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MULTISTAGE GEAR PROJECT OF THE ELECTRICAL CAR SGR2016 FOR THE SILESIA GREENPOWER ORGANIZATION.

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Abstract: The subject of the paper, titled 'Multistage Gear Project Of The Electrical Car SGR2016 for the Silesian Greenpower organization', is an analysis of a chain vibrations in drive gear of SGR 2016 bolide. The exact moment of gear switch presents the greatest threat of a stability loss. Tests have been conducted in 2 stages. The first stage was the determining of derailleur setting. The adjusted component was a mechanism responsible for increasing the drag movement of derailleur cage. Adjustment of this element caused change in vibration characteristics in different system parts. That element was adjusted so oscillations were minimized. The second stage of studies was pinpointing the correct rotational speed, at which gears should be changed, using previously defined derailleur adjustment. Speed, at which the gear switch takes place, was chosen so the vibrations are lowest possible. Oscillation amplitude was recorded using high speed camera, and the data was analyzed using a dedicated High Speed AVI software. Speed measurement had been done using magnetic field sensor. All studies have been conducted on inertial system, simulating real life race conditions. Comparing all results, optimal driving parameters in terms of drive gear work stability, have been defined.

Keywords: DRIVE GEAR, VIBRATION, VEHICLE, DERAILEUR

1. Introduction

For a long time, constructor had a problem with an engine torque transmission. It was solved in numerous ways, each one had its advantages and disadvantages. Reliability is the most important condition, which all systems must fulfill. In case of a breakdown it is necessary to stop entire system. That causes extensive work breaks. Gears should be constructed in a manner that prevents such occurrences.

In this paper a drive gear in Silesian Greenpower electrical vehicle SGR 2016 was investigated. Silesian Greenpower is an interdisciplinary student organization that constructs and develop electric vehicles. Bolides constructed by this organization participate in Greenpower education Trust races.

Up to this point, the team used belt transmission. Race rules allowed outside help during takeoff, this gave merit to this solution. Change in these rules forced the team to seek new solutions to this problem. Many different ways of solving this problem were taken into consideration, among them, chain gear showed great promise. This kind of transmission has great starting properties, thanks to the ability to change gears during the race. This option was unavailable using the previous transmission.

During test phase of implemented chain gear, several issues arose. The greatest flaw of this solution was frequent stability loss. This could result in drivers safety being compromised. In every real system, there are vibrations present. In considered case, both the vehicle chassis and the transmission vibrate. There are certain speeds at which vibrations can interfere with each other causing effect similar to resonance. This phenomenon is most visible in transmission strand. Subsequently,

this phenomenon will be referred to as resonance. Shifting gears during resonance speeds may result in stability loss.

Deliberations were started with literature overlook. Most publications regarding chain gears show only single geared transmissions. Because of that most obtained information come from experts. The aim of our research is a modernization of chain gear, resulting in smoother and failure-free operation. [2]

Transmission ratio can be calculated from formula 1.1:

$$i = \frac{z_1}{z_2} \quad (1.1)$$

Where:

i – ratio [1];

$z_{1,2}$ - number of teeth on smaller and larger wheel [1]; [1]

Mounted gear has the ability to change transmission ratios. The chain is being moved from one gear to the other on the powered wheel. Powering wheel has a constant teeth number of 11. Passive wheel cassette has two gears mounted on it. The lower gear has 27 teeth and higher 23 teeth. Available ratios are:

$$i_1 = \frac{11}{27} \approx 0,41$$

$$i_2 = \frac{11}{23} \approx 0,48$$

2. Studies methodology description

Tests were conducted on a specifically designed test bench, which simulated conditions on the track. Vibrations were registered with a highspeed camera and analyzed using High Speed AVI software. Chain deviation from its nominal position was recorded in two planes, plane containing strand and perpendicular to it. Both are represented in picture 1.

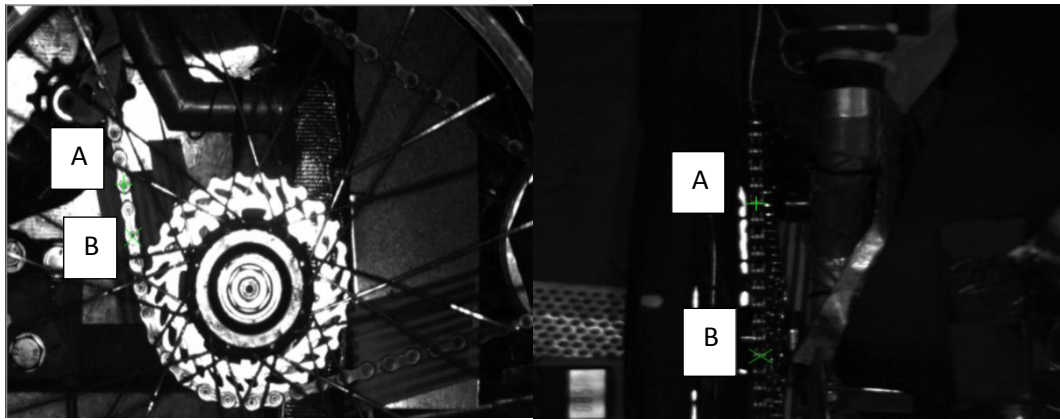


Fig. 1 Method of determination vibration; where 1 and 2 – base links.

Gear operation was divided into 3 stages. These stages were: start – from quiescence to gear switch, gear switch – brief moment before and after changing transmission ratios, maximal velocity - acceleration to speed limit. Subsequently, these stages will be referred to as such. Structure of regulation system is shown in picture 2.

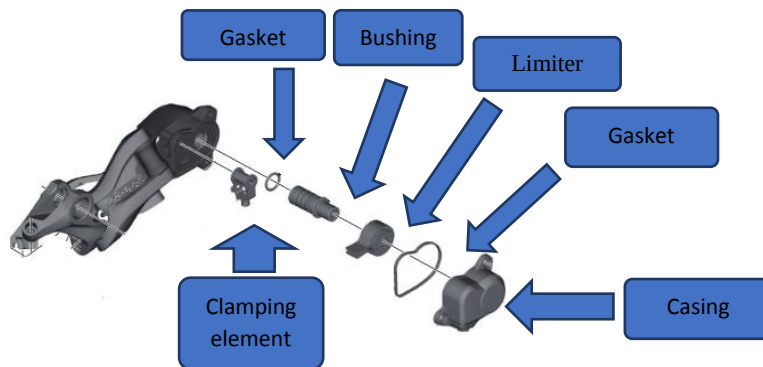


Fig. 2 Scheme of mechanism [3]

Main task of adjustment is to increase the movement drag of derailleur cage. In this paper this mechanism will be referred to as clutch. In studies assumed four configurations, 0;3;3,5 and 4. These numbers determine the number of rotations with which the adjustment screw was turned. Zero value means that the mechanism is switched off. Picture 3 shows a diagram of regulating device. This mechanism consists of screw, shaft and limiter. Limiter is used to increase wrap angle of bushing. Adjustment is done by pressing two plates in the left side of the image. The screw is being used to decrease the distance between plates in the limiter. Scheme of executive element is shown in picture 4.

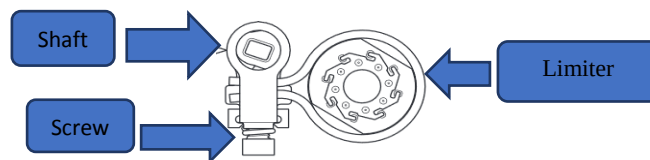


Fig. 3 Scheme of regulation mechanism [3]

Diagram in picture 4 shows the work principle of turning on mechanism in the clutch assembly. Engaging the clutch is done by pressing disks by rotating a lever.

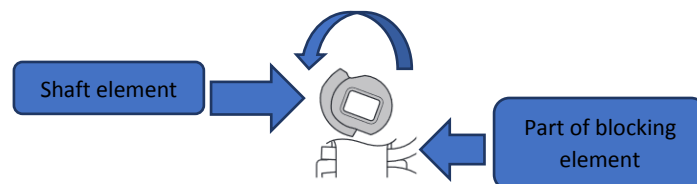
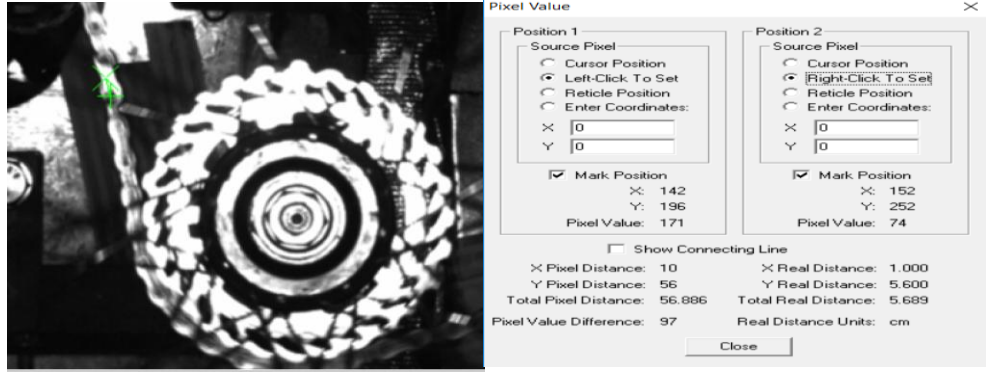


Fig. 4 Scheme of turning on mechanism [3]



Studies were conducted as follows:

- Determining of base link
- Determining the starting position of base link
- Selecting links in relation to which, translation was calculated
- Pointing link position during movement
- Calculating distance between, base and maximal positions.

Picture 5 shows starting position of the chain. Stripes in the background are the scale of reference to the vibration amplitude. One stripe has exactly 2 [mm] in width. Points, which are marked are the links were being analyzed during the test. Point coordinates were read off the pixel value module (Fig. 6).

Fig. 5 Method of determining vibration

Fig. 6 Pixel Value module

Next stage of vibration analysis was calculating the real distance translation of chain link.

$$p = x_1 - x_2 \quad (1.2)$$

Where:

p – distance between link positions [1],

x_1 – position of deviated link [1],

x_2 – position of base link [1].

$$p = 146 - 142$$

$$p = 4$$

$$x_r = \frac{2}{6}p \quad (1.3)$$

Where:

x_r – calculated translation[mm]

$$x_r = 1,33$$

3. Studies

Diagram 7 describes the correlation between vibrations amplitude and clutch setting in switch gear stage. It shows how clutch adjustment impacts vibrations in both planes. Higher clutch setting results in decreased vibrations in one plane and increased in the other.

From the diagram, it can be deduced that the best setting, in terms of work fluency, is setting number 3. However, this regulation has a negative impact on a fluency in the maximum velocity stage (Fig 8).

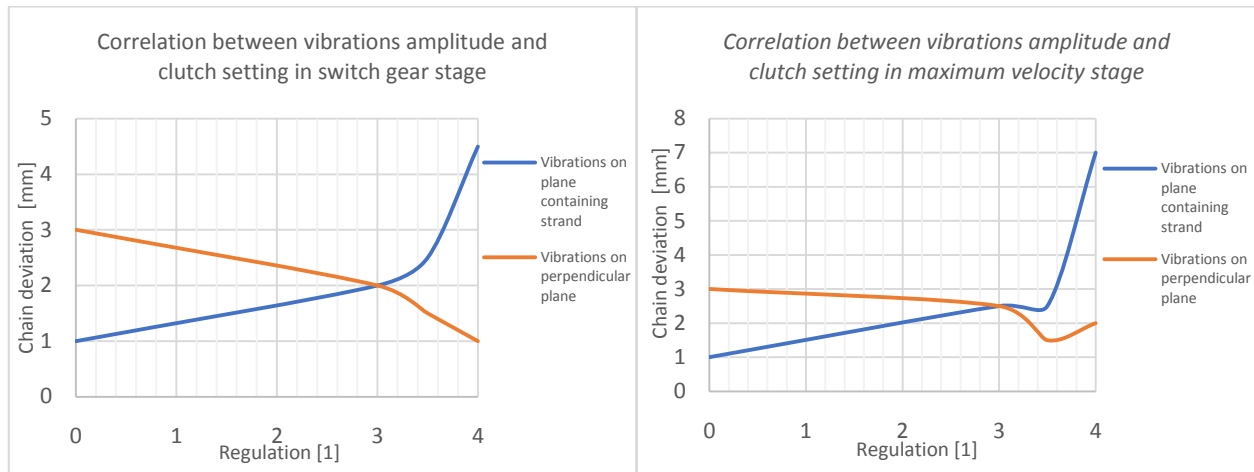


Fig. 7 Correlation between vibrations amplitude and clutch setting in switch gear stage

Fig. 8 Correlation between vibrations amplitude and clutch setting in maximum velocity stage

For this reason, it has been decided to further study setting 3,0 and 3,5. These studies had a purpose of determining the exact moment of vibration occurrence in switch gear stage, from this point vibrations increase in magnitude.

Another argument advocating for setting 3,5 is the fact that system oscillations were much higher before changing transmissions, which could be caused by resonance. This anomaly has adverse influence on SGR vehicle, due to an increased possibility of stability loss. This phenomenon was observed at a spot marked on diagram 9.

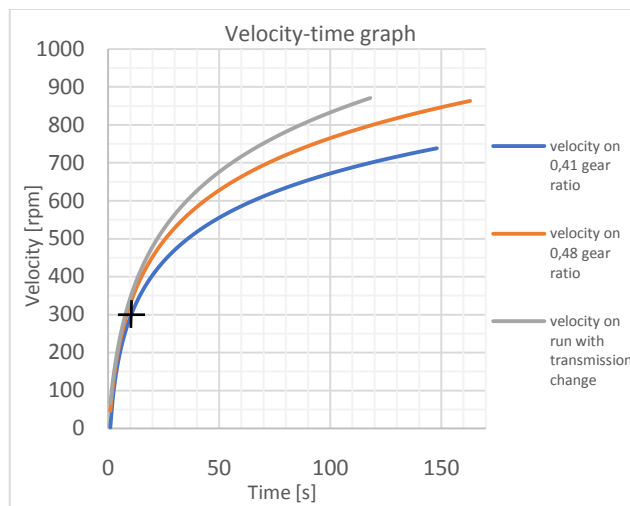


Fig. 9 Velocity-time graph using regulation 3,0

In order to unify the findings only 0,41 transmission ratio was taken into account. Times of resonance occurrences were confronted for settings 3,0 and 3,5.

Analysis revealed that for regulation 3,0 resonance occurred much sooner than for 3,5. Using setting 3,5 resonance was avoided by switching gears before it occurs (Fig 10).

Conclusions.

A correlation between vibrations amplitude and clutch setting in switch gear stage can be noticed. During work of the drive transmission system, the moment of changing gears represents the greatest threat of losing stability. Because of that, it is essential to adjust the system parameters to decrease the chance of stability loss. Conducted research showed, that increasing derailleur cage movement drag, causes vibration amplitude of the chain to increase in one plane and decrease in the other. Regulation 0 and 4,0 can not be chosen because of difference between vibration amplitudes in planes investigated. Clutch settings 1,0 and 2,0 show better work parameters of chain, however regulation 3,0 and 3,5 indicate a better strand stability.

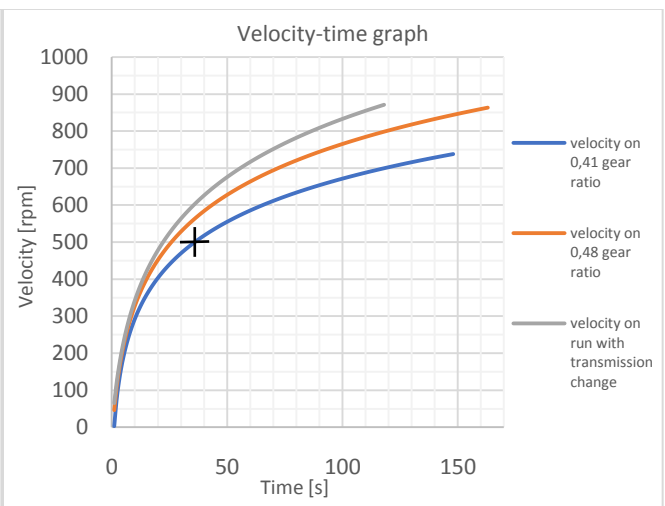


Fig. 10 Velocity-time graph using regulation 3,5

From among four different regulations, setting 3,5 shows greatest properties in all work stages.

It minimizes chain vibrations and decreases the chance of stability loss overall.

Literature.

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IMPROVING BENDING LOAD CAPACITY OF SPUR GEARS WITH INCREASING ROOT RADIUS

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Abstract: Gears are one of the most important machine elements in the industry. They are using many areas such as; automotive, energy, aviation, etc. Gears are exposed to higher loads day by day due to the increase in power and speed on the machines. Therefore, the stress values which are occur on the gear root is also increase. These stresses cause to damage on the teeth root. Thus the stress values have to be decrease to design optimum gear body. In this study, the effects of root radius on the gear bending stress are evaluated by using finite element method. At first, gears with standard root radius is investigated both DIN 3990 and finite element method. After the validation of finite element model, the root radius of the gear is taken as parameter. Gears with different root radiuses are analyzed by using finite element method. It is seen that with the increasement of gear tooth radiuses, the root stress is considerably decreased. With increasement of root radiuses the maximum principal stress reduced nearly 20%.

Keywords: SPUR GEARS, BENDING STRESS, LOAD CAPACITY, FINITE ELEMENT METHOD

1. Introduction

Nowadays gears have a widespread using areas such as automotive, energy, aviation etc. In these areas the demanding power and speed values increase so; gears are exposed to higher loads day by day. Consequently with increasing power and speed on machines, gear designs need to keep up with today's requirements. During the operation of gears each tooth is affected from static and dynamic loads. In lower rotation speeds, in proportion to transmitted torque, static loads are exposed on each tooth. With increasing of rotation speeds, dynamic loads begin to be effective. For longer fatigue life, quiet and safety operation of gears, dynamic loads need to be reduced. Bending stresses that occur at tooth root are one of the preventive factor for optimum gear design. Awareness of these stresses, especially in design stage, is also important for taking precautions to gear damage and improvement load carrying capacity [1]. Due to developments in engineering technologies gear's durability and load carrying capacity also can be modified with different ways such as change of root radius for reducing these stresses.

Widely using of gears in industry brings along various numerical and experimental methods. Numerical methods that enable to investigate effects of many parameters satisfactorily in small time periods are become dominated for gear damage analyses [1]. In this context many studies have been made in the literature; Ural et al. studied on the stress at the profile of tooth in cylindrical spur gears with ANSYS computer code and developed a model with 3 DOF using 3D solid 186 element. But the study contains only constant values of pressure angle number of teeth, modulus etc. [2]. Fetvaci and Imrak created a FEM model of a spur gear for examining tooth root stress. A macro code that developed by Fetvaci has used for applying tooth forces to model respectively in some points of engagement cycle. After their model's analysis with ANSYS, they compared the results with literature [1]. Karpat et al. investigated gears with asymmetric teeth. They developed a model which used for the modeling of tooth via finite element model and examined the root bending stress [3]. Fetvaci and Imrak proposed a new finite element gear mesh pattern which is suitable for investigations of bending stress analysis of spur gears during the engagement cycle [4]. Hasl et al. studied on a calculation method for tooth root stress of plastic gears and represented the results of calculated contact ratios and tooth root stresses with graphics [5].

Fetvaci and Imrak gave information about things to pay attention for modeling involute spur gears. Investigated stress accumulation that triggers the fatigue crack in tooth root due to loading conditions. A novel finite element method consisting of quadratic elements in the second order is also developed. The results of tension stresses in number of teeth between 20 and 40 are compared with values in the literature [6]. Wei analyzed the characteristics of involute spur gears by using finite element method. The contact stresses were examined using two-dimensional (2D) FEM models and the bending stresses in tooth root were examined using a three-dimensional (3D) FEM model [7]. Rajaprabakaran and Ashokraj studied on finite element model of spur gear with a segment of three teeth for investigating stress concentration [8].

There are many methods for investigating root bending stresses such as the finite element method which is one of the popular method but it is more reasonable that support the FEM with experimental and analytical methods for obtaining results with a better accuracy. In this context; Shaker, in his master thesis, studied on optimization of tooth – root profile for maximum load carrying capacity for spur and bevel gears and developed a novel approach to design tooth – root profile of spur and bevel gears for meeting industry's demand. Therefore, he used FEA and experimental method together [9]. Lisle et al. compared root bending stress according to ISO (The International Organisation of Standardisation) 6336:2006, AGMA (American Gear Manufacturers Association) 2101 – D04, ANSYS finite element analysis and strain gauge techniques. They compared the root bending stresses values of ISO-AGMA-FEA and FEA-strain gauge with graphics and evaluated the results [10].

In this study the effects of root radius on the gear bending stress is investigated by using finite element method. At first for the gears with standard root radius is investigated both finite element method and DIN 3990 standard. After the validation, gears which have various root radiuses are designed and analyzed. Features of gear model and mesh properties are given with tables. After creating FEA model the effect of tooth radius values on bending stress is investigated and the results are discussed.

2. Material and Method

There are various methods have been developed for the calculation of gear bending stress. These methods can be classified as analytical methods, numerical methods and experimental methods. ISO 6336 and DIN 3990 are the most well-known methods and standards in the literature. These standards are quite similar to each other and calculations are made according to the following assumptions.

- The critical section of the tooth is the thickness of the tangent to the root of the tooth, making an angle of 30 ° starting from the symmetry of the tooth
- Compression stress which is caused by the radial component of normal force on the gear can be neglected.
- The tooth load is considered to influenced by the addendum circle in DIN 3990 / Method C and ISO 6336 / TC 60 Method C.

According to DIN 3990, the maximum tooth bending stress in the spur gears is calculated according to Eq.(1).

$$\sigma_{F0} = \frac{F_t}{b \cdot m_n} Y_F \cdot Y_S \cdot Y_\epsilon \cdot Y_\beta \quad [1]$$

Where F_t is the tangential force, b is tooth width, m_n is normal module;

Y_F is form factor;

$$Y_F = \frac{6 \cdot (h_{Fa}/m_n) \cos \alpha_{Fan}}{(S_{Fn}/m_n)^2 \cos \alpha_n} \quad [2]$$

Y_S is stress correction factor;

$$Y_S = \left(1,2 + 1,3 \cdot \frac{S_{Fn}}{h_{Fa}} \right) \left(\frac{S_{Fn}}{2 \cdot \rho_f} \right)^{1/[1,21+2,3(h_{Fa}/S_{Fn})]} \quad [3]$$

Y_ϵ load sharing factor;

$$Y_\epsilon = 0,25 + \frac{0,75}{\epsilon_\alpha} \quad [4]$$

Y_β is helix factor, in this study the gear type is defined as spur gear thus the Y_β can be considered as ‘1’ also the finite element analyses is conducted by using single tooth model so the load sharing factor is also defined as, ‘1’.

In this study, the effects of root radius on the gear bending stress are evaluated by using finite element method (FEM). Also for validation, DIN 3990 standards have taken into account. After the validation of finite element model, the root radius of the gear is taken as parameter. Gears with different root radiuses are analyzed by using finite element method. ANSYS static structural module is selected for the finite element model creation.

Table 1. Features of FEA gear models

Material	Steel
Young Module (N/mm ²)	210000
Poisson Ratio	0.3

Gear material is defined as standard steel whose properties are given in Table.1. The gear material is defined as an isotropic due to the low stress levels.

The gear geometry is defined as a three tooth model. The whole gear is not preferred because of the calculation time. The mesh structure is consisted of 44353 hexahedral elements and 179875 nodes. The mesh properties are given in Table 2.

Table 2. Mesh properties

Mesh Structure	Hexahedral
Number of Element	44353
Number of Nodes	179875
Mesh Type	Solid187

The boundary conditions and the mesh structure of the finite element model is given in Fig 1. The model is fixed on both sides and at the bottom. The static load is applied on the tip of the gear.

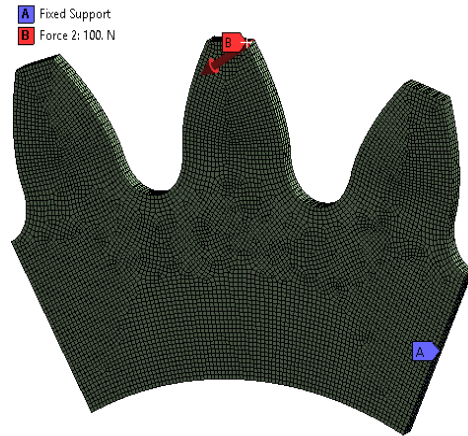


Fig 1. Finite element model

The pressure angle takes different values on the addendum circle and pitch circle. The standard pressure angle for the spur gear is 20°. The pressure angle is determined according to the following equation on the addendum circle.

$$\tau_0 * \cos \alpha \tau_{(i)} * \cos \alpha_{(i)} \quad [5]$$

where τ_0 : radius of pitch circle, $\alpha=20^\circ$ pressure angle on the pitch circle, $\tau_{(a)}$: addendum circle on the gear, $\alpha_{(a)}$: pressure angle on addendum circle.

Different root radius values are selected as a design parameter to reduce gear bending stress as $\rho = 0,1^*m, 0,2^*m, 0,3^*m, 0,375^*m$ (standard) and $0,47^*m$ respectively. The gears, which are used in this study, properties are given in Table 3.

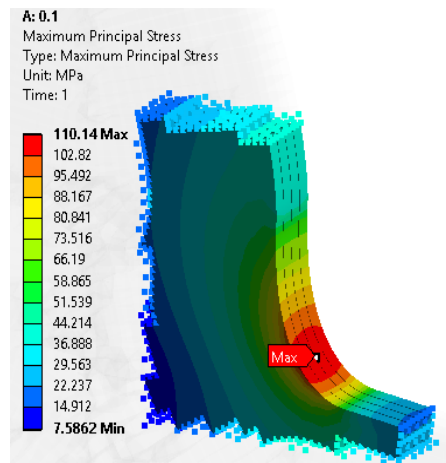
Table 3. Gear properties

Module (mm)	4
Tooth number	20
Pressure angle (°)	20
Face width (mm)	1
Normal force (N)	100
Root radius	0.1*m , 0.2*m, 0.3*m, 0.375*m 0.47*m

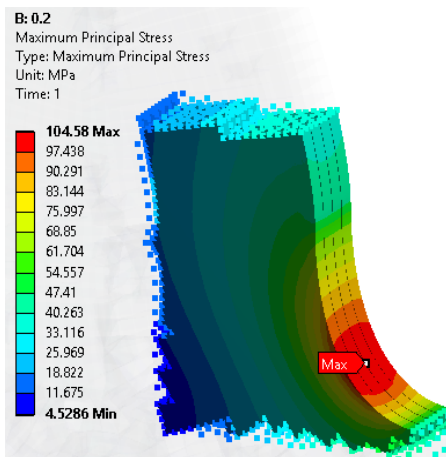
3. Results and Discussion

The maximum principle stress results are evaluated after the finite element analysis calculation. The maximum stress is seen at the root region of the tooth as in in Fig 2. At first gear which has standard root radius is investigated by using both finite element method and DIN 3990. According to the DIN 3990 form factor is defined as $Y_F = 2.87$, stress correction is defined as $Y_S = 1.55$, Y_ϵ and Y_β can be taken 1. Thus, the maximum stress is defined as 95,004 MPa by using DIN 3990. The finite element model result can be seen in Fig 2 at 94.141 MPa. The differences between two methods can be negligible so, the finite element model is validated for the standard spur gears.

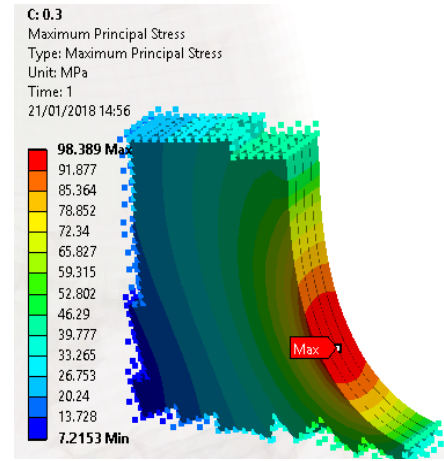
After the verification process of the finite element model the root radiuses of the spur gears are varied from 0.1*m to 0.47*m. The results can be also seen in Fig 2.



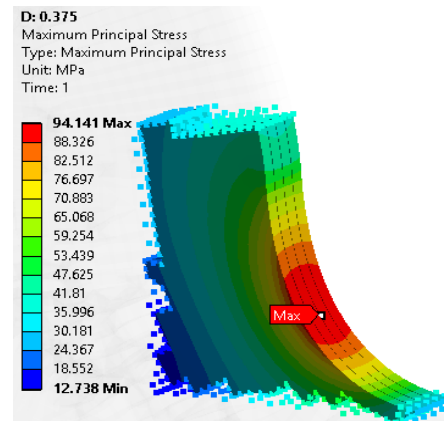
$\rho = 0,1 \cdot m$



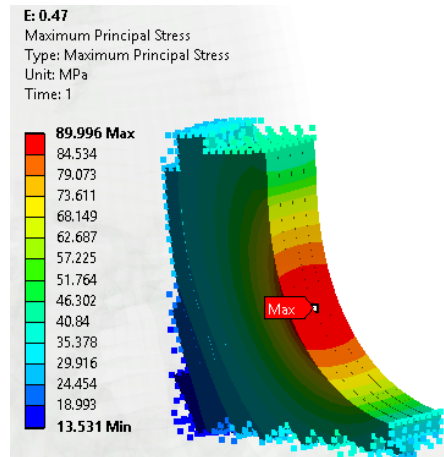
$\rho = 0,2 \cdot m$



$\rho = 0,3 \cdot m$



$\rho = 0,37 \cdot m$ (standard)



$\rho = 0,47 \cdot m$

Fig 2. Results of FEA, maximum principal stresses for different root radiuses

The maximum stress is seen when the root radius is the minimum values at 0.1*m. The maximum stress is 110 MPa, the minimum stress is occurred, when the root radius is maximum value at 0.47*m. The minimum stress value is 89 MPa for the maximum root radius.

The effect of root radius on the maximum bending stress is seen in Fig. 3. It is clearly seen that, when the root radius increase the maximum bending stress which is occurred in the root region is

considerably decreased. Nearly 20% of stress decrease is achieved only the increase of the root radius.

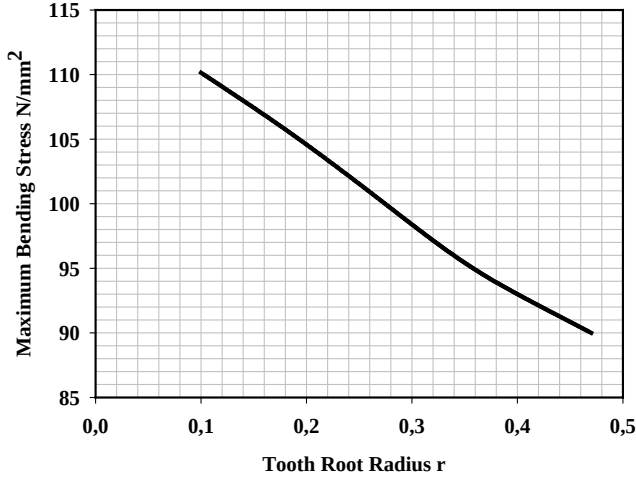


Fig 3. Effect of tooth root radius on bending stress

4. Conclusion

Most of gears give failure due to the high bending stress. Therefore, in the design phase of the gears, the bending stress should be minimized. Increasing gear root radius is a powerful way to decrease gear bending stress. To see the effect of gear root radius on the gear bending stress finite element model is created. Firstly gear with standard radius is analyzed and validated by using DIN 3990 analytically. After that the root radius values varied from $\rho = 0,1 \cdot m, 0,2 \cdot m, 0,3 \cdot m, 0,375 \cdot m$ (standard) to $0,47 \cdot m$ respectively and maximum bending stresses are calculated by using finite element method. With increase of root radiuses the maximum principal stress reduced nearly 20%.

5. References

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THE IMPACT OF THE CONSTRUCTIVE PARAMETERS OF THE BUMPER OVER THE CONSEQUENCES DERIVING FROM THE PROCESS OF COLLISION

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Abstract: With the analysis and synthesis of the process of crashing, the impact of every factor is estimated in the generation and the development of the accident, and conclusions are drawn regarding the impact of particular elements from the vehicle over the consequences from the accident.

The bumper system is a component of the vehicle intended to reduce the effects of the bumps on the front and the rear of the vehicle, its components and the shell. The cost of the damage and the protection regarding the bumper are the main criteria used to evaluate the efficiency of the impact of the bumper system, especially at low impact velocity.

This paper is representing the results of a deducted analysis and an investigation of the influence of the bumper system alongside its constructive parameters to the extent of the damage in low impact velocity. The investigation is primarily relating to statistical analysis of average damage, overall weighted average damage and variations in the amount of damage.

KEYWORDS: bumper system, accident damage, overall weighted average damage.

1. Introduction

The automotive industry has been known to be very competitive as far as its design and material usage are concerned. The automotive industry always faces greater market pressure to develop high quality products more quickly at lower costs, reduce weight in order to improve fuel efficiency and costs. One of the many purposes of the traffic jam safety measures is a construction of a safe vehicle which regarding its functions is supposed to represent the least influencing factor in the generation of causes that lead to creating detrimental consequences over all participants in the traffic jam and its environment, and, at the same time, in case an accident has occurred, to reduce or to completely baffle all possible consequences. The vehicle, regarding its constructive-technical and exploitative assets, the maintenance during the exploitation and the method of commandeering in traffic is one of the most important factors regarding safety. Starting from this particular definition regarding a vehicles safety in traffic, a broad space for investigation of all parameters is unclogged, which through the stage of projecting will lead to increase in the total value of safety regarding vehicles.

Systematizing the characteristics of the safety of the vehicle according to its specifics, they can be divided into three categories:

- Characteristics of active safety, which include all parameters that have an impact on the possibility of occurrence of an accident;
- Characteristics of passive safety, which include the parameters that have a preventive impact over the consequences of the accident; when the vehicle is partaking in an accident, the construction of the vehicle and the construction of its parts and units and their setting, to enable minimal or no injuries to the passengers in the vehicles and the incoming pedestrians, and also minimize the material damage of the vehicles, participants in the accident, the road and the environment.
- Characteristics of a catalytic safety, which include the parameters which indirectly contribute to the occurrence of the accident or increment the consequences of that accident.

The characteristics of the passive safety refer to all of the parts, units and elements which affect the consequences of the prompted accident in any way. The process of crashing is complex and dependent on various factors and their correlation. The subject of investigation in this paper is confirmation of the contribution of the bumper, with its constructive parameters to the consequences of the crash where the participants are moving with low speed. Through analysis of series of experimental crashes carried out in controlled

conditions, certain conclusions will be drawn concerning the estimate of the safety capabilities of the vehicle regarding its bumpers.

There are two significant aspects of the impact of the vehicle on the aftereffects of the accident regarding the bumper:

- The impact over the size of the injuries on a pedestrian regarding the incursion of the vehicle with the pedestrians.
- The impact over the size of the damage of the vehicles participants in the accident.

The main target of this paper is expansion and deepening the knowledge about the impact of the bumper with its constructive characteristics on the damage of the vehicle low impact velocity.

2. The role of the bumper in the safety capacities of the vehicle

The automobile bumper is a structural component of a vehicle that contributes to the improvement of the total asperity of the vehicle and its protection at front or rear impact. The bumper, firstly and mostly is intended to protect the body, the headlights or the stoplights, the indicators, the hood, the coolers and the other safety-bounded components of the vehicles in a low impact velocity.

But of course, the most important factor that affects the outcome of the accident is the impact velocity. The size of the damage of the vehicles is proportional to the impact velocity of the vehicles that partake in the accident. At higher impact velocities their impact over the damage of the vehicles is primary and dominant and the impact of the vehicle with its constructive characteristics and in that context also the bumper, is secondary. At low impact velocities of the vehicles the impact of the construction of the vehicles over the aftereffects of the accident is significant.

In 2002 IIHS conducted a research in traffic accidents in five big cities, recorded in their native departments for estimation of damage of the insurance companies. One of the conclusions of this study was that 14% of the accidents in the urban areas were impact with low impact velocities.

The bumper system is generally composed of four main elements: bumper cover, absorber, bumper carrier and holders, which are used for attachment with the body (the shell) of the vehicle.

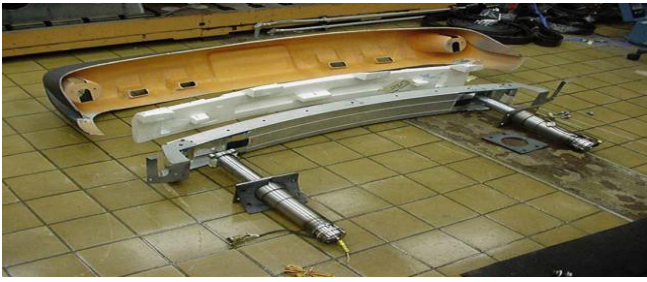


Figure 1: *The main components that constitute the bumper system and their relation [2]*

The leading restricting constructive factor for the bumper system regarding the construction of the entire vehicle is its volume. The bumper cover for the most part has a design function while its function regarding the vehicle's safety is minor. Because of that, and in order to achieve low production costs, this element at today's modern vehicles is only fabricated out of plastic: polystyrene, polycarbonate or acrylonitrile butadiene styrene. The carrier is the most significant element of the bumper system which protects the vehicle against frontal impacts and rear impacts. The carrier is manufactured out of steel sheet, aluminum, fiberglass, composite material or plastic. Commonly, an element which absorbs the energy from the crash is set between the carrier and the cover. Unlike the carriers, the absorbers are made out of low density materials. The bumper system is attached to the shell of the vehicles via holders through a rigid frame or via elastic framework as the newer constructive solutions suggest, using special mechanisms so-called shock absorbers, which have additional meaning, to absorb a part of the kinetic energy of the crash.

The geometry, stability and the capability to absorb energy from the crash are the key qualities of a good bumper system. Its width and length and the vehicle's height position, its capability to prevail the integrity, form and position before and after a low impact collision. The bumper also has to be constructed in a way to be manageably mending, easy and by low costs, after a collision at low speed of the vehicles.

There are various concepts used while projecting the bumpers, even for same class vehicles or models from the same manufacturer. The bumper system is a compromise of its design, its capability to absorb a part of the energy of the crash and its manufacturing costs. Some manufacturers pay special attention to the style and the visual effect on the account of the safety possibilities of the bumper which results in high damage costs in collision at low speed of the vehicles.

By analyzing the behavior of the bumpers, in the implementation of a controlled series of collisions, a relevant conclusion can be drawn that today's modern bumpers are not improved with safety features in relation to bumpers in older vehicles.

3. Regulatory normative regarding the vehicle bumpers

The need for reaching a certain level of standardized quality of the bumpers sets the necessity for establishing a certain regulative in this area.

ECE Regulation No 42, adopted by the United Nations Economic Commission for Europe, requires the vehicle's safety system to continue to operate normally after the front or rear of the vehicle is under the influence of a pendulum set at 455 mm above

the ground, loaded or unloaded at speed from 4 km per hour, along the entire length, i.e. 2.5 km per hour when operating in the corner on the bumper.

49 CFR part 581, American standard, prescribes requirements regarding the vehicles performance in a collision, front or rear, at low speed. The requirements apply to both the front and rear bumper of the vehicle, with a demand to prevent damage to the body and other equipment when hitting a barrier at a speed of 2 miles per hour along the entire length of the bumper i.e. 1miles per hour in a hit of the corner on the bumper.

Canadian regulation is very similar to the American. Also, this area is subject of regulation from the norms, the rulebooks of many scientific organizations, but not the NCAP programs for assessing the safety capabilities of the vehicle.

4. Realization of an engineering experiment. Discussion.

In this paper, by using statistical methods, the dependence of the height of the damage to the vehicle on the characteristics of the bumper system is analyzed and assessed, in the event of a low-speed collision, that is, the subject of statistical analysis are the results obtained from a series of experimental collisions in controlled conditions, realized by IIHS [3], according to the Bumper Test Protocol (Version VIII, September 2010) [4]. In this case, four types of impacts of the vehicle in a stationary obstacle are analyzed, which is simulating a bumper of another vehicle at rest: impact with front bumper, corner impact with front bumper, impact with rear bumper, corner impact with rear bumper.



with rear bumper and corner impact with rear bumper.

Figure 2: *Impact with front bumper; corner impact with front bumper; impact with rear bumper; corner impact with rear bumper*

The analyzed vehicles are classified into the following four groups: mini urban vehicles, small urban vehicles, medium-sized vehicles and limousines.

The subject of calculation and analysis is the average level of damage segmented by vehicle groups and impact type, overall weighted average damage (OWAD), level of variation and source of variations. In order for the measured sample to be considered relevant for further processing and withdrawing valid conclusions, it is necessary to be made a revision in the case of existence of rough errors and their elimination and to check out the fulfillment of the conditions of normality and homogeneity of the measured sample. For the four groups of vehicles, or their database, using the Grabs test, a conclusion can be drawn as the absence of a rough error. The high values of the p - indicator, significantly larger than the adopted level of risk of 5%, confirm the basic hypothesis that the data in the analyzed bases of the four groups of vehicles, follow the regularity of normal distribution.

TABLE 1: Average costs of damage from the performance of experimental crashes for the four groups of vehicles.

From the display it is obvious that the average costs for the calculated damage of the vehicles for the front impact are significantly higher in relation to rear impact whether it is a full or corner impact. Considering the configuration of the vehicle, the greater compactness in the rear as well as the existence of essential parts in the front of the vehicle, this conclusion is completely understandable. To see the level of variation, we will analyze the characteristic sizes of the normal distribution of the so-called overall weighted average damage (OWAD). OWAD is calculated when the amount of damage from the front and rear full impact is multiplied by two, and then collected with the amounts of damage to the corner front and rear impact. The amount thus obtained is divided by six and the value of the OWAD is obtained.

Group of vehicles	FULL IMPACT		CORNER IMPACT	
	Front (\$)	Rear (\$)	Front (\$)	Rear (\$)
Mini urban vehicles	2.119	1.388	2.161	706
Small urban vehicles	2.553	1.308	1.505	888
Medium-sized vehicles	2.244	1.455	1.938	1.193
Limousines	3.421	1.657	2.614	1.055

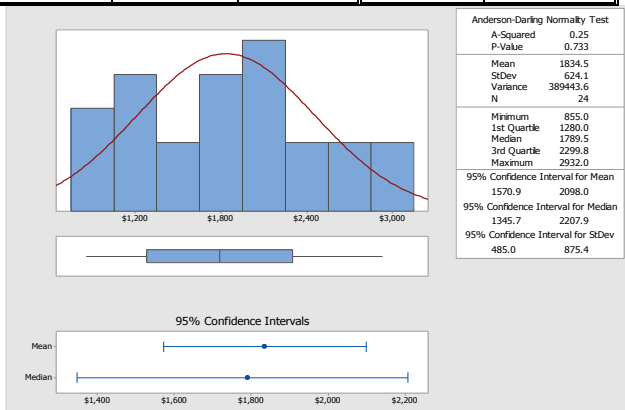


Figure 3: Diagram of probable distribution of OWAD for the group of medium-sized vehicles.

Although the number of motor vehicles that were the subject of this analysis differed significantly in each of the four groups, the obtained results show a high degree of coincidence. The height of the OWAD (table 2) ranges from \$ 1,718 in the group of small urban vehicles up to \$ 2,463 for the limousine group.

What is particularly noticeable are the huge coefficients of variation in the height of the OWAD, ranging from 32% to 36%. What is the reason for such variations in the level of damage in the groups?

TABLE 2: Values of the characteristic sizes in the analysis of the OWAD values for the four groups of vehicles.

Group of vehicles	No. of vehicles in the group	Average OWAD (\$)	Standard Deviation (\$)	Coefficient of variation (%)
Mini urban vehicles	7	1.776	609	34 %
Small urban vehicles	21	1.718	558	32 %
Medium-sized vehicles	24	1.834	624	34 %
Limousines	11	2.463	884	36 %

Since it is about analysis of the results of destructive tests, the ANOVA method is used to determine the contribution of both factors: the type of vehicle and the impact side (front or rear) on the total variations in the group. In doing so, we must start with the hypothesis for the homogeneity of the examined series of samples, that is, all the samples in the examined series are sufficiently identical for us to be able to consider that we operate with the same types.

From Table 3, it can be seen that the influence of the vehicle on the OWAD of the vehicle type with its structural features and in this context on the bumpers, is significantly greater in relation to the impact side – front or rear. This influence is stronger in full, compared to corner impacts, i.e. at vehicles with larger mass and dimensions compared to vehicles with smaller dimensions and mass.

TABLE 3: Determining the contribution of the type of the analyzed vehicle and the impact side on OWAD

Group of vehicles	FRONT OR REAR FULL IMPACT		FRONT OR REAR CORNER IMPACT	
	% Influence of vehicle type	% Influence of impact side	% Influence of vehicle type	% Influence of impact side
Mini urban vehicles	70%	30%	54%	46%
Small urban vehicles	70%	30%	54%	46%
Medium-sized vehicles	100%	0%	80%	20%
Limousines	95%	5%	73%	27%

5. Conclusion

Thanks to the results of the experimental impacts published by the IIHS, potential buyers are able to obtain adequate information about the amount of damage to vehicles at impacts conducted under controlled conditions. It is a motive for vehicle manufacturers to work on improving the performance of bumpers at vehicles, in

order to reduce the consequences of a traffic accident at low speeds of the vehicles.

Today's modern vehicles do not have bumpers with better impact resistance compared to older models. In support of this conclusion is the fact that the OWAD for all four groups of analyzed vehicles are higher than \$ 1,500, the boundary for poor and unacceptable quality of the bumper, according to the criteria established by IIHS. At the same time, the high level of variations in the OWAD in the realized experiment also speaks of the great possibilities for making improvements of the bumper system regarding the safety measures, starting from the design process and construction, selection of material, up to its testing.

Finally, the experiment and its results, as well as the requirements of customers for bumpers with an increased level of impact resistance, impose the question of the expediency of a more rigorous legal standardization of the quality of the bumper system.

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THEORETICAL AND NUMERICAL ASPECTS REGARDING THE THERMOELASTIC BEHAVIOUR OF RUBBERLIKE POLYMERS

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Abstract: Vehicle components made of rubber usually exhibit large deformations. Cyclic finite deformations may induce increasing temperature in hyperelastic materials. This case - where changes in deformation and in temperature occur simultaneously - is called coupled thermomechanical problem. Both the mechanical and thermal processes have their own governing equations, that is why special techniques are needed for the computation. A special technique will be presented for solving coupled problems, this is operator split method. The goal of this paper is to show how to solve the coupled thermomechanical problem by the principle of virtual power and the principle of virtual temperature, and how to apply them together.

Keywords: RUBBER, HIGH DEFORMATIONS, THERMODYNAMICS

1. Introduction

Rubber can be classified as a so-called hyperelastic polymer which has a typical geometrical and material nonlinear behavior. It means that the relationship between displacements and internal forces can be described by functions whose order is higher than linear. The geometrical nonlinearity is easy to handle mathematically, however the material nonlinearity is only described approximately [1], [2]. Independently of the experimental investigations which deal with the material behavior of rubber, a number of theoretical works treated rubber as an ideally nonlinear elastic, in particular hyperelastic material. One of the properties of the constitutive equations of hyper-elastic material is that stresses are derived from stored elastic energy function. Hyper-elasticity can be described by particularly convenient constitutive equation given its simplicity and it constitutes the basis for more complex material models such as elastoplasticity, viscoplasticity, and viscoelasticity [1].

Furthermore, the task becomes more complicated because of some features of rubber parts. The temperature of rubber increases significantly. Therefore, the temperature- and displacement fields are coupled, and it means that special solving algorithms are required [7]. So the equations of mechanics and thermodynamics are coupled. As described above, the goals of this paper are the following:

It is necessary to summarize the applied equations and the basic physical laws which are responsible for the theoretical background [8,9]. Clarification of these relationships is essential because the material laws of rubber cannot violate those basic physical laws. It is necessary to extend these relationships like balance of linear momentum and balance of angular momentum, the first and second law of thermodynamics to high deformation of rubber and rubberlike polymers. After this, it will follow the solution of the mechanical and heat conduction problem and the coupled thermomechanical problem by using the operator split method. In this paper, arrows above letters denotes vectors and double underline denotes tensor.

2. Governing equations

2.1 Equilibrium of linear momentum

The differential formulation of the equilibrium of linear momentum in the current configuration is

$$\rho \dot{\underline{\underline{v}}} = \underline{\underline{\sigma}} \cdot \underline{\underline{\nabla}} + \underline{\underline{f}} \quad (1)$$

where ρ is the mass density, $\underline{\underline{v}}$ is the velocity, $\underline{\underline{\sigma}}$ is the Cauchy stress, $\underline{\underline{f}}$ is the volume force.

2.2 Equilibrium of angular momentum

The next equality shows the differential form of the balance of the moments.

$$\underline{\underline{\sigma}} = \underline{\underline{\sigma}}^T \quad (2)$$

2.3 First law of thermodynamics

When deformations repeatedly occur, significant increase in temperature can be observed. The differential form of the first law of thermodynamics is in the current configuration

$$\dot{\rho}e = [-\underline{\underline{\nabla}} \cdot \underline{\underline{q}} + h] + \underline{\underline{\sigma}} \cdot \underline{\underline{l}} \quad (3)$$

where e is the internal energy per unit mass, $\underline{\underline{q}}$ is the heat flux, h is the heat source, $\underline{\underline{l}}$ is the velocity gradient, $\underline{\underline{l}} = \underline{\underline{F}} \cdot \underline{\underline{F}}^{-1}$, $\underline{\underline{l}} = \underline{\underline{v}} \circ \underline{\underline{\nabla}}$.

2.4 Second law of thermodynamics

The behaviour of viscoelastic materials is described by the second law of thermodynamics. The second law of thermodynamics in the current configuration can be written as

$$\dot{\eta}T\rho \geq -\underline{\underline{\nabla}} \cdot \underline{\underline{q}} + \frac{\underline{\underline{q}} \cdot \underline{\underline{\nabla}}T}{T} + h \quad (4)$$

where η is the entropy per unit mass and T is the absolute temperature. It will be practical to change the variable from entropy per unit mass to temperature by applying the Legendre transformation and by using the Helmholtz-free energy

$$\psi = e - \eta T \quad (5)$$

Substitute the Eqn. (5) into to the Eqn. (3) and subtract Eqn. (3) from Eqn. (4) the following expression will be generated

$$-(\dot{\psi} + \eta \dot{T})\rho + \underline{\underline{\sigma}} \cdot \underline{\underline{l}} - \frac{\underline{\underline{q}} \cdot \underline{\underline{\nabla}}T}{T} - D \geq 0 \quad (6)$$

which is known as Clausius-Duhem inequality [2].

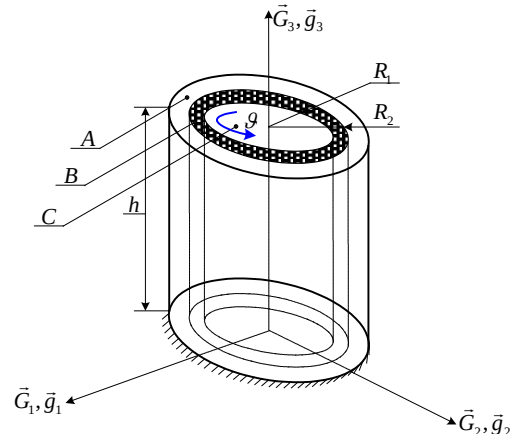


Fig. 1. Mechanical model of a silent block

3. Solution of the coupled thermomechanical problem

3.1 Principle of virtual power

Eq. (1) can be generated in the following form:

$$\underline{\underline{\sigma}} \cdot \nabla + \vec{f} = \vec{0} \quad (7)$$

The solution of Eq. (15) can be generated by finite element method. The basis of the finite element method is an energetical principle, in this case this is the principle of virtual power [4,7].

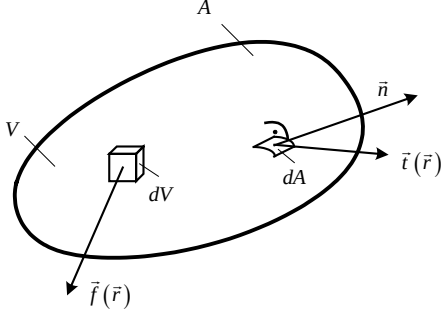


Fig. 2. Types of loads: $\vec{t}$ traction, $\vec{f}$ force per unit volume

Multiplying both sides of Eq. (7) by function $\vec{v}^*$ and integrating the following form will be generated:

$$\int_{(V)} \vec{v}^* \cdot \underline{\underline{\sigma}} \cdot \nabla dv + \int_{(V)} \vec{v}^* \cdot \vec{f} dv = 0 \quad (8)$$

where $\vec{v}^*$ is called virtual velocity field, which is the difference of two different kinematically admissible velocity field and has the following properties: continuous and it can be derivated at least once. The arrow placed above the Cauchy-stress tensor denotes the effect of the Hamiltonian differential operator. Using the $\vec{v}^* \cdot \underline{\underline{\sigma}} \cdot \nabla = \vec{v}^* \cdot \underline{\underline{\sigma}} \cdot \nabla + \vec{v}^* \cdot \underline{\underline{\sigma}} \cdot \nabla$ identity the next expression is generated:

$$\int_{(V)} \vec{v}^* \cdot \underline{\underline{\sigma}} \cdot \nabla dv - \int_{(V)} \vec{v}^* \cdot \underline{\underline{\sigma}} \cdot \nabla dv + \int_{(V)} \vec{v}^* \cdot \vec{f} dv = 0 \quad (9)$$

The first member of Eq. (9) can be converted by Gauss's theorem and the second member can be converted by the application of $\vec{v}^* \cdot \underline{\underline{\sigma}} \cdot \nabla = \underline{\underline{\sigma}} \cdot (\vec{v}^* \circ \nabla)$ identity, where two point means a scalar multiplication between tensors, therefore Eq. (9) results in the following form:

$$\int_{(V)} \vec{v}^* \cdot \underline{\underline{\sigma}} \cdot \vec{n} da - \int_{(V)} \underline{\underline{\sigma}} \cdot (\vec{v}^* \circ \nabla) dv + \int_{(V)} \vec{v}^* \cdot \vec{f} dv = 0$$

The above equation refers to the current configuration so the integration has to be done on the deformed shape of the body. It will be expedient to convert this equation to the reference configuration. Due to this transformation the integrations refer to the undeformable shape of the body.

Let us consider a hyperelastic continuum body in the reference configuration, its volume is indicated by V and its surface is indicated by A . It is known that dv can be expressed by dV

$$dv = JdV \quad (10)$$

and the connection between the surfaces da and dA can be expressed by the Nanson-formula [2]

$$\vec{n} da = J \underline{\underline{F}}^{-T} \cdot \vec{N} dA \quad (11)$$

where $\underline{\underline{F}}^{-T}$ is the inverse of the transpose of the deformation gradient, $\vec{N}$ is the normal vector of surface A , J is the determinant of the deformation gradient. Applying of Eq. (10) and Eq. (11) the Eq. (9) can be obtained, as it can be seen in the next expression:

$$\int_{(A)} \vec{v}^* \cdot \underline{\underline{\sigma}} \cdot \underline{\underline{F}}^{-T} \cdot \vec{N} J dA - \int_{(V)} J \underline{\underline{\sigma}} \cdot (\vec{v}^* \circ \nabla) dV + \int_{(V)} J \vec{v}^* \cdot \vec{f} dV = 0 \quad (12)$$

where $\underline{\underline{P}} = J \underline{\underline{\sigma}} \cdot \underline{\underline{F}}^{-T}$ is the first Piola-Kirchoff stress tensor. The first Piola-Kirchoff stress tensor does not refer to the reference configuration thus it needs to be converted. Multiplying the first member of the left side of Eq. (12) by the $\underline{\underline{I}} = \underline{\underline{F}} \cdot \underline{\underline{F}}^{-1}$ identity tensor and using the $\nabla = \nabla_0 \cdot \underline{\underline{F}}^{-1}$ expression, the following equation will be generated:

$$\int_{(A)} \vec{v}^* \cdot \underline{\underline{F}} \cdot \underline{\underline{F}}^{-1} \cdot \underline{\underline{P}} \cdot \vec{N} dA - \int_{(V)} J \underline{\underline{\sigma}} \cdot (\vec{v}^* \circ \nabla_0) \cdot \underline{\underline{F}}^{-1} dV + \int_{(V)} \vec{v}^* \cdot \vec{f} J dV = 0 \quad (13)$$

where $\underline{\underline{S}}$ is the second Piola-Kirchoff stress tensor which refers to the reference configuration. The surface of the considered body A can be divided into two parts, namely A_u surface where the displacement is defined and A_p where the traction is defined.

The $\underline{\underline{S}} \cdot \vec{N}$ product determines the value of $\vec{p}_0$ which is the traction vector in the reference configuration. Rearranging the second term of the left side of Eq. (13) the first Piola-Kirchoff stress tensor can be appeared:

$$\int_{(A_p)} \vec{v}^* \cdot \underline{\underline{F}} \cdot \underline{\underline{S}} \cdot \vec{N} dA + \int_{(A_u)} \vec{v}^* \cdot \underline{\underline{F}} \cdot \underline{\underline{S}} \cdot \vec{N} dA - \int_{(V)} J \underline{\underline{\sigma}} \cdot \underline{\underline{F}}^{-T} \cdot (\vec{v}^* \circ \nabla_0) dV + \int_{(V)} \vec{v}^* \cdot \vec{f} J dV = 0 \quad (14)$$

which can be transformed to the second Piola-Kirchoff stress tensor applying the $\underline{\underline{I}} = \underline{\underline{F}} \cdot \underline{\underline{F}}^{-1}$ expression:

$$\int_{(A_p)} \vec{v}^* \cdot \underline{\underline{F}} \cdot \vec{p}_0 dA + \int_{(A_u)} \vec{v}^* \cdot \underline{\underline{F}} \cdot \underline{\underline{S}} \cdot \vec{N} dA - \int_{(V)} \underline{\underline{F}} \cdot \underline{\underline{F}}^{-1} \cdot \underline{\underline{P}} \cdot (\vec{v}^* \circ \nabla_0) dV + \int_{(V)} \vec{v}^* \cdot \vec{f} J dV = 0 \quad (15)$$

The members of this equation can be divided into two groups: virtual power of internal forces (P_{int}) and virtual power of external forces (P_{ext}).

$$P = \int_{(V)} \underline{\underline{S}} \cdot \underline{\underline{F}}^T \cdot (\vec{v}^* \circ \nabla_0) \cdot \underline{\underline{F}} dV - \left(\int_{(A_p)} \vec{v}^* \cdot \underline{\underline{F}} \cdot \vec{p}_0 dA + \int_{(A_u)} \vec{v}^* \cdot \underline{\underline{F}} \cdot \underline{\underline{S}} \cdot \vec{N} dA + \int_{(V)} \vec{v}^* \cdot \vec{f} J dV \right) = 0 \quad (16)$$

Thus the virtual power of internal forces is equivalent the virtual power of external forces [1,2,9].

$$\begin{aligned}
& \underbrace{\int_{(V)} \underline{\underline{S}} \cdot \underline{\underline{F}}^T \cdot (\vec{v}^* \circ \nabla_0) \cdot \underline{\underline{F}} dV}_{P_{int}} = \\
& = \underbrace{\int_{(A_p)} \vec{v}^* \cdot \underline{\underline{F}} \cdot \vec{p}_0 dA}_{P_p} + \underbrace{\int_{(A_i)} \vec{v}^* \cdot \underline{\underline{F}} \cdot \underline{\underline{S}} \cdot \vec{N} dA}_{P_{ext}} + \underbrace{\int_{(V)} \vec{v}^* \cdot \vec{f}_0 dV}_{P_v}
\end{aligned}
\quad (17)$$

Denoting by P the difference of the two virtual power, P is the function of the position vector $\vec{r}$, the function $\vec{v}^*$ and the temperature T due to the thermal expansion.

$$P = P(\vec{r}, T, \vec{v}^*) \quad (18)$$

The thermal expansion is denoted by the next differential equation:

$$\frac{\partial J}{\partial T} = 3\alpha_0 J \quad (19)$$

where α_0 is the linear thermal expansion coefficient. Assuming that the temperature variations are not significant ($T - T_0 \ll T_0$), where T_0 is the reference temperature), the linear thermal expansion coefficient can be considered constant and in that case the solution of Eq. (19) can be determined by the next expression [2]:

$$J(T) = e^{3\alpha_0(T-T_0)} \quad (20)$$

Searching the vector field $\vec{r} = \vec{r}(\vec{R})$ (deformed body) which satisfies the nonlinear equation. Assuming that the mechanical processes are quasi-static processes thus the considered system is in a mechanical equilibrium in each time moment. It means that the acceleration of the points of the continuum body can be assumed zero and the other quantities do not depend on time. Since equation $P = 0$ is nonlinear the solution is generated by Newton-Raphson-method:

$$P(\vec{r}_k, \vec{v}^*) + DP(\vec{r}_k, \vec{v}^*)[\vec{u}] = 0 \quad (21)$$

where $\vec{r}_k$ ($k=0$) is an arbitrary vector, $DP(\vec{r}_k, \vec{v}^*)$ is the Gateaux-derivative [1] and $\vec{u}$ is the increment of displacement. After k th iteration steps

$$\vec{r}_{k+1} = \vec{r}_k + \vec{u} \quad (22)$$

the iteration will be stopped when the increment of displacement $\vec{u}$ decreases under the certain margin of error:

$$\frac{|\vec{u}|}{|\vec{r}_k - \vec{R}|} < h \quad (23)$$

3.2 Principle of virtual temperature

In order to solve the coupled thermomechanical problem, it is necessary to solve the heat conduction equation, too [7]. Considering an elastic element which has the property that the total mechanical energy is reversible. The free energy of the body is the function of the strain and temperature. Dissipation comes only from heat conduction.

Starting from the first law of thermodynamics Eq. (8) and changing the variable from entropy to temperature by using the following

expressions: $e = \psi + \eta T$, $\dot{e} = \dot{\psi} + \dot{\eta} T + \eta \dot{T}$, $\psi = \psi(T, \underline{\underline{E}})$, $\eta = -\frac{\partial \psi}{\partial T}$

$\eta = -\frac{\partial^2 \psi}{\partial T^2} \dot{T} - \frac{\partial^2 \psi}{\partial T \partial \underline{\underline{E}}} \cdot \underline{\underline{E}}$ the equation of heat conduction in the

reference configuration is

$$\rho_0 c \dot{T} = -\nabla_0 \cdot \vec{q}_0 + h_0 + \left(\underline{\underline{S}} - \rho_0 \frac{\partial \psi}{\partial \underline{\underline{E}}} + \rho_0 T \frac{\partial^2 \psi}{\partial \underline{\underline{E}} \partial T} \right) \cdot \underline{\underline{E}} \quad (24)$$

where ρ_0 is the mass density in the reference configuration, c is

the heat capacity, $\vec{q}_0$ is the heat flux vector in the reference configuration, h_0 is the heat source, where $\underline{\underline{E}}$ is the Green-Lagrange strain tensor. Correspondingly, the mechanical equation of the heat conduction equation can be solved by finite element method. The solution can be determined by the principle of virtual temperature. Assuming that the considered material is perfectly elastic the non-recoverable part of the mechanical power is:

$$\underline{\underline{S}} - \rho_0 \frac{\partial \psi}{\partial \underline{\underline{E}}} = \underline{\underline{0}} \quad (25)$$

Thus, the equation of heat conduction can be generated in the following form:

$$\rho_0 c \dot{T} = -\nabla_0 \cdot \vec{q}_0 + h_0 + \rho_0 T \frac{\partial^2 \psi}{\partial \underline{\underline{E}} \partial T} \cdot \underline{\underline{E}} \quad (26)$$

Multiplying both sides of Eq. (26) by function $\theta(\vec{r}, t)$ and integrating both sides, the following form will be generated:

$$\int_{(V)} \rho_0 c \dot{T} \theta dV = - \int_{(V)} \nabla_0 \cdot \vec{q}_0 \theta dV + \int_{(V)} h_0 \theta dV + \int_{(V)} \rho_0 T \frac{\partial^2 \psi}{\partial \underline{\underline{E}} \partial T} \cdot \underline{\underline{E}} \theta dV \quad (27)$$

where is $\theta(\vec{r}, t)$ called virtual temperature field, which has the similar properties

such as virtual velocity field, so it is continuous and it can be derivated at least once. Consequence of the second law of thermodynamics is that the heat flux can be expressed by the negative gradient of the temperature field [7]:

$$\vec{q}_0 = -\underline{\underline{\kappa}} \cdot \nabla_0 T \quad (28)$$

where $\underline{\underline{\kappa}}$ is the heat conduction tensor. Applying Gauss's theorem:

$$\begin{aligned}
\int_{(V)} \rho_0 c \dot{T} \theta dV &= - \int_{(A)} \theta \underbrace{\vec{q}_0 \cdot \vec{N}}_{q_N} dA - \int_{(V)} \theta \nabla_0 \cdot \underline{\underline{\kappa}} \cdot \nabla_0 T dV + \\
&+ \int_{(V)} h_0 \theta dV + \int_{(V)} \rho_0 T \frac{\partial^2 \psi}{\partial \underline{\underline{E}} \partial T} \cdot \underline{\underline{E}} \theta dV
\end{aligned} \quad (29)$$

Considering that the surface of the body can be divided according to the boundary conditions into two parts, the first member of the right side of Eq. (29) consists of two parts. The first part is the integral on the surface A_T where the temperature is given, the second is the integral on the surface A_q where the heat flux is given, as is presented in Fig. 3.

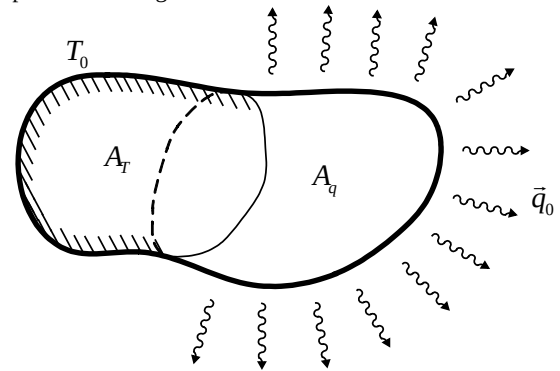


Fig. 3. Boundary conditions: defined temperature, defined heat flux

Furthermore, assuming that the considered material is isotrop in respect of heat conduction, thus $\underline{\underline{\kappa}} = \kappa \underline{\underline{I}}$, where $\underline{\underline{I}}$ denotes the identity tensor, the equation of heat conduction will have the next form:

$$\int_{(V)} \rho_0 c \dot{T} \theta dV = \int_{(A_r)} \kappa \theta (\nabla_0 T) \cdot \bar{N} dA - \int_{(A_i)} \theta \bar{q}_0 \cdot \bar{N} dA - \int_{(V)} \kappa (\theta \nabla_0) \cdot (\nabla_0 T) dV + \int_{(V)} h_0 \theta dV + \int_{(V)} \rho_0 T \frac{\partial^2 \psi}{\partial E \partial T} \cdot E \theta dV \quad (30)$$

Directing Eq. (30) to zero, the principle of virtual temperature can be determined which can be used for the approximate solution of the differential equation Eq. (26)

$$\Upsilon = \int_{(V)} \rho_0 c \dot{T} \theta dV + \int_{(A)} \theta q_N dA + \int_{(V)} \kappa \theta \nabla_0 \cdot \nabla_0 T dV - \int_{(V)} h_0 \theta dV - \int_{(V)} \rho_0 T \frac{\partial^2 \psi}{\partial E \partial T} \cdot E \theta dV = 0 \quad (31)$$

where Υ is function of the $\bar{r}$ position vector, the θ function and the T temperature, $\Upsilon = \Upsilon(\bar{r}, T, \theta)$. Searching the temperature field $T(\bar{r}, t)$ which satisfies the equation $\Upsilon = 0$.

3.3 Solution of the coupled thermomechanical problem

For the solution of the coupled problem, a special technique will be presented which is called operator split method. The solution is divided into two parts by the above mentioned method. First of all, the mechanical problem has to be solved with constant temperature:

$$P = P(\bar{r}, T = \text{const.}, \bar{v}^*) = 0 \quad (32)$$

The result of Eq. (32) is the $\bar{r}$ function. It will be followed by the solution of the heat conduction problem with constant $\bar{r}$ function, in the following manner:

$$\Upsilon = \Upsilon(\bar{r} = \text{const.}, T, \theta) = 0 \quad (33)$$

Substituting the resulted temperature field into the mechanical equation, the mechanical equation has to be solved again. These above mentioned two steps have to be repeated till the variation of the $\bar{r}$ function and temperature decrease under these margin of errors:

$$\frac{|\bar{r}_k - \bar{r}_{k-1}|}{|\bar{r}_k - \bar{R}|} < h_1 \quad (34)$$

and

$$\frac{|T_k - T_{k-1}|}{|T_k - T_0|} < h_2 \quad (35)$$

where h_1 and h_2 are margins of errors and T_0 is the reference temperature.

In order to solve the thermomechanical problem, it is necessary to use the weak formulation and to do the linearization of the non-linear formulation of the mechanical problem. The position vector is determined from the weak form of the mechanical problem, the temperature field is determined from the weak form of the heat conduction problem. The flowchart of the numerical solution is presented in Fig. 4.

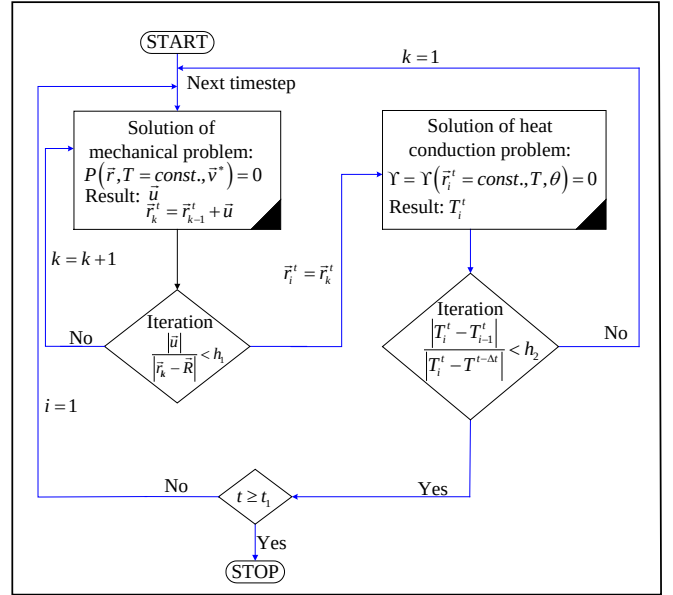


Fig. 4. Flowchart of solution of the coupled thermomechanical problem

4. Summary

We represented an algorithm which allows to calculate strain changes and temperature changes of the rubber part of a vehicle component under certain conditions. In the future we would like to develop a solving computer program in order to apply it as a thermodynamically consistent description.

The present numerical algorithm is the basis of the further fatigue and lifetime-calculations.

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POSSIBILITIES FOR CONTROL OF SEMI-ACTIVE SHOCK ABSORBERS IN ORDER TO REDUCE CASES OF SUSPENSION JOUNCES WHEN BRAKING

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Abstract: In this work a model of a truck when braking is created. The model takes into account the deflections in the suspension and vibrations of the sprung mass. The differential equations of the model are worked out. Numerical experiments are performed and some results are given. It is shown that if the shock absorber damping ratio is increased when vehicle braking, it is possible to reduce the pitch angle and the jounces in the suspension.

Keywords: SEMI-ACTIVE SUSPENSION, SHOCK ABSORBER, JOUNCE STOP, MODELING, SUSPENSION

1. Introduction

When the ground vehicles brakes, the wheel suspension travel may be spend and shocks may occur as a result of the inclusion of the jounce stops at maximum suspension deflection. The phenomenon is known as a suspension "slam" or "jounce" in the suspension, an amalgamation of the words jump and bounce. In suspension terminology, it means the most compressed condition of a spring. For instance, many suspensions use jounce stop blocks to prevent frame-to-axle contact (slam). In addition, significant accelerations can cause extreme strains in the vehicle's construction, leading to dangerous cracks and breakdown of the chassis and suspension components. Avoiding frequent shocks is usually associated with the requirement to increase the dynamic stroke of the suspension. However, when designing the suspension, the designer is limited by the size of the individual units, the requirements for the low frame layout above the road surface, the overall planning of the vehicle, etc. The process of braking the trucks and buses, changing the dynamic stroke of the suspension and the load distribution of individual axles and wheels has been numerically and experimentally studied by a number of authors [1, 2, 3 and 4]. Experimental studies of the jounce phenomenon of trucks on typical road surfaces are published in [5].

The purpose of this work is to study numerically the possibility of reducing the "jounces" in the suspension of a truck by changing the characteristics of adjustable semi-active shock absorbers. The test will be carried out in the event of an impact corresponding to the inertia force when the vehicle braking.

2. Dynamic Model

The model used in this study is based on the model shown in the work [6]. Fig. 1 shows a scheme of the used model. It takes into account the mass of the vehicle, its moment of inertia around the transverse axis, the elasticity of the front and rear suspension and the damping of the shock absorbers.

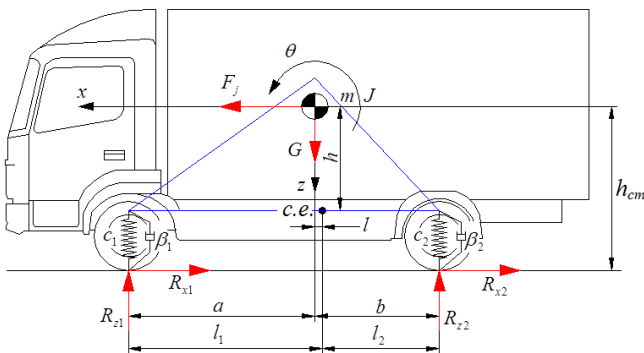


Fig. 1 Dynamic model scheme

The vector of the generalized coordinates is:

$$q = [x \quad z \quad \theta]^T$$

where x is the longitudinal displacement of the center of mass in the direction of movement of the truck, z - vertical displacement of the center of mass, θ - angular displacement of the body of the vehicle.

The following assumptions have been adopted:

- the characteristics of the elastic and damping elements are linear;
- the vehicle moves horizontally;
- ignoring the aerodynamic drag;
- the rolling resistance forces are neglected;
- the influence of the inertia moments of the rotating parts is neglected;
- it is assumed that the body angle is small (up to 15 °) and $\sin \theta = \theta$, $\cos \theta = 1$ is applied;

The differential equations describing the rectilinear motion of the vehicle are as follows:

$$m\ddot{x} = -R_x$$

$$m\ddot{z} = -F_{s1} - F_{s2}$$

$$J\ddot{\theta} = -aF_{s1} + bF_{s2} + M$$

where $R_x = R_{x1} + R_{x2}$, and R_{x1} , R_{x2} the longitudinal reactions in the contact between the wheels and the road when the vehicle is braking.

$$F_{s1} = F_{sp1} + F_{sh1} = c_1(z + a\theta) + \beta_1(\dot{z} + a\dot{\theta})$$

$$F_{s2} = F_{sp2} + F_{sh2} = c_2(z - b\theta) + \beta_2(\dot{z} - b\dot{\theta})$$

are forces in the front and rear suspension that arise when the system is pulled out of its static equilibrium, such as:

$$F_{sp1} = c_1(z + a\theta)$$

$$F_{sp2} = c_2(z - b\theta)$$

are the elastic forces of the front and rear suspension, and:

$$F_{sh1} = \beta_1(\dot{z} + a\dot{\theta})$$

$$F_{sh2} = \beta_2(\dot{z} - b\dot{\theta})$$

are the forces of resistance created by the shock absorbers respectively from the front and rear suspension of the vehicle. Through them we can find the dynamic vertical reactions on the road:

$$R_{z1} = G_{w1} + F_{s1}$$

$$R_{z2} = G_{w2} + F_{s2}$$

where G_{w1} and G_{w2} are the static loads of the front and rear wheels.

$M = F_j \cdot h = m \cdot j \cdot h$ is disturbing moment when braking;

$F_j = R_x = R_{x1} + R_{x2}$ is force of inertia;

j is braking deceleration;

h is distance from the center of elasticity to the mass center of the truck in the vertical direction (Fig. 1).

To approximate calculation of the moment of inertia, [7, 8] is used:

$$J = m.a.b$$

where J is the moment of inertia around the transverse axis of the vehicle with full load.

The recommended damping ratio of the shock absorbers for roads in good condition is [7, 9, 10]:

$$\psi_{oi} = \frac{\beta_i}{2m_i\omega_{oi}} = 0,2 \div 0,3$$

where ψ_{oi} is a damping aperiodic coefficient, m_i - the mass of the front or rear axle, ω_{oi} - its natural frequency which is:

$$\omega_{oi} = \sqrt{\frac{c_i}{m_i}}, \text{ rad/s}$$

Then for the overall damping ratio of the two shock absorbers from the front axle of the vehicle can be written:

$$\beta_1 = 2.\psi_{o1}m_1\omega_{o1} = 2,0.2.3400.7,0 = 9520 \text{ N.s/m}$$

and for the overall resistance coefficient of the two dampers at the rear axle:

$$\beta_2 = 2.\psi_{o2}m_2\omega_{o2} = 2,0.2.4100.7,5 = 12300 \text{ N.s/m}$$

When driving on uneven roads it is recommended that the damping aperiodic coefficient to be $\psi_{oi} = 0,6 \div 0,8$ [10, 11 and 12] and the critical aperiodic coefficient at which oscillations are aperiodic is $\psi_{oi} = 1$ [13] and is determined by the formulas:

$$\beta_1 = 2.m_1\omega_{o1} = 2.3400.7,0 = 47600 \text{ N.s/m}$$

$$\beta_2 = 2.m_2\omega_{o2} = 2.4100.7,5 = 61500 \text{ N.s/m}$$

3. Numerical Simulation

The numerical simulation is performed in program field of MATLAB. The values of the parameters are given in Table 1.

Table 1: Parameters of the dynamic model

Parameter	Symbol	Value	Dimension
Truck mass	m_0	5000	kg
Load mass	m_l	2500	kg
Full mass	m	7500	kg
Moment of inertia	J	33582	kg.m ²
Front suspension stiffness	c_1	166600	N/m
Rear suspension stiffness	c_2	230625	N/m
Front suspension damping	b_1	var	Ns/m
Rear suspension damping	b_2	var	Ns/m
Distance	h_{ec}	1,6	m
Distance	h	1,0	m
Distance	a	2,32	m
Distance	b	1,93	m
Static load – front axle	G_{w1}	33,355	kN
Static load – rear axle	G_{w2}	40,221	kN

The total braking force is described by trapezoidal law, analogous to the theoretical law of variation of braking deceleration in the braking diagram. Fig. 2 shows the variation of the front suspension deflection with low $\beta_1 = 9520 \text{ N.s/m}$ and high damping ratio $\beta_1 = 30000 \text{ N.s/m}$ of the front shock absorbers. The braking deceleration increase from 0 to 8 m/s in time interval 0,1 s.

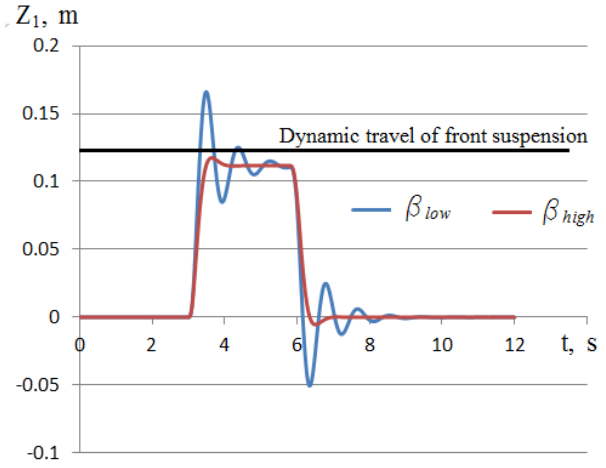


Fig. 2 Deflection of front suspension with low(blue line) and high damping (red line)

When the truck brakes with maximum brake deceleration there is a risk of a slam or jounce in the suspension. If the damping ratio of the shock absorbers increased at the beginning of braking (similar to the brake assistant control logic) the slams can be avoided (Fig. 2).

4. Conclusion

The considered model enables to study the vehicle pitch angle and jounces in suspension when vehicle brake. There are possibilities for reducing the suspension jounce if appropriate logic is used to control the semi-active suspension. With fast performance system can achieve good results and improve the comfort and reliability of the vehicle.

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THE INFLUENCE OF DISTANCE CALCULATION ON THE LOCATION OF CENTRAL WAREHOUSE

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Abstract: Choosing the right warehouse location is very important question which should be considered with great care, because it has a direct influence on the operating cost. For solving this problem many methods are on disposal. According to the way of determination of new warehouse location these methods can be divided in two groups': qualitative methods and quantitative methods. Quantitative methods are often used in practice from the reason greater objectivity and possibility to analyse obtained results. Most often used methods are: gravity method, load distance method, Weber problem and many others. For calculating of distance in the load distance method two approaches are in use rectilinear distance and Euclidean distance. Neither one of these two approaches credibly describes transport routes because they not taken into consideration any physical obstacles that might occur between potential warehouse locations. With the goal to determine the influence of distance calculation on the result of the load distance method, choice analysis of optimal location of warehouse for retail stores, shops in the urban area was performed and presented.

Keywords: LOGISTICS, MATERIAL FLOW, WAREHOUSING, LOCATION, TRANSPORT COST

1. Introduction

Warehouse is an inevitable part of logistic chain and without it binding of production, transport and consumption would not be possible. The purpose of the warehouse is best visible in retail stores which one have variable and unpredictable demand for goods, and the request for warehouse service is often issued during the working time of retail stores and it have to be fulfilled quickly. In contrary the shelves of retail stores will be empty and the customer will go to other retail stores. It is quite clear that the warehouse cannot be near to all of users of warehouse services, and it is also quite clear that their needs will better satisfied if the warehouse is able to respond as soon as possible. Placing the warehouse on location which will provide good services for all users is of crucial importance.

For solving of this problem many methods are on disposal. According to the way of determination of new warehouse location these methods can be divided in two groups': qualitative methods and quantitative methods. Qualitative methods are based on analysis of specific number of key factors which are important for choosing of warehouse location. Every one of those factors are analytically evaluated in form of points. The number of points, for every specific key factor, is determined by the analyst which is in charge for solving the problem of new warehouse location. The analyst determine these points subjectively in accordance with personal considerations. It is quite clear that some other analyst can evaluate same location differently which ultimately can lead to the selection of another warehouse location. These methods can be used for choosing the final warehouse location or they can be used in initial stage of determination of warehouse location for reducing of number of potential locations from which optimal warehouse location will be chosen with some other method. Quantitative methods used for solving the problem of warehouse location are methods that use clearly defined analytical indicators. These analytical indicators are: distance, transport cost, amount of transported material, investment cost, etc. Quantitative methods for solving the problem of warehouse location that are primary based on transport cost are used when relatively large transport costs are expected, i.e. when the transport distances are long [1]. Some of these methods are: *center of gravity*, *load distance*, *Weber's problem*, *Launhardt graphical method*, *Rockstroh method*, *northwest angle method*, etc.

2. Load distance method

Load distance method is approach for determining warehouse location based on analysis of distance between two location and amount of cargo transported between them. The goal of this method is to find location which will minimize the amount of cargo transported between warehouse and warehouse users and at the same time ensure the decrease of transport cost. In this way the load

distance method, alongside spacious distance takes in consideration the amount of transported cargo as an equally important factor. By applying the *load distance* method, a numerical value is obtained that represents the sum of products of the users distance from the given warehouse location and the amount of transported load between them. Numerical value is marked as *LD*, according to the name of this method "*Load Distance*" and according to [2] it is calculated as:

$$LD_j = \sum_{i=1}^n D_{ij} \times W_i \quad (1)$$

, where are:

LD_j - numerical value of *load distance* method

D_{ij} - distance between i and j warehouse location

W_i - amount of transported cargo from i location

As it can be seen from the equation (1), value of LD_j is determined by the distance between two location and by the amount of cargo transported between them [3]. Distance D_{ij} can be calculated as length of hypotenuse of imaginary triangle, which is formed by x and y coordinates of two locations, as is presented on Figure 1, where locations are shown and imaginary triangles which they form. Above mentioned distance is known as *Euclidean distance* and it can be calculated as [2],[4],[5]:

$$D_{ij} = \sqrt{(x_i - x_j)^2 + (y_i - y_j)^2} \quad (2)$$

The D_{ij} distance can be calculated as *rectilinear distance* between two points with 90° turn [3],[6],[7] i.e. the distance presents the sum of the lengths of cathetus, Figure 1.

$$D_{ij} = |x_i - x_j| + |y_i - y_j| \quad (3)$$

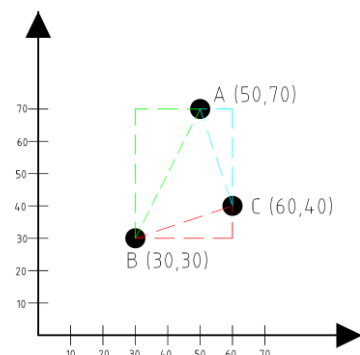


Fig. 1 Position of potential warehouse locations

Cargo or load W_i represents the amount of transported material and it is determined by demand of specific location for specific type of goods, and it can be expressed through number of transport cycles or number of transported units from proposed location of warehouse to the i location of user of warehouse services.

The decision for warehouse location is made by comparison of LD values for proposed locations and the optimal location is the one with lowest LD value. As it can be seen from equation (1) LD is in direct dependence from amount of transported material and distance D_{ij} . Distance determination ways are shown in equation (2) and (3). From those equations it can be seen that these two ways do not describe credibly realistic state of the transport paths nor do they take into account any physical obstacles that may occur between the two potential locations.

3. Determining of central warehouse location on specific example

To show the impact of distance D_{ij} determining methods on to the choice of the central warehouse location using the *load distance* method, in this paper is presented concrete example of determination of potential location of warehouse for chain of retail objects in the urban area. Presented locations of retail objects are users of warehouse service and they are also potential locations where the central warehouse could be positioned next to the retail facilities.

During the calculation of LD value for all the users of central warehouse services it is assumed that the transport intensity W_i is constant. Although this assumption does not correspond to realistic conditions, this allows clear determination of the dependence of the method of calculating the D_{ij} distance on the selection of the location of central warehouse.

The arrangement of the users of central warehouse services for which is necessary to determinate an adequate central warehouse location is required is presented in Figure 2.a. The service users are assigned by the names A, B, C, etc. For the given locations of the users of the central warehouse service, their position are determined by x and y coordinates, that were used for determination of the respective distance D_{ij} .

By analysing of equations (2) and (3) for determining of D_{ij} it can be concluded that is not important where the center of coordinate system is placed as long as the distance between the points are unchanged, with the condition that all the location of warehouse users are placed in same quadrant of coordinate system. By applying of these limitations, the coordinates for each given location i.e. the point were determined, Figure 2.b. The coordinates of users of central warehouse services are non-dimensional values because they are in function of the image resolution according to which they are determined.

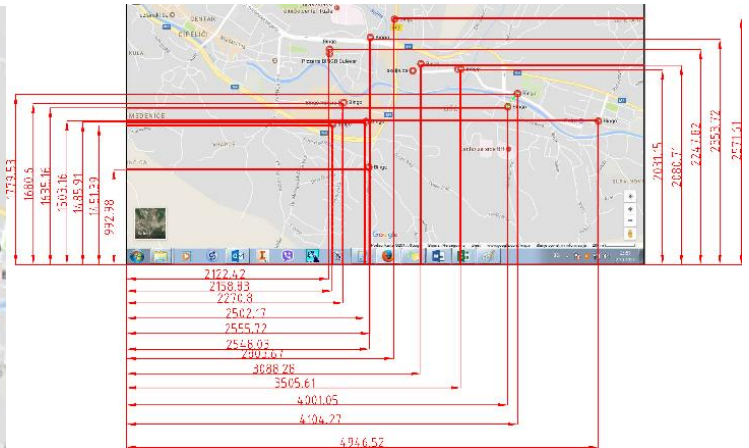


Fig. 2 Position (a) and determining coordinates (b) of "users" of central warehouse services - clients..

Based on the obtained coordinates, the distance D_{ij} was calculated in both pre-presented ways, using a constant value for W_i with the aim of neutralizing of its influence on the choice of location of central warehouse.

The results of calculation of the LD values for arrangement of the users of central warehouse services presented on Figure 2 are shown on the Figure 3, where can be seen that the location D is optimal location when the distance is calculated by *Euclidean distance*, and by applying *rectilinear distance* calculation location H is optimal choice for new central warehouse location. In both cases in those optimal locations of the central warehouse, factor LD has the lowest value. But the question is still remains, which location is optimal for central warehouse D or H

Although using the *rectangular* and *Euclidean* approach of the distance determination allows us to calculate the LD and determine the location of the central warehouse in accordance with the set criteria, the way of calculating of the distance D_{ij} can significantly deviate from the real conditions, as can be seen on the Figure 4. Based on the foregoing, it can be concluded that by applying of real distance for the same conditions, the *load distance* method could give different results.

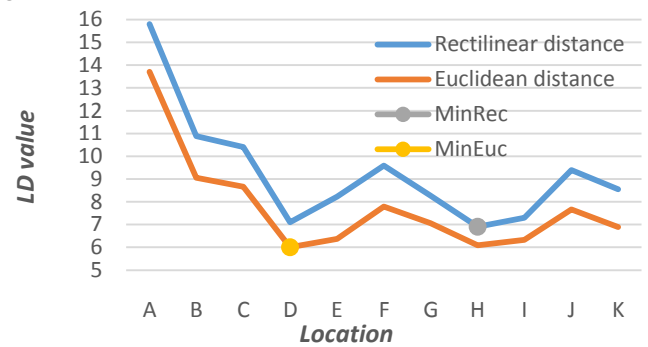


Fig. 3 Comparison of the results for Euclidean and rectilinear distance calculation in load distance method.

With the aim to analyse the influence of the way of determining the distance between the potential locations of the central warehouse and the users of warehouse services on the results of *load distance* method, the same positions of users of central warehouse services are used as in previous example, Figure 2.

The real distance are determined using the *Google maps* software for all location as it is shown on the Figure 4.c. For determined distances, calculation of the LD values was performed in accordance with the previously presented methodology. The results of that calculation are presented in the Table 1. Based on the obtained values, there is an obvious deviation from the results obtained in the previous two cases as can be seen on the Figure 5.

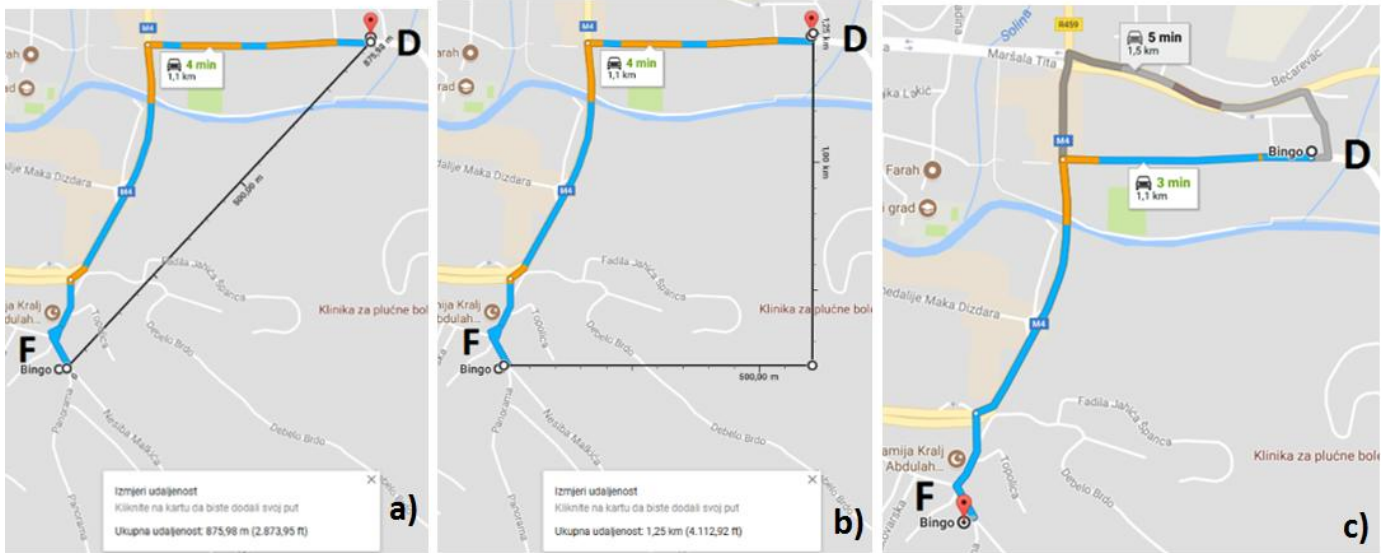


Fig. 4 Different ways to determine D_{ij} distance between two potential locations: a) Euclidean distance: $D_{ij} = 0,875$ [km], b) rectilinear distance: $D_{ij} = 1,25$ [km], c) real distance determined by transport routes: $D_{ij} = 1,1$ [km].

Using the first two approaches location D was the optimal location of the central warehouse with the minimum LD value for *Euclidian* distance calculation, and location H when *rectangular* distance calculation was applied. However, when real values of distances defined by “real” transport routes determined by *Google Maps* are used, the optimal location of the central warehouse with a minimum value of LD is the location E . In this case, the total distance that the transport vehicles will cross if location D is selected instead of location E is 2.85 [km], and if the location H is selected instead of E location the total distance that the transport vehicles will cross is 3,01 [km]. The resulting deviations certainly do not represent a small and negligible distance which the transport vehicles will cross, which ultimately creates the prerequisites for lowering the total transport costs.

4. Conclusion

The cost of transport is a very important item in the business of a company, and it can often be the one of items that secures market competitiveness, which is particularly evident in consumer products, products of low weight but with large volume, etc. Choosing wrong warehouse location can lead to decrease of income, and eventually to reduction of the market competitiveness. Many methods have been developed with the purpose to determine optimal warehouse location, and every of them have it good and bad sides as well as area of application. What is common to all those methods is that they use the same or similar mathematical algorithm to determine distances between the potential warehouse locations. One of the methods used for determining new warehouse location is *load distance* method. *Load distance* method is very simple to use and it don't have complicated

mathematical procedure which would require a high degree of knowledge and experience for its usage.

The paper presents the approach by which load distance method tests the optimal location of warehouse, as well as the ways to determine coordinates of potential locations of warehouse using available software such as *Google maps* and any of the 2D Design software such as *Solid Edge 2D*, *BabaCad*, *nanoCAD*, etc.

By analyzing of mathematical equations for determination of the distance between potential warehouse locations it have been noticed that resulting distances do not credibly describes transport routes as they are in reality, neither do the take in to consideration any physical obstacles that might occur between two locations. Specifically, in the example of usage of *Euclidean* equation for calculation of distances, the resulting distances i.e. transport routes are the shortest possible, that is more appropriate to air transport. By usage of *rectilinear* equation for calculation of distances the resulting distances, i.e. transport routes can be presented in the form of two straight lines intersecting at 90[°] who connects two potential warehouse locations, and this distance is slightly longer than the calculated *Euclidean* distance. Applying of these two approaches for calculation of the distances in some cases can results in different values of calculated distances for the *load distance* method, which additionally increases the distrust in its application.

The real transport routes, used for movement of transport vehicles, are consisted from a series of changes of directions that further increases the distance between two locations. It should also be taken in to consideration that all roads that are easy to use for movement of light transport vehicles do not provide same conditions for movement of heavy transport vehicles.

Table 1: Determining the optimal warehouse location by applying load distance method using real distance.

Location	W_i	Real distance [km]										
		A	B	C	D	E	F	G	H	I	J	K
A	1	0	0,65	0,85	1,1	1,4	2,6	2,6	2,5	2,5	2,2	3,4
B	1	0,65	0	0,24	0,45	0,7	1,6	1,6	1,8	1,8	1,6	2,4
C	1	0,85	0,24	0	0,45	0,75	1,6	1,6	1,8	1,8	1,6	2,4
D	1	1,1	0,45	0,45	0	0,35	1,5	1,2	1,4	1,4	1,1	2,7
E	1	1,4	0,7	0,75	0,35	0	0,85	0,9	0,8	0,8	0,85	1,7
F	1	2,6	1,6	1,6	1,5	0,85	0	0,8	0,7	0,7	1,6	1,1
G	1	2,6	1,6	1,6	1,2	0,6	0,8	0	0,3	0,1	1,4	1,4
H	1	2,5	1,8	1,8	1,4	0,8	0,7	0,3	0	0,21	1,3	1
I	1	2,5	1,8	1,8	1,4	0,8	0,7	0,1	0,21	0	1,1	1
J	1	2,2	1,6	1,6	1,1	0,85	1,6	1,4	1,3	1,1	0	0,35
K	1	3,4	2,4	2,4	2,7	1,7	1,1	1,4	1	1	0,35	0
LD		19,8	12,84	13,09	11,65	8,8	13,05	11,9	11,81	11,41	13,1	17,45

On the example presented on the Figure 3.c, it can be seen that from point *D* to point *F* there are two routes, the first one with the distance of 1.1 [km] and the second one with 1.5 [km]. Assuming that the movement of transport vehicles is forbidden through the first shorter transport route then the transport vehicles must take second longer route.

Figure 4.c shows the procedure for determining the real distance between two locations, which results in the actual path that the transport vehicles will make with real distance value, as well as with the possibility of taking into account traffic restrictions for certain types of the vehicles

The Figure 5 shows the results of implementation of *load distance* method for all three distance calculation approaches for the same settings as they are presented on the Figure 2. Given to the fact that the calculated *Euclidean* and *rectilinear* distance D_{ij} have different values, the resulting value of the *LD* coefficient is also different, which ultimately leads to suggestions of different warehouse locations. On the other hand, by usage of real distances in calculation, results in location that differs from the optimum location obtained by application of *rectangular* or *Euclidean* distance. By applying of real distances, calculated results are credible and they cannot be throw in doubt. As a matter of fact, an obtained results i.e. distances corresponds to the actual situation on the ground, and the approach of determining of real distances between potential locations is a lot simpler than the measuring of the coordinates and calculation of the *Euclidean* and *rectangular* distances.

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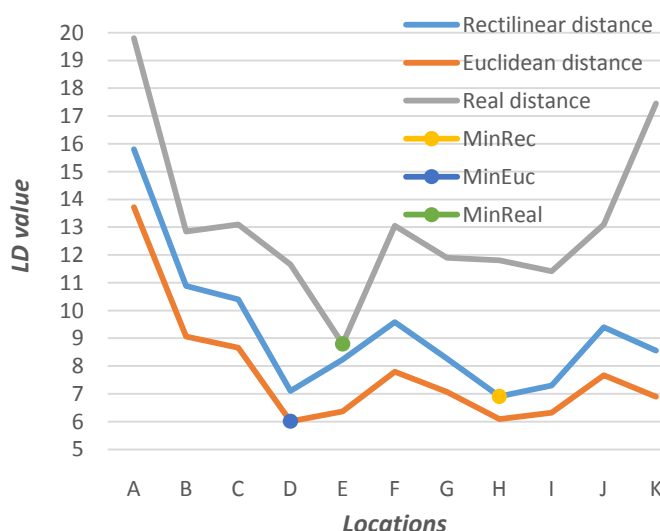


Fig. 5 Comparison of the results for load distance method for all three approaches of distance calculations.

WORK SAFETY AND ERGONOMICS AT THE WORKPLACE AN EXCAVATOR OPERATOR

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Abstract: *Ergonomics plays an important role in shaping safe and hygienic working conditions. Its main purpose is to increase the productivity of work in a way to not cause adverse health effects for the employee. In the article the profession of excavator operator, taking into account the scope of activities performed the qualifications and skills necessary to perform this profession were described. In the further part of the work, hazards occurring in the working environment of the excavator operator were identified. Then, selected ergonomic factors influencing the comfort of earthmoving machine operator were analyzed, in particular those that could cause excessive strain on the musculoskeletal system. Authors of the work focused primarily on discussing the construction and equipment of the excavator cabin as those elements that fundamentally determine the safe, ergonomic and comfortable work. The measurable effect of the work is to propose preventive works reducing the negative impact of these factors on the human body.*

Keywords: ERGONOMICS, OCCUPATIONAL HEALTH AND SAFETY, EXCAVATOR OPERATOR, WORKING ENVIRONMENT

1. Introduction

In Poland, Labor Codex [14] regulations oblige employers to provide employees with safe and healthy working conditions with appropriate use of the achievements of science and technology. This obligation can be implemented in many ways, for example by appropriate organization of work, ensuring compliance with health and safety rules, or responding to all the needs of adapting measures ensuring protection of life and health of people to changing working conditions. The obligation to provide health and safety at work also rests with the constructors and manufacturers of machines and other devices constituting work equipment. They should protect the employee from the negative impact of work environment factors, in particular: injuries, excessive vibrations, vibration, radiation, electric shock, the action of hazardous chemicals or other harmful work factors. In addition, these machines and devices should meet the technical requirements contained in Polish Standards [12, 13] and take into account the principles of ergonomics.

Ergonomics deals with issues related to the adaptation of the material environment and working conditions to the anthropometric possibilities, needs and limitations of man. It focuses on the proper shaping of the workplace, which will ensure the maximum possible work efficiency, at the lowest possible human biological cost [16]. Therefore, ergonomic solutions are characterized by proper adjustment to human anthropometry, comfort or lack of negative impact on the human body. The aesthetics are also important factor. The application of ergonomic principles at the workplace can bring many benefits, among which one should distinguish: improvement of productivity and quality of work, reduction of the workload, reducing the number of errors, increasing the safety of performed work (reducing the incidence of occupational diseases and reducing sickness absence) and increasing job satisfaction, which can also affect better motivation to work [7, 16].

In this article, considerations were taken in the field of work safety and ergonomics as excavator operator. Work related to the operation of this type of machine is heavy and physically demanding work. The excavator operator is exposed to many threat related to both the operation of the machine and the work environment. In addition, the operator's nuisance is related to the seated position of the body and the monotony of the activities performed, which can cause excessive strain of the muscular and skeletal systems. For this reason, it is necessary to adapt the basic working environment, which is the excavator cabin to human anthropotechnical conditions. The basic parameters of the excavator cabin in the field of safety and ergonomics include: protection against the effects of overturning (ROPS - Roll-Over Protective Structures) and falling objects from the height (FOPS - Falling Object Protective Structures), the position of the cabin in the machine, lighting, seat and the arrangement of control elements. Providing ergonomic working conditions affects the operator's safety, reduces the risk of accidents in which other people can

participate, and ensures the comfort of the employee. It is also important to equip the operator with appropriate personal protective equipment, depending on the place of work and working conditions.

2. Excavator operator profession

2.1. A description of the work and the way it is performed

The excavator operator is a profession consisting in the provision of services in the field of specialized earthworks and auxiliary works related to construction and mining. The basic duties of the machine operator include excavation, loading and handling of earth masses, loosening and transporting of spoil, sorting and deployment of materials in the landfill. In addition, the operator performs reloading and transport work, as well as cleaning works. The machine operator is also obliged to control its technical condition, replace operating fluids and remove minor defects [5, 8, 11]. The full scope of duties of the excavator operator is presented in Table 1.

Table 1: Responsibilities of the excavator operator

Before making earthworks
– preparation of the machine with equipment for earthworks, their repair and maintenance in accordance with the Operation Manual; – transporting the machine to the place of work; – preparation of the work area and works front in accordance with the rules and regulations of health and safety, ergonomics, fire protection and environmental protection; – getting acquainted with the survey of work, geodetic guidelines; – checking the technical condition of the machine and setting the levers and working units in the correct position.
During earthworks
– leveling the area using a excavator ; – excavation, ground loosening using a excavator; – arranging and sorting of backhoe work products; – controlling the quality of work performed; – performing works in accordance with the technological requirements specified for a given process and machine type; – reliable tracking of the machine environment in order to avoid collisions with other vehicles and objects and objects, as well as accidents involving people; – quick response to incorrect machine operation.
After making earthworks
– replacement of operating fluids and removal of minor defects in the excavator; – performing maintenance work; – reporting to the management about any observed defects and other irregularities related to the operation of the machine, removal of which is beyond the capabilities of the operator; – participation in acceptance and technical tests carried out after machine repairs; – preparing a daily report on the work performed; – cooperation with people supervising the work; – transporting the machine to the place of its garage.

2.2. Working environment

The working environment of the excavator operator are construction sites, gravel pits, sand pits, quarries, mines and machine parking spaces. The excavator operator works both inside and outside the cabin, therefore it is exposed to changing weather conditions. It should be emphasized that the ambient temperature has a important impact on the working conditions and the quality of the works performed. The operator works in a one- or three-shift system. The excavator operator is exposed to such dangerous and harmful factors as: noise, vibration, air dust, explosion, fire or electric shock. In addition, the work of the operator is associated with health threat associated with diseases of the musculoskeletal system and rheumatic diseases [8].

According to the International Occupational Risk Safety Data Sheet, the following hazard factors exist in the backhoe operator's work environment:

Factors that can cause an accident

- microclimate (cold, hot);
- fire, explosion;
- falls from a higher level to a lower one;
- slips and falls at the same level;
- lifting weights.

Physical factors

- crushing;
- collision;
- lighting;
- noise;
- vibration;
- electric current.

Chemical factor

- dust;
- chemical substances.

Ergonomic, psychosocial and work organization factors

- sitting body position;
- stress;
- work monotony.

In addition, among the threats at the excavator operator's position, you can indicate [11]:

- improper technical condition of the machine;
- terrain conditions when carrying out earthworks;
- failure to comply with traffic regulations;
- failure to comply with the safety rules during operation, inspection and repair of the machine;
- servicing the excavator without the required qualifications.

Incorrect operator behavior, improper work organization or work positions, as well as poor technical condition of the machine can cause accidents at work.

2.3. Education, qualifications and health and psychophysical requirements necessary to work in the profession of a excavator operator

The profession of a excavator operator can be carried out by a person who has: a minimum of 18 years, primary education, a current medical certificate stating that there aren't contraindications to the profession issued by a medical doctor and completed a specialized training and obtained a positive exam result. In addition, candidates starting the training should have at least one month's work experience in a position related to technical and operational servicing of machines in a given specialty. Otherwise, they are obliged to complete additional practical training [5, 8, 11].

In Poland, the legal basis for the training and qualification of machine operators and other technical equipment for earthworks (including excavator), construction and roadworks is the Regulation of the Minister of Economy of September 20, 2001 on health and

safety at work while using machines and other technical, construction and road equipment (Journal of Laws No. 118, item 1263) [10]. According to this document confirming the right to operate a backhoe loader and other working machines is the "Book of the working machine operator" with an appropriate entry regarding the type of allowances received, issued by the Institute of Mechanized Construction and Rock Mining. These rights in Poland are valid indefinitely. The basic requirement, necessary to undertake work in the profession of a backhoe loader operator, is to have the right to move the vehicle on public roads (driving license category B or T).

These rights can also be recognized in other European Union countries, depending on whether the occupation of the excavator operator is classified in the given country as a regulated or unregulated profession. In the case of a regulated profession, official recognition is required, which is carried out by the competent authorities of the target country. In addition, when there are major differences in the field of education or occupation, the recognition of professional qualifications is associated with an adaptation traineeship or an aptitude test. In turn, if in a given European Union Member State, the profession of an excavator operator is an unregulated profession, then the employer decides about employing an employee and recognition of his professional qualifications, pursuant to Directive 2005/36 / EC [2].

Based on the database of regulated professions [3], it can be concluded that the occupation of an excavator operator is a regulated profession in Norway, Iceland, Ireland and Bulgaria. In these countries, exercising the profession of an excavator operator by persons who have obtained their professional qualifications in another EU member state is possible only after the official recognition of professional qualifications.

The work of the excavator operator is characterized by high requirements regarding knowledge and driving skills, because even a small error can cause huge losses and expose to loss of life and health of people. In addition, servicing this type of machine requires the ability to concentrate, perceptive and reflex. Due to the specifics of the work performed, the excavator operator should demonstrate accuracy, reliability, care for safety, as well as technical abilities and manual dexterity.

Good health is required of the operator of the working machines, in particular the absence of defects in the organs of sight, hearing, movement and disturbances of balance, combined with resistance to long-lasting effort. Because the operation of work machines is a heavy and exhausting work, especially due to psychophysical loads. In addition, people undertaking the job of a backhoe operator should demonstrate the ability to cope in stressful and crisis situations.

The health contraindications preventing work in the discussed occupation include diseases of the musculoskeletal system (spine), heart disease, circulatory and respiratory system. Additionally, it should be noted that the exclusion for performing occupations related to driving vehicles is diagnosed dependence on alcohol and psychoactive substances. Due to strict health requirements, employees should undergo periodic medical and psychotechnical tests [11].

3. Analysis of selected ergonomic factors at the profession of a excavator operator

The excavator operator performs most of its duties in a sitting position. A working place is used when [4]:

- work requires high accuracy and precision over a longer period of time;
- a high degree of body and balance stability is required;
- all necessary items for work are located in an easily accessible place and do not require lifting above 150 mm;
- work does not require the use of high force and carrying loads over 4.5 kg;

- precise foot control is required.

The sitting, forced position of the operator's body causes that employees performing this profession are exposed to the ailments of the skeletal and muscular system. Diseases of the musculoskeletal system are characterized by one of the largest sickness absence of vehicle drivers.

In addition, in the working environment of the excavator operator, a significant risk factor for the health of employee is the work monotype, which causes of excessive strain certain parts of the body used to carry out professional activities. This factor may contribute to a decrease in concentration, and consequently, be the cause of an accident.

The excavator operator usually performs its work in the open air, in various weather conditions. Depending on the season, it is exposed to low or high temperature, high humidity, rain or snow. The indicated weather conditions may cause diseases of the lower and upper respiratory tract, disturbances of concentration and influence visibility (snow, bright sun).

The indicated factors have a fundamental impact on the safety, ergonomics and comfort of the excavator operator. Therefore, the cabin of the working machine should be characterized by appropriate construction and equipment.

3.1. Protective structure of the excavator cabin

The cabin is the basic equipment for earthmoving machinery (including excavator). Its main purpose of the application is to increase the safety and comfort of the operator during the work. The cabin protects the operator primarily against excessive noise, physical load and atmospheric factors. In addition, the floor of the cabin lined with a special mat allows to minimize the level of vibration. The ergonomics and operator comfort can also be influenced by such solutions as: rear side windows opening completely or partially improving ventilation, heated seat, or fresh air / recirculated air heater.

The basic requirement that must be met by currently offered excavator is to equip the cab with protective structures: ROPS (Roll-Over Protective Structures) and FOPS (Falling Object Protective Structures). The purpose of these constructions is to ensure an appropriate level of safety for the machine operator. The ROPS protective structure protects the worker from the effects of machine rollover during work. Its basic elements include: frame, brackets, mounting, support sockets, bolts, screws and suspension. Requirements that should be met by such a construction are set out in the standard PN-EN ISO 3471:2009 Earth-moving machinery -- Roll-over protective structures -- Laboratory tests and performance requirements [13]. The second type of protective construction is FOPS construction, which protects the operator from the effects of falling from the height of objects. Basic guidelines in this respect are specified in the standard PN-EN ISO 3449:2009 Earth-moving machinery -- Falling-object protective structures -- Laboratory tests and performance requirements [12].

In order to adequately protect the operator from hazards, the ROPS / FOPS protective structures should not be modified in any way. It is forbidden to weld, drill, cut or assemble parts. In the situation when the machine overturns or damages the cabin structure, it should be replaced with a new one. Assembly and dismantling of the cabin may only be carried out by specialized services. In addition, the machine should not be additionally loaded by mounting additional parts or adherence to the maximum weight guidelines indicated on the cabin. The operator is also obliged to carry out periodic inspections of the technical condition of the cabin, which are aimed at eliminating cracks, loosening screws and other damages. It should be emphasized that proper operation, systematic control [6, 9].

3.2. Visibility and lighting in the excavator cabin

The assurance of good visibility of the work field is of great importance in the work of the excavator operator. For this purpose, it is necessary to optimally place the cab in the working machine. In modern excavators, lift cabins are mounted, which allow changing its position from side to side, forward and backward, providing the operator with sufficient visibility during work. In addition, when constructing machines, certain positioning parameters of the eye ellipse are assumed and a reference line is set. The vehicles are constructed in such a way as to ensure the widest possible and the best visibility, taking into account the position of the posts, which is particularly important when performing precise tasks.

The level of visibility is not only influenced by the proper cabin construction and its location, but also by the equipment in the appropriate quality and size of the glass. Large, rounded windscreens and wide side windows guarantee visibility in all directions. A good solution used by machine manufacturers are also appropriately tinted glass protecting against excessive light (sunlight).

The operator of the excavator works at different times of the day, sometimes also in the evening and at night. Therefore, lighting is required to ensure proper visibility and operator comfort. The currently produced machines use lighting, which is characterized by high luminous efficiency. Correct lighting not only improves visibility, but also reduces the psychophysical load of the operator and increases the level of work safety.

3.3. Entering and exiting the excavator cabin

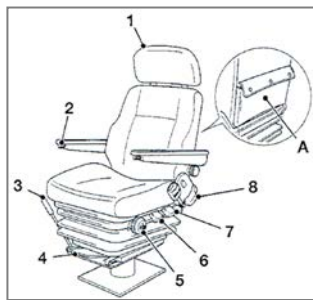
The safety of the excavator operator depends not only on the structural elements inside the cab. The external parts of the cabin are also important. Some of them include entry levels. They should guarantee safe entry and exit of the cabin, which is why they are made of non-slip materials. The correct operator behavior is also very important. Before entering or exiting the cab, make sure that the machine is stopped and properly parked. When entering and exiting the excavator, be facing the machine and use three support points (fig. 1). In addition, it is forbidden to use the rudders of the machine or steering wheel as handles [1, 9].



Fig. 1. Support points on the inside of the excavator's cabin [1]

3.4. Seat (armchair) of the excavator operator

Work in a sitting position is much less physically aggravating as opposed to a standing position. Nevertheless, long-term sitting of operators in the cabin may negatively affect the load on the lumbar spine. Therefore, long-term sitting is difficult for a man. It can cause spinal pain syndrome, flabbiness of the abdominal and back muscles, pressure on internal organs, as well as breathing difficulties. Therefore, the most important influence on shaping the correct position of the working machine operators is the seat (armchair). It should be properly adjusted to ensure work comfort and reduce operator fatigue. In addition, the height of the seat should be set so as to ensure free access of the operator to the rudders of the machine. In addition, the optimal seat height should allow the brake pedal to be fully depressed while the back touches the backrest [1]. An example of a excavator seat is presented in fig. 2.



- A - pocket
 1 - headrest
 2 - armrests
 3 - 360° rotating plate
 4 - forward / reverse adjustment
 5 - weight
 6 - height and inclination (front)
 7 - height and inclination (back)
 8 - seat back angle

Fig. 1. Construction of the operator's seat [9]

In terms of ergonomics, the basic elements of the seat include [4]:

- **headrest** – is intended to hold the head of the machine operator, and consequently prevents spinal injuries during sudden head movement backwards. It also affects the comfort of work, allowing a comfortable head support;
- **armrests** – they provide additional support for the body and allow relief of the neck and shoulder section; they affect the stabilization of the upper limbs, which is necessary in the operator's work. In addition, armrests help in sitting and standing, lining them with a soft fabric protects against an ulnar injury;
- **seat height** – it should be adjusted to ensure proper access to machine control devices and proper leg placement (thighs should be placed horizontally and the leg should be vertically);
- **forward / reverse position adjustment** – allows you to adjust the optimal position of the driver's seat with respect to the pedals and control and monitoring elements; it affects the safety and comfort of the operator's work, without causing any postural loads; in addition, the adjustment of the forward / back position together with the seat height adjustment provides the operator with adequate leg space, which also reduces the effects of static load and discomfort. For machine operators, the optimal height of the recess in a sitting position should be 70 cm (but not less than 66 cm);
- **rotary plate** – it allows to change the position of the operator's seat to any angle;
- **backrest reclining angle** – determines the pressure on the intervertebral discs; Increasing the inclination angle means that the greater part of the operator's weight is transferred to the backrest, which reduces the compressive force acting on the disks. The optimal inclination of the backrest should be 100° – 110°
- **the angle of the seat panel** – increasing the inclination angle of the seat plate allows you to maintain proper contact with the backrest and prevents the sliding of the operator; however, a larger inclination angle is not conducive to the proper lordosis (bending) of the spine.

3.5. Ergonomic arrangement of control elements

Performing by the operator of a excavator specific earthworks is associated primarily with the operation of control devices. Ensuring their optimal position, in accordance with the employee's anthropometry, influences the safety and comfort of work, and makes it more efficient and less stressful.

Machine manufacturers must make the following rules when making decisions regarding the arrangement of control devices [4]:

- quick operations requiring high precision are done with fingers or hands; therefore, buttons, tilting switches and rotary knobs are preferred for this type of operation;
- operations requiring strength and relatively lower precision are best done with upper limbs (with arm involvement) and lower limbs; for this type of operation it is best to use levers, cranks, handlebars and pedals;

- control elements should be such that they can be easily reached (in the optimal motor zone); manual controls should be arranged with the elbow and the shoulder.

The control devices, which should be compatible with the movement stereotype, are important in efficient and safe driving of the excavator. This means that the movement of the control device forward, up, to the right corresponds to positive effects (activation, start-up, parameter increase). On the other hand, the movement of the control element towards oneself, downwards, to the left causes negative effects (switching off, stopping, reducing the parameter). In addition, vehicles should pay special attention to the unification of the location of control devices, which means that a group of devices should always have the same location and move in the same directions [4].

The optimum position of control elements in vehicles requires taking into account the dimensional relations between the operator and the control station. In addition, the design of the site should take into account the user's reach zones. Control elements in the cab space are located in three zones: first, second and third degree. Table 2 presents their distribution and general characteristics.

Table 2: Comfort service zone in the excavator cabin

COMFORT SERVICE ZONE IN THE EXCAVATOR CABIN	
ZONE 1 device operation does not require detaching the shoulders	– light switches; – direction; – sound signal; – hand brake; – on / off switch; – tool control levers; – wiper; – sprinklers.
ZONE 2 device operation does not require detaching the lumbar region from the backrest	– blower on / off switch; – air conditioning control; – sun visors; – cabin light buttons; – electric lighter.
ZONE 3 device operation takes place within limited comfort, but does not interfere with other elements	– engine cover openings; – handles for opening windows; – window latches; – seat adjustment elements; – the main circuit breaker of the electrical system.

Safe and comfortable driving also requires proper positioning of the steering wheel in relation to the pedal. The tests [15] carried out indicate that there is a relationship between the position of the pedal relative to the seat and the resistance posed by the pedal, and the time and value of moving the foot while driving. In addition, the value of the pressure on the pedal and its duration is correlated with the load on the lower limbs. Prolonged peeling on the pedal with excessive force contributes to reducing the accuracy of control, and thus lowering the level of safety of work. In addition, performing this activity may cause lower limb and above all knee disorder.

In order to reduce the load on the knee joint, it is necessary to properly position the seat in relation to the pedal so that the angle in the knee joint is between 120° – 130°. In this angle range the load value is low and the pressure force is highest [15].

3.6. Personal protective equipment for excavator operator

Personal protection means are aimed at eliminating the danger resulting from the threat associated with the technological process or the work environment. They are usually used when other methods of protection are inadequate [7].

Personal protection measures very often affect the ergonomics and comfort of work. Some of them may cause employees to feel

uncomfortable and therefore it is necessary to strive to minimize their wearing time. It is recommended that the use of personal protection measures is only due to the working conditions and needs at a given place of work. In addition, to ensure the maximum possible level of employee protection, places and areas of use should be clearly marked. The basic means of personal protection at the excavator operator position include:

- protective helmet;
- protective gloves;
- respiratory mask;
- safety glasses;
- hearing protectors;
- safety shoes;
- protective mask;
- dust mask;
- protective clothing;
- clothing ensuring visibility (reflective vest);
- safety harness.

Summary

Based on the reading of this article, the following conclusions and final statements can be formulated:

1. The profession of an excavator operator may be carried out by a person who is 18 or older, has basic or vocational education, completed specialist training and passed the exam with a positive result, and received a "Book of a working machine operator" with an appropriate entry regarding the type of entitlements received.

2. The excavator operator is exposed to many dangerous and harmful factors, among which the highest level of occupational risk is characterized by: noise, vibration, electric shock or dust.

3. The work of an excavator operator is a tedious and tiring work, requiring perceptiveness, continuous concentration, precision, as well as dexterity of upper and lower limbs. In addition, the machine operator works in a sitting, forced position, which means that their ability to change the position of the body is small. In addition, he performs a monotypic work, which also affects the load on the skeletal and muscular system of the employee and the decrease in concentration that can cause accidents.

4. The direct position of the excavator operator's operator is the space of the machine's cabin in which he spends most of the day's work. Its construction and equipment depends on the comfort and safety of the employee and other people in the immediate vicinity. Therefore, all elements of the cabin should be adapted to human anthropogenic conditions, be characterized by easy and intuitive operation, and be appropriately located.

5. In terms of ergonomics, the protective structure of the cabin against the effects of overturning (ROPS) and falling from the height of objects (FOPS), operator's seat (seat), cab position and lighting guaranteeing proper visibility as well as the arrangement of control devices are of great importance. In addition, the safety and comfort of the operator's work also depends on his behavior, how to operate the machine, compliance with the general rules of health and safety and the use of protective measures.

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MEASUREMENT OF ELECTROSAFETY PARAMETER WHEN PROVIDING THE ELECTRIC POWERS ON BOARD THE SHIP FROM COAST

ИЗМЕРЕНИЕ ПАРАМЕТРОВ ЭЛЕКТРОБЕЗОПАСНОСТИ ВО ВРЕМЯ ЭЛЕКТРОСНАБЖЕНИЯ СУДОВ ОТ БЕРЕГОВЫХ СЕТЕЙ

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Abstract: The world market offers the greater nomenclature of the facilities for measuring parameters of electrosafety, as there is always a demand on them. The majorities of facilities do not guarantee accuracy or in general cannot value the measured parameter unless they do execute requirements and conditions of their operation. In particular, measurement of the resistance grounding device (GD), at resistivity of the soil more than 500 Ohm.m and in large concourse of the underground communication (seaports) requires the high qualification of the operator and large number of measurements. For instance, in the article unpinned way of measurements GD is considered based on measurement of phase voltages of network with deafergrounded of neutrality in connection with the investigation sidebar of the grounding device contour. The Offered way of the determination of the resistance GD, in basic range (0.1-10 Ohm) allows to value the garbled resistance GD with necessary accuracy.

KEYWORDS: GROUNDING DEVICE, DEAFERGROUNDED NETWORK, SEAPORT COMMUNICAITION

1. Introduction

The Electric power worked out on ship, on prime cost vastly cherish electric powers, worked out powerful coast power station. When parking in port ship possible to connect to coast power system and remove their own generator units from work. In this case it is spared fuel and butters, withdraws the noise in engine rooms, is freed part personnel for repair work.

For presenting the electric powers on ship quays port must be corresponding to image are equipped with. The Difficulty consists in that in coast set of alternating current is mainly used three-phase system with grounded neutral, but on court zero point windings generator from body ship is insulated.

At connection of the ship power station to coast network necessary body ship safely to ground to exclude the electrocorrosion of the body [1] .

2. Preconditions and means for resolving the problem

At supply ship with use the three-phase transformer grounding vein of the ship cable serves for earth of the body ship.

While standardizing and designing the **grounding device (GD)** the probability of injuring person by electric current is taken into consideration. In no area of the technology and in general in real life it is impossible to provide full safety for people. Probability of the contiguity with dangerous voltages can be sharply reduced by careful design of GD.

The Distributing networks with nominal inter phase voltage of 380V execute the fourconducting and threeconducting. They are connected to networks with higher voltage through lowering transformers. Neutral of windings of the undermost voltage of the transformer is grounded. Its

connected to grounded substations. Zero wire is grounded repeatedly.

The rates on GD require conditions, which they must satisfy [2]. The main are requirements, defining conditions of electrosafety.

For GD, used for grounding of the networks of the secondary voltage 380/220 B, potential GD must not exceed 125 B, but its resistance-4 Ohm.

To calculate GD - means to define potentials in any point of the space under given current, in particular potential GD, and also potentials in typical points of the surfaces of the land. The land usually is not of the same origin in composition and moisture content. Its resistivity changes when moving horizontally and in depth. As a rule, theoretical calculations without concrete measurements of resistivity of the soil, give greater inaccuracy, therefore results of theoretical calculation are used beforehand, on stages of the projection.

Available electroinstallations require periodic energo checkup, preventive maintenances and checking the requirements of electrosafety, periods of which are stipulated in acting documents. In Georgia the checking of the specified requirements must be produced much more often, since a big part of electroequiments has worked out its resource. The requirements of acting rules of the SAFETY and Rules to Technical Usage of Electromounting install the rates a parameters of electrosafety and periods their periodic checking, to which priority is given to:

1. The measurement of the resistance of insulation;
2. The measurement of the resistance of the loop "phase-0";
3. The measurement of the resistance of the grounding device.

The world market offers the greater nomenclature of the facilities for measuring parameters of electrosafety, as there

is always a demand on them. These facilities, as a rule, measure the specified parameters separately that requires from buyer rather significant financial expenses (refer to, for instance, nomenclature of one of the world leader producing measuring instrument, company "Sonel" www.sonel.pl).

The majority of facilities do not guarantee accuracy or in general cannot value the measured parameter unless they do execut requirements [3] and conditions of their operation. In particular, measurement of the resistance GD, at resistivity of the soil more than 500 Ohm.m and in large concourse of the underground communication (seaports) requires the high qualification of the operator and large number of measurements.

Sometimes resistivity soil outside of sidebar of the earth forms more than 500 Ohm.m (the utter asphalt, stony or sandy soil,...). Consequently it's necessary to jab in current and potential electrodes (the pin) soil enough deeply, but if it fails jab several pins, having connected them in parallel that is possible if there are not found power or high-tension cables in the earth. So we must be sure beforehand, that there are not such cases. In these cases, the determination of the site underground communication dispenses for the customer are more expensive, than measurement of the resistance of the earth itself. Here it's impossible to measure only. For reception of the reliable result it is necessary to do as minimum several measurements and if their results differ more than 10 %, continue driving the pins in the other place.

At first thought this is an exclusive matter, but practically we come across it often, in the seaport and particularly in the central part of any big city.

The conclusion is one: alternative unpinned ways of measurements of the resistance of the grounding are necessary.

Such works have already been carried out, however they are found at a rate of patent and recommendation moreover majority of them are insufficient to provide the main range of the measurements (0.1-10 Ohm) with necessary accuracy (5-10 %).

3. Objective and research methodologies

We Offer our variant of measurements of the resistance of the grounding, in basic range of the measurements (0.1-10 Ohm) [4].

The main idea is such a: If on zero wire (N) two equal opposite phase of the current, are passed the fall of the voltage on it will be equal to zero regardless of resistances of the zero wire (Z_0).

Let's consider the most simplest electric circuit (fig. N1) of the passing of the current of the load (I_n), under its connection between phase and zero in single-phase network with deafgrounded of neutrality of the power transformer.

Assume, we have measured the voltage on load (the points 2;3)- U_n , and without it - U_{xx} under given resistance of the load Z_{23} . We shall define how far will the voltages U_n and U_{xx} differ.

$$U_{xx} - U_n = \frac{U_n}{Z_{23}} \cdot (Z_{01} + Z_{12} + Z_{03}) \quad (1)$$

If we connect point 3 with grounding device GD, the expression (1) will look like:

$$U_{xx} - U_n = \frac{U_n}{Z_{23}} \cdot (Z_{01} + Z_{12} + \frac{Z_{03} \cdot Z_{GD}}{Z_{03} + Z_{GD}}) \quad (2)$$

From (2) follows that any GD of the load reduces the fall of the voltage to network and promotes the increasing of electrosafety.

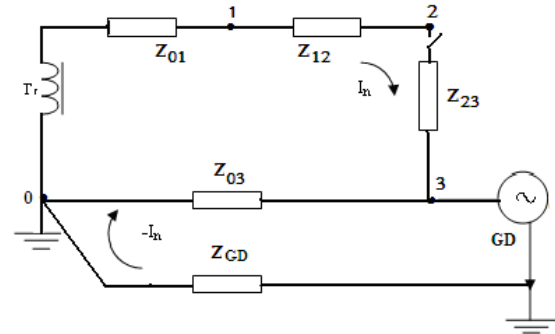


Fig. N1. EQUIVALENT SCHEMES to SINGLE-PHASE NETWORK UNDER LOAD

Tr-supplying transformer; Z_{01} -internal resistance of the transformer; Z_{12} -resistance phase wire; Z_{23} -resistance of the load; Z_{03} -resistance of the zero wire; GD-separate sidebar of the earth (grounded device); Z_{GD} - a resistance grounding network.

Assume, that between point 3 and GD we shall connect the power source with the known voltage U_2 moreover we select it in such a way, that contour current (the contour GD, 0, 3, GD) was equal to a current of the load and inverse on direction, then equality is fair:

$$U_2 = U_n \cdot \frac{Z_{GD}}{Z_{23}} \quad (3)$$

From which follows that in this case it is possible to define the resistance of the grounding Z_{GD} , regardless of the resistances of the zero wire:

$$Z_{GD} = \frac{U_2}{U_n} \cdot Z_{23} \quad (4)$$

If we measure not voltage (U_n), but current of the load (I_n) (for instance, current mite), that expression (4) is simplified:

$$Z_{GD} = \frac{U_2}{I_n} \quad (5)$$

Inaccuracy of the measurement of the resistance GD will depend on inaccuracy of the measurement of the voltage and current moreover if their relative inaccuracy are equal or close, then resistance GD will be calculated maximum exactly:

$$\frac{\Delta Z_{GD}}{Z_{GD}} = \frac{\Delta U_2}{U_2} - \frac{\Delta I_n}{I_n} \quad (6)$$

But must not be forget about that concourse of the voltage has its own (internal) resistance so under responsible measurements (for instance, under normalized importance of the resistance GD less 0.5 Ohm) it is necessary to take into account.

It is enough to measure the resistance of the concourse of the voltage once to hereinafter if required, correct the results of the measurement GD, entering corresponding adjustment.

The essence of the second method is that the method of measuring the resistance of the earthing switch in a network with a blindly grounded neutral provides for the transmission of the load current included between the phase conductor and the earthing switch and the determination of the desired parameter on the basis of the magnitude of the current flowing through the load. In addition, the load current is branched in two potentiometers with the formation of one of the nodes of the measuring bridge circuit composed by these potentiometers together with the zero wire and the required earthing resistor; The first potentiometer is connected to the neutral wire, and the second - to the earthing switch; the balance of the bridge is monitored by means of a zero indicator connected to the diagonal of the bridge between the earthing switch and the first potentiometer. The resistance of one or both potentiometers is adjusted until the bridge is balanced, and the resistance of the earth electrode is determined taking into account the ratio of the resistance of the potentiometers at the moment of bridge equilibrium.

FIG. 2 is an electrical circuit for explaining the nature of the proposed method for measuring the resistance of the earthing switch. The source of a single-phase network of an industrial frequency (50 Hz) is shown which, via a phase conductor with resistance R_c , through a load resistor R_z is connected to a measuring bridge circuit whose arms are formed by potentiometers R_1 and R_2 , zero wire with resistance R_0 and the required resistance of the earthing switch R_x .

For the indicated measurement scheme, the equilibrium condition of the bridge has the form:

$$\frac{R_2}{R_x} = \frac{(1-K) \cdot R_1}{R_0 + K \cdot R_1} \quad (7)$$

where K is the known part of the resistance of the potentiometer R_1 .

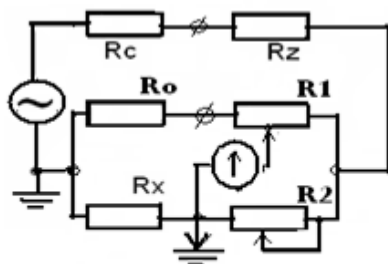


Fig. 2

From the relation (7) it follows that the resistance of the zero wire R_0 participates in the measurement of the resistance of the earth electrode R_x , but if the resistance of the potentiometer R_1 is chosen, for example, three orders of magnitude greater than R_0 , then it can be neglected.

In view of this

$$R_x = \frac{K}{(1-K)} \cdot R_2 \quad (8)$$

that is, a direct measurement of the resistance of the earthing switch is provided.

4. Conclusion

Various variants of the device for implementing the proposed method for measuring the resistance of the earthing switch are possible, for example, the use of bridge arms with a close inductive coupling (transformer bridges), the use of resistive, capacitive bridges and bridges of other types.

The Offered way of the determination of the resistance R_D , in basic range (0.1-10 Ohm) allows to value the garbled resistance R_D with necessary accuracy.

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EVALUATING THE IMPACT OF SECURITY MEASURES ON CONTAINER SUPPLY CHAINS

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Abstract: The present article analyses the development and impact of the container security initiatives on maritime supply chains. Security issues concerning container liner shipping require complex approach and application of integrated IT systems on national and international levels. During the last two decades various initiatives have been applied to ensure for higher level of security of container transportation. The background and aim of these initiatives are studied in detail and their effect is evaluated. The results show that container transportation still has vulnerable nodes and the application scope of security initiatives is to be widened despite the high costs.

Keywords: CONTAINER LINER SHIPPING, CONTAINER SECURITY, PORT SECURITY, MARITIME SUPPLY CHAIN

1. Introduction

During the last two decades several international regulations have been introduced focusing on security issues of containerized cargo flows. These are the International Ship and Port Facility Security (ISPS) code, the Container Security Initiative (CSI), the 24-hour Advance Vessel Manifest Rule. To support the global supply chain as concerns proactive approach for incidents prevention and tracing of containers, ports and logistics stakeholders have introduced new technologies based on real-time information systems [4]. The transportation of containerized cargoes requires efficient supply chain management and relevant security measures. European ports have been given the opportunity to establish the required security level in response to newly set international standards. The European Shipping Containers Surveillance system, implemented via various EU funded projects includes a number of recommendations – standardization, national regulations, policy guidelines. The present article analyses the effect of the global container security measures. The processes of logistics chain of container transportation are presented, outlining the vulnerable nodes as concerns security issues. The positive and negative effects of the security measures are assessed via quantification of direct costs to logistics stakeholders.

2. Structure of container supply chain and security issues

Container supply chain is characterized by complex interactions between numerous subjects, production areas, regulating bodies and polices. At the beginning of the container supply chain are the shippers who require the services of intermediaries that will ensure for the international transportation of containers including maritime transportation. At the other end of the container supply chain are the consignees who require timely and quality delivery of goods. Most of the container cargo flows are initiated on the basis of commercial interactions and relations between sellers and buyers. In most cases, however, it is the shipper who disposes of the exact information about the type and quantity of cargoes shipped in containers. The latter is of fundamental importance as concerns the security of container supply chains. Due to the relatively medium-scale of the shippers' enterprises, these companies do not have access to resources for increasing the supply chain security level.

Forwarders, on the other hand, have better overview of the supply chain but their "hybrid" role both as carriers for their clients or shippers for carriers can be a prerequisite for hindered access to information regarding the cargoes. Forwarders companies are predominantly medium-sized companies that are not able to fulfill the costly security measures.

The surveillance and monitoring of containerized cargo flows is the responsibility of each governmental body as well as of the customs office. Customs are responsible for securing that containers

and the cargoes are customs cleared during exports, transit and import procedures.

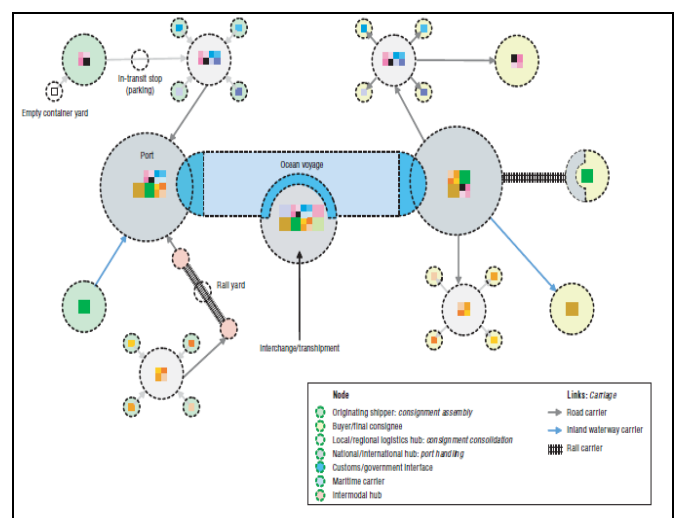
The security of the container supply chain is responsibility of all participants and any disruption of security can compromise the entire supply chain. The vulnerability of containers due to environmental factors is related to railways warehousing, road transport stoppages, during storage and loading at container terminals. As concerns these vulnerable nodes all efforts should be focused on ensuring the physical safety of storage areas and minimization of unauthorized access. The monitoring of containers transportation should be implemented within "real-time" environment and at the right moment, i.e. there should be reliable information at any moment about the location of the containers.

The physical flows of container supply chains constitute the movement of the containers and represents the material flows from a security point of view. In general, the network of nodes and edges in containers supply chains consists of several processes (Figure 1):

- consolidations of cargoes;
- transportation to the port of loading;
- handling at the port of loading;
- transportation by sea;
- handling at the port of discharge;
- land/inland waterways carriage to the consignee.

More efficient routing of containers with minimum stoppages during transportation and decreased storage time increase the safety of cargoes transportation and ensure for higher revenues for all participants.

Figure 1: Container supply chain for exports [2]



3. Container supply chain security measures

Container supply chain security measures can be classified as follows:

- focusing on the monitoring of the container content;
- focusing on the containers integrity;
- aiming at the ensuring of safety of the environment during the transit and handling of containers;
- related to the monitoring of container transportation within the entire supply chain;
- ensuring and usage of supply chain information.

Being a complex structure, each container supply chain element is aiming at optimization of its own processes. According to the well known principle in logistics management, the aggregating of individually optimized relations in some cases results in non-optimal supply chain. Non-harmonized practices, incompatible operations and information management systems, uncoordinated regulations, both on national and international level, can lead to vulnerability of the security system due to lack of coordinated approach.

As concerns the security of physical flows the following should be considered:

- the containerization point is of prime importance as concerns security since it is the last point where the contents of the container can be visually identified and compared with the respective invoice or waybill. Until the moment of decontainerization all information regarding the content of the container will be evident only in the cargo documents (freight manifest, Bill of Lading, etc.)
- containers are most vulnerable when they are standstill which means that security measures are most important in those nodes where containers are being handled or stored;
- crossing of international borders includes extensive customs control that leads to potential delays;
- most of the containers traffic transits through at least one sea port which levels of security and relevant security measures are at a different level.

There are two major types of physical surveillance of containers: X-ray scanning (non-intrusive inspection) and direct physical examination. The latter usually involves at least 8-10 hours per container which can lead to potential delays for the entire delivery.

The Container Security Initiative is structured around the concept of “pushing the border back”, i.e. identifying the security risk at the point of origin and before shipment and serving as protection against pertaining risks during containers transportation. Presently, there are over 50 ports that have been approved for applying the Container Security Initiative (SCI) as shown in Figure 2.

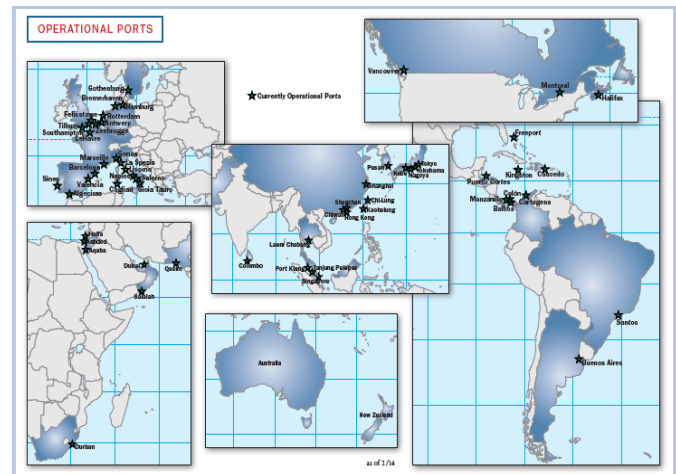
As of April, 2008 the European Union and the USA have signed an agreement for activation and expansion of cooperation in customs procedures and mutual cooperation as concerns containers security. The agreement concerns cooperation for ensuring of containers transportation safety and related issues and is applicable to all containers carried by sea, irrespective of their origin, that are imported, handled or transiting through EU and USA.

The container security initiative (CSI) consists of the following four elements:

- high-risk containers are identified via automated information;
- containers are pre-screened for high-risk identification;
- usage of special equipment for detection and screening of high-risk containers ensuring of inspections without delay;

- usage of smarter containers that are tamper-proofed [6].

Figure 2. Ports, applying the Container Security Initiative [6]



The “24-Hours Rule” is based on automated information for identification of higher risk containers. As of 2002, all carriers are obliged to submit electronic cargo manifest to the US Customs before cargo loading. The mentioned rule is also applicable to transit and empty containers as well as bulk and conventional general cargo shipments. The cargo manifest is the document that legalizes any cargo carried by a seagoing vessel and contains information about the shipper, consignee, notify party, port of origin, port of destination and cargo description. In this way customs authorities are closely monitoring the shipment content along with the time periods needed for the container transportation. The container is being tracked if risk is identified or dangerous goods are carried in the container. The mentioned information, transmitted electronically, is used both for exports and imports. It is the responsibility of the carrier to ensure for information provision which is accurate and complete and is submitted at the required time. Some states do not disclose information contained in cargo declarations for security reasons until the process of cargo manifest filing is completed – the relevant information might be published only after loading is completed and the vessel has left the port.

Screening systems usually use x-rays, gamma-rays machines and GPS. The mentioned technology allows for fast inspection without delay apart from the technological time needed for the screening. This type of equipment can identify specific materials which can potentially pose a risk to the environment.

4. Cost analysis of container supply chain security measures

The costs pertaining to the implementation of security measures can be indirect and direct. The latter are the capital costs, necessary for the design and implementation of the security network:

- purchase of new equipment for physical structure protection;
- adoption and/or implementation of security regulations;
- implementation of security policies and regulations;
- employment of trained security personnel.

Costs that are classified as indirect are the costs related to the system operations: equipment maintenance, application of efficient management strategies, response costs to incidents, costs related to management and operations recovery and reconstruction of infrastructure.

For the ocean carriers the costs include the following elements [1]: costs for setting up a new system for documentary transactions, costs for increased communication, personnel training, increased labor costs (cargo handling and inspection, operations of special equipment), usage of security-related equipment. For example, vessels are equipped with Automatic Identification System (AIS). Carriers also use software for monitoring and tracing of container movements for preventing incidents and illegal actions. As for specific technologies being used - Global Positioning System (GPS), Radio Frequency Identification (RFID) and electronic seals of containers also enhance the tracking of containers. It could be necessary that special security equipment is installed on board the vessels to enhance the global security measures.

The costs of security measures that impact the entire container supply chain are related to:

- time delays due to security inspections, whereas the costs will be higher at the very beginning of the transportation process; delays in the delivery time will eventually lead to penalties to parties and/or damages due to decreased cargo quality;
- costs for providing cargo manifest in advance, whereas this requirement is based on the “24-Hours Rule”; although the information contained in the manifest is not disclosed at an early stage same could be used for illegal purposes and losses related to the latter are borne by several participants in the supply chain;
- the introduction of additional state taxes related to the design and maintenance of the security infrastructure and equipment;
- increased insurance premiums despite the applied new security systems on board ships and in ports.

In addition to the expected higher revenues, the most important benefit from increased security measures is the increase of cargo flows to and from ports applying higher security level system.

Risk analysis and risk management, performed by the various participants of the container supply chain, will enhance the evaluation of potential gains and reduce the cost of investments. Decision makers are to explore the fixed costs for application of the international regulations, increase costs for communication exchange along with increase of operational expenses, inspection and personnel training costs. It should be noted that the economic benefits of increased security measures for container supply chains are more evident in the long run therefore strategic management is to implement such measures.

According to [3] the cost benefit analysis of a new automated cargo manifest estimates direct saving to American importers only at USD 22.2 billion over 20 years and savings of USD 4.4 billion for the American government over the same period.

The CSI requires that all foreign states invest in special equipment (screening, detection) and provides guidance to port management via trained personnel. However, recent evaluations have shown that the overall costs of increased security measures will be borne by the clients and will be calculated as additional costs per container. Depending on the type of port management, the state or the municipality will initially invest in the required special equipment which returns are calculated on the basis of increase of prices for shippers, carriers and port services and, finally, of the final prices of goods.

Given the higher costs of security measures application the less developed countries/ports will bear the largest burden of market share loss. Due to the high competition in the shipping market, both tramp and liner, these companies/ports will inevitably face reduced revenues and loss of market position. According to [5] the CSI program proposed by the US will be difficult to implement by some small developing countries because the cost per port for that program varies between USD 1-5 billion. Table 1 presents an approximate estimation of related costs for the implementation of obligatory and optional security measures.

Measures		Initial costs (approximate, mln USD)	Yearly cost (approximate, mln. USD)
Port facility assessment	Security	27,9	8
Port facility plan	Security	N/A	N/A
Port facility officer	Security	N/A	N/A
Port training	Facility	N/A	N/A
Port facility equipment/staff	Security	N/A	N/A
24-Hour Automated Manifest System	Security	281.7 to 10 000	281.7 to 10 000
Container Security Initiative	Security	N/A	N/A

Table 1. Cost of container supply chain security measures [7]

When implementing only the obligatory requirements as per the IMO conventions (for example, ISPS), apart from the optional requirements (CSI), less developed countries will inevitably face decrease of cargo flows and cargo turnover at ports as supply chain participants would prefer faster and more secure transportation. Even in the medium run the latter will distort the international trade patterns.

5. Conclusion

The present paper analyzes the framework of security measures in container supply chains, outlining the strengths and weaknesses of the mandatory and optional security measures. The existing security measures aim at the protection of international and national interests and ensuring of safe environment.

All the participants of the container supply chain are challenged by the fact that there are no universal standards concerning the container transportation apart from the mandatory regulations of IMO at ports and for sea carriage (ISPS, SOLAS). The other edges of the container supply chain – road and rail transport, as well as inland waterways, are still the vulnerable parts of the system.

The available solutions for security enhancement of container supply chains, however, involve certain delays in cargo flows. All participants should bear the cost and responsibility of the applicable measures. Shippers are responsible for the containers content and provide the relevant info via automatic cargo manifests transfer to the national customs authority as per the “24-Hours rule”. On the other hand, ocean carriers are responsible for vessel’s deviation from the customary route and are obliged as per ISPS code to provide advance ships’ arrival information to port authorities. The strict security procedures of the CSI will lead to delays in the transportation process especially for carriers and shippers. A detailed analysis has been made regarding the cost of the security measures and its effect on the container supply chain. The results show that the benefits from increased security measures are higher for all supply chain participants, allowing for protection from usual hazards during the transportation process.

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RESEARCH OF INTELLIGENT TRANSPORT SYSTEMS MANAGEMENT OF CONVOY OF UNMANNED VEHICLES WITH THE LEAD PILOT VEHICLE FOR WORK IN THE NORTH OF THE RUSSIAN FEDERATION IN THE ARCTIC AND ANTARCTIC

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Abstract: In the Russian Federation the problem of creating a transport system to control movement of the column of unmanned vehicles to enhance the efficiency and safety of passenger transportation in remote regions of the North, the Arctic and Antarctica is solved for the first time. The study was first developed mathematical models and algorithms for control and interaction of intelligent transport systems of traffic control columns, control system movement of the lead pilot vehicle of a column of unmanned vehicles, the motion control system driven unmanned vehicle convoys of unmanned vehicles.

KEYWORDS: INTELLIGENT TRANSPORTATION SYSTEM, UNMANNED VEHICLE, MOTION CONTROL SYSTEM, VISION, NAVIGATION, COMMUNICATIONS, ACTUATORS, RADAR, CAMERA, ALGORITHMS, AND SOFTWARE.

1. Introduction

During the development of electric vehicles and unmanned vehicles [1] in the Russian Federation, a number of works on the study of intelligent transport systems are carried out. One of the objects of the research is the intelligent traffic control system for unmanned vehicles with a leading pilot vehicle, improving the efficiency and safety of cargo and passenger traffic in remote regions of the North, the Arctic and Antarctica.

Leading international firms such as Daimler-benz, Volvo, IVECO, MAN, Scania etc. are actively working on the creation of autopilots for commercial vehicles in the convoy with a pilot car driven by drivers.

However, repetition of the foreign technical solutions providing satisfactory functioning of control systems of unmanned trucks as a part of columns in road climatic conditions of the Western Europe appears impossible for road climatic conditions of the Arctic zone of the Russian Federation.

It is generally accepted that the use of unmanned cargo vehicles in the convoy is the most effective, in which the following results are expected:

- improvement of road safety, as it minimizes the negative impact of the human factor, which according to statistics is the cause of almost 80 % of road accidents;
- achieving fuel savings of up to 20 % ;
- increase in transportation productivity by 1.3-1.4 times;
- providing comfortable working conditions for drivers in driven trucks;
- minimization of harmful effects on the environment;
- reducing the need to maintain a large staff of professional drivers with high wages;
- the possibility of integration of unmanned transport systems into the technological process of enterprises, primarily large transport and logistics centers, ports, etc., ensuring their continuous round-the-clock operation.

The problem of increasing the productivity of road transport is a priority. The prospects of logistics are based on the development of transport and infrastructure technologies. At present, domestic and foreign transport enterprises implementing a modern logistics technology, transportation and freight handling: the telecommunication system forwarding, terminal system for the carriage of goods, transportation of "door to door", etc.

According to well – known estimates, the cost of transportation of goods ranges from 20 to 70% of the total cost of logistics, while the price of goods transport component has a different share depending on the type of products: 2...3% – for electronics, 5...6% – for food, 7...12% – for machinery and equipment, 40...60% - for raw materials, 80...85% for mineral products. According to experts' calculations, the introduction of information systems can increase the average speed of vehicles by 10 ... 30 km/h.

From the standpoint of the organization of freight transport in General, a promising step is to unite the idea of unmanned transport and cargo convoy. As a result, there are two types of transport systems and infrastructure for them:

- "road train" with a virtual trailer;
- convoy (with virtual connections);
- intelligent logistics center.

2. Prerequisites and means to solve the problem.

Road and climatic conditions of the Arctic zone of the Russian Federation are characterized by the following features:

- abnormally low temperatures (up to -60 ° C), impeding the start of engines and limiting the normal operation of the technical means of autopilots;
- heavy traffic conditions on off-road terrain allowing operation of vehicle only in winter conditions;
- undirected terrain that makes it difficult to visually determine the location of the pilot vehicle on the route;
- snow drifts, which does not allow to recognize a road marking and impeding the functioning of the devices technical vision;
- constancy of reduced visibility in conditions of polar night;
- unpredictable influence of geomagnetic situation and state of the earth troposphere and ionosphere in polar latitudes on the conditions of uninterrupted reception of satellite navigation signals.

The technical solutions of control systems developed in this applied scientific research and experimental development should take into account the above peculiarities of the road and climatic conditions of the Arctic zone of the Russian Federation.

In the course of the work, the following goals were set:

- creation of an intelligent traffic management system column unmanned vehicles with a leading pilot vehicle, improving the efficiency and safety of cargo and passenger traffic in remote regions of the North of the country, the Arctic and Antarctica;
- obtaining significant scientific results, allowing to move to the creation of new types of cargo vehicles in unmanned design, providing a significant increase in freight traffic;
- market launch of new intellectual goods vehicles with a qualitatively new higher technical characteristics;
- ensuring better connectivity of the territory of the Russian Federation through the creation of intelligent transport and telecommunication systems, as well as holding leading positions in the creation of international transport and logistics systems, development and use in the North of the country and the Arctic and Antarctic.

For example, in the North of Russia, the Arctic and the Antarctic, the use of modern cruise control systems is impossible without improvement. Sensors serial systems do not work properly at low temperatures, which can lead to emergency situations.

In snowfall, a large amount of snow sticks to the front bumper of the car and partially closes the front radar view [2]. this, in turn, leads to inaccurate detection of the road situation and increases the risk of collision [3].

The climate of the North of Russia is characterized by strong winds that have a strong impact on the vehicle. This impact may result in unstable acceleration/deceleration of vehicles by the active

cruise control system, which in turn can cause severe discomfort to the driver.

The software part of the existing cruise control systems takes into account the current value of the vehicle speed, the value of the speed of approach with the driving car and the relative distance. Based on these data, the algorithms calculate a target deceleration for the asphalt pavement of the roadway. The presence of snow and ice on the roads increases the braking distance, which in turn reduces the efficiency of the active cruise control system and can lead to a traffic accident [4]. In order to ensure an adequate level of safety, a number of calibrations are needed that will increase the distance between vehicles in pursuit mode. Also changes logic of operation: the system is the cruise control needs to begin braking at a larger distance to the leading vehicle, but with a lower maximum deceleration. This configuration will minimize the impact.

By analyzing the characteristics of the chassis design of an unmanned cargo vehicle, it is possible to draw the following conclusions.

The chassis of the unmanned cargo vehicle intended for operation as a part of a column in the conditions of the Far North, the Arctic and Antarctica shall possess high passability for what should be equipped with the drive of at least 4 wheels.

The chassis of an unmanned cargo vehicle can be equipped with a power plant of any type used in automotive technology, but it is preferable to use the electric drive of the driving wheels without speed transmissions. This conclusion is formulated on the basis of the analysis of world experience, according to which almost all modern unmanned vehicles (both freight and passenger) are equipped with electric drive.

Modern models of unmanned cargo vehicles are based on the technologies of leading manufacturers of cars and automotive components for electronic driver assistance systems (ADAS).

In the conditions of operation of a column of unmanned vehicles with a pilot driving vehicle in the conditions of the Far North, the Arctic and Antarctica there are additional restrictions associated with natural features characteristic of these areas.

One of them should include:

- lack of navigational guidance on a virtually homogeneous desert area;
- polar night periods of up to six months, making it difficult to visually detect obstacles on the lane;
- the lack of paved roads and with recognizable markings;
- increased frequency of failures in satellite navigation due to changes in geomagnetic conditions and other external factors (interference);
- abnormally low temperatures (up to minus 60 ° C and below) and winds with speeds up to 25 m / s and above;
- snow drifts that make it difficult for visual recognition of the lane itself;
- low grip of the tyres when driving on ice, snow and dirt ruts filled with water;
- the increased fuel consumption of vehicle caused by low speeds of the movement on the lowered transfers and practically the round-the-clock mode of operation of internal combustion engine in the conditions of abnormally low temperatures.

In view of the above, the navigation system, including satellite, inertial and wheeled navigation system, is required to provide high-precision positioning solutions in the conditions of loss of radio visibility of satellites, the drift of coordinates of the inertial system and wheel errors. Accuracy can be raised through the use of a weather forecast system, which will give an understanding of the roadway condition and traffic conditions [5].

3. Results and discussion

Mathematical models and control algorithms are the conceptual core of intelligent control systems, largely determines the technical characteristics and basic consumer properties of the final product. One of the obvious and main requirements for mathematical models and algorithms used is their adequacy in the entire operating range of parameter changes, along with the possibility of their use for indirect measurements and for the formation of control actions.

3.1 Description of mathematical models of longitudinal movement of the center of mass of the driven unmanned vehicles

For the development of control algorithms and interaction of intelligent transport control system column below is a description of the mathematical models of longitudinal motion of the center of mass of the guided unmanned . To determine the control actions on the engine, transmission and brake system adequate mathematical model of longitudinal motion of the center of mass of the vehicle.

As a vector of control actions is considered.

(1) $U = (U_1, U_2, U_3)^T$, when
 U_1 – gearbox transmission number $1 \leq U_1 \leq U_{1max}$; $U_1 = 0$ – corresponds to the neutral state of gearbox;

(2) $U_2 = 2\sin^2(\frac{\varphi_{ap}}{2})$ – the control action of the accelerator corresponding to the rotation angle of the throttle φ_{ap} engine;

(3) $U_3 = P_T P_{Tmax}^{-1}$ – control action on a braking system equal to the ratio of pressure P_T to its maximum allowable value P_{Tmax} .

The longitudinal motion of the center of mass of the car is described by a system of differential equations, in which the first equation of the system is a special case of the second Newton's law, and the second equation is the definition of speed as a derivative of the path [6]:

$$(4) \begin{cases} \dot{V}_m = a_m = a_{dT}(U) - k_x m_0^{-1} V_m^2 - k_{rp} g - \alpha_T g \\ \dot{L}_m = V_m \end{cases}$$

when

$a_m = \dot{V}_m$ – longitudinal acceleration of the centre of mass;

(5) $a_{dT} = a_d(U_1, U_2) - a_T(U_3)$ – traction and acceleration created by the engine to the transmission to the drive wheels (a_d) and brake deceleration (a_T), generated by the braking system;

(6) $k_x = 0,5 C_x S_x \rho$ – drag coefficient, aerodynamic drag;

C_x – specific coefficient of aerodynamic drag;

S_x – cross-sectional area (midsection) of the object;

m_0 – vehicle weight;

k_{rp} – the coefficient of rolling friction of the tires;

α_T – pitch angle equal to the angle of slope of the road surface to the horizon;

g – acceleration of gravity.

Drag coefficient, aerodynamic drag k_x and rolling friction of tires k_{rp} unknown in advance and determined experimentally.

3.2. Development of the algorithm of brake control

The problem of development of the brake control algorithm is solved for the adaptive cruise control system, advanced emergency braking system and auxiliary braking system.

The principal differences of the considered algorithm of automatic braking of the developed systems of power-driven vehicles from those used in existing foreign systems of active safety are:

- installed on modern vehicles, automatic emergency braking (AEBS) systems are used in braking ABS function, therefore, they have all the disadvantages of anti-lock systems, namely the cyclic principle of action, which does not allow for effective braking on slippery and uneven surfaces. The algorithm does not depend on ABS and works on the principle of limiting the braking deceleration, taking into account the slippery surface;
- preventing collisions with passing objects in the rear hemisphere is not part of the functions of the currently used systems. However, the danger of such collisions and the severity of their consequences is no less significant than in the front hemisphere.

To describe the algorithm of automatic braking should describe the conditions of collisions of objects when moving in the same lane.

Consider the differential equations of distances L_1 and L_2 between the driven car and the obstacles in the front and rear traffic lights on the same lane:

$$(7) \begin{cases} \frac{dL_1}{dt} = V_{np1}(t) - V_m(t); \\ \frac{dL_2}{dt} = V_m(t) - V_{np2}(t), \text{ when} \end{cases}$$

$V_m(t)$ – speed of driven car,

$V_{np1}(t)$ and $V_{np2}(t)$ – speed front and rear obstacles.

Figure 1 shows the relative position of the front and rear obstacles on the lane..

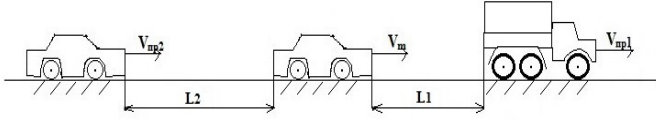


Figure 1-Relative positioning of the front and rear passing obstacles on the lane

The solution of differential equations (8), presented on the time interval $(t \div t_s)$ to the complete stop of objects:

$$(8) \begin{cases} L_1(t_s) = L_1(t) + \int_t^{t_s} [V_{np1}(\tau) - V_m(\tau)] d\tau; \\ L_2(t_s) = L_2(t) + \int_t^{t_s} [V_m(\tau) - V_{np2}(\tau)] d\tau. \end{cases}$$

A natural condition for preventing collisions is the implementation of inequalities $L_1(t_s) > 0$ and $L_2(t_s) > 0$, which are converted to the following:

$$(9) \begin{cases} L_1(t) > \int_t^{t_s} [V_m(\tau) - V_{np1}(\tau)] d\tau; \\ L_2(t) > \int_t^{t_s} [V_{np2}(\tau) - V_m(\tau)] d\tau \end{cases}$$

The right-hand sides of the inequalities (9) represent the lower bounds of safe distances ΔL_{rp1} и ΔL_{rp2} to front and rear hurdles.

Given the time lag of actuation of the brake systems of the vehicle τ_m and rear obstacles τ_{np2} the lower bounds of the safe range ΔL_{rp11} and ΔL_{rp21} at intervals, delays are:

$$(10) \begin{cases} \Delta L_{rp11} = 0,5\hat{a}_{np1}\tau_m^2 + (V_m - V_{np1})\tau_m; \\ \Delta L_{rp21} = 0,5\hat{a}_{tm}\tau_{np2}^2 + (V_{np2} - V_m)\tau_{np2}, \text{ when} \end{cases}$$

$\hat{a}_{np1}$ – projected estimation of the deceleration of the front obstacle;

$\hat{a}_{tm}$ – forecast estimate of deceleration of the driven vehicle.

The second pair of lower boundaries of safe distances ΔL_{rp1} and ΔL_{rp2} determined at the time interval until the complete stop:

$$(11) \begin{cases} \Delta L_{rp12} = V_m\tau_m + 0,5V_m^2\hat{a}_{tm}^{-1} - 0,5V_{np1}^2\hat{a}_{np1}^{-1}; \\ \Delta L_{rp22} = V_{np2}\tau_{np2} + 0,5V_{np2}^2\hat{a}_{np2}^{-1} - 0,5V_m^2\hat{a}_{tm}^{-1}. \end{cases}$$

The resulting bounds of the safe range ΔL_{rp1} and ΔL_{rp2} defined as the maximum of a pair $(\Delta L_{rp11}, \Delta L_{rp12})$ and a pair of $(\Delta L_{rp21}, \Delta L_{rp22})$:

$$(12) \begin{cases} \Delta L_{rp1} = \max[\Delta L_{rp11}, \Delta L_{rp12}] \\ \Delta L_{rp2} = \max[\Delta L_{rp21}, \Delta L_{rp22}] \end{cases}$$

In this case, the inequalities are fulfilled

$$(13) \begin{cases} L_1(t) > \Delta L_{rp1}(t); \\ L_2(t) > \Delta L_{rp2}(t), \end{cases}$$

means that the collision avoidance conditions are met both on the lag interval and on the interval to a full stop.

To prevent collisions with an oncoming obstacle $V_{np1} < 0$ the value of the boundary distance is

$$(14) \Delta L_{rp1} = V_m\tau_m + V_{np1}\tau_{np1} + 0,5V_m^2\hat{a}_{tm}^{-1} + 0,5V_{np1}^2\hat{a}_{np1}^{-1}$$

Analysis of the conditions for collision prevention, presented in the form of the problem of dynamic stabilization of distances, shows that the highest values of boundary distances is characteristic for the oncoming obstacles [7]. Uncertainty of estimates of delays in braking of the driven vehicle and obstacles, along with uncertainty of their decelerations in case of manual control creates difficulties of adequate forecasting of boundary distances.

The solution of the problem of dynamic stabilization of safe distances [8] or deceleration a_{tm} the driven vehicle is determined from the boundary distance equations provided

$$(15) L_1 = \Delta L_{rp1} \text{ и } L_2 = \Delta L_{rp2}.$$

So, in particular, brake deceleration a_{tm} , sufficient to prevent collision with a forward fixed, passing or oncoming obstacle is

equal to:

$$(16) a_{tm} = \begin{cases} \bar{a}_{tm}, \text{ если } \Delta L_{m1}^* > 0 \text{ и } \bar{a}_{tm} \leq a_{tmax}; \\ a_{tmax}, \text{ если } \Delta L_{m1}^* \geq 0 \text{ или } \bar{a}_{tm} > a_{tmax}, \text{ when} \end{cases}$$

$$(17) \bar{a}_{tm} = 0,5V_m^2(\Delta L_{m1}^*)^{-1};$$

$$(18) \Delta L_{m1}^* = \begin{cases} [L_1 - V_m\tau_m + 0,5V_{np1}^2\hat{a}_{np1}^{-1}], \text{ если } V_{np1} \geq 0; \\ [L_1 - V_m\tau_m + V_{np1}\tau_{np1} - 0,5V_{np1}^2\hat{a}_{np1}^{-1}], \\ \text{ if } V_{np1} < 0; \end{cases}$$

$\hat{a}_{np1}$ – assessment of the developed brake deceleration of the front obstacle;

Braking deceleration a_{t1} , sufficient to prevent a collision with the rear of the associated constraint on the delay interval braking τ_{np2} equally:

$$(19) a_{t1} = \begin{cases} \bar{a}_{t1}, \text{ если } \bar{a}_{t1} \leq a_{tmax}; \\ a_{tmax}, \text{ если } \bar{a}_{t1} > a_{tmax}, \end{cases}$$

when $\bar{a}_{t1} = 2[L_2\tau_{np2}^{-2} + (V_m - V_{np2})\tau_{np2}^{-1}]$.

3.3. Development of algorithm of interaction of CU in the column

After switching to the operating mode, following in the column the leading vehicle, the leading vehicle, on which the Bluetooth access server devices are located, begins the exchange with the driven unmanned vehicles equipped with Bluetooth client modules [9].

Performing the functions of a server (hereinafter server) device with Bluetooth access, transmits to customers (driven by unmanned vehicle) information, on the basis of which the task of managing the course of driven unmanned vehicles is formed.

Messages are exchanged between the server and clients in accordance with the Protocol of exchange that guarantees the delivery of the message.

The exchange session always initializes the server, to verify communication, the server, with the operator specified frequency, sends a command to all clients to send telemetry, the clients, if they have a new regular or emergency telemetry, sends it to the server.

According to the results of telemetry processing, the control program of the column or the operator sends to the selected unmanned vehicle control command to perform certain actions specified in paragraph 3.1.

Driven unmanned vehicles transmit to the server telemetric information necessary for trouble-free movement as part of the column.

In the case of identifying obstacles on the lane column in front of one of the unmanned vehicles, the unmanned vehicle, in the presence of (according to the engineer reconnaissance), the second lane begins the maneuver detour obstacles, with a notification over the communication channel leading unmanned vehicle otherwise the unmanned vehicle is stopped and sends a notification about an emergency stop [10,11].

In the event that one of the unmanned vehicles makes an emergency stop associated with the diagnosed malfunctions, the emergency unmanned vehicle transmits to the driving car a message about the emergency stop.

3. Conclusion

The analysis of results of the carried-out researches in the field of perspective directions of development of systems of functioning of the unmanned vehicle allows to formulate the following conclusions: the principles of construction of multilevel information management systems of the unmanned vehicle and their hierarchy are defined; the review of technical characteristics of available devices and systems of technical vision is carried out; the comparative analysis of opportunities and technical characteristics of systems of navigation and orientation of the unmanned vehicle; modern intelligent communication systems of unmanned vehicles and their technical characteristics are considered..

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AIR COOLED DIRECT INJECTION DIESEL ENGINE MAIN OPERATING PARAMETERS ANALYSIS DURING THE CHANGE IN ROTATIONAL SPEED

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Abstract: Four-stroke air cooled, direct injection diesel engine Torpedo BT4L912 during rotational speed variation was investigated in this paper. By using the measurement results obtained at engine brake was calculated several main operating parameters in each engine rotational speed. Rotational speed was varied from 1000 rpm to 2300 rpm. Calculated engine operating parameters are engine torque, effective power, engine brake mean effective pressure, specific effective fuel consumption and volume fuel consumption per engine process. The highest obtained engine torque amounts 338.8 Nm and the highest engine effective power amounts 71.76 kW. Specific effective fuel consumption has the lowest value of 197.42 g/kWh at engine rotational speed of 1800 rpm. Regarding several calculated engine operating parameters, optimal operating point of the analyzed engine is at 1800 rpm.

Keywords: DIESEL ENGINE, DIRECT INJECTION, ENGINE BRAKING, ENGINE OPERATING PARAMETERS

1. Introduction

Experimental measurements are the basis for internal combustion engine operating parameters analysis, [1] and [2]. Parallel to internal combustion engine measurements, numerical simulations have been developed to ensure easier, faster and much cheaper investigations of engine operating parameters, regardless if it was investigated gasoline or diesel engines [3].

If in the focus was diesel engines, as in this paper, it should be noted that today was known several types of diesel engine numerical models: 0D (zero dimensional) models [4], multizone models [5], quasi dimensional models [6] and [7], while the last and most detailed ones are CFD (Computational Fluid Dynamics) models [8]. In order to determine the accuracy and precision of each numerical model, they must necessarily be validated in several different measurement points of the tested engine. Because of that fact, experimental engine measurements are inevitable.

To reduce diesel engine emissions and improve engine operating parameters, researchers are intensively involved in implementing combustion of different alternative fuels in existing diesel engines. A complete review of green fuels as alternative fuels for diesel engines is presented in [9] while the review of performance, combustion and emission characteristics of bio-diesel fuelled diesel engines presented authors in [10].

In this paper was presented change in the main operating parameters of four-stroke air cooled, direct injection diesel engine Torpedo BT4L912 during rotational speed variation. Based on the measurement results obtained at engine brake was calculated several main operating parameters and presented for a various engine rotational speed. Those operating parameters were engine torque, engine effective power, engine brake mean effective pressure, specific effective fuel consumption and volume fuel consumption per engine process. The change of last two operating parameters was calculated by using data for fuel density and fuel tank volume, along with the time for the fuel consumption from the fuel tank. For the wide range of engine rotational speeds was obtained recommended operating area of the analyzed engine.

2. Air cooled, four-stroke, direct injection diesel engine specifications

The investigated engine was four-stroke air cooled, direct injection diesel engine Torpedo BT4L912. The main engine characteristics along with necessary brake and fuel specifications are presented in Table 1.

Table 1. Engine and brake specifications

Number of cylinders	4
Cylinder diameter	102 mm
Cylinder stroke	120 mm
The total operating volume	3922 cm ³
Cylinder cooling	With air
Brake arm length	0.714 m
Fuel tank volume	100 cm ³
Fuel density	830 kg/m ³
Fuel lower heating value	42700 kJ/kg

3. Engine measurement results and measuring equipment

Engine measurement was performed on the engine test bench in the company Torpedo, Rijeka, Croatia. Used measuring equipment in the majority is owned by the same company, Fig. 1.

Brake force was read directly on the brake Schenk U1-30. The engine rotational speed was measured by inductive encoder on the brake shaft. Volumetric fuel consumption was measured by using a photocell.

Cylinder pressure was measured with a data acquisition device (analogue-digital system). An analogue signal was amplified through the amplifier and converts to digital signal, which can be further processed. The data acquisition device has a microprocessor MC 68020 (32 Bit and 16.7 MHz) along with co-processor MC 68881 (16.7 MHz). In this device are included two analogue-digital converters (resolution of 12 Bit) with maximum of $2 \cdot 4 \cdot 10^5$ measurements in second (five measurements for one engine crank angle). Cylinder pressure was measured in the first engine cylinder with quartz sensor Kistler - Type 7061 which measurement range is from 0 to 200 bars.

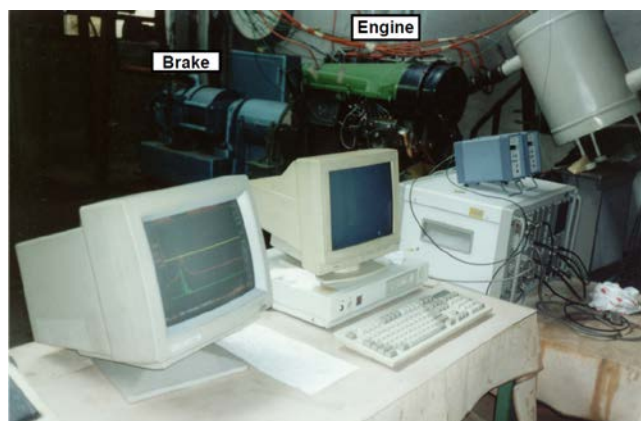


Fig. 1. Diesel engine Torpedo BT4L912 connected to brake during the measurements

Engine measurement was carried out in a way that after engine start, maximum rotational speed has been achieved without any load. After that the braking force gradually increased and the engine rotational speed decreased. For the engine analysis provided in this paper, necessary measured operating parameters are presented in Table 2 and those are: engine rotational speed, brake force and time for the fuel consumption from the fuel tank.

Table 2. Torpedo BT4L912 obtained measurement results

Test No.	Engine rotational speed (rpm)	Brake force (N)	Time for the fuel consumption from the fuel tank (s)
1	2300	406.98	17.9
2	2200	436.40	17.8
3	2000	453.07	19.1
4	1800	474.64	23.7
5	1600	460.91	23.2
6	1500	460.91	24.5
7	1400	457.97	25.9
8	1200	437.38	29.6
9	1000	388.34	35.7

4. Equations for calculating engine main operating parameters according to measured results

At each engine rotational speed torque was calculated according to the equation:

$$M = F \cdot R \quad (1)$$

where M (Nm) is torque, F (N) is brake reaction force and R (m) is the brake arm's length on which the reaction force is measured - Table 1.

Engine effective power delivered to power consumers was calculated by using an equation:

$$P_{ef} = \frac{M \cdot 2 \cdot \pi \cdot n}{60 \cdot 1000} \quad (2)$$

where P_{ef} (kW) is engine effective power and n (rpm) is engine rotational speed.

Engine brake mean effective pressure was calculated according to the equation:

$$p_{ef} = \frac{P_{ef}}{n_a \cdot V_{tot,op}} \cdot 6 \cdot 10^5 \quad (3)$$

where p_{ef} (bar) is an engine brake mean effective pressure and $V_{tot,op}$ (cm³) is the total engine operating volume - Table 1. Engine active rotational speed n_a (rpm) which takes into account only the engine processes with fuel injections is equal to:

$$n_a = \frac{2 \cdot n}{\tau} \quad (4)$$

where τ is engine stroke (analyzed engine is a four-stroke engine).

Specific effective fuel consumption was calculated by using an equation:

$$b_{ef} = \frac{V_{ft} \cdot \rho_f}{t_{ft} \cdot P_{ef}} \cdot 3.6 \quad (5)$$

where b_{ef} (g/kWh) is specific effective fuel consumption. V_{ft} (cm³) is a fuel tank volume - Table 1, ρ_f (kg/m³) is fuel density - Table 1 and t_{ft} (s) is time for the fuel consumption from the fuel tank - Table 2.

Volume fuel consumption per engine process was obtained by using an equation:

$$V_{pp} = \frac{V_{ft}}{n_a \cdot t_{ft} \cdot z} \cdot 6 \cdot 10^4 \quad (6)$$

where V_{pp} (mm³/proc.) is volume fuel consumption per engine process and z is the number of engine cylinders - Table 1.

5. Change in engine calculated main operating parameters with discussion

For each measurement point, at each engine rotational speed, engine torque was calculated by using equation (1). At the maximum engine rotational speed of 2300 rpm, engine torque amounts 290.5 Nm, Fig. 2. With the decrease in the engine rotational speed from maximum value, engine torque firstly continuously increases. The highest value of engine torque was obtained at 1800 rpm and amounts 338.8 Nm. During further decrease in the engine rotational speed from 1800 rpm, engine torque continuously decreases and at the lowest rotational speed of 1000 rpm engine torque has the lowest value of 277.2 Nm. As analyzed engine is air cooled type, available torque range from 277.2 Nm to 338.8 Nm for observed rotational speeds is satisfactory.

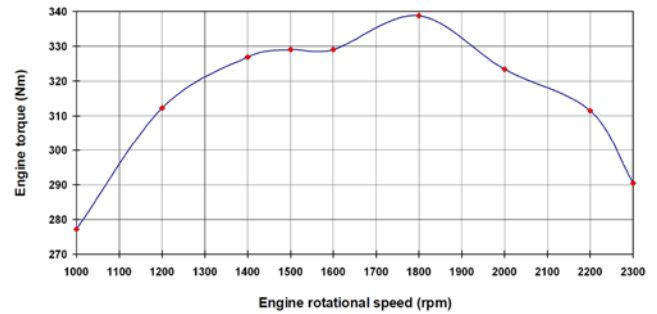


Fig. 2. Engine torque change in relation to engine rotational speed

Engine effective power which is delivered by the outlet coupling to the power consumers was calculated for each measured engine rotational speed according to equation (2). At the lowest engine rotational speed of 1000 rpm, engine effective power is the lowest and amounts 29.03 kW, Fig. 3. With the increase in the engine rotational speed from the lowest one, engine effective power continuously increases up to 2200 rpm. At 2200 rpm, effective power has the highest value which amounts 71.76 kW. In the range from 2200 rpm to 2300 rpm, engine effective power decreases and at the highest rotational speed (2300 rpm) effective power is equal to 69.97 kW.

If compared the change of analyzed engine torque, Fig. 2, with change of engine brake mean effective pressure, Fig. 4, it can be seen that both parameters have identical trends. Brake mean effective pressure was calculated by using equations (3) and (4). At the lowest engine rotational speed of 1000 rpm is observed the lowest brake mean effective pressure which amounts 8.88 bars.

With the increase in the engine rotational speed from the lowest one, brake mean effective pressure continuously increases and reaches the maximum value of 10.85 bars at the 1800 rpm. From the engine rotational speed of 1800 rpm up to 2300 rpm, brake mean effective pressure continuously decreases and on the highest engine rotational speed of 2300 rpm it amounts 9.31 bars.

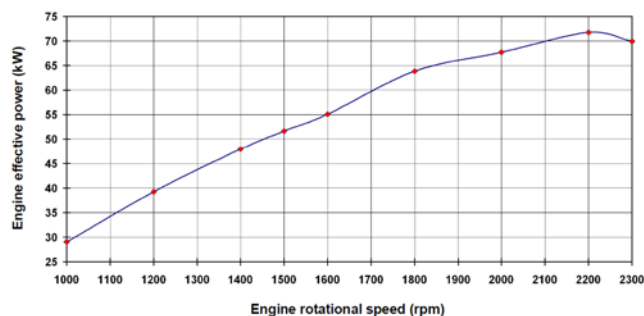


Fig. 3. Engine effective power change for various observed engine rotational speeds

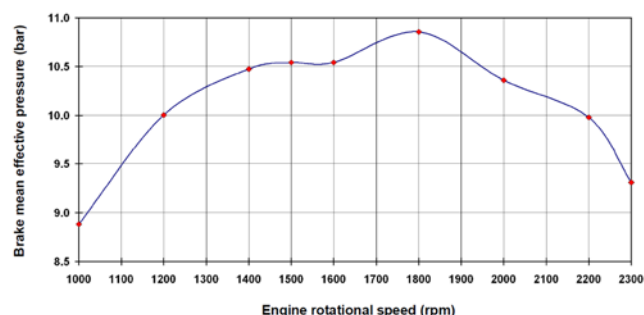


Fig. 4. Change in brake mean effective pressure for various engine rotational speeds

Specific effective fuel consumption was calculated by using measured values for each observed engine rotational speed, according to equation (5). Trend of specific effective fuel consumption of the analyzed engine is similar to diesel engines investigated by the other authors, [11] and [12]. With the increase in the engine rotational speed, specific effective fuel consumption firstly decreases to the lowest value, after which follows its increase. In Fig. 5 is not presented trendline for specific effective fuel consumption change, calculated values for each engine rotational speed were connected with a line in order not to approximate the real data.

At the lowest engine rotational speed (1000 rpm) specific effective fuel consumption is the highest and amounts 288.33 g/kWh. With the increase in the engine rotational speed from the lowest one, specific effective fuel consumption decreases until 1800 rpm. At 1800 rpm specific effective fuel consumption has the lowest value which amounts 197.42 g/kWh. From 1800 rpm up to the highest engine rotational speed of 2300 rpm, specific effective fuel consumption continuously increases and on 2300 rpm it has a value equal to 238.58 g/kWh.

Volume fuel consumption per engine process (VFCPP) was calculated for each measured engine rotational speed, according to equation (6). At the lowest engine rotational speed of 1000 rpm, VFCPP amounts 84.03 mm³/proc, Fig. 6. Increase in the engine rotational speed from 1000 rpm causes that VFCPP firstly slightly increase at 84.46 mm³/proc. (1200 rpm) after which follows the continuous decrease up to 1800 rpm. At the 1800 rpm, VFCPP has the lowest value which amounts 70.32 mm³/proc. From 1800 rpm to 2000 rpm VFCPP increases and amounts 78.53 mm³/proc. at 2000 rpm. Finally, from 2000 rpm until the highest engine rotational speed, volume fuel consumption per engine process continuously decreases and amounts 72.87 mm³/proc. at the 2300 rpm.

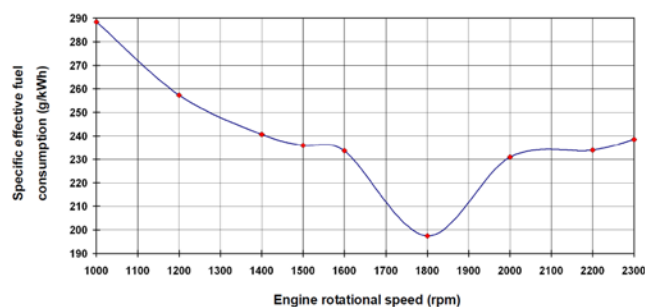


Fig. 5. Specific effective fuel consumption change for various engine rotational speeds

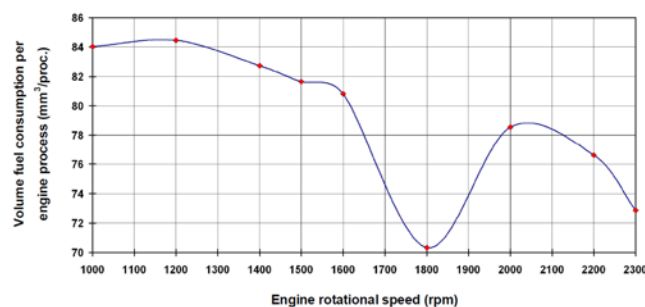


Fig. 6. Volume fuel consumption per engine process change for various observed engine rotational speeds

6. Conclusion

In this paper were analyzed changes in the main operating parameters of four-stroke air cooled, direct injection diesel engine Torpedo BT4L912 during rotational speed variation. Several main operating parameters for each measured engine rotational speed was calculated in order to obtain complete insight into the engine operating characteristics.

The highest obtained engine torque amounts 338.8 Nm and the highest engine effective power amounts 71.76 kW. Highest torque and highest effective power were not obtained at the same engine rotational speed. Engine brake mean effective pressure was the same trend as engine torque. The highest value of engine brake mean effective pressure amounts 10.85 bars.

The lowest specific effective fuel consumption was obtained at 1800 rpm and amounts 197.42 g/kWh. At the same engine rotational speed was also obtained the lowest volume fuel consumption per engine process which amounts 70.32 mm³/proc.

The final conclusion, which can be derived from the presented calculated results, is that the optimal operating point of the analyzed engine Torpedo BT4L912 is operating point at 1800 rpm. In that operating point engine has the highest torque and the lowest specific effective fuel consumption. Also, in that operating point analyzed engine has the lowest volume fuel consumption per process.

7. Acknowledgments

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INVESTIGATION OF GRAIN BIOMASS PROPERTIES AS AN ALTERNATIVE FUEL

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Abstract: In connection with the observed dynamism of changes on the energy market regarding generation of energy from sources other than coal, more attention was paid to the use of "clean", low-emission technologies. Bearing in mind the need to search new, alternative energy sources, the aim of the study was to investigate and analyze the properties of grain biomass, such as rice and corn, as a fuel. To achieve the goal, the research problem has been formulated: how the fragmentation degree and the type of biomass affects such energy properties as heating value and heat of combustion. To resolve the problem, the properties of white rice, black rice, red rice and corn before and after grinding were analysed. The results show that white rice has the biggest value of heat of combustion before grinding, red rice – the lowest. White rice has the biggest heating value for whole grain, red rice – the lowest. The research allows to state that grinding operations result in increasing energetic properties of biomass. It can be also assumed that biomass is a good substitute of fossil fuel.

Keywords: RICE, CORN, HEATING VALUE, HEAT OF COMBUSTION, ALTERNATIVE FUEL

1. Introduction

Biomass has been used by humankind as a source of heat since immemorial time. Many different definitions of the concept of biomass can be found in the literature. According to the Ordinance of the Minister of Economy and Labour dated 9 December 2004, "biomass" refers to solid or liquid biodegradable substances of plant or animal origin, obtained from products, waste and residues from agricultural and forestry production and from industries processing waste, as well as fractions of other biodegradable waste [1].

The current trend to use biomass to replace coal is mainly related to the EU goals concerning CO₂ emission reduction [2], [3]. As compared to burning coal, emissions of harmful substances, including CO₂, are much lower when burning biomass, as it's shown in table 1 [4], [5].

Table 1. Calorific values and emissions of harmful gases for coal and biomass [4]

Pollutants emitted to the environment	Fuel / heating value [MJ/kg]			
	Coal/25	Straw/17	Wood/19	Waste/10-40
Carbon dioxide	100	-	-	128
Nitrogen oxides	0.3-0.4	0.16	0.16	0.55
Sulphur oxides	0.5-10	0.07	-	1
Dusts	0.05	0.20	0.20	0.11

Biomass used as a source of energy can be in many different forms: waste wood, straw, energy plant plantation crops, organic waste (manure, sewage sediments, waste from sugar refineries, distilleries, etc.), as well as liquid biofuels or biogas. Of the biomass types listed above, straw and energy plants can be most suitable for heat and power generating plants [6], [7].

Energy can be obtained from both, processed and unprocessed biomass. Conversion can occur in the process of direct combustion, gasification or processing into liquid fuels. The simplest and cheapest way to obtain energy from biomass is direct burning in special boilers, co-burning with such conventional energy sources as coal or heating oil and burning processed biomass products: methanol, ethanol, biogas or biodiesel [8] - [10].

The world biomass supply is estimated to deliver ca. 44 EJ of energy per year, which accounts for ca. 10% of the global energy consumption. The available supply that can be used reaches 276 EJ/year [11]. Table 2 shows the technical potential of biomass in selected European countries.

Table 2. The technical potential of biomass in selected Central European countries [11]

Central Europe	Technical potential of biomass [PJ/year]
Austria	368
Czech	299
Germany	1200-1700
Hungary	100-190
Italy	1000-1200
Poland	927
Slovakia	40-90
Slovenia	20-53

Energy can also be obtained from grain, e.g. from such commonly cultivated crops as corn, rice, wheat and other cereals that cannot be processed into food or feed due to bad quality. Many grains - whole or ground - remaining as waste from production processes, could be successfully used as a valuable fuel material or as an input to a biogas production plant.

The aim of the study was to investigate and analyze the properties of grain biomass, such as rice and corn, as fuel. To achieve the goal, the research problem was formulated as a question: how the fragmentation degree and the type of biomass affects such energy properties as heating value (W) and heat of combustion (Q_w). To resolve the problem, the following properties were investigated: moisture content, particle size and shape, heat of combustion and heating value of white rice, black rice, red rice and corn before and after grinding.

2. Materials and methods

2.1. Analyzed grains

The study involved samples of four types of cereals: corn, white rice, red rice and black rice. Table 3 shows a list of cereals tested and their characteristics.

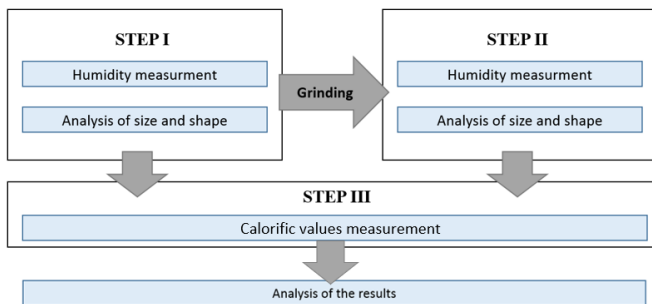
Rice and corn are considered to be plants with the largest acreage globally. Corn is usually ground for animal feed, for food industry and also for energy purposes in biogas plants. Corn is very commonly used as heating fuel in the USA and Canada. Special corn-burning stoves have been designed for this purpose there. Shredded rice is used to make products such as rice noodles. It can also be a valuable energy source in combustion processes. Rice waste is most commonly used in the form of briquettes. The fact that rice and corn are used for energy generation should not cause any deficit of these materials in the food processing industry, considering the huge volume of global production of these crops [12] - [16].

Table 3. Characteristics of tested materials [own work]

CHARACTERISTICS OF TESTED MATERIALS				
	Red rice	White rice	Black rice	Corn
Grains				
Application	Food industry	Food industry, Energy purposes ex. briquette production	Food industry	Animal feed, food industry, biogas plants, solid fuel

2.2. Research plan

Tests were performed for each sample in accordance with the plan shown in Figure 1.

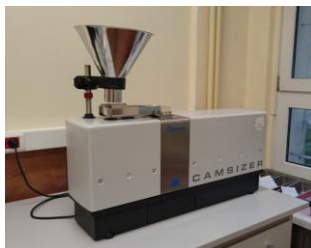
**Fig. 1.** Research plan of particles size and shape of biomass [own work]

The research was divided into three parts. The first step was humidity measurement and whole grains size analysis. The samples were then ground. In the next step, an analysis of ground material moisture content was performed, followed by the analysis of particle size using the CAMSIZER. The third step was energy content examination and finally - the results obtained were analysed.

2.2. Methods

Samples of grain were ground in the five disc mill located in the Laboratory of Comminution Research at the Faculty of Mechanical Engineering UTP in Bydgoszcz.

Before and after grinding, particle size analysis was performed using Retch CAMSIZER analyser. The analyser takes photographs of the falling particles and enables the shape and size of fractions to be determined, as well as their percentage share in the sample.

**Fig. 1.** CAMSIZER analyser [own materials]

The particle size analysis results were used as a basis for determining the degree of grain fragmentation. It can be determined using a number of methods, e.g. the maximum degree of fragmentation i_g :

$$i_g = \frac{D_{\max}}{d_{\max}} \quad (1)$$

$D_{\max}$ - arithmetic mean of diameters of the largest grains fed to the grinder, $d_{\max}$ - arithmetic mean of diameters of the largest grains in the product of grinding.

The maximum degree of fragmentation is biased due to the fact that the grains being fed into the grinder and the grinding product are not ideally spherical. The degree of fragmentation can also be

determined from the dependence at the 80% degree of fragmentation (2) based on the knowledge of the sample particles size distribution. In this study, an 80% degree of fragmentation is used for biomass tests.

$$i_{80} = \frac{D_{80}}{d_{80}} \quad (2)$$

D_{80} - the sieve mesh opening size allowing 80% of the grains though, d_{80} - the sieve mesh opening size allowing 80% of the grinding product through.

As the next step, the heat of combustion was determined for corn and rice samples in a calorimetric bomb (Fig. 3), in oxygen at 25°C, according to the PN-81/G-04513 standard. The water equivalent of calorimeter was determined according to the PN-71/G-04062 standard, using 99.5% benzoic acid C_6H_5COOH . The sample combustion heat is: $Q = 26417$ kJ/kg.

**Fig. 2.** The workstation used for determining the heat of combustion [own materials]

3. Research results

The first step of the research was humidity measurement. Table 4 presents the results of humidity analysis. The results show, that white rice was characterized by the highest humidity (before and after grinding). Black rice had the lowest humidity before and after grinding. Humidity of tested materials increased after grinding.

Table 4. Results of analysis of humidity of tested biomass grains [own study]

HUMIDITY OF THE SAMPLES TESTED [%M]				
Name Form	Red rice	White rice	Black rice	Corn
Grains	12,630	13,600	8,723	9,378
Shredded	13,176	13,896	11,527	12,264

Four samples with different degrees of fragmentation were selected for each biomass category. Table 5 shows the results of the particle size analysis for the three rice types and corn grains. Based on the equivalent grain diameters (D_{80} i d_{80}), fragmentation degree i_{80} was determined for each of the samples (table 5).

Table 5. Results of particles size and fragmentation degree [own study]

		Sample no			
Grain type	Characteristics	1	2	3	4
Black Rice	d_{80} [mm]	1.10	1.35	1.30	1.20
	D_{80} [mm]	2.08	2.08	2.08	2.08
	I_{80}	1.89	1.54	1.60	1.73
Red rice	d_{80} [mm]	1.10	1.35	1.30	1.20
	d_{80} [mm]	2.08	2.08	2.08	2.08
	I_{80}	1.89	1.54	1.60	1.73
White rice	d_{80} [mm]	1.20	0.75	0.80	1.30
	D_{80} [mm]	2.18	2.18	2.18	2.18
	I_{80}	1.82	2.91	2.73	1.68
Corn	d_{80} [mm]	2.35	1.65	2.00	2.55
	D_{80} [mm]	7.85	7.85	7.85	7.85
	I_{80}	3.34	4.76	3.93	3.08

With the fragmentation degree determined, grain calorific values were analysed. In the case of whole grains, white rice shows both the highest heating value (16401 kJ/kg) and the highest combustion heat (17575 kJ/kg), while the lowest values were recorded for red rice ($W=15972$ kJ/kg, $Q_w=17142$ kJ/kg) (Fig. 4). The results of tests for whole grains of corn or rice are consistent with the values presented in other studies [17]-[21].

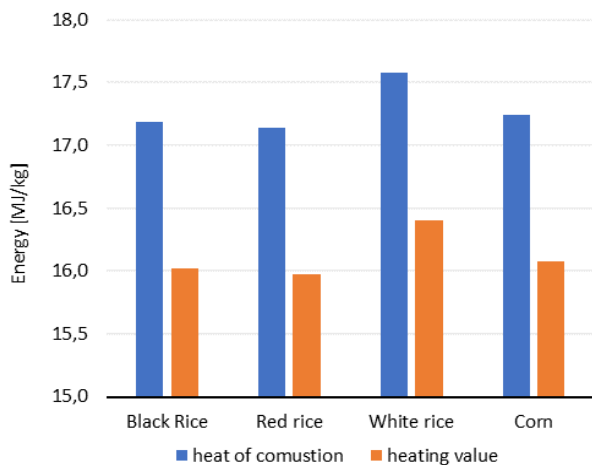


Fig. 4. Combustion heat and heating value of grains before grinding [own study]

After grinding, both the heating value and combustion heat of the biomass samples increased as compared with the values recorded for whole grains (Fig. 5-6). The highest increase of the heating value and combustion heat was recorded for black rice (by 4.5% and 4.3% respectively), the lowest – for white rice (by 0.3% and 0.3%).

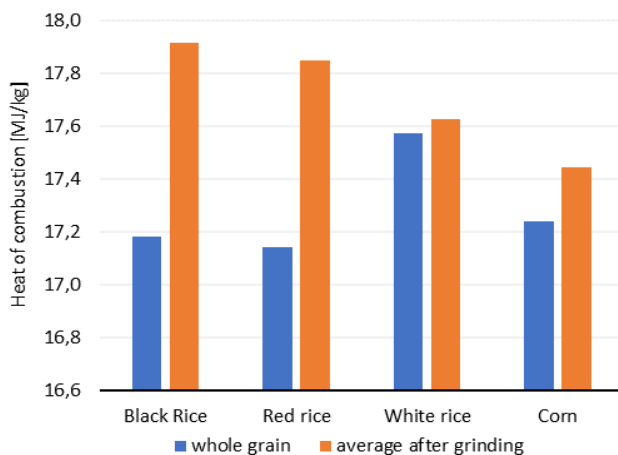


Fig. 5. Combustion heat of samples, before and after grinding [own study]

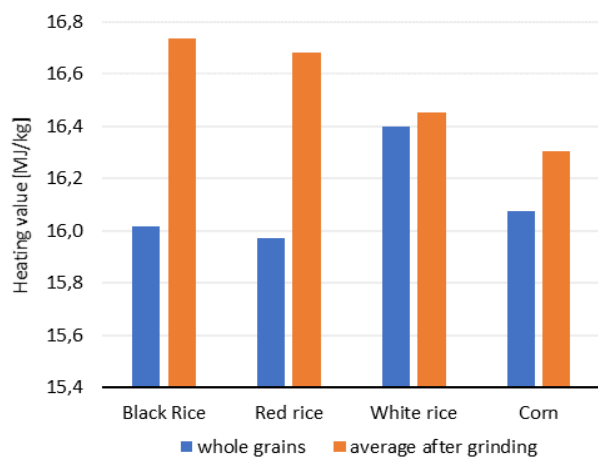


Fig. 6. Heating value of the samples before and after grinding [own study]

Figures 7-10 illustrate heating values of the samples tested, depending on the fragmentation degree. The best results of heating values after grinding were obtained for black rice (17154 kJ/kg) with the fragmentation degree (1.23), the worst for corn (15836.5 kJ/kg) for fragmentation degree (3.08). The highest combustion heat (18321 kJ/kg) was recorded for black rice with the fragmentation degree (1.23), while the lowest – for corn (16923 kJ/kg) with the fragmentation degree (3.08).

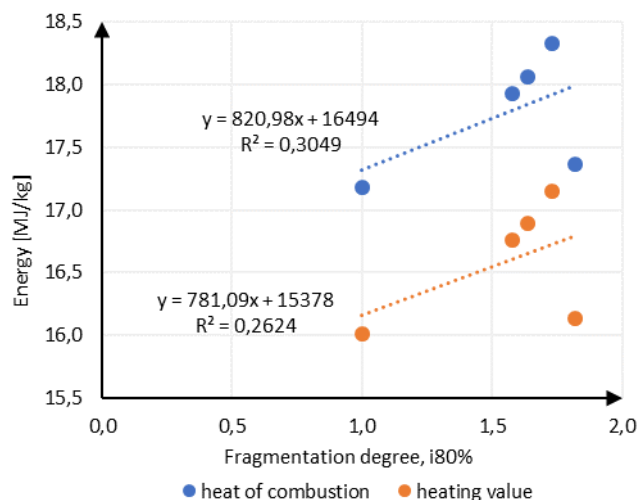


Fig. 7. Combustion heat and heating value of black rice, depending on the fragmentation degree [own study]

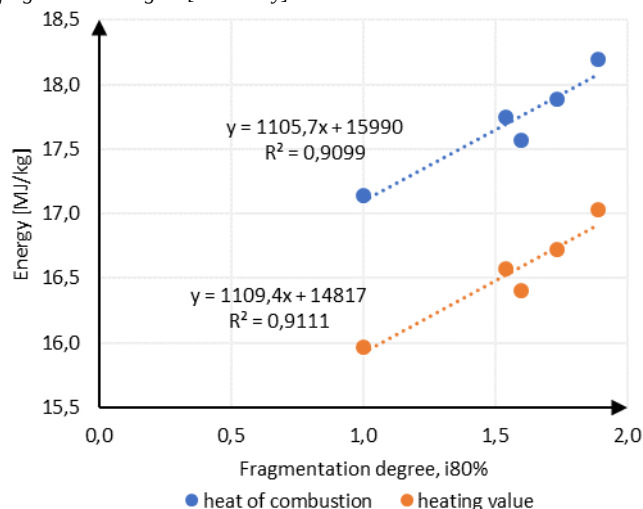


Fig. 8. Combustion heat and heating value of red rice, depending on the fragmentation degree [own study]

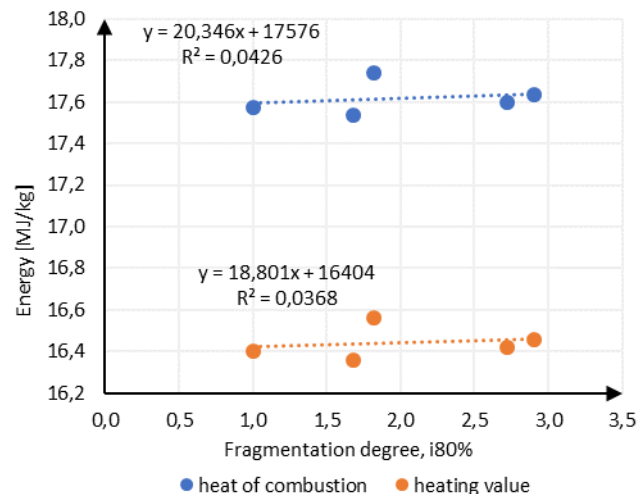


Fig. 9. Combustion heat and heating value of white rice, depending on the fragmentation degree [own study]

The results obtained do not allow of the unambiguous determination of the heating value and the heat of combustion. Yet, both the heat of combustion and the heating value can be observed to grow together with the degree of fragmentation. In a general case, the statement that grinding causes energy obtained in the process of grain burning to grow can be considered as true. Similar results can be found in studies [22] - [24].

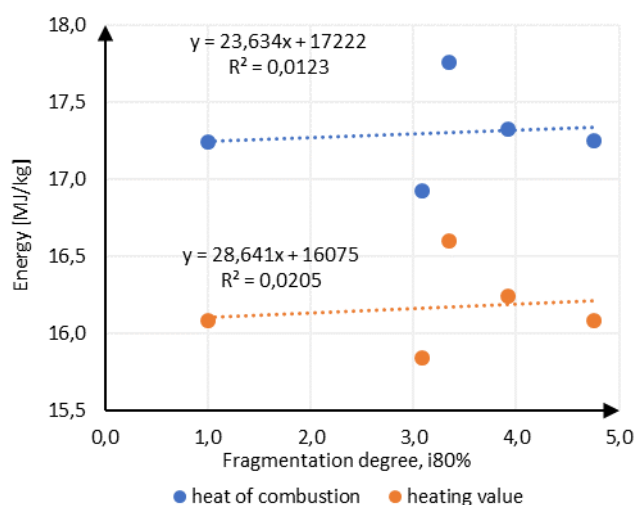


Fig. 10. Combustion heat and heating value of corn, depending on the fragmentation degree [own study]

4. Summary and conclusion

The results show white rice has the biggest value of heat of combustion before grinding, red rice - the lowest. White rice shows the biggest heating value for whole grain, red rice - the lowest. The research revealed that the values of heat of combustion and heating value increased for almost all tested samples after grinding. The best results of heating values after grinding were obtained for black rice (17154 kJ/kg) with the fragmentation degree (1,23), the worst - for corn (15836,5 kJ/kg) for fragmentation degree (3,08).

The research allows to state that grinding operations results in an increased energy properties of biomass. It can be also assumed that biomass is a good substitute of fossil fuel, owing to the high heating values of tested grains. The results show that of all the materials tested, ground black rice was the best energy source. For complete analysis of the tested grains properties as fuel, the exhaust gases analysis should be done.

Acknowledgments

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NUMERICAL SIMULATION ON THE VIBRATION OF A TEST BED WITH ENGINE WITH DUAL MASS FLYWHEEL

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Abstract: This paper describes the results of numerical simulations of a vehicle engine with dual mass flywheel coupled with dynamometer. The differential equations of the model are given. The mass moments of inertia and the torsional stiffness are taking into account. Numerical simulations are carried out. Natural frequencies of the system are determinate. Bode diagrams of the system are shown and analysed. Recommendations to reduce vibration by using an additional flexible coupling are given.

Keywords: DUAL MASS FLYWHEEL, TRANSMISSION, DRIVELINE, DYNAMOMETER, DYNAMICS, VIBRATION

1. Introduction

The Dual Mass Flywheel (DMF) is widely used in modern diesel or gasoline direct injection engines. Obtaining the performance characteristics of the engines is important part of the engine research and development process. For testing the Internal Combustion Engines (ICE) the dynamometer test beds are widely used.

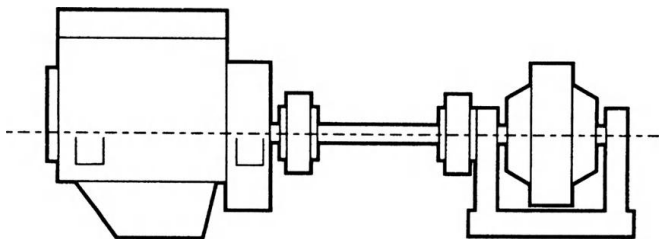


Fig. 1 Engine test bed [1]

The mechanical part of test bed shown in Fig. 1 consists of Internal Combustion Engine, propeller shaft (cardan), dynamometer and mountings.

The dynamometers are used, also when the transmission tests are performed. The test bed setup shown in Fig. 2, a) is intended for testing the internal combustion engine and transmission only [2]. In this case, an appropriately suited dynamometer is used to apply a load to the unit under test's cardan shaft.

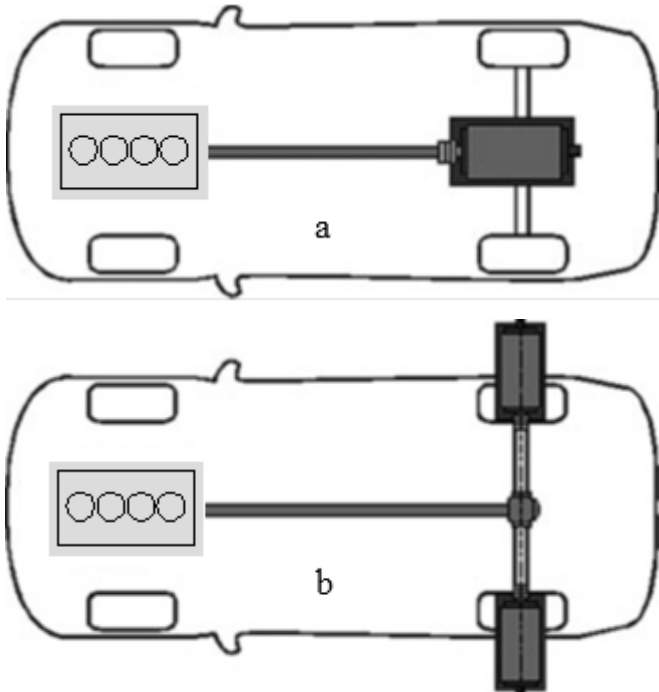


Fig. 2 Powerpack test bed for standard drives [2]

The setup shown in Fig. 2, b) allows testing of the transmission and the rear-axle final drive and differential.

Figure 3 presents the typical test bed setup for vehicles with front-wheel drive [2].

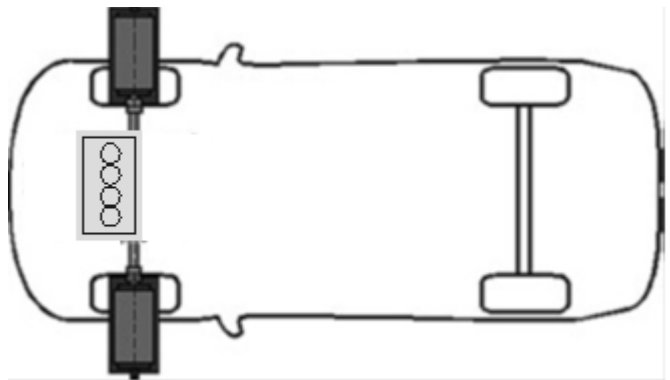


Fig. 3 Powertrain test bed for front-wheel drives [2]

The test bed setup with four dynamometers shown in Fig. 4 is used for testing four-wheel drive vehicle's powertrains.

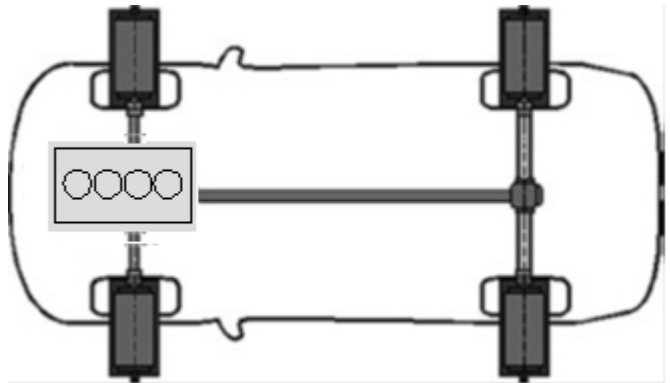


Fig. 4 Powertrain test bed for four-wheel drives [2]

In any case of the test bed layout concepts shown above, the inertia of the moved vehicle, reduced to the crankshaft is missed. The mass moment of inertia of the dynamometer rotors is much smaller than the vehicle mass inertia. Thus, the values of the natural frequencies of the dynamical system are much higher than the natural frequencies of the system with moved vehicle. As result the vibrations of the test bed increases. Some test mechanics and engineers welded the primary and secondary mass of the dual mass flywheel, or they used substituting single mass flywheel. This does not lead to good results with modern direct injection engines with high power received from a less number of cylinders.

In this regard, the purpose of this publication is to draw out the differential equations of an engine-dynamometer dynamic model with dual mass flywheel and to carry out numerical simulations. To

gives recommendations and technical solution for reducing vibration in test beds when engine with DMF is used.

2. Dynamic Model

A test bed dynamical model is shown in Fig. 5. The model consists of two elements of inertia, spring and damper. They are:

$$J_1 = J_c + J_{f1}$$

J_c – mass moment of inertia of the engine crankshaft;

J_{f1} – mass moment of inertia of the primary mass of the flywheel.

$$J_2 = J_{f2} + J_{ps}$$

J_{f2} – mass moment of inertia of the secondary mass of the flywheel;

J_{ps} – mass moment of inertia of the propeller (cardan) shafts;

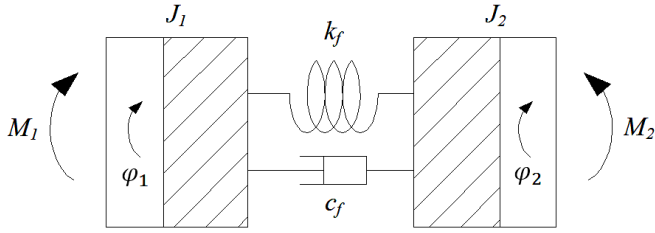


Fig. 5 Equivalent 2 DOF engine-dynamometer dynamic model

k_f, c_f – spring stiffness and damping of the dual mass flywheel;

M_1 – exciting moment from the engine;

M_2 – the dynamometer resistant moment.

Some companies offer rubber elastic couplings [3] that are mounted between the flywheel and the dynamometer. The spring stiffness and damping of the additional coupling are k_c and c_c .

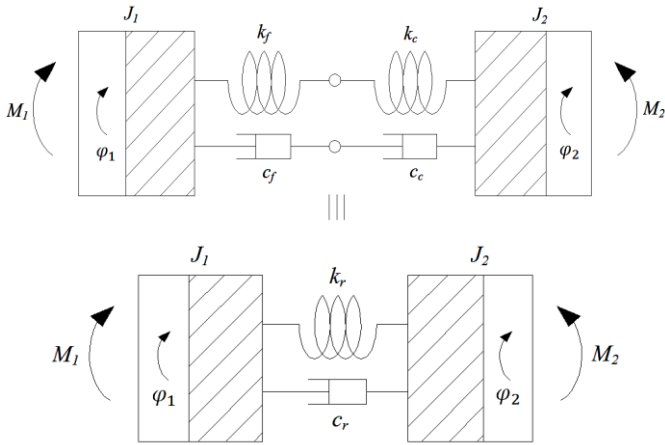


Fig. 6 Equivalent 2 DOF engine-dynamometer dynamic model with additional elastic coupling

Then according to Fig. 6 the reduced stiffness of the spring will be:

$$k_r = \frac{k_f \cdot k_c}{k_f + k_c}$$

The same holds for the dampers:

$$c_r = \frac{c_f \cdot c_c}{c_f + c_c}$$

The differential equations of the model in Fig. 5 are:

$$J_1 \ddot{\varphi}_1 + k_f (\varphi_1 - \varphi_2) + c_f (\dot{\varphi}_1 - \dot{\varphi}_2) = M_1$$

$$J_2 \ddot{\varphi}_2 - k_f (\varphi_1 - \varphi_2) - c_f (\dot{\varphi}_1 - \dot{\varphi}_2) = -M_2$$

And the differential equations of the model in Fig. 6 are:

$$J_1 \ddot{\varphi}_1 + k_r (\varphi_1 - \varphi_2) + c_r (\dot{\varphi}_1 - \dot{\varphi}_2) = M_1$$

$$J_2 \ddot{\varphi}_2 - k_r (\varphi_1 - \varphi_2) - c_r (\dot{\varphi}_1 - \dot{\varphi}_2) = -M_2$$

3. Numerical Simulation

Natural frequency of the system is:

$$\omega_{nat} = \sqrt{\frac{k(J_1 + J_2)}{J_1 \cdot J_2}}, \text{ rad/s}$$

In overcritical operating conditions ($\omega_{ex} > \omega_{nat}$), it must be ensured that the minimum excitation frequency will in all operating points will remain to a sufficient degree above the natural frequency [3]. The exciting frequencies (ω_{ex}) are, hence, the basic frequency (number of work cycles per unit time) and their integral multiples. They are proportional to the crankshaft speed. All of these exciting frequencies can resonate with one of the natural frequencies (Fig.7).

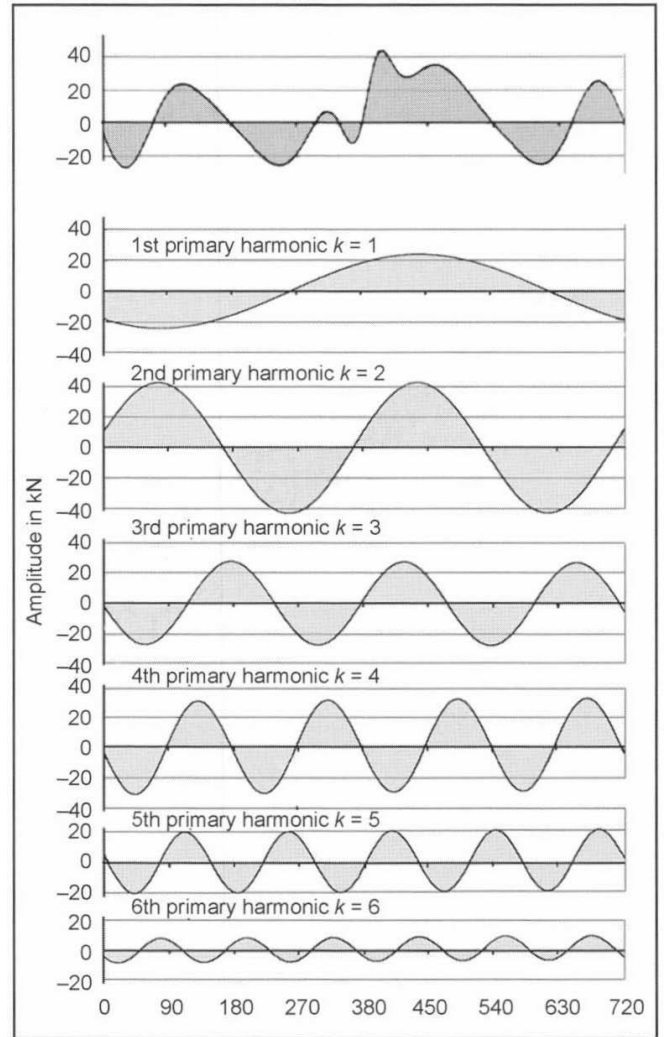


Fig. 7 Fourier analysis of a tangential force diagram: The tangential force curve is composed of the first six harmonics [4]

The numerical simulations are performed in program field of MATLAB. The Bode diagram in logarithmic scale is shown in Fig. 8. It can be seen that when the crankshaft is coupled with the dynamometer, using only a dual mass flywheel, the maximum ($\omega_{nat}=95,9$ rad/s) of the amplitude response (blue line) is within the engine operating range (between 94,2 and 628 rad/s). This can lead to undesirable vibrations in the test bench. When using a single mass flywheel without any elastic coupling, the maximum offsets to

the higher frequencies again in the engine operating range (red line - $\omega_{nat}=372$ rad/s).

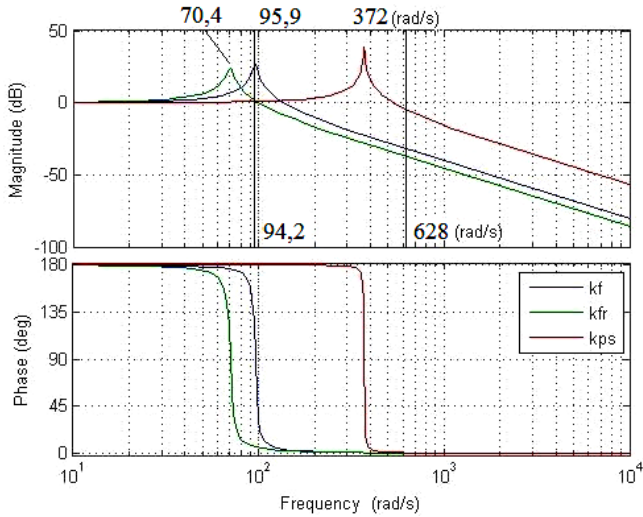


Fig. 8 Bode diagram of the system with DMF (blue line), DMF and high flexible coupling (green) and single mass flywheel without any elastic coupling (red). $k_f=600$; $k_r=323$; $k_{ps}=9000$ N.m/rad

Only when the crankshaft is coupled with a double mass flywheel and highly flexibility coupling to the dynamometer rotor, the maximum of the amplitude-frequency characteristic, offsets to the left, below the minimum engine speed (green line - $\omega_{nat}=70,4$ rad/s). In this way there is good damping of the vibrations of the first and higher order in the whole operating range of the engine.

4. Conclusion

The considered model enables to study the vibration in test bench with engine with dual mass flywheel. The numerical experiments with different variants of coupling the engine to the dynamometer are considered. The simulation shows that it is necessary to use a flexible coupling to reduce the total stiffness in the system. Thereby reducing the vibration transmitted from the engine to the test bed.

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