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Mobility & **V**ehicle **M**echanics

Motorna **V**ozila i **M**otori

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[1] INFLUENCE OF PROTECTED STURCTURES ON TRACTOR SAFETY

Branka Grozdanić, Velimir Petrović, *IMR Institut, Beograd, Serbia*

UDC: 331.454:656.137

Abstract

Tractor has integrated protective structure configuration as sort motoring rolling stock, own motion fulfils in driving ways. When tractor as elementary units of agriculture mechanization is driven in traffic on motor ways, he must complete all safety measures and it technical demands. If necessary safety demands are not fulfil, he could be potentially dangerous drag-operating machine. From this reasons it is necessary total protection of tractor driver.

The purpose of tractor driver protection is to mount protection structures on tractor chassis, which is necessary to fulfilled safety measures for tractor driver in the case of tumbling during exploitation time in filed usage and in transport.

For increasing tractor safety, are adopted solutions in working space in tractor cabin, with goal that lead to better tractor ergonomic.

Key words: tractor, protective structure.

UTICAJ ZAŠTINIH STRUKTURA NA BEZBEDNOST TRAKTORA

UDC: 331.454:656.137

Rezime: Traktor, kao vrsta motornog vozila, svoje kretanje ostvaruje na putnim deonicama koje su vrlo složene konfiguracije. Kada se traktor, kao osnovna jedinica poljoprivredne mehnaizacije, kreće u saobraćaju na putu, on onda mora da ispunjava sve bezbedonosno – tehničke zahteve. Ukoliko se traktor ne koristi prema određenim pravilima bezbednosti, isti je potencijalno vrlo opasna vučno-pogonska mašina. Iz tih razloga je neophodna potpuna zaštita rukovaoca traktora.

U cilju zaštite rukovaoca traktora, na traktore se postavljaju zaštitne strukture , koje imaju zadatak da zaštite rukovaca u slučaju prevrtanja traktora tokom njegovog korišćenja u polju i u trasportu.

U cilju povećanja bezbednosti traktora, primenjuju se rešenja koja se odnose na zaštitu radnog prostora vozača traktora, sa ciljem da se omogući obavljanje radova uz što manje psihofizičke napore.

Ključne reči: traktor, zaštitna struktura.

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INFLUENCE OF PROTECTED STURCTURES ON TRACTOR SAFETY

Branka Grozdanić [1], Velimir Petrović

UDC: 331.454:656.137

INTRODUCTION

Tractor protective structure, i.e. safety cabin or roll bar for protection agriculture in forestry mechanisation is protective part of vehicle with role to reduce injury possibilities for driver in unwanted roll of tractor during normal usage. Protective structure is specific for free space which characterise in inner structure blanket or in space which is restricted by line-up straight lines by edge surface structure and by any part of tractor vehicles which can be in contact with road (floor), and which can not keep tractor in certain position in the case of roll. Most technical safety devices on tractor are cabin for tractor, seat, and steering wheel, braking device, lights for road and signal lights. Safety cabin has purpose that fulfils for tractor driver: protection for atmospheric condition, protection of noise and vibration, good visibility, forced ventilation and possibility of installing air conditioning.[1]

SAFETY AND SECURETY IN TRACTOR OPERATIONAL FUNCTION

For safety driving, tractor it need that working space i.e. cabin, and represent concord completely adjusted to human body. Beside effort to made working space comfort to driver there are still missing links to advantages in constructive-technical characteristics in ergonomic design, manifested in the shape of commands etc [1].

Researching of working space is base on assuming the least of all measure for volume of working space, and entry to it; tractor drivability as the number of most necessary positions for exit from vehicle in the crash case. In testing protective structure based for ``seating place of driver entry and exit`` are specificity for construction measures in agriculture tractors which are: at least measures of driver entry, number and position of at least exit in the case of danger and at least possible volume from inner free space.

Beside this testing, for cabin mounting in tractor vehicles it is necessary that cabin content static and dynamic testing methods, as acceptance procedures as well in crash test. Static method of testing protected structure assumes testing on horizontals loads and testing on forced pressure. Researching with horizontal loads means exposure front, back and sideway surface of protected structure. Pressure testing assumes application of vertical loads with beam that is transversely mounted opposed to upper parts of protected structures.

Dynamic method of testing protective structure assumes researching by appliance of impact and pressure testing.

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Impact testing of protective structure assumes exposure front and backside of protective structure of weight impact as 2000 kg swinging arm. Pressure testing implies application of vertical loads by vertical force on testing protective structure by meaning stiff beam (with radius of 250 mm).

Safety space with inner space of tractor is define on reference plate, by assuming that safe plate is rounding horizontal with seat and steering gear in static and dynamic testing, stay normal in tractor and firm under safety structure. Safety space is in figure 1.

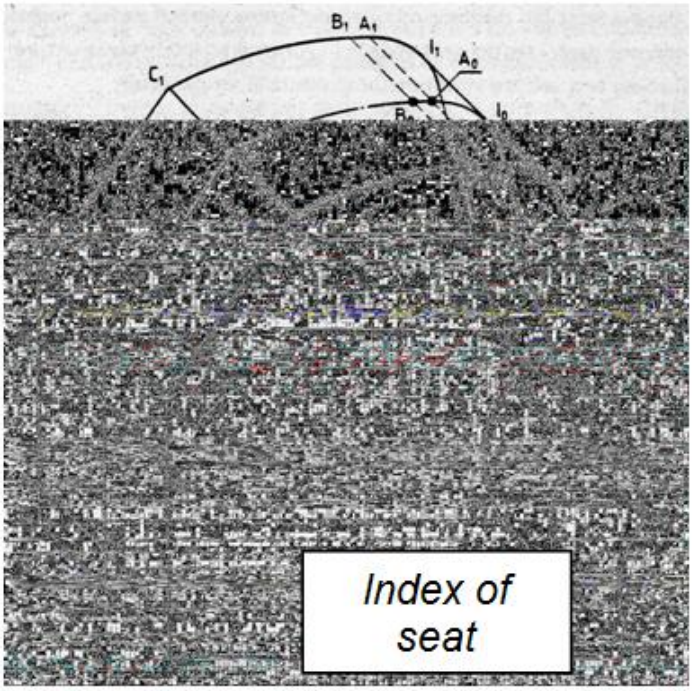

Figure 1: *Safety space*

INFLUENCE OF WORKING SPACE ON TRACTOR SAFETY

Working space consider al least min volume for driver space of any unmoving parts of structure which is available to driver during tractor usage for safety conducting tractor vehicle from driver's seat in any controlling direction.

Standards for testing working environment are declare for at least min measures of working space and min entry space for tractor vehicle conducting space, as well as number, position and at least min safety exits of safety structure i.e. cabin. Standard for testing ``driver's seat, entry and exit-measures`` are specifications for construction measures for agriculture tractor and consider on min measures for entry door, number and position and min measures for exit for inner free driving space.[2]

In figures 2. and 3. are declared min measures for driver entry doors.

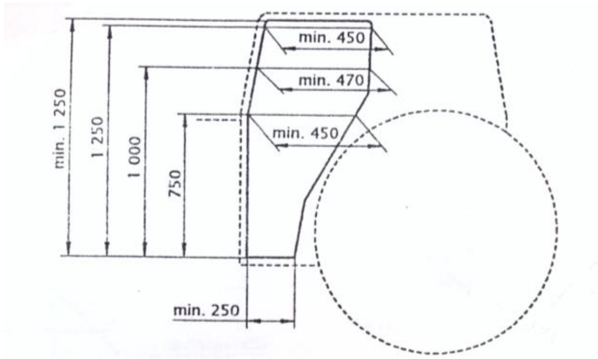

Figure 2: *Declared min measures of passage way for entry door measures for driver entry[3]*

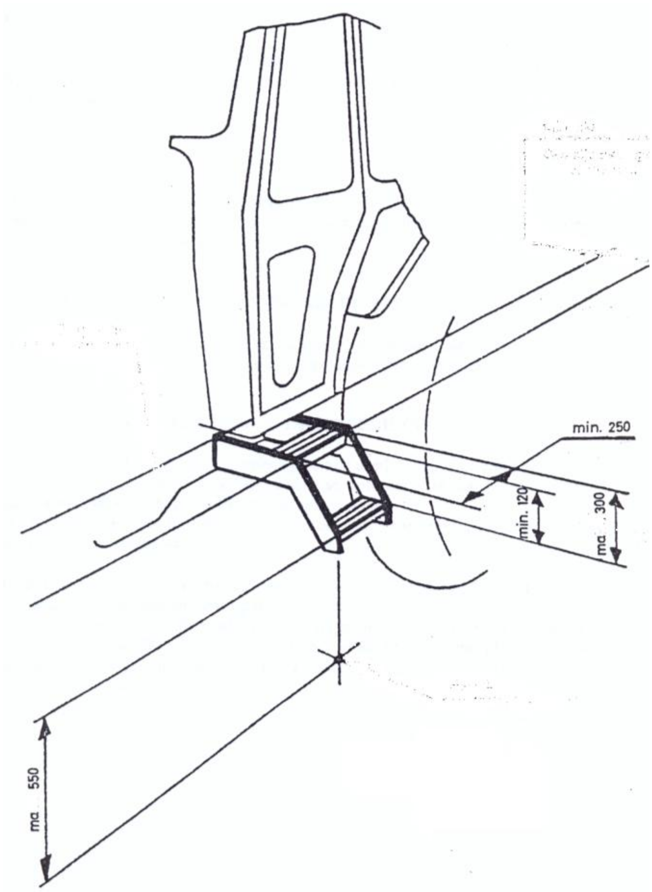

Figure 3: *Declared min [3]*

Min measures of cabin inner free space are defined on figures 4a and 4b. These declared measures are derived in vertical reference surface, tractor length measure and on center of steering wheel. These declared measures are defined for cabins with only one seat for driver. In case of danger there must be present three exits, each on different side of cabin. Front and back side as roof of cabin must consider as possible exit path (danger passage) ways. [3]

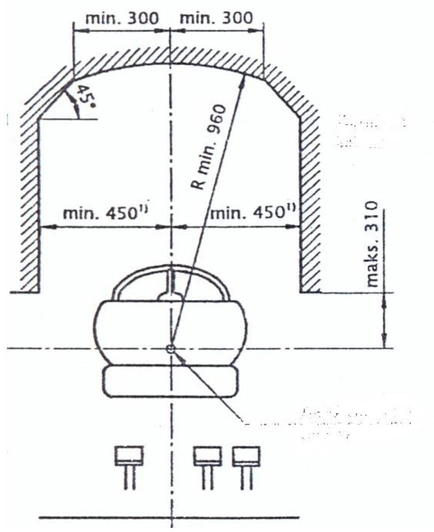

Figure 4a: *Min measures of cabin inner free space [3]*

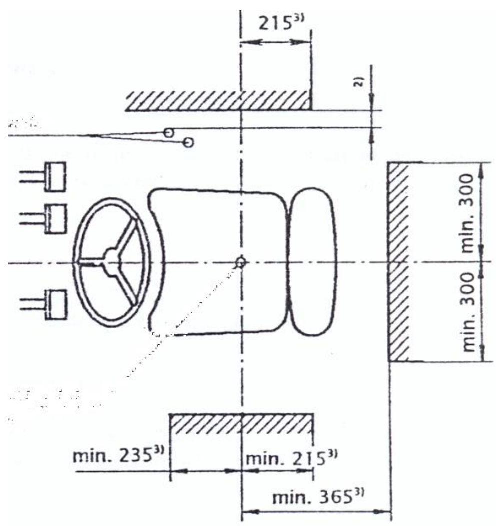

Figure 4b: *Min measures of cabin free space [3]*

TEST RESULTS

Industry of Engine Manufacturing in Rakovica, Belgrade, Serbia had conducted safety testing on cabins installed on tractors type R65-12BS., as well as testing on driver's seat, and testing for enter and exit on cabins by standard ISO 4252/1997. Tested safety frame is made from beam of cube cross section profile and bending steel body-lateral space (roof and mud covers).Safety structure –cabin, is joined on tractor chassis with four rubber mounting. Cabin has two side windows as middle widow and two doors. Tested cabin is shown on figure 5.

Figure 5: *Tested safety cabin [2]*

Measures and comparing of tested cabin are given in tables 1, 2 and 3.

Table 1**:** *Measures on doors for entry and exit (from figures 2 u 3)*

	Measures on drawing (mm)	Measured values (mm)
Min.	450	470
Min.	470	540
Min.	450	620
Min.	250	260
Max.	550	520
Min.	250	265
Max.	300	300

Table 2**:** *Inner free space in cabin (from figure 4a)*

	Measures on drawing (mm)	Measured values (mm)
Min.	300	420
Min.	300	420
Max.	310	100
R Max.	960	970

***Table 3 –** Inner free space in cabin (from figure 4b)*

	Measures on drawing (mm)	Measured values (mm)
Min.	300	480
Min.	300	480
Min.	215	320
Min.	235	320
Min.	215	320
Min.	365	460

By analyzing results are evaluated results with geometric control of cabin safety structure R 65-12 BS and all condition state it can conclude that dimension control of inner free space is satisfying for assumed and declared standards which is legislative for tractor working safety space.

CONCLUSION

In the purpose of increasing tractor's driver safety and others conductors in traffic, there are declared legislative measures that must be accomplished during tractor manufacturing. In tractor industry, it is necessary to follow this measures as well designing and mounting safety structures by EEC demands and EEC standards. This means that tractor cabin is necessary tool for safety purposes but they need to be test during research phase. Tested by legislative standards, safety cabins could be mounted on tractors, and entry serial manufacture in tractor industry.[3]

ACKNOWLEDGMENTS

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[2] - IMR Institute Report No. 230.12.

[3] Standard ISO 4252 / 1997 year.

[1] CENTRAL COMMUNICATION UNIT FOR VEHICLES WITH DIESEL ENGINE

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UDC: a) 629.3.054
b) 629.113.5 : 681.587.7

Abstract

In modern vehicles, electronic control units, which exchange information in order to utilize the resources more efficiently, are used for control of operation of some subsystems. A special central communication unit, which directs and filters information gained by data exchange between subsystems' control units, is used to synchronize the subsystems. The central communication unit is specific to every model of the vehicle because it integrates all subsystems in a whole providing necessary information for operation of all subsystems and meeting specific demands of vehicle model. Hardware and software realization of electronic control unit built in Zastava Florida vehicle with diesel engine is described in the paper.

Key words: communication, electronic control unit, CAN protocol.

CENTRALNA KOMUNIKACIONA JEDINICA ZA VOZILA SA DIZEL MOTOROM

UDC: a) 629.3.054
b) 629.113.5 : 681.587.7

Rezime: U savremenim vozilima za upravljanje radom pojedinih podsistema koriste se elektronske upravljačke jedinice koje međusobno razmenjuju informacije u cilju efikasnijeg korišćenja resursa. Za sinhronizaciju podsistema koristi se posebna centralna komunikaciona jedinica koja usmerava ili filtrira informacije dobijene razmenom podataka između upravljačkih jedinica podsistema. Centralna komunikaciona jedinica je specifična za svaki model vozila jer integriše sve podsisteme u celinu obezbeđujući neophodne informacije za rad svih podsistema i zadovoljavajući specifične zahteve modela vozila. U radu je opisana hardverska i softverska realizacija elektronske komunikacione jedinice ugrađene u vozilo Zastava Florida sa dizel motorom.

Ključne reči: komunikacija, elektronska upravljačka jedinica, CAN protokol.

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CENTRAL COMMUNICATION UNIT
FOR VEHICLES WITH DIESEL ENGINE

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UDC: a) 629.3.054
b) 629.113.5 : 681.587.7

INTRODUCTION

Motor vehicle, as a complex system, has several subsystems (engine, brakes, suspension, steering, active and passive safety systems, signalization, diagnostics ...). Each subsystem has corresponding electronic control unit. The same information is frequently required for operation of different electronic control units, so the information exchange is necessary. Also, synchronization of operation of all subsystems is necessary. Information exchange is conducted by using a serial communication network between the subsystems' control units. A special central communication unit, which directs and filters information gained by data exchange between subsystems' control units, is used to synchronize the subsystems.

Some subsystems (for example, engine, ABS, air bags ...) with their control units may be used on several vehicle models of the same or different producers. Particularity of each vehicle model manifests itself through Central communication unit, which integrates all subsystems in a whole, providing necessary information for operation of all subsystems and meeting specific demands of vehicle model.

Factory "Zastava Automobiles" has built a diesel engine DV4DT (made by Peugeot factory), which acquires some of information necessary for normal engine operation with the help of communication network based on CAN protocol, on Zastava Florida vehicle in order to modernize the vehicle and broaden a model range. Also, part of information on engine operation that is being transferred for operation of the instruments and signal bulbs on the dashboard may have been acquired only through communication network. In vehicles made by Peugeot, it is achieved through a special central communication unit, CCU, (Peugeot factory calls it BSI) that cannot be used in Zastava Florida vehicle. Thus, it has been necessary to develop a specific Central Communication Unit for Zastava Florida vehicle with a diesel engine. A small budget for development and potential small series of vehicles did not allow the engagement of some specialized firm like Bosch in development of the Central communication unit.

TECHNICAL DEMANDS FOR REALIZATION OF THE CENTRAL COMMUNICATION UNIT

The Central Communication Unit (CCU) exists in every modern vehicle that meets strict ecological and safety regulations. The control part of the unit is realized with the help of

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microprocessor in which the software is placed in EEPROM memory and may be updated if there is a need to replace the model in production or if software error is observed in exploitation.

Vehicle models produced in large series have Central communication units developed specially for those vehicles.

Vehicle models produced in small series use standard hardware units and software is developed and adjusted to the vehicle model for which it is used for. There are a few producers specialized for adoption of software of standard hardware communication units to the vehicle and the leader in this field is Bosch.

Software in control units, as well as the used algorithms, based on which the software is written, represent producer's business secret and they are not presented in public. One part of the information that is being exchanged through communication network is public and its form is prescribed by corresponding regulations (OBD for example), while the rest of the information is not documented and thus it is unavailable for public use.

Depending on the number of the subsystems built in the vehicle, there may be a swamp during information flow, so some information may lose their relevancy after some time, if only one communication network is used. In order to prevent a swamp of communication networks, the Central communication unit may be connected with two or more communication networks. Using of several communication networks demands the networks to be divided according to priority and communication speed through them.

By analysis of communication demands in Zastava Florida vehicle with diesel engine, conclusions are reached that the communication unit should have one communication channel for data exchange between engine Electronic Control Unit (ECU) and the Central communication unit and that additional electronic control units of some vehicle subsystems like ABS or airbags that have not been implemented at the start of the production, may be later connected to communication bus.

Concept of communications in Zastava Florida vehicle with diesel engine is shown in figure 1.

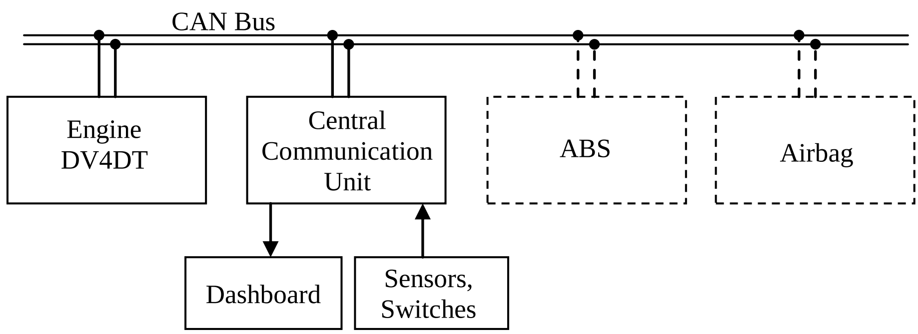

Figure 1: *Concept of communications in Zastava Florida vehicle*

The central communication unit acquires and processes information necessary for operation of diesel engine and dashboard. Accuracy of the information is checked by algorithms and, when their accuracy is established, they are coded and transformed in numeric binary form in order to send them through CAN bus to diesel engine's ECU.

The total travelled distance of the vehicle calculated on the basis of information on vehicle speed acquired from the engine's ECU is transferred to the engine's ECU as numeric information. The total travelled distance is permanently memorized in the CCU and it is needed by the ECU for forming of OBD information. If some information is not reliable or accurate, the engine's ECU is also informed by corresponding message.

The Central communication unit sends the following information as binary information to the ECU:

- brake switch state,
- inertial switch state,
- air-conditioning switch state and
- fuel level in reservoir.

A result of algorithmic check is sent with each of this information.

Based on numeric information separated from the engine's ECU messages, CCU extracts information necessary for operation of signal bulbs and instruments on the dashboard and for operation of some devices on the vehicle.

CCU on the dashboard controls operation of the following signal bulbs:

- diagnostic bulb (MIL),
- exceeding of the engine's maximum temperature,
- occurrence of water in diesel fuel,
- overheating of the diesel engine and
- proper function of the CCU.

Based on numeric information from the engine, analogue signals necessary for operation of the following instruments are generated in the CCU:

- speedometer,
- engine speed,
- engine temperature.

As an answer to air-conditioning turnign on, the engine's ECU sends information when it is permited to turn on the air-conditioning and, based on that information, the CCU engages the relays that turn on the air-conditioning.

HARDWARE AND SOFTWARE REALIZATION OF THE CENTRAL COMMUNICATION UNIT

A prototype of the Central communication unit for Zastava Florida vehicle with diesel engine type DV4DT, made by Peugeot, is made with microprocessor that has a communication module with CAN protocol and RAM and EEPROM memory on it.

Industrial prototype is made with double-sided printed circuit-board with SMD components chosen in such a way that it may work in broad temperature range from -40°C to 85°C as demanded from electronic components that are to be mounted on motor vehicles. The printed circuit-board with electronic components and a connector is placed in a metal box which decreases electro-magnetic disturbances that may unfavourably influence the operation of electronic devices. Connecting the prototype to vehicle's electric installation is done by 25-pins connector. Complete prototype of the CCU is placed under the steering wheel in immediate vicinity of the dashboard in order to reduce the length of the cables connecting the dashboard and the CCU.

A view of CCU prototype with printed circuit-board partially pulled out is shown in figure 2.

Figure 2: *Prototype of the central communication unit*

Functional zones, marked in figure 3 on , may be seen on the printed circuit-board of the CCU.

In figure 3, zone 1, there is a microcontroller with an oscillator that operates on frequency of 24 MHz. In zone 2, there are output drivers for switching on and off of the bulbs on the dashboard and of the relays for switching on the air-conditioning. Output drivers have overload protection and self check if output section is in a short circuit and cut-off and they transfer that information to the microcontroller. In zone 3, there are analogue outputs for starting the instrument for engine temperature measurement and pulse generators for operation of the instrument for vehicle and engine speed measurement. Zone 4 contains components for conditioning and multiplexing of input signals from the switch.

Conditioning and multiplexing is realized with the help of a special integrated circuit that has voltage overload protections built in. In zone 5, there are drivers for CAN bus access with filters for prevention of electro-magnetic disturbances that appear during passing of impulse signals through electric installation. Zone 6 is a connector. In zone 7, there is a voltage stabilizer that provides 5V voltage for microcontroller operation. Connector for input and update of software from microcontroller is placed in zone 8.

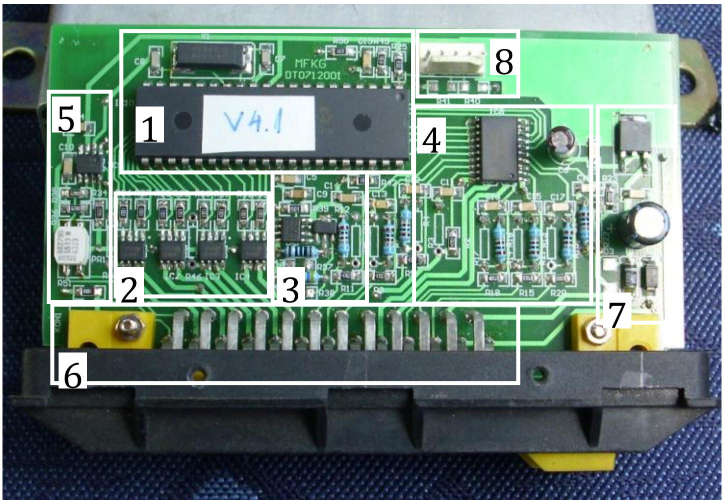

Figure 3: *Functional zones of the Central communication unit*

Electric circuit layout for connecting the Central communication unit to the electric installation of vehicle is shown in figure 4. Wires that are used for connecting to the CAN bus are realized as twisted pairs in order to eliminate electromagnetic disturbances.

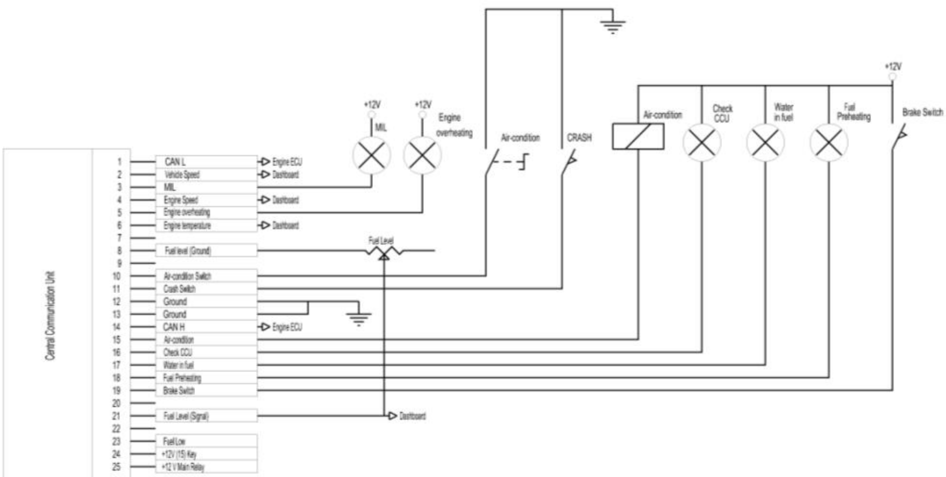

Figure 4: *Electric circut layout of the Central communication unit*

Programming code is placed in microcontroller's EEPROM memory and written in assembler in order to achieve maximal operation speed, in order words that CCU may perform control in real time. Operation in real time is necessary for mathematical transformation of numeric information gained for the engine's ECU into impulse train for sensor of vehicle and engine speed and into analogue voltage for temperature indicator.

Algorithms used in the programming code are formed on the basis of own experiences and information available from literature. Numeric information acquired from the engine's ECU, logically crossed with information acquired at the CCU's input, is used in algorithms for checking the accuracy of information generated by CCU.

Electric circuit diagram and algorithm for checking if output driver is functioning and if really desired activity, e.g. switch on the bulb, is realized (figure 5). Testing of the proper functioning is especially important for reliable determination of proper indication or malfunction of MIL bulb.

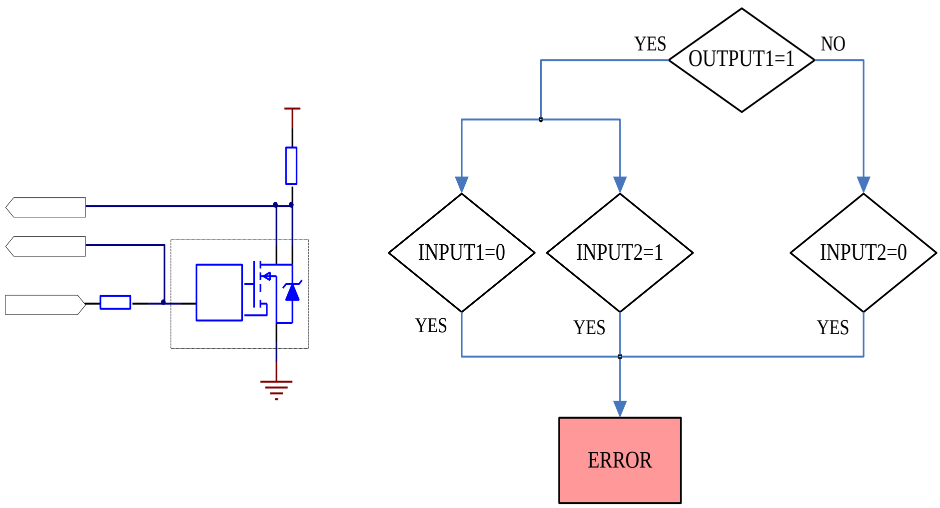

Figure 5: *Algorithmic test of proper function of CCU's output drivers*

Turning on of the bulbs and relays is realised simply by supplying the voltage of the logical unit to the input of the output driver, OUTPUT1. Algorithmic testing if load, Rp, that may be a bulb, a relay or some other resistance actuator, is really switched on, demands the use of two inputs of microcontroller, INPUT1 and INPUT2. With INPUT1, checking if the load's state is adequate to desired is performed and with INPUT2 testing of the driver itself (if it is in short circuit, overheated ...) is performed based on driver's self test.

In the case when there is less than minimum fuel in the reservoir, the engine's ECU switches to a special regime of operation. In order to test if fuel sensor is functioning properly and to determine the fuel low level, fuel gauge must be used which equivalent scheme is shown in figure 6. Algorithm for testing of fuel gauge proper function is presented in the same figure. Since the signal from the fuel gauge is analogue, analogue to digital conversion of this signal is necessary. Based on values obtained by that conversion, it may be determined if the gauge is cut off (CODE 11) or if it has connection to the ground (CODE 01).

Fuel level gauge is cut off if analogue signal measure by microcontroller is higher than:

$$V_{\max} = \frac{\mathrm{POT} + \mathrm{R2}}{\mathrm{POT} + \mathrm{R1} + \mathrm{R2}} \mathrm{VCC} \tag{1}$$

where POT is total resistance of the fuel gauge.

Fuel gauge is connected to ground if analogue signal measured by microcontroller is less than:

$$V_{\min} = \frac{\mathrm{R1}}{\mathrm{POT} + \mathrm{R1} + \mathrm{R2}} \mathrm{VCC} \tag{2}$$

Limit value of fuel minimum is experimantally determined.

Example of the applied algorithms, where engine ECU's information is used, is control of proper function of the brake switch (if information that the brake is pressed is correct), given in figure 7.

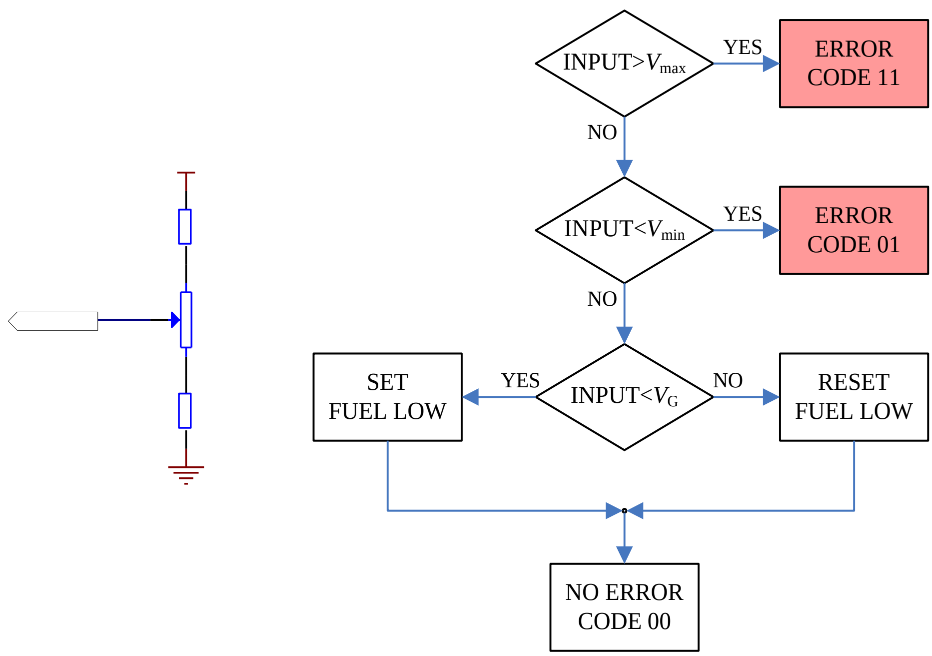

Figure 6: *Algorithmic checkihg of fuel gauge*

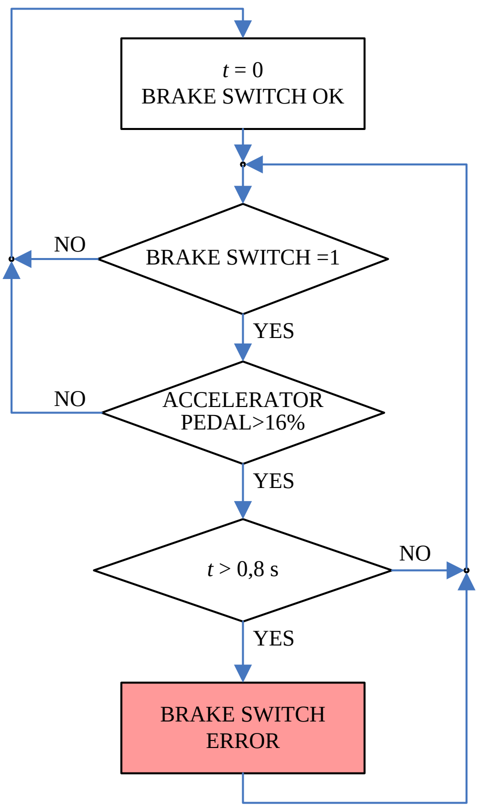

Figure 7: *Algorithmic checking of break switch proper function*

It is considered that the brake switch is defective algorithmically if:

- the brake switch is activated,
- the gas pedal is in position 16% higher than at full throttle,
- previous two conditions exist longer than 0,8 s.

Gas pedal position is obtained in numeric form from the engine's ECU through communication network and it is followed by information if this information is correct. If information is not correct, testing of the proper function of the brake switch is not performed and the CCU sends information that information on brake switch state is not correct.

Malfunctioning of the Central communication unit is signalled to the driver by especial bulb at the dashboard. Conditions for the CCU failure are non existing communication on a CAN bus, malfunction of output drivers (cutting off, short circuit or overload) and algorithmically determined failures of the input parameters.

CONCLUSIONS

Central communication unit for Zastava Florida vehicle with diesel engine is successfully both hardware and software realized. During realization, it has been shown that, in addition to functional characteristics of the device, it is necessary to take care of reliability and accuracy of information transferred to other subsystems. Specific algorithms developed for specific purpose during realization are built into software for testing the accuracy of information and for checking if output organs have properly done what has been to them. They are very often more complex and program demanding than realization of control goals.

ACKNOWLEDGMENTS

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[1] THE STUDY OF MOTOR VEHICLES SUITABILITY FOR MOUNTING COMBAT SYSTEMS

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__Slavko Muždeka,__ Military academy, Belgrade, Serbia

UDC: 623.437.4:623.438.2/.7

Abstract

At the beginning of the development a new all-terrain vehicle one of the most all-inclusive research is the research work on its suitability to have the superstructure built. Within the research work presented in this paper a number of analyses have been made with regard to the vehicle conception, mounting weapons, and also the tests with regard to the vehicle characteristics. In this paper the term suitability of the all-terrain vehicle for building superstructure on it has been introduced, i.e. the factor of suitability of the all-terrain vehicle for built-in superstructure.

The suitability factor of the all-terrain vehicle for built-in superstructure has been defined in several ways and from a number of viewpoints, i.e. a variety of criteria has been applied and they depend on the purpose of the superstructure. On the basis of the analysis that has been made for the vehicle with four axles it can be concluded that it is possible to satisfy the requirements of customers for the axis load and with a less number of axles.

The load bearing factor possibility has been defined as the ratio of the geometrical characteristics of the vehicles frames (longitudinal supporters) that are being compared.

The similar analysis has been made from the aspect of the vehicle stability and vehicle maintenance. By defining the factors of additional load it was possible to explicitly define the load increase on the tested section for two specific usage conditions and also to evaluate the successful way to build-in superstructure onto this type of vehicle.

Key words: all-terrain vehicle, combat vehicle, suitability, combat system.

PROUČAVANJE POGODNOSTI MOTORNIH VOZILA ZA MONTIRANJE BORBENIH SISTEMA

UDC: 623.437.4:623.438.2/.7

Rezime: Na početku razvoja jednog novog terenskog vozila jedno od najsveobuhvatnijih istraživanja je istraživanje njegove pogodnosti za nadgradnju. U okviru istraživanja koje je

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prikazano u ovom radu sprovedeno je niz analiza koje se odnose na koncepciju vozila, ugradnju oruđa, zatim ispitivanje karakteristika vozila.

U radu je uveden pojam pogodnosti terenskog vozila za nadgradnju, odnosno faktor pogodnosti terenskog vozila za nadgradnju.

Faktor pogodnosti terenskog vozila za nadgradnju je definisan na više načina i sa više gledišta odnosno primenjeni su različiti kriterijumi a oni zavise od namene nadgradnje.
Iz analize koja je sprovedena za vozila sa četiri mosta može se zaključiti da je moguće zadovoljiti zahteve korisnika za osovinskim opterećenjem i sa manjim brojem mostova.

Faktor mogućnosti prijema opterećenja definisan je kao odnos geometrijskih karakteristika ramova (uzdužnih nosača) vozila koja se upoređuju. Slična analiza je sprovedena sa aspekta stabilnosti vozila i održavanja vozila.

Definisanjem faktora dodatnog opterećenja eksplicitno se određuje povećanje opterećenja na posmatranom mestu za dva karakteristična uslova upotrebe i omogućava sagledavanje uspešnosti nadgradnje ove vrste vozila.

Ključne reči: terensko vozilo, borbeno vozilo, pogodnost, borbeni sistem.

THE STUDY OF MOTOR VEHICLES SUITABILITY FOR MOUNTING COMBAT SYSTEMS

Novak Vukčević [1]***, Slavko Muždeka***

UDC: 623.437.4:623.438.2/.7

INTRODUCTION

The special-purpose modern combat systems perform complex functions in the extreme working conditions. In order to satisfy the users` requirements in the development process of these systems it is necessary to define the objective (aim) function of the system on the basis of the systemic approach and consequently on such defined objective function to set out concrete requirements for individual sub-systems. Considering that one of the basic requirements is the establishing of a quality system with the shortest development cycle period and minimal costs, the justified need presupposes that new systems be developed by the integration of existing sub-systems into development of which considerable know-how has been invested and which performed quite well in practice. This approach in development can ensure a considerable flexibility and the possibility to satisfy users` requirements, but can also be unsuccessful. The reasons for a possible development failure lie in the fact that the objective function need not secure the sub-systems of top quality characteristics that can in the best way contribute to the objective function as a whole [1]. Another reason, apart from the technological capabilities of the manufacturers that can always be a limiting factor, is related to the way in which the requirement has been defined, i.e., the objective function. Namely, the user frequently defines some requirements in general terms, and other requirements, however, in detail and strict way, so that, having in mind the fact that the requirements are almost in each case mutually confronted, there exists a genuine need to make correction in the course of their development. There is always evident interaction between the user (the customer) and the constructor, i.e., the manufacturer in developing the combat systems. It should appropriately ensure that the quality of the system be assessed in the right way only in the exploitation stage. On the other hand, one of the major characteristics of a combat system is its suitability for maintenance and modernization, which is at higher levels frequently entrusted to the manufacturer, so that the importance of interaction between the manufacturer and the user becomes quite significant.

One of the basic requirements related to the combat systems performances is their mobility, which presupposes that the systems have been mounted on the platforms of various motor vehicles, whereby this requirement has been satisfied. The motor vehicle onto which the combat systems are mounted (super-structured) apart from ensuring mobility should also meet a number of requirements which in the end should ensure that a combat system is not viewed separately from the platform, but that the term combat system presupposes a vehicle together with its superstructure. In order to accomplish this it must be ensured that the interaction of the combat system and the vehicle be such that it should secure functioning of the system in all conditions with accomplishment of all assigned performances. This paper

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represents an attempt to define general characteristics of vehicles as platforms for mounting combat systems, the superstructure characteristics and the way of its connectivity with the vehicle, as well as characteristics which describe the suitability of a specified category of a vehicle for mounting a special combat-purpose system.

THE BASIC CHARACTERISTICS OF VEHICLES AS PLATFORMS FOR COMBAT SYSTEMS BUILT-IN SUPERSTRUCTURE

When considering the possibility of vehicles selection for a combat system platform one should make the analysis in at least three fields – the analysis of the vehicle characteristics in terms of ensuring the mobility as a basic requirement for the integration of the combat platform, the analysis of the combat system purpose and the level of protection that the vehicle should ensure, as well as the analysis of the workloads of a vehicle which are the result of a combat system integration.

The mobility as one of the basic characteristics of the modern combat systems is ensured by different performances of a vehicle. Given the consideration of various vehicles types onto which a combat system can be mounted it is interesting to analyze the following characteristics of a vehicle: towing performances, possibilities of clearing obstacles (manoeuvrability), braking performances, steering capability and stability. While considering the towing performances the limiting criterion is not frequently the maximum speed, as in the case with transport and passenger vehicles, but the possibility of clearing an assigned steep terrain with a specified speed, medium or maximum movement speed outside the road network, the characteristics of acceleration, the minimal speed etc. The possibility of overcoming obstacles is also being analyzed from the aspect of the possibility of clearing obstacles of little volume, low coefficient of surface adherence and a high coefficient of rolling resistance, high obstacles (a vertical wall etc.), obstacles like ditches, water obstacles and so on. Considering the braking performances there are no specific requirements, except that the criteria for assessment of the efficiency of a braking system in certain segments can be more complex. The steering performances are very important, especially bearing in mind the fact that here we deal with the vehicles of large dimensions that are supposed to move across rough terrains or along the roads that have limited geometry. Stability is very significant and complex characteristic which has a major impact on the vehicle selection. It is important to note that the stability is analyzed as a static – through the analysis of the vehicle performance at acceleration stage, braking, moving round a bend and similar. It is especially interesting to consider the vehicle stability while accomplishing the objective function of a combat system.

The analysis of a combat system and the safety level that a vehicle should ensure gives the answer to a basic question – whether the combat armoured vehicle can be accepted as a platform or whether the platform can represent a highly manoeuvrable and passable all-terrain vehicle. In the category of combat armoured vehicles three types of vehicles are being considered – heavy combat vehicles based on the tank platform, light combat caterpillar-type vehicles based on the combat infantry vehicle and combat vehicles based on the platform of the armoured wheeled vehicle. In the category of the highly manoeuvrable all-terrain vehicles the all-terrain wheeled vehicles specially developed for applications in defence matters are being considered. These vehicles are different from commercial vehicles since their sub-systems have been so designed to ensure a high degree of manoeuvrability, and also have special sub-systems that ensure certain additional performances. Considering the fact that application of combat vehicles largely influences the safety level it is clear that the armoured platforms will be the candidates for those combat systems that are exposed to the direct enemy fire and the operation which can be accomplished if the crew is protected in the armoured section. Another major criterion for platforms selection is the unification of vehicles within a combat unit. This criterion can be even more important, so that, for instance, the tank platform is in most cases also used as a bridge pillar, that is to say, as a logistic vehicle in tank units because in this way the logistic support is simplified and the vehicle crews training is made easier in the process of combat systems exploitation. In figure 1 there is a tank as a basic combat weapon, as well as combat systems developed on the tank platform – bridge pillar and tow tank. The selection preference for a combat vehicle selection as a platform for a combat system can be also be related to an available contingent of armoured vehicles. Namely, of late many armies have at their disposal a larger number of tanks and armoured carriers of older generations which are no longer necessary for their basic use. There is a tendency to carry out the so-called conversion which encompasses also the modernization of a vehicle as a platform and in this way ensure vehicles that will be suitable for super-structured combat systems.

1a)

1b)

1c)

Figure 1: *The heavy tracked combat vehicle as a platform for different super-structured combat systems, a) tank, b) mechanised bridge, c) logistic vehicle*

The all-terrain vehicles of high manoeuvrability whose principal use is the transportation of troops and equipment are of major interest as the platforms of mounted combat systems. These vehicles are in most cases used for various mounted combat systems whose basic use is related to the operations outside the immediate combat contact. Since we deal here with vehicles whose manufacture price, as well as the costs of logistic support in the course of the life cycle are significantly lower than the price and the costs of the life cycle of armoured vehicles, of late there has been done significant research work in order to ensure the integration of different platforms onto the vehicle of this type. The unification as a major parameter for the selection of combat platforms also in the selection preference of this kind has a major impact. Thus, for instance, within artillery units, in case the artillery weapons are being mounted onto the vehicle, as a platform can be used a vehicle that can perform some other functions – transportation of ammunition, troops etc. In figure 2 is shown the vehicle with the variant of a built-in crate in case it is a transportation vehicle, as well as self-propelled multiple rocket launcher mounted onto the same vehicle.

2a)

2b)

Figure 2: *The all-terrain vehicle of high manoeuvrability as a platform for various mounted combat systems, a) transport vehicle, b) self-propelled multiple rocket launcher*

The analysis of the workloads of vehicles that are result of a mounted combat system has a great significance in the vehicle selection process. Apart from the mass and the superstructure mass centre of gravity position as a parameter for the vehicle selection that influence the performances of the vehicle itself, a great importance is also given to the way of establishing connectivity between the vehicle and superstructure, the way of distributing the load, the ratio of workloads in case the combat system does not perform its function and the case when the combat system performs its function, geometrical characteristics of a combat system in various situations and so on. An important element for analysis of workloads is also the fact that in case of armoured vehicles the connectivity between the superstructure and the vehicle is established through the armoured section, and in case with the all-terrain vehicles through a high manoeuvrability via the frame.

THE VISUALISATION OF A VEHICLE DESIGN CONCEPTION

The development of a combat system integrated onto the motor vehicle platform on the one hand presupposes the selection of the most suitable platform (of a vehicle), and on the other, the selection of the most suitable way to mount a combat system onto the platform. Both these tasks imply the analysis of a large number of possible variants which is not feasible without the application of modern methods and computer modelling techniques. The computer modelling is in this case a powerful tool for the analysis of a large number of solutions and the selected solution optimization.

Since the vehicle is an assembly of mutually interdependent integral parts, which accomplish the assigned objective by their joint operations, the systemic approach to the analysis of such a technical system enables us to analyze the characteristics of one section or a system from the aspect of its impact on other sections, i.e. on its place and role in the whole system. As the basic functions of a vehicle have in most cases mutually confronted requirements, it is necessary to achieve their mutual coordination, in other words their simultaneous satisfaction, so that it is necessary to start with the requirement for the whole vehicle, and then to make the analysis of the system impact on its component parts.

Systemic engineering is supposed, on the basis on the previously set requirements, to enable the production of specification of necessary operational characteristics of the technical system with the proposal of the most appropriate configuration and the organization of the major sections. In order to achieve this it is necessary to ensure a high degree of confidence in the results obtained without the prior production of the physical prototypes, that is, on the analysis of the virtual model. By applying the systemic approach we define not only the requirements in relation to the developed system, in other words its components, but the way in which to achieve their satisfaction.

In the course of the combat system designing integrated onto the vehicle the level of built-in characteristics should constantly be checked up and compared with the set requirements prior to the designing of the physical prototype. Thus the need arises to develop the appropriate methodology by which it is possible to design, simulate, check, and even test the system at the level of the virtual prototype.

Today the designers have at their disposal a large support and assistance in the modern development techniques of the so-called “virtual” models which make it possible to assess the requirements satisfaction of the monitored system prior to the completion of the first physical prototype. In this way the controlled virtual environment is established in which the preconditions for the meeting of basic requirements of the modern vehicle development are achieved.

A great advantage of designing by means of the virtual modelling process is reflected in the possibility to prevent human errors in the course of designing, which enable the simulated tests. These tests, considering that they are conducted on the virtual model, offer an objective assessment of the right and successful solution prior to the start of the physical model test.

In this way is also changed the role of tests which have so far been conducted exclusively on the physical model. These tests are now preceded by virtual tests. The visualization in essence represents a mediator between the monitored output characteristics of the system, that is being simulated and the constructor (designer). The software for visualization enables the designers to use the tri-dimensional visual data, creating by this the idea of the performance of a designed system or its components in real time, in the course of performing the objective function.

The designers have nowadays at their disposal a large number of a variety of programs from the virtual modelling domain. Software starts mainly from the 2D drawing, then through 3D parametric modelling to checking up of kinematics, dynamics, heat and other calculations and virtual tests of the 3 D models. In figure 3 is shown the 3 D model of the frame of an all-terrain vehicle [2] with the elements of an addition frame, stands (legs) and the elements of superstructure (mount structure).

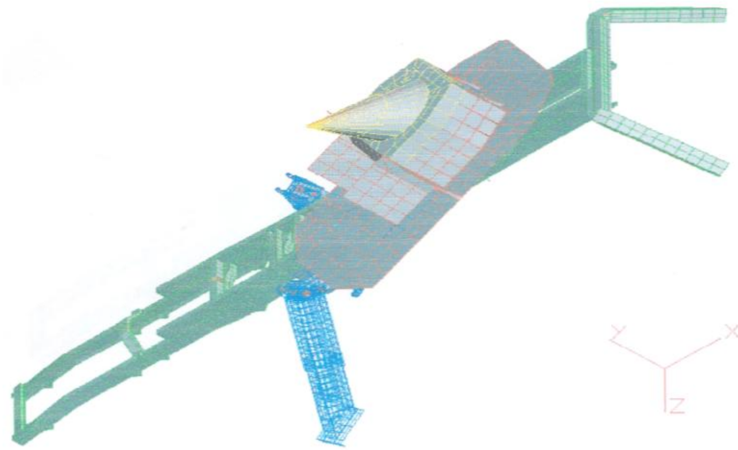

Figure 3: *3 D model of the frame of a measurement vehicle with the elements for superstructure construction*

THE ANALYSIS OF THE ALL-TERRAIN VEHICLE SUITABILITY FOR MOUNTED ARTILLERY SYSTEM

In the analysis of the conception of the special superstructure vehicle, from the aspect of suitability for superstructure construction (bearing in mind that it will carry powerful weapons) the initial criteria should be defined consequently leading to the designing conception. In case of the mounted artillery system onto the all-terrain vehicle it is important to consider the following:

- the barrel position of a weapon in the march formation (backwards or in the direction of moving),
- the barrel position in the combat position (most likely in the opposite direction from the moving, driving),
- the allowance of ammunition size (space),
- requirements in terms of building in accessory equipment,
- the number of the crew,
- requirements for an additional cabin for the crew,
- angle of fire in line, left and right of the vehicle longitudinal axis,
- necessary elevation,
- requirements for the ammunition magazine containing projectiles and
- requirements for existence of an electro-hydraulic system for the vehicle stabilization at a firing stage, movement of the weapon in assigned directions and elevation and loading of weapon.

Within the framework of development research [3, 4] which has been conducted in the Military Technical Institute the analysis of suitability of all-terrain vehicles for mounted artillery system has been made. The analysis has been made for two groups of vehicles:

- all-terrain vehicles of the drive formula 8x8 and
- all-terrain vehicles of the drive formula 6x6

To make the analysis of suitability of all-terrain vehicles for mounted artillery weapons a selection has also been made of the number of bridges and the number of drive bridges. The total number of bridges depends on the total mass of weapons and possible specific pressure onto ground. The specific pressure depends on the axis load and the contact surface between wheels and ground, and is conditioned by the ground capacity.

Considering that this weapon will be employed on the terrains and on the surfaces with little capacity, the axis load should be within the limits from 5.000 daN to 8.000 daN. Consequently, for the expected total mass of weapons of 25 000 kg to 30 000 kg, the total number of bridges should be from 3 to 4. If the maximum adhesion weight is planned to be achieved, it is necessary for all wheels to be drive wheels since in this case the adhesion weight equals the overall mass of the weapons, which is the main condition for ensuring the maximum mobility and manoeuvrability of a vehicle.

The analysis of positioning the weapon onto the all-terrain vehicle has been made for vehicles FAP 3032 BS/A 8x8; KAMAZ 63501, 8x8; TATRA 8x8; URAL 8x8; MZKAT 8x8; FAP 2026 6 t, 6x6; KAMAZ 43118, 6x6; TATRA 6x6; URAL 6x6 and MZKAT 6x6. Within the preliminary analysis of satisfying basic technical requirements the following activities have been carried out:

- measurements of masses and their positions aboard the vehicle – the coordinates of center of gravity have been defined,
- the towing-dynamic calculations have also been conducted - towing-dynamic characteristics have been thus calculated,
- axle load has been defined,
- parameters of the longitudinal static stability of a vehicle have been calculated,
- on the basis of static stability at the side camber the maximum angle of the side camber has been calculated and
- the analysis has been made for the tires capacity for each vehicle.

By this analysis a conclusion has been drawn that from the aspect of performances the most suitable vehicle for superstructure building is КАМАЗ 63501, 8x8.

The criteria for assessment of suitability for artillery systems superstructure

In order to obtain exact indicators on the vehicle suitability for artillery system superstructure in the research work [5] a detailed analysis has been made with the aim of identification of the parameters for the quantitative assessment of suitability from various aspects. In this regard the specific factors that ensure the afore-mentioned assessment have been introduced.

The criteria of suitability of an all-terrain vehicle with built-in superstructure can be defined in a variety of ways and a variety of viewpoints since the assessment criteria can also be different and depend on the purpose of a superstructure, that is, on the customer`s requirements.

The main criteria that will be applied to give the assessment of suitability for building superstructure should establish the capabilities of a vehicle to carry a load. The possibility of receiving a load depends on the resistance of the construction to bending and twisting of each concrete vehicle. This possibility depends on the geometrical characteristics of longitudinal and transversal frame supporters and on the character of their connection (riveting, total or partial welding, bolted joint etc) and the characteristics of the frame construction material. Since the geometrical characteristics of longitudinal supporters have the primary impact on the possibility of receiving a load, it has been established that the factor of the possibility of load receiving should equal the resistance moment of the vehicle frame. On the basis of calculations about the resistance moments the vehicle type ranging can be carried out and draw a conclusion on the vehicle suitability from the aspect of load receiving and the necessity of building an additional frame for receiving a load on the part of the superstructure. In the case study it was established that the vehicle frame from the family KAMAZ possesses two times larger resistance moment than the vehicle belonging to the family FAP, so that according to this criterion the advantage has been given to vehicle of the KAMAZ family.

The second criterion to assess the suitability of a vehicle for a superstructure building is the vehicle stability. The vehicle stability with mounted artillery weapons depends on the firing line elevation, the position of the system centre of gravity, overall system mass and the distance between the supporting points (with the vehicles these are wheels, if there are not any additional supporting legs or stands). From this it can be concluded that for the assigned firing line elevation and constant overall system mass, the vehicle stability at the moment of firing a projectile is subjected to both the centre of gravity position and the tires tracks. Thus it could be stated that the vehicle stability measurement represents the tires tracks and the elevation of the system`s centre of gravity. On the basis of such statement the factor of stability can be defined as:

The factor of stability at the moment of a projectile firing F_{st} can be defined as:

$$F_{st} = B \cdot h \tag{1}$$

where the following denotes:

- B – the width of the vehicle tracks,
- h – the height of the vehicle centre of gravity

By analyzing the value of the stability factor for different vehicles it has been concluded that there is not a significant difference, i.e. all analyzed vehicles have nearly the same suitability of being super-structured in terms of stability. It is the consequence of the law-regulated limitation of the maximum allowed vehicle width of up to 2.500 mm, which entails the tire tracks of almost 2000 mm, while the similar centre of gravity elevation with all vehicles is the result of the similar design. The absolute value of the stability factor with all vehicles was such that vehicles in this regard did not satisfy the stability requirements at the moment of projectile firing at all assigned angles. Thus it was defined that the meeting this requirement presupposes designing the additional supporters in the form of legs (stands) which should satisfy this requirement in all projected firing conditions.

The third criterion that can contribute to the assessment of the vehicle suitability for superstructure building is the possibility of maintenance. In the maintenance possibility analysis special attention has been paid to the superstructure impact on maintenance possibility of a basic vehicle. The maintenance possibility evaluation is a complex matter since it is difficult to define precise criteria for its presentation. Generally speaking, it is quite clear that the vehicles with the special superstructures have as a rule lesser maintenance possibility than the vehicles without superstructures.

Especially significant criterion for assessment of vehicles superstructure suitability is the additional load to vehicle structures as the consequence of the superstructure building. Within all vehicle structures, according to the previous experimental tests, it has been established that the largest loads occur on the vehicle frame, i.e. in the section where it joins the weapon junction. In order to more thoroughly evaluate the effect of successful mounting of weapon it is necessary to evaluate the additional load degree according to the adequate factors. In this case the additional load factor has been defined (F_{do}) which represents the ratio of weapon load in different conditions of exploitation. It has been considered in two characteristic cases:

- Additional load of the vehicle frame at projectile firing. Here the additional load factor is defined as the ratio of the vehicle frame strain at the moment of live projectile firing and the strain at the moment of the hydro shell firing. The hydro shell is used because the hydro shell tests can be conducted outside the test range (proving ground) which corresponds the firing range intended for concrete ballistic elements (since there is no projectile) and it is much cheaper because the projectile is not used. The recoil power effect with the hydro shell testing is nearly equivalent to the recoil power effect of the corresponding projectile. During these tests it has been concluded that the change of recoil effect in weapon with hydro shell and live shell is not equivalent although the recoil power effect in both cases is nearly equivalent.
- The additional vehicle frame load at the moment of projectile firing and in marching position. The additional load factor is here defined as the ratio of frame strain in case when the superstructure performs its purposeful function (in our case it is at the moment of projectile firing) and the strain in the case when the superstructure does not perform its function, i.e. in the course of movement (in concrete case it is in the marching position, when there is no projectile firing).

The addition load factor in the first specific case can be analytically shown in the following way:

$$F_{do} = \frac{\Phi_b}{\Phi_h} \qquad (2)$$

where

- Φ_b – the vehicle frame strain in the moment of live projectile firing, at the observation place [$\frac{N}{mm^2}$],
- Φ_h – the vehicle frame strain in the moment of hydro shell firing, at the observation place [$\frac{N}{mm^2}$],

The additional load factor in the second specific case can be shown analytically in the following way:

$$F_{do} = \frac{\Phi_b}{\Phi_m} \tag{3}$$

where - Φ_m – the vehicle frame strain in the marching position, in the observation place [$\frac{N}{mm^2}$].

The analysis of the additional load factors shown in (2) and (3) points to the fact that the most influential parameters are the frame surface and the thickness of the frame wall. It is further noticed that the theoretical evaluation of the defined factors is significantly subjected to the hindering circumstances because of unknown influences of the simulation models on their value, and in the experimental part of the work, on the basis of measurement results, the concrete values of load factors have been established and they range from 1,36 to 2,38.

Figure 4: *The artillery system integrated onto the mobile platform*

By defining the factors of additional load explicitly the increase of load in the observation place for two specific conditions of performance has been shown and it has also enabled insight into the superstructure successfulness of this vehicle type. On the basis of defined criteria and methodology defined in [5] the vehicle selection was made with presentation of the prototype of the integrated combat platform shown in figure 4.

CONCLUSIONS

The integration of combat system onto the mobile platform in the form of a motor vehicle ensures better performance quality of the very system because the combat system in this case realize additional performances mostly in terms of its mobility. When evaluating the suitability of a vehicle for superstructure building it is necessary to pay attention to a large number of affecting parameters which should satisfy the criteria that lead to the evaluation of the system successfulness. Considering that the combat system being mounted does not itself perform the function, but that the term combat system indicates the integral construction consisting of a vehicle and a combat system it is obvious that the quality of this complex system will depend not only on the sub-system quality (a vehicle and superstructure) but also on the quality of realizing their connectivity. By introducing the criterion for assessment of vehicle suitability with built-in combat system superstructure, and also the factors which quantify individual criteria the systemic approach of analyzing this task is ensured and also conditions were obtained for the results to be satisfying, saving time and resources for construction and evaluation of the physical prototypes.

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[1] CRASH ANALYSIS OF COMPOSITE SACRIFICIAL STRUCTURE FOR RACING CAR

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UDC: 678-19:796.71]:614.8

Abstract

Driver safety in case of accident is a main problem when developing a new vehicle design, and this is of particular importance when dealing with racing car. This is due to the higher velocity that generally characterizes the operation of a racing car with respect to a urban use vehicle. To reduce the development and testing costs of a new safety design, it is recommendable to use computational crash simulations for early evaluation of safety behaviour under vehicle impact test. The numerical simulation of the crash behaviour by using finite element methods and the lightweight design of the sacrificial structure made in composite is presented in this paper. It was developed for an auto-cross racing car, with a small formula style which was later manufactured, tested and demonstrated on the race competition. Final results show the comparison between the impact attenuator device in composite and the same one in aluminium, both connected with body structure. The main idea of the research was to demonstrate energy absorbing capabilities of a thin-walled crash box during the frontal impact, with the lowest initial deceleration. In order to initialise the collapse in a stable way, the design of the composite sacrificial structure has been completed with a trigger which is consisted of a very simple smoothing (progressive reduction) of the wall thickness. Initial requirements were set in accordance with the 2008 Formula SAE rules and they were satisfied with the final configuration.

Key words: sacrificial structure, lightweight design, composites.

ANALIZA SUDARA KOMPOZITNE ABSORBUJUĆE STRUKTURE SPORTSKOG TRKAČKOG AUTOMOBILA

UDC: 678-19:796.71]:614.8

Rezime: Bezbednost vozača u slučaju saobraćajne nesreće predstavlja jedan od glavnih aspekata koji se uzima u obzir prilikom razvoja novog koncepta vozila. Naročito je od velike važnosti pri projektovanju sportskih trkačkih automobila koji dostižu znatno veće vrednosti brzina u odnosu na putnička vozila. U cilju smanjenja troškova kao i vremena potrebnog za razvoj i testiranje savremene strukture automobila, preporučljivo je koristiti kompjuterske simulacije zbog rane procene zadovoljenja bezbedonosnih zahteva vozila prilikom sudara. U ovom radu su predstavljene numeričke simulacije udara korišćenjem metode konačnih elemenata, kao i projektovanje lakog kompozitnog absorbera energije. Ova struktura je razvijena za potrebe sportskog vozila koje je stilski dizajnirano u obliku

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formule, a u sledećoj fazi proizvedeno, testirano i korišćeno u auto trkama. Završni rezultati prikazuju poređenje absorbera energije napravljenog od kompozitnih materijala i aluminijuma, povezanih sa školjkom automobila. Glavna ideja ovog istraživanja je da se prikažu sposobnosti absorpcije energije tankozidnih struktura prilikom frontalnog sudara, uz dobijanje što manje vrednosti usporenja. U cilju inicijalizacije kolapsa, sa aspekta stabilnosti projektovanje kompozitnog absorbera je obavljeno korišćenjem triger mehanizma, koji podrazumeva progresivnu redukciju debljine. Početni projektantski zahtevi postavljeni na osnovu 2008 Formula SAE pravila u potpunosti su zadovoljeni finalnom, predloženom konfiguracijom.

Ključne reči: absorber energije, projektovanje lakih struktura, kompozitni materijali.

CRASH ANALYSIS OF COMPOSITE SACRIFICIAL STRUCTURE FOR RACING CAR

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UDC: 678-19:796.71]:614.8

INTRODUCTION

Carbon fibre composites have shown to be able to perform extremely well in the case of a crash, and are being used to manufacture dedicated energy-absorbing components, both in the motor sport world and in the construction of aerospace structures. In case of impact, the kinetic energy of the vehicle should be dissipated in a very fast but stable, regular and controlled way so that passengers can survive. In metallic structures, this energy absorption is achieved by plastic deformation, while in composite structures it relies on the material diffuse fracture: i.e. on a completely different mechanism. Excellent performance of composites, sometimes better than those of the metallic similar structures, can be obtained by choosing appropriately the mechanical (the stacking sequence, the number of layers, the type and quantity of fibres and matrix) and geometrical (the beam section shape, the wall thickness, the extremity joints) design parameters [2].

This paper is presenting the design and numerical simulation of impact event for the frontal safety structure of car body for the formula SAE vehicle developed by the Polytechnic of Turin team. On figure 1 we can see the final solution of the racing car, which was produced in the collaboration with partner enterprises in Turin and successfully demonstrated in several racing competitions in Europe. The production strategy of this car consists in a concurrent analytical and experimental development, from the initial conceptual design and coupon testing, through the stages of element and subcomponent engineering, to final component manufacturing.

Figure 1: *The racing car of Polytechnic of Turin*

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After initial analysis of the response of the designed impact attenuator in aluminium, which is presented in previous activities [3], it is developed the same sacrificial structure but in composite. Since the composite is relatively new material, requires constant evaluation in automotive applications to ensure equal if not greater performance than their metallic counterpart. The main scope of this work is the comparison of crashworthiness response of composite and aluminium impact attenuators and development of optimal solution that can be implemented in final production.

The performed numerical simulations put in evidence the needs for the design of a good energy absorbing structure and also the requirements for a good design of the links between attenuator and Body in White. The idea is to develop an efficient solution in terms of absorbed energy versus weight ratio, which leads to lightweight approach.

Complete geometry was done by means of the software for 3D modelling Catia. The finite element model of the structure was developed by means of the software code Hyper Mesh. Finally the crash event simulation has been developed by means of the Radioss code.

The main aspects of the research were oriented toward the dissipation of the kinetic energy during the front impact that should be as progressive as possible and toward the evaluation of the initial deceleration which has to be as little as possible. It is well known that the optimal solution in absorbing energy for this type of application is to obtain a nearly flat diagram of the impact force, that means a nearly constant value of the deceleration [6]. All initial requirements, which gave good results at the end of simulation, were done regarding the 2008 Formula SAE rules. These constraints are mainly oriented to dimensions of the attenuator, mounting of the parts, material selection, construction of attachments and final results requirements.

GEOMETRY OF THE MODEL

The Society of Automotive Engineers has introduced a series of regulations to ensure that race cars conform to stringent safety requirements and build quality, in order to be granted race-worthiness certification. These criteria include a series of static loads applied to the chassis, which guarantee the strength and integrity of the survival cell and a series of requirements on the location and impact characteristics of the energy-absorbing device [8]. Each year the number and severity of these requirements increases in line with ongoing research and development in crashworthiness or in response to real life accidents.

In order to meet the requirements of Formula SAE 2008 competition, the attenuator must guarantee specific performances in terms of average deceleration values and minimum acceptable dimensions during impact. Moreover the assembly of the sacrificial structure is subjected to the following conditions according to reference [21]:

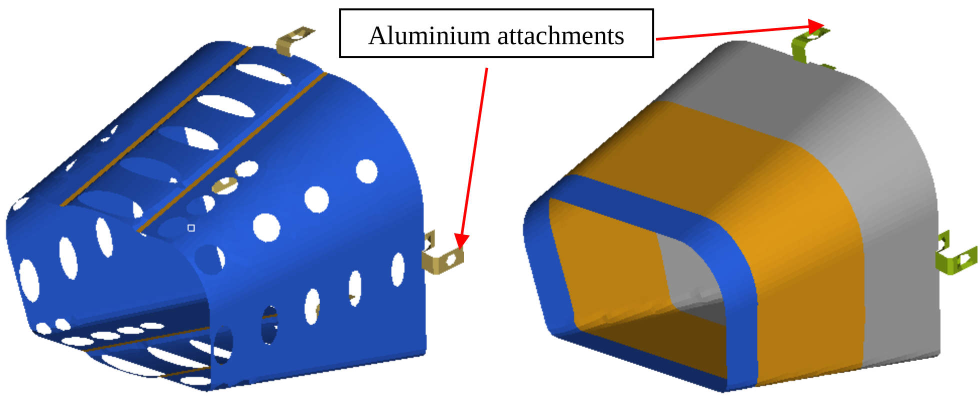
- The impact attenuator must be installed in front of the bulkhead;
- It must be at least 200 mm long (along the main axis of the frame), 200 mm wide and 100 mm high;
- It must not penetrate the front bulkhead in case of impact;
- It must be attached to the front bulkhead by welding, or at least, 4 bolts (M8, grade 8,8);

- It must guarantee safety in case of off-axis and off-centre impact;

The analysed energy absorber (figure 2) consists of a truncated pyramidal structure with an almost rectangular section. The pyramidal structure allows to obtain a major stability during the progressive crushing, while the rectangular section has rounded edges to avoid stress concentrations.

In the picture of figure 2(a) which is showing the aluminium impact attenuator structure, some holes are well visible in all the four side walls of the impact attenuator structure. They are operating as triggers, that means have both the scope to decrease the weight of the structure itself and to obtain a better crush behaviour by decreasing the first peak of the collapse force. By putting holes at appropriate positions, especially on fillet surfaces, it is obtained the sacrificial structure in aluminium with good impact performance and in particular, smaller initial deceleration pick during the crash test.

In the picture of figure 2(b) is shown the absorbing device in composite. In comparison with the same one in aluminium, it doesn’t have holes and properly shaped aluminium plates. The design of sacrificial structure has been completed with a trigger which consists in a smoothing (progressive reduction) of the wall thickness in order to reduce locally the resisting section. This trigger is intended both to reduce the value of the force peak and to initialise the structure collapse in a stable way. Three different wall thickness are introduced and shown with different colours on figure 2 (b). Also, the carbon fibre component is offering great deal of weight saving with respect to the aluminium structure. Comparing the masses, the composite structure is 40 % lighter.



(a) Mass=0,544kg (b) Mass=0,307kg

Figure 2: *Impact attenuator structure: (a) in aluminium; (b) in composite*

One of the important SAE constraints is about the construction solution of attachments between impact attenuator and bulkhead. After several consultations with judges of SAE 2008 from Germany, the final solution for the design of links was chosen, which gave good results from the point of view both of practical implementation and, finally, of good impact behaviour, i.e. desired amount of absorbed energy and suitable level of deceleration.

These final links are C shaped. These links are independent parts of the system. They require bolts for the connection between car body and impact attenuator. In the images of the figure 3, we can see the front impact attenuator structure and the links assembled to that structure.

In the picture of figure 3, we can see complete structure; it consists of three different parts: car body frame, impact attenuator and links.

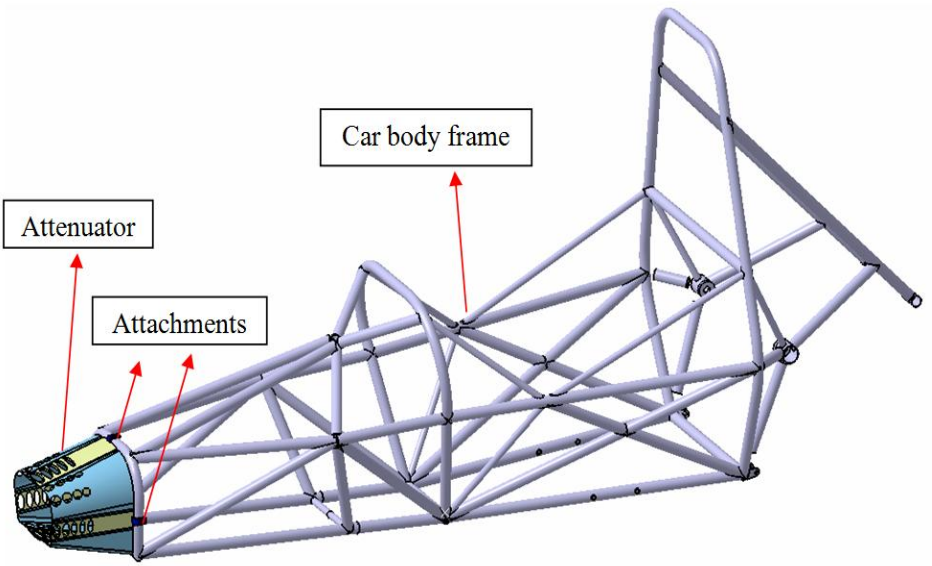


Figure 3: *Complete race car structure*

FINITE ELEMENT MODEL

Complete simulation was done using finite element model developed in Hyper Mesh software. It is very important to notice that it was not used all car body structure, but only the frame that is the most important part from the point of view of the vehicle front impact and driver safety. Unnecessary parts were cut off before making the mesh of the finite element model.

Three different types of materials are used: steel S275JR UNI EN 10025 (Fe430), 6082T6 aluminium alloy (this type of the alloy is anticorrosion material good for energy absorption but quite hard for bending) and composite T300/5208 carbon-epoxy.

Aluminium impact attenuator and links were made of previously mentioned aluminium alloy and the car body frame was made of steel. Also, this type of attenuator includes two beams which have different properties in comparison with the rest of attenuator. The properties of parts are: impact attenuator shell thickness is 1.5mm, impact attenuator beam thickness is 3mm, attachment thickness is 3mm and tube thickness is 2.5mm. Parts of the aluminium sacrificial structure with attachments are presented on figure 4.

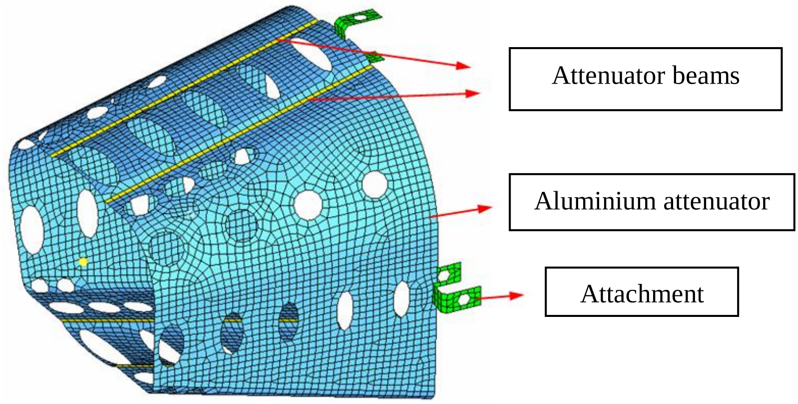


Figure 4: *Parts of the aluminium impact attenuator*

For this simulation materials were modelled as isotropic with elastic – plastic characteristics. The description of the material characteristic will be made by means of the usual Ramberg–Osgood plastic law together with the Johnson – Cook factor to account for the strain rate effect (Law 2 in RADIOSS). Adopted values of modelling parameter can be seen in table 1.

Table 1: *Material properties used for aluminium attenuator simulation*

Property	**Aluminium alloy 6082T6**	**Steel S275JR UNI EN 10025 (Fe430)**
Density	2.7 * 10-3 g/mm3	7.8 * 10-3 g/mm3
Poisson's ratio	0.33	0.3
Young's modulus	70000 MPa	206000 MPa
Yield strength	428.5 MPa	275 MPa
Hardening parameter	327 MPa	591.4 MPa
Hardening exponent	1	0.7108
Strain rate coefficient	0.00747	0.03
Ultimate strength		430 MPa

The finite element model of the composite absorbing structure is shown on figure 5. As previously mentioned, the triggering mechanism is introduced in terms of progressive reduction of wall thickness. In particular, there are three different wall thickness zones that are presented on next figure by green, red and blue colour:

- Green zone - laminate thickness = 0,8 mm simulated with 4 plies
- Red zone - laminate thickness = 1,3 mm simulated with 6 plies
- Blue zone - laminate thickness = 1,4 mm simulated with 7 plies

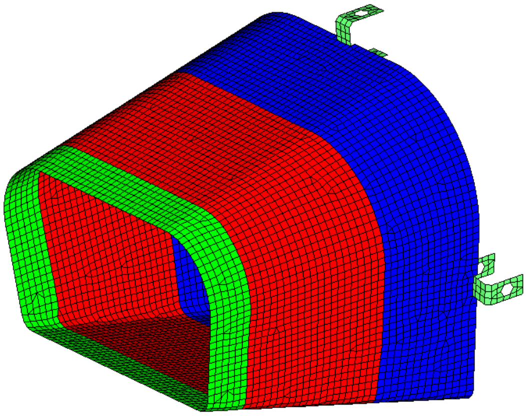


Figure 5: *Composite impact attenuator*

The composite T300/2508 is modelled with stacking sequence [0°,90°] and unidirectional fibre. The element property type used for modelling composites in solver Radioss v44 is Type10-Composite Shell. It allows the possibility of modelling up to 100 layers with constant thickness, variable layer orientation, constant material properties and constant reference vector. Integration is performed with constant stress distribution for each layer. The used material law is COMPSH (25) with orthotropic elasticity, two plasticity models and brittle tensile failure. The plasticity model is based on the Tsai-Wu criterion which allows to model the yield and failure phases [17]. In table 2 are presented material characteristics, necessary for composite modelling.

Table 2: *Material properties used for composite attenuator simulation*

Property	**Composite T300/5208 carbon-epoxy**
Density	1.6 * 10-3 g/mm3
Young modulus in fibre longitudinal direction	181 GPa
Young modulus in transverse direction	10.3 GPa
Poisson’s ratio	0.280
Shear modulus $_{12}$	7.170 GPa
Shear modulus $_{23}$	4.020 GPa
Shear modulus $_{31}$	7.170 GPa
Longitudinal tensile strength	1.500 GPa
Transverse tensile strength	0.040 GPa
Longitudinal compressive strength	1.500 GPa
Transverse compressive strength	0.246 GPa
Compressive strength in direction 12	0.246 GPa
In plane shear strength	0.040 GPa

A rigid wall with a mass of 300 kg is added, it models the obstacle impacted during the test. The structure is moving with initial velocity of 7m/s. There is friction between the rigid wall and attenuator. The friction coefficient has the value of 0,2 between attenuator components and 0,4 between rigid wall and attenuator.

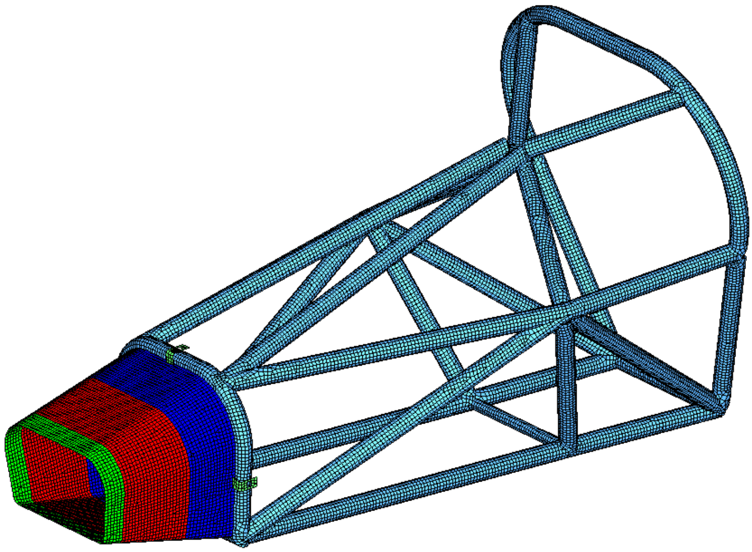


Figure 6: *Finite element model of the structure with composite attenuator*

When analysing the impact attenuator structure linked to the body frame, not all the body frame was modelled. Constraints were added in substitution of the rear part of car body, that was cut off, because it is not relevant for the present analysis.

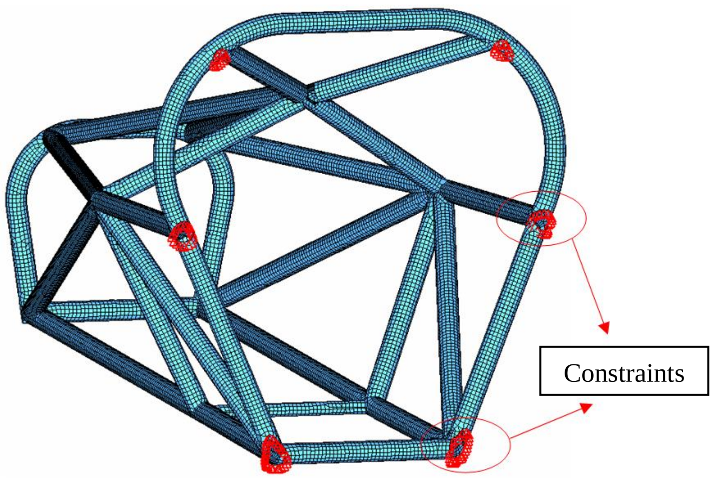


Figure 7**:** *Finite element model with specific view on constraints*

In finite element model, there are several contacts between parts which do not allow for penetration. In pictures of figures 6 and 7, we can see the representation of finite element model with mesh and constraints.

The complete model, which consists of composite impact attenuator, linkages and front part of the car body, has 215590 elements and 38194 nodes. We used 3 node and 4 node element types.

RESULTS

Most of the relevant results obtained by the performed analysis will be illustrated and discussed in the following paragraphs.

First of all we will see the results obtained in the simulation of the structure with impact attenuator in aluminium in comparison with the results obtained in the simulation of the solution of the composite impact attenuator linked to the frame structure. Results have been analysed by means of the software code Hyper View. The duration of the simulation is 60ms. In figures 8 and 9 we can see the crushed shape of the two different models in two subsequent positions during the impact: in figure 8 we are about at 14ms of the crash event while in figure 9 we are at the end of the crash. On the left side we see the results obtained when considering the impact attenuator structure made in aluminium while on the right side we see the results obtained with the impact attenuator made in composite. Both structures have almost the same behaviour and absorption of the energy in the first half of the simulation, which is presented on figure 8. But, at the end of the simulation, we can see on figure 9 that the aluminium impact attenuator is more deformed, in particular in the lateral walls. The difference in displacement and deformation is almost 50% with respect to the composite attenuator at the end of the simulation. This means that the composite structure is showing better crash behaviour.

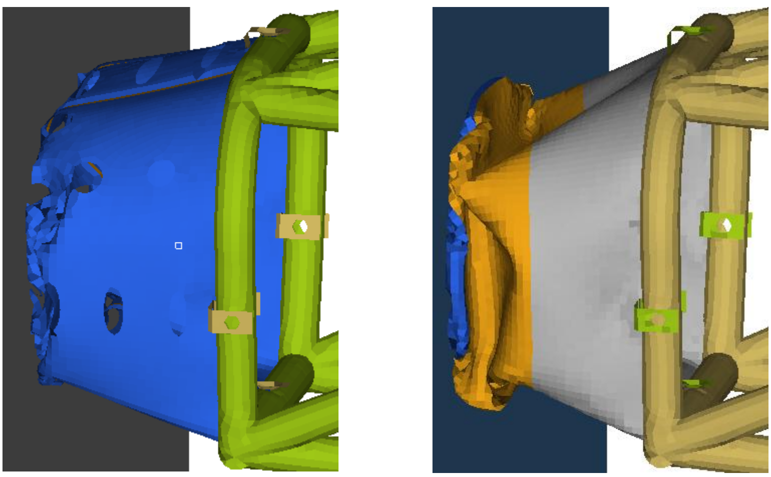


Figure 8: *The position of two structures during the impact at 14ms of crash event*

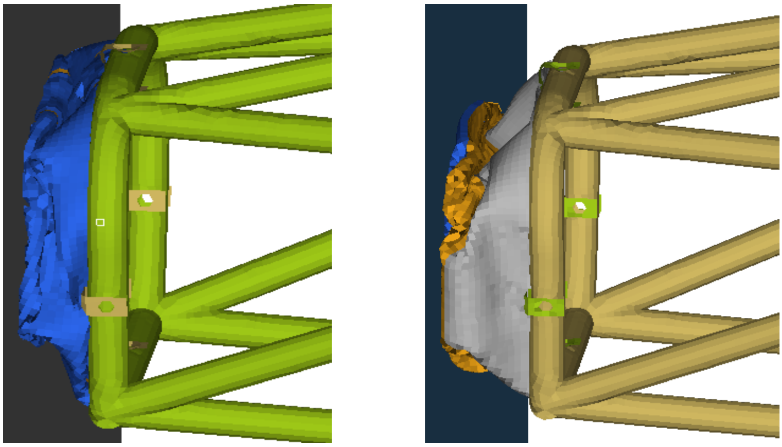


Figure 9: *The final position of two structures during the impact*

Figure 10 shows the diagram of the trend of kinetic energy in time, the responses of two different models are reported: the car body with impact attenuator in aluminium and the same one in composite. It is worth of note that all curves are starting from the initial energy value of 7350 J. The behaviour of the composite attenuator structure during the crash impact is better with respect to the aluminium one almost from the begin of crash evolution. In particular, the kinetic energy is absorbed faster by the composite structure than by the aluminium one. The zero value of the kinetic energy is reached after 38 ms of the simulation in the case of composite device and 47 ms when we have the aluminium attenuator. This means that the composite structure has a more progressive collapse.

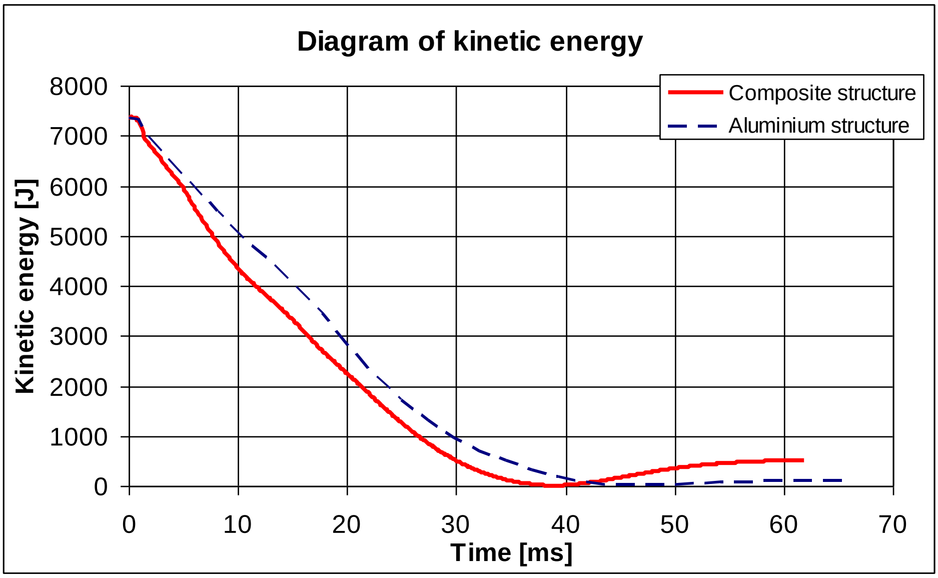


Figure 10: *Comparison of kinetic energy of the models with aluminium and composite sacrificial structure*

On figure 11 we can see the diagram of the history of the velocity in x direction for both cases. The before mentioned results are confirmed in the same manner.

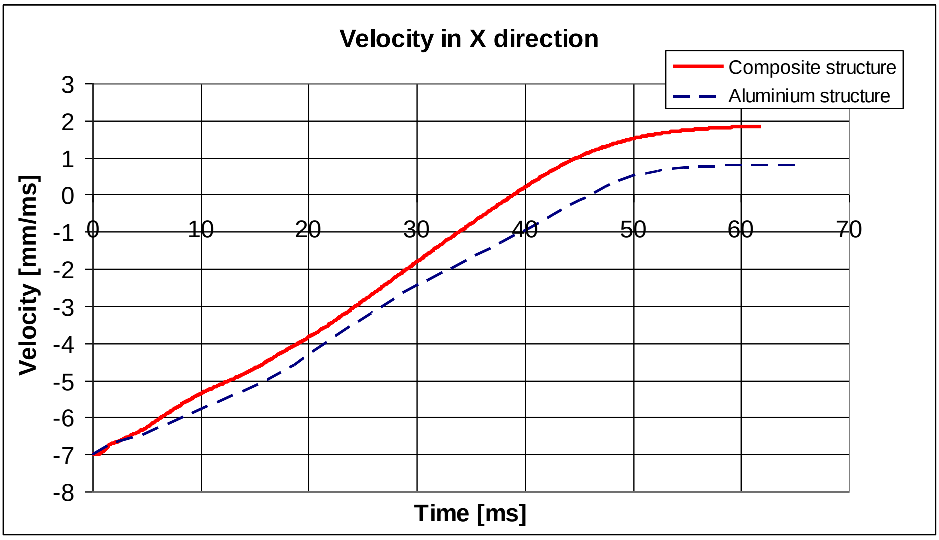


Figure 11: Numerical results of the velocity of the aluminium and composite impact attenuator

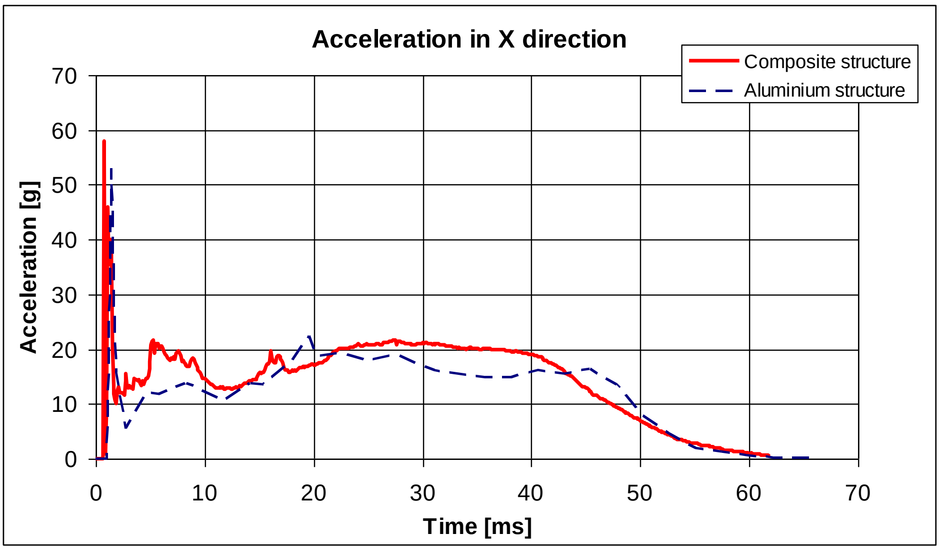


Figure 12: Comparison of decelerations for two attenuator models: in aluminium and in composite

On figure 12 we can see the diagram of the evolution of deceleration during the time for both solutions. The first peak is due to the initialisation of the structure collapse and, for aluminium sacrificial structure, has the value of 0,52 mm/ms^2, which is 52g. This value in the case of composite attenuator is almost the same, and it became 58 g. Between 6 and 25 ms of the simulation process, the behaviour of these two structures is almost the same. On figure 13 the force displacement diagram is presented. The force is directly dependent on acceleration, and that's why the behaviour is almost the same. The first maximum peak of the aluminium structure is about 145 kN and for composite structure 155 kN.

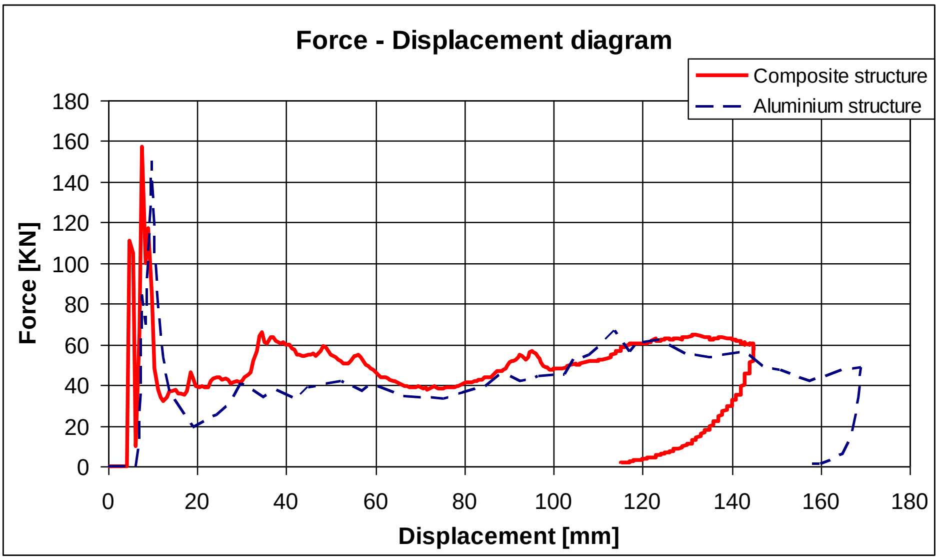


Figure 13: *Force – displacement diagram for aluminium and composite attenuators*

In both diagrams of figures 12 and 13 it is well visible that the composite impact attenuator response is quite good being close to the optimal absorber one from the aspect of initial pick of the force and acceleration. After initial peaks, we obtain a nearly flat diagram of the impact force, that means a nearly constant value of the deceleration. Also, it is recommended in SAE rules that "average deceleration of the vehicle must not exceed 20 g". On figure 12 it is well visible that the average acceleration of the assembly that consists of car body and aluminium impact attenuator is about 14 g and in the case of composite attenuator is higher but not exceeding 20 g.

CONCLUSIONS

The paper has developed a careful analysis of the crash behaviour of the impact attenuator structure that has been designed to equip the formula SAE car of the Politecnico di Torino racing team. The simulations performed by the finite element methods permit to point out some very important facts, that are giving strong arguments about the quality of results got in this work.

First of all the design of the energy absorbing structure has to allow a progressive evolution of the phenomenon avoiding as much as possible the presence of force peaks (that means deceleration peaks). The ideal solution is a flat diagram of the resisting force in time.

As a second capital point, achieved results put in evidence that the adoption of composite materials not only leads to lighter structure but also gives the possibility to design structures that have crash performance even better than aluminium structures. The composite impact attenuator is lighter 40% respect the same one in aluminium.

From the analysis of the diagrams, we can conclude that the collapse evolution of both suggested attenuators is progressive and stable. It is also possible to note the presence of a quite high first peak of the load that is due to the initialisation of the structure collapse. In any case the value of the maximum deceleration (that is in correspondence to the first peak) in the case of aluminium attenuator structure is about 0.52 m/ms^2 (52g). In the case of composite structure the maximum deceleration is about 58g, that means a very small difference with respect to previously mentioned and reliable solution. Also, the average deceleration of the aluminium structure is 14g and in the case of composite solution also less than 20g. This values fits well with the requirements of SAE 2008 rules, as they require that the adopted solution leads to an average deceleration lower than 0.2 m/ms^2 (20g). It means that crush initialisation triggers improved significantly the structural response, that is very important for composite sacrificial structure. Also, the absorption of the kinetic energy of the composite attenuator showed stable behaviour and better response respect to the aluminium solution.

All these results confirm that the composite attenuator structure is very good from the point of view of frontal impact and it is possible to absorb energy on good way.

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[1] IDENTIFICATION OF THE MOTORCYCLE STEERING PROPETRIES

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UDC: 629.118.6-5: 656.13

Abstract

The Motorcycle as a two – wheeled vehicle in interaction to rider behaviour essential influences on the traffic safety. With this aspects this paper aimed to development a combined theoretical – experimental approach to investigation rider – motorcycle system. The basic model necessary to development methodology, design experimental system, planing and carrying out experimental tests, then to data processing and results interpretation was selected and adapted from our previous study. Formed experimental system to motorcycle testing consists from two measuring subsystem. The first, supported on the electronic measuring system Spider – 8 HBM, with seven measured variables transducers and two alternative transducers. The second, supported on the compact autonomous vibration analyzer measuring instrumentation with two transducers – accelerometers and six measured directions. Experimental investigation are conducted in the curves of real road condition. Processed data are analysed with respect to longitudinal velocity and its influences to traffic safety quantities. The some news investigation possibilities which allows development system and proposed methodology are pointed out.

Key words: motorcycle, experiment, identification, steerig properties.

IDENTIFIKACIJA UPRAVLJIVOSTI MOTOCIKLA

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Rezime: Motocikl kao vozilo sa dva točka, u interakciji sa vozačem, bitno utiče na bezbednost saobraćaja. Sa ovog aspekta ovaj rad ima za cilj razvoj kombinovanog teorijsko – eksperimentalnog pristupa za istraživanje sistema vozač – motocikl. Bazni model neophodan za razvoj metodologije, projektovanje eksperimentalnog sistema, planiranje i izvođenje eksperimenata, zatim za obradu podataka i interpretaciju rezultata, izabran je i prilagođen iz naših prethodnih istraživanja. Formirani eksperimentalni sistem za ispitivanje motocikla sastoji se iz dva merna podsistema. Prvi je baziran na elektronskom mernom sistemu Spider -8 HBM, sa sedam davača mernih veličina i još dva alternativna davača. Drugi je baziran na kompaktnom samostalnom analizatoru vibracija sa dva davača ubrzanja i šest mernih pravaca. Eksperimentalna istraživanja su sprovedena u krivinama u uslovima realnog puta. Obrađeni podaci su analizirani sa aspekta uticaja podužne brzine na pokazatelje bezbednosti saobraćaja. Istaknute su neke nove mogućnosti istraživanja koje nudi razvijen sistem i predložena metodologija.

Ključne reči: motocikl, eksperiment, identifikacija, upravljivost.

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IDENTIFICATION OF THE MOTORCYCLE STEERING PROPERTIES

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UDC: 629.118.6-5: 656.13

INTRODUCTION

The dynamical characteristics of motorcycles in interaction with rider behaviour have the essential influence on the traffic safety. On the other hand, motorcycle as a two-wheeled vehicle have much complexity and distinction from those of a four – wheeled vehicle. Namely, dynamics of a motorcycle is very complicated themselves and difficult to study. Besides, the rider used control strategies, as well his motion relative to motorcycle, affect to system stability and controllability.

Motorcycle handling properties have been intensive studied during later 50.years. The subject of these studies were formulation a suitable approach to modelling and mathematical description of motorcycle dynamics as system with multi rigid bodies [1]. The influence of the motorcycle subsystem rigidity to its lateral dynamics is aim research in paper [2]. The results of the motorcycle oscilatory behaviour on the straight and curved path presented in paper [3], shown four typical modes as named, capsize, roll, weave, wobble, which influence to properties of lateral dynamics. The rider control actions have also been studied by several authors [4], [5],[6] , and many importand facts were found. But actual riders control of the motorcycles is very complicated and influenced to traffic safety so this researching task has today priority. In the literature there are many papers on the steering properties of the motorcycles but only a few deal with experimental research [7].

Based on the above mentioned problem this paper aimed to development a combined theoretical–experimental method to investigation rider–motorcycle system by driving manoeuvres relevant on the traffic safety. Basic model, experimental system and proposed procedure are described and verified in next chapters.

THEORETICAL CONSIDERATION

The partial models of rider – motorcycle system used in this paper are based on the our previous study [8]. The structure and model parameters was selected and adapted according to requirements for development of methodology, design experimental system, planing and carrying out of experimental tests, also, for combining theoretical – experimental approach to data processing and results interpretation.

Two characteristic models, derived from [8], upon that extended and adapted for use in this paper are presented in Fig. 1 and 2. Fig. 1, shows the partial model of rider–motorcycle behaviour in transversal plan during curved motion. The total mass is divided into two

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parts, m_1 – sum of motorcycle mass and rider lower body mass at height h_1 from the ground, m_2 – mass of rider upper body at height h from the ground, as well as at distance h_2 from the hip pivot point S. The roll angle of system is denoted with ψ. From this direction rider can lean on the right or on the left for angle ±γ about longitudinal axis crossing the point S, parallel to system lean axis, x.

The model configuration in Fig. 1, depends from the rider used strategy. The upright position, as well leaning system from centre curve, in figure denoted as *out*, converges to unstable system motion. But, leaning toward the centre curve, denoted as *in*, balances the gravitaionl F_{gi} and centrifugal F_{ci}, i=1, 2, forces and stabilizes the system motion. Also, the every change of system position, defined by angle ψ, in transvesal plane, disturbes dynamical balance. On the other hand, the change of rider lean angle, γ, taken as his control input, according to consideration given in study [8], essential influentes to driving safety.

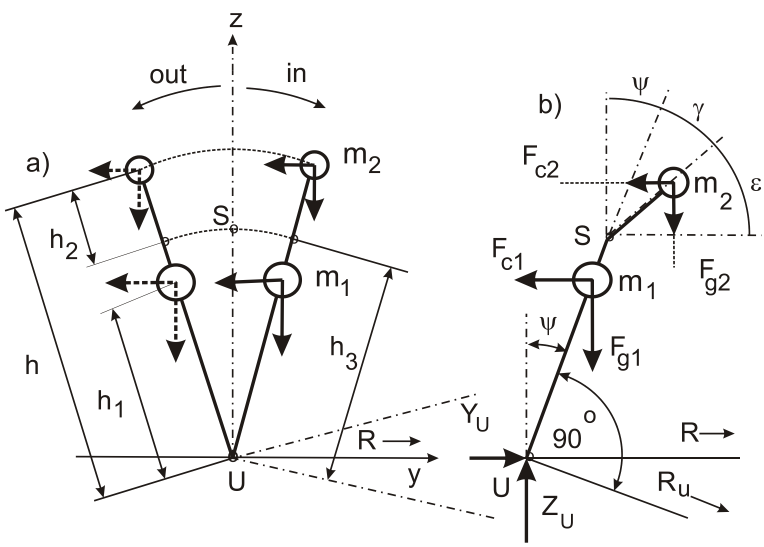

Figure 1: *Model of rider–motorcycle roll motion and rider leaning*

According to presentation and notation in Fig. 1, can be written the condition of dynamical balance for considered system in general form:

$$\Sigma Z_i = 0, \quad \Sigma Y_i = 0, \quad \Sigma M_{ui} = 0 \qquad (1)$$

Based on the equations (1) two characteristic cases are further derived:

(1) steady state motion without rider leaning, γ = 0,

$$Z_u = F_{g1} + F_{g2} = (m_1 + m_2)g = mg \qquad (2)$$

$$Y_u = F_{c1} + F_{c2} = (m_1 + m_2)v^2 / R = mv^2 / R \qquad (3)$$

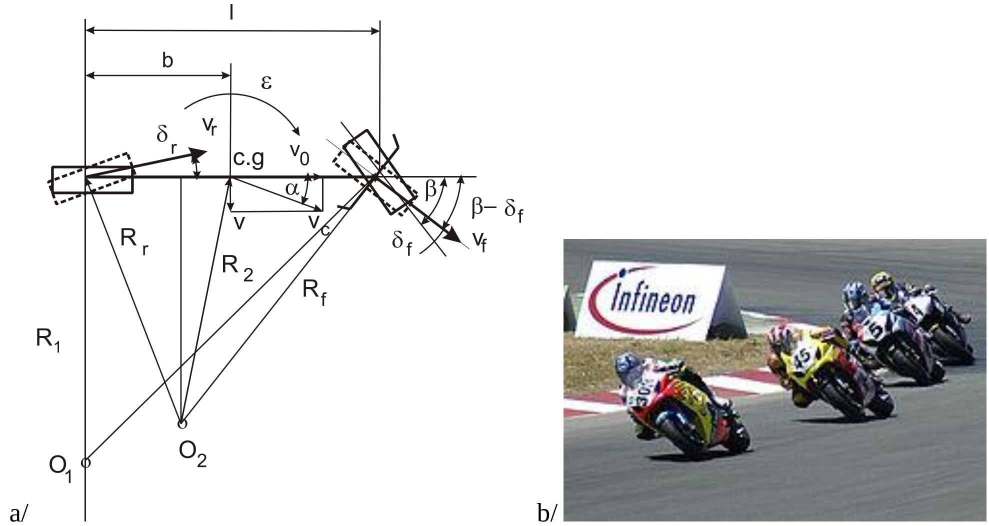

Figure 2: *a/ model of rider–motorcycle lateral motion, b/ riding strategies*

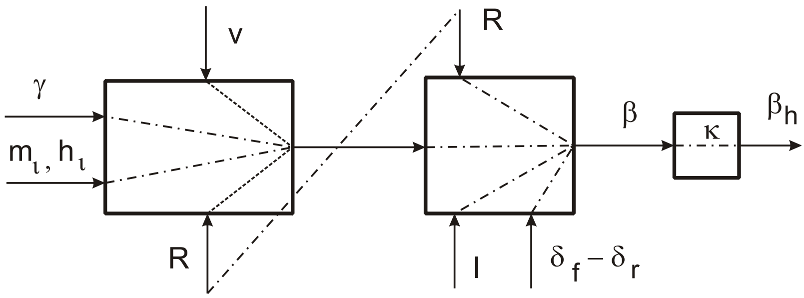

Figure 3: *Identification model of rider–motorcycle system*

$$\Sigma M_{ui} = F_{c1} h_1 \cos\psi + F_{c2} h \cos\psi - F_{g1} h_1 \sin\psi - F_{g2} h \sin\psi = 0 \tag{4}$$

From equations (2) and (3) follows,

$$Y_{u\varphi} = \varphi mg \geq Y_u = mv^2 / R \quad \rightarrow \quad v_{sl} = \sqrt{\varphi g R} \tag{3a}$$

where v_{sl} – maximum motorcycle stable velocity with respect to lateral sliding, φ - coefficient of tire/road lateral friction.

From equation (4) follows,

$$[(m_1 h_1 + m_2 h) v^2 / R] \cos\psi \leq (m_1 h_1 + m_2 h) g \sin\psi \rightarrow v_{sz} = \sqrt{Rg \tan\psi} \tag{4a}$$

where v_{sz} – maximum motorcycle stable velocity with respect to lateral capsize, ψ – roll angle.

The system rider – motorcycle roll angle for above considered case is a function of velocity v, and curve radius, R:

$$\psi_1 = \arctan(v^2 / Rg) = f_1(v, R) \tag{5}$$

(2) steady state motion with rider leaning, γ≠0,

According to above given condition in equation (1), $\Sigma M_{ui} = 0$, and presentation in Fig. 1b, can be written equation,

$$\begin{aligned}&(m_1h_1 + m_2h_3)g\sin\psi + (m_2h_2)g\sin(\psi+\gamma) = \\ &= (m_1h_1 + m_2h_3)(v^2/R)\cos\psi + (m_2h_2)(v^2/R)\cos(\psi+\gamma)\end{aligned} \tag{6}$$

Thus, rider leaning includes additional influenced factors in equation (5). Namely, now the roll angle is a complex function of system parameters m_i, h_i, motorcycle velocity v, curve radius R, and rider lean angle γ, as follows,

$$\psi_2 = f_2(m_i, h_i, v, R, \gamma) \tag{7}$$

for value of γ = 0, equations (7) turns into equation (5). This is a check of correctness of derived equation (6).

The partial model of rider – motorcycle system lateral motion in Fig. 2, was used in this paper to estimation of the basic relation between the motorcycle steer wheel motion parameters, as angle, torque, and parameters motion in lateral direction, as lateral acceleration, velocity, as well as yaw rate and so on. Based on the model in Fig. 2 and Fig. 1b, may be presented following alternative relation between steer fork motion and vehicle position coordinates:

- front wheel steer angle versus rear wheel path radius, without roll motion and sideslip, as follows, $\beta = \beta (l, R_1)$,

$$tg\beta = l / R_1 \tag{8}$$

- front wheel steer angle versus rear wheel path radius, with roll motion, without sideslip, as follows, $\beta = \beta (l, R_r, \psi)$,

$$tg\beta = l / R_{u1} \quad \to R_{u1} = R_1 / \cos\psi \tag{9}$$

where ψ – roll angle is given by equation (5).

- front wheel steer angle versus motorcycle path radius, with roll motion and sideslip angles, as follows, $\beta = \beta$ (l, R, v, ψ, δ_1, δ_2),

$$\beta = \beta[l, R_2, \psi_i = f_i(...), \delta_f, \delta_r] \tag{10}$$

where δ_f, δ_r – cornering angles of front and rear motorcycle wheel, respectively, $\psi_i = f_i$ (...) – roll angle function one of two relation , (5) or (7), dependently from considered influenced factors.

Based on the Fig. 1 and 2, as well as equation from (1) to (10), an identification model for motorcycle steering properties, as a block diagram, is given in Fig. 3., where χ – front fork angle in longitudinal plan, β_h – handlebars steer angle. The others signs given in Fig. 1, 2.

EXPERIMENTAL SYSTEM

Experimental system in this research includes, rider – motorcycle – measuring device for testing in real driving condition. Design of experiment and measurement devices development to investigation of motorcycle steering properties has primarly goal in this paper. An optional experimental system concept is propsed. This means, that dependently from actual research tasks, available transducers and measuring equipment, its mass and size, required accuracy, available space in vehicle, and so on, different measuring configuration can be formed.

The block diagram of measuring equipment on the motorcycle tested is shown in Fig. 4., and characteristic segments in Fig. 5 to Fig. 7. The signs in Fig. 4, mean: 1 – D – steering torque dynamometer, 2 – contactless steering angle transducer, 1a, 2a – transducers alternative, for previous given 1 and 2, seted on the motocycle steering damper, further, 3, 4, 5, 6, 7 – an inductive accelerometer completion to measurement of motorcycle yawing, rolling and rider leaning acceleration or alternative option 8 – with two triaxial accelerometers. Complete measuring system includes two subsystem: first, based on the support of S – PC – Spider 8 – PC measurement electronics, second, based on the VA – Vibration analyzer, compact autonomous measurement instrumentation, [9].

Fig. 5a shows the view of two types of displacement transducers used to steer angle measurement, ’’Novotechnik ’’ products. In Fig. 5b, is presented the connection of displacement transducer with steering damper. Fig. 6a, shows connection of three transducers (from left to right, triaxial accelerometer ASC, load cell Z8 HBM, linear displacementmeter) with electronic measuring system Spider – 8 HBM during of measured channels setting. All the signal conditioning , transducers excitation , amplification, digitalization and computer connection are realized by means of Spider – 8 HBM. The motorcycle steering damper used as support to setting of displacement transducer is shown in Fig. 6b. The triaxial accelerometer 4520-002 B&K is connected with motorcycle frame as presented in Fig. 7a, and with vibration analyzer 4447 B&K according to Fig. 7b.

Above presented measuring system allows to investigation of system rider – motorcycle – road in different riding condition with possibility to inclusion or expulsion some measured options dependently from actual research tasks [14]. In this paper the adequate option was used to system testing by curved motion and varied velocity.

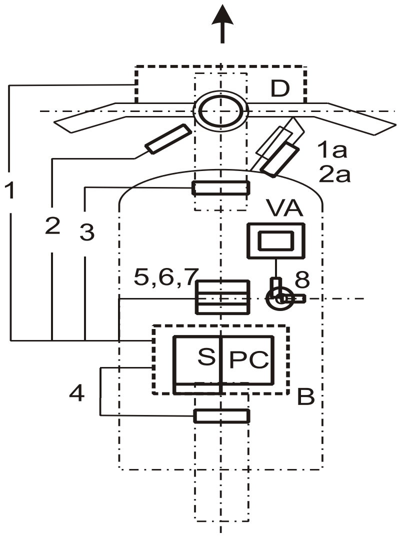

Figure 4: *Block - diagram of motorcycle measuring equipment*

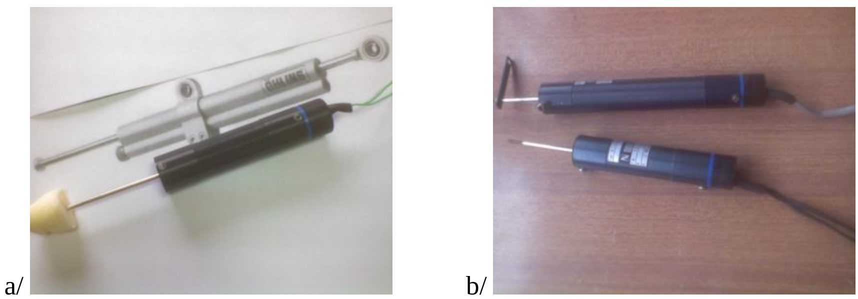

Figure 5: *The view of displacement transducers Novotechnik:a/ connecting with steering damper, b/ comparison two different types*

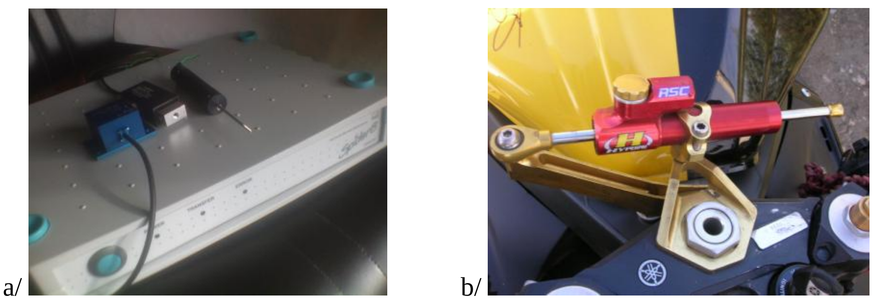

Figure 6: *a/ triaxial accelerometer, load cell and linear displacementmeter connected to measuring system Spider – 8 HBM, b/ steering fork damper*

Figure 7: *Triaxialaccelerometer and vibration analyzer*

RESULTS

The preliminary experimental tests, aimed to check of measuring system reliability and verification of proposed methodology, were carried out on the curved paths. The varied combination of curve parameters and motorcycle velocity values were chosen. For given curve, ride velocity is varied step by step, keeping its steady value during one passage. Characteristic segments of curved motion time history, for example, input transient curve, middle curve part of constant radius, output transient curve, were focused to further data processing, results analysis and presentation. According to planed research, in this paper, the results of system behaviour at steady state in middle curve part are presented, by pointed out the influence of riding velocity.

Measured data as system input and output variables, denoted in Figures 1, 2, 3, were processed and graphical presented in Fig. 8 to 12, as follows. Fig. 8 a, shows relatioship between system roll angle and longitudinal velocity in riding domain from 8 to 20 m/s. As can see from this presentation the system roll angle is a nonlinear function of velocity. The estimated norm of residuals, as a measure of curve fitting goodness, indicates that polinomial of 4. degree is acceptable approximation. This quantity pass over theoretical value of 2 obtained for simplified simulation models. Similar trend in view of deviation from theoretical linear direction shows relationship between yaw velocity and longitudinal velocity in Fig. 8b. The reasons for these deviation are unincluded affected factors by theoretical consideration and coducted simulation.

The angle of motorcycle steering fork turning, named in Fig. 9 a, as steering angle, decreases by increasing of longitudinal velocity. This means that for given configuration and parameters and by motion in chosen curve, system exibits so called oversteering tendency, one term often used in automobile technics, [11], [12], [13]. More information about this vehicle propreties will be given by analysis Fig. 12.

The intersting finiding follows by comparison the change of rider lean angle with longitudinal velocity , Fig. 9b, and change roll angle in Fig. 8a, at equal condition. As can see, the both curve show similar trend. But, rider lean angle has at lower velocity negative initial values. Mutual relation between of these two quatities, shown in Fig. 11b, approximate linear, indicates the rider used strategy. The relationship between lateral acceleration and longitudinal velocity , presented in Fig. 10a is of fundamental importance for this research, from more reasons. The first, as measure of lateral load with respect to roll angle stability limits and tire adhesion limits. The second, as basic measured variable suitable to identification some unmeasured or hardly measured variables, for example vehicle path radius, steering angle turning and so on, based on the principle virtual sensors [10]. The curve fitting with second order polinomial is a good approximation as can see in Fig. 10a. Fig. 10b, shows mutual relation of steering angle and lateral acceleration, also with second order polinomial approximation and corresponding the norm of residuals ploted in Fig. 11 a.

The components of steering angle are presented in Fig. 12a. The change of curve 1, shows direct influence of longitudinal velocity, until the curve 2 shows part of steering angle caused by tire cornering propreties, also indirect influences. By comparison curve in Fig.12 a ,b, can be concluded, that component 2 contributes to above mentioned oversteerig tendency. This is an important result about the motorcycle steering properties which indicates to possibilities of proposed methodology.

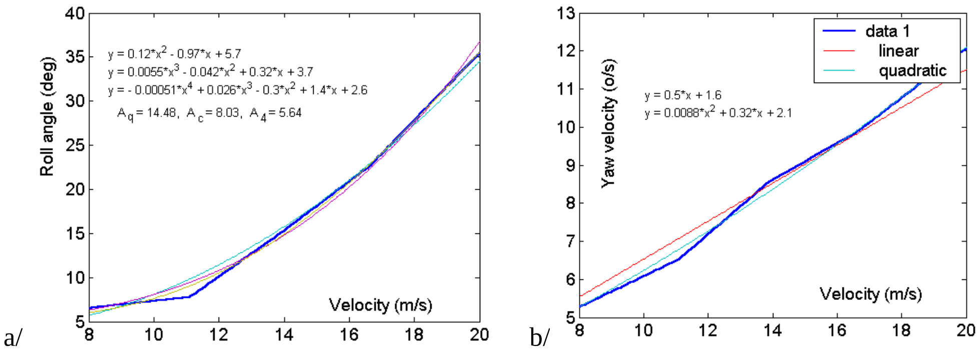

Figure 8: *a/ roll angle, b/ yaw velocity, versus longitudinal velocity by steady state curved motion*

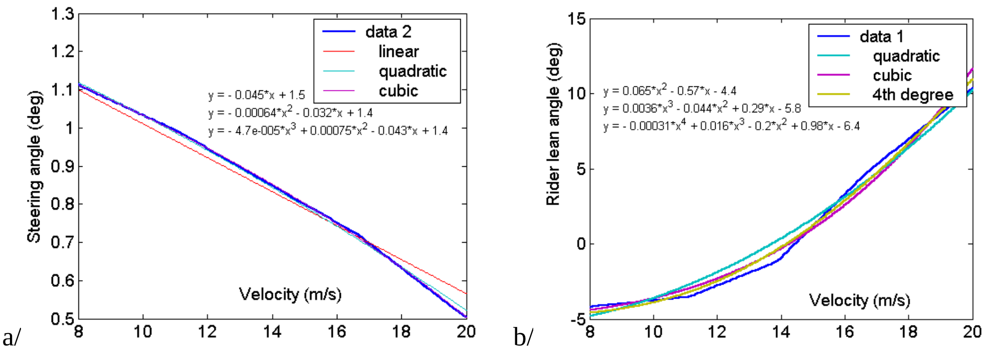

Figure 9: *a/ steering angle, b/ rider lean angle, versus velocity by steady state curved motion*

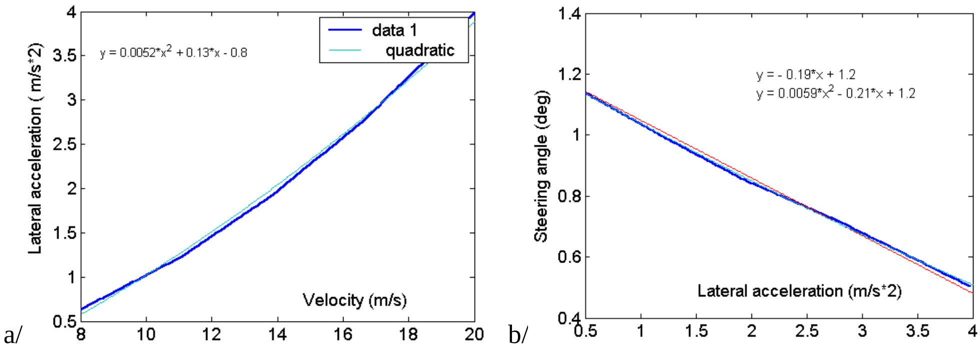

Figure 10: *a/ lateral acceleration versus velocity,*
b/ steering angle versus lateral acceleration

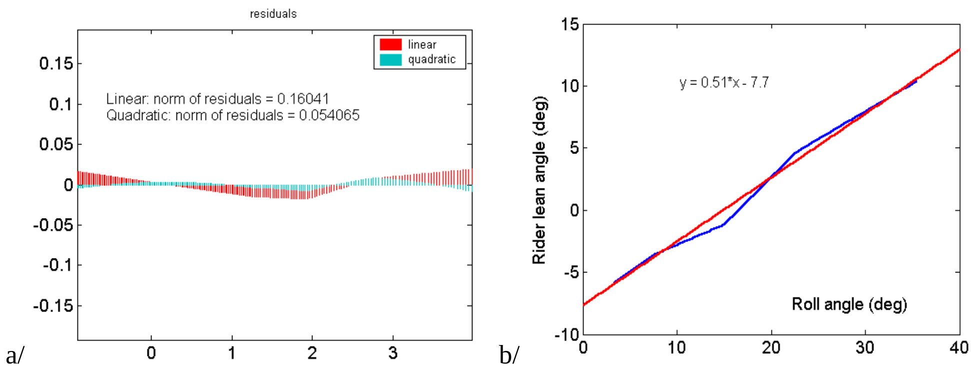

Figure 11: *a/ validation of curve fitting in Fig. 10b,*
b/ relation between rider lean angle and roll angle

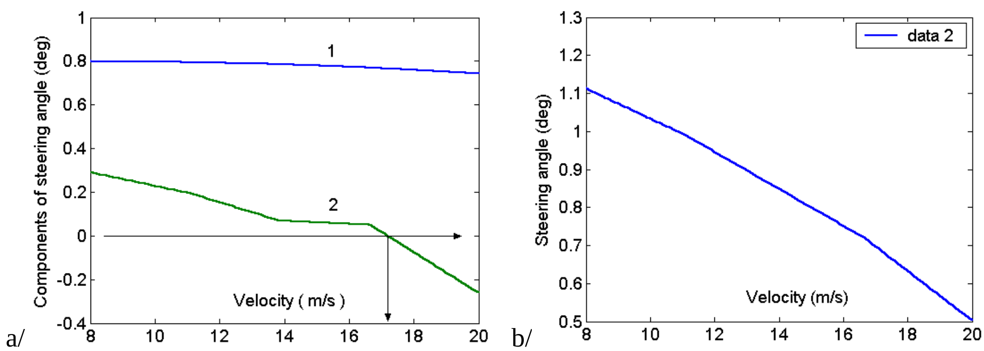

Figure 12: *a/ components of steering angle, b/ sum of steering angle, versus velocity.*

CONCLUSIONS

The dynamical characteristics of motorcycles in interaction to rider behaviour have the essential influence on the traffic safety. In the literature there are many papers on the steering properties of motorcycle but only a few deal with experimental research. To design an adequate experimental equipment an appropriate basic model of system must be chosen. Limited motorcycle space and dimension, influence of masses ratio to measurement accuracy require the application of compact measuring devices. The preliminary results of experimental tests obtained during motion in the curve of real road condition indicates to essential influence of motorcycle velocity on the stability limits quantities as measures to assessment steering properties and levels of its active safety. These results, also discover news possibilities to integration advanced simulation methods with modern measuring electronics , supported by logistics systems to solution , above pointed out, actual problems of motorcycle dynamics.

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[1] CONTRIBUTION IN DEFINING PARAMETERS OF THE EQUIVALENT SYSTEM OF TORSIONAL VIBRATIONS WITH THE SIGNIFICANT INTERNAL DAMPING ELEMENTS

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Abstract

In the development of internal combustion engine as propulsion systems for road vehicles, a large emphasis is on increasing the efficiency by means of down-sizing. On the one hand it requires the reduction of specific weight and specific volume of internal combustion engine, that includes reducing mass of individual parts like the crank mechanism, and on the other hand it is expected to increase the effective engine parameters per unite of volume. Considering that internal combustion engines belong to the category of the extremely dynamic mechanical systems, any change in the masses and in the construction of the components requires a detailed vibration phenomena analysis.

In the paper are presented aspects of choosing equivalent torsional vibration systems, ways in defining the individual elements of the system, with special emphasis on the elastic torsional vibration damper as an element of highly nonlinear characteristics of internal damping and stiffness.

Key words: torsional vibration, internal damping, elastic damper.

DOPRINOS DEFINISANJU PARAMETARA EKVIVALENTNOG TORZIONO OSCILATORNOG SISTEMA SA ELEMENTIMA SA ZNAČAJNIM UNUTRAŠNJIM PRIGUŠENJEM

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Rezime: U okviru razvoja motora sus, kao pogonskog agregata za drumska vozila, veliki akcenat se stavlja na povećanje efikasnosti putem tzv. down sizing-a. S jedne strane zahtjeva se smanjivanje specifične težine i specifične zapremine motora sus, što zahtjeva smanjenje mase pojedinih dijelova uključujući i krivajni mehanizam, a s druge strane očekuje se povećanje efektivnih parametara po jedinici zapremine. S obzirom da motor sus spada u kategoriju visoko dinamičkih mašinskih sistema, svaka promjena u masama i konstrukciji dijelova zahtijeva i detaljno razmatranje oscilatornih fenomena.

U okviru rada predstavljeni su aspekti izbora ekvivalentnog torziono oscilatornog sistema, načina definisanja pojedinih elemenata sistema, s posebnim naglaskom na elastični

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prigušivač torzionih oscilacija kao element sistema sa izrazito nelinearnim karakteristikama unutrašnjeg prigušenja i krutosti elastičnog elementa.

Ključne reči: torzione oscilacije, unutrašnje prigušenje, elastični prigušivač.

CONTRIBUTION IN DEFINING PARAMETERS OF THE EQUIVALENT SYSTEM OF TORSIONAL VIBRATIONS WITH THE SIGNIFICANT INTERNAL DAMPING ELEMENTS

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INTRODUCTION

Internal combustion engine is a cyclic machine that develops its power from associated chemical energy of liquid or gaseous fuel that is released through the combustion process. Because of the cyclic process, the unevenness of fuels quality and the stochastic combustion process that takes place in a very short period of time in the cylinder, internal combustion engine is subjected to uneven mechanical loading. To obtain a more uniform speed of rotation of IC engine it is common to set additionally a flywheel. But even so, there is a variation of the current effective torque which represents the excitation of torsional vibrations of the crankshaft. The occurrence of vibrations is undesirable and it is because of the additional dynamic load of the crankshaft that can lead to its fracture, which is especially pronounced in more powerful IC engines.

Since the torsional vibration system of IC engines, including its additional elements in the form of pumps, additional drives, etc., is very complex, it is very difficult to describe it mathematically in exact from, so it is necessary to set a model that will be a dynamical equivalent to the actual system. The parameters identification phase of the equivalent IC engines torsional vibration system in practice faces a number of difficulties that are associated with certain unknowns. With the aim to mathematically solve the problem it is common in everyday engineering practice to use approximations, which give good enough agreement with experimental results. The fact that a lot of different empirical and half-empirical approaches for parameters identification of the IC engine torsional vibration system are existing, indicates the complexity. Knowing the critical working areas of the IC engine, devices and systems can be designed that enable shifting of critical regime from the working area of the system, or, which is in practice more often, which would provide damping of critical vibration amplitudes.

There is a large number of different solutions for torsional vibration dampers taking benefit of damping properties of different materials. Commonly for the IC engine applications viscous and viscous-elastic dampers are in use. According to their simple design, reliability, diversity and economy in production the viscous – elastic dampers have priority.

Viscous-elastic dampers use, such as the word itself says the viscous-elastic properties of polymers (rubber) for damping torsional vibrations. However, the properties of polymers are not constant and vary greatly with temperature, frequency and vibration amplitudes of

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the system. The existence of such an element in torsional vibration system, despite all its undoubted benefits, brings unknowns that have to be defined, especially in very responsible systems.

The following gives an example of determining the equivalent torsional vibration system with special emphasis on possible ways of presenting system damping as a highly nonlinear component.

1. EQUIVALENT TORSIONAL VIBRATION SYSTEM OF THE IC ENGINE

Due to limited knowledge and often complicated mathematical apparatus that follows original presentation of the torsional vibration system, it is necessary for practical application to develop equivalent systems that are simpler and which also reflect with sufficient accuracy events in the real system.

Often in use is a in line equivalent system where all elements of the vibration system are reduced to a single axis, of course with the condition of equality of potential and kinetic energy of the real and equivalent systems. The in-line equivalent torsional vibration system with torsional vibration damper for a 6 cylinder in-line, water cooled diesel engine is shown in the figure 1. Moments of inertia (Θ) for the cylinders are different because of different contra weights at throws of crankshaft.

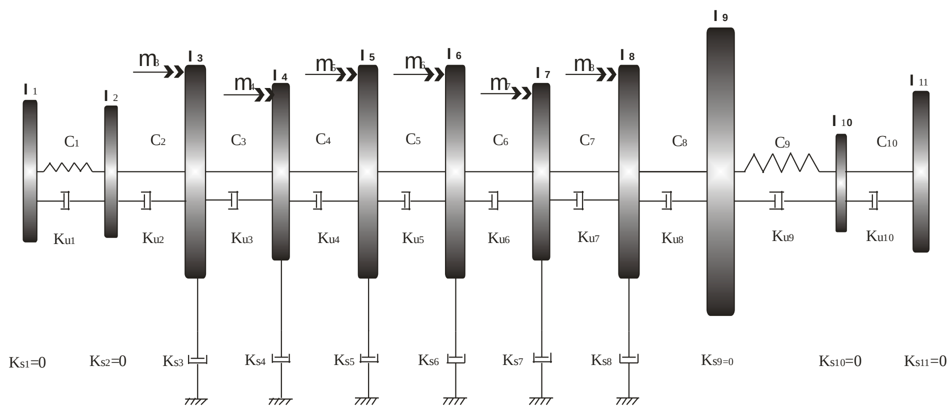

Figure 1: *Equivalent torsional vibration system of a IC engine*

1.1 Parameter determination of the torsional vibration system

After adoption of the equivalent torsional vibration system it is necessary to define its parameters. Determination of a set of parameters of the equivalent torsional vibration systems such as moments of inertia of concentrated masses (Θ), reduced length (l) and torsional stiffness of shaft parts (c), is extensively described in many literature sources, especially in [2, 3, 4].

Furthermore, it is necessary to determine the excitation torque (M) by IC engine cylinders. The most reliable estimation of torque is through measurement, more exactly through measurement of indicated pressure in cylinder of IC engine for the given case. Having in mind great demands for carrying out such measurements, different half empiric methods for estimation of the uneven part of the excitation torque, presented in form of Fourier series, are in use. This methods differs one from another in number of influence parameters taking in to account [6].

For a complete definition of a torsional vibration system it is necessary to determine the parameters that define the damping in the system.
There are different approaches to describe the damping, but basically they are:

- Describing the damping of the entire system based on the known properties of the system or structure, without determination of damping for individual (local) structural elements;
- Describing the damping of the system based on the known properties of certain structural elements of the system; (describing the damping of the system components and based on them damping of the system);
- Describing the damping on hand of describing behaviour of materials using the laws of continuum mechanics (constitutive equations for materials).

In this paper total system damping is separated to internal (K_u) and external (K_s) damping. Internal damping represents the material damping, that highly depends on geometric shape, material properties and vibration frequency. The internal damping (K_u)is estimated on base of using empiric relation presented in [9]:

$$K_u = \frac{c\,\mathrm{M}_1^{1/2}}{a\,\omega\,\sqrt{\left(R_0^3 - R_i^3\right)\dfrac{\pi}{2}}} \tag{1}$$

where are: c- stiffness, M_1- excitation torque , ω- circular vibration frequency, R_0, R_i- outer and inner half diameter of shaft, a - coefficient that depends on kind of material.

External damping represents resistance of moving parts of the IC engine (pistons, connector rods, crankshaft, etc.) and other auxiliary engine devices and is very difficult for explicit determination. A common approach is to use a combination of empirical recommendations, such as [3, 8, 9], and through confirmation of the selected damping on the basis of calculation the vibration amplitude, which again can be experimentally determined.

A particular problem is the introduction of the contribution of the torsional vibration damper in vibration model. The problem occurs primarily because of the nonlinearity of the damping and its dependence on a number of parameters that are changeable with the IC engine regimes. In figure 2, for purposes of illustration, are shown the changes of stiffness and damping of torsional vibration dampers depending on the relative amplitude of vibration [1], for a concrete torsional vibration damper of an engine (figure 1). Because of great number of quantities that can be encountered in the literature as a measure to define the internal damping of viscous-elastic dampers, within table 1 an overview of commonly used terms for their definition and relationships between these quantities is given.

Table 1. *Correlation between damping quantities*

	K_u	δ	ξ	D	Λ
Damping coefficient K_u	K_u	$2I\delta$	$2\,I\omega_0\xi$	$I\omega_0 D$	$\frac{I\,\omega_d\Lambda}{\pi}$
Suppression coefficient δ	$\frac{K_u}{2I}$	δ	$\omega_0\xi$	$\omega_0\frac{D}{2}$	$\frac{\omega_d\Lambda}{2\pi}$
Damping of Lehr ξ	$\frac{K_u}{2\sqrt{cI}}=\frac{K_u}{2I\,\omega_0}$	$\frac{\delta}{\omega_0}$	ξ	$\frac{D}{2}$	$\frac{\Lambda}{2\pi}\frac{\omega_d}{\omega_0}$
Loss factor D	$\frac{K_u}{\sqrt{cI}}=\frac{K_u}{I\,\omega_0}$	$\frac{2\delta}{\omega_0}$	2ξ	D	$\frac{\Lambda}{\pi}\frac{\omega_d}{\omega_0}$

Table 1 *Correlation between damping quantities (continues from previous page)*

	Ku	δ	ξ	D	Λ
Logarithmic decrement Λ	$\frac{\pi K_u}{I\,\omega_d}$	$\frac{2\pi\delta}{\omega_d}$	$\frac{2\pi\xi}{\sqrt{1-\xi^2}}$	$\frac{\pi D}{\sqrt{1-\frac{D^2}{4}}}$	Λ
Band wide of frequency response $\Delta\eta=\frac{\Delta\Omega}{\omega_0}$	$\frac{K_u}{\sqrt{cI}}=\frac{K_u}{I\,\omega_0}$	$\Delta f=\frac{\delta}{\pi}$ $\Delta\Omega=2\delta$	2ξ	D	$\frac{\Lambda}{\pi}\frac{\omega_d}{\omega_0}$
Dynamic amplification factor (η = 1) M_r	$\frac{\sqrt{cI}}{K}$	$\frac{\omega_0}{2\delta}$	$\frac{1}{2\xi}$	$\frac{1}{D}$	$\frac{\pi}{\Lambda}\frac{\omega_0}{\omega_d}$
Damping ratio $\Psi=\frac{W_D}{W_{pot}}$	$\frac{2\pi\Omega K_u}{c}$	$\frac{4\pi I\,\Omega\delta}{c}$	$4\pi\eta\xi$	$2\pi\eta D$	$2\eta\Lambda\frac{\omega_d}{\omega_0}$
tg ε_a Loss angle	$\frac{\Omega K_u}{c}$	$\frac{2I\,\Omega\delta}{c}$	$2\eta\xi$	ηD	$\frac{\eta\Lambda}{\pi}\frac{\omega_d}{\omega_0}$

In the table 1 significance of tag is as follows:

$\omega_0=\sqrt{\frac{c}{I}}$ – Natural frequency of the system without damping

$\omega_d = \omega_0\sqrt{1-\xi^2}$ - Natural frequency of the system with damping

Ω - Frequency of excitation

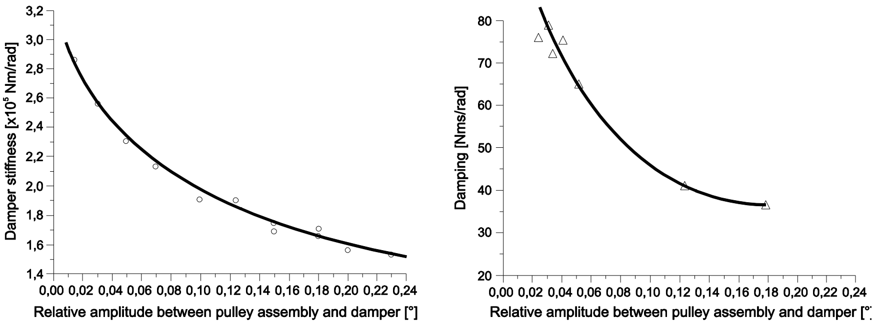

Figure 2: *Change of stiffness and damping of torsional vibration damper, for a concrete example, as function of relative vibration amplitude*

In the mass production of torsional vibration dampers for IC engines, as a measure of quality assurance, measuring of characteristic quantities of the damper is done. Common are measurements for estimation of the dampers nominal frequencies at prescribed temperature conditions. This measurement should confirm that the previously established parameters, like dynamic stiffness (c_d) and loss angle ($tg\varepsilon_a$), after production of multiple dampers are kept at the required tolerances.

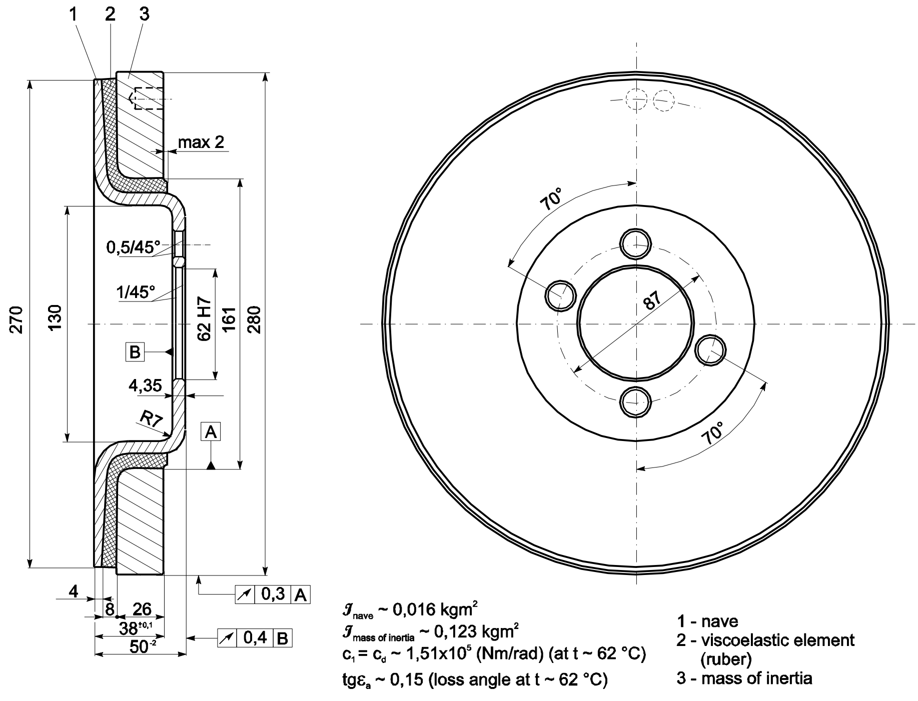

Figure 3: *Viscous-elastic torsional vibration damper*

Figure 3 shows a drawing of a particular torsional vibration damper with the basic parameters provided by the manufacturer.

2. SOLVING AND VERIFICATION OF EQUIVALENT TORSIONAL VIBRATION SYSTEM

After identifying all the necessary parameters of the system, it is necessary to establish a system of equations describing the considered processes.

Torsional vibrations of the individual disks presented in the model (figure 1) can be described trough system of equations (2) in matrix notation:

$$[\mathcal{J}]\{\ddot{\vartheta}\}+[K_u]\{\Delta\dot{\vartheta}\}+[K_s]\{\dot{\vartheta}\}+[C]\{\Delta\vartheta\}=\{M\} \qquad (2)$$

where are: ϑ twisting angle, $\dot{\vartheta}$ and $\ddot{\vartheta}$ derivation of twisting angle, $\Delta\dot{\vartheta}$ difference of twisting velocity of neighbouring disks, $\Delta\vartheta$ difference of twisting amplitudes of neighbouring disks. Knowledge of all parameters from system of equations leads to solve twisting amplitudes of the pulley of the IC engine. The pulley assembly is the most common location for measurement of vibration amplitudes. The obtained system of equation is solved by MatLab.

For the purpose of verifying torsional vibration model IC engine with following data is analysed:

- Natural aspirated, in line 6 cylinder diesel engine with firing order 1-5-3-6-2-4;
- diameter/stroke of piston 123 mm/140 mm; compression ratio ε = 16,5;
- max power/engine speed: 143 kW/2200 rpm;
- torsional vibration damper on the specific engine is taken from figure 3. In one example the parameters of the torsional vibration damper are taken from figure 3 (c_1=1,51 10^5 Nm/rad, K_{u1}=2,51 Nms/rad ($tg\varepsilon_a$=0,15)), and in the second example parameters are estimated according to experimental analyses and calculations.

By analyzing the results obtained by calculation and their comparison with experimental results, figure 4, it can be concluded that the approach of establishing an equivalent inline torsional vibration system, and the way of determining the parameters of the system, are good enough and acceptable for engineering practice, where the results obtained could be useful for evaluating the efficiency of torsional vibration dampers [1, 7].

Figure 4 presents comparative calculated and experimental results as a result of observation of significant excitation orders and 2nd mode vibrations of the previously described IC engine with a torsional vibration damper. Far better agreement of calculated and experimental results is in the case of knowing the real (nonlinear) characteristics of the elastic element of torsional vibration damper.

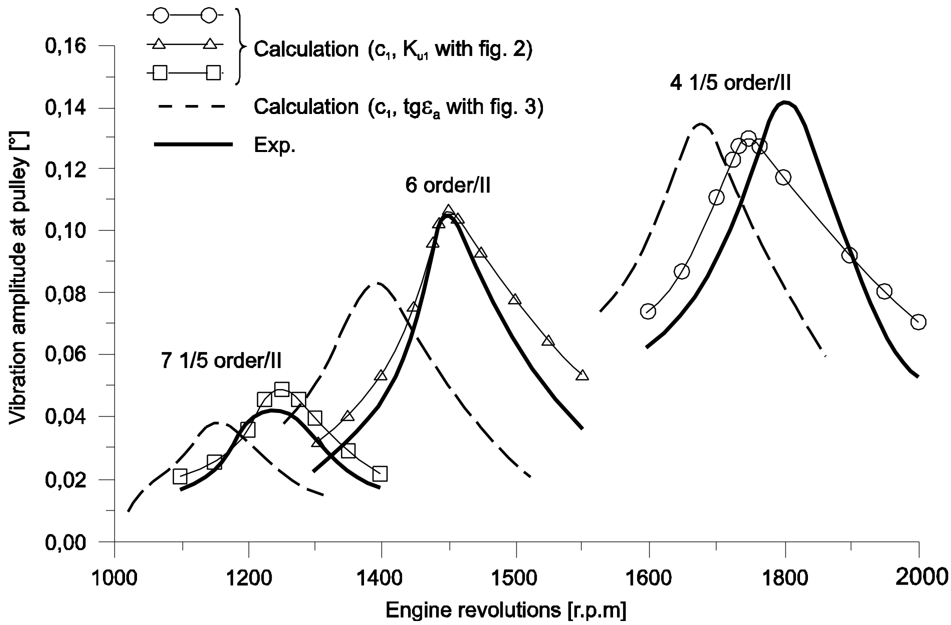

Figure 4: *Vibration amplitude at engine pulley, as a function of engine revolution, for interesting excitation orders and II mode vibrations*

Based on results achieved in this manner, values of dynamic amplitudes of twisting tension on individual part of the engine crankshaft can be calculated which is helpful in determining the dynamic endurance of the engine crankshaft.

CONCLUSIONS

Complicated torsional vibration systems of IC engines, for the purposes of engineering practices, can be effectively replaced by the so-called in line equivalent torsional vibration systems. The calculation results of torsional vibration of certain parts of the system, obtained by the proposed method of calculation, agree quite well in character and the absolute values with the real-measured values. The biggest problem in determining the individual parameters of equivalent torsional vibration systems is the nonlinear damping of torsional vibration damper. Based on comparisons of calculated and experimental results it is confirmed that the entire process of analysis of torsional vibrations, from physical model, ways of determining the input parameters, mathematical models to numerical solutions for the everyday practice of analyzing the influence of certain parameters is largely satisfactory, with provided knowledge of the nonlinear characteristics of elastic element of torsional vibration damper.

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