigue life, results that

- for the conventional punching the impact cycles at which the brake disk teeth will resist is $3.25 \cdot 10^6$ cycles
- for the smooth punching the impact cycles at which the brake disk teeth will resist is $9.19 \cdot 10^6$ cycles.

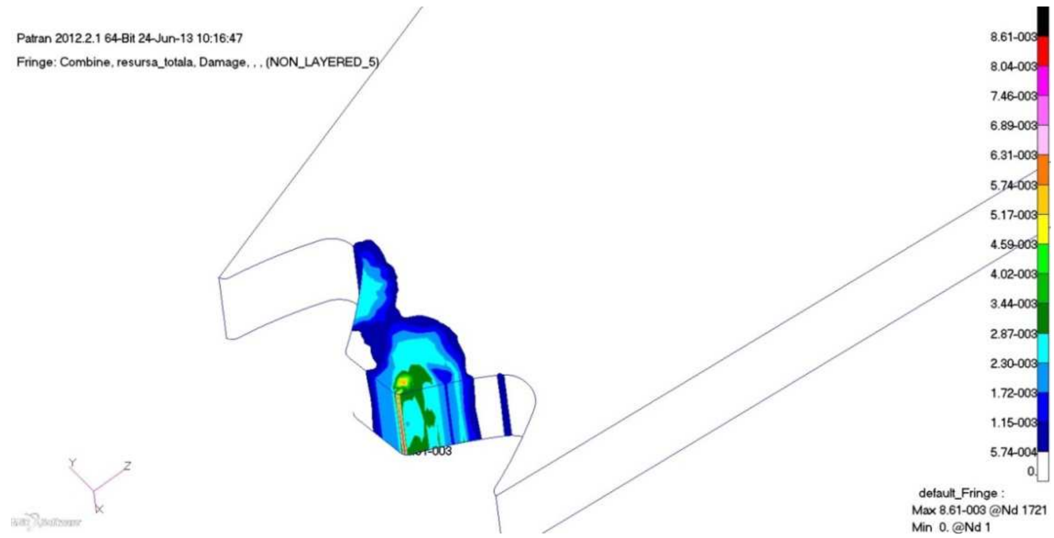

Figure 5. Cumulative fatigue damage for the conventional punching. Maxim value $8.61x10^{-3}$.

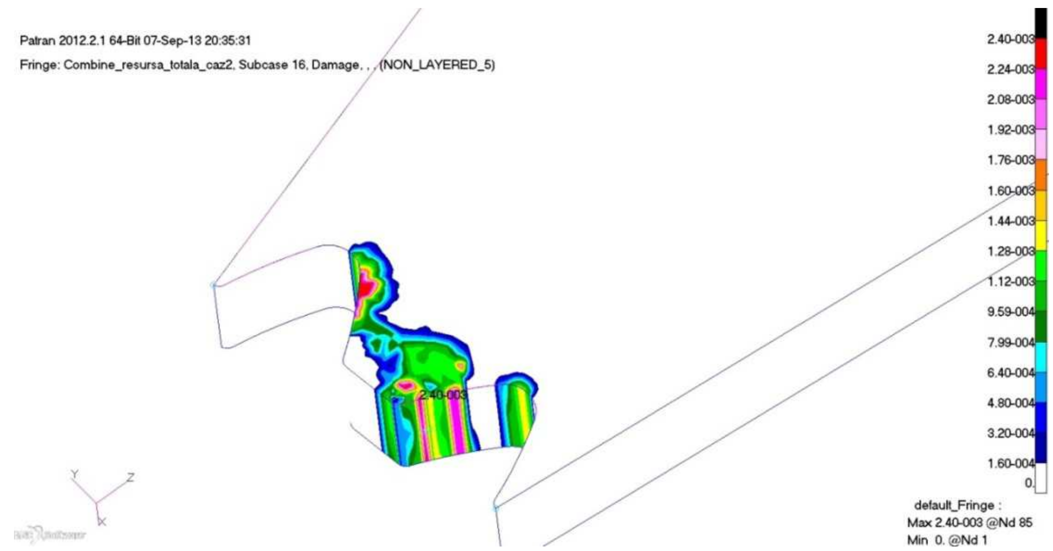

Figure 6. Cumulative fatigue damage for the smooth punching. Maxim value $2.4x10^{-3}$.

3 Conclusions

The technological method used to manufacture the brake disk has a significant influence on the disk durability. Using the smooth punching method the durability will be around three times greater.

References

[1] Explicit Nonlinear User's Guide – MSC Corporation 2012.
[2] MSC Fatigue User's Guide – MSC Corporation 2012.

THE EVALUATION OF KINEMATIC MEASURES WITHIN THE PROCESS OF OVERTAKING MOTOR VEHICLES

EVALUAREA MĂRIMILOR CINEMATICE ALE PROCESULUI DEPĂŞIRII AUTOVEHICULELOR

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Abstract: *In the present study are evaluated the kinematic measures within the process of overtaking motor vehicles from a mathematical point of view, in different driving situations represented through physical models. It is considered that accidents can be prevented once the driving situations are known. In the evaluation of the kinematic measures which characterize the overtaking of motor vehicles the following are taken into account: the variants of making an overtake, frequently came across during driving; the consecutive steps of the overtaking process; the environmental conditions - the general ambiance, the weather conditions, the day-night alternance, the most unfavourable time intervals, the limitation of visibility, the reduction of road adherence etc. - with influence over the human factor; the length and speed of each vehicle involved in the overtaking process. For each of the approached overtaking variants and the different states of the driver - expecting danger; having a normal behaviour in situations which present an imminent danger; driving during dawns and dusks; having the number of perceived elements bigger than four in the decision-making process etc., the obtained results show the variations of the safe distances between the vehicles, both while exiting the driving lane and during the initiation and the ending of the re-entrance in the lane of the overtaking vehicle, depending on the perception-reaction time of the driver-vehicle ensemble and the traveling speed of the vehicles. Taking into account the development tendency of the systems which improve the qualities of the vehicles regarding the avoidance of producing accidents, the developed calculation module may underlie the future overtaking assistance systems.*

Keywords: *motor vehicle, driver, overtake, safe distance, traffic accident.*

Rezumat: *În lucrare se evaluează, din punct de vedere matematic, mărimile cinematice ale procesului depăşirii autovehiculelor, în diferite situații din conducerea auto surprinse prin modele fizice. Se consideră că odată ce sunt cunoscute situațiile din conducerea auto, pot fi prevenite accidentele rutiere. La evaluarea mărimilor cinematice care caracterizează depăşirea autovehiculelor se ține seama de: variantele de efectuare a depăşirilor, frecvent întâlnite în practica conducerii auto; etapele consecutive ale procesului depăşirii; condițiile de mediu - ambianța generală, condițiile meteorologice, alternața noapte-zi, intervalele orare cele mai defavorabile, limitarea vizibilității, reducerea aderenței carosabilului etc. - cu influență asupra factorului uman; lungimea şi viteza fiecărui autovehicul implicat în procesul depăşirii. Pentru oricare din variantele de depășire abordate și diferitele stări ale conducătorului auto - se aşteaptă la pericol; are un comportament normal în situațiile care reclamă un pericol iminent; circulă în perioadele de răsărit şi crepuscul; numărul de elemente percepute, în vederea luării unei decizii, este mai mare de patru etc. -, rezultatele obținute surprind variațiile distanțelor de siguranță între autovehicule, atât la desprinderea din coloană cât și la inițierea și sfârșitul revenirii în coloană a autovehiculului care efectuează depășirea, în funcție de timpul de percepție-reacție al ansamblului conducător-autovehicul și vitezele de deplasare ale autovehiculelor. Ținând seama de tendința de dezvoltare a sistemelor care îmbunătățesc calitățile autovehiculelor referitoare la evitarea producerii accidentelor, modelul de calcul dezvoltat poate sta la baza proiectării unor sisteme de asistare la depășire.*

Cuvinte cheie: *autovehicul, conducător auto, depăşire, distanță de siguranță, accident rutier.*

1. Introduction

In street traffic the environmental conditions - the general ambiance, the weather conditions, the day-night alternance, the most unfavourable time intervals, the limitation of visibility, the reduction of road adherence, the day of the week, the hours of the day etc. - have a significant influence over the human factor [1, 2, 3, 5, 6, 7, 9, 11].

The perception-reaction time of the driver is variable, taking into account his age and tiredness,

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the climatic conditions, the number of external stimuli which may affect the state of the driver. For the situations which present imminent danger, a value of the perception-reaction time between 0.8 and 1 second reflects the normal behaviour of a driver of age 25 to 35 years old, well rested, with a medium driving experience, which normally looks forward and not being previously averted of a possible accident danger [1, 2, 3, 10, 14]. Compared to the situations in which the driver does not expect any danger and normally looks forward, if he is previously averted or if he drives on a road section or in specific danger generating conditions (if he expects danger), his perception-reaction time is shorter with up to 40% [1, 2, 3, 10]. If the perception-reaction time is shorter, the decisive manoeuvre will be done more quickly and the chances of avoiding or eliminating the accident increase.

In some situations, the perception-reaction time increases as it follows [1, 2, 3, 5, 8, 9, 10]:

- with 15…20% in the conditions of driving on slippery roads (wet, snowy, with slime or glaze);
- with 15…50% when the number of perceived elements in the decision-making process is over four;
- with 20…30% for the dawn and dusk periods;
- with 25…50% in reduced visibility conditions (rain, snow, fog, dark);
- with approximately 50% while the cellphone is used;
- with approximately 160% in case of momentary blindness from the powerful glow of the headlights of another vehicle, if an obstacle is spotted during the recovery period or immediately after it.

During the overtaking process [1, 2, 3, 5, 10, 14] a relatively large number of not easily predictable elements cauzality reports need to be perceived and analyzed (the speed of the overtaking vehicle and of the one that needs to be overtaken; the distance between the vehicle which intents to overtake and the one which will be overtaken; the positions of each vehicle according to the width of the road; the speed of the vehicle from the opposite lane; the distance between the vehicle intending the overtake and the one coming from the opposite direction etc.). In such cases a medium driver perception-reaction time of 3 seconds is recommended [1, 2, 3] both for the one overtaking and for the one being overtaken.

The different overtaking variants taken into account, frecquently came across while driving, are presented through physico-mathematical models. For the evaluation of kinematic measures within the process of overtaking vehicles, a numerical calculus model has been developed in which are taken into account the conditions in which an overtake is made (the overtaking variants, the traveling speed of the vehicles, the possibility of strong brakes, the state of the driver, the nature and state of the road etc.) and which allow the user to obtained the desired results with graphical interpretations.

2. The numerical evaluation method

2.1. The steps of the overtaking process

In the present study the steps of the overtaking process are taken into consideration for the situation in which no vehicle approaches from the opposite direction.

The numerical calculus model developed in the MathCAD program is based on the physical phenomena within the consecutive steps of the process of overtaking vehicles (Figure 1) [1, 2, 3, 10, 14]:

- the initial step, which takes place on the S_i distance (vehicle 1 executes an S-shaped movement corresponding to exiting the lane and retreating on a parallel direction with vehicle 2);
- the step of the parallel traveling of the two vehicles on the S_p distance, having a safe lateral distance D_t between the longitudinal axes of the vehicles;
- the final step with a trajectory also shaped like an S, but on the S_r distance, during which vehicle 1 exits the overtaking lane and comes back on the initial one.

The duration of traveling an S-shaped route (clothoid arcs) by vehicle 1 (Figure 1) with the speed v, both during the initial and the final step of the overtaking process may be calculated with the following relation [1, 2, 3, 10]:

$$t = \sqrt{\frac{D_t}{1.56 \cdot \varphi_t}}, \text{ [s]}, \qquad (1)$$

in which φ_t is the adherence coefficient on transverse direction, characterized by the nature and state of the road.

2.2. The variants of performing an overtake used in the study

Of all the variants of performing overtakes, frequently came across during driving, the following are mentioned (Figure 1) [1, 2, 3, 10]:

- *variant A*: vehicle 1 travels with the speed $v_1 = v_2$ behind vehicle 2, at a safe distance S_1. When the possibility of overtaking occurs, vehicle 1 accelerates with a_{med} and begins detaching, so that at the end of the first step (after traveling the distance S_i) it reaches the speed $v_{1i} > v_2$ ($v_{1i} = v_1 + a_{med} \cdot t$). After the parallel travel on the S_p distance (with the same acceleration a_{med}), at the end of the step vehicle 1 reaches the speed $v_{1p} > v_{1i}$, ($v_{1p} = \sqrt{v_{1i}^2 + 2 \cdot a_{med} \cdot S_p}$) and when its front passes with S_3 the front of vehicle 2 ($S_3 > L_1$, $S_3 = L_1 + S_{3s}$), vehicle 1 starts coming back on the initial lane without accelerating. The S_{3s} distance is considered so that between the rear of vehicle 1 and the front of vehicle 2 there is a t_{3s} interval of approximately 2…3 seconds [4]. Thus, it is considered that on the S_r distance of the final step vehicle 1 is traveling at a constant speed v_{1p}, and after its return on the initial lane there is a S_4 safe distance between the two vehicles;
- *variant B*: vehicle 1, having the speed $v_1 > v_2$ ($v_1 = ct.;\ v_2 = ct.$), begins overtaking 2, starting from a safe distance S_1. When the rear of vehicle 1 passes with S_{3s} the front of vehicle 2 it starts returning on the initial lane, so that after the return between 1 and 2 there is a safe distance S_4. In the overtaking process, v_1 but also v_2 are maintained constant and $v_{1p} = v_{1i} = v_1$;
- *variant C*: vehicle 1 travels with a constant speed $v_1 > v_2$, but when it arrives at a safe distance S_1 behind 2, seeing that it is possible to overtake, starts exiting the lane and simultaneously accelerates. Afterwards, vehicle 1 performs an overtake similar to *variant A*;
- *variant D*: similarly to *variant C* until vehicle 1 starts returning on the initial aisl with its speed being v_{1p}, after which it considers continuing its traveling with the same uniformly accelerated movement. After vehicle 1 travels the S_r distance, it reaches the speed $v_{1r} > v_{1p}$, ($v_{1r} = v_{1p} + a_{med} \cdot t$), and during the end of the overtake between the two vehicles there must be a S_5 safe distance.

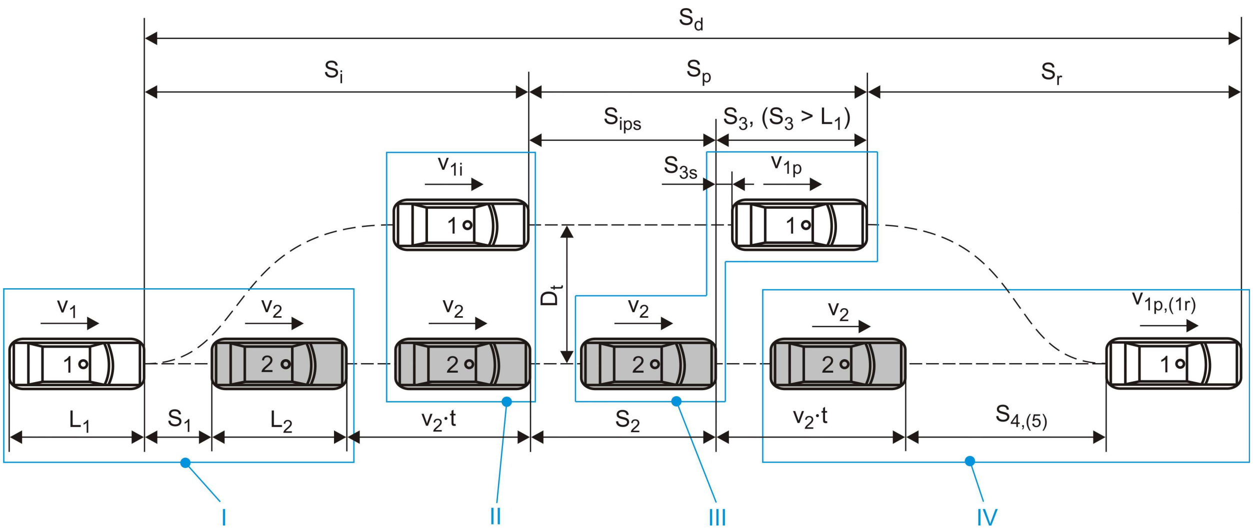

Figure 1. The positions of the vehicles during the consecutive steps of the overtaking process. 1 - the vehicle which makes the overtake; 2 - the vehicle which is overtook (v_2 = const.); $L_{1,2}$ - the length of vehicle 1, respectively of vehicle 2; I - the position in which vehicle 1 starts exiting the lane to overtake; II - the position in which vehicle 1 reaches the speed $v_{1i} > v_1$ and starts a parallel travel with vehicle 2; III - the position in which vehicle 1 reaches the speed $v_{1p} > v_{1i}$ and starts coming on the initial lane when its back passes with S_{3s} the front of vehicle 2; IV - the position in which vehicle 1 comes back in the lane and behind it is the front of vehicle 2 after the distance $S_{4,(5)}$ (the end of the coming back in the lane of vehicle 1).

For each of the variants mentioned in the development of the calculus model are taken into account the situations in which the vehicles would brake strongly [1, 2, 3, 10]:

- when vehicle 1 exits the lane, vehicle 2 might strongly brake;
- at the end of the overtake, vehicle 1 might strongly brake right after returning on the initial lane.

These situations are taken into consideration to evaluate the possibility of avoiding car collision during the overtaking process.

2.3. Notations used in the calculus model according to the conditions in which the overtake is made

For each of the *A, B, C* and *D* overtake *variants* are taken into consideration different states of the driver, symbolized so: *a* - expecting danger; *b* - normal behaviour in situations which present imminent danger; *c* - the conditions of overtaking on humid roads; *d* - the number of perceived elements in the decision-making process is bigger than four; *e* - for the dawn and dusk periods.

In the reference situation for both the state of the driver (e.g. - *a*) and the overtake variant (e.g. - *A*) the notation used will be as *„a-A"*. If a certain state of the driver is reffered to more than one overtake variant, for example, state *(a)* refers to both variant *C* and variant *D*, the used notation will be as *„a-C,D"* etc.

The values of perception-reaction times to brake of the driver-vehicle ensemble for both the one who performs the overtake and for the one who is being overtaken, depending on the state of the driver, are considered so: $t_{pr(a)} = 0.48…0.6\ s$; $t_{pr(b)} = 0.8…1\ s$; $t_{pr(c)} = 0.92…1.2\ s$; $t_{pr(d)} = 1…1.5\ s$; $t_{pr(e)} = 0.96…1.3\ s$.

Also, for each of the *A, B, C* and *D* overtake *variants* are taken into consideration different natures and states of the road on which the vehicles travel, symbolized as such: *nsr1* - dry concrete-asphalt road; *nsr2* - humid concrete-asphalt road.

In reference situation for both the overtake variant (e.g. - *A*) and the nature and state of the road (e.g. - *nsr1*) the notation will be used as *„A-nsr1"*. If a certain nature or state of the road refers to more than one variant of overtake, for example if the nature and state of the road *(nsr1)* refers to both variant *C* and variant *D*, the notation will be used as *„C,D-nsr1"* etc. In reference situation for the state of the driver (e.g. - *a*), the overtake variant (e.g. - *A*) and the nature and state of the road (e.g. - *nsr1*) the notation will be used as *„a-A,nsr1"*.

To define the adherence coefficients which characterize the nature and state of each considered road ($\varphi_{nsr1} = 0.7…0.8$; $\varphi_{nsr2} = 0.45…0.55$, [1, 2, 4, 8, 9, 12, 13] - on longitudinal direction) in the numerical calculus model, the variable $n = 1…2$ will be used, so: $n = 1$ takes into consideration the road *nsr1* and $n = 2$, the road *nsr2* (in calculations the medium values of these are $\varphi_n = \varphi_{med_n}$). For the roads with longitudinal leaning under an angle α, in the place of the φ_n coefficient, the global adherence coefficient φ_{0_n} is taken into consideration, coefficient given by a relation such as: $\varphi_{0_n} = \varphi_n \cdot \cos\alpha \pm \sin\alpha$, ("+" ascension; "–" descent). In this study, the considered road is horizontal ($\alpha = 0$).

In the case of overtakes, the traction force (tangential longitudinal) of the driving wheels of the vehicle have high values and act simultaneously with a transverse force, producing a significant reduction of the adherence coefficient on transverse direction φ_{t_n} .This is necessay to avoid transverse or tangential sliding,making sure that the resultant of the two forces - longitudinal and transverse - does not exceed the maximum adherence force when their measures and directions modify. In such situations, for transverse acceleration comfort maintaining conditions [1, 2, 3] it is considered that the overtakes will pass off with a transverse direction adherence coefficient of $\varphi_{t_n} \cong 0.8 \cdot \varphi_{a_n}$, and the sliding coefficient of $\varphi_{a_{jd}}$ represents approximately 80% of the longitudinal direction adherence coefficient φ_n.

If in the numerical model it is necessary to use an *(M)* measure which varies between a minimum value *(M_{min})* and a maximum value *(M_{max})*, considering a variable *j* which comprises values of the considered measures in the interval *($M_{min}…M_{max}$)*, a relation can be define which can be generally available for the developed calculus model, such as: $M_j = M_{min} + (j-1) \cdot \frac{M_{max} - M_{min}}{10}$, $j = 1…11$.

In order to underline the distance traveling timing S_i or S_r, the relation (1) may adapt to the considered variables, so: $t_{j,n} = \sqrt{\frac{D_{t_j}}{1.56 \cdot \varphi_{t_n}}}$.

According to vehicle class/segments, the dimensional characteristics differ from a class to another [11]. In the calculations the lenghts of the *compact* class vehicles are taken into consideration ($L_1 = L_2 = 4.1…4.74\ m$: $4.1…4.45\ m$ - the ones who have the hatchback car body; $4.4…4.74\ m$ - the ones who have the cabriolét car body, berlin - sedan - or break) [11], these ones being considered family vehicles. Taking into consideration that the width of a driving lane is of at least *3.5 m* and taking into account the widths of the *compact* class vehicles (approx. *1.74…2 m*, without the rearview mirrors) [11], in the calculations a value of approx. *3…3.25 m* is used for the lateral safety distance between the longitudinal axes of the vehicles involved in the overtaking process.

2.4. The safety distances within the process of overtaking vehicles

2.4.1.The safe distance between the vehicles in the detachment from the lane moment

When vehicle 1 exists the lane, to avoid a collision with vehicle 2 who might strongly brake, vehicle 1 should be at a safe distance S_1 (Figure 1) from vehicle 2 which allows vehicle 1 to start the same manoeuvre. The responsibility of respecting this safe distance is the obligation of vehicle 1 who performs the overtake.

Taking into account for both vehicles equal braking efficiencies, this is possible for each of the presented overtaking variants (Figure 2…4), with the following conditions met [1, 2, 3, 10]:

$$S_{1\begin{pmatrix}A\\B\\C,D\end{pmatrix}(a,b,c,d,e)\begin{pmatrix}j\\j,n\\j,n\end{pmatrix}} \geq \begin{pmatrix} S_1, \text{figure 2 (variant } A) \\ S_{11}+S_{12}, \text{figure 3 (variant } B) \\ S_{13}+S_{14}, \text{figure 4 (variants } C,D) \end{pmatrix}, \tag{2}$$

in which: $S_1 = v_1 \cdot t_{pr}$, [m]; $S_{11} = v_1 \cdot t_{pr}$, [m]; $S_{12} = \dfrac{v_1^2 - v_2^2}{2 \cdot d_{ebr}}$, [m]; $S_{13} = \dfrac{v_{1,3-4}^2 - v_1^2}{2 \cdot a_{med}}$, [m]; $S_{14} = \dfrac{v_{1,3-4}^2 - v_2^2}{2 \cdot d_{ebr}}$, [m]; $v_{1,3-4} = v_1 + a_{med} \cdot t_{pr}$, [m/s], where: t_{pr} represents the perception-reaction time in braking of the driver-vehicle ensemble 1, measures in [s]; d_{ebr} - deceleration of a strong brake, measured in [m/s^2]; a_{med} - the medium acceleration afferent to the overtake, measured in [m/s^2]; the speeds v_1 and v_2 are measures in [m/s].

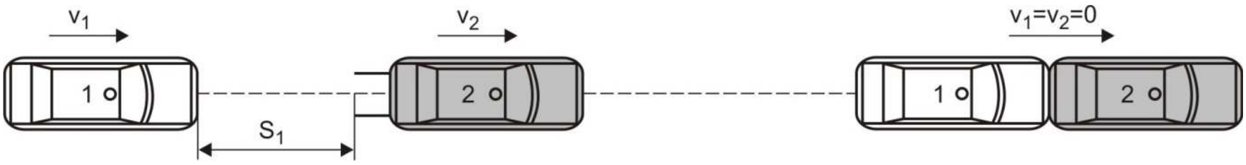

Figure 2. The safe distance in the detachment from the lane moment (variant A overtake).

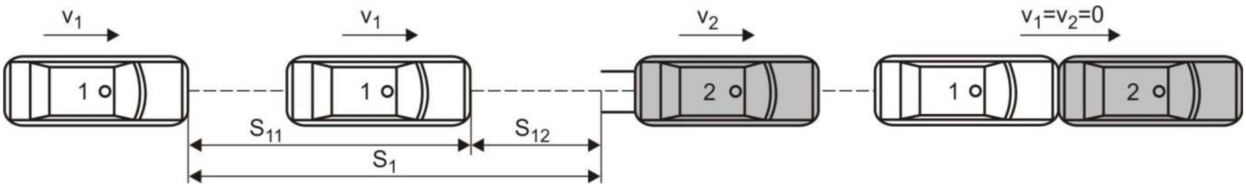

Figure 3. The safe distance in the detachment from the lane moment for an overtake with constant speed (variant B overtake).

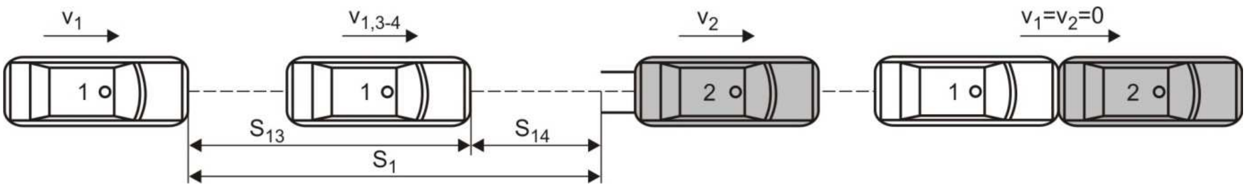

Figure 4. The safe distance in the detachment from the lane moment for an uniformly accelerated overtake (variants C and D overtakes).

To underline the variation of the safe distance S_1 in the detachment from the lane moment, according to the perception-reaction time t_{pr} of the driver-vehicle ensemble which performs the overtake, for the different overtake *variants A, B, C* and *D* and different natures and states of the road, the speeds of the vehicles v_1, v_2, the time t_{pr}, the deceleration of a strong brake d_{ebr}, are considered as such: $t_{pr} = 0.48 \ldots 1.5\ s$; $d_{ebr(A,B,C,D)_n} = \varphi_{med_n} \cdot g,\ m/s^2$, g being the gravitational acceleration; $v_{1(A)_{ct.}} = 50\ km/h$; $v_{1(B,C,D)_{ct.}} = 60\ km/h$; $v_{2(A,B,C,D)_{ct.}} = 50\ km/h$. In this calculus step in the developed numerical model the speeds $v_{1(A)_{ct.}}$, $v_{1(B,C,D)_{ct.}}$ and $v_{2(A,B,C,D)_{ct.}}$ are considered constant, having the mentioned values, and the medium accelerations [1, 2, 3] of the vehicle that overtakes, afferent to the overtake *variants A, C* and *D*, corresponding to the steps of the overtaking process (Figure 1), are considered as such: $a_{med(A)} = 1.42\ m/s^2$; $a_{med(C,D)} = 0.88\ m/s^2$. For the overtake *variant B* the speed of the vehicle overtaking is constant throughout the whole overtaking process, the medium acceleration $a_{med(B)}$ afferent to this overtake variant is null.

Also, to underline the variation of the safe distance S_1 in the detachment from the lane moment according to the traveling speed of the overtaking vehicle, in case of a normal behaviour of the driver in situations that present imminent danger, the traveling speed of the vehicle which is overtook is considered the same ($v_{2(A,B,C,D)}$ = *30…50 km/h)* for each of the *overtake variants A, B, C* and *D*. The traveling speed of the overtaking vehicle is considered as such: for overtake *variant A* - equal to that of the overtaken vehicle *($v_{1(A)} = v_{2(A,B,C,D)}$ = 30…50 km/h)*, but for the overtake *varians B, C* and *D* presented in the study - higher than the one of the overtaken vehicle *($v_1 > v_2$; $v_{1(B,C,D)}$ = 40…60 km/h)*. In this calculus step in the numerical calculus model, the speeds $v_{1(A)}$, $v_{2(A,B,C,D)}$ and $v_{1(B,C,D)}$ are comprised between the minimum and maximum mentioned values.

To determine the safe distance S_1 in the detachment from the lane moment according to the presented overtake variants and the conditions they meet, the relations (2) [1, 2, 3, 10] is adapted to the numerical calculus model, so:

$$S_{1\begin{pmatrix}A\\B\\C,D\end{pmatrix}(a,b,c,d,e)_{\begin{pmatrix}j\\j,n\\j,n\end{pmatrix}}} = \begin{pmatrix} v_{1(A)_j}\cdot t_{pr(a,b,c,d,e)_j} \\ v_{1(B,C,D)_j}\cdot t_{pr(a,b,c,d,e)_j} + \dfrac{v^2_{1(B,C,D)_j} - v^2_{2(A,B,C,D)_j}}{2\cdot d_{ebr(A,B,C,D)_n}} \\ \dfrac{\left(v_{1(B,C,D)_j} + a_{med(C,D)}\cdot t_{pr(a,b,c,d,e)_j}\right)^2 - v^2_{1(B,C,D)_j}}{2\cdot a_{med(C,D)}} + \\ + \dfrac{\left(v_{1(B,C,D)_j} + a_{med(C,D)}\cdot t_{pr(a,b,c,d,e)_j}\right)^2 - v^2_{2(A,B,C,D)_j}}{2\cdot d_{ebr(A,B,C,D)_n}} \end{pmatrix}. \tag{3}$$

2.4.2. The distance travelled by the overtaking vehicle, corresponding to the initial overtake step, the detachment from the lane and the retreat on a parallel direction to that of the overtaken vehicle

After traveling the safe distance S_i (Figure 1),

$$S_{i\begin{pmatrix}A\\B\\C,D\end{pmatrix}_{j,n}} = v_{1\begin{pmatrix}A\\B,C,D\\B,C,D\end{pmatrix}_j}\cdot t_{j,n} + \begin{pmatrix}a_{med(A)}\\a_{med(B)}\\a_{med(C,D)}\end{pmatrix}\cdot\frac{t^2_{j,n}}{2}, \tag{4}$$

coreesponding to the initial overtake step, vehicle 1 reaches the speed viteza v_{1i},

$$\begin{pmatrix} v_{1i\begin{pmatrix}A\\C,D\end{pmatrix}_{j,n}} \\ v_{1i(B)_j}\end{pmatrix} = \begin{pmatrix} v_{1\begin{pmatrix}A\\B,C,D\end{pmatrix}_j} + a_{med\begin{pmatrix}A\\C,D\end{pmatrix}}\cdot t_{j,n} \\ v_{1(B,C,D)_j}\end{pmatrix}. \tag{5}$$

For the overtake *variants A, C* and *D* the S_i distance may be calculated through the following relation:

$$S_{i\begin{pmatrix}A\\C,D\end{pmatrix}_{j,n}} = \frac{v^2_{1i\begin{pmatrix}A\\C,D\end{pmatrix}_{j,n}} - v^2_{1\begin{pmatrix}A\\B,C,D\end{pmatrix}_j}}{2\cdot a_{med\begin{pmatrix}A\\C,D\end{pmatrix}}}. \tag{6}$$

2.4.3 The distance travelled by the overtaking vehicle, corresponding to parallel traveling with vehicle 2 moment

After traveling the S_p distance (Figure 1) vehicle 1 reaches the speed v_{1p}, defined according to the overtake variant, as such:

$$\begin{pmatrix} v_{1p\begin{pmatrix}A\\C,D\end{pmatrix}_{j,n}} \\ v_{1p(B)_j}\end{pmatrix} = \begin{pmatrix} \sqrt{v^2_{1i\begin{pmatrix}A\\C,D\end{pmatrix}_{j,n}} + 2\cdot a_{med\begin{pmatrix}A\\C,D\end{pmatrix}}\cdot S_{p\begin{pmatrix}A\\C,D\end{pmatrix}_{j,n}}} \\ v_{1i(B)_j}\end{pmatrix} \tag{7}$$

where:

$$S_{p\begin{pmatrix}A\\B\\C,D\end{pmatrix}_{\begin{pmatrix}j,n\\j\\j,n\end{pmatrix}}} = S_{ips\begin{pmatrix}A\\B\\C,D\end{pmatrix}_{\begin{pmatrix}j,n\\j\\j,n\end{pmatrix}}} + S_{3\begin{pmatrix}A\\B\\C,D\end{pmatrix}_{\begin{pmatrix}j,n\\j\\j,n\end{pmatrix}}}. \tag{8}$$

The S_{ips} distance(Figure 1) may be expressed like this:

$$\begin{pmatrix} S_{ips\begin{pmatrix}A\\C,D\end{pmatrix}_{j,n}} \\ S_{ips(B)_j}\end{pmatrix} = \begin{pmatrix} v_{1i\begin{pmatrix}A\\C,D\end{pmatrix}_{j,n}}\cdot t_{ips_j} + a_{med\begin{pmatrix}A\\C,D\end{pmatrix}}\cdot\dfrac{t^2_{ips_j}}{2} \\ v_{1i(B)_j}\cdot t_{ips_j}\end{pmatrix}, \tag{9}$$

where t_{ips} (approx. 1…2 s) is the duration of traveling this distance.

To make a safe overtale, the S_3 distance (Figure 1) has to be bigger than the L_1 length of vehicle 1 which overtakes vehicle 2 [1, 2, 3], for each of the presented overtake variants. Thus, the distance S_3 (Figure 1) is given by the following relations:

$$S_{3\begin{pmatrix}A\\B\\C,D\end{pmatrix}\begin{pmatrix}j,n\\j\\j,n\end{pmatrix}} = S_{3s\begin{pmatrix}A\\B\\C,D\end{pmatrix}\begin{pmatrix}j,n\\j\\j,n\end{pmatrix}} + L_{1_j}\ , \qquad (10)$$

in which the S_{3s} distance travelled in the t_{3s} time can be expressed as such:

$$\begin{pmatrix} S_{3s\begin{pmatrix}A\\C,D\end{pmatrix}_{j,n}} \\ S_{3s(B)_j} \end{pmatrix} = \begin{pmatrix} \dfrac{\left(\sqrt{v^2_{1i\begin{pmatrix}A\\C,D\end{pmatrix}_{j,n}} + 2\cdot a_{med\begin{pmatrix}A\\C,D\end{pmatrix}} \cdot S_{ips\begin{pmatrix}A\\C,D\end{pmatrix}_{j,n}}} + a_{med\begin{pmatrix}A\\C,D\end{pmatrix}} \cdot t_{3s_j}\right)^2}{2\cdot a_{med\begin{pmatrix}A\\C,D\end{pmatrix}}} - \\ -\dfrac{v^2_{1i\begin{pmatrix}A\\C,D\end{pmatrix}_{j,n}} + 2\cdot a_{med\begin{pmatrix}A\\C,D\end{pmatrix}} \cdot S_{ips\begin{pmatrix}A\\C,D\end{pmatrix}_{j,n}}}{2\cdot a_{med\begin{pmatrix}A\\C,D\end{pmatrix}}} \\ v_{1i(B)_j} \cdot t_{3s_j} \end{pmatrix}. \qquad (11)$$

To be able to initiate the detachment from the lane and the return on the initial lane of vehicle 1 (Figure 1), the travelled distance ($S_{ips1} = S_i + S_{ips} + S_{3s}$) must be bigger than the distance ($S_{ips2} = S_1 + L_2 + v_2 \cdot t + S_2$). During ($t_{ips} + t_{3s}$) the traveling of the distance ($S_{ips} + S_{3s}$) by vehicle 1 (Figure 1), vehicle 2 travels the distance S_2, which can be expressed as such:

$$S_{2(A,B,C,D)_j} = v_{2(A,B,C,D)_j} \cdot (t_{ips_j} + t_{3s_j})\ . \qquad (12)$$

By calculating the distances S_{ips1} and S_{ips2} according to the overtake variant, the state of the driver, respectively the nature and state of the road,

$$S_{ips1\begin{pmatrix}A\\B\\C,D\end{pmatrix}_{j,n}} = S_{i\begin{pmatrix}A\\B\\C,D\end{pmatrix}_{j,n}} + S_{ips\begin{pmatrix}A\\B\\C,D\end{pmatrix}\begin{pmatrix}j,n\\j\\j,n\end{pmatrix}} + S_{3s\begin{pmatrix}A\\B\\C,D\end{pmatrix}\begin{pmatrix}j,n\\j\\j,n\end{pmatrix}}\ ; \qquad (13)$$

$$S_{ips2\begin{pmatrix}A\\B\\C,D\end{pmatrix}(a,b,c,d,e)_{j,n}} = S_{1\begin{pmatrix}A\\B\\C,D\end{pmatrix}(a,b,c,d,e)\begin{pmatrix}j\\j,n\\j,n\end{pmatrix}} + L_{2_j} + v_{2(A,B,C,D)_j} \cdot t_{j,n} + S_{2\begin{pmatrix}A\\B\\C,D\end{pmatrix}_j}\ , \qquad (14)$$

it will be verified if $S_{ips1} > S_{ips2}$.

2.4.4. The distance travelled by the overtaking vehicle by detaching from the overtake lane and returning on the initial lane, corresponding to the final overtake step

After traveling the distance corresponding to the final overtake step S_r (Figure 1), defined according to the considered overtake variant,

$$\begin{pmatrix} S_{r\begin{pmatrix}A\\B\\C\end{pmatrix}_{j,n}} \\ S_{r(D)_{j,n}} \end{pmatrix} = \begin{pmatrix} v_{1p\begin{pmatrix}A\\B\\C,D\end{pmatrix}\begin{pmatrix}j,n\\j\\j,n\end{pmatrix}} \cdot t_{j,n} \\ v_{1p(C,D)_{j,n}} \cdot t_{j,n} + a_{med(C,D)} \cdot \dfrac{t^2_{j,n}}{2} \end{pmatrix}, \qquad (15)$$

vehicle 1 remains at v_1 speed of reaches the speed v_{1r}, ($v_{1r(D)_{j,n}} = v_{1p(C,D)_{j,n}} + a_{med(C,D)} \cdot t_{j,n}$).

For the overtake *variant D* the S_r distance may also be expressed through the following relation:

$$S_{r(D)_{j,n}} = \frac{v^2_{1r(D)_{j,n}} - v^2_{1p(C,D)_{j,n}}}{2\cdot a_{med(C,D)}}\ . \qquad (16)$$

2.4.5. The total distance travelled by the overtaking vehicle

For each of the overtake variants the total overtake distance S_d (Figure 1) may be calculated through:

$$S_{d\begin{pmatrix}A\\B\\C\\D\end{pmatrix}_{j,n}} = S_{i\begin{pmatrix}A\\B\\C,D\\C,D\end{pmatrix}_{j,n}} + S_{p\begin{pmatrix}A\\B\\C,D\\C,D\end{pmatrix}\begin{pmatrix}j,n\\j\\j,n\\j,n\end{pmatrix}} + S_{r\begin{pmatrix}A\\B\\C\\D\end{pmatrix}_{j,n}}\ . \qquad (17)$$

2.4.6. The safe distance between the vehicles when the overtaking vehicle returns on the initial lane

The safe distances S_4, respectively S_5 (Figure 1, Figure 5) between the vehicles at the end of the overtake must be [1, 2, 3] big enough for vehicle 2 not to hit vehicle 1 when the latter would start a strong brake right after returning on the initial lane.

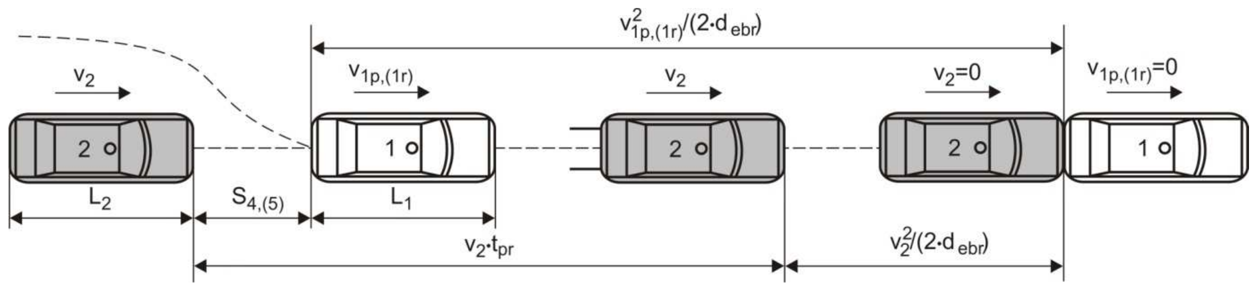

Figure 5. The safe distance when returning to the initial lane.

If right after the overtake vehicle 1 has the speed $v_{1p,(1r)}$ (Figure 5) and it starts to brake, then it would stop on a distance $v^2_{1p,(1r)}/(2\cdot d_{ebr})$. For vehicle 2 not to hit it (Figure 5) it should stop on a maximum distance of $(v_2\cdot t_{pr}+v_2^2/(2\cdot d_{ebr}))$, which is equal to the distance $(S_{4,(5)}+v^2_{1p,(1r)}/(2\cdot d_{ebr}))$, from where it results [1, 2, 3] that the safe distance $S_{4,(5)}$ (Figure 5), adapted to the considered variables (the nature and state of the road, the overtake variant and the state of the driver) is given through the following:

$$\begin{pmatrix} S_{4\begin{pmatrix}A\\B\\C\end{pmatrix}_{j,n}} \\ \\ S_{5(D)_{j,n}} \end{pmatrix}_{rel.(18)} = v_{2(A,B,C,D)_j}\cdot t_{pr(a,b,c,d,e)_j} - \frac{1}{2\cdot d_{ebr(A,B,C,D)_n}}\cdot\left[\begin{pmatrix} v^2_{1p\begin{pmatrix}A\\B\\C,D\end{pmatrix}\begin{pmatrix}j,n\\j\\j,n\end{pmatrix}} \\ \\ v^2_{1r(D)_{j,n}} \end{pmatrix} - v^2_{2(A,B,C,D)_j}\right]. \tag{18}$$

Meeting the condition (18) depends on the S_3 distance (Figure 1) when vehicle 1 starts the return move on the initial lane and when vehicle 1 is returned on the initial lane, the $S_{4,(5)}$ distance between the vehicles is determined [1, 2, 3] from the equality:

$$S_3+S_r=v_2\cdot t+S_{4,(5)}+L_1. \tag{19}$$

For each of the considered overtake variants, the S_3 distance (Figure 1) is determined according to the relations (10) and (11), and the S_r distance from which vehicle 1 detaches from the overtake lane and returns on the initial lane is determined according to the relations (15) and (16). In the moment of return on the initial lane of the overtaking vehicle, it is at the $S_{4,(5)}$ distance from the overtaken vehicle, point in which vehicle 1 reached the speed $v_{1p,(1r)}$. According to the nature and state of the road on which the vehicles travel, the safe distance $S_{4,(5)}$ is adapted to the variables refering to the overtake variant, respectively to the nature and state of the road, thus:

$$\begin{pmatrix} S_{4\begin{pmatrix}A\\B\\C\end{pmatrix}_{j,n}} \\ \\ S_{5(D)_{j,n}} \end{pmatrix}_{rel.(20)} = \begin{pmatrix} S_{3\begin{pmatrix}A\\B\\C,D\end{pmatrix}_{j,n}} \\ \\ S_{3(C,D)_{j,n}} \end{pmatrix} + \begin{pmatrix} v_{1p\begin{pmatrix}A\\B\\C,D\end{pmatrix}\begin{pmatrix}j,n\\j\\j,n\end{pmatrix}}\cdot t_{j,n} \\ \\ \dfrac{v^2_{1r(D)_{j,n}}-v^2_{1p(C,D)_{j,n}}}{2\cdot a_{med(C,D)}} \end{pmatrix} - v_{2(A,B,C,D)_j}\cdot t_{j,n}-L_{1_j}. \tag{20}$$

For vehicle 1 to return safely on the initial lane, during the overtake process the following condition must be met [1, 2, 3]:

$$\begin{pmatrix} S_{4\begin{pmatrix}A\\B\\C\end{pmatrix}_{j,n}} \\ \\ S_{5(D)_{j,n}} \end{pmatrix}_{rel.(20)} \geq \begin{pmatrix} S_{4\begin{pmatrix}A\\B\\C\end{pmatrix}_{j,n}} \\ \\ S_{5(D)_{j,n}} \end{pmatrix}_{rel.(18)}. \tag{21}$$

3. The obtained results

According to the calculus method the results are obtained with graphical interpretation of kinematic measures within the process of overtaking vehicles.

Thus the travel speeds of the vehicles $v_{1(A)_{ct.}}$, $v_{1(B,C,D)_{ct.}}$ and $v_{2(A,B,C,D)_{ct.}}$, *the variation of the safe distance* S_1, in the detachment from the lane moment, according to the perception-reaction time t_{pr} of the overtaking driver-vehicle ensemble for different overtaking *variants A, B, C* and *D* and different natures and states of the road (see the relations 2 and 3) is underlined in figure 6.

In order to overtake, in the detachment from the lena moment it is necessary to ensure a safe distance S_1 between the vehicles, according to the travel conditions and the state of the driver. For each of the considered overtake variants the safe distance S_1 rises simultaneously with the rise of the perception-reaction time at the braking of the driver-vehicle ensemble 1 (see Figure 6, table 1).

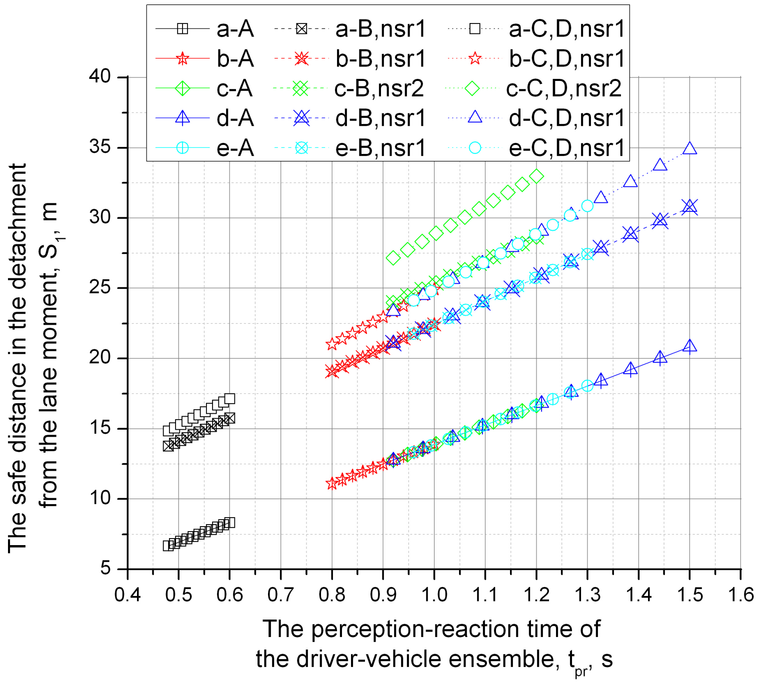

Figure 6. The variation of the safe distance in the detachment from the lane moment in accordance with the perception-reaction time (t_{pr}) of the overtaking driver-vehicle ensemble, for different overtake variants and different natures and states of the road.

Table 1.
The variation of the safe distance in the detachment from the lane moment according to the overtake variant and the perception-reaction time *(t_{pr})* of the overtaking driver-vehicle ensemble, on the *nsr1* road, in the state of the driver *b*.

The overtake variant	***b-A(t_{pr})***	***b-B,nsr1(t_{pr})***	***b-C,D,nsr1(t_{pr})***
b-A(t_{pr})	-	-39.59%	-45.40%
b-B,nsr1(t_{pr})	+65.54%	-	-9.61%
b-C,D,nsr1(t_{pr})	+83.14%	+10.63%	-

In regard to the situation which present an imminent danger, in the case of a normal behaviour of the driver, the *safe distance* S_1 in the detachment from the lane moment according to the perception-reaction time of the overtaking driver-vehicle ensemble varies depending on the state of the driver, the overtake variant, respectively the nature and state of the road according to figure 7 (with the traveling speed of the vehicles $v_{1(A)_{ct.}}$, $v_{1(B,C,D)_{ct.}}$ and $v_{2(A,B,C,D)_{ct.}}$).

The variation of the safe distance S_1 in the detachment from the lane moment according to the traveling speed of the overtaking vehicle in the case of a normal behaviour for the driver in situations which present imminent danger (see the relations 2 and 3) is underlined in figure 8 (with the traveling speeds of the vehicles $v_{1(A)}$, $v_{2(A,B,C,D)}$ and $v_{1(B,C,D)}$ comprised between their minimum and maximum values).

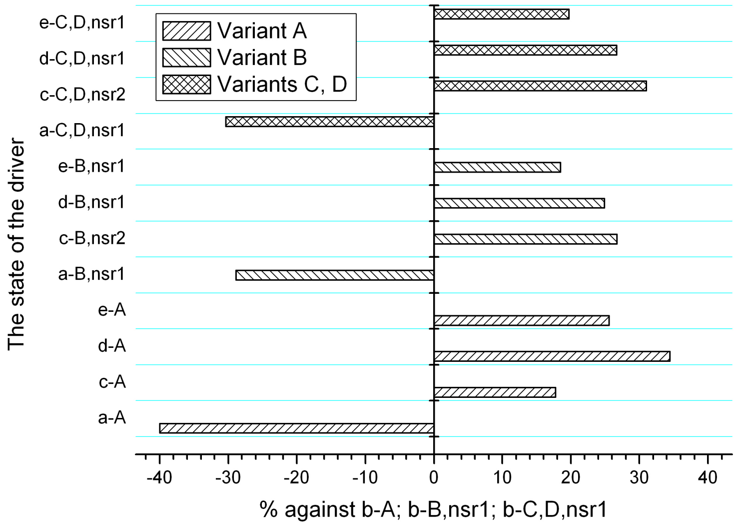

Figure 7. The variation of the safe distance in the detachment from the lane moment according to the state of the driver, taking as a base comparison the travel of the vehicles on nsr1 and the case of a normal behaviour of the driver in situations which present imminent danger (state b).

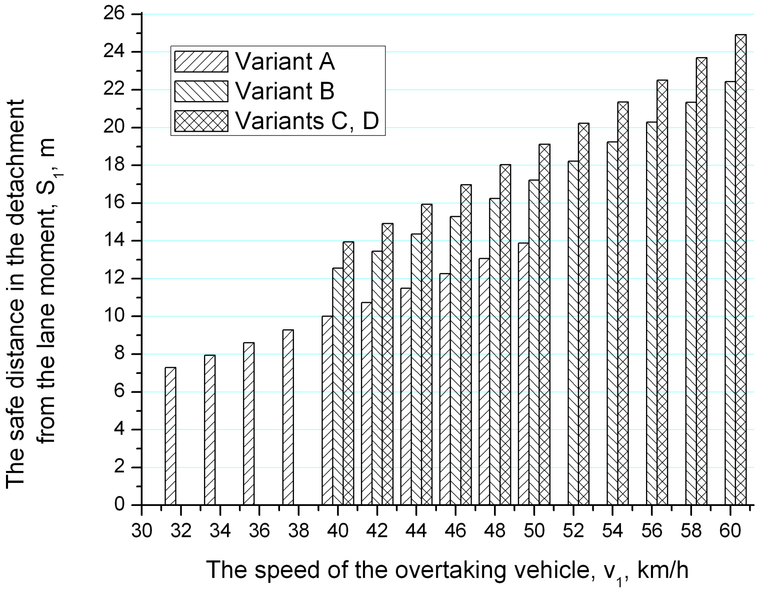

Figure 8. The variation of the safe distance in the detachment from the lane moment according to the traveling speed (v_1) of the overtaking vehicle, on the nsr1 road, in the case of normal behaviour of the driver in situations which present imminent danger (state b).

The safe distance S_1, influenced both by the variation of the perception-reaction time of the driver-vehicle ensemble 1 and by the traveling speed (the traveling speed of the vehicles $v_{1(A)}$, $v_{2(A,B,C,D)}$ and $v_{1(B,C,D)}$ comprised between their minimum and maximum values) gets bigger proportionally with the speed increase (see Figure 8). Besides the influence of the perception-reaction time of braking of the driver-vehicle ensemble 1 over the safe distance S_1, the traveling speed of the vehicles also significantly influences this distance, the latter varying according to the traveling conditions, respectively the used overtake variant (table 2).

Table 2.
The variation of the safe distance in the detachment from the lane moment according to the overtake variant and the traveling distance *(v_1)* of the overtaking vehicle, on the *nsr1* road, in the state *b* of the driver.

The overtake variant	*b-A*$_{(v_1)}$	*b-B,nsr1*$_{(v_1)}$	*b-C,D,nsr1*$_{(v_1)}$
b-A$_{(v_1)}$	-	-41.66%	-47.45%
b-B,nsr1$_{(v_1)}$	+71.40%	-	-9.92%
b-C,D,nsr1$_{(v_1)}$	+90.28%	+11.02%	-

With the traveling speeds of the vehicles $v_{1(A)}$, $v_{2(A,B,C,D)}$ and $v_{1(B,C,D)}$ comprised between their minimum and maximum values, the *variation of the safe distance* S_1 (see the relation 3) according to the traveling speed of the overtaking vehicle and the state of the driver (*a, b, c, d, e*) for the overtake *variants A, B, C* and *D* may be followed in figure 9. The conditions referring to the traveling speeds of the vehicles are considered the same with the ones in figure 8, and the meaning of the notations from figure 9 is the same as the one from figure 6.

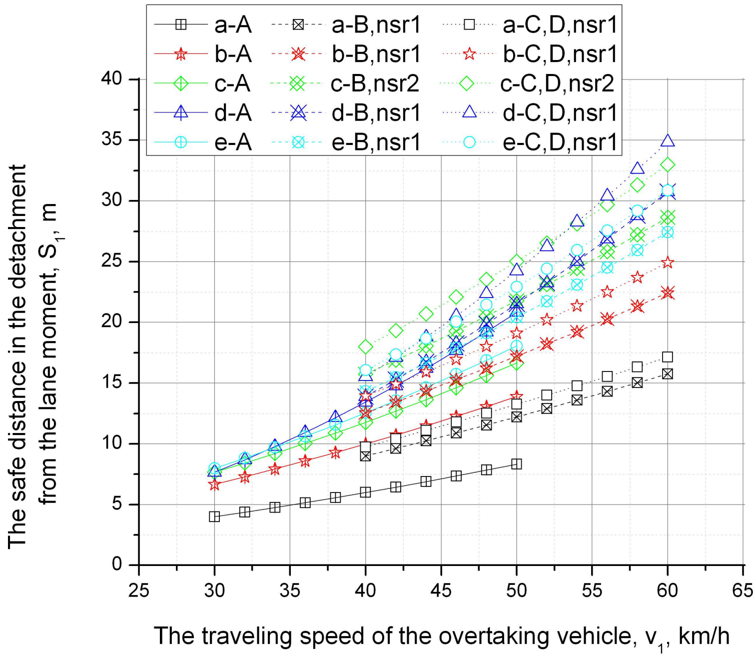

Figure 9. The variation of the safe distance in the detachment from the lane moment according to the traveling speed of the overtaking vehicle and the state of the driver.

Against the situations which present imminent danger in the case of normal behaviour of the driver, the *safe distance* S_1 in the detachment from the lane moment according to the traveling speed of the overtaking driver and the state of the driver varies according to the overtake variant and the state of the driver, according to the results presented in table 3 (the traveling speeds of the vehicles $v_{1(A)}$, $v_{2(A,B,C,D)}$ and $v_{1(B,C,D)}$ are comprised between their minimum and maximum values).

Table 3.
The variations of the safe distance in the detachment from the lane moment according to the traveling speed of the overtaking vehicle and the state of the driver, taking as base comparison the case of normal behaviour of the driver in situations which present imminent danger, with movement on *nsr1 (b-A; b-B,nsr1; b-C,D,nsr1)*.

The state of the driver and *nsr* \ **The overtake variant**		***A***	***B***	***C, D***
a	*nsr1*	-40%	-29.11%	-30.69%
c	*nsr2*	+18.02%	+26.69%	+31.05%
d	*nsr1*	+36.15%	+26.06%	+28.03%
e	*nsr1*	+26.04%	+18.88%	+20.25%

The variation on the S_i *distance* according to the overtake variant, the nature and the state of the road on which the vehicles travel and the traveling the detachment from the lane route time (see the relations 4…6) is underlined in figure 10 and accorrding to the speed v_{1i} in figure 11.

The comparative results referring to the S_{ips1} and S_{ips2} *distances* (see the relations 11…14) according to the nature and state of the road, the overtake variant and the state of the driver are presented in figure 12.

According to figure 13, depending on the nature and state of the road on which the vehicles travel, the S_p *distance* varies according to the overtake variant and the v_{1p} speed (see the relations 7…11).

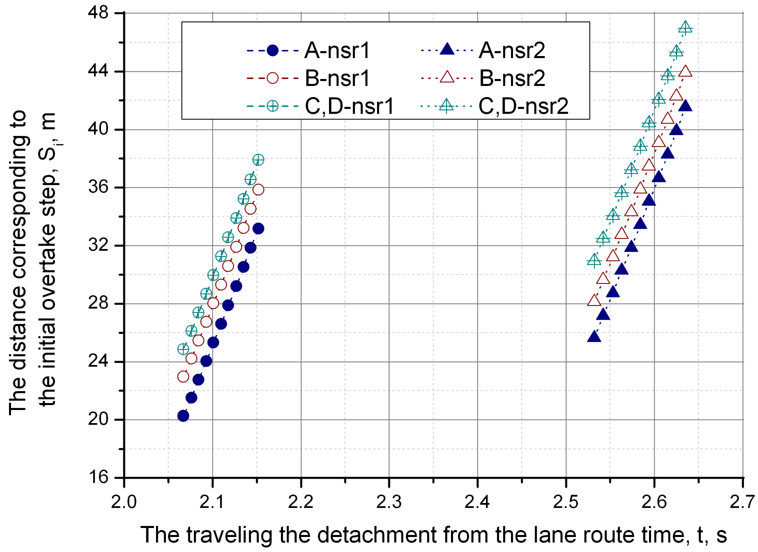

Figure 10. The variation of the S_i distance corresponding to the initial overtake step according to the traveling of the detachment from the lane route time and the nature and state of the road.

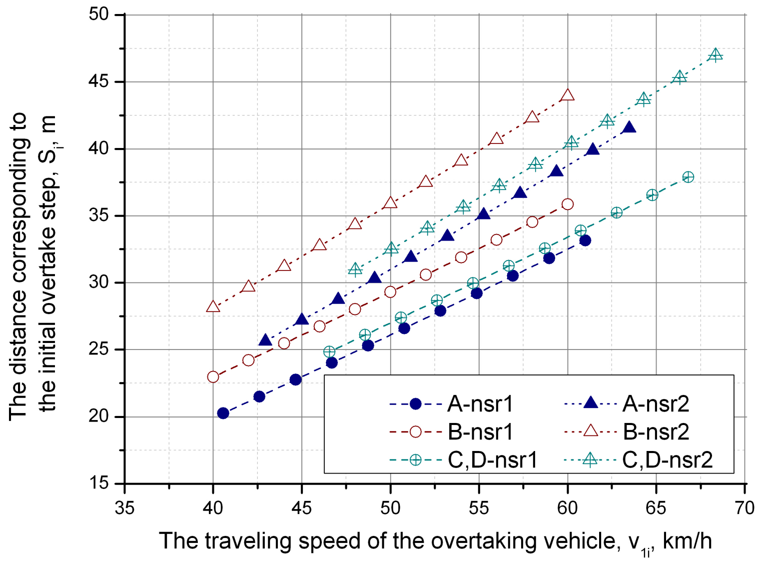

Figure 11. The variation of the S_i distance corresponding to the initial overtake step according to the traveling speed over the overtaking vehicle and the nature and state of the road.

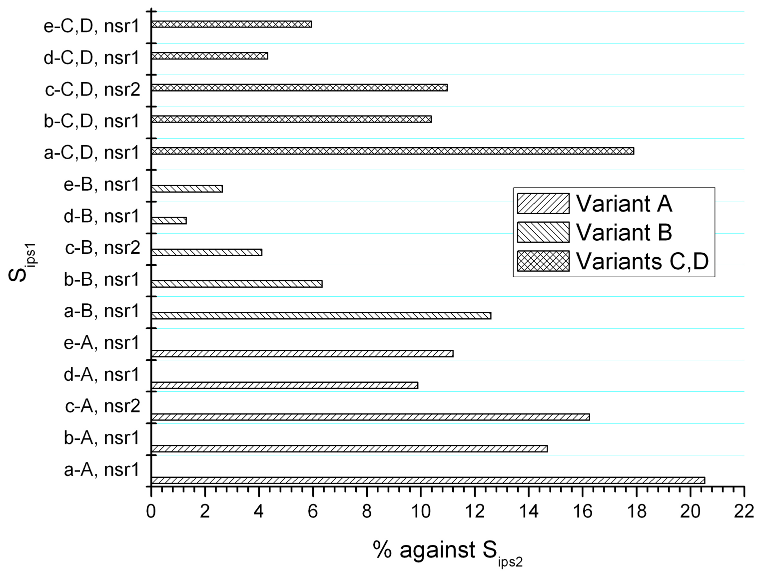

Figure 12. The comparative results referring to the S_{ips1} and S_{ips2}, ($S_{ips1} \leftrightarrow S_{ips2}$) distances according to the nature and state of the road, the overtake variant and the state of the driver.

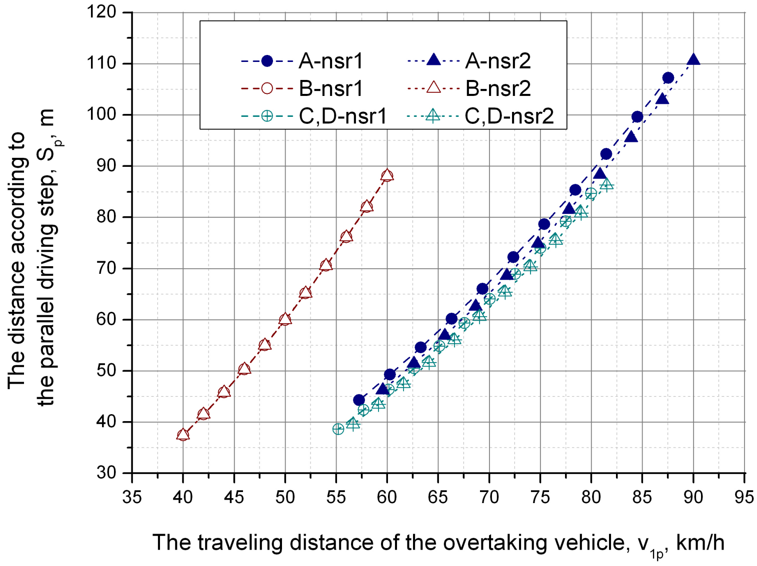

Figure 13. The variation of the S_p distance corresponding to the parallel driving step according to the traveling speed of the overtaking vehicle and the nature and state of the road.

Depending on the nature and state of the road on which the vehicles travel, the S_r *distance* varies according to the overtake variant and the time traveling the return on the initial lane route (see the relations 15, 16) as seen in figure 14 and according to the $v_{1p,(1r)}$ speed as seen in figure 15.

The total overtake distance S_d (Figure 1) (see the relation 17) varies depending on the nature and state of the road on which travel the vehicles involved in the overtaking process, the traveling speed of the overtaking vehicle and the overtake variant, according to figure 16.

Depending on the nature and state of the road on which the vehicles travel, the $S_{4,(5)}$ *distance* (see the relations 18…20) varies according to the overtake variant and the time traveling the return on the initial lane route as seen in figure 17 and according to the $v_{1p,(1r)}$ speed as seen in figure 18.

The comparative results referring to the safe distance when the overtaking vehicle returns on the initial lane (see the relation 21) according to the nature and state of the road, the overtake variant and the state of the driver are underlined in figure 19.

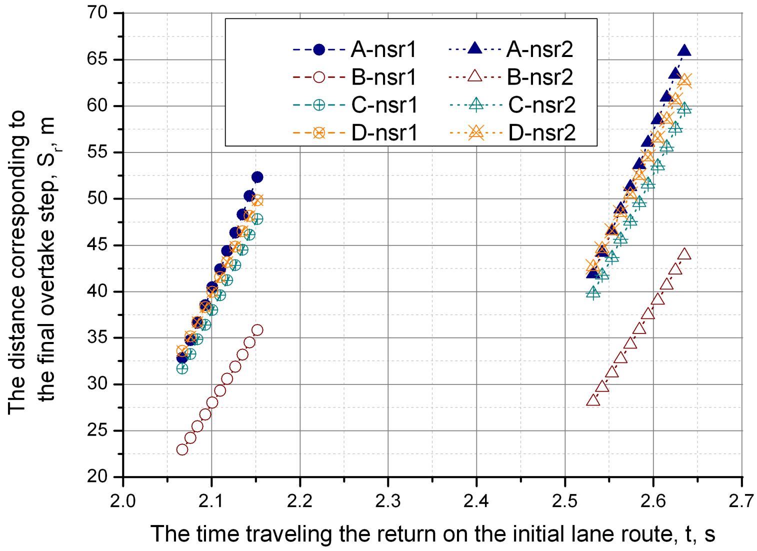

Figure 14. The variation of the S_r distance corresponding to the final overtake step according to the time traveling the return on the initial lane route and the nature and state of the road.

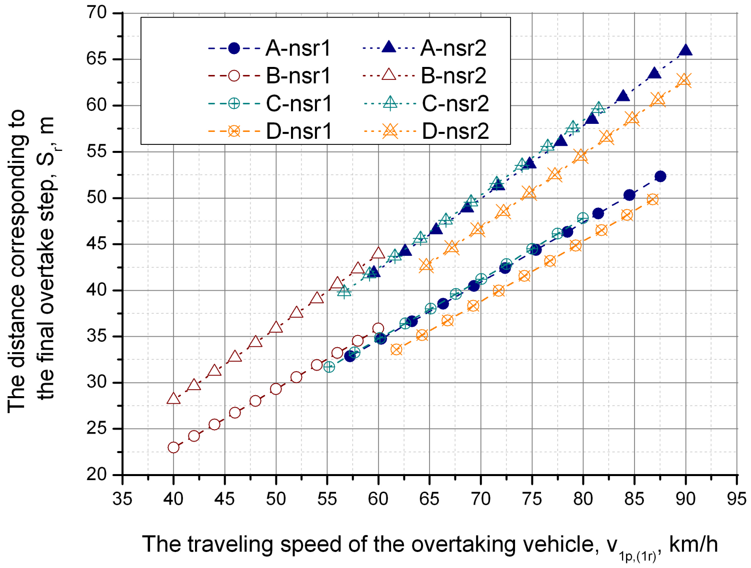

Figure 15. The variation of the S_r distance corresponding to the final overtake step according to the traveling speed of the overtaking vehicle and the nature and state of the road.

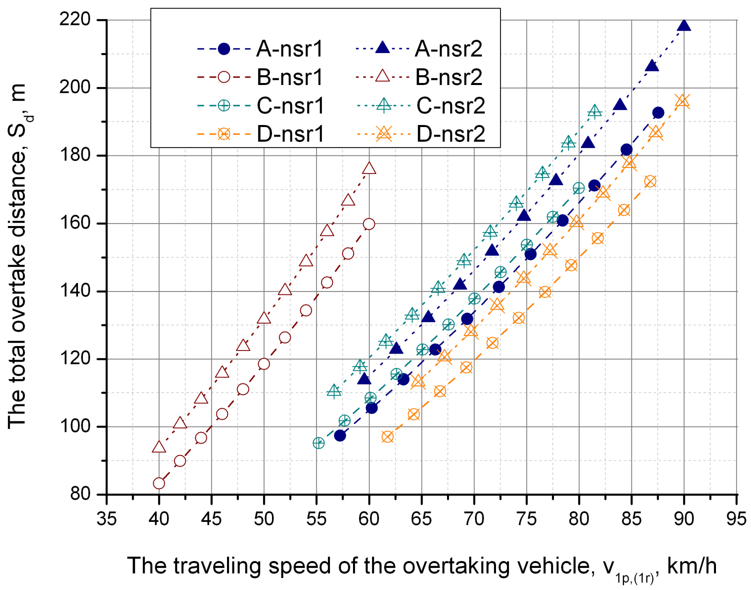

Figure 16. The variations of the total overtake distance S_d according to the traveling speed of the overtaking vehicle and the nature and state of the road.

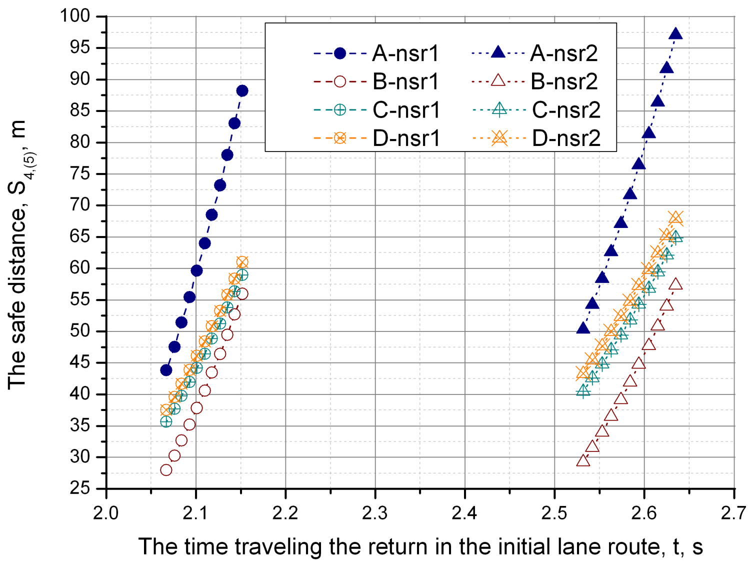

Figure 17. The variation of the $S_{4,(5)\text{-rel.}(20)}$ safe distance necessary when returning on the initial lane according to the time traveling the return in the initial lane route.

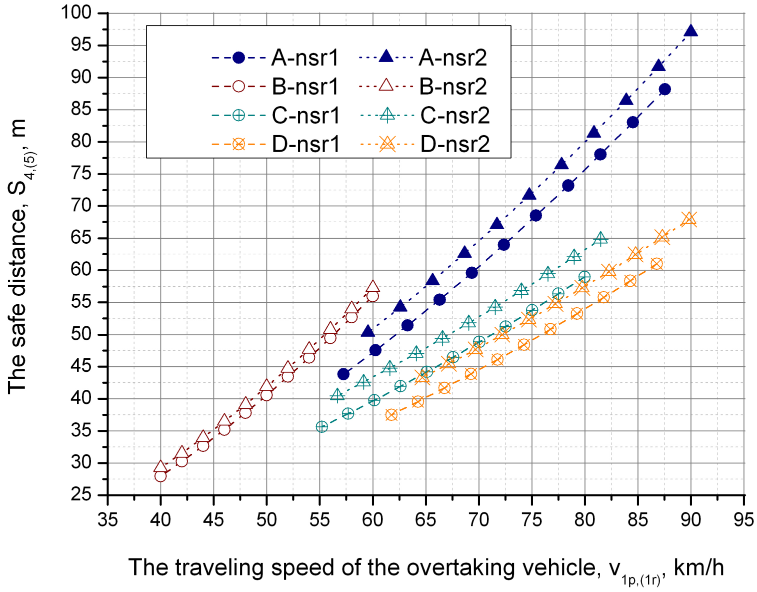

Figure 18. The variation of the $S_{4,(5)\text{-rel.}(20)}$ safe distance necessary when returning on the initial lane according to the traveling speed of the overtaking vehicle $v_{1p,(1r)}$.

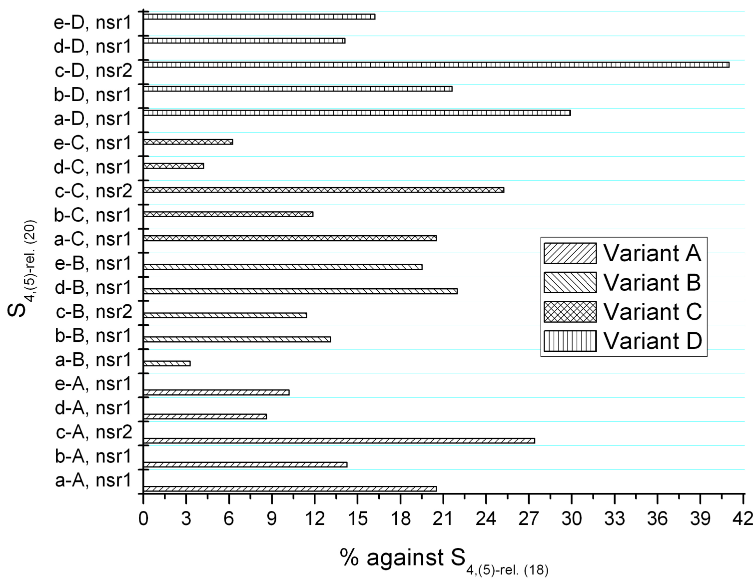

Figure 19. The comparative results referring to the safe distance when returning on the initial lane of the overtaking vehicle ($S_{4,(5)\text{-rel.}(20)} \leftrightarrow S_{4,(5)\text{-rel.}(18)}$) according to the nature and state of the roand, the overtake variant and the state of the driver.

Taking into account the comparative results referring to the $S_{4,(5)}$ distance from figure 19 it is found that condition (21) is met. As a consequence, in the conditions considered in the study, vehicle 1 can safely return on the initial lane.

4. Conclusions

The following can be concluded after browsing through this entire study:

– taking as base comparison the case of normal behaviour of the driver in situations which present imminent danger, the safe distance in the detachment from the lane moment according to the perception-reaction time of the overtaking driver-vehicle ensemble and depending on the traveling speed of the overtaking vehicle and the state of the driver, presents approximately the same variation tendency for each of the studied overtake variants;

– the obtained results referring to the safe distances specific to the overtaking vehicles steps reflects the necessity of keeping these adequate distances for each considered overtake variants;

– the increase of the overtaking vehicle speed implies he necessity of a bigger safe distance, respectively an important growth of the stopping distance (the traveled space in the perception-reaction time interval and the actual braking distance) and the driver cannot compensate this through a faster reaction, thus contributing to the accident risk growth;

- the correct estimate of the traveling speed of the vehicles and an apreciation corresponding to the needed safe distances for an onvertake performance increase the traffic safety;
- the developed numerical model for evaluating the kinematic measures within the process of overtaking vehicles allows the changing of the entry data, the taking into consideration of other conditions of performing ovetakes, respectively getting the results with graphical interpretation;
- the usage of the computerized analysis through the advantages it offers (the reduction of the projecting time, the simulation of different functioning conditions, the wide applicability in interest domains etc) becomes a useful and necessary instrument for the contemporary engineers who work in the projection, the building, the development and the safety of the vehicles, but it is to be underlined the fact that the usage of computerized analysis is not a necessary and suffiicient condition in issuing some final order considerations, but because of the complexity of presently developed mathematical models may be a trustworthy instrument used by the specialists;
- the paper can be expended by studying the overtake process in the situation when another vehicle approaches from the opposite lane; in these situations the evaluation of the dilemma zone (situated between the critical zone of stopping the overtake manoeuvre and the one with its continuation), in which neither the abandoning the overtake manoeuvre nor the one to continue it is sure.

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