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ABOUT THE CALCULATION OF EXHAUST GAS PARAMETERS OF A DIESEL ENGINE WITH MEDIUM CAPACITY

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(Received 19 September 2018; Revised 11 Octomber 2018; Accepted 13 Octomber2018)

Abstract. *The article touches upon the issue of the problem of calculating the parameters of the exhaust gas and the solid dispersed phase of the diesel engine of the car of statistical average power. The purpose of the paper is to determine the parameters of the exhaust gas and to select suitable methods for its purification. The results of calculating the parameters of the exhaust gas and solid dispersed phase can be used in the design of the device for cleaning the exhaust gases of the diesel engine of internal combustion of a car.*

Key-words: *diesel engine exhaust gas components, exhaust gas density, dispersed composition of solid fractions of exhaust gases, devices of dry, wet and electrostatic cleaning of exhaust gases*

1. INTRODUCTION

To optimize the design of the device protected by the patent [1], to purify exhaust gases (EG) of an internal combustion engine of the diesel type, which includes three degrees of trapping of solid and liquid fractions of exhaust gas, it is necessary to determine the expected parameters for the dispersion composition of solid exhaust gases of exhaust gases, as well as operating parameters gases: composition, density, temperature, pressure etc. The purpose of the article is to determine the expected parameters of the exhaust gases of a diesel engine for calculating the design of the device for cleaning exhaust gases of a diesel engine of statistical average power.

2. INITIAL PARAMETERS OF EXHAUST GASES OF THE CAR

To calculate the size of the device for gas purification, the following initial parameters of the exhaust gases must be known: the volume of the exhaust gas under normal conditions - VEG (m3/s); the mass of the gas at the entrance to the device - MEG (kg/s);density of the exhaust gas at the inlet to the device ρEG (kg/m3);temperature of exhaust gases tEG °C; humidity of exhaust gases to be purified - HEG (kg/m3); excess pressure or rarefaction of the gas to be cleaned ± PEG (kPa); barometric pressure - Pbar (kPa); gas composition for the components - rN2, rO2, rH2O, rCH, rCO2, rCO, rNOx fraction (the sum of the components should be one); concentration of solid fractions in the gas to be purified - gs (g/m3), the number of soot fraction - NPM, the particulate matter carbon black - PMdi (μm); (i = 1,2, ... N); percentage composition of carbon black having particle size di and less -PCi, (%) (i = 1,2, ... N); density of soot particles - ρsp, (kg/m3); required degree of gas purification from soot - ηs (%).

3. CALCULATION OF THE EXHAUST GAS PARAMETERS OF A CAR WITH AN AVERAGE STATISTICAL POWER ENGINE

The amount of exhaust gas and other engine parameters depend on power, fuel consumption, engine type, operating mode etc. The power of the diesel engine for calculating the design of the device for cleaning the exhaust gases protected by the patent [1] we take equal to the statistical average power of the engines of cars sold in 2012 in Germany. According to the Center for Automobile Research at the University of Essen, the engine power was PEA = 138 hp (101.5 kW).

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The leader among diesel engine manufacturers in the US, CUMMINS, has published data for a diesel engine type NT855G6 (2010) with a rated output of PEC=326 hp(240 kW) [2].
This engine consumes - V'c= 67.0 l/h of fuel and 1320 m3/h of air during operation in nominal power mode and produces exhaust gases in the volume V'EG = 3 858 m3/h at a temperature of tg= 574 °C.
If we assume that the average statistical power engine has a design similar to the NT855G6 engine, then it can be assumed that the average statistical power engine can have exhaust volume and other parameters in proportion to the parameters of the American engine. This assumption allows us to determine the following parameters for an engine with an average statistical power.
The volume of exhaust gases in the nominal power mode will be:

$$VEG = V'EG \cdot PEA / PEC \tag{1}$$

VEG = 3 858 · 138 / 326 = 1 633 [m3/hour] = 27,2 [m3/minutes] = 0,45 [m3/s]

Consumption of fuel:

$$Vc = V'c \cdot PEA / PEC = 67{,}0 \cdot 138 / 326 = 26{,}4\ [l/h] = 0{,}44\ [l/min] = 0{,}00733 [l/sec] \tag{2}$$

The temperature of the exhaust gas is the same with the engine type NT855G6: t'EG=574 °C.
Specific content of soot (carbon C) in the exhaust gases of diesel engines under full load conditions is 0.25 - 2.0 g/(kW/h) [6].
We take the specific release of soot gESRS = 2.0 g/kWh, the emission of soot in the nominal power mode for the average statistical engine will beup to:

$$GSRS = PEA \cdot gESRS = 101.5 \cdot 2.0 = 203.0\ [g/h] \tag{3}$$

In this case, the specific concentration of soot particles in the exhaust gases will be:

$$GCSRS = GSRS / VEG = 203.0 / 1633 = 0.124\ [g/m3] \tag{4}$$

In the rated power mode, burning one liter of fuel produces soot

$$GSl = GSRS / Vc = 203.0 / 26.4 = 7.69\ [g] \tag{5}$$

Volkswagen's 2009 engines, with a power of 103 kW, consume FSC =7.4 liters per 100 kilometers and produce an exhaust gas volume per 100 kilometers:

$$VSVW = VEG \cdot FSC / Vc = 1633 \cdot 7.4 / 26.4 = 457{,}7 [m3/100\ km] \tag{6}$$

At the same time, a Volkswagen engine the 2009 of the model probably has in the exhaust gases at the exit of the engine cylinders the next amount of PM of soot

$$GSSVW = GSl \cdot FSC / 100 = 7{,}69 \cdot 7{,}4 / 100 = 0{,}569\ [g/km] \tag{7}$$

The exhaust gases must be cleaned and contain the amount of PM of soot no more than: GSS = 0,005 g/km, according to the EURO - 6 emissions standards (diesel).
To ensure the EURO-6 norms for the permissible emission of soot per 1 km from an engine with an average power of N = 103 kW, the required degree of purification should be not less than:

$$\eta rd = (GSSVW - GSS) / GSSVW = (0{,}569 - 0{,}005) / 0{,}569 = 0{,}99 \tag{8}$$

Let's define the expected density of the exhaust gas of a diesel motor of average power in the nominal power mode. The exhaust gas of the diesel engine is a mixture of exhaust gas components.
We will select the main components of the exhaust gas of the diesel engine (under normal conditions) and determine the expected density of the exhaust gas of the diesel engine as a homogeneous gas, replacing the mixture of exhaust gas components of the diesel engine -ρ (kg/m3).

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$$\rho EG = \sum ri \cdot \rho l \quad (9)$$

where ρi - is the density of the *i*-th gas, is calculated as the density of this gas, in the amount of *mi* kilograms in the volume *wi* (partial volume) at a mixture pressure of PEG and the temperature of the exhaust gases *t*EG °C.

Exhaust gases contain more than 200 different hydrocarbons (CnHm - ethane ρe = 1,25; methane ρm = 0,67; ethylene ρet = 1,17; benzene ρb = 3,25; propane ρpr= 1,83; acetylene ρac= 1,08, etc.) [7]. Taking into account the low concentration and the wide variety, we take the average density for hydrocarbons ρCH = 1,25equal in gas density ethane.
For exhaust gas of a diesel engine:

$$\rho' EG = rN2\ \rho N2 + rO2\ \rho O2 + rH2O\ \rho H2O + rCH\ \rho CH + rCO2\ \rho CO2 + rCO\ \rho CO \quad (10)$$

where:
rN2, rO2, rCO2 –volume concentration in fractions of a unit
ρN2,ρO2,ρCO2 – density of components (under normal conditions), kg/m3 [7]
ρ'EG= 0,77*1,165+0,15*1,330+0,075*0,749+0,001*1,25+ 0,002*1,830 + 0,002*1,164=1,161 [kg/m3]

The density of the exhaust gases *ρEG* at the inlet to the cleaning device is calculated for the temperature tEG= 500 °C, since the gases are partially cooled when they pass through the gas line to the cleaning device:

$$\rho EG = \rho' EG \cdot (Pbar + PEG) \cdot 273 / (Pbar \cdot (+273 tEG)) \quad (11)$$

where:
Pbar- the barometric pressure, Pbar = 101.3 kPa;
P'EG- the pressure in the flue at the exit from the engine cylinders, according to [9] we take - P'EG = 147 kPa (1,5 kg / cm2)
ρEG= 1,161 · (101,3+ 147)·273 / (101,3 · (+ 273 500)) =1,005 [kg/m3]

According to Charles's law with the same volume of gas

$$V = const \rightarrow P'EG / t'EG = PEG / tEG \quad (12)$$

Therefore, the pressure of the exhaust gas at the inlet to the gas purification device will be equal to:

$$PEG = P'EG \cdot tEG / t'EG = 147 \cdot 500 / 574 = 128\ [kPa] \quad (13)$$

4. CALCULATION OF EXHAUST GAS FRACTIONS

The formation of both a carbon and a complex structure of microparticles in a flame when the fuel burns in an ICE combustion chamber is, in general, a process of condensation.
As a nonequilibrium thermodynamic process, it is determined by a large number of parameters.
To determine the relative volume and weight of the solid and liquid fractions of the parameters of the exhaust gases, we use the results of research carried out by V.V. Brazovsky[3].
V.V. Brazovsky conducted a study of the dispersed composition of exhaust gases using the registration of a hologram on the matrix of a high-speed digital camera and special software methods for processing the obtained holograms. A holographic image of an ensemble of microparticles (soot) was obtained for a 1 × 1 cm2 exhaust gas flow section, thanks to the use of a helium-neon laser.
The histogram of the distribution of the solid and liquid fractions of the KamAZ-740 engine was obtained with a combustion engine load of 160 kW and 2600 rpm, a coolant temperature of 80 ° C, a sampling step of 0.5 μm in this measurement [3].
KamAZ-740.662-300 (Euro-4) diesel engine with turbocharging, with intercooling of charge air, electronic control and fuel system of "Common Rail" type.
The common rail injection system is a modern fuel injection system for diesel engines.

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Therefore, the results of the study of the condensed phase of the KamAZ-740 engine are applicable for modelling the volume, weight and distribution of solid and liquid fractions of the diesel engine of an average car.

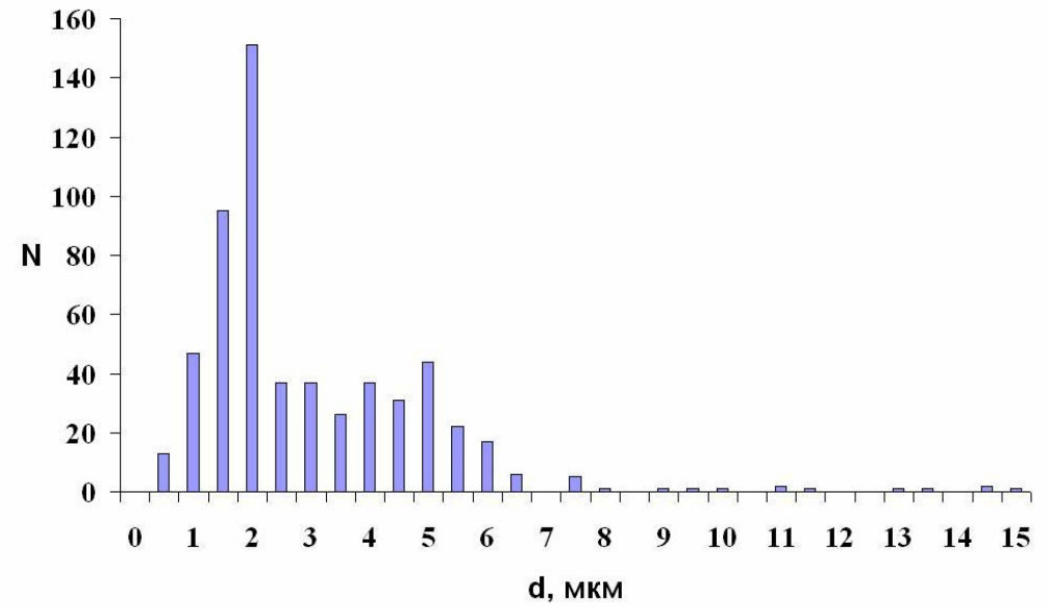

Figure 1. The distribution of N (d) (here N is the number of particles, d is the diameter in μm).

Based on the approximate histogram of the particle distribution over the diameters (Figure 1), we determine the relative volume and weight of the exhaust gas fractions that will be more efficiently captured at the corresponding stage of the exhaust gas cleaning for combustion in the combustion chamber in accordance with the patent [1].
The main methods for cleaning exhaust gases from aerosol, solid or liquid particles are: "dry" method, "wet" method and under the influence of electrical forces.
The cyclone works on a dry basis, it is effective for dust with a particle size of more than 10 μm [5] and reliably operates at temperatures up to 500 ° C.
A wet scrubber works on the wet method, it is effective at an average particle size of ≥ 0.1 μm.
Electro filters are used for fine purification of gases from aerosol particles of 0.01 to 100 μm at a gas temperature of up to 400 - 450 ° C. The proposed device for cleaning the exhaust gases of an internal combustion engine [1] is shown in Figure 2.
The device is designed for cleaning the exhaust gases: in the first stage by a conical cyclone, on the second stage by a wet dust collector of the scrubber type, and at the third stage by an electro filter type apparatus. Taking into account the provision of effective purification of up to 97.5% (we accept up to ηc= 0.97) for dust with a particle size of more than 10 μm [4], we take for the first stage of the device [1] (cyclone) cleaning particles of 10 to 16 μm in size.
Taking into account the provision of effective purification of up to 96-98% (we accept up toηsc = 0.97) and more with an average particle size of 1-2 μm, we assume for the second stage of purification of the device [1] (scrubber) the calculated particle size range from 2.5 up to 10.0 microns.
Taking into account the provision of effective purification to 96-98% (we accept up to ηef = 0.97) and more with an average particle size of 0.01 μm and higher at a gas temperature of up to 400 - 450 ° C, we accept for the third stage of purification of the device [1] (electrostatic precipitator), the calculated particle size range is from 0.01 to 2.25 μm.
Let us calculate the relative volume of particles of the same diameter of the solid and liquid fractions of the exhaust gas, according to the distribution shown in Figure 1, according to the formula:

$$Vi=Ni*1/6\ \pi di3 \qquad „ \qquad (14)$$

where:
Ni- is the number of particles in the i-th fraction,
di -is the diameter of the i-th fraction, μm.

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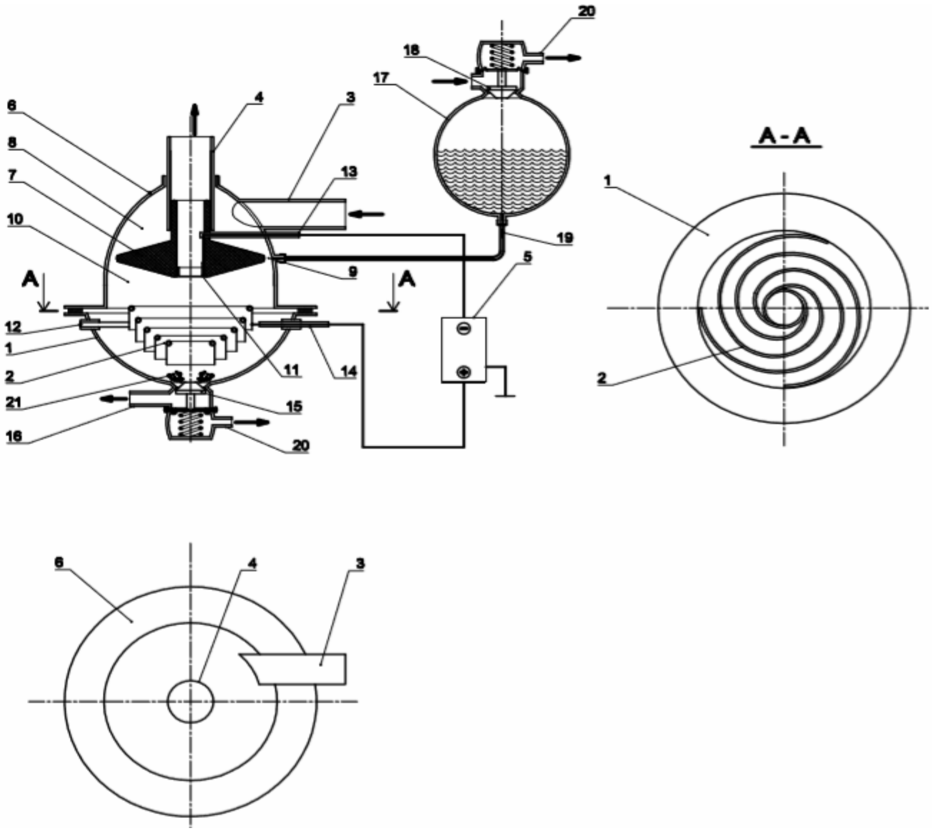

Figure 2. Device for cleaning of exhaust gases [1]

The total relative volume of a group of fractions with a diameter of 0.5 to 2.5 μm, which will be captured by the electrostatic precipitator ΣVce = 1 208,7 μm3
The total relative volume of a group of fractions with a diameter of 2.5 to 10.00 μm that will be captured by a scrubber ΣVcs= 19029,03 μm3
The total relative volume of a group of fractions with a diameter of 10.0 to 16.0 μm that will be captured by the cyclone ΣVcc = 18976,13 μm3
The total relative volume of solid and liquid fractions in the exhaust gas:

∑ *VPM 0-15=ΣVoe+ΣVcs+ΣVcc* =1 208,70+19029,03+18 976,13=39 213,9 [μm3] (15)

The specific gravity of the soot particles, for particles of diesel soot Gss, lies within the narrow limits of 1800-2100 kg / m3 [5] and can be assumed equal to:
Gss =1950 kg / m3= 1, 950 mg/mm3 = 1, 950 · 10-6mg/μm3

Let's calculate the relative weight of the solid and liquid fractions of the exhaust gas of each group of particles in accordance with their volume:

∑*Gce* =*ΣVce* · *Gss* = 1 208,70 · 1, 950 · 10-6= 0,00236 [mg] (16)

∑*Gcs* =*ΣVcs* · *Gss* = 19029,03 · 1, 950 · 10-6= 0,0371 [mg] (17)

∑*Gcc* =*ΣVcc* · *Gss* = 18 976,13 · 1, 950 · 10-6= 0,0370 [mg] (18)

The total total relative weight of the solid and liquid fractions of the exhaust gas will be:

∑*G* =∑ *Goe*+∑*Gcs*+ ∑*Gcc* = 0,07646[mg] (19)

Based on the average data on the emission of a diesel engine, we determine the specific release of soot in the exhaust gases,gs = 2.0 g / kWh[6].

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$\sum GSS = PEA \cdot gs$ = 101,5 · 2,0 =203,0 [g/ hour] (20)

When the engine with an average power of PEA= 101.5 kW in the rated power mode in one cubic meter of waste gas will work

$gs = \sum GSS / VEG$ = 203,0 / 1633 = 0,124 [g/m3] (21)

We calculate for the average power engine the maximum weight of each group of solid and liquid fractions of the exhaust gas that must be captured by the device at the appropriate stages of purification when the engine is running at rated power:

$\sum G1 = \sum GSS \cdot \Sigma Gcc / \sum G$ = 203,0 · 0,0370/0,07646= 98,23 [g/ hour] (22)

$\sum G2 = \sum GSS \cdot \sum Gcs / \sum G$ = 203,0 · 0,0371 /0,07646= 98,50 [g/ hour] (23)

$\sum G\,3 = \sum GSS \cdot \sum Gce / \sum G$ = 203,0 ·0,00236/ 0,07646= 6,27 [g/ hour] (24)

In the proposed device, the purification of waste gases is carried out in a system of connected devices, therefore the expected overall purification degree is:

$\eta cd = 1 - (1 - \eta c) \cdot (1 - \eta sc) \cdot (1 - \eta ef)$ = 1 – (1– 0.97) · (1– 0.97) · (1– 0.97)= 0,9999 (25)

The calculated total purification level of the proposed device [1] ηcd= 0.9999 - corresponds to the degree of purification ηrd = 0.99, which is necessary to comply with the EURO 6 norms for the permissible emission of soot per 1 km.

5. CONCLUSIONS

The calculation of the expected parameters of the dispersion composition of solid exhaust gases of exhaust gases, as well as the working parameters of gases: composition, density, temperature, pressure, etc., is carried out in the article. The parameters of the exhaust gases can be used to calculate the design of the apparatus for cleaning the exhaust gases of diesel power of average power.
The device [1] for cleaning the exhaust gases of an internal combustion engine, which includes three extraction levels of solid and liquid exhaust gas fractions, can potentially provide the car with EURO-6 emission standards of soot emission for 1 km.

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GENERALIZED POISSON LINEAR MODEL FOR FATAL CRASHES ANALYSIS

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Abstract. *Mathematical models help researchers to investigate physical phenomena, out of which motor vehicle crashes are a part of this investigation domain. To understand and develop technical solutions, in order to reduce the number and severity of road accidents, different mathematical methods could be used to analyze accident data. Statistical evaluations are based on collected data contained in different databases. Computational programs are able to deliver proper results of a complex mathematical equation and also reduce significantly the calculation time. In this paper was integrated the generalized Poisson linear model into the statistical program R-Studio, in order to evaluate fatal crashes. This approach could be used in many different studies that are focusing on categorical variables evaluation, which characterize a road accident type.*

Key-words: *Generalized Poisson Linear Model, R-Studio, Fatal Road Accidents*

1. INTRODUCTION

Generalized linear models contain a standard definition of different regression types, which describe the effect of co-variables on response variable through a linear predictor.

Thus, the mean value $\mu = E(y|x)$ of response variable y is described by a linear predictor $\eta = x'\beta$, through an inverse link function $\eta = g(\mu)$ [3].

In order to evaluate complex combinationsof potential risk factors that produce road accidents the R-Studio computational program was used. It uses the R language aimed to provide statistical and graphical results. R-Studio is based on open source idea, so that everybody could actively improve this free software system [1].

Fatal road accidents represent the worst scenario involved in road crashes therefore it is highly needed to improve the traffic safety. The risk factors that produce fatal accidents on curved and inclined plane on Romanian traffic roads were evaluated using the Poisson generalized linear model and R-Studio.

The database contains crash information during a five year period between 2008 and 2012 [7].

2. METHODS

Road accidents have a low probability of occurrence, so that Poisson distribution could be successfully used in evaluating the categorical variables.

It is important to mention that Poisson distribution introduces the condition that the expected value is equal to variance, which represents the mean value [2].

The probability mass function of Poisson distribution is described by the following equation [4]:

$$f(y|\lambda) = P(Y = y) = \frac{\lambda^y \exp(-\lambda)}{y!}, \qquad y = 0,1,\ldots \tag{1}$$

where λ is the mean of a Poisson distribution representing the number of events per interval.

Thus, the mean value of Poisson distribution $\lambda_i = E(y_i|x_i)$ could be defined by the linear predictor $\eta_i = \beta_0 + \beta_1 x_{i1} + \cdots + \beta_k x_{ik}$ and logarithmic link function, described by the linear regression [5]:

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$$\log(\lambda_i) = \eta_i = \beta_0 + \beta_1 x_{i1} + \cdots + \beta_k x_{ik} \quad (2)$$

where x_{ik} represent the explanatory variables and β_k are the parameters that describe the expected modification of the log mean value, when the value of explanatory variable x_{ik} modifies with one unit.

In some cases the modification effect of the response value depends on the interaction between explanatory variables.
Eq. 3 presents the simplest regression model with interaction effect between explanatory variables x_i and z_i [3]:

$$\log(\lambda_i) = \beta_0 + \beta_1 x_i + \beta_2 z_i + \beta_3 x_i z_i + \varepsilon_i \quad (3)$$

where $\beta_3 x_i z_i$ represents the interaction between x_i and z_i and ε_i is the error term.
R-Studio function which describes the generalized Poisson linear model is stored under glm ( ) function, without interaction effect, as follows [6]:

$$GLM <-glm\ (y\sim A\ + B\ , family\ = poisson(link = log), dat = counts) \quad (4)$$

The interaction effect between explanatory variables A and B is cached in R-Studio subroutines by the following formula [6]:

$$GLM <-glm\ (y\sim A * B\ , family\ = poisson(link = log), data = counts) \quad (5)$$

Output data can be extracted using the following R functions:
- anova(object_1, object_2): compares two models;
- coef (object): extracts regression coefficients;
- deviance (object): extract the sum square of residuals;
- formula (object): returns the models formula;
- plot (object): function used to plot diagrams;
- residuals (object): extracts the residuals matrix;
- summary (object): returns the main results of the regression. [6].

The analysis of the fatal crashes using generalized Poisson linear model is based on the presumption that the observed data $y_i \in \{0,1,2,\ldots\}$during a random time period follows a Poisson distribution of the expected mean value $y_i \sim Po(\lambda_i)$, where λ_i represents the mean value defined by the following function of linear predictor η_i^{lin} [3]:

$$E(y_i) = \lambda_i = \exp(\eta_i^{lin}) \quad (6)$$

3. DATA ANALYSIS

The road accident database used in this paper contains information regarding date and time when accidents occurred, general information about infrastructure, how the accidents were produced, safety measures and visibility factors. The database is structured in Microsoft Excel format so that the information could be uploaded in different statistical programs, in this case R-Studio. In order to handle the data in the statistical program it was necessary to convert the .xls files into .csv format.
Generalized Poisson linear model handles count data, therefore the number of fatal crashes that occurred on curve and inclined plane was chosen as response variable.
In order to verify if the Poisson distribution is suitable for this study was calculated the mean and the variance of the total number of deaths during five year period of the study:

mean(Road_Accidents$Total_Number_of_Deaths)→ 0.1043181
var(Road_Accidents$Total_Number_of_Deaths) → 0.1227122

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It can be seen that the variance has a slightly higher value than the mean, which shows that the data is slightly overdispersed, but insignificantly.
To evaluate only the fatal crashes that occurred in curve (C) and on inclined plane (IP), the subset function of the statistical program was used in order to filter the data.
One remark regarding inclined plane: road accidents were classified in up-gradient and down-gradient inclined plane.

Data_1 ← subset (Road_Accidents, IP==”Up-Gradient” | IP==”Down-Gradient”)
Data_2 ← subset (Data_1, C==”Curve”)

The function factor was used to encode the accident data as categorical variable, in order to avoid unwanted calculation errors.

Data_2$IP ← factor (Data_2$IP, levels = c (“Up-Gradient” , “Down-Gradient”))
Data_2$C ← factor (Data_2$C, levels = c (“Curve”))

To find the suitable approach of analyzing fatal crashes on curved and inclined plane a chi squared test was done using anova function, in order to determine if the interaction effect (INT) between explanatory variables has influence or not.

With_INT = glm (Number_of_Deaths ~ IP * C, Data_2, family = poisson)
Without_INT = glm (Number_of_Deaths ~ IP + C, Data_2, family = poisson)
anova (With_INT, Without_INT, test = “chisq”)

Table 1.
ANOVA Analysis of the suitable model

Model	Resid. Df	Resid. Dev	Df	Deviance	Pr(>Chi)
Without_INT	59752	30874	-	-	
With_INT	59750	30866	2	7.7265	0.021*

Regarding the results listed in Table 1, it was concluded that the model with interaction effect has a significant low p-value (Pr(>Chi)), which means that the analysis should be conducted taking into account the interactions between explanatory variables.
Quasi-Poisson test was additionally implemented to investigate the level of overdispersion, because the conditional mean value does not equal the conditional variance.
The dispersion parameter for quasi-Poisson family was around the value 0.1 which falls into Poisson tolerance condition.

3. CONCLUSIONS

In this paper a mathematical methodology used in road accident analysis is presented, together with computational support of the R-Studio statistical program. General Poisson linear models for count data could be successfully used to determine risk factors that produce road accidents, with the possibility of taking into account the interaction effect between explanatory variables. Mathematical support integrated in computational programs could solve complex problems involved in road accidents.
One significant factor investigated for fatal crashes on curved roads and inclined plane was found to be lateral impacts between a vehicle and a pole, or any similar object, but this subject was studied in detailin another technical paper [7].

ACKNOWLEDGMENT

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STUDIES AND RESEARCH REGARDING OPTIMAL CARGO LOAD OF A TRANSPORT VAN USING CARMAKER SOFTWARE

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Abstract. *The main objective of this paper is to find the optimal positioning of a cargo load in a given van, using simulations implemented in CarMaker. The chosen cargo weight is 1000 kg (close to the limit of the chosen van, 1350 kg), and the factors that are used as input are: the type of driver (aggressive, defensive and normal – for each type of driver the longitudinal acceleration and decelerations vary and also the maximum velocity and lateral), the road that was kept the same and was built so that the driver has to follow several curves left and then to the right with decreasing curve radius, the manoeuvres are set as a maximum speed on the given road (100 km/h, adapted by the driver so that the van stays on the road), and the tires were kept the same. The 1-ton load was placed in the van in 9 points on two planes with different heights and all the simulations were compared to the unloaded van. Also, an 80-kg driver was input. The results that were exported from all the simulations were: the total damp force of all the wheels, the yaw angle, the pitch angle, the roll angle, the yaw rate (measured in degrees/second), the pitch rate and the roll rate all correlated to the velocity of the vehicle. CarMaker offers a lot of output information regarding the vehicle behaviour due to the complexity of the equations behind the software. Another advantage is the coupling possibility with AVL InMotion so that the virtual engine is replaced with the real engine coupled to the DynoRoad dynamometer in the Testing and Homologation Laboratory of the Automotive Engineering and Transport Department (Technical University of Cluj-Napoca). To ensure the uniformity of the forces that occur while driving on a road with curves, in all the elements of the suspension system, the load must be placed in the centre area of the van. However, the idea of the simulations was the lack of side doors of some vans that require an un-cantered placement of the cargo.*

Key-words: *simulation, carmaker software, optimal cargo load*

1. INTRODUCTION

In early studies, researchers like Heidelberg et. al. [1] and BostelMan et. al. [2] were interested more in the way of loading and placement of cargo during the manufacturing or for Naval Research.
In more recent researches, automation has taken over the loading of the cargo, so the research has gone towards that direction, like Teller et. al. [3], studied a voice-comandable robotic forklift working alongside humans in minimally-prepared outdoor environments; McLaughlin et. al. [4] researched a detector placement optimization for cargo containers using deterministic adjoint transport examination for SNM detection; while Turanov et. al. did an analytical investigation of cargo displacement during the movement of rolling stock on a curved section of a track.
Further researches underlined the development of a decision support system for air-cargo pallets loading problem and also problems regarding analytical modelling cargoes displacement in wagon and tension in fastening, researches made by Chanet. al. [6] and Turanov et. al. [7].
Also Kothawadeet. al. [8] and Patilet. al. underlined the importance of space optimization methodologies, the trend being toward automation, like shown by Kim et. al.
Modern methods allow the user to simulate the effect of the cargo with consideration to the vehicle aerodynamics, the driver (the aggressivity of the driver), the influence of the road, maneuvers, and more importantly the effect of the cargo placement on the vehicle suspension durability, by monitoring the angles of the vehicle and the rate (Figure 1).

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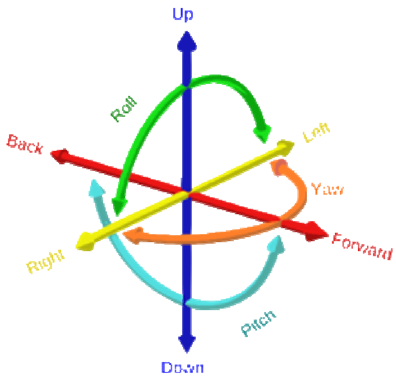

Figure 1. Monitored vehicle angles

2. METHODOLOGY

Using the AVL InMotion system, and its CarMaker software, a simulation was done, to research the influence of the placement of the cargo on the angles of the vehicle and the rate (yaw rate, pitch rate and roll rate) because they influence the ware of the suspension system directly.
The input data of the simulation are: the vehicle was chosen, a Mercedes van with back loading possibility only and a maximum load capacity of 1300kg (in the simulation, thecargo load was considered 1000kg and the 80 kg driver, like shown in Figure 2); the vehicle body and aerodynamics are presented in figure 3; the road was implemented in CarMaker by using segments, a straight line so that the vehicle can get up to 100 km/h, and then a series of left and right curves with decreasing radius like shown in figure 4; the driver was set to normal, with an allowed acceleration of 3 m/s, lateral acceleration of 4 m/s and a deceleration of -4 m/s; the imposed maneuvers were 100 km/h speed with a manual gear shifting by the smart driver.

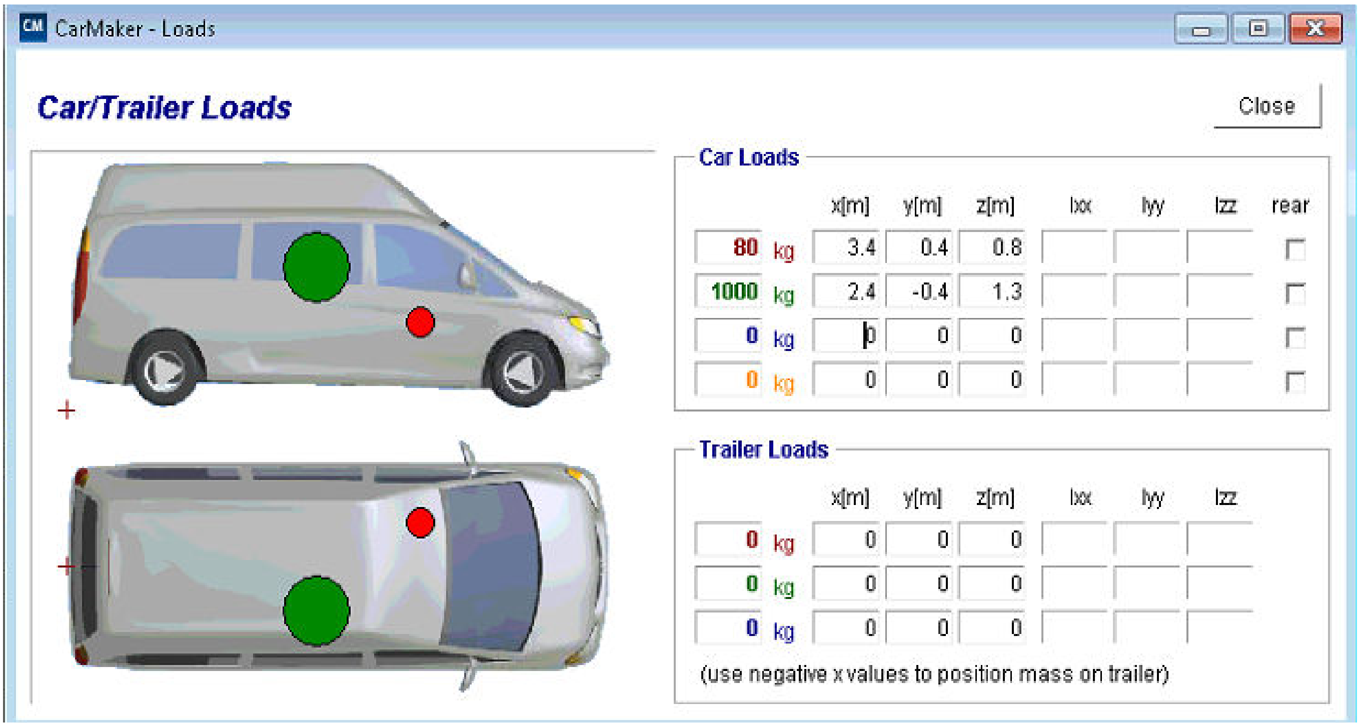

Figure 2. Chosen vehicle and loads

The cargo placement strategies are presented in figure 6.
Two planes were chosen, the upper plane at 1.3m and the lower at 0.8m (on the floor of the van) and three sections: the central section, the left section (placed at -0.4m) and the right section at 0.4m.

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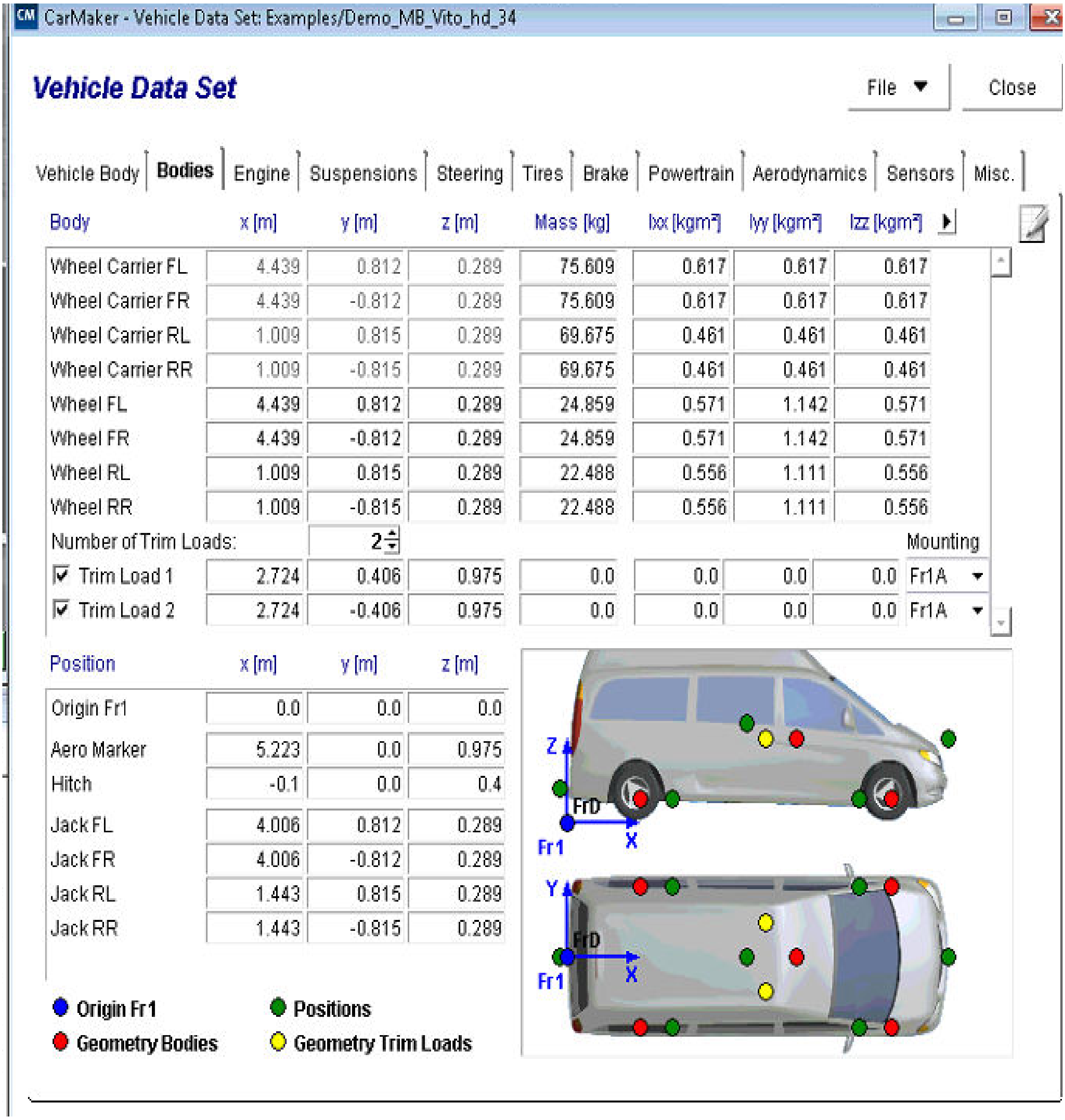

Figure 3.a Vehicle bodies

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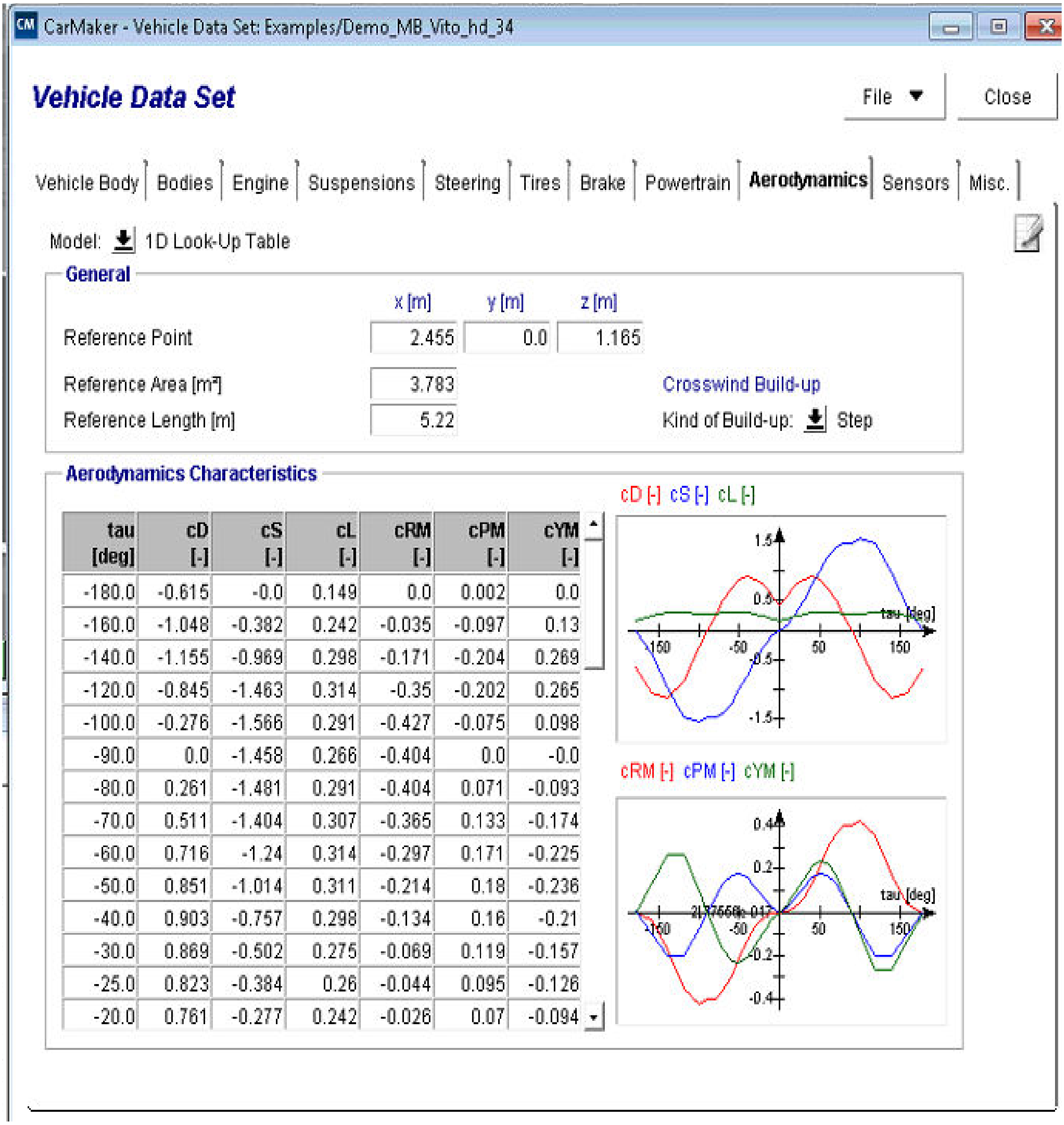

tau [deg]	cD [-]	cS [-]	cL [-]	cRM [-]	cPM [-]	cYM [-]
-180.0	-0.615	-0.0	0.149	0.0	0.002	0.0
-160.0	-1.048	-0.382	0.242	-0.035	-0.097	0.13
-140.0	-1.155	-0.969	0.298	-0.171	-0.204	0.269
-120.0	-0.845	-1.463	0.314	-0.35	-0.202	0.265
-100.0	-0.276	-1.566	0.291	-0.427	-0.075	0.098
-90.0	0.0	-1.458	0.266	-0.404	0.0	-0.0
-80.0	0.261	-1.481	0.291	-0.404	0.071	-0.093
-70.0	0.511	-1.404	0.307	-0.365	0.133	-0.174
-60.0	0.716	-1.24	0.314	-0.297	0.171	-0.225
-50.0	0.851	-1.014	0.311	-0.214	0.18	-0.236
-40.0	0.903	-0.757	0.298	-0.134	0.16	-0.21
-30.0	0.869	-0.502	0.275	-0.069	0.119	-0.157
-25.0	0.823	-0.384	0.26	-0.044	0.095	-0.126
-20.0	0.761	-0.277	0.242	-0.026	0.07	-0.094

Figure 3.b Vehicle aerodynamics

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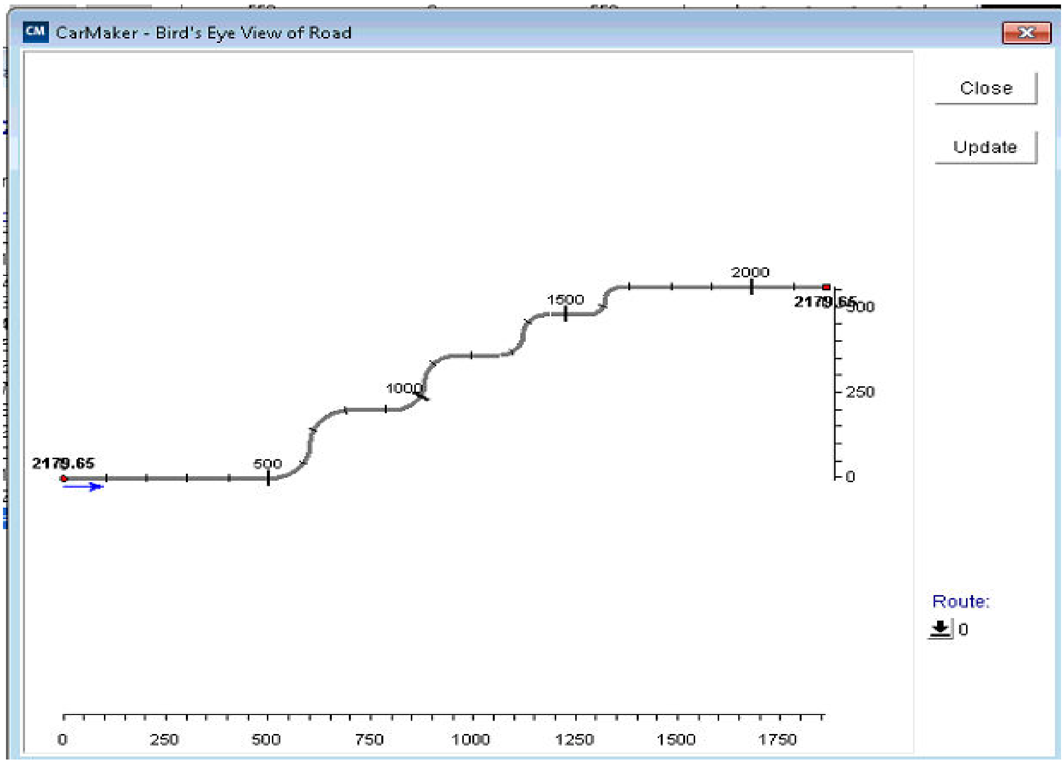

Figure 4. Implemented road in CarMaker

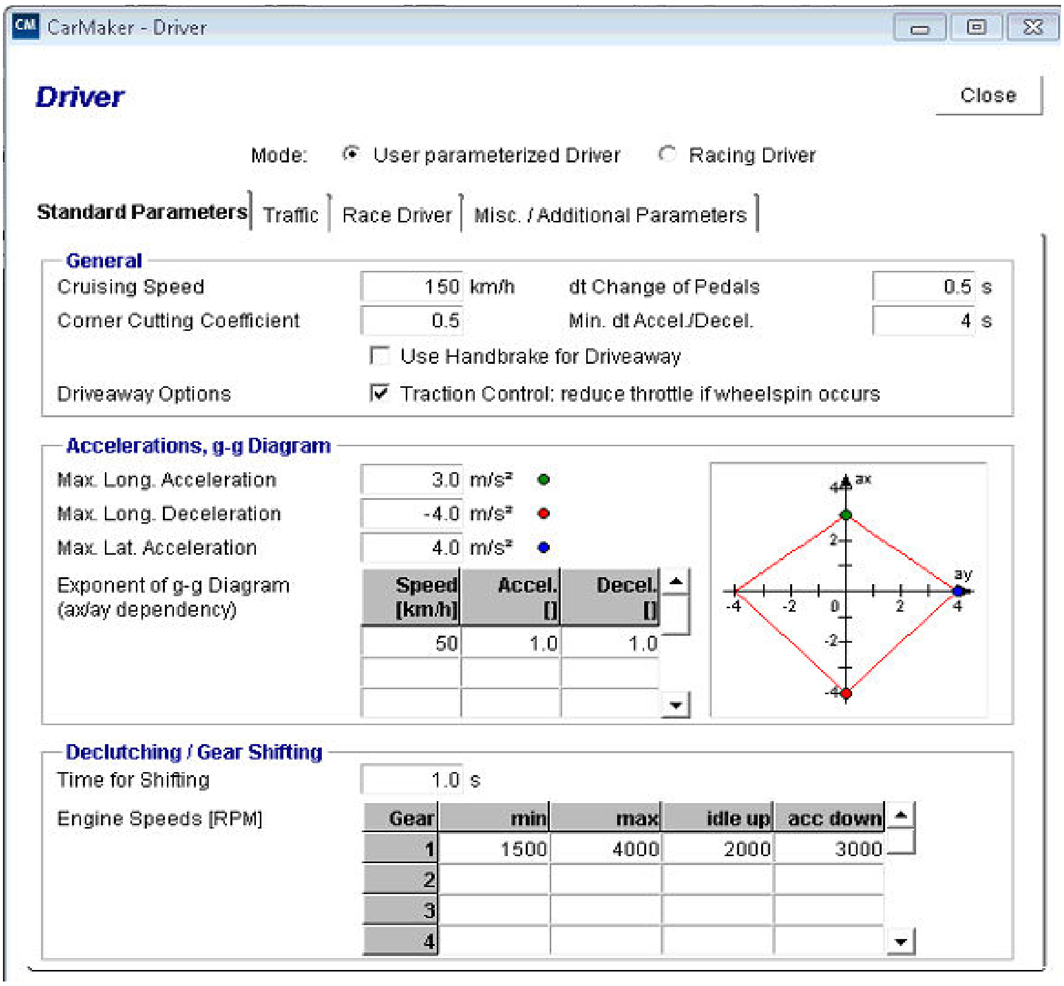

Figure 5.a Driver in CarMaker

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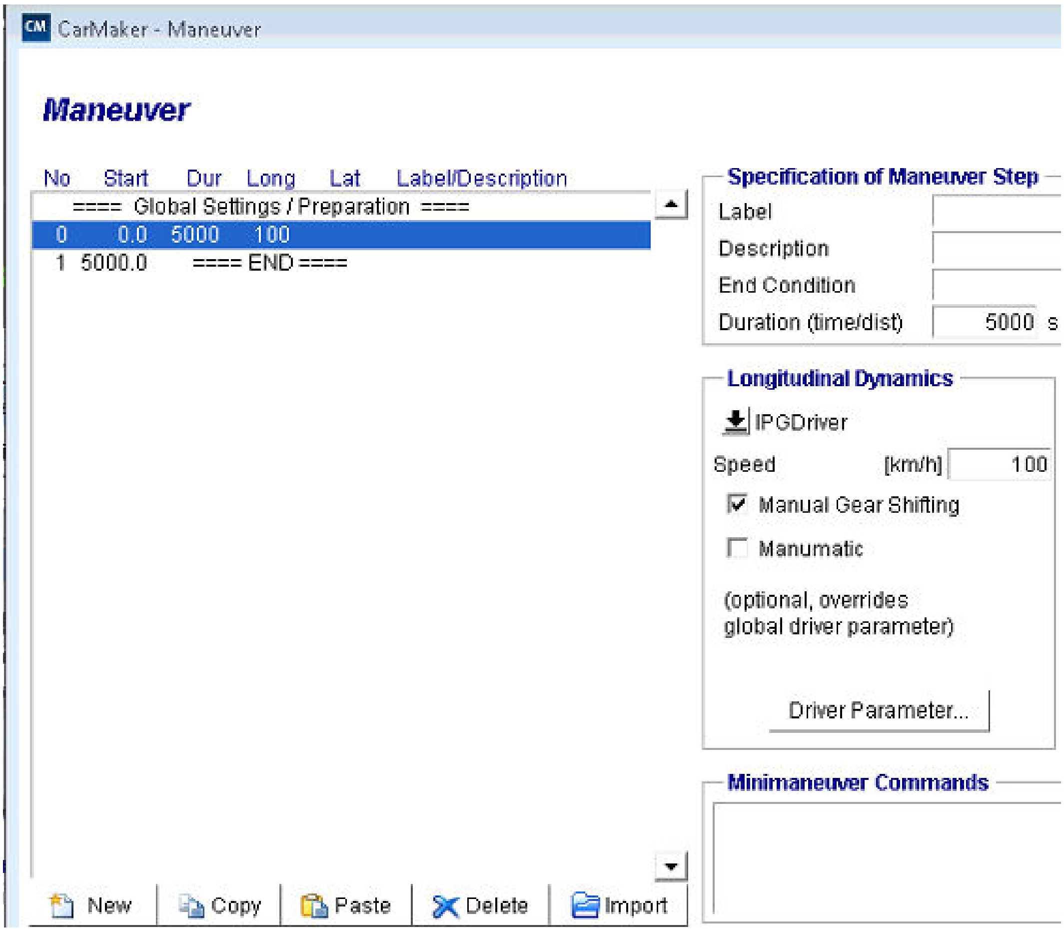

Figure 5.b Maneuvers in CarMaker

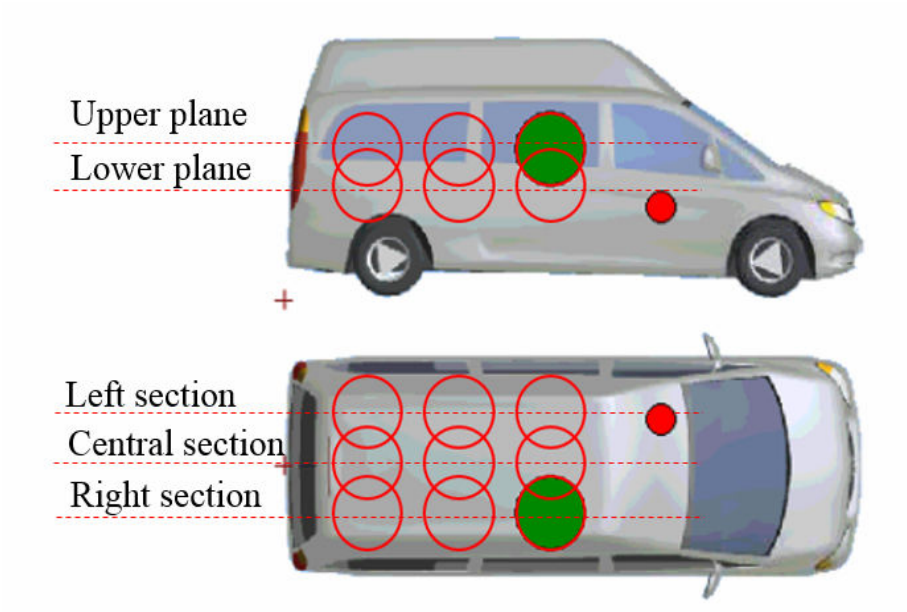

Figure. 6. Cargo placement strategies

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Table 1.
Study cases and the notation

Case number	Placement of the cargo (x,y,z coordinate) [m]			Case number	Placement of the cargo (x,y,z coordinate) [m]		
1	No load			11	1.7	0	1.3
2	1.7	0	0.8	12	2.4	0	1.3
3	2.4	0	0.8	13	1	0	1.3
4	1	0	0.8	14	1	0.4	1.3
5	1	0.4	0.8	15	1	-0.4	1.3
6	1	-0.4	0.8	16	1.7	0.4	1.3
7	1.7	0.4	0.8	17	1.7	-0.4	1.3
8	1.7	-0.4	0.8	18	2.4	0.4	1.3
9	2.4	0.4	0.8	19	2.4	-0.4	1.3
10	2.4	-0.4	0.8				

3. RESEARCH AND RESULTS

The first set of results that were extracted are presented in Figures 7 and 8: the Total damp friction force for all 4 wheels, for case (1) and (15) respectively.

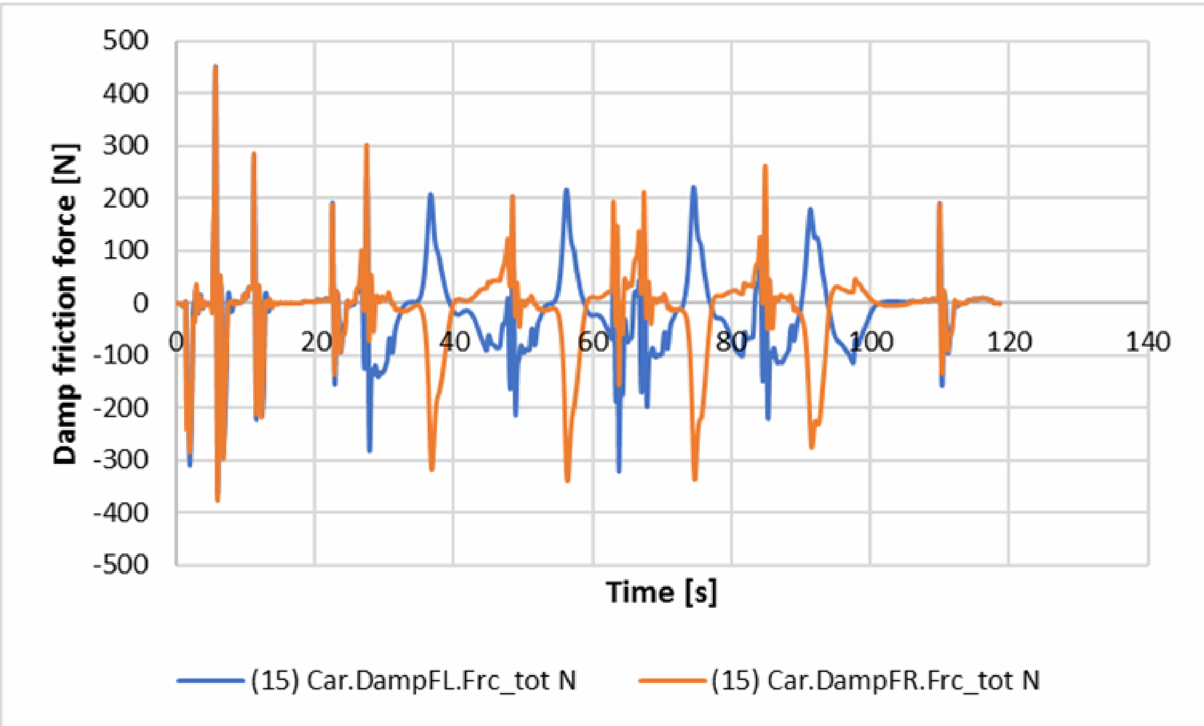

Figure 7. Total damp friction force for front wheels for case (15)

The most significant cases to compare are (1), (2), (9) and (15) because (1) is unloaded, (2) is a central low placed cargo, while (9) is a front right low placed cargo and (15) is a back left high cargo.

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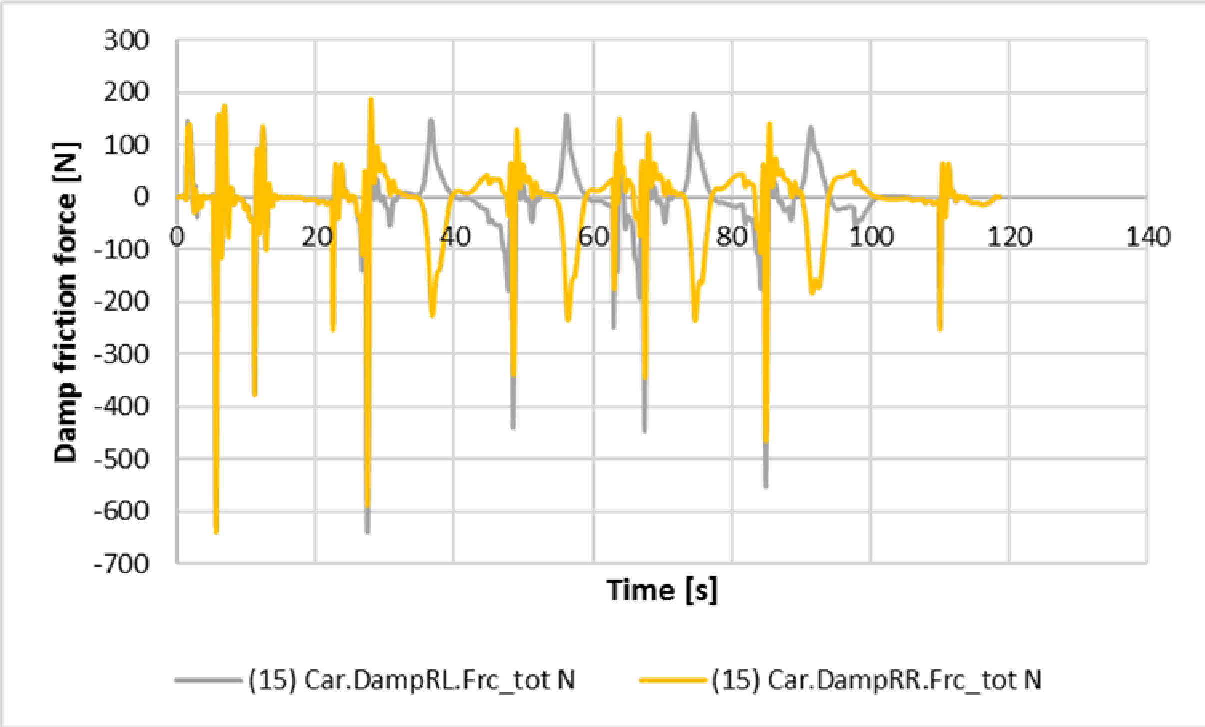

Figure 8. Total damp friction force for rear wheels for case (15)

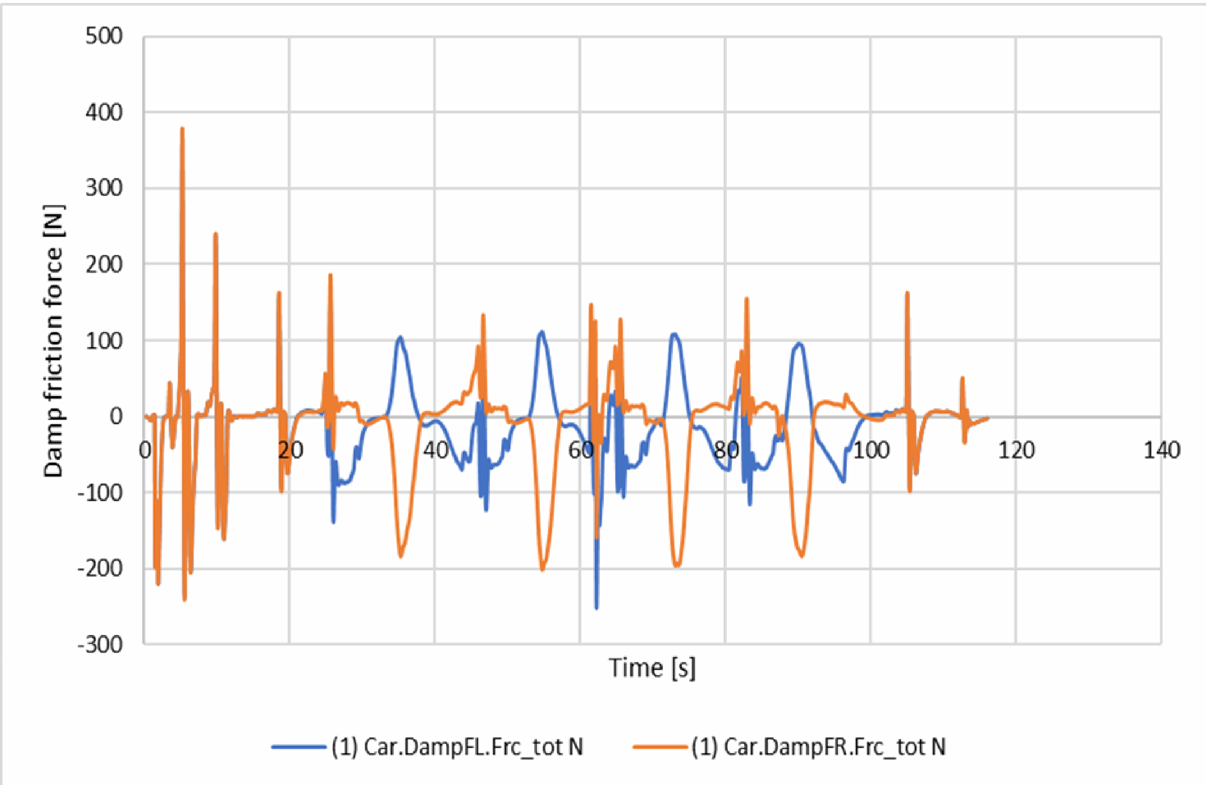

Figure 9. Total damp friction force for front wheels for case (1)

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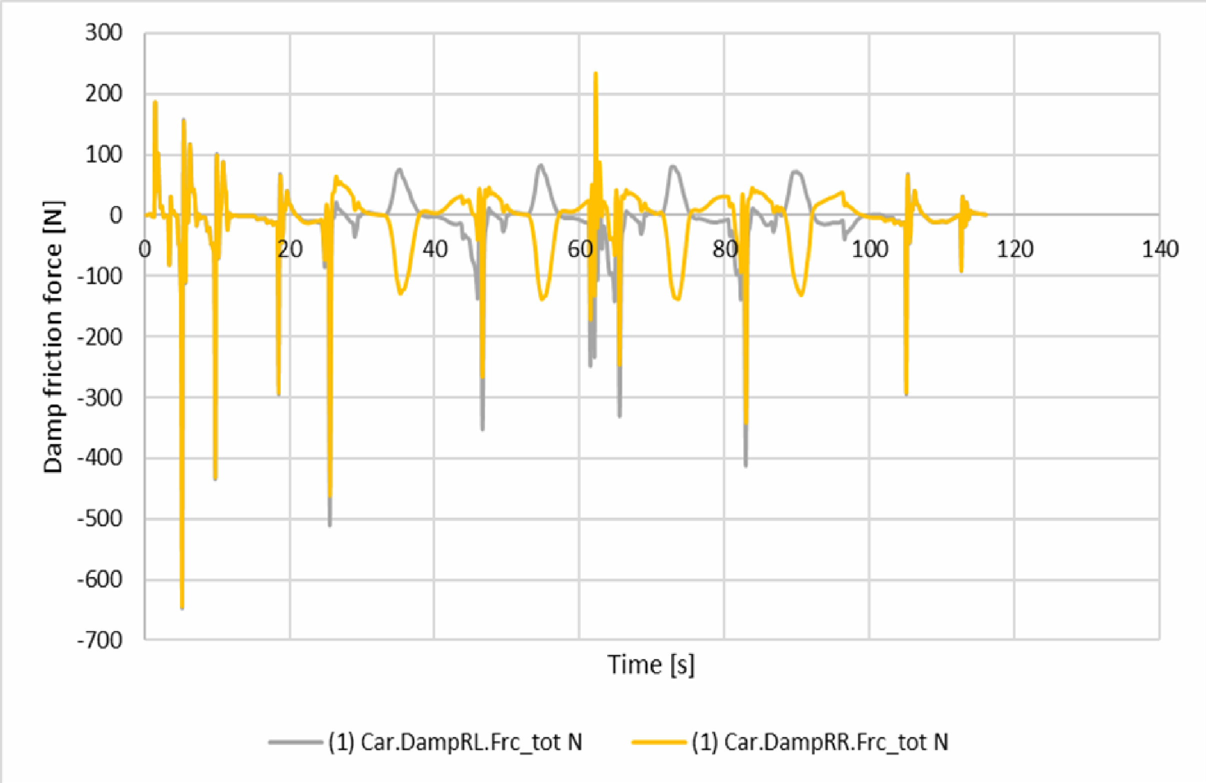

Figure 10.Total damp friction force for rear wheels for case (15)

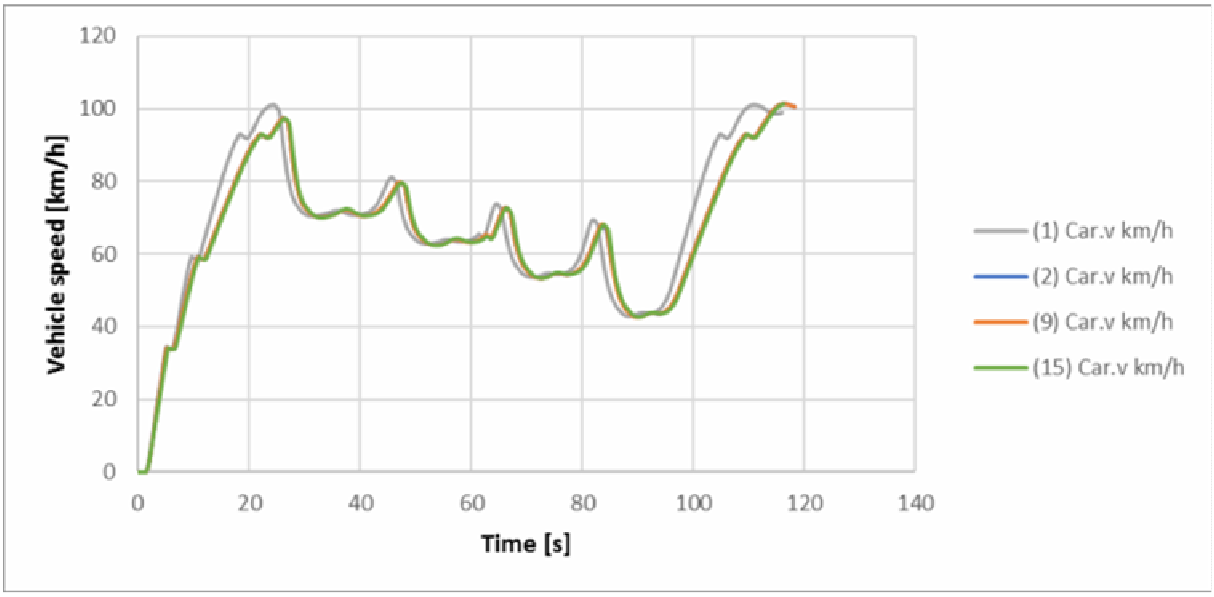

Figure 11. Vehicle velocity variation for cases (1), (2), (9) and (15)

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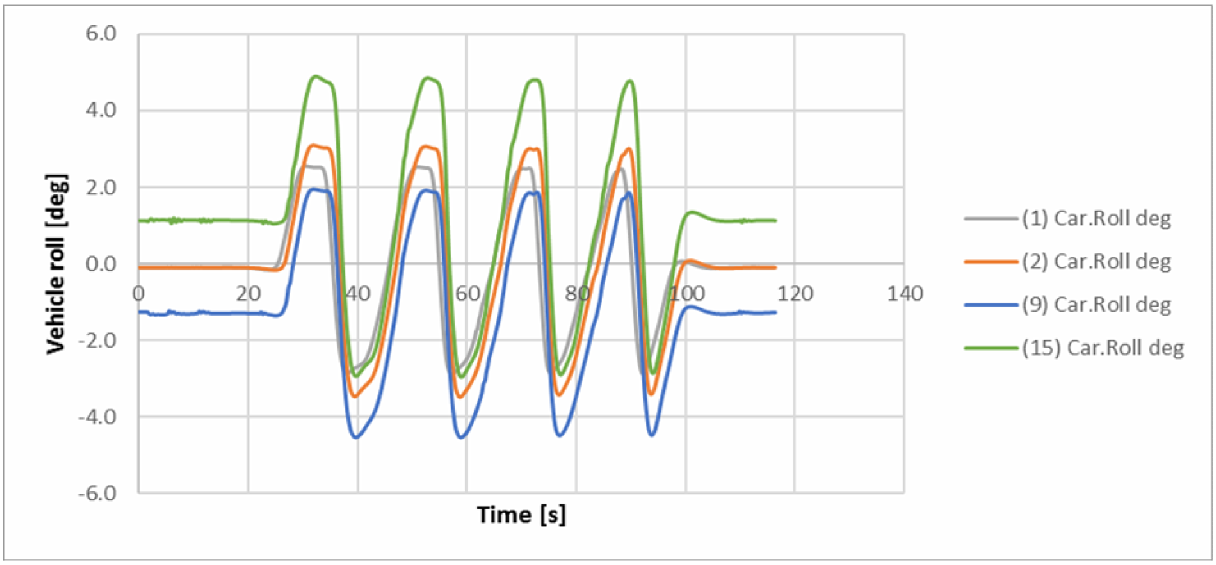

Figure 12. Vehicle roll angle variation for cases (1), (2), (9) and (15)

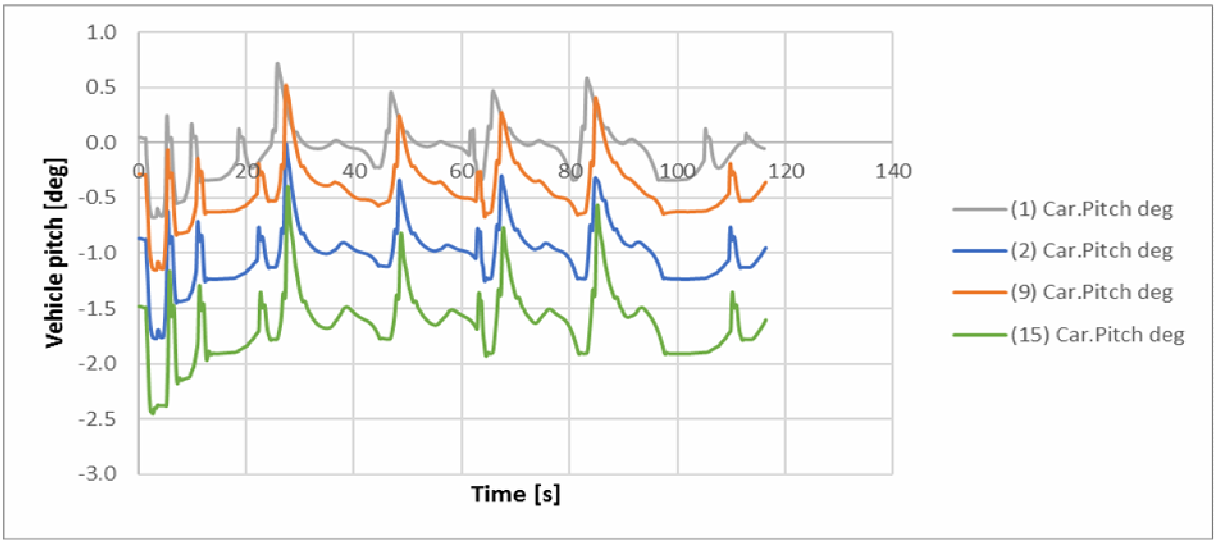

Figure 13. Vehicle pitch angle variation for cases (1), (2), (9) and (15)

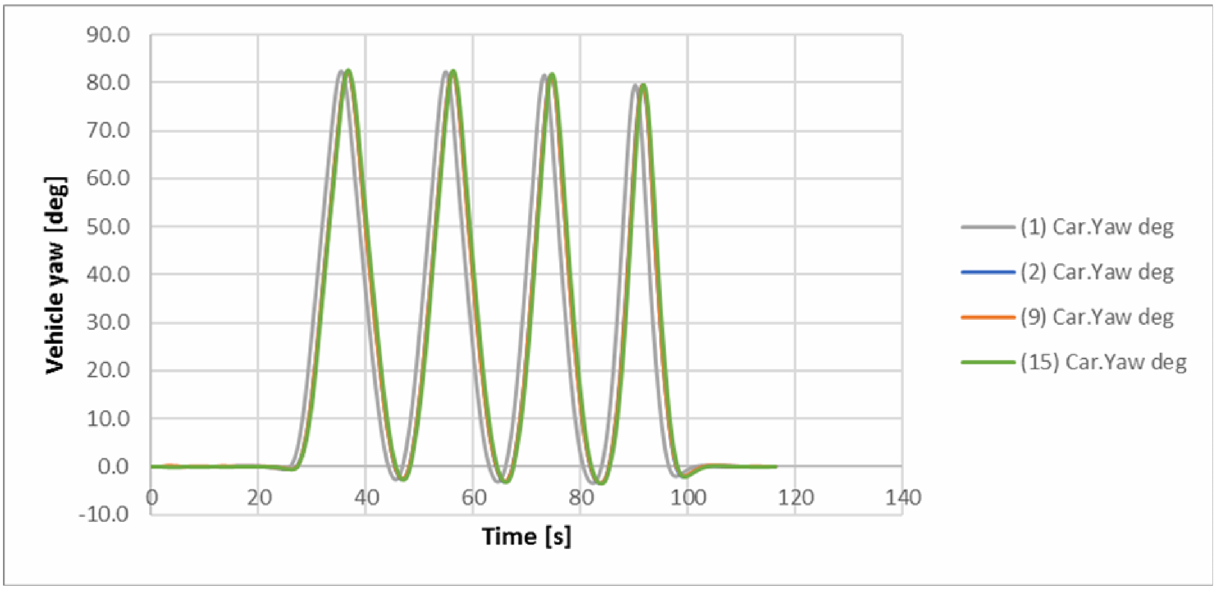

Figure 14. Vehicle pitch angle variation for cases (1), (2), (9) and (15)

4. CONCLUSIONS

AVL InMotion is a complex system that allows the user to insert data for simulation, while one of the components is real instead of virtual, making it a Hardware in the Loop simulation/test.
That would be the next step of the current research.
For the current simulations, it can be observed that the Total damp friction force has a similar variation in both cases (1) and (15), but due to the 1-ton load on case (15), the Total damp friction force on the Front Right wheel is bigger, from 380 N to 440 N for case (15).
For the Left wheels the variation is similar, but for case (15) the Total damp friction force doubles from 100 N to 210 N and the variation is more aggressive (has a peak).
Despite the different placement of the cargo, the velocity of the vehicle can be maintained the same by the smart driver, that is why only a variation between cases (1) and (2) can be observed.
Variations for cases (2), (9) and (15) overlap.
Due to the cargo weigh, the vehicle cannot reach the desired velocity (100 km/h) because the vehicle must take the first curve.
The vehicle roll angle variation was presented, and the conclusions are: the highest vehicle roll of 5 degrees is for case (15) because of the high placement of the cargo, while for a central low positioning (case (2)), the angle does not vary only with 0.6 degrees from the unloaded van (1).
By placing the cargo on the right like in case (9), the vehicle roll angle will be bigger on the negative side, but about the same as in case (15).
The influence of the cargo placement can be observed also in the vehicle pitch angle variation.
The smallest pitch of the vehicle is achieved for the empty van case (1). For case (9), because the cargo is placed close to the center of gravity of the van, the pitch varies from -1.2 to 0.5 degrees, followed by larger variations for cases (2) (from -1.7 to 0degrees) and (15) (from -2.49 to -0.4 degrees).
The vehicle pitch angle has an overlap for cases (2), (9) and (15) due to the added cargo, and the placement does not influence it, only in comparison to case (1).
As a general conclusion, the optimal placement of the cargo is low, as close as possible to the center of gravity of the vehicle and centered to the vehicle axis.
Even though if placed on the left or right, the roll angle can be corrected by the driver, but the vehicle velocity will be directly influenced.

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SIMULATION ENVIRONMENT FOR FATIGUE ANALYSIS OF CAR WHEEL RIM UNDER RADIAL FATIGUE TEST

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Abstract: *The importance of the wheel for the car is clear. The wheels with mounted tyres have to carry the vehicle weight and to handle with the steering system, providing the cushioning effect. The automobile wheel must be strong enough to perform the above functions supporting static and dynamic loads. Durability design and analysis is an essential element in achieving the mentioned requirements and involves a multi-disciplinary approach. If it is designed a new wheel rim, the prototype should be passing the radial and cornering fatigue test. This paper presents the simulation environment for car wheel rim under radial fatigue test with the purpose to evaluate the wheel rim strenght. This paper reviews also, some modern durability philosophies and the computer based tools available to meet the needs of design, test and development engineers.*

Key-Words: *Steel wheel rim, fatigue analysis, stain life, static analysis.*

1. INTRODUCTION

The wheel rim is a transfer element between the tire and the vehicle. The main requirements of an automobile wheel rim are the following: a) it should be as light as possible so that unsprung weight is least; b) it should be strong enough to perform the carriage function of the vehicle weight supporting static and dynamic loads; c) it should be possible to remove or mount the wheel rim easily. These wheel rims are the most important for cars to run. They must be very safe because they are related to life. Due to the continued rotational motions, the wheel rims of automobiles receive repetitive stresses. If this period becomes long-term, a fatigue fracture can take place [1][2]. Steel and light alloy are the foremost materials used in a wheel rim, however some composite materials together with glass-fiber are being used for special wheel rims. Stamped rims are made of steel, a light constructional material, which is pressed and then welded. They are also prepared because steel has poor corrosion resistance. Finally, steel wheel rims are usually coated with enamel or varnish. The main advantages of steel wheel rims are: a lower price than alloy wheels; flexibility, especially when running into a hole in the road, or when running over objects lying on the road; they absorb shocks and protect the car's body from deformations, which is very important for driving safety. The main disadvantage of steel wheel rims is their weight. The main objective of this article is to evaluate the durability of a steel wheel rim of the car under radial fatigue test using finite element method [3][4]. Fatigue analysis is preceded by finite element model validation using theoretical and experimental modal analysis. This changeable load is applied on the wheel rim structure considering the initial stress field produced by the tire pressure and the inertial forces generated by the rotation of the wheel.

2. THEORETICAL AND EXPERIMENTAL MODAL ANALYSIS

In the automotive industry, experimental modal analysis is used in a new vehicle development to realize different components and assemblies for road dynamic loads. Testing is important to cover some aspects in simulation process using finite element method. Theoretical and experimental modal analysis of the wheel rim implies the following steps: construction a finite element model of the rim to deduce modal parameters, experimentally determination of the modal parameters of the wheel rim under test, correlation of experimental measurement with simulation results [5][6].

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The simulation and experimental studies are carried out with free-free boundary conditions. The 3D model of the wheel rim was created using CATIA and the file was exported in STEP format into ANSYS program. The mesh is made with 10-node tetrahedral structural solid elements using an element edge length of 5 mm. The total number of nodes and elements is 196894 and 197083 respectively. The wheel rim material is Dual Phase Steel DP600-HR with elastic modulus E=2e11 Pa, density ρ=7850 kg/m3, Poisson's ratio υ=0.3, tensile yield strength YS=4.38e8 Pa and tensile ultimate strength UTS=6.16e8 Pa.
The experimental modal tests and analyses are carried out using vibration measurement and modal analysis software PULSE 12 (Brüel & Kjær). The test procedure comprises data acquisition from impulse hammer, curve fitting of measured frequency response functions for modal parameter estimation.
The test bench is shown in Figure 1 and the accelerometer positions are presented in Figure 2.

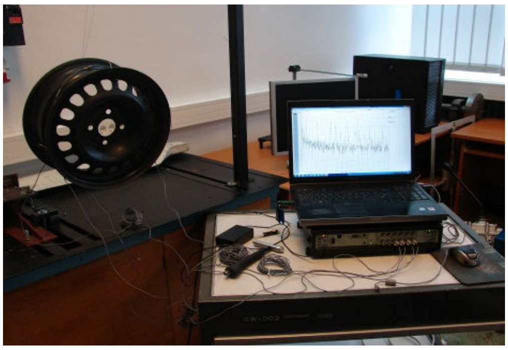

Figure 1. PULS 12 (B& K) test bench

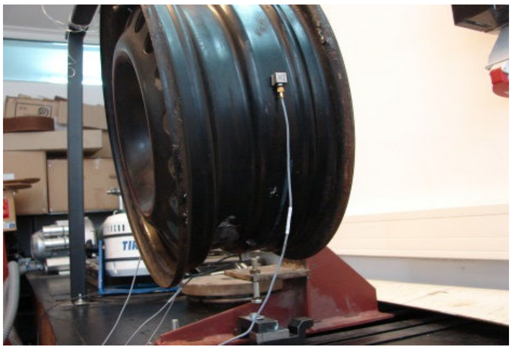

Figure 2. Accelerometer positions on wheel

Four theoretical modes (1, 3, 7 and 16) are shown in Figure 3, Figure 4, Figure 5 and Figure 6. Also, the natural frequencies obtained by test are specified for every displayed mode.

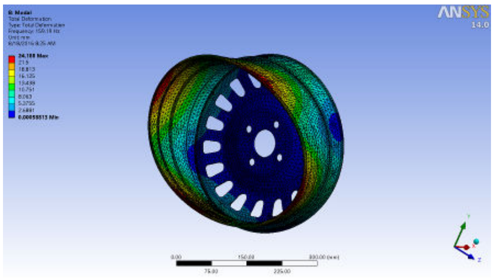

Figure 3. Mode 1 - 159 Hz (test: 178 Hz)

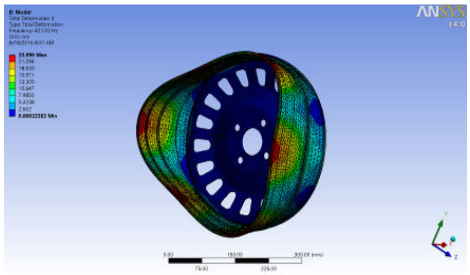

Figure 4. Mode 3 - 421 Hz (test: 422 Hz)

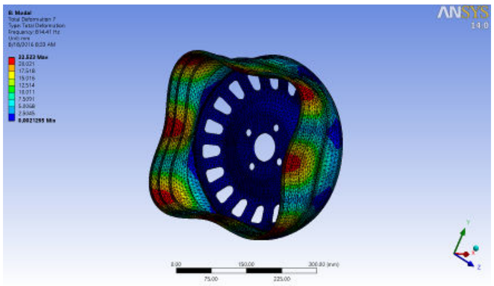

Figure 5. Mode 7 - 814 Hz (test: 814 Hz)

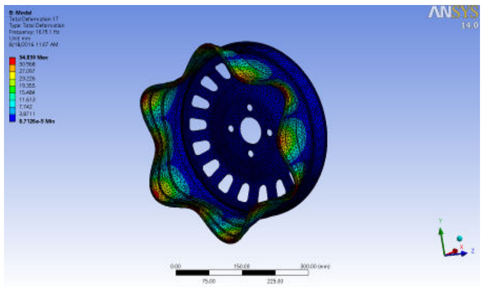

Figure 6. Mode 16 - 1619 Hz (test: 1650 Hz)

The four accelerometers (type 4507 B) are mounted in four points located at 90° to each other. With the modal hammer (type 8206-003) kicks are applied in points located between accelerometers. The force generated by the hammer was considered as input quantity and as response was selected the speed vibration so the recorded accelerations had to be integrated.
Some of the obtained results, natural frequencies and viscous damping coefficient, using information in measurement point, are shown in Table 1.

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Table 1.
Some natural frequencies and viscous damping coefficients obtained by test

Quantity	0°	90°	180°	270°
f [Hz]	178	178	178	178
ζ [%]	0.0132	0.0131	0.0131	0.0131
f [Hz]	422	426	422	426
ζ [%]	0,00448	0,00511	0,00591	0,00402
f [Hz]	814	814	814	814
ζ [%]	0,00061	0,00067	0,00066	0,00076

Performing experimental modal analysis helped in certifying the subsequently simulation results.
The validated FE model will be used now to evaluate the fatigue life of steel wheel rim under radial fatigue test.

3. ANALYSIS OF THE STEEL WHEEL RIM UNDER RADIAL FATIGUE TEST

3.1. Fatigue analysis using FEA platform

The finite element is a mathematical numerical method for solving ordinary and partial differential equations having the capability to solve complex problems represented in differential equation form. Classical methods alone usually cannot provide adequate information to determine the safe working limits. At the moment the finite element method is used almost to solve structural problems in engineering [7]. Fatigue is an important problem for components subjected to repeated loadings and it is a difficult design challenge to resolve. Practice has shown that large percentage of structural failure is attributed to fatigue and it continues to be the focus of fundamental and applied research. Variable loadings applied on a component or structure at stresses allowable for static loadings may cause the appearance of one crack. Under cyclic loading these cracks may continue to grow and can generate a failure when the remaining structure can no longer carry the loads. A methodology to predict crack initiation in the fatigue damage assessment of metallic structures typically used in ground vehicle industry is presented in this paper. The finite element model is integrated with a notch stress–strain analysis method. Local loads are modeled with linear elastic FE analyses considering the radial load as being zero-based fatigue loads. The computed stress–strain response is used to predict the fatigue crack initiation life using effective strain range parameters.

3.2. Wheel rim model

Figure 7 and Figure 8 show the CAD model of the wheel rim and FE model respectively. The meshing was performed using the mesh generate options in the ANSYS Workbench.

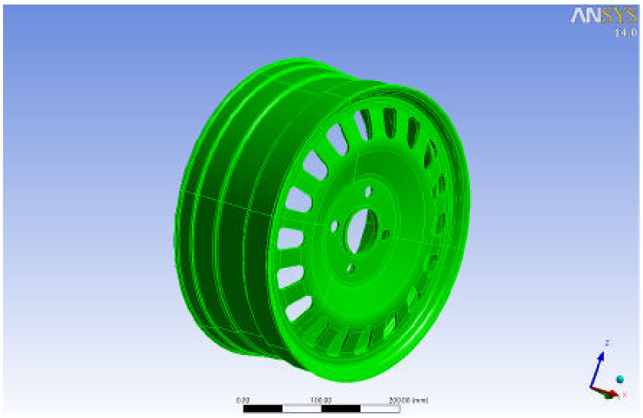

Figure 7. CAD model of the wheel rim

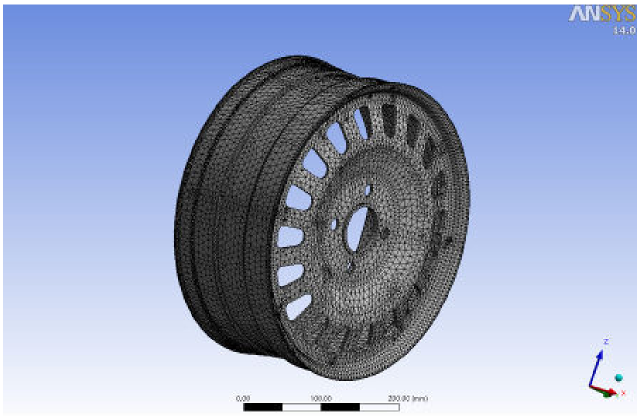

Figure 8. FE model of the wheel rim

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3.3. Boundary conditions and loadings

The FE wheel rim model is fixed around the holes through which the screws pass to clamp the wheel rim on the disk spindle. The tire characteristics are 195/65R15. The rolling diameter is 634.5 mm. It is assumed that the car speed is 80 km/h which corresponds to an angular speed of the wheel of 75.26 rad/sec. The tire pressure is 30 psi (0.207 N / mm2) and it is applied on the outer surface of the wheel rim. The total weight of the car is 1600 kg and the radial reaction force on the wheel rim is 3924 N, applied on the surfaces corresponding to the tire bead seats. Figure 9 shows the following boundary conditions and loads: fixed surface around the clamped holes, inertial load and tire pressure.
Figure 10 presents the way considered to apply the radial load.

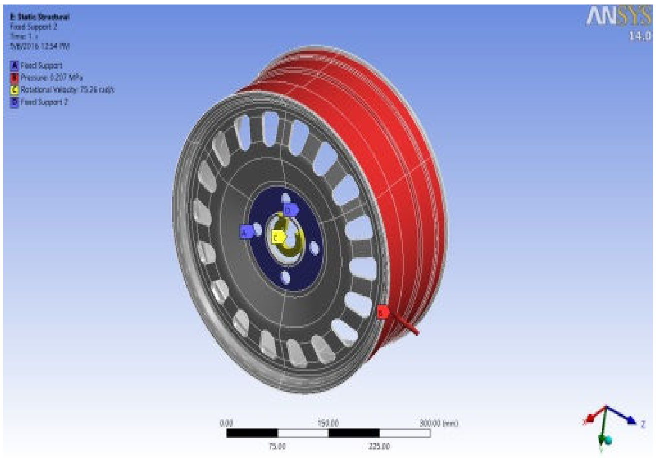

Figure 9. Boundary conditions

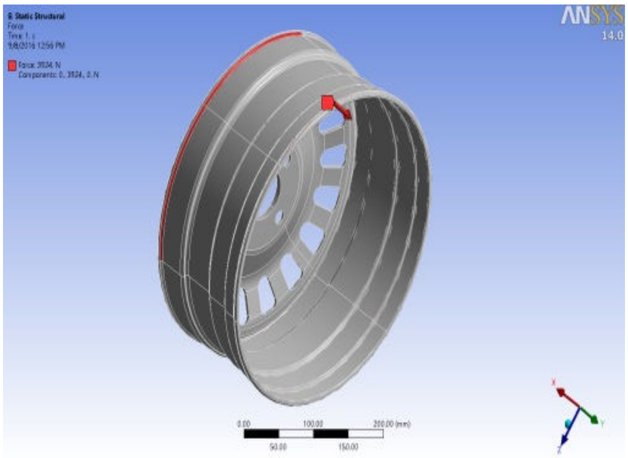

Figure 10. Application of the radial force

3.4. Fatigue analysis process and results

The total weight is distributed on all four wheels and it is balanced with vertical reaction forces from the road through the tires. These loads constantly compress the wheel rims in radial direction. The radial load becomes a variable load with the rotation of the wheel while the car is running. The evaluation of wheel rim fatigue strength under cyclic radial load is an important performance characteristic for structural integrity. Fatigue life prediction using the stress approach is mostly based on local stress, because it is not possible to determine nominal stress for the individual critical areas. Fatigue analysis is done in two stages. In the first step, the static elastic tensions are evaluated. These are generated by the simultaneous action of inertial forces generated by the rotation of the wheel and the tire pressure. Due to the action mode of the mentioned loads, the maximum von Mises stress is lower when the two loads simultaneously act. In the next step, a cyclic radial load is applied, generated by the rotation of the wheel. In this paper it is analyzed the car wheel rim fatigue under cyclic radial load using the local strain-life method. Unlike the stress-based fatigue life prediction, in which only elastic stresses and strains are presented, the strain-life method takes into consideration the local cyclic plastic deformations.
Most components may appear to have nominally cyclic elastic stresses, but notches, welds or other stress concentrations may present local plastic deformations. The wheel rims are made of steel plate which is in the first time stamped (pressed) and then the resulted components are welded.

The relation of the total strain amplitude εa, total elastic amplitude ε_a^e, total plastic amplitude ε_a^p and the fatigue life in reversals to failure (2Nf) is expressed in the subsequent form:

$$\varepsilon_a = \varepsilon_a^e + \varepsilon_a^p = \frac{\sigma_f^{'}}{E}(2N_f)^b + \varepsilon_f^{'}(2N_f)^c \qquad (1)$$

The strain-life data are the following: $\sigma_f^{'}$ - fatigue strength coefficient, b - fatigue strength exponent, $\varepsilon_f^{'}$ - fatigue ductility coefficient, c - fatigue ductility exponent.
Schematic of a total strain-life curve is shown in Figure 11.
The strain-life parameters for the car wheel rim material, Dual Phase Steel DP600-HR are shown in Table 2.

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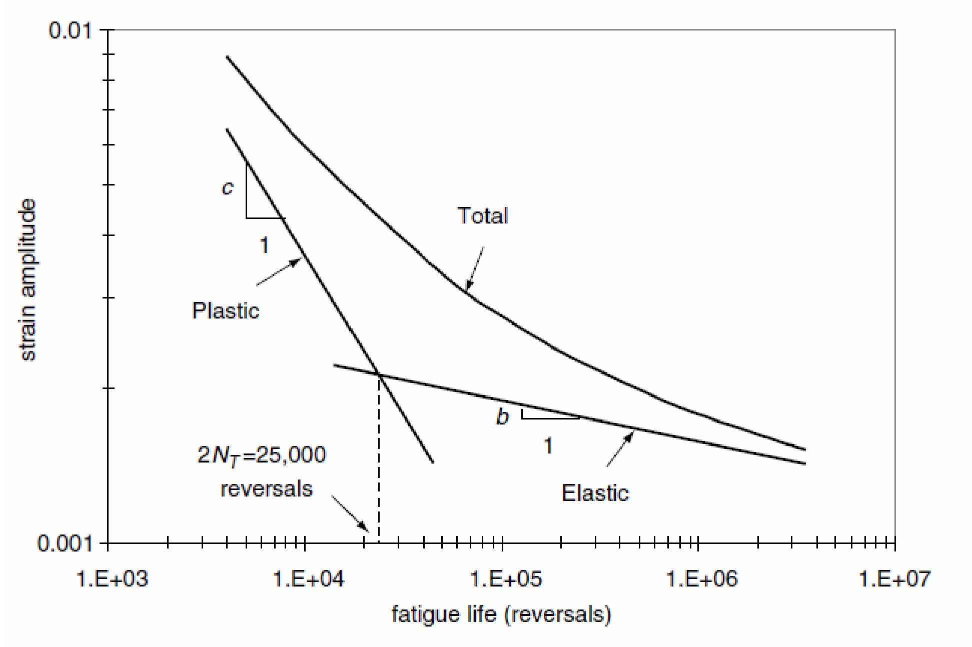

Figure 11. Schematic of a total strain-life curve

Table 1.
Strain-life parameters for Dual Phase Steel DP600-HR

Table of Properties Row 10: Strain-Life Parameters

	A	B	C	D	E	F
1	Strength Coefficient (Pa)	Strength Exponent	Ductility Coefficient	Ductility Exponent	Cyclic Strength Coefficient (Pa)	Cyclic Strain Hardening Exponent
2	1.007E+09	-0.087	2.441	-0.757	9.04E+08	0.114

Fatigue analysis is done using Fatigue Tool of the Ansys Workbench 14.0 platform. Over the static loading represented by the inertial forces and the tire pressure it overlaps the pulsating radial force. This operation is achieved using Solution Combination facility. Figure 12 presents the equivalent von Mises stresses, maximum and minimum principal stresses and maximum shear stresses in the wheel rim. The extreme stress values are in the region of one ventilation holes.

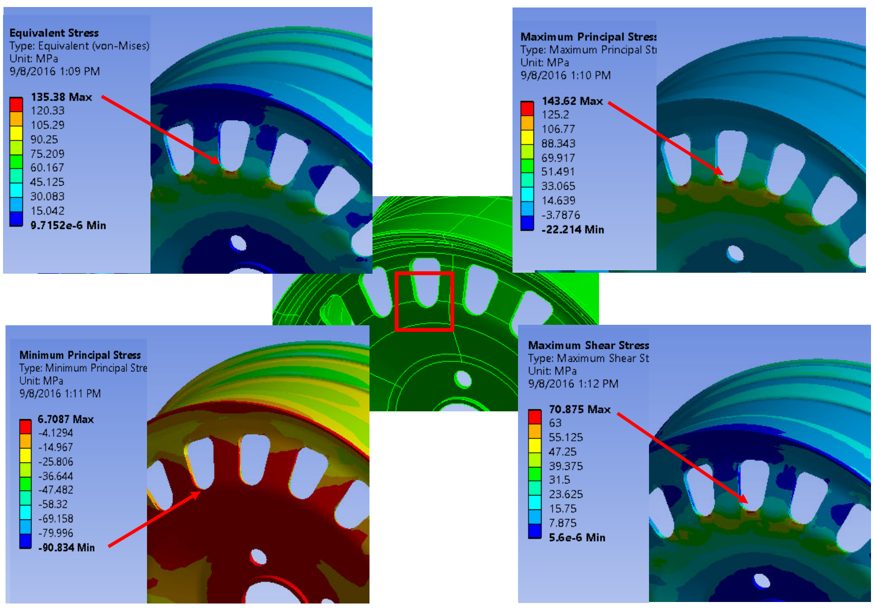

Figure 12. Equivalent, Maximum and Minimum Principal and Maximum Shear Stresses

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Details of Fatigue Tool are the following: Analyze Type → Strain Life, Mean Stress Theory → Morrow, Stress Component → Max Principal, Infinite Life → 1e9 cycles. It is considered that 1 cycle corresponds to 1 rotation of the wheel. The fatigue analysis result is shown in Figure 13, the minimum life being 8.83e8 cycles in the region of one ventilation holes.

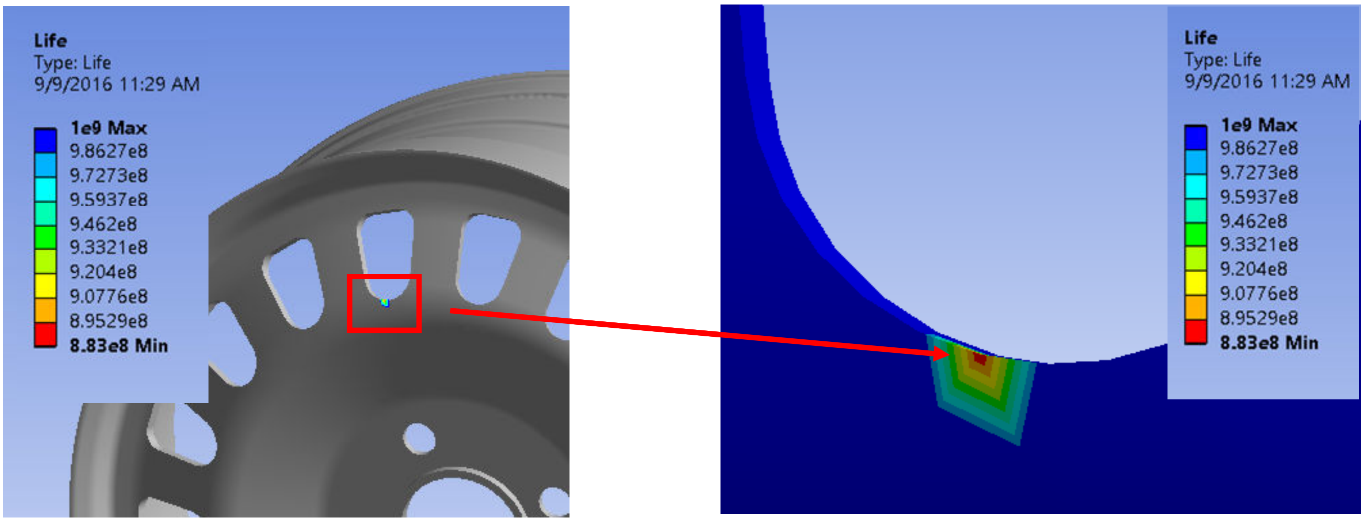

Figure 13. Fatigue analysis results

During the static analysis of the steel wheel rim, it was discovered that the maximum stress concentration occurred in the region of one ventilation hole. It is observed that when the wheel rotates at a speed of 718.7 rpm with tire pressure of 30 psi (0.207 N / mm2) and under the radial load of 3924 N, the fatigue crack begin to propagate at the point of maximum stress concentration.
The fatigue material properties are based on uniaxial stresses but in the reality stress fields are usually multiaxial. Biaxiality indication is defined as the ratio between the principal stress smaller in magnitude and the larger principal stress. The principal stress nearest zero is ignored. A biaxiality of zero corresponds to uniaxial stress, a value of –1 corresponds to pure shear, and a value of 1 corresponds to a pure biaxial state. Figure 14 shows biaxiality indication under mentioned loads.

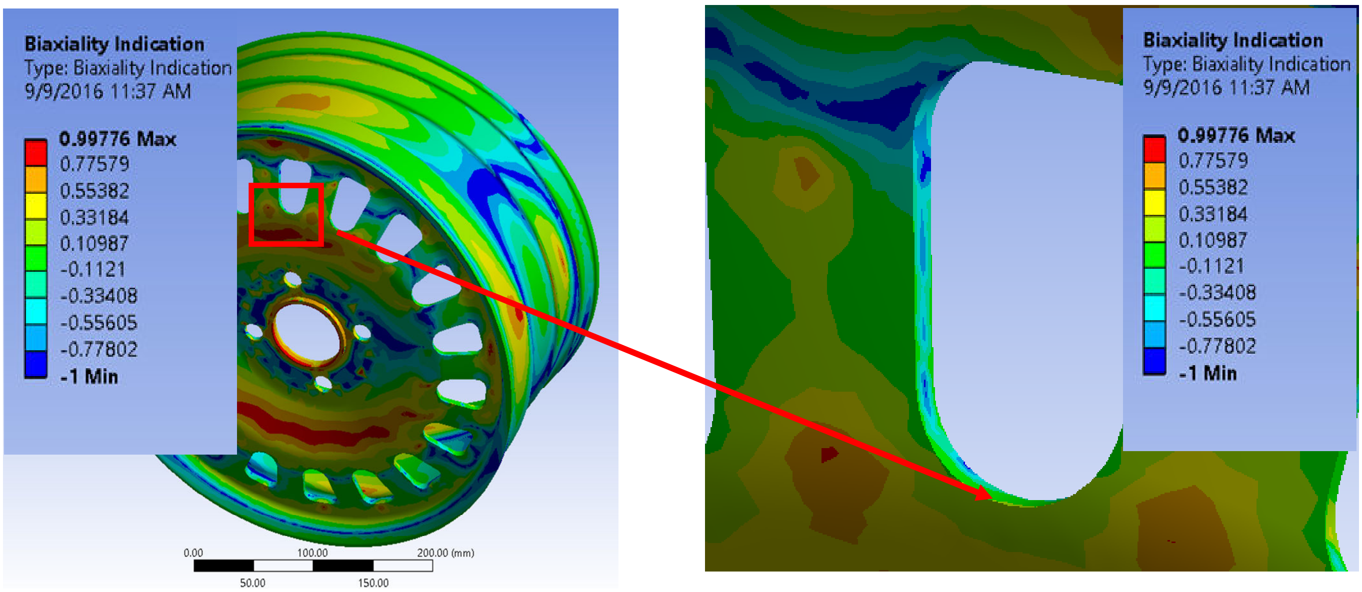

Figure 14. Biaxiality indication

As it can be seen, the preponderance of this model is under a pure uniaxial stress.
The most damaged point occurs at a point of predominantly uniaxial stress.

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It results as fair to use the steel properties obtained by uniaxial fatigue testing of a DP600-HR specimen. Fatigue Sensitivity shows how the fatigue results change as a function of the loading at the critical location on the model, Figure 15.

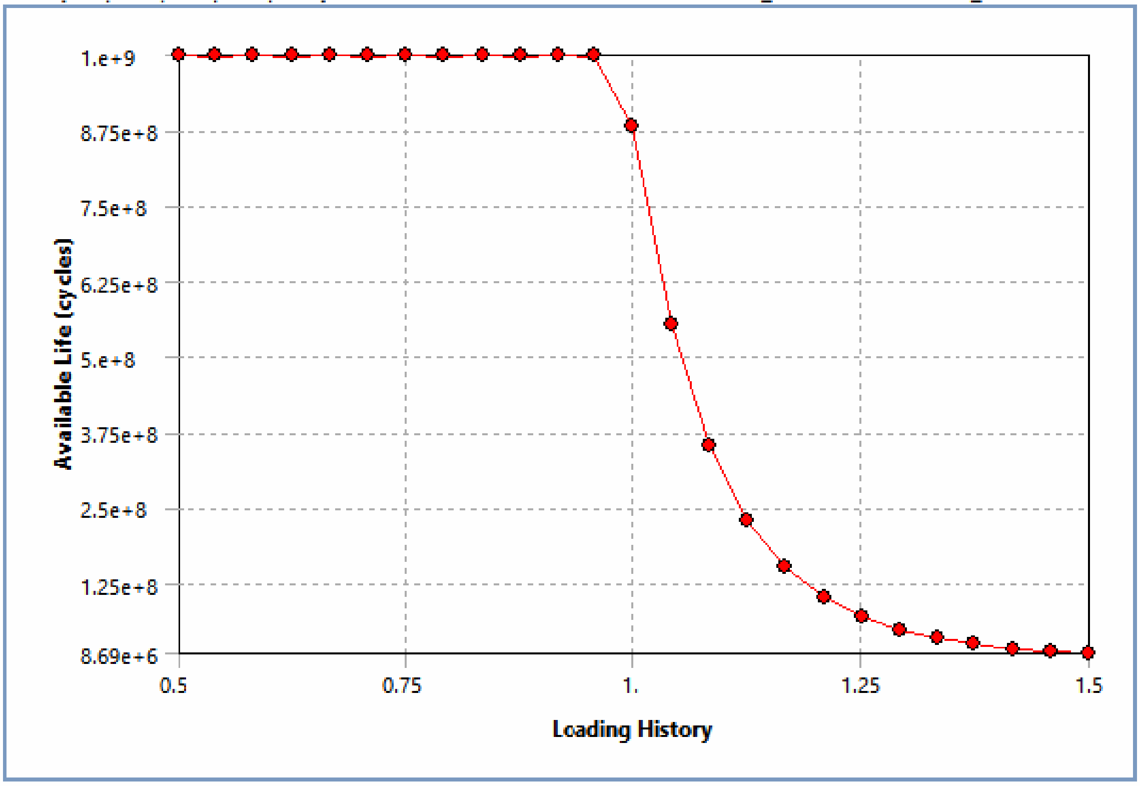

Figure 15. Fatigue sensitivity

In this paper the sensitivity is determined for the life if the FE load was 50% of the current load up to 150% of the current load. A value of 100% corresponds to the life at the current loading of the model. The fatigue life for current load is 8.83e8 cycles, and decreases to 8.69e6 cycles (10161%), if the load increases to 150%.

4. CONCLUSIONS

The radial fatigue test is intended to find the structural performance of a wheel rim for normal use on passenger cars, light trucks and multipurpose vehicles. The wheel rim has to withstand repeated radial loading for a defined number of cycles in order to pass the test.
This paper presents a computational methodology for fatigue estimation of the wheel rim subjected to the repeated radial loading. In the first step it is made an experimental modal analysis to validate FE model and to confirm the subsequently simulation results.
The validated FE model will be used now to evaluate the fatigue life.
The fatigue life of a wheel rim is calculated from the stress values obtained from static analysis, based on the local strain approach in conjunction with linear elastic FE stress analyses.
The stress–strain response at a material point is computed with a cyclic plasticity model coupled with a notch stress–strain approximation scheme.
The predicted failure locations are in the same sections to the real crack initiation regions and are reliable with other reported analysis.

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ISSN 2457 – 5275
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Frequency: Quarterly
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Format: online, English
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Type: Open Access
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Summary – on December 31, 2018

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