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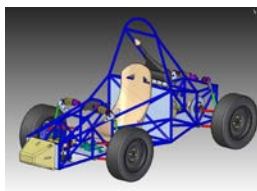
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de Automobile
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FORMULA STUDENT



Studiul mișcării squish în cilindrii motoarelor diesel ● Analiza datelor experimentale caracteristice funcționării m.a.i. ● Validarea sistemului de monitorizare indirectă a presiunii din pneu ● Evaluarea vibrațiilor autovehiculelor la deplasarea în teren accidentat ● Studiul comportamentului aerodinamic al unui kart electric folosind CAD și CFD ● Diagnosticarea elementelor pompei injector

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DICȚIONAR EXPLICATIV PENTRU ȘTIINȚĂ ȘI TEHNOLOGIE AUTOVEHICULE RUTIERE

Autori: **Gheorghe FRĂȚILĂ, Cristian ANDREESCU, Cornel VLADU,**
Raluca MOISESCU, Cornelia STAN, Marius TOMA, Gabriel CRISTEA

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Dicționarul de față reunește și completează termenii publicați în volumele I și II ale Dicționarului Explicativ pentru TRANSPORTURI – Autovehicule Rutiere, apărute în anii 2006 și respectiv 2009, în seria DICȚIONARE EXPLICATIVE pentru ȘTIINȚĂ ȘI TEHNOLOGIE, coordonată de regretatul acad. Gleb Drăgan. El se referă la terminologia domeniului AUTOVEHICULELOR RUTIERE și este realizat pe baza fondului terminologic standardizat pe plan mondial.

Având la obârșie construcția simplă a unei trăsături cu motor, automobilul a evoluat spectaculos pe parcursul unui secol, devenind un sistem complex la realizarea căruia contribuie specialist din multe domenii: mecanică, electronică, hidraulică, știința materialelor, optică, automatizări și prelucrarea informației, design și altele. Acestora li se adaugă un mare număr de oameni implicați în activități conexe realizării automobilului propriu-zis, activități care au căpătat dimensiunile unor adevărate „industrii”: comercializare, mentenanță, asigurări, organizarea traficului și transportului rutier, definirea regulamentelor tehnice privind automobilul și urmărirea respectării lor etc. În aceste condiții, fondul de termeni cuprinși într-un dicționar explicativ referitor la domeniul autovehiculelor rutiere ar trebui să ajungă la dimensiuni apreciabile –după unii specialiști, până la 15.000-16.000. Întrucât, așa cum s-a arătat mai sus, prezenta lucrare se încadrează într-o serie de dicționare elaborate pe diferite domenii, s-a considerat că pentru evitarea unor suprapuneri, să fie prezentate termenii specifici autovehiculelor rutiere, iar dintre cei aparținând științelor fundamentale (matematică, fizică, chimie ș.a.) sau celor tehnice generale (rezistența materialelor, organe de mașini, electrotehnică, electronică, termotehnică, informatică ș.a.) să fie abordați doar cei mai importanți și intim legați de ingineria autovehiculelor. Forma în limba română, termeni și definiții, au rezultat în urma consultării unui număr larg de factori tehnico-economici, de cultură și educație apropiați domeniului, prin anchetă publică.

Autorii au dedicat realizarea lor promotorului învățământului superior de autovehicule rutiere din țara noastră, profesorul universitar inginer Constantin Ghiulai și memoriei academicianului Gleb Drăgan, inițiatorul secției DICȚIONARE EXPLICATIVE pentru ȘTIINȚĂ ȘI TEHNOLOGIE.



SISTEME DE PROPULSIE HIBRIDE ELECTRICE PENTRU AUTOMOBILE – PROTOTIPURI VIRTUALE

Autori: **Valerian CROITORESCU**

Editura Politehnica Press
Anul apariției: 2016
ISBN 978-606-515-694-4

Lucrarea tratează, într-o manieră sistematică și amplă, modalități de dezvoltare prin modelare și simulare a sistemelor de propulsie hibride electrice și a elementelor componente specifice unor astfel de sisteme de propulsie. Astfel, lucrarea prezintă modalitățile prin care se dezvoltă prin modelare un sistem de propulsie hibrid electric, care funcționează folosind arhitectura paralelă post-transmisie. Dezvoltarea acestui sistem de propulsie se face într-o manieră modulară, fiind facilă introducerea ulterioară a altor elemente componente necesare. Elementele specifice studiate și dezvoltate în lucrare sunt mașina electrică cu reluctanță variabilă, transmisia cu variație continuă și bateriile de acumulare cu element activ principal litiu. Pentru fiecare dintre acestea lucrarea prezintă modalitățile de dezvoltare a modelelor funcționale și modelelor termice, care au fost integrate ca elemente componente ale modelului de sistem de propulsie hibrid electric, care la rândul său face parte dintr-un model de autovehicul.

Lucrarea cuprinde o introducere, bibliografia și următoarele capitole:

1. Modelul global al sistemului de propulsie hibrid electric
2. Modelarea mașinii electrice
3. Modelarea transmisiei
4. Modelarea bateriilor
5. Managementul energetic al sistemului de propulsie



PROIECTE ÎNCHISE, PROIECTE VECHI, PROIECTE NOI

CLOSED PROJECTS, OLD AND NEW



Adunarea Generală Extraordinară a „ATA – *Associazione Tecnica dell'Automobile*”, convocată în 18 noiembrie 2016, a hotărât dizolvarea societății după o activitate de aproape șapte decenii.

O pierdere semnificativă pentru lumea specialiștilor automobiliști.

Luând act de acest fapt, executivul FISITA a operat modificările necesare pe site-ul său.

ATA – *Associazione Tecnica dell'Automobile* – asociația profesională din Italia similară SIAR, a fost înființată la 3 ianuarie 1948 la inițiativa unui grup de automobiliști și motorști.

Unul dintre promotorii și fondatorii ATA (și primul redactor șef al revistei „Giornale e Atti dell'ATA”) – dr. Gino Pestelli, în discursul inaugural, considera ATA drept un spațiu de manifestare a excelenței italiene în domeniul automobilului: „*Per il passato, per il presente, per l'avvenire*”.

ATA a fost concepută ca loc de întâlnire nu doar a inventatorilor și creatorilor în domeniul autovehiculelor, ci și a constructorilor, ziariștilor, antreprenorilor, diferiților experți.

ATA s-a constituit ca o asociație în esență culturală, care viza înțelegerea reciprocă și cooperarea armonioasă între cei care își desfășurau activitatea în domeniu, în cel mai larg înțeles: un spațiu de diseminare a cunoștințelor tehnice, de promovare a cercetării științifice, un centru de promovare a inițiativelor utile, în special practice, pentru dezvoltarea industrială, comercială și socială.

ATA a contribuit la constituirea la 24 ianuarie 1948 a FISITA (*Fédération Internationale des Sociétés d'Ingénieurs des Techniques de l'Automobile*) și la organizarea primului Congres Internațional al FISITA (23 Septembrie 1948) cu prilejul celui de al 31-lea Salon Internațional al Automobilului de la Torino. Numeroase alte manifestări științifice mondiale, europene și naționale au fost găzduite de ATA de-a lungul timpului.

Odată cu constituirea ATA a apărut și publicația lunară a asociației „Giornale e Atti dell'ATA”. Primul număr a fost datat 15 ianuarie 1948. Începând cu 2014 revista s-a reorganizat sub numele „Ingegneria dell'Autoveicolo”, cu 6 apariții și un tiraj de 30.000 exemplare anual.

Editura ATA a permis publicarea unui număr important de titluri din domeniul ingineriei autovehiculelor. Un fond bibliografic de documentare important a fost la dispoziția membrilor societății.

Încă din 1949 ATA a constituit „Grupul studenților ATA”, ceea ce mai târziu a devenit „ATA Università”. Implicarea și promovarea tinerilor automobiliști a fost o preocupare a ATA, dintre aspectele importante în această direcție remarcând „Premiile ATA” adresate tinerilor ingineri și Formula ATA ce cuprinde Formula SAE Italy și Formula Electric și Hybrid Italy (prima ediție 2005, ultima 2016).

ATA a avut anual circa 3000 membri și peste 100 membri firme/colective universitare.

ATA: deja istorie, un proiect cu un final nedorit și trist...

Când am primit informația privind desființarea ATA, am accesat mai întâi site-ul FISITA și am constatat absenta Italiei și ATA în rândul societăților europene membre FISITA.

Apoi, pe site-ul ATA (www.ata.it) am găsit convocarea adunării generale extraordinare având la ordine de zi dizolvarea asociației.

Nu am găsit informații privind cauzele dispariției ATA.

Am observat doar câteva elemente legate de apariția revistei „Ingegneria dell'Autoveicolo” (la SIAR avem colecția revistei, ultimul număr primit fiind mai/octombrie (!) 2015), frecvența redusă a unor acțiuni ATA la nivel național etc.

ATA a sprijinit SIAR prin diverse acțiuni și numeroși membri ai SIAR au prietenii membri ATA! Regretăm dizolvarea ATA! Sperăm într-o rapidă renaștere a ATA!

Observând însă această întâmplare nu putem să nu ne punem întrebări privind viitorul SIAR!

O sumară analiză scoate la iveală două tendințe, contradictorii.

Prima, pozitivă, generatoare de optimism, constată o creștere importantă a numărului de membri SIAR: mai mult de dublare în ultimii patru ani! Și, pe această linie, cu rezerve importante de creștere dacă mai mulți studenți (licență, master) ar fi atrași ca membri SIAR Junior în cadrul structurilor teritoriale constituite în principalele centre universitare.

Pe de altă parte, factori diverși au determinat o anumită diminuare a acțiunilor membrilor SIAR privind:

- publicarea de lucrări în revistele „Ingineria automobilului” și „Romanian Journal of Automotive Engineering”;
- participarea la Congresul internațional anual al SIAR și alte manifestările tehnico-științifice;
- publicarea de lucrări științifice, manuale, cursuri universitare etc.
- colaborarea și organizarea de activități comune cu mediul economic/industrial specific;
- promovarea în societate a domeniului ingineriei autovehiculelor, transporturilor și siguranței rutiere etc.

Dispariția ATA ar putea constitui un moment de reflexie asupra proiectelor SIAR! Asupra coeziunii asociației noastre, a inițiativei, implicării și punerii în valoare a entuziasmului tinerilor și experienței membrilor cu vechime în SIAR!

Proiecte mai vechi sau mai noi (Concursul național studentesc „Prof. ing. Constantin Ghiulai” – ediția a IV-a – secțiunea „Dinamica autovehiculelor”, ediția I-a „CAD Catia”, Competiția Kart Low-Cost, Formula Student și multe altele.) așteaptă să fie dezvoltate și valorificate!

Un moment de analiză, coeziune, inițiativă și acțiune la care invit pe toți membrii societății noastre!

Prof. dr. ing. Minu MITREA, Secretar General SIAR

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INTERVIU CU ING. ADINA-MARIA SOROȘTINEAN

INTERVIEW WITH MISS ADINA-MARIA SOROȘTINEAN



Adina Sorostinean was a finalist at the EuroBrake Student Innovation Competition 2016 in Milan, Italy. Adina has continued to study within the mobility engineering sector and has broadened her horizons by travelling to Paris to take up a Masters programme.

We caught up with Adina to talk all things mobility, EuroBrake and where she sees herself in a few years...

Tell us about your Masters programme, what are you focusing your studies on?

In September this year I started a Masters in Mobility and Electric Vehicles as a student of the Group Renault Corporate Foundation. The courses take place at five Universities in Paris and Lille and they address all the problems related to electric vehicles and their infrastructure...

Did you always want to be an automotive engineer/what inspired you?

I was always passionate about discovering how things around me work and I loved mathematics and physics in school. Therefore I decided to become an engineer but the passion for vehicles came afterwards...

Did you do any work experience/internships?

I understood pretty quickly that if I want to progress in the automotive domain I have to gain a lot of experience in a short time. During my studies I took part twice in the Erasmus+ exchange program. After graduation I realised that I was still far from having the experience that I wanted so I started another internship. This time was closer to the industry at BMW Munich...

What advice would you give to students who are keen to enter the automotive engineering industry?

Automotive engineering is a very challenging field and it requires a lot of dedication but it is also very rewarding because as an automotive engineer you literally make things move...

What do you hope to be doing in your career in 5 years' time?

The automotive sector is facing a big change with two important

directions: the electric vehicle and the autonomous vehicle...

In 3 words how do you see the current automotive engineering industry?

Revolutionary, innovative, challenging.

Read the full interview with Adina here:

Cititi interviul cu Adina-Maria aici:

<https://www.fisita.com/yfia/careers/tips>

http://www.fisita.com/documents/Adina_Interview_BW.pdf

Adina-Maria Soroștinean a absolvit în anul 2015 Academia Tehnică Militară din București, Facultatea de Mecatronică și Sisteme Integrate de Armament, specializarea „Echipamente și sisteme de comandă și control pentru autovehicule”.

Ing. Adina-Maria Soroștinean este din 2014 membră a Societății Inginerilor de Automobile din România – SIAR și a FISITA – International Federation of Automotive Engineering Societies.

About FISITA



FISITA is the international network for automotive mobility engineers, representing over 210,000 engineers in 36 countries

- Established in 1948, FISITA provides a global platform for knowledge exchange between industry and academia. FISITA is also a leading advocate for the education of young engineers, creating pathways for future talent through its established links with the industry

- FISITA helps guide the future direction of the automotive mobility engineering profession by contributing to the development of safe, sustainable and affordable mobility solutions

- FISITA organises a number of global events, in addition to EuroBrake, including the FISITA World Automotive Summit – an exclusive annual meeting of industry leaders, and the FISITA World Automotive Congress – a forum for industry experts, engineers and executives to exchange ideas and discuss the trends that drive the automotive mobility industry forward

- For more information about FISITA see www.fisita.com.

Interviul a fost publicat prin amabilitatea FISITA.

This interview is published through FISITA courtesy.

Foto: Giuseppe Maco



STUDIES REGARDING THE INFLUENCE OF THE SQUISH IN-CYLINDER MOVEMENT OF THE AIR IN A DIESEL ENGINE

STUDII PRIVIND INFLUENȚA MIȘCĂRII DE SQUISH ÎN INTERIORUL CILINDRULUI LA MOTOARE DIESEL

REZUMAT

Obiectivul acestei lucrări a fost stabilirea influenței mișcării de squish din interiorul cilindrului asupra turbulențelor din interiorul cilindrului unui motor cu aprindere prin comprimare.

Simulările au fost făcute cu ajutorul softului AVL FIRE ESE DIESEL, pentru a simplifica simularea și timpul de simulare (comparativ cu AVL FIRE). Motorul ales este Mercedes-Benz E220 tip OM611 220 CDI, 92 kW, deoarece pistonul poate fi secționat și analizat pentru a putea introduce date cât mai precise în simulare.

Pentru a studia influența mișcării de squish s-a modificat distanța dintre capul

pistonului și chilasă (standard fiind 1 mm, și modificări la 0,5 mm, 1,5 mm și 2 mm) dar raportul de comprimare a fost menținut constant pentru a nu modifica influența asupra performanțelor motorului.

Prin modificarea mișcării de squish, se schimbă viteza de curgere a aerului și implicit turbulențele induse în camera de ardere, de aceea performanțele motorului deferă. Studiile viitoare vor analiza și influența mișcării de squish asupra emisiilor poluante ale aceluiași motor.

Keywords: Squish motion, intake, CFD Simulation, AVL FIRE, AVL ESE DIESEL.

1. INTRODUCTION

Currently, the study of the in-cylinder movement of the air during the intake process for the Diesel engine has been done (a major concern) because the quality of the burning process and therefore the emission quantities can also be changed through a simple modifications of shapes for the piston and intake pipes (manifold), which

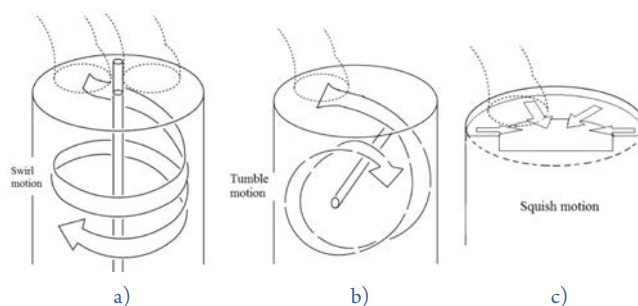


Fig. 1. In-cylinder motion of the air: a) Swirl, b) Tumble and c) Squish

is why many researchers have emerged in this direction [1][2][3]. The influence of the piston bowl was demonstrated by Jovanovic et al. [4][5] and also many CFD studies were made in this direction [6].

From a simplified point-of-view, there are two types of ideal flow patterns in an engine cylinder: *swirl motion* (with the cylinder axis as the axis of rotation and the flow entering tangentially through the intake ports) and *tumble motion* (orthogonal to the cylinder axis, the axis of motion moves as the cylinder expands and stays halfway between the top cylinder wall and the cylinder head at the bottom). Both are rotational motions, however, the axis of rotation is different in each case. Depending on the type of engine, one of these patterns is considered optimal because it maximizes the mixture of injected fuel and air, resulting in a homogeneous combustion.

2. IN-CYLINDER MOVEMENTS OF THE AIR FOR THE DIESEL ENGINE

2.1. Swirl and Tumble Movement

The flow of air admitted into the engine cylinders is usually characterized by a swirl movement, represented in figure 1.a., the tumble movement, represented in figure 1.b., and turbulence intensity. Swirl and tumble are widely known controlled turbulences that may exist within the cylinder, either separately or combined. These movements are created during the intake stroke, the piston moves towards BDC and the fluid passes over the inlet valves. Depending on the design of the inlet movements, swirl and / or tumble are created and preserved (or slowly dissipated) into the cylinder when the intake valves are closed. Swirl motion lasts longer than tumble and can affect the post-oxidation process [7].

The swirl number is the angular velocity of swirl motion around the central axis of the cylinder, while the angular velocity of the tumble movement is perpendicular to the axis of the cylinder.

Where there are both these types of movements, they combine to create a single large vortex. Normally, the swirl motion is used in direct injection Diesel engines, while tumble is used in spark ignition engines.

During the engine cycle (the time in which the swirl motion is created and the valves are closed) the swirl movement changes between compression and combustion. At the end of the compression stroke, the rotation of the swirl motion is larger and the flow of air is forced into the combustion piston bowl. The radius of the swirl motion is reduced, increasing the angular velocity. When the piston moves toward BDC, the opposite is true. The flow also slows down due to friction with the combustion chamber walls [7].

2.2. Squish Movement

The region where the squish motion occurs, represented in figure 1.c., is located on the outer radius of the flat area of the piston. This area is between the piston and the cylinder head, designed to be of approximately 1 mm, which is important for a Diesel engine. When the piston reaches TDC (Top Dead Center) the squish movement also contributes to guide the fuel that is injected. For a given compression ratio (higher compression ratio offers higher efficiency at a certain level), the heat transfer at TDC is relatively high and by reducing the surface, the heat transfer is also diminished (the surface, the heat transfer, ...).

Air trapped in the volume between the piston and the cylinder head is forced toward the center of the cylinder and creates a powerful flow. This flow (squish) can slightly influence combustion process timing after fuel injection.



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3. MATERIALS AND METHOD

3.1. Simulation Software

The software that was used to simulate the combustion process is AVL FIRE ESE DIESEL.

AVL FIRE is a multifunctional software (the latest generation 3D CFD -Computational Fluid Dynamics). It is developed and continuously improved to solve the most demanding problems in terms of geometric complexity, physical and chemical. CFD is a branch of fluid mechanics which uses numerical methods and algorithms to solve and analyze problems that involve fluid flows.

In order to simulate the 3D flow, mixture formation, combustion and pollutant formation, the CFD modeling engine with direct injection is used. It also makes it possible to analyze the interaction between fuel and air movement admitted into the combustion chamber. The main advantage of modeling is that it minimizes the CFD simulation time, specifically the numerous studies executed in a relatively short time, based on computing power. The main interest in flow visualization is admitted cylinders extraction and analysis of virtual motion swirl and tumble motion [8].

Generally the level of information that can be provided by a visualization technique increases with the size of the input data.

3.2. Simulated Engine

The simulated engine is the OM611 220 CDI, 92 kW (Mercedes-Benz E220). This engine was chosen because the engine is available for further practical analysis.

The simulations were made for different distances between the piston and the cylinder head at TDC, keeping the same compression ratio by slightly changing the profile of the piston bowl.

The steps that followed in AVL FIRE ESE DIESEL for a single simulation were:

- the General Data for the engine were introduced (engine layout, number of cylinders, compression ratio, crank radius, connecting rod length and piston offset);
- the piston parameters were introduced from an actual section of a piston;
- the injector properties were introduced;
- the mesh was generated (2D and 3D) and checked for irregular faces;
- the data for calculation were introduced (crank angle for the start and end of the simulation, engine speed, boundary conditions for the piston, liner and axis);
- fluid properties and initial conditions were introduced;
- the solver control parameter tree was completed (k-zeta-f turbulence model, energy output and no two-stage-pressure correction) and the maximum number of iterations was set to 100;
- output control was set to ensure proper 2D and 3D results;
- combustion model was set to Coherent Flame Model - ECFM-3Z;
- the spray was set, taking into account the fuel consumption at full throttle for the given engine speed;
- the simulation was started and the results were exported using Report Generator.

The first simulation was made for a 0.5 mm distance between the piston and the cylinder head, where a maximum velocity of the intake air was 86.778 m/s at 714 deg CA. The simulation can be seen in figure 2.

The second simulation was made for a 1 mm distance between the piston and the cylinder head, where the maximum velocity of the air was obtained at 710 deg CA (56.949 [m/s]), as shown in figure 3.

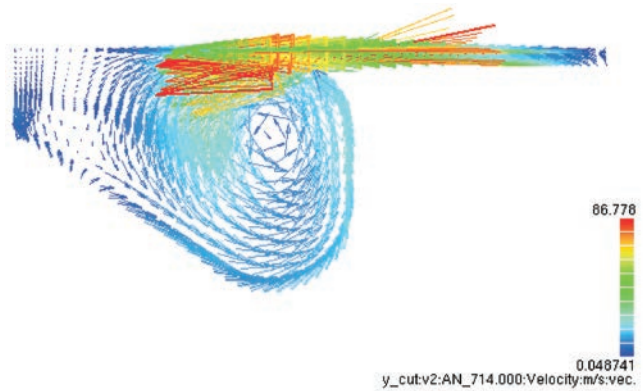


Fig. 2. Maximum velocity of the air for 0.5 mm gap at 714 deg CA

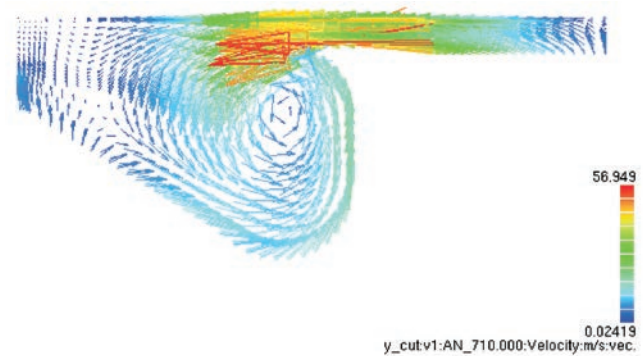


Fig. 3. Maximum velocity of the air for 1 mm gap at 710 deg CA

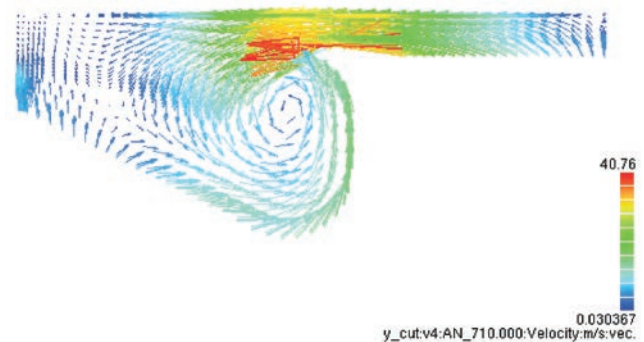


Fig. 4. Maximum velocity of the air for 1.5 mm gap at 710 deg CA

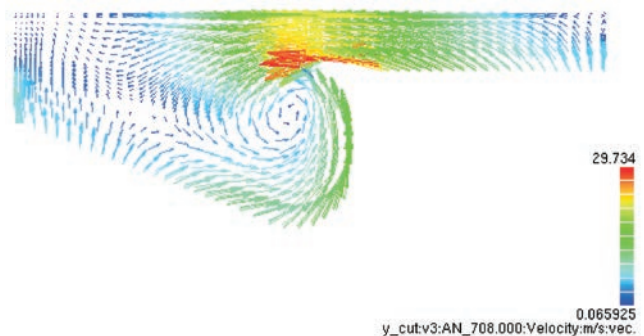


Fig. 5. Maximum velocity of the air for 2 mm gap at 708 deg CA

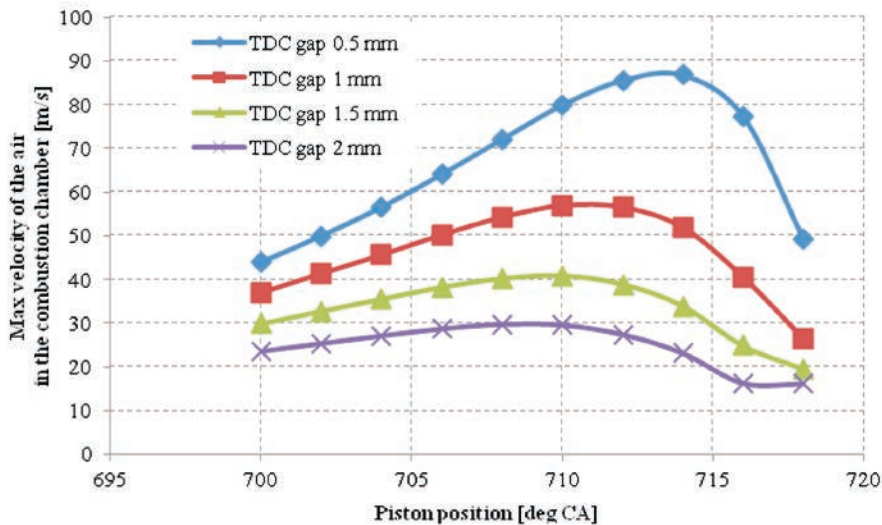


Fig. 6. Maximum velocity of the air in the combustion chamber for all simulation cases

The third simulation was made for a 1.5 mm distance between the piston and the cylinder head, and the forth simulation for a 2 mm distance. The simulations show a velocity of 40.76 [m/s] at 710 deg CA, and 29.734 [m/s] at 708 deg CA respectively (figures 4 and 5).

4. RESULTS

The results of the simulations are presented in table 1 and figure 6.

In table 1, the maximum values for each simulation were outlined with a green background.

The effective power was also monitored and the values are presented in table 2.

5. CONCLUSION

When trying to modify the squish motion of the air, all velocities and turbulences change inside the combustion chamber, therefore the performance of the engine differs.

Knowing the maximum velocity of the air inside the combustion chamber, the effective power of the engine was also monitored to see consequent modifications. The maximum power of the engine was obtained at 1

mm gap. For a 1.5 mm gap the power was lower with 4.17% and for a 2 mm gap the power was lower with 17.16%. For a gap of 0.5 mm, the expected tendency of the power would be a small rise, but the simulation has shown a lower power with 4.26%, therefore the recommended value for the TDC gap is 1 mm.

Future studies will be made on the same engine to see the influence of the TDC gap and implicitly the influence of the squish on pollutant emissions.

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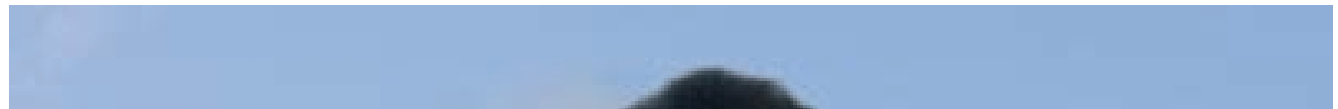
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Table 1. Values for maximum velocity of the air inside the combustion chamber, in all four simulated cases

Piston position [deg CA]	700	702	704	706	708	710	712	714	716	718
TDC gap 0.5 mm	43,8 3	49,8 9	56,4 6	64,0 5	72,0 7	79,9 3	85,5 9	86,7 7	77,3 3	49,2 1
TDC gap 1 mm	37,0 5	41,4 1	45,6 8	50,2 5	54,2 6	56,9 4	56,5 7	51,8 9	40,6 1	26,4 8
TDC gap 1.5 mm	29,8 3	32,7 2	35,5	38,1 7	40,1 6	40,7 6	38,8 6	33,8 8	24,8 6	19,5 2
TDC gap 2 mm	23,5 8	25,3 8	27,1 1	28,7 1	29,7 3	29,5 4	27,4	23,1 5	16,3 3	16,1 9

Table 2. Values for effective power after the simulations with different gaps



STUDIUL FUNCȚIONĂRII MOTOARELOR AUTOVEHICULELOR PE BAZA ANALIZEI DATELOR EXPERIMENTALE

THE STUDY OF ENGINES FUNCTIONING BASED ON EXPERIMENTAL DATA

REZUMAT

Lucrarea evidențiază cele trei aspecte importante și inseparabile ale abordării sistematice a funcționării motoarelor și a dinamicii autovehiculelor: luarea în considerare a interacțiunii om-vehicul-teren, tratarea deplasării cu algoritmi specifici teoriei sistemelor și analiza datelor experimentale cu algoritmi specifici teoriei semnalelor. În cadrul lucrării, abordarea sistemică a funcționării motoarelor și a dinamicii autovehiculelor se bazează pe datele experimentale obținute la încercări, cu ajutorul cărora se analizează mișcarea și se obțin modele matematice ale deplasării prin algoritmi de identificare a sistemelor. De asemenea, sunt redate principalele metode de analiză a datelor experimentale, care apelează la teoria probabilității, teoria informației, analiza de corelație și analiza dispersională; în plus, sunt evidențiate

posibilitățile oferite de analiza în timp, în frecvență și în timp-frecvență a datelor. Algoritmii de identificare și procedurile de analiză evidențiate asigură studiul dinamicității și economicității autovehiculelor fie folosind direct datele experimentale, fie utilizând modelele matematice stabilite pe baza acestora și aplicând conceptele și algoritmi specifici teoriei sistemelor. Datele experimentale au fost obținute prin încercările unui automobil cu control electronic al funcționării și folosind aparatură de achiziție și stocare a datelor furnizate de calculatorul de bord și preluate de la tractoarele incorporate din fabricație.

Keywords: electronic control systems, main statistical characteristics, time-frequency analysis, vehicle engine.



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1. INTRODUCTION

Up to present, the study of engine functioning has generated theoretical and experimental studies based on the knowledge of researchers and the practical possibilities for investigation at their disposal; moreover, the approaches were based on the technical level of the engine itself (classic or electronic control systems). The theoretical developments in the various disciplines, the entry of investigative equipments becoming more and more efficient and the equipment of the engines with electronic control systems (on-board computer, embedded sensors and actuators) are the main factors which have influenced the techniques of the theoretical and experimental study of engines functioning [1][2][3][4][8][9][10].

2. EXPERIMENTAL RESEARCH

To register the functional parameters, during the experimental researches one can use the diagnostic testers, some of them shown in Fig. 1: Opel (Fig. 1a), Daewoo family (Fig. 1b), Dacia Logan (Fig. 1c) and Skoda, VW, Audi, SEAT (Fig. 1d).

In consequence, the study of engines functioning based on experimental data presents at this point the following specific characteristics: it is a systemic approach, taking into account the action of the driver; it uses concepts and algorithms that have not been used in literature, but that belong to the system dynamics (neural networks, genetic algorithms, fuzzy, neuro-fuzzy algorithms, fractals, extremal analysis, symbolic analysis, tensor analysis etc.); it applies all three techniques to the processing of the experimental data (through the time analysis, frequency analysis



Fig. 1. Testers for the acquisition and storage of data from on-board computer

and time-frequency analysis); it uses new algorithms and methods for the study of engine's dynamics and fuel saving; it approaches unitarily the whole operation by three inseparable components: identification (the establishing of the mathematical models based on experimental data), control and diagnosis [3][4][5].

Fig. 2 and Fig. 3 presents some results acquired from testing the Skoda Octavia car which was equipped with on-board computer and used both petrol and LPG (liquefied petroleum gas).

The graphs on the left side of Fig. 3 show the values of engine speed n and the graphs on the right side show the values of the throttle shutters' position x , the last marking the engine load. The graphs from Fig. 3b and Fig. 3d show that during the experiments, when it performed normal movement, the engine operated at partial loads, so not the full load $x = 100\%$.

Fig. 4 highlights some functional dependencies between the throttle shutters' position x and the engine torque M_e , with Skoda Octavia using petrol in Fig. 4a and with Skoda Octavia using LPG in Fig. 4b.

Based on the experimental values, one can calculate the main statistical characteristics of the first order that are frequently used in literature, as

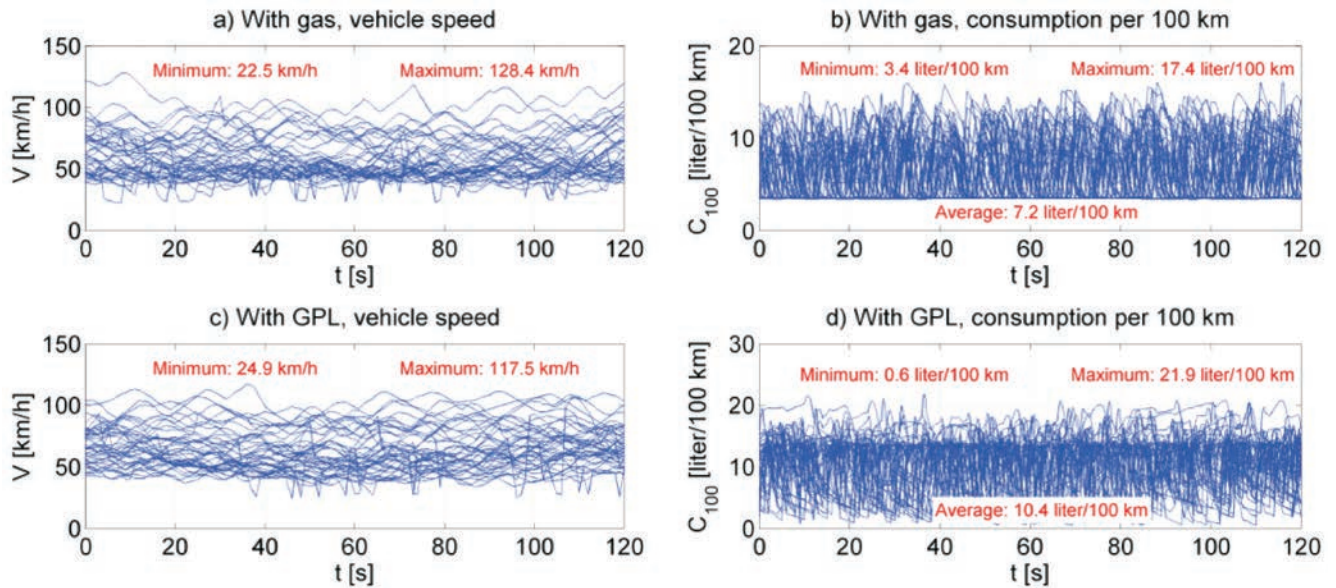


Fig. 2. Speed and fuel consumption per 100 km, 40 samples with gas and 40 samples with LPG, Skoda Octavia car

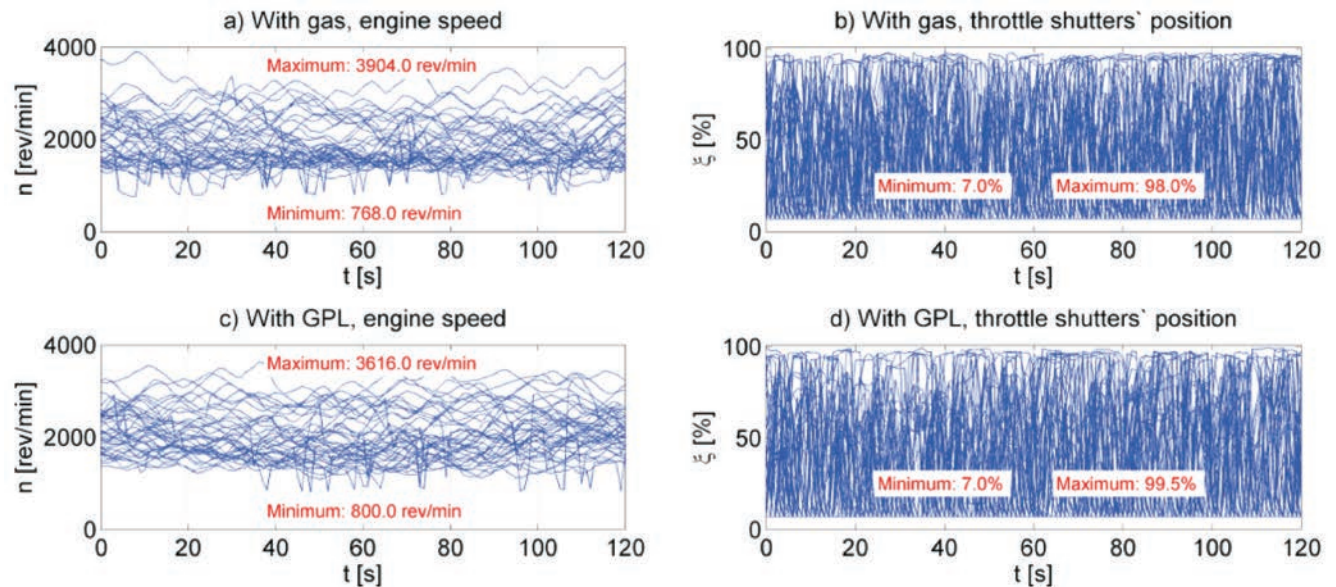


Fig. 3. Engine speed and throttle shutter's position, 40 samples with gas and 40 samples with LPG, Skoda Octavia car

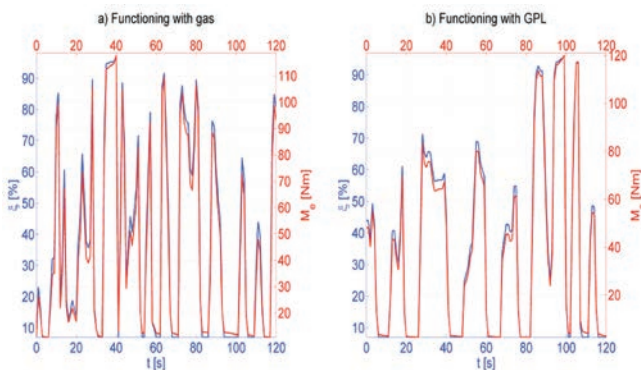


Fig. 4. Throttle shutter's position and engine torque, SB12 sample with gas and SG12 sample with LPG, Skoda Octavia car

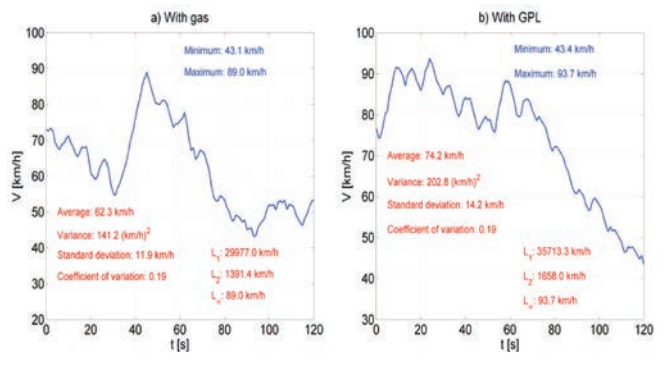


Fig. 5. Main statistical characteristics of the first order of speed

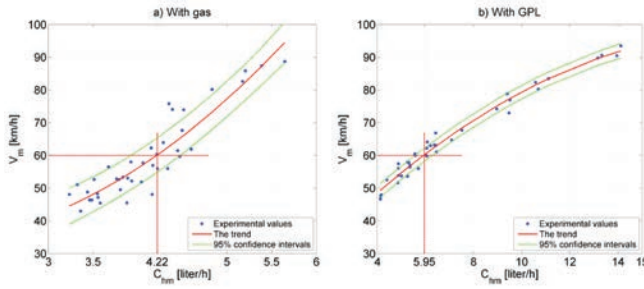


Fig. 6. The estimation of the average consumption to achieve a required average speed, 40 samples with gas and 40 samples with LPG

shown in Fig. 5 for the speed belonging to a sample with petrol (Fig. 5a) and a sample with LPG (Fig. 5b) of a Skoda Octavia car.

The graphs in Fig. 5 shows, in addition to known parameters (average, variance, standard deviation, coefficient of variation etc.) others used in statistics and electronic control, such as norms of measurements; in electronic control there are being used the H_2 algorithm (which uses rule 2) and the H_∞ algorithm (which uses infinite norm). Also, on the basis of the experimental data there can be estimated the fuel consumption required to achieve a required average speed, as shown in Fig. 6. Indeed, for example, from Fig. 6 it results that in order to achieve an average speed of $V_m = 60$ km/h there is required a mean fuel consumption $C_{100m} = 4.22$ liters/100 km of gas, or $C_{100m} = 5.95$ liters/100 km of LPG, meaning a 41% increase. In the graphs of Fig. 6 there are shown the experimental values, the trends in the targeted addition and the 95% confidence intervals.

Fig. 7 shows the densities of probability of the engine's torque M_e for eight experimental samples, of which four are with gas (upper graphs) and four are with LPG (lower graphs). These graphs show that the torque is not subject to the law of Gauss for normal distribution, the curves are not symmetrical to their maximum and present more than one peak.

Processing all the experimental data obtained from tests has led to the

conclusion that no functional size is subject to the distribution laws, frequently used in classical statistics (Gauss, Weibull, exponential, logarithmic etc.), which requires stochastic analysis of engine functioning, for example by using the bootstrap algorithm [5].

Also, processing the experimental data has shown that dependencies between sizes are emphasized in a non-linear manner; this aspect can be confirmed by applying the correlation analysis, the coherent analysis and the bispectral frequency analysis [4][6][7][8].

For example, the bispectral frequency analysis, which uses the rank III cumulant, establishes the bispectral of any discrete dynamic series $y[n]$:

$$S_{3y}(v_1, v_2) = \sum_k \sum_r C_{3y}(k, r) e^{-j2\pi v_1 k} e^{-j2\pi v_2 r} \quad (1)$$

in which $k \in (-\infty, \infty)$, $r \in (-\infty, \infty)$.

The rank III cumulant is determined with the ratio:

$$C_{3y}(k, r) = M \left\{ y^*[n] y[n+k] y[n+r] \right\} \quad (2)$$

which represent an extension of the classical autocorrelation function; the sign "*" is the complex-conjugate of the discrete dynamic series $y[n]$:

In formula (1), the frequency bandwidth v_1 is assigned to the linear constituent and v_2 is the frequency bandwidth assigned to the non-linear constituent. For example, Fig. 8 shows the results of the engine power bispectral analysis from one experimental sample; because the images from Fig. 8b, Fig. 8c and Fig. 8d are not empty, the engine power dynamic series also contain a non-linear constituent.

The most important consequence of this aspect is that the engine's functioning has to be described by non-linear mathematical models.

The unsteady character of the engine's functioning confirms itself through a time-frequency analysis of experimental data [4][10]. The time-frequency analysis uses different Cohen transforms, wavelet transforms, Stockwell transforms etc.

Therefore, for example, the Stockwell transform which represents an

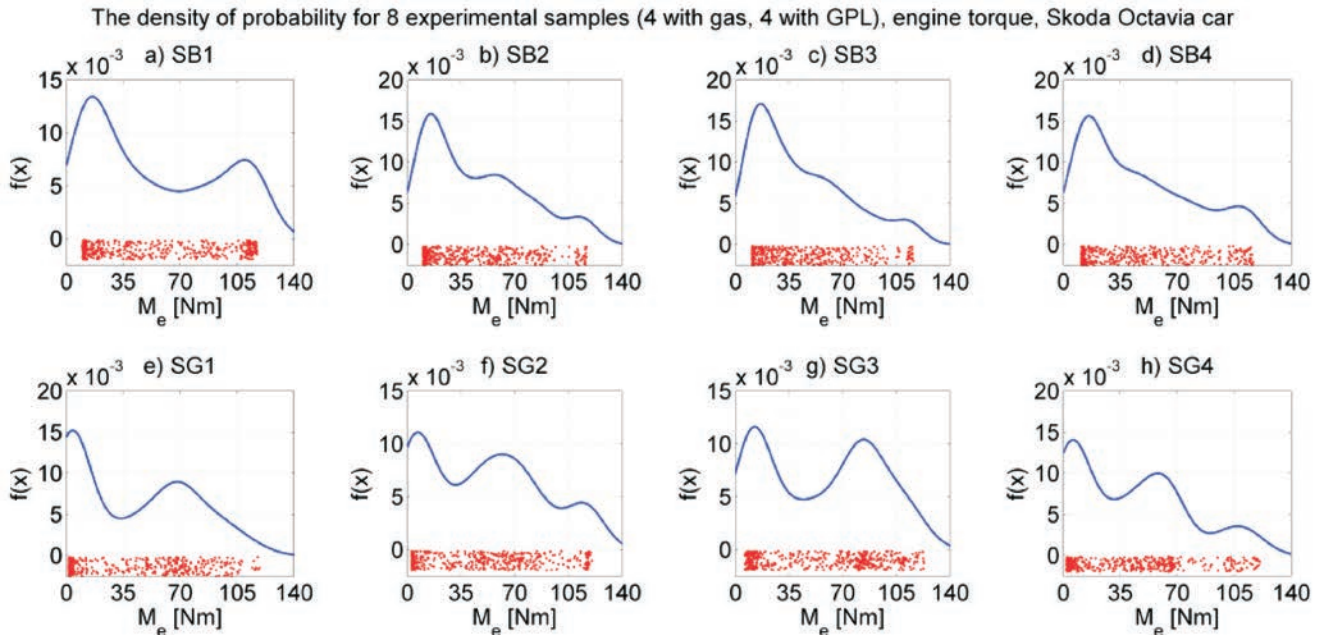


Fig. 7. The estimation of density of probability for 8 experimental samples, the engine torque

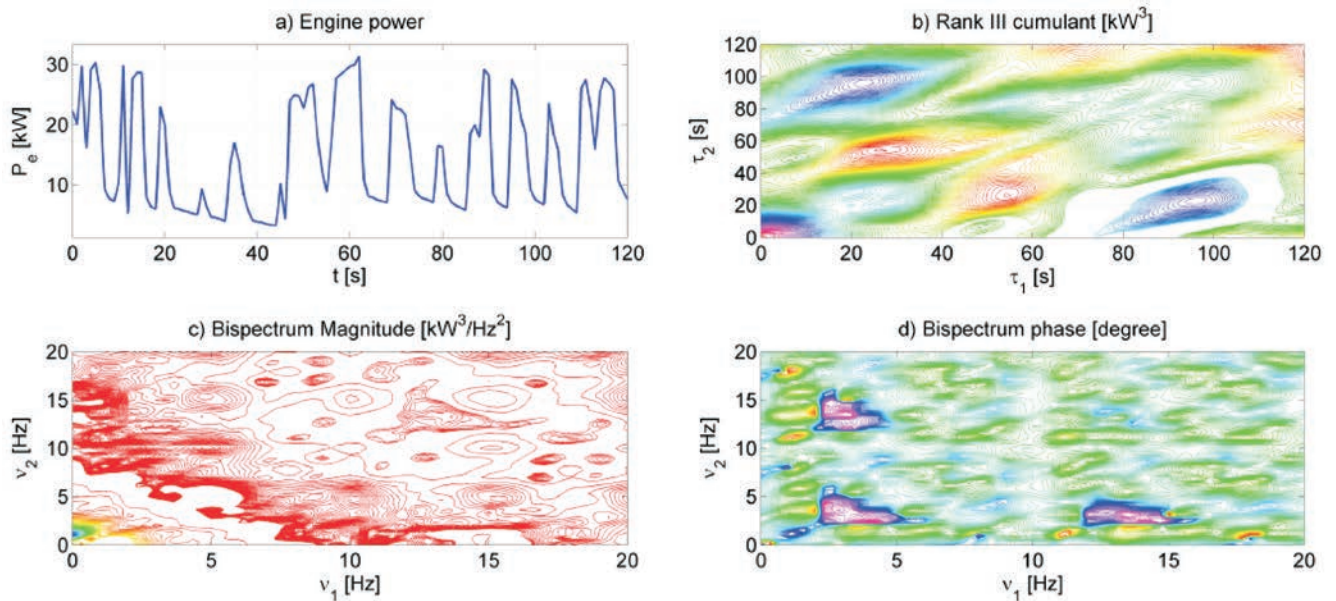


Fig.8. The bispectral analysis of the engine power, SB23 sample with gas

extension of the wavelet transform, through a phase correlation, is defined by the ratio [4][10]:

$$Y(\tau, \nu) = \int_{-\infty}^{\infty} y(t) \frac{|v|}{\sqrt{2\pi}} e^{-\frac{(t-\tau)^2 v^2}{2}} e^{-j2\pi \nu t} dt \quad (3)$$

in which the expression that marks the wavelet extension is $(t, \tau - \text{time}, \nu - \text{frequency})$:

$$\varphi(\tau, \nu) = \frac{|v|}{\sqrt{2\pi}} e^{-\frac{(t-\tau)^2 v^2}{2}} e^{-j2\pi \nu t} \quad (4)$$

Fig. 9a presents the analysis in classical frequency, which uses the Fourier transform (the graph presents the relative amplitude, that is the current value divided by the maximum value of the amplitude), and Fig. 9b presents the time-frequency analysis using the Stockwell transform.

As can be seen from Fig. 9b, only the time-frequency analysis shows the time layout of the harmonic constituent with high energy application.

The graphs from Fig. 9 show two frequency values (2.4 Hz and 5 Hz) at which the Fourier amplitude has two peaks; Fig. 9c shows the instantaneous values for the amplitude.

3. CONCLUSION

The study of engines functioning based on experimental data implies using all three techniques to the processing of the experimental data (through the time analysis, frequency analysis and time-frequency analysis) and also using new algorithms and methods for the study of engine's dynamics and fuel saving.

On the basis of the experimental data there can be estimated, for example, the fuel consumption required to achieve a required average speed etc.

Also, based on experimental data we can establish mathematical models of engine functioning, including the analytical expressions of their static characteristics [4][9].

In conclusion, the most important aspect is that the engines functioning has to be described by mathematical models with unsteady time coefficients, for example by using neural networks or neuro-fuzzy algorithms.

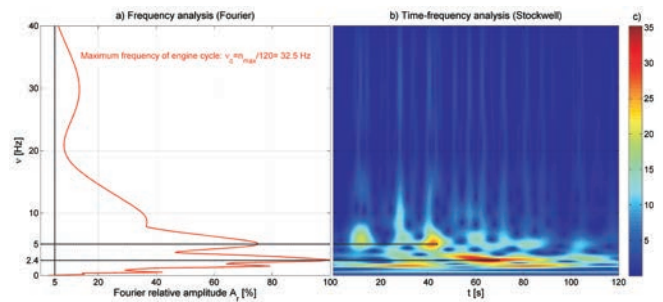


Fig. 9. The spectral analysis (in frequency and time-frequency) of the engine torque

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A DEVELOPMENT OF THE VALIDATION SYSTEM OF INDIRECT TPMS

STUDIU CU PRIVIRE LA VALIDAREA SISTEMULUI DE MONITORIZARE INDIRECTĂ A PRESIUNII ÎN PNEU

REZUMAT

Sistemul de monitorizare indirectă a presiunii anvelopei (ITPMS) folosește software-ul care este încorporat în controlul electronic al stabilității autovehiculului (ESC). Spre deosebire de TPMS-ul direct, această soluție constructivă poate contribui la reducerea costurilor și a masei autovehiculului pentru că nu include senzori și conexiunile aferente. Pentru a determina presiunea joasă a anvelopei ITPMS folosește o analiză a razei de rulare și o analiză în frecvență. Analiza în frecvență permite monitorizarea variațiilor de presiune pentru toate roțile autovehiculului, simpla analiză a variației razei de rulare fiind insuficientă.

Din cauza caracteristicilor tehnice descrise mai sus, ITPMS poate determina presiunea joasă numai atunci când autovehiculul este în mișcare și apare o influență importantă asupra anvelopei și a condițiilor de rulare. Modelarea neliniară a pneului și a condițiilor de rulare este dificilă, ceea ce face ca și simularea pe bancul de încercări a condițiilor de rulare pentru un autoșasiu elementar să fie, de asemenea, greoie. Lucrarea prezintă un sistem de verificare conceput pentru evaluarea eficienței a procesului de dezvoltare a ITPMS pentru un șasiu dat.

Keywords: Indirect tire pressure monitoring system, MILS, simulator, test data management

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1. INTRODUCTION

By recall cases of Firestone tires and Ford, the US government mandated that ensure the stability of the tire with a tire pressure monitoring system (TPMS) in the car [1]. TPMS legislation of the US government made TPMS legislation of countries such as EU, Korea

that vehicles are equipped with TPMS mandatorily [4]. TPMS technique is divided into direct method that uses a sensor and indirect method that does not use a sensor. Since it is getting competitive in the automotive market, OEM want to lower the cost of a vehicle equipped with indirect TPMS on vehicles of less than C segment [5]. Hyundai Autron, Hyundai Mobis and Hyundai Motors did a mass product by developing ITPMS. This paper introduces the equipment and validation process to establish effective verification of ITPMS in the process of developing indirect TPMS and to explain the effect.

2. RELATED WORK

2.1. Indirect TPMS

ITPMS is only software that is embedded in ESC for monitoring the air pressure of the tire without any machine or electronic components. The car mounted with ESC has a wheel speed sensor to measure the wheel speed of each wheel. ITPMS checks the air pressure of the tire using

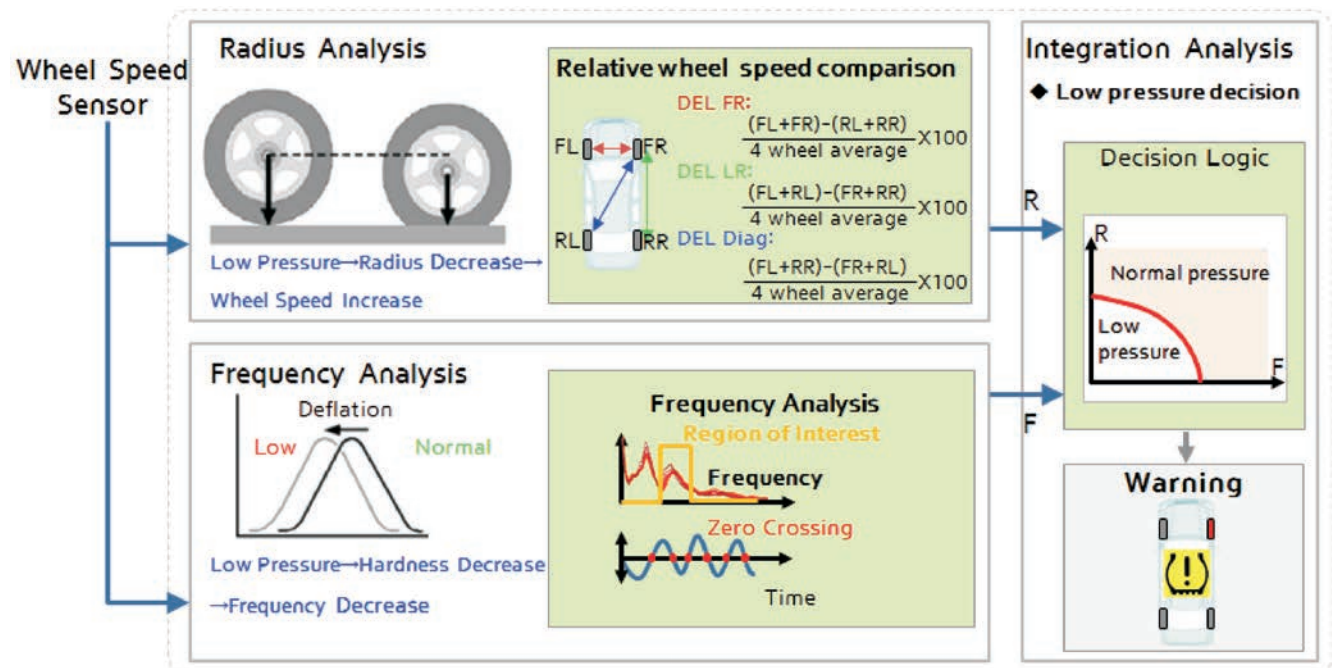


Fig. 1. ITPMS Main Concept

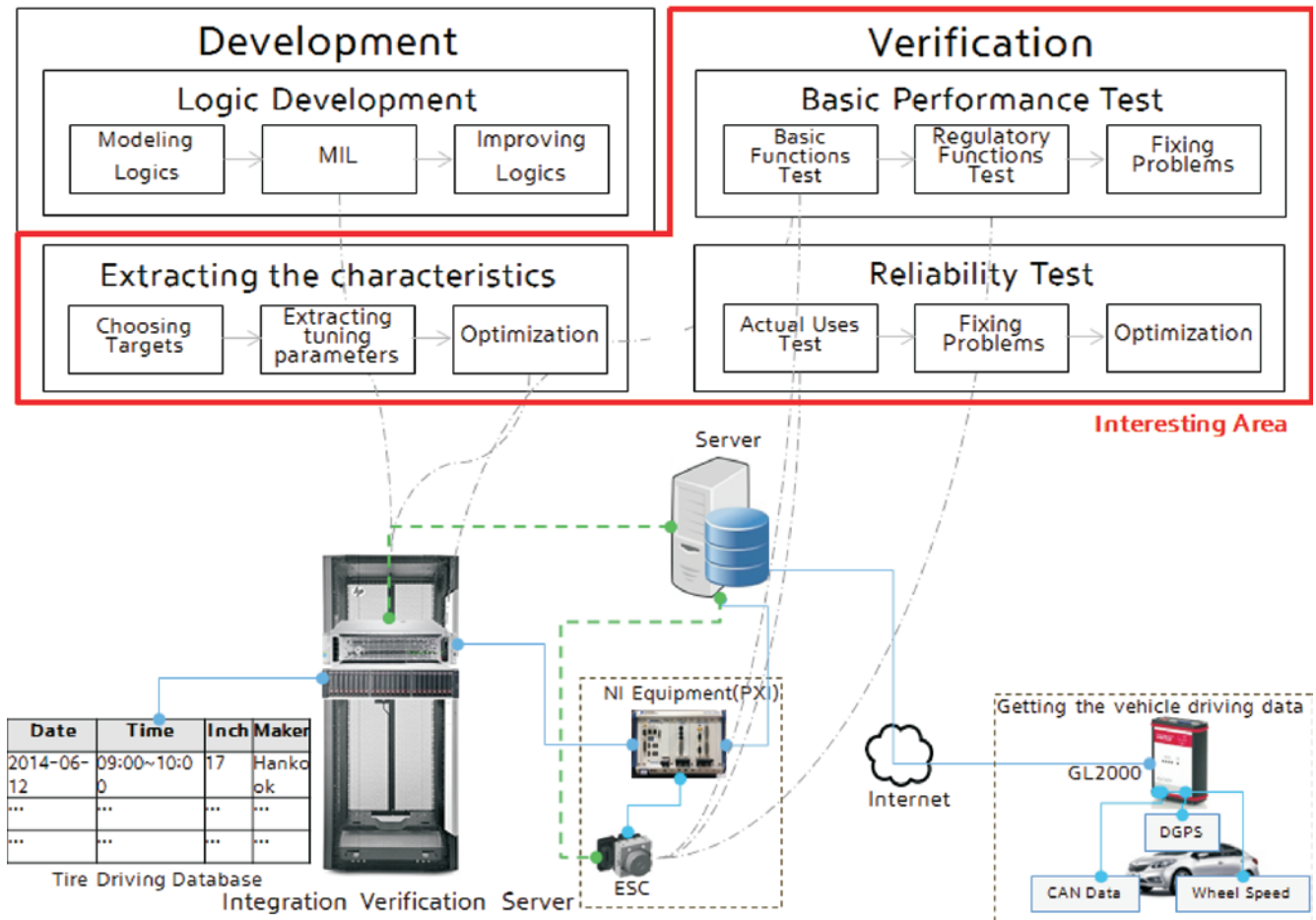


Fig. 2. ITPMS Development & Verification Process

wheel speed sensors value that is the input data of the radius analysis and frequency analysis [6]. In case of the radius analysis, the relative change is measured in the radius of the running of the tire according to the change in pressure. Thus, in the case of 1-3 low pressure of the tire, which is accompanied by the wheel difference, it is possible to detect a low pressure. There are two kinds of methods for the radius analysis such as using the difference in wheel speed [7], creating a dynamic model of the vehicle with the adaptive filters [8]. On the other hand the case of the low pressure four wheels, all four wheels cannot be detected by the radius analysis that uses the relative changes. To overcome the disadvantages of this radius analysis method, we use a frequency analysis.

The frequency analysis determines whether the low pressure by detecting a frequency variation of the tire caused by the pressure difference. In other words, the frequency is lower than normal pressure balloon tire. Therefore, frequency analysis unlike the radius analysis can easily detect low pressure four wheels. Frequency analysis method is how to extract the frequency range of interest by using the fast Fourier transform (FFT) [9], making the tire model with adaptive filter [8] and zero crossing algorithms for counting whether to change the code after passing through the signal processing [10]. (Figure 1) shows the ITPMS main concept.

2.2. ITPMS evaluation equipment

ITPMS evaluation equipment is divided in two methods such as using the plant model of the vehicle and the tire, using the data acquired from the actual vehicle. The method using the plant model of the tire and the

vehicle has the advantage of being able to try a variety of driving environments for validation [11]. However, there is a disadvantage of less validity for vehicles and tires from the actual vehicle development stage. Therefore, in the actual vehicle development step it is required to be the operation for acquiring the verification data from the actual vehicle and the tire. But there are few related researches.

3. STRUCTURE OF THE PAPER

In Chapter 4 we brief overview about the process and equipment to verify the ITPMS. Chapters 5 is to learn the system for managing obtained data from the vehicle and in chapter 6, by using the acquired data, we look at the server-based verification and MILS environment for the tuning method. In chapter 7, by using the obtained data, we investigate the simulator to verify ITPMS on the actual hardware. Finally, chapter 8 shows a tool to support ITPMS verification process and the reliability test result.

4. ITPMS VERIFICATION PROCESS AND EQUIPMENT

4.1. Process

ITPMS is operated based on the vehicle driving data. Analyzing the test results or actual data need no distortion to the experiment, and also requires a lot of time. An effective development process is necessary in order to develop in a short time. We made a process and conducted developing according to the process (Figure 2). ITPMS is developed and validated in three stages:

- Extracting the characteristics: In the test for extracting the characteristic values of the tire, we extract the tuning parameter for all tires mounted on

a vehicle. The tuning parameter is used to calibrate the threshold of low pressure decision. For example, if the A vehicle can be equipped with the five kinds of tires, we measure the data from the vehicle through the test mode by mounting each tire in a vehicle to extract the characteristic of the tire. By using the measured data, the tuning parameter optimization based on the verification server to be described later is performed.

- **Basic Performance Test:** Once extraction is complete, to confirm the basic performance and regulatory performance a real vehicle test is conducted. Not to test all tires, we choose edge-tiers that mean having insensitive or sensitive characteristics and test them. The test can be performed for all tires mounted on the vehicle. However, because of many resources and time consuming, the test cases are made with respect to the peculiar tire characteristics. If a problem occurs in the basic performance testing, tuning operations carried out again.
- **Reliability test:** After the basic performance is secured, considering the variety of actual use in real road conditions, we conduct a test for reliability verification. If a problem occurs similarly in this case, we solve the problem through the tuning. It is difficult to carry out the test again when problems occur because of a lot of resources and time consumption. For this reason, we acquire the vehicle data from each test. If problems occur, using the acquired data without performing again the test is repeated by using the simulator described later and the verification server.

4.2. Equipment

ITPMS verification equipment is consisted of three modules. The first is a data acquisition device attached to the vehicle which transmits data of the vehicle to the server. The second is to manage the data collected by the data acquisition device as ITPMS verification server, and performs tuning and verification based on the data obtained and ITPMS model. Finally ITPMS simulator that uses the obtained data is performed for the verification on the ITPMS hardware.

5. THE DATA ACQUISITION DEVICE

ITPMS necessary data is a counter value of the wheel speed sensors, vehicle driving information (Brake signals, acceleration, engine torque, Yawrate etc.), global positioning system (GPS) information. We use vector GL2000 to obtain all of above information. GL2000 is equipment that can have CAN channel, GPS signal. The driving information of the vehicle is acquired through the CAN, and the wheel speed sensors of the vehicle are obtained at ESC controller via a high speed CAN. Data acquisition is completed and is transmitted to the ITPMS verification server using the long term evolution (LTE) modem.

6. ITPMS VERIFICATION SERVER

6.1. Data Management Environment

The data from the obtaining apparatus is collected by ITPMS verification server. The collected data with the test information which tester made is recorded into the database. CAN data which has driving information and the wheel counter value of the vehicle after the reading is completed, is respectively converted into a data format that can be performed in the data format that can be implemented with the simulator and the verification server.

6.2. Verification server environment

ITPMS verification server is made of the parallel computing environment to perform the tuning parameters and verify ITPMS functions. The work environment is consisted of MATLAB and Simulink of MathWorks. In case of a need to use all of the acquired data to perform a tuning or verification, because car data is so much larger data capacity, the efficient parallel processing is required to perform the tuning and verification. For

such parallel processing in hardware the verification server is composed of one server and four workstations that perform the verification practically and by using the parallel tool box of MathWorks performs parallel processing in software.

6.3. Tuning and verification

As it described in process, we extracts the necessary data by using the test data to extract the characteristic of the tire. The data to be extracted is a reference value for determining the low pressure and a corrected value that is used in the radius analysis. We obtain the respective data by each tire and set a representative value that can covers the tire to be mounted on the vehicle by using a statistical technique. In this way using the parameters and the acquired data, the basic performance and the reliability test is performed for verifying that the ITPMS accurately determines the low or normal condition. At this time, using the receiver operating characteristic (ROC) charts we track whether MISS ALARM and FALSE ALARM appear in ITPMS.

7. ITPMS SIMULATOR

7.1. Scope of the simulator

When verifying ITPMS with the 20 ~ 30 thousand km driving data, using the parallel computing takes about 3 hours. Therefore, even when the tuning parameters or ITPMS software is changed, by using the verification server, there is no great burden.

However, in order to verify ITPMS on the hardware, because the simulator has to enter the data in real-time, it cannot be verified for all the input data as ITPMS verification server. In case of using ITPMS simulator, we make test cases using a limited ITPMS data to verify the malicious running mode or fault diagnosis and verify them.

7.2. Configuration simulator



The ITPMS simulator (Figure 3) was configured using PCI extensions for instrumentation (PXI) of National Instrument (NI). It can perform a simulation with up to two ECUs. Using the digital board to simulate the wheel speed sensor input data, and uses the CAN card and a digital input/output card to simulate the CAN and IG ON / OFF, Brake signal. The simulator carries out in the fault diagnosis mode and the verification mode for verifying the general performance.

Fig. 3. ITPMS Simulator

8. CONTROL TOOL AND EXPERIMENT

We can proceed according to the simulator control, data management, test cases generation, validation result management by a tool to support the ITPMS development process.

In the data management mode, it transforms the raw data acquired in the vehicle to the data format for the test environment and stored in the database. Based on the stored data and generates a test case for the test purpose. Test cases written in the script form can dynamically have a variety of test items. Once the verification is complete, the report documents are automatically generated and the generated document is written

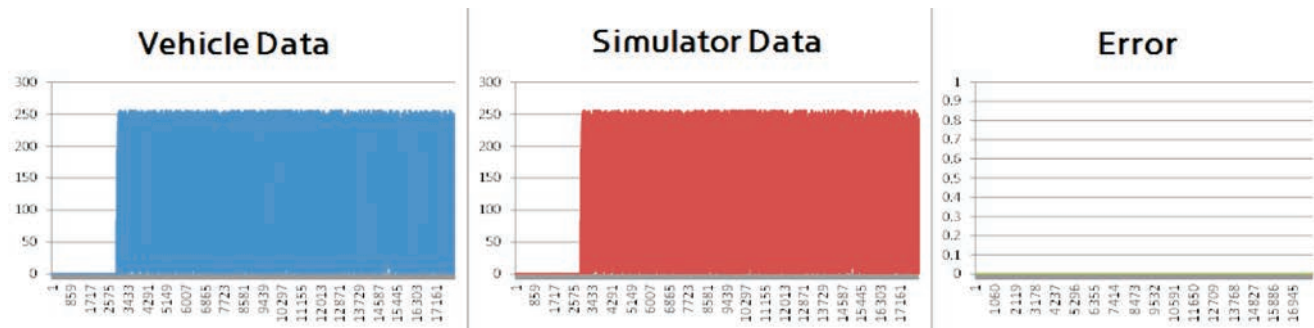


Table 1. Errors between a vehicle and a simulator

	Del FR	Del LR	Del Diag	Fre FL	Fre FR
Vehicle	0.36	0.36	0.19	36.5	44.16
Simulator	0.36508	0.35186	0.20016	36.4827	44.131
Error	0.00508	0.00814	0.01016	0.0173	0.029

Table 2. Errors of repeatability tests

	Del FR	Del LR	Del Diag	Fre FL	Fre FR
Average	0.36508	0.35186	0.20016	36.4827	44.131
Standard deviation	0.006356	0.002447	0.005914	0.021302	0.033516
Error(100%)	1.741197	0.695535	2.95513	0.05839	0.075947

to the database easier to manage history. Because of the difficulty of the same experiment in the vehicle, a simulator is extremely useful. However, if the results are different, the simulator cannot be used. We compared to the output of the vehicle and performed a test of repeatability in order to verify that the output values are same. Figure 4 shows that the wheel speed value of vehicle and simulator is same.

According to the test results, it was confirmed that the simulator outputs the same result. However, even though we control with the same clock, as a physical limitation between the simulator and the ECU, an error occurs in the portion where the pulse occurs at the same time. It was confirmed that the range of small errors that do not affect the result. The following is a data comparing the results of testing important factors for determining the low-pressure value.

Del FR, Del LR and Del Diag represent the difference of the left/right, front/rear and diagonal of each wheel. Fre FL/FR indicates the frequency value of the front wheel. By relative standard deviation (RSD) method, factor analysis results show the RSD values of Del FR, Del LR, Del Diag is less than 3%, RSD value of Fre FL/FR that needs for precise comparison is less than 1% (Table 2). It means that there is no problem as the performance evaluation simulator.

9. CONCLUSION

By the above construction of process and verification equipment, we can effectively do the ITPMS development and validation. If we received a simple change requests from OEM or tier1, it takes more than a month without building the verification process and equipment. We can reduce the development period to a week by the solution about the process and equipment.

Fig. 4. Comparing the wheel speed value of ITPMS simulator and a vehicle

Also by utilizing the existing accumulated data, the quality of ITPMS was a remarkably improved by performing verification for the modified software.

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EQUIPMENT AND METHODS USED FOR EVALUATION OF VEHICLE'S SYSTEMS VIBRATIONS ON ROUGH TERRAIN

ECHIPAMENTE ȘI METODE DESTINATE EVALUĂRII VIBRAȚIILOR AUTOVEHICULELOR LA DEPLASAREA ÎN TEREN ACCIDENTAT

ABSTRACT

Whatever the purpose of research in the automotive area, a decisive role in obtaining relevant and compelling data on research is given by the test equipment, the data acquisition, and methods chosen for their achievement. Mechanical vibrations, especially those arising upon reading this of trails highly rough, in which amplitudes are of very high or are perceived for long periods of time, causes the serious damage to the

human body: biological, mechanical and psychical. Obtaining accurate data concerning vibrations and shocks products in an area of rough terrain, as well as understanding thereof, it creates the possibility of establishing new technological approaches which to lead in reduce injuries caused by them on the human body.

Keywords: vibrations, data acquisition, equipment



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1. INTRODUCERE

Oricare ar fi scopul cercetării în domeniul automobilelor, un rol decisiv în obținerea de date relevante și convingătoare privind cercetarea este dată de echipamentul de testare, achiziția de date și metodele alese pentru realizarea acestora. Vibrații mecanice, în special cele care apar în urma parcurgerii de trasee extrem de dure în care amplitudinile sunt

foarte mari sau sunt percepute pentru perioade lungi de timp, provoacă deteriorarea gravă a organismului uman: biologic, mecanic și psihic.

Obținerea de date exacte cu privire la vibrațiile și șocurile produse într-o zonă de teren accidentat, precum și înțelegerea acestora, creează posibilitatea de a stabili noi abordări tehnologice care să conducă la reducerea leziunilor provocate de acestea asupra corpului uman.

Scopul avut în vedere la realizarea cercetării îl reprezintă obținerea unor date reale și concludente asupra valorilor vibrațiilor, care afectează factorul uman pe timpul deplasării cu viteză ridicată, într-un teren accidentat. Astfel că, în urma parcurgerii unor trasee accidentate, echipamentele trebuie să permită obținerea unor valori reale ale vibrațiilor existente, pentru masele suspendate și nesuspendate ale autovehiculului.

2. ECHIPAMENTE UTILIZATE PENTRU ACHIZIȚIA DATELOR DIN TEREN[3][4]

Echipamentul ales pentru cercetare a fost atent selectat, astfel încât să servească cât mai mult posibil domeniului de aplicare propus. De asemenea, montarea acestora de către autori s-a realizat în conformitate cu nevoile specifice cercetării, astfel încât să poată reda date despre conformația traseului urmat, a accelerațiilor induse de traseu, cât și a vibrațiilor transmise către scaun și conducătorul vehiculului.

În desfășurarea experimentului au fost utilizate următoarele echipamente, fiecare furnizând o serie de date, utile analizelor ulterioare:

* Pentru identificarea poziției autovehiculului în raport cu terenul și realizarea traseului urmat a fost utilizat un sistem tip SpeedBox produs de firma Race-Technology, compus dintr-o antenă de recepție GPS și un modul de procesare a datelor (fig.1).

* Pentru determinarea valorilor de dinamicitate ale vehiculului pe timpul



Fig. 1. Antena GPS



Fig. 2. Sistemul DL-10



Fig. 3. Amplasarea sistemului de senzori tri-axiali RLVB



Fig. 4. Senzori vibrații tip 4504 A



Fig. 5. Senzor vibrații tip 4447

deplasării a fost utilizate două dispozitive tip DL-10 (fig.2), care furnizează date PVT (Poziție, Viteză, Timp), montate în partea centrală a bordului, cât și în spatele primului rând de scaune.

* Pentru obținerea datelor referitoare la caracteristicile drumului: tangaj, ruli, girare a fost utilizat un sistem de achiziție a datelor compus dintr-o unitate centrală și două accelerometre pe trei axe tip RLVB IMU03, produse de firma Racelogic. Sistemul complet a fost montat și fixat pe suprafața pedalieră a autovehiculului, astfel încât să poată înregistra orice șoc sau denivelare din teren (fig.3).

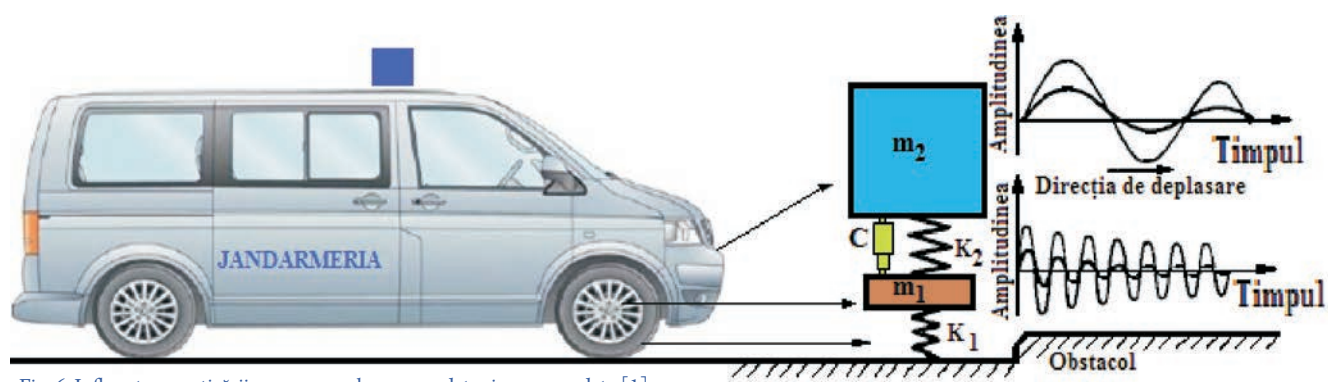


Fig. 6. Influența amortizării asupra maselor suspendate și nesuspendate [1]

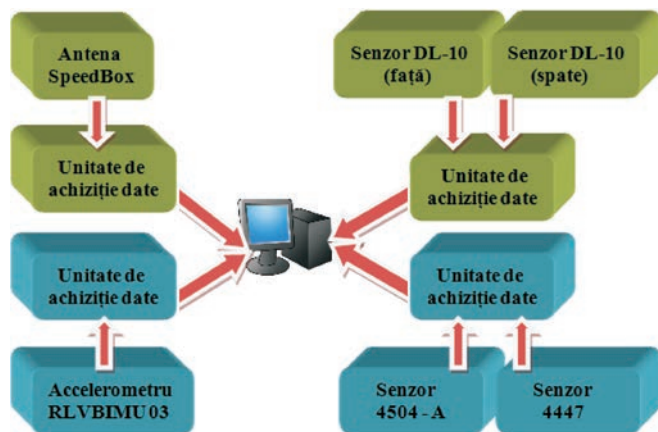


Fig. 7. Schema bloc de achiziție a datelor

* Pentru obținerea datelor legate de valorile vibrațiilor transmise către structura scaunului și către conducătorul de autovehicul s-au utilizat senzori tip 4504 A, montați pe structura cadru a scaunului și un senzor tip 4447 amplasat pe suprafața șezutului scaunului (fig.4 și 5).

3. IDENTIFICAREA DATELOR ACHIZIȚIONATE

În orice regim de deplasare, cât și pe orice categorie de drum, deplasarea autovehiculului este însoțită de apariția vibrațiilor și a șocurilor (fig.6).

Denivelările căii de rulare creează șocuri asupra roții, acestea fiind transformate în oscilații cu diferite frecvențe prin intermediul pneului și al suspensiei, fiind transmise ulterior către structura de rezistență a vehiculului și de aici către scaune și pasageri. Imaginea următoare (fig.7) prezintă schema bloc privind achiziția datelor despre terenul abordat, cât și despre valorile vibrațiilor recepționate adoptată de autori.

Datele recepționate de la antenă au permis identificarea corectă a traseului urmat pe timpul deplasării (fig.8), iar cele provenite de la accelerometrele tri-axiale au identificat caracteristicile terenului (tangaj, ruliul, girație) [5].

În imaginea următoare (fig.9) se pot observa variațiile de nivel ale terenului, pe toată lungimea traseului, recepționate cu ajutorul sistemului GPS.

În imaginea următoare (fig.10) pot fi observate valorile accelerațiilor verticale care se manifestă pe toată lungimea traseului.

Datele înregistrate de sistemul DL-10 sunt secvențe NMEA "GPRMC" scrise sub formă de text, conform standardului NMEA 0183.

Acest tip de secvențe conține un minim de date recomandate PVT (poziție, viteză, timp) pentru determinarea soluției pe baza semnalului GPS recepționat:

* DL-10 față:

– secvența de început

\$GPRMC,094053.400,A,4418.5023,N,02803.4489,E,0.01,81.94,130516,,,D*54-

– secvența de final

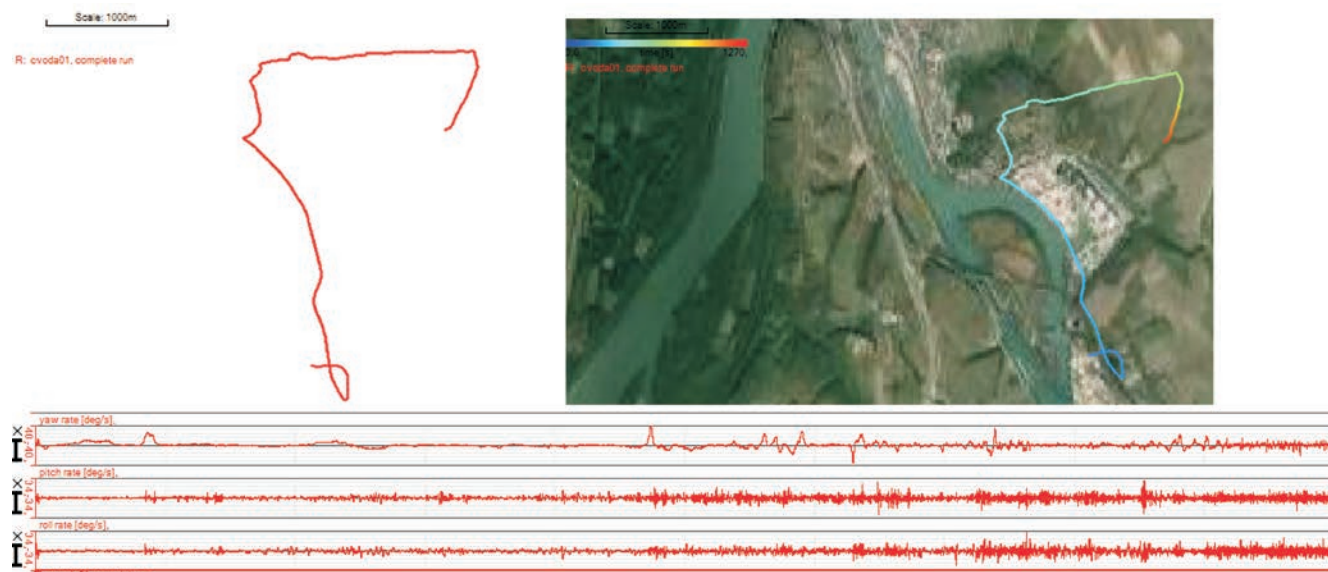


Fig. 8. Traseul analizat și variațiile caracteristice ale terenului

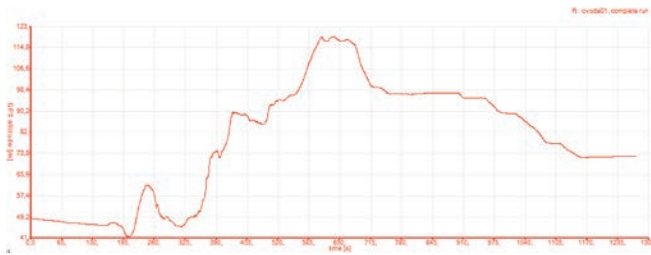


Fig. 9. Variațiile altitudinii terenului înregistrate prin intermediul GPS

\$GPRMC,095145.100,A,4420.0085,N,02804.0713,E,0.00,327.15,130516,,D*60

Explicație date înregistrate:

\$GPRMC = identificatorul 094053.400 / 095145.100 = ora de început / sfârșit a înregistrării: 09,40:53 / 09,51:45

A = semnal activ

4418.5023,N / 4420.0085,N = latitudinea: 44° și $18,5023^{\circ}$ N / 44° și $20,0085^{\circ}$ N

02803.4489,E / 02804.0713,E = longitudinea: 28° și $03,4489^{\circ}$ E / 28° și $04,0713^{\circ}$ E

0.01 / 0.00 = viteza față de sol în noduri = 0,018km/h / 0,00km/h

81.94 / 327.15 = direcția de deplasare în grade

130516 = data înregistrării: 13.05.2016

D*54- / D*60 = sumă de control

De asemenea, pe durata experimentului, pentru determinarea vibrațiilor existente la nivelul scaunului șoferului, cât și pentru aflarea valorilor primite de către conducătorul autovehiculului pe timpul deplasării în teren accidentat,

Datele înregistrate prin intermediul senzorului 4447 (Brüel & Kjær) sunt prezentate în imaginea următoare (fig.11).

4. CONCLUZII

Relevanța cercetării este dată de o serie de factori, cu importanță majoră pentru modalitățile de abordare a analizelor vibrațiilor primite de structura autovehiculului, cât și de personalul aferent, în timpul deplasărilor în teren accidentat.

Astfel, putem preciza că:

– utilizarea unor echipamente adecvate scopului propus, facilitează obținerea unor date precise și reale din teren;

– prin folosirea echipamentelor cu senzori GPS se pot obține corelații semnificative între diferitele echipamente utilizate pentru sistemul de achiziție a datelor;

– adoptarea unor metode originale de montare a echipamentelor de recepție a semnalelor la nivelul structurii autovehiculului, crează posibilitatea obținerii unor date bine direcționate scopului propus;

– corelarea unui număr mai mare de echipamente crează posibilitatea obținerii unor date cât mai precise asupra fenomenelor urmărite.

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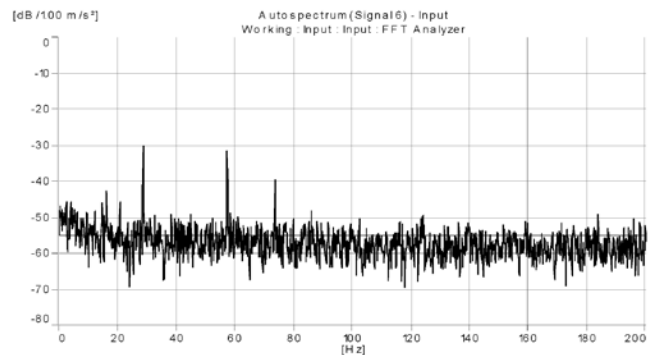


Fig. 11. Semnal măsurat pe verticală, pe scaunul șoferului

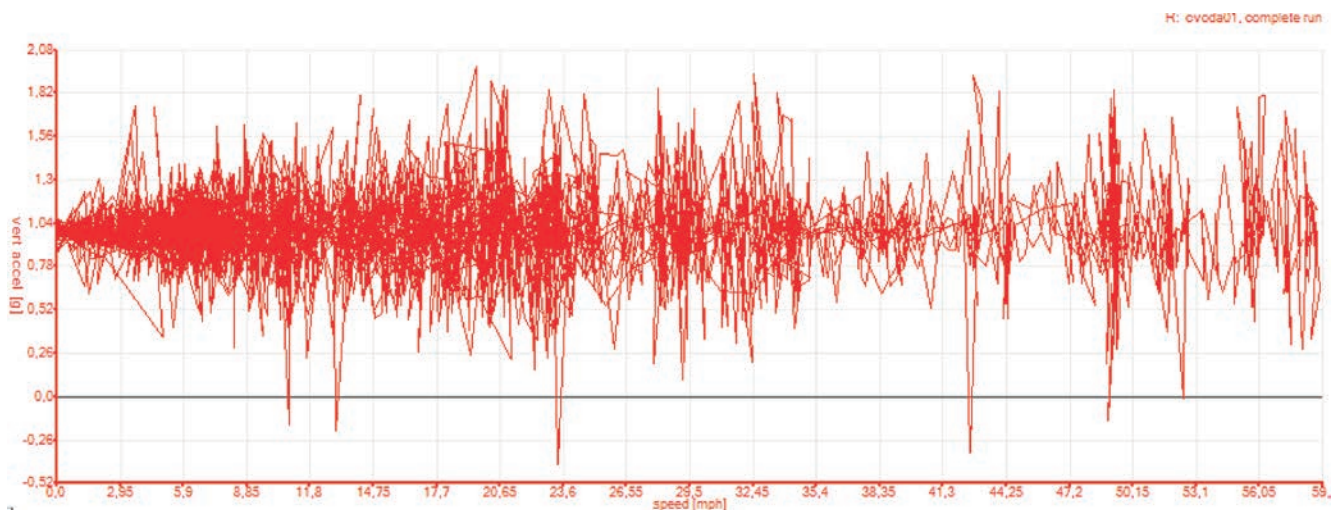


Fig. 10. Valorile accelerațiilor verticale în funcție de viteza autovehiculului

THE STUDY OF THE AERODYNAMIC BEHAVIOR OF AN ELECTRIC KART USING CAD AND CFD METHODS

STUDIUL COMPORTAMENTULUI AERODINAMIC AL UNUI KART ELECTRIC PRIN FOLOSIREA METODELOR CAD ȘI CFD

REZUMAT

Lucrarea prezintă aspecte ale modelării, simulării și rezultate folosind metode CAD (Computer Aided Design) și CFD (Computational Fluid Dynamics). Această simulare a implicat folosirea programelor Catia V5 și Ansys Fluent și a fost aplicată pe un kart electric dezvoltat în cadrul Centrului de Cercetare Ingineria Automobilului al Universității din Pitești.

Pornind de la kart-ul existent a fost construit un model geometric în Catia V5. Respectivul model a fost folosit pentru a simula comportamentul aerodinamic cu ajutorul programului Ansys Fluent.

Rezultatele simulării precum și concluziile sunt prezentate la finalul lucrării.

Keywords: electric kart; aerodynamics



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1. INTRODUCTION

In this paper it is presented a model for studying the aerodynamic behavior of an electric kart. In comparison with a petrol powered kart, the electric kart has batteries placed between the driver's seat and the side bumpers. For the studied case, presented in Fig. 1 and Fig. 2, due to their flat surface, the batteries could have a negative impact on the karts' aerodynamic performance. An electric kart was chosen for this study due to the

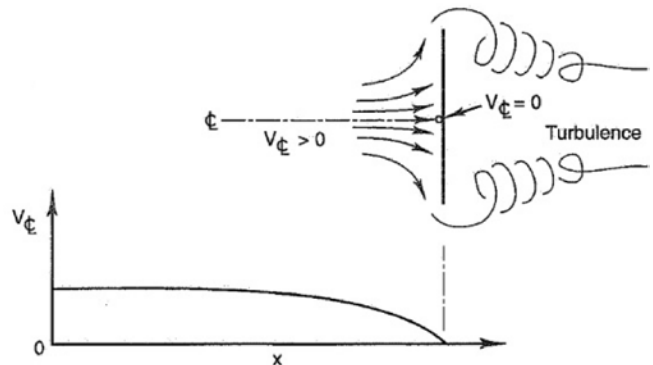


Fig. 1. Dynamic or "stop" pressure, "q" [1]

lack of air and sound pollution.

If a large flat plate is inserted into an airstream at right angles to the flow, then air particles moving in the column of air near the center would be decelerated to a stop at the plate and all of their kinetic energy would be converted to an increase in static pressure. This increase in static pressure is called the "stop" pressure, numerically equal to the dynamic pressure [1].

2. METHODOLOGY

In the first stage, a CAD model was created using CATIA V5 software. The numerical model is generated from the geometrical model. After generating the numerical model, the aerodynamic behavior is simulated by CFD, with the help of ANSYS FLUENT software.

2.1. Geometric Model

With the sets of commands and functions available through CATIA V5 a

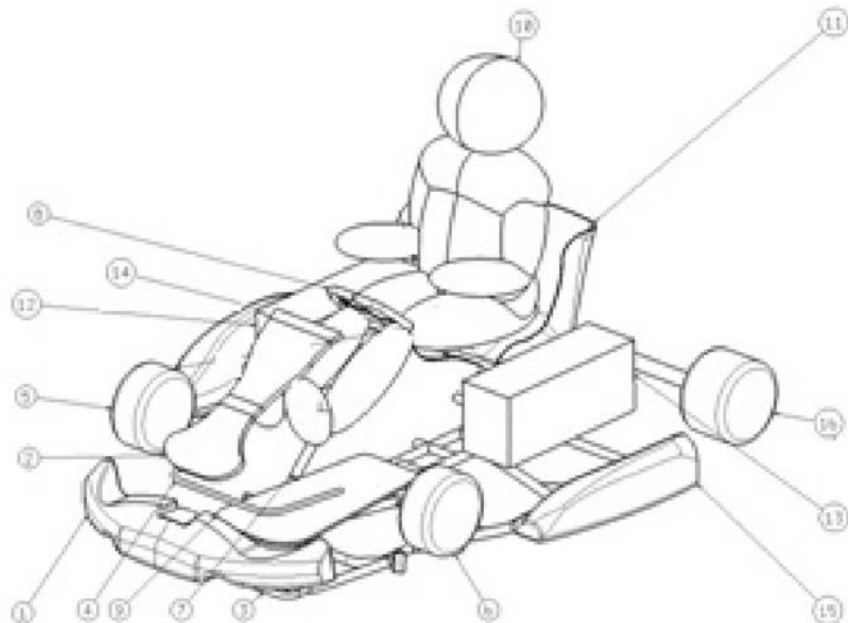


Fig. 2. Geometrical model and components: 1 – Front bumper; 2 – Steering column protection; 3 – Chassis; 4 – Frame (front protection); 5 – Right front wheel; 6 – Left front wheel; 7 – Steering column; 8 – Steering wheel; 9 – Floor panel; 10 – Driver; 11 – Seat; 12 – Right batteries; 13 – Left batteries; 14 – Side left protection; 15 – Side right protection; 16 – rear axle (assembled).

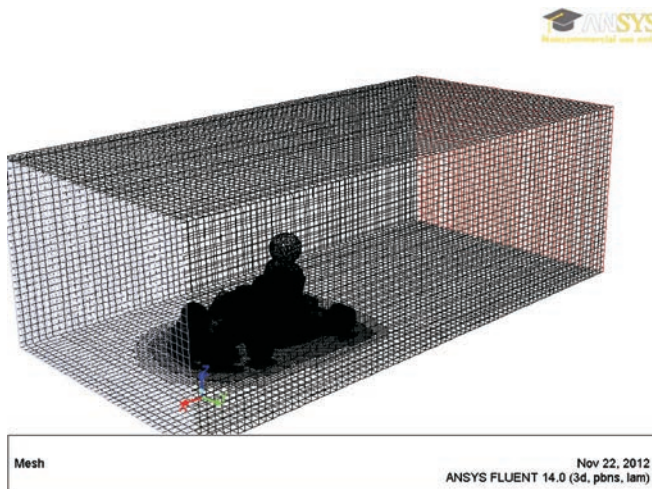


Fig. 3. Computational grid

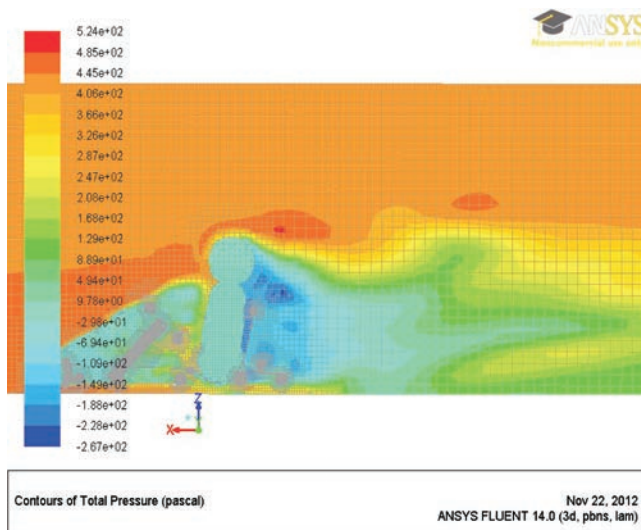


Fig. 4. Numerical results of total pressure (velocity 25 m/s)

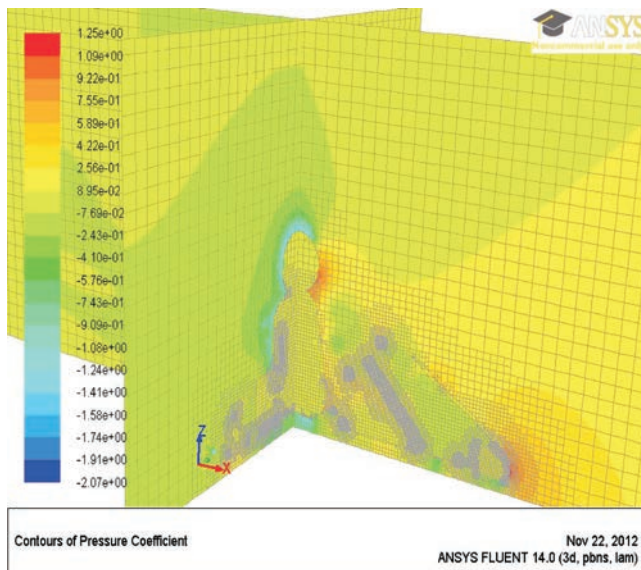


Fig. 5. Numeric results of the pressure coefficient

Table 1. Calculated aerodynamic drag forces for fluid speed $v=15$ m/s

Zone	Pressure	Viscous	Total
Front bumper	-8.564	-0.047	-8.611
Steering column protection	-4.549	-0.011	-4.560
Chassis	-3.627	-0.050	-3.677
Frame (front protection)	-0.006	0.000	-0.006
Right front wheel	-0.811	-0.020	-0.831
Left front wheel	-0.810	-0.021	-0.830
Steering column	0.080	0.001	0.081
Steering wheel	-0.812	-0.004	-0.816
Floor panel	-0.100	-0.012	-0.111
Driver	-22.050	-0.051	-22.101
Seat	-1.274	-0.005	-1.279
Right batteries	-7.910	-0.012	-7.922
Left batteries	-9.096	-0.013	-9.109
Side left protection	-0.866	-0.041	-0.907
Side right protection	-2.027	-0.046	-2.073
Engine	-3.947	-0.003	-3.950
Rear axle (assembled)	-8.321	-0.055	-8.376
Net	-74.689	-0.391	-75.079

Table 2. Calculated aerodynamic drag forces for fluid speed $v=20$ m/s

Zone	Pressure	Viscous	Total
Front bumper	-15.396	-0.063	-15.458
Steering column protection	-8.328	-0.015	-8.343
Chassis	-7.441	-0.068	-7.509
Frame (front protection)	-0.072	-0.001	-0.074
Right front wheel	-0.736	-0.027	-0.763
Left front wheel	-1.401	-0.025	-1.425
Steering column	0.049	0.000	0.049
Steering wheel	-2.144	-0.005	-2.148
Floor panel	-0.170	-0.017	-0.186
Driver	-38.546	-0.067	-38.613
Seat	-3.408	-0.006	-3.414
Right batteries	-14.208	-0.017	-14.225
Left batteries	-15.720	-0.018	-15.739
Side left protection	-3.926	-0.061	-3.987
Side right protection	-2.637	-0.044	-2.682
Engine	-4.793	-0.005	-4.798
Rear axle (assembled)	-14.728	-0.071	-14.799
Net	-133.605	-0.510	-134.115

geometric model of the kart and driver was generated. The geometric model corresponds to a medium-sized individual with a height of 1750 mm.

The generated CAD model and components are shown in Fig. 2.

This model was used taking into account the dimensions of the kart and the anthropometric dimensions of the users of this kart. In CATIA V5, model type dummies are used for evaluation of ergonomics performance of various technological operations. A simplified model of the driver will reduce the necessary computational resources. Therefore, an ellipsoid model was developed and it corresponds to a medium-sized individual with a height of 1750 mm.

2.2. Numerical Model

The computational grid (Fig. 3) is defined by taking the geometrical model as a carrier defined above. The numerical model defines the computing grid necessary for numerical solving of the set equations describing the fluid flow. At this stage, the boundary conditions are defined (wall,

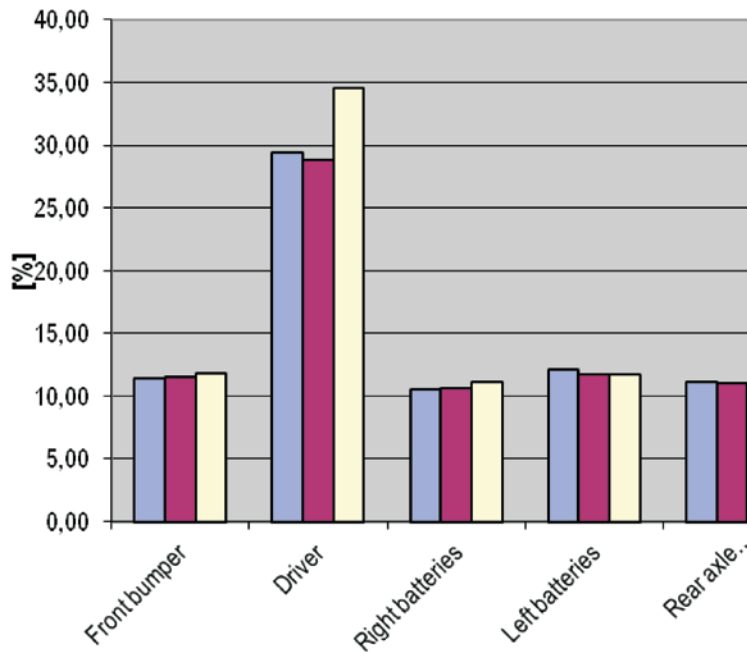


Fig. 6. Geometric elements with significant contribution in generating aerodynamic drag force

interaction area, entrance area, output area) and the conditions necessary to establish the set of equations describing fluid flow are also created.

The geometric model of the electric kart and driver represents the structure – defined as a wall – around which airflow is studied. The outer faces of the geometric body representing air are defined as the borders of the field of computing. These borders can be associated with different properties such as wall, input or exit. The numerical values that are initialized to these areas are the air flow rate that gives the speed of the vehicle and pressure. If the output value of the surface pressure is equal to atmospheric pressure thereby defining a free flow conditions. Solving numerical model Table 3. Calculated aerodynamic drag forces for fluid speed $v=25$ m/s

Zone	Pressure	Viscous	Total
Front bumper	-24.04	-0.08	-24.11
Steering column protection	-11.86	-0.02	-11.88
Chassis	-10.12	-0.07	-10.19
Frame (front protection)	-0.09	0.00	-0.09
Right front wheel	-2.33	-0.03	-2.36
Left front wheel	-3.39	-0.03	-3.42
Steering column	0.34	0.00	0.34
Steering wheel	-1.44	-0.01	-1.45
Floor panel	-0.26	-0.01	-0.28
Driver	-70.30	-0.08	-70.38
Seat	3.28	-0.01	3.27
Right batteries	-22.65	-0.02	-22.67
Left batteries	-23.87	-0.02	-23.89
Side left protection	-4.17	-0.08	-4.25
Side right protection	-4.64	-0.06	-4.69
Engine	-6.37	-0.01	-6.37
Rear axle (assembled)	-21.03	-0.09	-21.12
Net	-202.94	-0.61	-203.54

provides the user with a series of results necessary for evaluating the aerodynamic performance.

2.3. Conditions of simulation and turbulence model

The influence of the ground on the main aerodynamic characteristics of the car, drag and lift, is studied in two ways, commonly used in wind tunnels, respectively without ground effect (fixed wheels and no relative motion between car and road), and with the moving wall approach [2]. This study is carried out considering fixed wheels and no relative motion between kart and road for three velocities of 15, 20 and 25 m/s. Analyses were performed in steady state, for a reference pressure of the air $p=1$ At. The SST (Shear Stress Transport) turbulence model is used to solve the simulation process.

3. RESULTS

The pressure diagram plots the pressure values obtained by solving the numerical model. Information about total, static and dynamic pressure and pressure coefficient is available.

Figure 4 shows the variation of total pressure for the studied kart. Figure 5 shows the calculated value of the pressure coefficient.

4. CONTRIBUTION OF THE KART'S COMPONENTS IN GENERATING DRAG

In order to evaluate the contribution of the kart's components to the aerodynamic force, a set of 3 numerical tests were made at speeds of $v_f=15$ m/s, 20 m/s and 25 m/s. Table 1, Table 2 and Table 3 show the results of calculated aerodynamic drag forces that each component generates.

In order to highlight the results, the figure 6 shows, in percentages, the geometric elements with significant contribution in generating the aerodynamic drag force.

5. CONCLUSION

Analyzing the results presented in Figure 6 one can see that within reasonable limits of $+/-2\%$ the batteries have about 25% contribution in generating drag at various speeds. It is quite significant and it is a negative impact on aerodynamic behavior. This happens due to the front vertical flat surface of the batteries. The problem may be solved by placing wind deflectors in front batteries or use protective covers with a shape that would have less contribution in generating drag. Another possibility would be to place the batteries in the middle of the kart, between the steering column and the seat.

This computational model allows further investigation of the influence of driver size and can be used to optimize vehicle outer shape for improved aerodynamics.

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Fig. 7. Pressure drop test

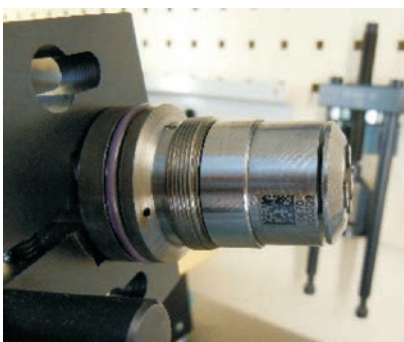


Fig. 8. EUI body



Fig. 9. Hartdridge test bench

Table 1. Results of the tests on the Delphi EUI

EU l	T1 [bar]	T2 [10 s]	T3 [s]	Valve ure	Fig- ure	Observations
1	335	yes	10.5	Fig.10		stuck nozzle
2	342	yes	5.2	Fig.11		stuck valve
3	341	yes	1.3	Fig.12		unidentified small part
4	339	yes	1.35	Fig.13		
5	342	yes	2.5	Fig.14		
6	339	yes	1.7	Fig.15		

2.2. Bosch UIP

The first analysis consists is focused on the six unit pumps that equipped the OM 502 LA-542 engine of an MB Actros 1840 LS truck (11946 ccm, 290 kW, Euro 2, 1998 year). The engine of the vehicle presents heavy start,

and no abnormal geometry of the nozzle's holes and corresponding injection parameters (injection pressures, opening times, injection periods, a.o.), thus excluding the injector units as responsible of the malfunctioning of the engine.

In these conditions, the whole attention was concentrate on the six unit

either in cold or warm conditions and has altered performances (power, torque, speed stability). The suspicions lead to the injection systems, for instance also for the unit pumps and/or the injector's nozzles (Fig.1 and 2). The separately testing of the injectors, including nozzles, by visual observation under the microscopic procedure and on specific test bench, reveals that there are no obstructions or leaks



Fig. 10. Valve EUI 1



Fig. 11. Valve EUI 2

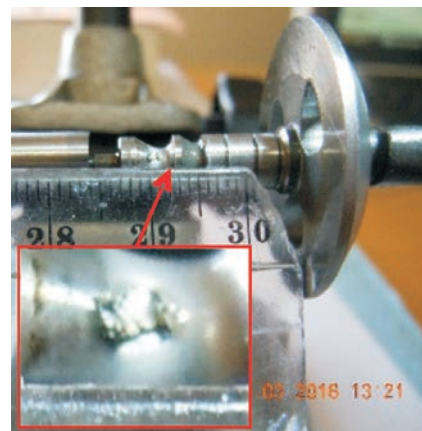


Fig. 12. Valve EUI 3




Fig. 13. Valve EUI 4



Fig. 14. Valve EUI 5



Fig. 15. Valve EUI 6



Test Type

Diagnostic Test

Operator Name

GEZA

Test Time

2016/03/25 17:17:18

Owner Name

ERLEND DIESEL SERVICE

EUI/EUP Information

Part Number

8113941

Serial Number

00026218ED

Original Code

0300070

Result Details

No.	Result Information	Value	Min	Max	Units	Status
1	Electrical Results					
2	- SCV Resistance	4.266	3.500	5.100	Ohms	✓
3	- Valve Test OK	-	-	-	-	✓
4	Diagnostic Results					
5	- Results - OUT OF LIMITS!	-	-	-	-	✗
6	- Purge	-39.0	-25.4	25.4	mm3/st	✗
7	- Base Load	1.616	1.200	1.900	Volts	✓
8	- Response Time - T3	39.0	-75.0	75.0	µS	✓
9	- Peak Pressure	-396.0	-200.0	200.0	bar	✗
10	- Rated Delivery	-41.8	-11.4	11.4	mm3/st	✗
11	- Peak Torque Delivery	-57.3	-16.5	16.5	mm3/st	✗
12	- Idle Delivery	-11.1	-8.5	8.5	mm3/st	✗

Fig. 16. Rejection of EUI 1

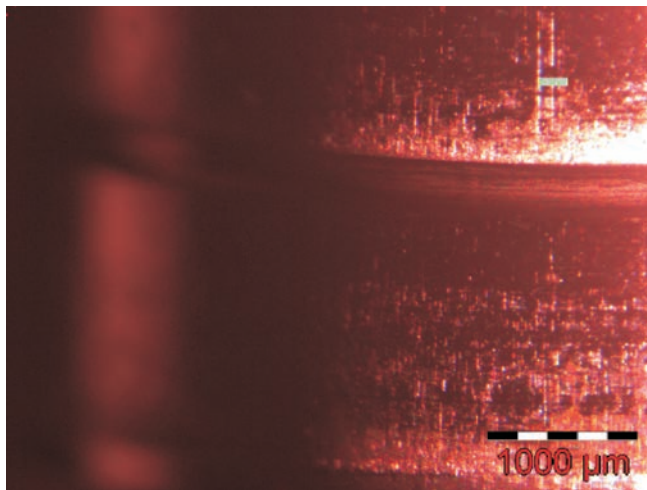


Fig. 18. Valve pin micro groove



Test Type

Diagnostic Test

Operator Name

GEZA

Test Time

2016/04/01 14:18:28

Owner Name

ERLEND DIESEL SERVICE

EUI/EUP Information

Part Number

8113941

Serial Number

00024835DD

Original Code

270010

Result Details

No.	Result Information	Value	Min	Max	Units	Status
1	Electrical Results					
2	- SCV Resistance	4.220	3.500	5.100	Ohms	✓
3	- Valve Test OK	-	-	-	-	✓
4	Diagnostic Results					
5	- Results - OUT OF LIMITS!	-	-	-	-	✗
6	- Purge	-28.0	-25.4	25.4	mm3/st	✗
7	- Base Load	1.634	1.200	1.900	Volts	✓
8	- Response Time - T3	53.7	-75.0	75.0	μs	✓
9	- Peak Pressure	-262.8	-200.0	200.0	bar	✗
10	- Rated Delivery	-32.0	-11.4	11.4	mm3/st	✗
11	- Peak Torque Delivery	-38.7	-16.5	16.5	mm3/st	✗
12	- Idle Delivery	-8.4	-8.5	8.5	mm3/st	✓

Fig. 17. Rejection of EUI 2

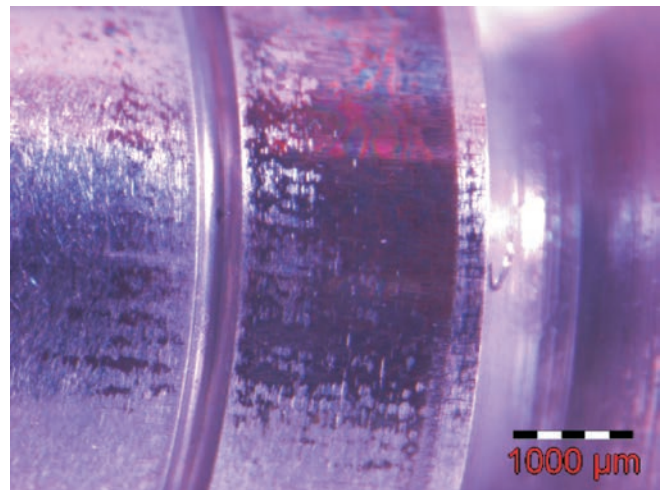


Fig. 19. Valve pin erosion

pumps, as the most important part of the injection system that could generate working inaccuracies.

In Fig.1 is presented one of the injection pump unit (UIP), which was tested for working parameters (injection pressure, pressure drop, leaks, flow, etc.).

All the six UIP were tested individually, in order to precisely identify the whole problems of the injection system.

Because the working condition refer also to the flow measurement, there were performed such tests on EPS 815 test bench and were followed up multiple parameters and demanding conditions requisites, for example fuel/conditioning oil temperature, pressure, flow on different conditions (start up point, warm up, clean run, conditioning, rated point, cooling).

All these parameters were directly acquired and analyzed through specific interface of the test bench and some of this information, for rated point measurements, is presented in Fig.3.

The faults identified on the valve-seat assembly of the six UIP, through microscope visualization, were similar and consist of many erosion marks caused, most probably, to the impurities in the fuel and compromising of the fuel filter, due to long time exploitation between changes.

Thus are presented images captured on valve seat, Fig.4, valve, Fig.5, and valve pin, Fig.6, respectively, with the erosion spread marks on the jointed surfaces (in these pictures the parts were separated, after dismounted, for direct investigation).

2.3. Delphi UIP

The tested D12D diesel engine of the Volvo FH 42 truck (12.130 ccm, 309 kW, Euro 3, 2005 year) was equipped with six Delphi E3 Electronic Unit Injectors (EUI), Fig.2, and the symptoms claim by the owner consists of impossible normal start (only with additional volatile substances injected/sprayed in intake pipe), variable idle speed and poor performances (power/torque).

The injection system works at high pressure (maximum around 2500 bar) and the "mechanical" parts are controlled through an electronic control unit (ECU) of the diesel management structure.

As a whole assembly, consisting either of high pressure pump but also with the valve-nozzle injector unit, the possible damages that distort the working parameters must be separated, and each part must be distinctly diagnosed.

Following technical specification of the EUI manufacturer (Delphi), the units were tested for:

T1. Nozzle Opening Pressure – NOP, that must be around 250 – 300 bar. The precise value is recorded only in the technical documentation of each specific type of nozzle/injector, and is not directly presented in this article;

T2. Seat Tightness – ST, presume the direct observation of possible leak of fuel/calibration oil that could occur on the nozzle hole's surface, when the pressure inside the injector are maintained at least 10 s, at a pressure with 15 bar less than the pressure prescribed for NOP;

T3. Back Leakage – BL, that reveal the period of maintaining the drop of

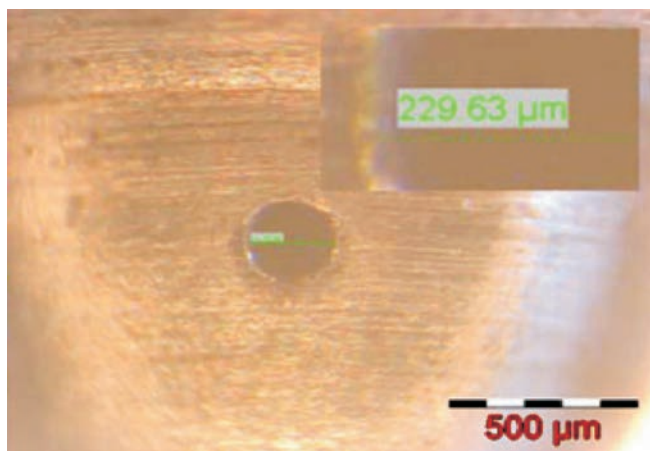


Fig. 20. Measured nozzle's hole

pressure between 170 to 140 bars, inside the nozzle. Optimum period of the pressure drop must be in 3 – 30 s range;

T4. Testing of the EUI in the Hartdridge AVM 2 – PC – 20hp test bench.

Fig.7 presents the testing device for pressure drop measurement of the nozzle, Fig. 8 presents the special unmounting/mounting device of the EUI and an EUI body (consisting of hydraulic body, valve, electric coil, a.o.) and Fig. 9 presents the Hartdridge AVM 2 – PC -20hp test bench, prepared for EUI test. Table 1 presents the values obtained for the T1 – T3 test items, applied to all six EUIs, and also some comments about the status of each injector/part when was unmounted.

2.4. Results and Discussions

According to Table 1 results it can be observed that all the EUI's nozzles present proper values for the NOP and only the first two EUIs (1 and 2) present appropriate values for the BL test. Because only these two/three types (T1 – T3) of results are not enough for validate the testing operation of the EUIs, all the units will be tested also on the Hartdridge AVM 2 – PC -20hp test bench. After the complete tests of the EUIs on the test bench, also the EUI 1 and 2 were rejected ("Out of Limits" results, see Fig.16 and 17), because internal testing procedure consisting on "Peak Pressure" – line 9, "Rated Delivery" – line 10 and "Peak Torque Delivery" – line 11, were not passed.

Thus it was necessary supplementary investigation on the internal damages of the hydraulic assembly of the EUI that were performed on the stereo microscope. The visualization of the corresponding joint surfaces reveals pronounced erosion marks (micro grooves, Fig.18 and micro abrasive marks Fig.19) on the high pressure valve liners and pins, all of these as consequences of the impurities in the fuel or compromising of the fuel filtration. Were excluded the inappropriate material quality because on these surfaces were not identified point-size regions where pitting occurs and also there was no thermal hardening or chemical treatment of these surfaces applied.

The micro groove presented in Fig.18 reveal the action of some abrasive particles (the dimension of the groove is 21.96 μm, see details in Fig.17) which generate axial wear on material and Fig.19 present the spreading of the wear surface to up to 35 – 45% of the lateral surface of the sealing/joint surface of the valve, that generate the inappropriate working conditions of the whole EUI. Due to these advanced erosion of the valve liner – valve pin joint, for all the EUIs, the high pressure "sealing" are compromised, the leaks and pressure drop are high and the EUI were rejected by the Hartdridge testing equipment, and the only possibility too fix the claims of the malfunctioning of the engine is to replace all these EUIs.

In the same time, for all the six injectors visual analysis was performed, on the stereo microscope, to identify possible alteration of the nozzle's holes. For

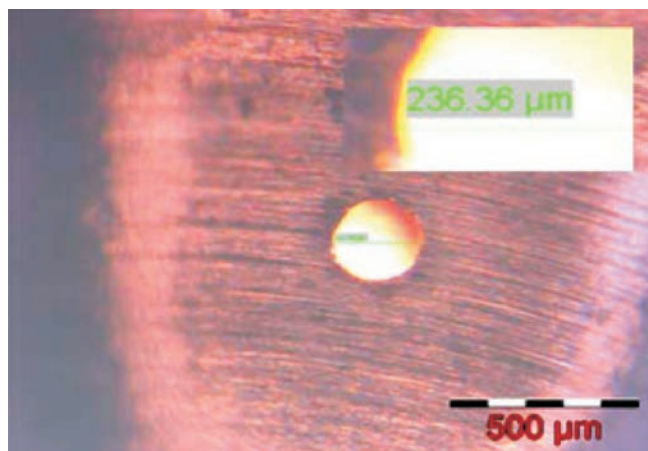


Fig. 21. Nozzle's hole with internal light source

each of the fifth holes of each nozzle the optical measured dimensions, using an internal light source (see Fig.21) were in the 220 – 240 μm range, Fig. 20 and 21. According to this visual inspection were observed compromising of the nozzle's surface and was established that these defects occur as consequences of the steel-brush cleaning of the injectors, in order to remove the soot and unburned hydrocarbons deposits on EUI.

3. CONCLUSIONS

The diagnosis of the injection system of a Diesel engine is a high technology operation due to the high level of the testing equipment demands. Thus, only in specialized laboratories or facilities, on dedicated devices and following specific and coded information (not open/explicit values), these tests applied on injection system parts, for validating their performances, could be applied. The testing equipment manufacturer could offer the possibility to test not only the same brand/manufacturer parts but also to test some other parts from other manufacturer, but, in most of the cases, the validation and coding of the injector assembly, in order to be installed on the Diesel engine, could be performed exclusively on some dedicated equipment. The tests made on the Bosch's injection pump units that were claimed for malfunctioning on an Euro 2 heavy truck 12 liters engine, reveals the erosion marks and grooves on the valve assembly (valve seat and joint surfaces), these damages being produced as consequences of fuel impurities and fuel filtration alteration. In this case the injector assembly performed on specific parameters and only the injection pump units are responsible for engine's malfunctioning. For the Euro 3, 12.1 liters tested Diesel engine, that were equipped with Delphi's Electronic Unit Injectors (EUI), only the whole diagnosis, on separate parts and for the whole assembly of each EUI, reveals all the troubles of these equipment. Because the general leaks and pressure drops are considered only as guided information, there were necessary direct testing of each EUI, performed on the special test bench, and all of the tested EUIs were rejected, in direct accordance with the technical specification of the manufacturer. Supplementary investigations performed on the hydraulic assembly identified also massive erosion marks and grooves, produced by the same fuel contamination and fuel filtering discredit.

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Deadlines

- 17.04.2017: Submission of abstract
22.05.2017: Notification of acceptance
24.07.2017: Final paper



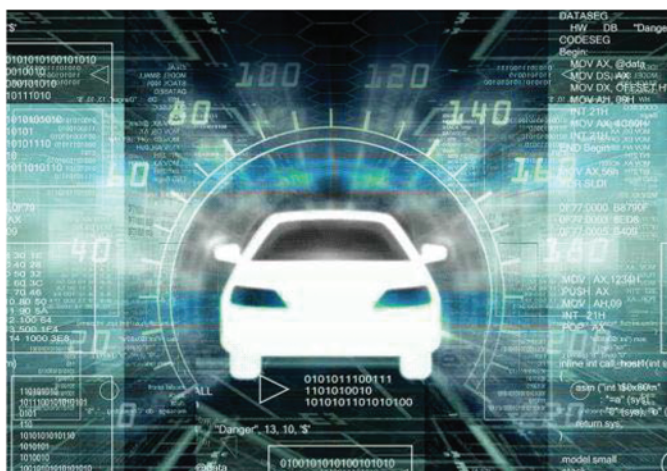
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