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Abstract: Low temperature combustion in homogeneous charge compression ignition (HCCI) engines is an alternative combustion technique to existing automotive spark ignition and diesel engines. The present study is aimed at fundamental understanding of phenomena affecting combustion and emissions of gasoline fuelled HCCI engines with internal gas re-circulation. The experiments that yielded the results were performed using dedicated engine test stand with single cylinder research engine. Different control parameter sweeps were accomplished to provide comprehensive data on engine operating parameters, combustion evolutions and emissions under variable conditions. Experimental analysis was also supported by engine cycle simulation to provide more comprehensive data on in-cylinder processes.

Keywords: ENGINES, LOW TEMPERATURE COMBUSTION, HOMOGENEOUS CHARGE COMPRESSION IGNITION

1. Introduction

Homogeneous charge compression ignition (HCCI) is nowadays emerging technology in the field of piston combustion engines. In the light of future emissions regulations, it appears to be the most promising solution for automotive propulsion systems fuelled with gasoline. This novel combustion technique provides significant reduction of fuel consumption and engine-out nitrogen oxides (NOX) emissions at low and medium engine load/speed conditions. HCCI combustion system utilizes auto-ignition of homogeneous in-cylinder charge, thus it is a combination of two acknowledged principles of internal combustion engines operation. The cylinder is fed with a homogeneous mixture, typical of spark ignition engines, and combustion is initiated by compression temperature, typical of diesel engines.

One of the first efforts in HCCI combustion implemented in a 2-stroke engine was reported by Onishi et al.\(^1\). Large amounts of residuals in the cylinder of 2-stroke engine provided end of compression temperature high enough to invoke the auto-ignition of gasoline. Najt and Foster\(^2\) presented the first results of a 4-stroke gasoline HCCI engine. Additional energy for gasoline auto-ignition under low compression ratio of the engine was provided by heating-up of the air-fuel mixture in the engine intake system. These initial research demonstrated reduction of cylinder-out NOX emissions by 98% in comparison to spark ignition engines.

During the last decade, the more production feasible HCCI approach was developed. It utilizes internal exhaust gas recirculation (EGR) obtained by a negative valve overlap (NVO). The NVO technique allows auto-ignition of high octane number fuels (gasoline like) at compression ratios typical of spark ignition engines\(^3\). In order to trap a sufficient amount of exhaust in the cylinder, an exhaust valve is closed before top dead center (TDC) in the exhaust stroke. The trapped residuals are then compressed and expanded after TDC. To avoid excessive backflows into the intake port, the opening of an intake valve is retarded. This technique is also referred to as controlled auto-ignition (CAI)\(^4\). Example traces of in-cylinder pressure, temperature and valve lifts during engine operation in the NVO mode are shown in Fig. 1.

Combustion process in an HCCI engine is initiated by increase in temperature inside the combustion chamber and starts at multiple sites simultaneously. In contrast to spark ignition and diesel engines, combustion process is very rapid and results in realization of the close to ideal Otto cycle. It increases thermal efficiency of the engine. Due to volumetric combustion of the mixture, temperature in the combustion chamber is uniform, and considerably lower than maximum temperatures in case of other combustion systems. Additionally, high EGR rates are provided, which further contribute towards temperature reduction. Thus, low temperature combustion is realized which produces extremely low NO\(_X\) emissions and smokeless exhausts\(^5\).

Ignition and combustion processes in HCCI engines are governed primarily by chemical kinetics of hydrocarbons oxidation. Auto-ignition timing is primarily controlled by compression temperature histories. It is also affected by mixture composition and its ignitability\(^6\). However, it should be noted that aforementioned parameters cannot be separately and directly controlled in practical engines.

![In-cylinder pressure, temperature and valve lifts during HCCI engine operation using NVO technique.](image)

Fig. 1 In-cylinder pressure, temperature and valve lifts during HCCI engine operation using NVO technique.

Thus, successful implementation of such engines will depend on fundamental understanding of the in-cylinder processes and development of control technologies. The most crucial challenges for development of HCCI engines are: control of combustion timing under variable operating conditions, expansion of operating range in the HCCI mode and engine operation under transient conditions\(^7\).

This study presents achievements of combustion engine group from Lublin University of Technology in the field of understanding of fundamental phenomena of HCCI engine operation. The presented results were collected during the past few years at a dedicated laboratory with a single cylinder engine. The purpose of this study was not to optimize engine operation to achieve the best operating parameters. Rather, it was intended to demonstrate the effects of different control strategies on combustion and exhaust emissions.

2. Experimental equipment

The experiments that yielded results presented in this study were carried out using dedicated single cylinder research engine. The engine was built in-house using special hydraulic engine head, which enabled fluent regulation of valve timings and valve lifts. Fully variable valvetrain was set to achieve NVO, and thus HCCI combustion. Swept volume of the cylinder was 489 cm\(^3\) and geometric compression ratio was 11.7. Fuel was directly injected into the combustion chamber with the use of solenoid swirl-type injector. The engine was fuelled with a pump-grade European Euro Super gasoline (95 research octane number). The engine was fitted with electrically driven vane compressor to increase intake pressure. Thus, it was able to be operated as naturally aspirated or boosted. The engine was installed on a test bench equipped with direct current dynamometer governed by an automation system. Fig. 2 shows appearance of the experimental engine.
Fig. 2 Appearance of the experimental test stand.

The engine test bench was equipped with all necessary measurement and control instruments. Fuel consumption was measured using a fuel balance and intake air flow was measured with a thermal mass flow meter. The engine was also equipped with a set of pressure and temperature transducers in order to control the thermodynamic conditions of all media; intake, exhaust, cooling liquid, etc. In-cylinder pressure was measured with the use of a miniature pressure transducer installed directly in the engine head. Pressure and other crank angle (CA) based parameters were recorded with a constant angular resolution of 0.1 °CA, where analog-digital converter was triggered by optical encoder. The composition of exhaust gas was measured with a Fourier transform infrared (FTIR) multi-compound analytical system. Additionally, oxygen content in exhaust gas and excess air ratio (λ) were provided by wide-band oxygen probe installed in the exhaust runner. Detailed descriptions of the experimental test stand can be found in10, 11.

3. Experimental results

Effects of valve timing

Valve timings are basic control parameters of the HCCI engines operated in the NVO mode. Valvetrain settings enable regulation of the quantity of exhaust gas trapped on the cylinder as well as amount of intake air. However, these two parameters are coupled and cannot be controlled separately. During the intake process the cylinder is partially filled with trapped exhaust. Thus, amount of aspirated air is affected by amount of internally re-circulated exhaust and its temperature12.

To demonstrate these complex effects, intake and exhaust valve timings were varied independently. Exhaust valve closing (EVC) was varied in a range from 627 °CA to 646 °CA, whereas intake valve opening (IVO) was varied between 77 °CA and 99 °CA. All valvetrain timings are expressed in domain of crank angle as shown in Fig. 1. This experiment was performed at constant engine speed of 1500 rev/min. Fuel was injected into the cylinder in a single dose during exhaust expansion, during the NVO period. Excess air ratio was maintained nearly stoichiometric.

Fig. 3 IMEP (a) and NOX emissions (b) with respect to valve timings; arrows show direction of NVO angle increase.

Fig. 3a shows that engine load expressed as IMEP is controlled solely by exhaust valve timing. Delay of EVC by 19 °CA resulted in change in IMEP from approx. 0.24 MPa to 0.35 MPa. Primarily this change was a result of variable amounts of aspirated air. At constant air excess, quantity of fuel was varied too, thus amount of chemical energy introduced to the cylinder was reduced at EVC advance. Additionally, engine load was affected by variable thermal efficiency. Thermal efficiency increased for increasing engine load, as lower was contribution of thermal losses to energy balance of the engine. It should be also noted that for constant exhaust valve timing, intake valve timing affected thermal efficiency. For all conditions, the highest efficiency was observed for moderate intake valve timings. While engine load was not strongly influenced by IVO, NOX emissions appeared to be affected by the intake valve timing to the high extent, as shown in Fig. 3b.

Fig. 4 In-cylinder pressure for variable EVC at IVO = 89 °CA (a) and for variable IVO at EVC = 640 °CA (b).

To provide more insight into the effects of valve timings on combustion, Fig. 4 shows in-cylinder pressure for variable EVC and variable IVO. It can be noted from Fig. 4a that the change in engine load as a result of variable EVC does not affect auto-ignition timing. Obviously peak pressure decreased for lower loads. In contrast, at constant load, variable IVO modified start of combustion timing, as shown in Fig. 4b. Both early and late IVOS resulted in combustion delay. Later combustion reduced peak in-cylinder pressure and temperature, which was reflected in the NOX emissions, shown in Fig. 3b. The delay of combustion was attributed to the cooling effects of early and late backflows through the intake valve. The pressure traces explain drop in thermal efficiency at extreme intake valve timings. Trivially, slower combustion reduced work produced by the engine cycle.

Effects of excess air

The previous section showed that load control of the engine using solely variable valve timings under constant λ enabled engine operation in a very narrow range of IMEP. Thus, to expand engine operation range it is necessary to combine variable valvetrain settings with variable λ. Experiments at variable λ were performed for four NVO angles. The configurations of EVCs and IVOS were set to provide maximum thermal efficiency under desired engine load. For EVC varied from 627 °CA to 646 °CA, IVO was varied from 89 °CA to 83 °CA accordingly. Fuel was injected in a single dose during exhaust expansion during the NVO period.

Fig. 5a shows the results for IMEP with respect to λ values, which were varied from stoichiometric to maximum attainable level, where limitation came from occurrence of misfires. It can be noted that even small increase in air excess resulted in relatively high drop in IMEP. This effect was attributed to variation in EGR rate. Increase in λ results in a decrease in exhaust temperature. When colder exhaust gasses are trapped, their amount increases, thus reducing amount of aspirated air. As a result, overall effect of λ on engine load is much stronger than one resulting from air excess
itself. One conclusion can be drawn from the above analysis that λ is an effective parameter for controlling of the engine load. Fig. 5b demonstrates how NOX emissions are sensitive to engine load and air excess. For stoichiometric mixture, where IMEP was varied from approx. 0.25 MPa to 0.35 MPa NOX emissions were increased 8 times, showing EGR impact on emission. In this range, EGR rate varied between 0.49 and 0.54, resulting in drop in peak temperature from 1708 K to 1510 K. It should be noted that emission levels of spark ignition engines under comparable operating conditions are approx. 12 g/kW h. The NOX emissions increased at slight λ increase above stoichiometric level, as a result of higher oxygen availability. Further increase in λ resulted in substantial drop in emissions due to increasing fuel dilution by air and re-circulated exhaust.

(a) IMEP (a) and NOX emissions (b) with respect to λ at variable valve timings.

Fig. 6 In-cylinder pressure (a) and temperature (b) for variable λ; EVC = 634 °CA.

Limitations of applicable λ values during HCCI operation result from delay of auto-ignition, as shown in Fig. 6a. When fuel chemical reactions are not commenced before piston top position, combustion will not take place due to dropping in-cylinder temperature.

It should be noted, that temperature of gasoline auto-ignition is not sensitive to λ; thus, combustion timing is controlled solely by compression temperature histories. Fig. 6b shows that end of compression temperature decreases for leaner mixtures, indeed. Substantial drop in peak temperatures clearly accounts for the reduction in NOX emissions.

Effects of boost

Meaningful effect of air excess on NOX emissions and combustion, shown in the previous subsection, reveals another possibility to further reduce emissions and control heat release by using boost. Additionally, application of boost enables increase of quantity of aspirated air and can be utilized in HCCI engines for extension of operating regime.

Tests that yielded results presented in this section were performed at 1500 rev/min. Likewise in previous experiments, fuel was injected during the NVO period. Valve timings were set at 640 °CA for EVC and 82 °CA for IVO. Boost pressure sweeps were applied at different amounts of fuel. In the first case 13.3 mg of fuel was injected, which provided stoichiometric mixture at naturally aspirated conditions. Boost pressure was increased from atmospheric to approx. 150 kPa absolute, providing variable air excess at constant fuelling. In the second case mass of fuel was increased to 18 mg in order to explore engine operation above limit resulting from insufficient amount of air under naturally aspirated conditions. At this condition, minimum boost pressure was increased to provide stoichiometric mixture. It was found that thermal efficiency at constant fuelling and variable intake pressure was nearly constant, thus achieved IMEP values were approx. 0.35 MPa and 0.5 MPa, accordingly. Values of λ were varied from slightly lean to 1.5 for less fuel injected and to 1.3 for more fuel injected.

(a) IMEP [MPa] (a) and NOX emissions (b) with respect to Boost pressure.

Fig. 7 NOX emissions with respect to boost pressure.

(a) In-cylinder pressure (a) and temperature (b) for variable boost pressure; IMEP ~ 0.5 MPa.

Fig. 8 In-cylinder pressure (a) and temperature (b) for variable boost pressure; IMEP ~ 0.5 MPa.
by factor of 5. For the highest boost pressure increase in the emissions was noted, however, it was attributed to unstable combustion at very lean mixture. Likewise, in the case of higher fuel dose, increase in boost pressure appeared to be an effective method for reduction of NO\textsubscript{X} emissions. Narrower range of applicable boost pressures resulted from increase in combustion harshness. It was found that increase in boost pressure advanced auto-ignition, as shown in Fig. 8a. Additionally, higher peak pressures were noted, which increased combustion noise and mechanical loads of the combustion chamber. Fig. 8b shows that advance in combustion resulted from increase in end of compression temperatures. At the same time, peak temperatures were reduced, justifying drop in NO\textsubscript{X} emissions.

4. Engine cycle modelling

Analysis of auto-ignition timing at variable boost pressures reveal a peculiar effect. Considering changes in the amounts of fresh air and retained residuals, as well as exhaust temperature, it was expected that boost application will result in drop of start of compression temperature and auto-ignition retard. However, experimental results provided relationships opposite to expected ones, as shown in Fig. 8. Increase of amount of aspirated air at constant fueling increased end of compression temperature and thus, resulted in advance of auto-ignition.

To provide detailed data on gas exchange and thermal balance between trapped residuals and aspirated air, experiments were complemented by zero-dimensional modelling of the engine cycle. Boost software from AVL was used for simulation. Computations were done at the same conditions as during experiments. The obtained results confirmed experimental observations that increase of intake pressure increased compression temperature, besides reduction of the EGR rate at elevated amounts of fresh air in the cylinder. Temperature at IVC event increased by 10 K approximately for intake pressure change from atmospheric to 150 kPa, as shown in Fig. 9. The results also clarified the reason for the delay of auto-ignition for increase of air excess under constant fueling conditions, shown in Fig. 6. Increase of excess air reduces temperature as well, besides compensation by EGR rate.

\[ \text{Fig. 9 Calculated temperature at intake valve closing (IVC) versus air excess ratio and EGR rate at variable intake pressure.} \]

5. Conclusions

Low temperature combustion of gasoline in an HCCI engine operated in the NVO mode was studied using variable control methods. Variable valve timings, air excess ratio and boost pressure were accomplished to provide experimental end modeling data on their effects on combustion and emissions. The collected data show fundamental principles of residual effected HCCI engine operation. The findings of the study can be summarized as follows:

1) At regulation of the engine via valvetrain settings exhaust valve timing determined fresh air intake, whereas intake valve timing had much smaller effect. However, intake valve timing had an impact on heat release rate, and therefore influenced thermal efficiency and emissions of nitrogen oxides.

2) Amount of fuel and resulting excess air ratio are effective methods for engine load control. Lean mixture boundary of the engine operation resulted from drop in compression temperature, which caused delay of auto-ignition. However, lean combustion produced extremely low NO\textsubscript{X} emissions.

3) Application of boost enabled extension of permissible engine load range. Additionally, increase in boost pressure and resulting fuel dilution further reduced NO\textsubscript{X} emissions.

4) Auto-ignition timing and boundaries of engine operation in an HCCI mode are resulting primarily from temperature of the mixture at the end of compression. The thermal state of the mixture is mainly affected by thermal balance of the gas exchange process.

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References


Optimum connecting rod design for diesel engines

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Abstract: One of the most critical components of an engine in particular, the connecting rod, has been analyzed. Being one of the most integral parts in an engine’s design, the connecting rod must be able to withstand tremendous loads and transmit a great deal of power. This study includes general properties about the connecting rod, research about forces upon crank angle with corresponding to its working dependencies in a structural mentality, study on the stress analysis upon to this forces gained from calculations and optimization with the data that gained from the analysis. In conclusion, the connecting rod can be designed and optimized under a given load range comprising tensile load corresponding to 360° crank angle at the maximum engine speed as one extreme load, and compressive load corresponding to the peak gas pressure as the other extreme load.

Keywords: CONNECTING ROD, OPTIMIZATION, DIESEL ENGINE

1. Introduction

During the design of a connecting rod, optimized dimensions allowing the motion of rod during operation should be taken into account in the calculation of variable loads induced in the system and the resulting data should be tested experimentally. Therefore, a connecting rod design and manufacturing, production technologies, materials performance analysis is laborious and long lasting process involving many steps. Computer-aided design and analysis programs which are being used widely currently, are part of the design of a more flexible manner and to ensure a shorter time changes in design. By applying the finite element method, more reliable and more realistic results are obtained in practice. Thus, instead of using all designed models, only the most appropriate model optimized numerically can be used for performance testing, providing significant time and cost saving.

Connecting rod’s loading state is quite complicated. It is exposed to high pressure loads due to the combustion in the chamber, is also repeatedly subjected to high tensile forces due to the inertia. Therefore, the reliability of this engine component is of critical importance. Due to the reasons mentioned above research topics such as connecting rod manufacturing technology, materials, performance simulation, fatigue are the subjects for both academic and industrial research. The connecting rod material and manufacturing technique have been demonstrated as the most important parameters for optimal design [1, 2]. Also, the stress distribution induced in the connecting rod depending on operating conditions, is examined in detail in related literature [3-5]. There are also remarkable researches involving the optimum design of the connecting rod which exposed to variable stress from the viewpoint of fatigue [6-9].

In the work presented here, a connecting rod is modeled numerically for a 4-cylinder, 4-stroke diesel engine. The most critical sections of connecting rod are determined based on mechanical strength. According to this, the optimum geometric parameters effecting connecting rod’s mechanical parameters have been optimized.

2. Detailed information for available connecting rod

The modeled connecting rod is used for a commercial 4-cylinder, 4-stroke diesel engine. Operating parameters of engine is as follows: Maximum cylinder pressure is 190 bar and compressive load at 2000 rpm will be 100 kN when engine reaches its maximum speed of rotation (5100 rpm) tensile load will be 15 kN. Nominal power of engine is 150 kW, connecting rod length is 145 mm, diameter of crankshaft is 43.2 mm, stroke of cylinder is 88 mm, and diameter of cylinder is 85 mm.

A set of forces resulting from various effects such as pressure inside the cylinder, inertia and friction induce stresses on connecting rod. Force caused by pressure inside the cylinder reaches its maximum value around the top dead center. Inertia forces results from the acceleration of moving elements. Numerical values of these forces are dependent on the type, rated power and rotational speed of engine. Forces resulting from the pressure especially gain importance on diesel engines due to high combustion chamber pressures. Inertia force reaches remarkable values at high speeds. Inertia forces with respect to crank angle is given in Table 1.

### Table 1: Inertia forces for various crank angles.

<table>
<thead>
<tr>
<th>Crank angle (°)</th>
<th>Inertia Forces (kN)</th>
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<tr>
<td></td>
<td>Fp_x</td>
</tr>
<tr>
<td>0</td>
<td>-4.5</td>
</tr>
<tr>
<td>80</td>
<td>-256.0</td>
</tr>
<tr>
<td>60</td>
<td>-152.8</td>
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<tr>
<td>90</td>
<td>153.6</td>
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<td>150</td>
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<tr>
<td>180</td>
<td>2.5</td>
</tr>
<tr>
<td>210</td>
<td>174.4</td>
</tr>
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<td>240</td>
<td>274.4</td>
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Fpx: Inertia force at x-direction; Fpy: Inertia force at y-direction

In industrial applications, I and H formed cross-section for connecting rod design are used extensively regarding lightness and manufacturing advantages. In this study, H-type section is selected in order to achieve easy manufacturing from the viewpoint of lightweight design and to form more suitable transient region between the rod and big-end.

Connecting rods are generally manufactured by forging. Therefore, high strength malleable steels should be used as connecting rod material. Depending on the application, connecting rods are made of carbon steels or steel alloys. The research carried out on the powder forged in the 1970s and 1980s has led to the making of this method industrially applicable. In Table 2, the modeled connecting rod’s mechanical properties are presented [10].
### Table 2: Mechanical properties of connecting rod material.

<table>
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<tr>
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<th>Measured</th>
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<tr>
<td><strong>Powder forged rod</strong></td>
<td></td>
<td></td>
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<tr>
<td><strong>Powder Metal</strong> (C-70)</td>
<td></td>
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<tr>
<td><strong>Tensile Strength (MPa)</strong></td>
<td>696 min</td>
<td>794</td>
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<tr>
<td><strong>Yield Strength (MPa)</strong></td>
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<td><strong>Elongation (%)</strong></td>
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<tr>
<td><strong>Specific gravity (g/cm³)</strong></td>
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<td><strong>Hardness (HB)</strong></td>
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<td>245</td>
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### 3. Modeling and Constitution of Finite Element Model for Connecting Rod

The literature review showed that there are three fundamental dimensions to be taken into account in a connecting rod design. These are:

- Small end inner diameter,
- Cross-section,
- Big end inner diameter.

These dimensions change with respect to type and properties of engine. For numerical analysis, a small end inner diameter, a big end inner diameter and a cross-section are defined based on given engine parameters. Some basic dimensions required for analysis is given Fig.1. According to this, the connecting rod is dimensioned by using empirical data recommended for basic parameters [11].

### 3.1 Finite Element Model

3D model and meshing of the designed connecting rod is shown in the Fig. 2. Meshing of model has been carried out by using tetra elements. The reason for using hexa elements instead of tetra elements is that connecting rod has a more detailed model surface relative to the crank journal and piston pin models. Discontinuities can be more easily handled in meshes with tetra elements.

### 3.2 Numerical Analysis

Numerical analysis is achieved in two stages. At the first stage, the compression effect resulting from the maximum gas pressure has been analyzed. At the second stage, the tensile effect occurring in the maximum speed due to inertia forces has been solved. The results of these analysis is shown in the Fig. 4. As seen from the Fig. 4, under compressive stress, some sections of connecting rod seems overloaded. But, under the tensile stress, same element gives more promising result regarding the material’s yield strength of 462 MPa. According to the results of the analysis, some connecting rod sections have insufficient strength under compressive stress. Unnecessary accumulation of material especially in the shoulder region has been observed in analysis. Therefore, in first step the connecting rod section is strengthened, and at the other step, an optimization technique is implemented to regions having excessive material. In addition, the most important region in the design of a connecting rod excluding big head, small head and rod is transition zone to the heads. The highest stresses are induced in these zones.
As a solution, it is assumed that an increase in the \( l_c \) and \( b_{sh} \) dimensions would be satisfactory. Since, the available connecting rod design gave good results in tension analysis, an increase in the \( l_c \) and \( b_{sh} \) dimensions make the situation better.

**4. Optimization of connecting rod**

At this stage, a practical method is used to reach an optimal geometry by constituting a drawing model with an ability of varying parametric dimensions utilizing an analysis tool embedded in the software. Basic static models give satisfactory results for obtaining idealistic geometry at the optimization stage. Different models derived from parametric model by changing critical dimensions have been analyzed by using finite element software to reach an ideal form. By using optimal dimensions obtained from optimization process and by taking into consideration the appropriate manufacturing methods, a connecting rod is designed as shown in the Fig. 5 (named Model-4). The wall thickness was increased in this design in order to make improvement against compressive stresses.

Although new design modifications on connecting rod gave promising results, but the desired improvements have not been reached. It has been found that, increasing wall thickness is not sufficient enough to get better analysis results. Since further increase on wall thickness would be unfavorable, it has been considered that some fundamental variations on connecting rod form would be more realistic.

As seen from Fig. 7 that in the modified model named as Model-5, the material is gathered on the transition zones from rod to big and small heads. Some remarkable changes have been carried out especially on the transition zone between rod and big head. Unnecessary parts have been removed from the shoulder region. Force lines on the transition zone between rod and small head have been improved. Furthermore, \( b_{sh} \) dimension in Fig. 1 has been increased.

The analysis results for Model – 5 is shown in Fig.8. A satisfactory amount of improvement is observed for this model. Although there are some local points around the transition zone in which the yield strength of material is exceeded, this is mainly due to the neglecting damping effect and absence of bush on the on connecting rod. Analysis of connecting rod under tensile stress gives satisfactory results as expected. The overstressed point on the analysis results table take place as a result of a discontinuity on
meshing process. This discontinuity has no effect on analysis process.

![Fig.8: Model -5 a) Compression analysis. b) Tension analysis](image)

### 4.1. Weight Reduction Studies

As a result of analysis conducted, satisfactory improvements are carried out for connecting rod. After this stage, some sections of connecting rod where the stress values are low can be redesigned from the viewpoint of weight reduction. Some remarkable amount of material was removed from cap section. Furthermore, material reduction was applied to the transition regions from the rod to big and small heads. Optimized connecting rod after weight reduction process is shown in Fig.9. Weight of connecting rod was reduced from 687.4 g to 663.7 g with this process.

![Fig.9: Optimized connecting rod](image)

Analysis result of connecting rod under compression for the final optimized design is shown in Fig. 10.

**Fig.10 : Analysis of final design connecting rod under compression.**

Transition from rod to big head is transferred to totally safe region according to the analysis for compressive stress. Even though the stresses on the transition section from rod to small head seems very close to yield strength, this is caused by not using a bush for modelling the system.

### 5. Conclusion

At end of this study, a new connecting rod design having optimized dimensions and sufficient strength has been suggested for given basic dimensions and operating conditions. At the first stage, the connecting rod formed from preliminary work was analyzed numerically and weak points of this design was determined. The optimum connecting rod geometry was reached by applying some design modifications to these points.

In this study, elastic properties for connecting rod material was used. Using plastic material data will be more satisfactory for analysis. Induced stresses can be more calculated more precisely by meshing the connecting rod finer and more detailed.

For preliminary work, linear static analysis will be sufficient in order to determine the basic geometry. But, for future work, bushing, crank bearings and damping effect can be taken into account. By introducing the bolts tightened with rated torque in order to obtain correct preload to analysis, it is possible to get results which have relatively good conformity with actual part.

Furthermore, these analyses can be compared to test data in order to increase reliability of design.

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TRAIN PATHS ALLOCATION AS A PART OF RAIL INFRASTRUCTURE CAPACITY RESEARCH

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Abstract: The paper is focused to relationship between rail infrastructure manager and railway undertaking as a subject in railway market. The rail capacity research offers response to questions about possibilities how to satisfy the railway undertaking requirements. The train path allocation is continuous process that is organised by European law in liberalised railway sector. This contribution provides a new look at the issue of rail capacity, its identification and provision for railway transport undertakings. The paper introduces the selected methodologies for railway infrastructure capacity calculation shortly and its importance for non-discriminatory capacity allocation.

Keywords: UIC METHODOLOGY, RAILWAY INFRASTRUCTURE, THEORETICAL CAPACITY, PRACTICAL CAPACITY

1. Introduction

Allowing undertakings to access to the railway infrastructure is one of the precondition for achieving a competitive railway services market. The direction of infrastructure development refers to the two key subjects generally: the state that creates transport policy at national level and railway infrastructure manager (IM). IM operates infrastructure under the permission and also offers it to the undertakings with a valid license for transport operation. Third subject can be a union of states with interest to create a transnational transport market with non-discriminatory terms. Thus determines the basic transport policy that is superior to transport policy at national level.

The key issue for IM is capacity management. There is need to know the infrastructure capacity. The capacity expresses the infrastructure manager’s business offered by the allocating of the train paths to the railway undertakings.

2. Guiding principles in determining the capacity in Slovakia compared to Croatia

The methodology including the detection procedures for capacity of infrastructure facilities was contained in prescription D24 CSD (Czechoslovakian railways). ŽSR (Railways of Slovak republic) as its successor took over the methodology in Regulation D24 ŽSR.

The theoretical and practical capacity

The capacity can be in principle divided into:

- theoretical (maximum),
- practical.

When calculating the maximum capacity any loss of time is not considering and it is assumed that the device capacity we determine is used exclusively for activities, for which it is intended and necessary technological blocking times follow up tightly and immediately without any loss of time.

When calculating the practical capacity we consider not only the need for maintenance of equipment or the fact that the equipment is also used for other purposes than for which it is primarily designed and used, but also time to eliminate the backup for possible defects or irregularities in traffic.

The capacity of equipment

The capacity can be determined for these railway equipments:

- track line,
- station gridiron,
- station line.

For determining capacity of track lines can be used graphically methodology, analytically methodology as well as combination of both.

Maximum capacity can be expressed by a formula:

\[ N_{\text{max}} = \frac{T}{t_{\text{obs}}} \]  

[technological operations/calculation period]

where:

- \( T \) calculated time (peak time or all the day) [min],
- \( t_{\text{obs}} \) average time for realization of the following technological operation (train drive, shunting operation etc.) [min].

If there is calculated a practical capacity, we could take into consideration the time for maintenance as well as the reserve time for elimination of delays, in which are primary traffic operations not possible. Practical capacity is expressed by a formula:

\[ n = \frac{T - (T_{\text{col}} + T_{\text{rail}})}{t_{\text{obs}} + t_{\text{col}} + t_{\text{rail}}} \]  

[technological operations per period]

where:

- \( T_{\text{col}} \) total time, in which is the facility out of order because of maintenance, inspection or revision [min],
- \( T_{\text{rail}} \) total time, in which is the facility occupied by another operations, that are not primary intended for this facility [min],
- \( t_{\text{col}} \) technological time of facility occupation by one technological operation [min],
- \( t_{\text{rail}} \) average time of probable mutual distortion of two operations (trains) in the places of potential threats (simultaneous drives impossible) accrued at one technical operation (train) [min].

The capacity is defined as a value of calculated trains of basic parallel train diagram (mostly represented by middle-distance freight trains or by the most frequent train category) or in an average trains (average value of time occupation – train sequence time).

To compare the analytical methods, the Faculty of Transport and Traffic Science of the University of Zagreb presented a quite similar methodology, that we can mention from this point of view. For the calculation of particular capacity is used a coefficient of elimination. There are used two different methods for capacity calculation. The first method deals with the maximum capacity in the number of trains or train pairs (the differentness), that are the most frequent on the line and these are used as a base. By using equivalents they transfer other train categories to this basic trains and the capacity is calculated such for parallel train diagram as for non-parallel train diagram. Other method determines the capacity
without isolating the category of trains and takes into consideration the probability of the influence of mutual relations of certain types of trains.

For example, the capacity of the line section (both directional one track line) is presented in the following formula:

$$N = \frac{1440 - T_{pr}}{T_{pg} \alpha_{ps}} \quad \text{[train pairs / day]}$$

where:

- $T_{pr}$ time of technological downtime [min],
- $T_{pg}$ graph period (means for both directional train pair) [min],
- $\alpha_{ps}$ coefficient of operation reliability [-].

That means, that capacity is detected for train pairs (supposition for only mutual trains) and the other circumstances that decrease the value of capacity are expressed by the coefficient in comparing with Slovak method that uses marginal amount of exact values in formula. But the whole issue is quite difficult and highly professional and slickly elaborated in detail.

3. Contemporary trends in the determination of the capacity worldwide

In the world are currently using many methodologies for detection of railway infrastructure capacity. In the Europe, the International Union of Railways (UIC) draw up regulations for capacity, which aims to unify previously used national methodologies of each European railway networks, so that the results of the assessment of the individual parts of the corridors are mutually comparable. Leaflet UIC 406 is not mandatory also allows infrastructure managers to use also the national methodology.

**UIC methodology**

Leaflet UIC 406 was adopted in 2004. It admits that capacity qua doesn’t exist, because it depends on the exploitation of infrastructure. Basic parameters from which depends the infrastructure capacity depends are the number of trains, average speed, stability and heterogeneity of timetable. Like the D24 methodology so the UIC methodology uses in determining the track section capacity a method of inserting an additional train paths to advance designed timetable. Calculation of total capacity must be applied by compression of train paths at restrictive track section. The compression of paths is performed regardless to the downstream of the track sections in so-called representative day and in a maximum (peak) operation (minimum 120 minutes).

There is space for research continuance. At first, the compression of train paths is well to apply for and double -track line. That is reason the path not into account the possible ordering the requirement for train path by searching a limiting partial section. Methodology does not taking into account the train path from the initial point to the end pint of the track section a method of inserting an additional train paths to advance designed timetable. Calculation of total capacity must be applied by compression of train paths at restrictive track section. The compression of paths is performed regardless to the downstream of the track sections in so-called representative day and in a maximum (peak) operation (minimum 120 minutes).

There is space for research continuance. At first, the compression of train paths is well to apply for and double -track line or for every track of one direction traffic. There is not exactly done how could we proceed it to the compression of a both directional train paths. On the other hand, the idea of relative capacity which depends at various aspects is one of the most important notions that we can follow as a hypothesis and based on that we can involve our ideas. That is the reason continuing research in this direction.

**Simulation as a capacity exploitation planning tool**

Due to different scientific theories we know several different types of railway simulations. Railway simulation models can be classified generally to:

- scope (macroscopic, microscopic),
- analytical approach (stochastic, deterministic),
- processing technique (synchronous/ parallel/ contemporary, asynchronous).

Many of the models used in practice are based on the theory of synchronous simulation, but we can deal with asynchronous simulation method as well. Some selected synchronous simulation tools are RailSys, RailPlan, VISION, OpenTrack, SIMONE, FALCO, TRANSIT, RAILSIM, SENA JŘ VT / ZONA CP VT.

After the first experience with the good-known simulation software OpenTrack we have got an unique opportunity to work with the RailSys as well as the OpenTrack at the University of Zagreb and thus evolve new ideas in this field by proceeding a simulation tests and make partial conclusions about the direction of new ideas. The research now is in progress by working in an infrastructure manager of RailSys and the first results are coming soon.

As example, the representatives of asynchronous simulation tools BABSI a STRESI are developed by German university RWTH Aachen.

### 3.1. Transcript of a new methodology design

While so far it’s been the trend in detection of railway infrastructure capacity to simplify and average many of inputs, such as the relating calculations to the "average" trains, respectively simplifying the elements of time, so in terms of technological progress it is not a big deal today to create a comprehensive system that would be as far as possible considering a diversity of these parameters and that would provide a space for detailed and transparent detection of capacity with identifying the bottlenecks as a problem of railway infrastructure. Such a system could make an interactive platform and could quick operational respond to an incident caused by the operation, respectively to more and new assigned paths. Another focus area is based on the fact that the infrastructure capacity analysis takes into account the integrity of the train path from the initial point to the end pint of the track section of a line and thus the capacity of railway section is detecting by searching a limiting partial section. Methodology does not taking into account the possibility ordering the requirement for train path using a part of the track section only. That is reason the path not always goes through the limiting partial section. The result is a distortion of the actual infrastructure capacity which may in some cases hinder to respond flexibly to supply the demand for transport paths and degrade the procedure to increase the competitiveness of railway transport to another modal transport.

### 3.2. Determination of the basic principles of capacity calculation in research

Tackling the issue of capacity of oven systems (infrastructure equipment, etc.) is a complex process in which basic rules of traffic operations must be followed while safeguarding the diversity that exists in this field in different countries. The basic premise is that practical capacity and required reliability of transport operations are consistent and as much as possible optimized.

The purpose of the new methodology proposal for capacity detection, respectively providing appropriate information, is the determination and the establishment of a procedure that can flexibly respond to immediate changes in traffic management and will be so full auxiliary tool for traffic planning that can be useful in relation to the railway infrastructure managers and railway undertakings. The basic scheme of the proposed methodology in which the undertaking enters into the process of selecting a suitable free train path is based on the following steps (Figure 2):

- **route choice** – customer (operator) in the interactive software application pre-selects train path, which may contain several lines,
- **basic state of capacity view** – the maximum capacity for the selected line and its status from preliminary timetable is shown (Figure 2),
- **track section choice** – in case of non route-long paths,
- **specifications for allocation of path** – detailed specification of the requirements for the allocation of such path, as a date and approximate time position of the path, train type (weight,
length, rolling resistance, traction, etc.), the base of which is recalculated into time elements needed for upgrade the capacity, along the number of paths (one, more or two-way path), use of station lines with a boarding platform edge or stops, the possibility (accepting) of staying on the journey (crossing over, prevention, traffic sequence), it means priority paths and so on.,

- **current and planned state view** – appears updated capacity on the selected track, which can then be confirmed by customer, respectively it will not be available and customer will be offered to change the planned route due to insufficient capacity of this option,

- **confirmation and incorporation of requirements** – after successful completion of a route selection for the request for capacity is incorporated as requirement into the system and appears as increased used capacity, respectively reduced free capacity in all the track sections of chosen route (in several lines). Output requirement then becomes the input for the infrastructure manager to be further processed in the process of allocation of railway infrastructure capacity.

**Fig. 1** Flow diagram of selection of a suitable route for planning of shipment

**4. Next steps research**

For the further steps in the research is used a graphic method, is similar to the UIC methodology. This is based on following:

- hypothesis that the capacity is relative, depending on a lot of aspects,
- obscure steps of using train path compression for one-track lines,
- each line section can be occupied by different amount of trains.

As we are trying to focus on a graphic method (geometrical relations), at first we need to solve a problem of the value, in which can we express a blocking space (or blocking stairs) of a train in the time-distance dimension. We are tending several forthcoming issues:

- to test the idea of Figure 2 in a simulation software (timetable stability),
- to propose the model of a new method for graphic detection of capacity,
- to draft an absolute addressing of blocked sections for this case,
- to make this usable for manual calculation as well as simulation outputs,
- to continue in the research by evolving next ideas.

**Fig. 2** Events of discount provision in terms of train path marketing

Figure 2 shows the basic states for supply of discount for railway undertakings based on their previous optional choice of the train path they would like to use. Basic train paths (black) are previously added in yearly timetable construction process or they are “ad hoc” paths, that are confirmed (or later paid) by railway undertakings and broke in by infrastructure manager. Red path means the theoretical middle distance path in the free space of time-distance traffic diagram. There are two basic situations that can be brightly used in marketing of infrastructure manager. If in that free space is possible to add more than one path (grey vs. green), than it is better to add the demanding path (green) as close to the basic path as possible and do not block the free space by only one inappropriate path in the middle. This can be achieved by offering discounts from the train path fees. Similar situation can relieve a timetable construction by using latitude setting for adding train paths. That means, that this paths are interrupted because of non-moving basic paths (cruising, preceding, etc.) and that is why this paths are easier inserted. Of course, this means longer and energy inefficient train drive that should be eliminated by discount again.

**5. Conclusion**

In the chain of transport services, especially in freight transport is increasingly coming to the fore system of “the pull” when determining and significant entity in the development of the transport market is the shipper, so the final element of the chain that determines the mode of its transport operations in the form of specified requirements for transport and by this basis is the requirement for transportation transmitted through the undertakings to the primary transport market and thus the infrastructure manager in the form of demand for railway infrastructure capacity. The common space of business is therefore railway infrastructure, which is provided to them based on their current license for operating on that infrastructure. However, to be fully maintained the principle of non-discriminatory access to the infrastructure for customers, the capacity (and its information) has to be properly identified and published by infrastructure manager, which is in its administration and for which is fully responsible. It is necessary to find a simple and transparent way to organize the resulting values into a usable format for marketing activities of infrastructure managers. This method may be used to:

- a detailed determination of railway infrastructure capacity,
- accurate allocation bottlenecks that reduce the capacity of rail infrastructure,
- continuous calculation of available capacity in the allocation of "ad hoc" paths,
• operational management and decision making at operational incidents,
• educational and research purposes.

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Abstract: The paper deals with analysis of the development of performances in freight rail transport in comparison with the development of gross domestic product as the main indicator measuring the economic performance of the country and on the other hand the performances in passenger rail transport in relation to the development of the average monthly wage as one of the most important indicator characterizing the living standard. The change in demand for transport services is analyzed with respect to the change in the price of the ticket in passenger rail transport and also the change in the price for transport of one tonne of goods in freight rail transport.

Keywords: MACRO-ECONOMIC INDICATORS, PASSENGER RAIL TRANSPORT, FREIGHT RAIL TRANSPORT, ELASTICITY

1. Introduction

The development of the economy in the country is influenced by the effective activity and cooperation of households, companies, the state and foreign countries. These four sectors are affected by each other. Consumers enter the market for their personal needs as a buyers, but also as a sellers. Their receipts from ownership of the production factors use for the purchase of goods, services, or savings. The companies produce goods, respectively offer the services, and come to the market for their sale. [1], [2].

The specific role in this cycle has a state, which should guarantee the appropriate conditions for companies but on the other hand also for the households, which are the main customers. [3].

Transport is one of the most important sectors of the economy. The share of transport in gross domestic product (GDP) is about 6% in Slovakia. According to the Statistical Office of Slovakia were employed almost 99 thousand people in transport sector in 2014 while 12.5% of this number were employed in railway transport. Therefore, it is relevant to examine the impact of economic indicators on the change of performances in rail passenger and freight transport.

2. Comparison of the transport performances and macroeconomic indicators

Economic situation of the country can be characterized by a number of basic indicators. The development of freight rail transport is compared in relation to the gross domestic product and the development of passenger rail transport depending on the average monthly wage.

2.1 Performances of freight rail transport in comparison with GDP

The most commonly used indicator characterizing the economic situation in the state is gross domestic product. It represents the value of final goods and services produced in the time period on the national territory by production factors owned by the citizens of the country or foreigners working in that country. It is that part of gross production which is intended for final consumption (not to the next production) and satisfies the final consumers. [4], [5].

Figure 1 shows the development of the GDP (in constant prices) and transport performances of freight rail transport (in tonne kilometers).

Fig. 1 The performances of freight rail transport in comparison with GDP

The gross domestic product of Slovakia regularly increased by more than one billion eur in the last period, however the most significant increase was between the years 2013 and 2014 (more than 2.5 billion eur) with the assumption of a continuation of this trend in the future.

The same development was also in the performances of freight rail transport in the analysis period when the increase between the years 2012 and 2014 was 1238 million tkm (almost 17%).

2.2 Performances of passenger rail transport in comparison with average monthly wage

Statistics indicate an average monthly level of wages of the employee for the whole economy of Slovakia, therefore include all sectors of economic activities with the exception of management or business receipts and also the receipts of their shareholders, military components, people on maternity and parental leave. The data are classified according to economic activity, geographical area, age groups or education.

Passenger kilometers (pkm) are the most appropriate measure for transport statistics and their comparison in passenger transport because an indicator the number of passengers can bring a high risk of double counting, especially in international transport.

Figure 2 shows the development of average gross monthly wage in the national economy in comparison with the performances of passenger rail transport.
The average monthly wage in Slovakia for the last five years reflects the development of the gross domestic product. An increase of almost 20€ is recorded each year, while the most significant increase was between the last two analyzed years (up to 34€). According to different economic situations of regions is the average monthly wage regarded as substandard in certain areas of Slovakia in relation to others where the wage is much lower.

The development of performances in passenger rail transport (expressed in passenger kilometers) showed a similar trend in the years 2010 - 2014, however the last two months of 2014 were significantly affected by the introduction of free transport for certain groups of passengers.

3. Price elasticity of demand for services in passenger and freight rail transport

Price elasticity of demand (PED) shows the relationship between price and quantity demanded and provides a precise calculation of the effect of a change in price on quantity demanded. The degree of response of quantity demanded to a change in price can vary considerably. The key benchmark for measuring elasticity is whether the coefficient is greater or less than proportionate. If quantity demanded changes proportionately, then the value of PED is 1, which is called ‘unit elasticity’.

PED can also be:
- Less than one, which means PED is inelastic.
- Greater than one, which is elastic.
- Zero (0), which is perfectly inelastic.
- Infinite (∞), which is perfectly elastic.

PED on a linear demand curve will fall continuously as the curve slopes downwards, moving from left to right. PED = 1 at the midpoint of a linear demand curve.

3.1. Price elasticity of demand for services in passenger rail transport

Cenová elasticita dopytu po doprave sa vypočíta ako pomer zmeny množstva vyjadreného najčastejšie prepravným výkonom a zmeny ceny. Pri cenovej elasticite sme skúmali závislosť výkonov železničnej osobnej dopravy od zmeny ceny cestovného lístka. Vzhľadom na to, že cestovné je v železničnej doprave stanovené pre jednotlivé tarifné pásmá, ktorých veľkosť nie je rovnaká, uskutočnili sme prepočet výšky cestovného tak, aby zohľadňovala rôzne vzdialenosti jednotlivých tarifných pásmí (Fig. 3).

Fig. 3 Calculation of average price for transport

We calculated the average price as the average of the prices for one passenger kilometer in the individual tariff zones. Prices in passenger rail transport are regulated by the state therefore does not change often. Figure 4 shows the price elasticity of demand for passenger rail transport in the years 2011-2014.

The coefficient of price elasticity is less than one in all cases, what means that the demand for services of public passenger rail transport is inelastic - it is not significantly affected by the change of the ticket price.

3.2. Price elasticity of demand for services in freight rail transport

Price elasticity of freight rail transport is focused on the dependence of the change in the rates for the transport of 1 tonne of goods from the transport performances. The rate for transport of 1 tonne of goods is set out for each tariff zone as well as in passenger transport (Fig. 5). The conversion is processed for each analyzed year, where the result is the average price for one tonne kilometre. The list of rates is referred in Tariff for freight rail transport – TR1.
We calculated the average price as the average of the prices for one tonne kilometer in the individual tariff zones. The used list of rates is for railway wagons, which are owned by the carrier.

Price elasticity of demand for freight rail transport in the years 2013 and 2014 is shown in Figure 6.

The coefficient of price elasticity is more than one in both cases, what means that the demand for services of freight rail transport is elastic. Performances of freight rail transport increased every year, although the rate for transport of one tonne of goods also increased also. This fact could be affected by increased interest in intermodal transport.

### 4. Conclusion

The good economic situation and living standards is also reflected positively on the increasing demand for services in passenger and freight rail transport. The increase in the number of passengers carried in the period was recorded each year during the period, even between years 2010 and 2014 it was more than 11%. Despite this fact, we are concluded on the basis of price elasticity that demand remains inelastic to change of price in passenger rail transport. This trend indicates that the price in passenger rail transport is not the most important factor for passengers in deciding on the choice of type of transport. The situation in freight transport was similar when the increase of performances was observed in comparing the years 2012 and 2014, despite the changes in the rates for transport of one tonne of goods.
DESIGN FOR ACTIVE FLUTTER SUPPRESSION AND MODEL VERIFICATION

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Abstract: The article deals with developing a mathematical model of non-rigid aircraft lifting surface with control surface controlled by pilot and supplementary control surface driven by control law. The purpose of this model is to determine if such as concept of control surface and supplementary control surface can be used for active flutter suppression on an aircraft structure. The supplementary control surface is placed next to the control surface at outboard side. The lifting surface is representing by an airfoil placed at 70% of a wing span.

A structural model is developed by means of Lagrange differential equations of second kind. Theodorsen model of thin oscillation airfoil with control surface is used for unsteady aerodynamic. Duhamel’s integral of Wagner function is carried out for transformation of unsteady aerodynamic to a time domain. The mathematical model is present in state space representation. There is exemplification of the critical flutter velocity calculation and a dynamical response of the structure. The supplementary control surface for flutter suppression with simplified model is added. Closed-loop feedback control system is formed and a several control laws are presents. The verification of open-loop model is done on behalf of the critical flutter speed comparison with FEM software for flutter analysis MSC Nastran and flutter analysis program developed at CTU in Prague. The article also presents work on an experimental verification of the open-loop model in aerodynamic tunnel.

Keywords: FLUTTER, ACTIVE SUPPRESSION, STATE SPACE REPRESENTATION, NASTRAN, UNSTEADY AERODYNAMIC

1. Introduction

Flutter is self-excited oscillation of aircraft structure which leads to complete destruction of an aircraft. It is defined by so called flutter velocity. Above this velocity a damping is positive so amplitude of oscillation increases within each cycle. Each type of aircraft has to prove that is flutter free up 1,2Design Velocity of an airplane, before a serial production start. There are several way how to move the flutter to higher velocity. The most common way is by mass balancing or by aerodynamic balancing of control surface. There are also some other ways [1]. A research in area of increasing the flutter velocity of an aircraft structure is aim at active suppression by a control surface to damp an oscillation above the flutter velocity [2] [3]. This paper is aim to verify a new approach of flutter suppression. This approach lies in splitting the control surface in two parts. The inboard part of control surface remains as usual controlled by pilot to maneuver with aircraft. The outboard part of control surface will be driven by control law to suppress undamped oscillation of an airfoil.

2. Structural model

The structural model of the airfoil was developed on behalf of Lagrange differential equations of second kind (LDE.II) for three degree of freedom in plunge, pitch and control surface rotation see the model at figure 1. An opposite orientation of coordination system is given by used aerodynamic model.

\[ [M_q] + [T_q](q) + [K_q](q) = [F_q] \]

where \( M_q \) - structural mass matrix, \( T_q \) - structural damping matrix, \( K_q \) - structural stiffness matrix, \( F_q \) - vector of unsteady aerodynamic forces and \( q \) - deformation vector are defined as follow:

\[ q = [y \quad \varphi \quad \delta]^T \]

\[ [M_q] = \begin{bmatrix} m & S_{\delta 0} & S_{\delta} \\ S_{\delta 0} & J & (x_k - x \delta) S_{\delta} + J \delta \\ (x_k - x \delta) S_{\delta} + J \delta & J \delta & I \end{bmatrix} \]

\[ [T_q] = \begin{bmatrix} 0 & b_y & 0 \\ b_y & 0 & 0 \\ 0 & 0 & b_{\varphi} \end{bmatrix}, \quad [K_q] = \begin{bmatrix} K_y & 0 \\ 0 & K_{\varphi} \end{bmatrix}, \quad [F_q] = \begin{bmatrix} Y \\ M \\ M_{\delta} \end{bmatrix} \]

where \( m \) – is mass, \( S \) – static moment, \( J \) – moment of inertia, \( x_k \) – hinge axis, \( x_{\delta} \) – elastic axis, \( b \) – viscous damping, \( K \) – stiffness, \( y \) – plunge, \( \varphi \) – pitch, \( \delta \) – control surface rotation, \( Y \) – lift, \( M \) – aerodynamic moment, \( M_{\delta} \) – hinge moment. Because this study is aimed on airfoil investigation without influence of wing span the units of mass matrix and aerodynamic vector are specific.

3. Unsteady aerodynamic

The mathematical model of airfoil unsteady aerodynamic forces is based on Theodorsen thin harmonically oscillation airfoil [4]. This model is commonly used for flutter analysis because consider not only aerodynamic forces as a function of free stream velocity but also take in account a change of lift and moment due to airflow pitch, control and surface rotation.

\[ Y = -\rho c^2 \left[ \pi \varphi \delta \psi \right] + \pi \delta^2 \left[ \varphi \psi + c \delta (\varphi + \delta \psi) \right] - 2\pi \rho \varphi \psi c \left[ \psi \delta + c \delta \psi \right] \]

\[ M = -\rho c^2 \left[ \pi \varphi \delta \psi \right] + \pi \delta^2 \left[ \varphi \psi + c \delta (\varphi + \delta \psi) \right] + \pi \psi \delta \left[ \varphi \psi + c \delta (\varphi + \delta \psi) \right] \]

\[ \rho c^2 \left[ \pi \varphi \delta \psi \right] + \pi \delta^2 \left[ \varphi \psi + c \delta (\varphi + \delta \psi) \right] \]

where \( \rho \) – density, \( c \) – semichord, \( v \) – air stream velocity, \( T_{11} \) – Theodorsen constant, \( C_{12} \) – Theodorsen function. The first group of terms of equations (2) – (4) are called non-circulatory or apparent mass component and they account for inertia of the fluid. The second groups of term products characterized by Theodorsen function \( C(k) \) are called circulatory components and they take in account an influence of the shed wake vorticity. Thus the vector of unsteady aerodynamic forces can be written as

Fig. 1 Model of the airfoil.
The integral can be written in state space form as \( (12) \):

\[
\phi(t) = \int_{0}^{t} e^{-A(t-s)} [B(t-s)]_1(s) ds
\]

An explicit expression for the Wagner function does not exist so for practical evaluation of Duhamel’s integral of Wagner function a subsonic approximation by second order exponentials function is used given by \( [5] \):

\[
\psi(t) = \Phi(t) = 1 - a_1 e^{(-\rho v t/c)} - a_2 e^{(-\rho v t/c)}
\]

\[
a_1 = 0.2048 \quad a_2 = 0.2952 \quad \beta_1 = 0.5577 \quad \beta_2 = 0.3330
\]

Thanks this approximation the solutions of the Duhamel’s integral can be written in state space form as \( [6] \):

\[
\begin{bmatrix} l_1 \\ l_2 \end{bmatrix} = \begin{bmatrix} 0 & 1 \\ -\beta_1 l_2 & -\beta_2 l_2 \end{bmatrix} \begin{bmatrix} l_1 \\ l_2 \end{bmatrix} + \begin{bmatrix} 0 \\ 1 \end{bmatrix} \psi(t)
\]

or

\[
\begin{bmatrix} \dot{l} \\ \dot{y} \end{bmatrix} = \begin{bmatrix} a_1 \beta_1 & a_2 \beta_2 \end{bmatrix} \begin{bmatrix} l_1 \\ l_2 \end{bmatrix} + \begin{bmatrix} 0 \\ 1 \end{bmatrix} \psi(t)
\]

4. Open-loop model

The open-loop model also called as a system is formed in state space representation by merging the structural model with non-circulatory part of aerodynamic load including a positive feedback formed by circulatory part of aerodynamic loads. The output of the model has to be in form of position, velocity and acceleration for all degree of freedom, for purpose of control law. A schema for model of System is presented at Figure 2.

The structural and aerodynamic model is given as follow:

\[
[M_a] \ddot{q} + [T_a] \dot{q} + [K_a] q = \begin{bmatrix} 0 \\ 0 \end{bmatrix}
\]

by transformation to state space representation we obtain

\[
\begin{bmatrix} \dot{x}_1 \\ \dot{x}_2 \end{bmatrix} = \begin{bmatrix} [M_t]^{-1} [K] & 1 \\ 0 \end{bmatrix} \begin{bmatrix} \dot{x}_1 \\ \dot{x}_2 \end{bmatrix} + \begin{bmatrix} 0 \\ 0 \end{bmatrix} [F_e]
\]

or

\[
\begin{bmatrix} y_1 \\ y_2 \end{bmatrix} = \begin{bmatrix} [C_t] \dot{y}_1 + [D_t] u \end{bmatrix}
\]

where \( u \) is the input vector.

The input to Duhamel’s integral represent by Eq. (13) and (14), has to be cut of about second derivation of DOF because they are not needed here. So the following reduction matrix is added in front of Duhamel’s integral:

\[
[K] = \begin{bmatrix} I & 0 \\ 0 & I \end{bmatrix}
\]

We can obtain the open loop model of system by merging equations (13), (14), (18), (19), (20), (21) according the schema on figure 2. Then the state space representation of system is

\[
\begin{bmatrix} \dot{x}_{sys} \\ y_{sys} \end{bmatrix} = \begin{bmatrix} A_{sys} & B_{sys} \\ C_{sys} & D_{sys} \end{bmatrix} \begin{bmatrix} x_{sys} \\ u_{sys} \end{bmatrix}
\]

with state vector \( x_{sys} = [x \ T]^T \) and \( y_{sys} = [\psi \ \dot{\psi} \ \ddot{\psi} \ \delta \ \dot{\delta} \ \ddot{\delta}]^T \), input vector \( u_{sys} = [0 \ 0 \ 0]^T \) and output vector \( y_{sys} = [\psi \ \dot{\psi} \ \ddot{\psi} \ \delta \ \dot{\delta} \ \ddot{\delta}]^T \).

Fig. 2 Schema of the closed-loop model

5. Closed-loop model

There was added the supplementary control surface with controller to the feedback loop of the system. The purpose is an active damping of an airfoil oscillation and shifting the flutter velocity to higher speeds. The supplementary control surface is intangible and non-rigid with aerodynamic load generated on behalf of supplementary control surface rotation \( \Theta \) only. The transport delays, servo transfer function or sensor noise is not taken in account. This paper is focused on testing if a possibility of the new
concept of flutter suppression is possible at all. The supplementary control surface hinge moment \( M_\Theta \) is defined as

\[
M_\Theta = \frac{1}{2} \rho v^2 b_c \theta (x_k - x_0) c^2 \theta
\]

where \( b_c = \frac{\alpha_k}{2c} \), \( c^0 = 2 \left(a \cos \left(1 - 2b_c \right) + 2b_c \right) \)

and will extend equation (5) as follow

\[
\{ F_0 \} = \begin{pmatrix} Y_{NC} M_{\Theta}^NC + M_{\Theta} M_{\Theta}^N \end{pmatrix}
\]

The angle of rotation \( \Theta \) will be driven by controller on behalf of the control variable chosen from system output \( y_{sys} \). There has to be also considered another two gain matrix and state space model for regulator to form the closed loop model. The first matrix is for picking a control variable on behalf of system output \( \{ K_4 \} \)

\[
\{ K_4 \} = [R_1 \ R_2 \ R_3 \ R_4 \ R_5 \ R_6 \ R_7 \ R_8 \ R_9]
\]

Where all constant \( R_i \) are zero except the one representing control variable which will be set \( \pm 1 \) where positive or negative sign is for generating hinge moment in phase or antiphase according to control variable.

The controller transfer function is transform to state space model and added to closed-loop in flowing form

\[
\{ x_{RS} \} = \begin{pmatrix} A_{sys} & B_{sys} K_4 C_{sys} \end{pmatrix} \{ x_{RS} \} + [0] \{ u_{RS} \}
\]

with input as a control variable and output as a supplementary control surface rotation angle \( y_R = \Theta \). The second gain matrix

\[
\{ K_4 \} = \begin{pmatrix} 1 \ \ 0 \ \ 0 \ \ 0 \ \ 0 \ \ 0 \ \ 0 \ \ 0 \ \ 0 \end{pmatrix}
\]

Generate the supplementary control surface hinge moment on behalf of rotation angle \( \Theta \), given by controller. Then the closed-loop model given in figure 2 is defined as follow:

\[
\{ x_{RS} \} = \begin{pmatrix} A_{sys} & B_{sys} K_4 C_{sys} \end{pmatrix} \{ x_{RS} \} + [0] \{ u_{RS} \}
\]

or

\[
\{ x_{RS} \} = [A_{RS}] \{ x_{RS} \}
\]

\[
\{ y_{RS} \} = \{ y_{sys} \}
\]

with state vector \( \{ x_{RS} \} = \{ y \ \ \varphi \ \delta \ \dot{y} \ \dot{\varphi} \ \dot{\delta} \ \l_1 \ l_2 \ z_{R1} \ z_{R2} \}^T \)

input vector \( \{ u_{RS} \} = [0 \ \ 0 \ \ 0]^T \)

and output vector \( \{ y_{RS} \} = [y \ \varphi \ \delta \ \dot{y} \ \dot{\varphi} \ \dot{\delta} \ l_1 \ l_2 \ z_{R1} \ z_{R2}]^T \).

6. Flutter velocity and dynamical response of the model

The velocity of flutter can be obtained by root locus. This is eigen-value analysis of system matrix by:

\[
det \left\{ A_{RS} - \lambda I \right\} = 0.
\]

where \( \lambda \) is eigenvalue. Each eigen-mode is representing by complex conjugate pair of \( \lambda = \lambda + i\omega \). The analysis is perform according to the selected variable i.e.: velocity, controller gain and so on. Each \( \lambda \) representing eigen-mode is given as

\[
\lambda = \omega (y + i\).
\]

where \( \omega \) – frequency, \( y \) – damping, \( i \) - complex unit. The lag root obtained by Duhamel’s integral has real part only and is defined as

\[
\lambda = \gamma_{LAG} \equiv \frac{\gamma_{LAG}}{i}.
\]

The flutter velocity is found from complex plot of \( \lambda = \phi(v) \) where any of eigenvalue cross imaginary axis of complex plot i.e. \( \text{Re}(\lambda) = 0 \). The dynamical response of model for given initial condition can be obtained by solving Eq. (33) and (34).

7. Verification of open-loop model

The verification process of system was done for the following input data which are based on data collected during flutter analysis of those airplanes: Sparrow [7], Phoneix [8], Via [9], and Magic [10]. Input data: \( c = 0.475 \text{m}; x_e = -0.238 \text{m}; x_k = -0.249 \text{m}; \) \( m = 0.086 \text{kg/m}; S_0 = 0.856 \text{kg m/\text{s}}^2; \ l_1 = 0.618 \text{kg m/\text{s}}^2; \ S_0 = 0.046 \text{kg m/\text{s}}^2; \ f_p = 25 \text{Hz}; \ f_0 = 47.2 \text{Hz}; \ g_y = 981 \text{m/s}^2; \rho = 1.225 \text{kg/m}^3; \) where \( g \) – structural damping.

The flutter velocity of open-loop model determined by means of root locus Eq. (35) is \( V_{FL} = 83.3 \text{m/s} \) for plunge. The verification of model derived in this paper was done on behalf of flutter velocity comparison with one obtained by FEM software for flutter analysis MSC.Nastran, flutter analysis program developed at CTU in Prague [11] and author’s K-Method [12] for the same input data. The results are listed in table 1.

<table>
<thead>
<tr>
<th>Method</th>
<th>MSC.Nastran</th>
<th>K-Method</th>
<th>CTU in Prague</th>
<th>Root locus</th>
</tr>
</thead>
<tbody>
<tr>
<td>( V_{FL} ) [m/s]</td>
<td>84.1 m/s</td>
<td>79.7 m/s</td>
<td>86.5 m/s</td>
<td>83.3 m/s</td>
</tr>
</tbody>
</table>

Table 1: Comparison of flutter velocity for different methods.

8. Results of closed-loop analysis

There were tested several type of control laws but only those presented here have some influence on flutter velocity. The main request for control law is to damped an airfoil oscillation to remain in steady flight above open-loop flutter velocity. The controller has to be also design so that does not respond on pilot control inputs for maneuvering of airplane or deformation of aircraft structure caused by turn maneuver. Practically this means not to respond in low frequencies, approximately up to 2Hz. The following table 2 presents the results of analyzed control model and obtained flutter velocities. The table also shows an angular rotation of supplementary control surface as a verification criterion of design correctness.

The results given by Gain controller doesn’t satisfy the request not to act in low frequencies. It was used for initial preview into the behavior of the model and for check if the idea of flutter suppression by supplementary control surface is possible. It also show that has to be investigate a positive feedback of controller, given by non-zero constant \( R_i \), because the negative feedback caused decreasing the critical flutter velocity in some cases.

The highest closed-loop flutter velocity is for test case ID 11 and 20 with control variable of \( \phi \). For this cases the flutter will not occur at all, the critical speed represents a torsional divergence velocity because \( \text{Re}(\lambda) = 0 \) for the lag root. But due the unacceptable high rotation of supplementary control surface at low velocity this options can’t be used.

The best trade-off between flutter velocity and rotation \( \Theta \) is test case ID 31 for Band pass controller with \( \dot{y} \) control variable where flutter velocity is 187m/s and \( \Theta = 20^\circ \) for high velocity and \( 40^\circ \) for low velocity. The root locus and dynamical response of test case ID 31 is on figure 3.
9. Experiment

There was carried out some works on experimental verification of this concept of flutter suppression. A very first model for study of flutter mechanism was built from thin plastic plate. A car was used for execution of an experiment and the flutter occurs at velocity about 16.6m/s.

There was build another experimental model for testing in aerodynamic wind tunnel at CTU in Prague. It was made from balsa and glass fiber composite with reinforcement beam flange from carbon composite. The model had two control surface and one supplementary control surface in the middle driven by controller. Figure 4 show ground vibration test of this model for determining the model input data for mathematical model. This model was destroyed during the experimental determination of open-loop flutter velocity. The problem was to high velocity increment. Construction of another experimental model is in progress.

![Fig. 3 Root locus and dynamical response for closed-loop test case ID 31](image)

10. Conclusion

On behalf of mathematical model derived here the idea of flutter suppression by supplementary control surface can be realized. The analytical calculation shows that the flutter was not suppressed but move to higher velocity only. For the best test cases the velocity was shifted about 225%. Future work on mathematical model will be aimed on inclusion of wing span, unsteady hinge moment of supplementary control surface. There have to be also considered an actuator dynamic and its time delay, sensor location and their noise.

References


![Fig. 4 Ground vibration test of experimental model](image)

Table 2: Results from closed-loop analysis

<table>
<thead>
<tr>
<th>ID</th>
<th>Controller transfer function</th>
<th>Control variable</th>
<th>Non-zero constant R</th>
<th>Constants of controller</th>
<th>Vₚ [m/s]</th>
<th>ξ₁ [°]</th>
<th>ξ₂ [°]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Gain controller Tₑ = G</td>
<td>y</td>
<td>R₁ = ±1</td>
<td>G=16</td>
<td>208</td>
<td>±30°</td>
<td>±40°</td>
</tr>
<tr>
<td>2</td>
<td></td>
<td>φ</td>
<td>R₂ = ±1</td>
<td>G=24</td>
<td>159</td>
<td>±15°</td>
<td>±40°</td>
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<tr>
<td>3</td>
<td></td>
<td>δ</td>
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<td>G=4.5</td>
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<td>±60°</td>
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<tr>
<td>4</td>
<td></td>
<td>γ</td>
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<td>G=3.5</td>
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<td>±100°</td>
<td>±100°</td>
</tr>
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<td>φ</td>
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<td>G=0.2</td>
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<td>±75°</td>
</tr>
<tr>
<td>6</td>
<td></td>
<td>δ</td>
<td>R₂ = ±1</td>
<td>G=0.018</td>
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<td>±30°</td>
<td>±30°</td>
</tr>
<tr>
<td>7</td>
<td></td>
<td>y</td>
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<td>G=3.00; n₁=5; n₂=31</td>
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<td>±50°</td>
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<tr>
<td>8</td>
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<td>φ</td>
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</tr>
<tr>
<td>9</td>
<td></td>
<td>δ</td>
<td>R₂ = ±1</td>
<td>G=0.20; n₁=5; n₂=500</td>
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<td>±40°</td>
<td>±60°</td>
</tr>
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<td>10</td>
<td></td>
<td>y</td>
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<td>±50°</td>
<td>±60°</td>
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<tr>
<td></td>
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<td>φ</td>
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<td>±80°</td>
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<td>γ</td>
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<td>±20°</td>
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<td>±50°</td>
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<tr>
<td>16</td>
<td></td>
<td>φ</td>
<td>R₂ = ±1</td>
<td>G=30.0; n₁=50</td>
<td>94</td>
<td>±60°</td>
<td>±50°</td>
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<tr>
<td>17</td>
<td></td>
<td>δ</td>
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<td>±50°</td>
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<td>φ</td>
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</table>
DRIVER HEALTH AND TRAFFIC SAFETY ON THE TOURIST ROUTES

1. Introduction

The driver is an essential link of the “driver-car-road-environment” system, its operator particularly [1]. The driver receives basic information on road conditions through observation. A great deal of information and its nature, which is constantly changing, for example when driving in mountain environments, does not allow for its timely absorbing and processing, and consequently for making the right decision. This process is especially pronounced at psycho-physiological changes in driver’s organism.

The system of organizing traffic safety health-care envisages a complex of the actions, which includes both medical prevention of road accidents and provision of medical assistance after these accidents.

One of the most fundamental factors of traffic safety is the health of the driver.

A person’s organism has specific requirements when driving in mountain environments. At an altitude of more than three kilometers, a person has trouble breathing because of oxygen deficiency. When driving for long period of time in mountain environments, oxygen deficiency leads us to headaches, reduction in sight and hearing, disorders of digestive system and a number of undesirable effects. All this leads to rapid and severe fatigue of driver.

Travelling in tourist destinations existing in mountains gives a great pleasure to many car enthusiasts (Pic. 1). Of course, overcoming the breathtaking ascents and downhill sections on the twisting mountain roads is much more interesting, as well as enjoying views over the beautiful mountains.

2. Preconditions and means for resolving the problem

Before starting driving on the mountain roads (Pic. 1), the tourist-drivers have to make an in-depth analysis of all those difficulties, which they could face when driving on the mountain roads, as well as it is important to continuously improve skills and moral-psychological qualities of driver.

To that end, the driver has to take into account several recommendations as follows:

To accurately assess your own capacities. For example, if the driver of another vehicle outpaces you, do not step on the gas and do not compete with him! Let him outpace you, and remember that in such case the increased speed is just a direct path to the hospital. Alternately, at best, you may meet this driver on the pass being too tired and nervous, thinking about what to do with this damaged car, since studies of transport-tourist routes road infrastructure of the Imereti region have shown that service centers are mostly situated only in cities and sometimes in densely populated areas.

Abstract: Based on the results of studies carried out in the field conditions, there are given some recommendations for drivers of vehicles moving on tourist routes with mountain road conditions, which will reduce likelihood of traffic accidents for them.

The paper also describes the results of studies carried out in accordance with the methodology of health check of drivers of vehicles moving on the tourist routes. As the evaluation parameters, there have been used several indexes as follows: the harmony of driver’s body structure and his/her physical development was evaluated by Quetelet-2index; death-birth and power indexes were used for evaluation of the power of the apparatus of external respiration and hand muscle strength, accordingly; the nature of nervous vegetal regulation was determined by means of systolic and diastolic pressure and chest compression frequency of hemodynamics indices; for evaluating the adaptation level by Kerdo index and Robinson index there was used the screening procedure.

Reliability of driver depends not only on his/her qualification and experience, but the state of his/her health and ability to work are of high importance as well.

KEY WORDS: VEHICLE’S DRIVER; TOURIST ROUTE; VEGETAL REGULATION INDEX; HEMODYNAMICS INDICES; SYSTOLIC AND DIASTORIC PRESSURE; CHEST COMPRESSION FREQUENCY; ADAPTIONAL LEVEL.
During the period of carrying out anthropometric screening, by meeting certain requirements, the period of three years.

Parameters, the observations were carried out on the tourist routes in poor health, especially the drivers, must comply with safety regulations. Some people are adapting rapidly to oxygen deficiency, environments, may even have the mild or severe forms of mountain sickness. So, those people, who are physically unprepared and are not accustomed to mountain environments have been subjected to a medical examination on the parameters, which affect hemodynamic indices (Table 1).

Table 1

<table>
<thead>
<tr>
<th>Year</th>
<th>Heart Ratio (f)</th>
<th>Systolic Blood Pressure (P&lt;sub&gt;1&lt;/sub&gt;)</th>
</tr>
</thead>
<tbody>
<tr>
<td>2014</td>
<td>72,1±2,50</td>
<td>118,1±4,48</td>
</tr>
<tr>
<td>2015</td>
<td>71,4±2,70</td>
<td>119,5±2,27</td>
</tr>
<tr>
<td>2016</td>
<td>71,1±2,60</td>
<td>121,3±2,49</td>
</tr>
</tbody>
</table>

Comparative Analysis ofDriver’s Hemodynamic Indices

The dangerous turn of the Imereti region’s transport-tourist route

When driving on the mountain routes, the most important factor is the strict observance of traffic rules by drivers of vehicles, as well as their health check system.

During intensive traffic flow in complicated mountain environments, high demands are placed to the state of health of a driver. In order to define driver’s health end-points, he is needed to have a medical check-up. Proper medical check-ups before setting out on the journeys represent an essential link in the prevention of road traffic accidents.

Any person, who is not accustomed to mountain environments, may even have the mild or severe forms of mountain sickness. So, those people, who are physically unprepared and are in poor health, especially the drivers, must comply with safety regulations. Some people are adapting rapidly to oxygen deficiency, but some of them need a long period of adjustment.

Health end-points, which may represent the basis for refusing the vehicle’s driver journey, are as follows:

- The presence of severe symptoms or complications of chronic diseases;
- Increasing or reducing heart rate and variation of the upper and lower rates of blood pressure that is typical of all tested drivers;
- The driver is under the influence of alcohol or other preparations (narcotic drugs and psychotropic preparations), which disturb their functional status.

For a comparative analysis of morphofunctional parameters, the observations were carried out on the tourist routes on the vehicle’s drivers of different age and physical abilities for the period of three years. During the period of carrying out anthropometric screening, by meeting certain requirements, the following have been measured: driver’s height, cm; weight, kg; chest circumference S, cm; lung capacity, V; right hand size, A, cm; left hand size, B, cm; systolic pressure P<sub>1</sub>, mm, Mm/hg; diastolic pressure P<sub>2</sub>, mm, Mm/hg, heart ratio, f, beats per minute.

As the evaluation parameter there also has been used the body-weight index, or Quetelet-2 index, which is calculated by the formula:

$$I_m = \frac{M}{h^2}$$

The body-weight index allows for assessing the level of compliance of the body-weight with its height, in other words the harmony of its physical development. The body-weight index evaluates indirectly if the mass of a person is sufficient, normal or excessive. According to foreign medical scientists, variation of the body-weight index within 25-27 kg/m<sup>2</sup> for men is ideal. As the evaluation parameter there also are used: the birth-death ratio K = V/M, which characterizes the power of the apparatus of external respiration, but the power index I<sub>p</sub> = A/M, determines the hand muscle strength.

Of high importance for the drivers of tourist buses in mountain environments is calculation of Kerdo index [2], or the index of nature of nervous vegetal regulation by the formula:

$$I_k = 1 - \frac{P_2}{f}$$

For quantitative evaluation of energy potential of human organism, there is used the Robinson index [4], which is calculated by the formula:

$$I_r = \frac{f \cdot P_1}{100} + \frac{0.011 \cdot W}{2} + 0.014 \cdot P_1 + 0.014 \cdot W + 0.009 \cdot M - 0.004 \cdot C + 0.009 \cdot h + 0.273,$$

where, W- age (years); C- gender (male -, female – 2).

The adaptation level of human organism to the environment “health level” is classified as follows: health first level – the state is optimal, satisfactory adaptation; health second level – the adaptation mechanisms are tense; health third level – unsatisfactory adaptation level, during which there are inconsistencies with the organism functioning mechanism; health fourth level – interruption of adaptation – status of patient before or during disease [3].

Hemodynamic indices represent the criterion of cardio-vascular system, which is influenced by the environmental factors. Thus, the drivers of buses moving on the tourist-transport routes in mountain environments have been subjected to a medical examination on the parameters, which affection hemodynamic indices (Table 1).
It has been established that during the entire period of research, heart ratio has a narrow range of frequencies, which indicates stability and tolerance of organism. Systolic and diastolic pressures tend to increase in line with the rise in age, especially among older drivers.

<table>
<thead>
<tr>
<th>Driver</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
</tr>
</thead>
<tbody>
<tr>
<td>Queletet-2 index</td>
<td>22,9</td>
<td>25,9</td>
<td>25,4</td>
<td>28,6</td>
<td>26,1</td>
</tr>
<tr>
<td>Assessment</td>
<td>1</td>
<td>2</td>
<td>3</td>
<td>2</td>
<td>2</td>
</tr>
<tr>
<td>Kerdo index</td>
<td>0,04</td>
<td>0,04</td>
<td>-0,13</td>
<td>0,01</td>
<td>-0,02</td>
</tr>
<tr>
<td>Assessment</td>
<td>2</td>
<td>2</td>
<td>3</td>
<td>2</td>
<td>2</td>
</tr>
<tr>
<td>Robinson index</td>
<td>85,3</td>
<td>111,3</td>
<td>82,1</td>
<td>109,8</td>
<td>109</td>
</tr>
<tr>
<td>Assessment</td>
<td>2</td>
<td>3</td>
<td>2</td>
<td>3</td>
<td>3</td>
</tr>
<tr>
<td>Adaptation level</td>
<td>2,35</td>
<td>2,94</td>
<td>2,21</td>
<td>3,19</td>
<td>3,09</td>
</tr>
<tr>
<td>Health level</td>
<td>1</td>
<td>2</td>
<td>1</td>
<td>3</td>
<td>2</td>
</tr>
</tbody>
</table>

Note: 1- below the norm; 2- normal value; 3- above the norm.

By Queletet-2 index, it has been established that the drives have a harmonic ratio of the body height and weight (Table 2), the only exception is one tested patient (Im ≈ 28,6). According to this parameter, he has excess weight, and another one has insufficient mass (Im = 22,9). So, as it is believed by psychologists, the body dysmorphic disorder may have a negative effect on human psychological health: impairment in self-assessment ability, causing depression.

The level of nervous vegetal system effect on blood vessels was assessed by Kerdo index (Table 2). The values of this index are within normal physiological parameters from 0,1 to -0,1. In addition to the driver, who had initial vegetation tonus with exuberant sympathetic effect (-0,13), one more driver has been identified with vegetotony tendency (-0,02).

For describing the quality of the exchange-energy processes occurring in the organism and myocardial oxygen consumption, there has been used the Robinson index. The upper (111,3) and lower (82,1) values of Robinson index confirm exuberance of sympathetic or para-sympathetic vegetation nervous system, but two of them disclosed the signs of cardio-vascular system regulation disturbance (109 and 109,8), and one of them had thecardio-vascular system regulation disturbance directly.

By the ratio of health level, the has been determined the adaptation level. The lower is the value of the adaptation level parameters, the higher is health level. In general, the drivers disclosed the positive result of health assessment and adaptation capacities (Table 2): satisfactory adaptation (first health level) was disclosed by 2 drivers, in other words, their organisms have high functional capacities; 2 drivers belonged to the functional stress group (second health level; and 1 driver disclosed unsatisfactory level of adaptation.

3. Conclusion

Good characteristics of nervous vegetal regulation indicate the optimal level of functional reserve of regulation and self-regulation mechanisms, which ensures adaptation to the ambient environmental factors.

The rivers of buses moving on the tourist-transport routes are needed to have medical check-ups on the parameters, which affect hemodynamic indices.

Thus and so, Reliability of driver depends not only on his/her qualification and experience, but the state of his/her health and ability to work are of high importance as well.

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TRAFFIC RISK MAPPING – RISK DISTRIBUTION AMONG THE MUNICIPALITIES IN MONTENEGRO

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Abstract: Accident analysis plays an important part in the strategy to reduce road accidents. To significantly reduce traffic fatalities and serious injuries on public roads, there is need to review the characteristics of motor vehicle accidents and identify the hidden patterns behind the accidents. Identifying traffic accident concentration area is important for road safety improvements. Determining the risk rate in road traffic and its spatial distribution may indicate the areas that are more at risk in traffic in proportion to others. Beside, such analysis can identify differences in the nature of the problem among spatial units on the territory of a country. An analysis of spatial traffic risk distribution among the municipalities on the territory of the Montenegro has been conducted. The results of the study indicate that there is a significant difference in the amount of traffic risk among the municipalities.

Keywords: TRAFFIC SAFETY, MOTOR VEHICLE ACCIDENTS, RISK MAPPING

1. Introduction

In Montenegro in the last ten years, the number of persons killed in road traffic is reduced. However, it is still not enough to be compared with the developed countries of the European Union. According to EU statistics, the rate in terms of fatalities per 100,000 populations in Montenegro in 2012 was 29.6; while in EU countries this indicator varies from 2.5 to 18.7, /1/. As well, Montenegro’s road network is high risk, including those segments with a high traffic volume, /2/. Many road crashes are preventable and history provides evidence that the right intervention can make a significant positive impact. In order to make progress in road safety, comprehensive data collection and in-depth data analysis are essential in terms of designing effective safety strategies, setting challenging targets, determining intervention priorities and monitoring program effectiveness. In May 2011, the United Nations launched a Decade of Action for Road Safety. The goal of the Decade (2011–2020) is to stabilize and reduce the increasing trend in road traffic fatalities, saving an estimated 5 million lives over the period. This report builds on the 2009 report, and provides additional data in a number of important areas. It is supported by a Global Plan for Road Safety, /3/. In Montenegro, National Coordination Board was established to monitor road safety parameters. In addition, government adopted Strategy for efficient traffic (2008), /4/, Strategy for improvement of road traffic safety 2010-2019, /5/, and Action plan for Strategy for improvement of road traffic safety implementation (2011), /6/. The main goal of this strategy is to reduce number of fatalities by 30% and injuries by 20% before 2014 (compared to 2007). The long-term goal is to reduce these figures by 50% and 30% respectively by 2019, compared to the same year.

To significantly reduce traffic fatalities and serious injuries on public roads, we need to review the characteristics of motor vehicle accidents and identify the hidden patterns behind the accidents’ records, referring mainly to the actual knowledge contained in the collision data rather than the raw data records themselves. Thus, identifying traffic accident concentration area is important for road safety improvements. The main objective of road safety ranking is to identify sections that carry the highest individual risk, i.e. where the probability of a fatal road accident is the highest, and sections that carry the highest societal risk and offer strong potential for reducing accident costs as a result of treatment undertaken by road managers.

2. Risk mapping

Risk maps are colour-coded maps showing the relative risk to an individual road-user of being involved in fatal and serious crashes or relative risk of fatal and serious injury on individual road sections across a road network. Risk maps are statistically designed to support national road safety strategies and add an extra layer of information alongside existing approaches. These maps each gave useful information in building up the overall picture of safety performance of the network. It was concluded that accident databases could be established that enabled comparative risk maps to be drawn for different municipalities, and provided useful information on the difference in risk rate. Focus of the risk mapping is to inform the road user how and where he needs to modify his behaviour to minimize risk – i.e. to enable him to recognize the sources of risk on different types of road. Alongside this should be the recognition that risk cannot be eliminated on all roads through infrastructure changes, and that the road user must take a share in the responsibility for a safe road system.

There is variation in methodology for identification of critical zones or black spots and checking road safety level, such as: road safety impact assessment, road safety audit, road safety inspection (periodic road safety inspection, night-time road safety inspection, railway crossing inspection, tunnel inspection, destination-sign inspection, inspection of other road signs and traffic devices, ad-hoc road safety inspection), in-depth studies, black spot management, network safety management, etc.

In the last few decades, road safety has become a high priority in many governments’ policy-making. During 2001, a process of risk rate mapping was developed by TRL for the European Road Assessment Programme, /7/, to enable the risk of fatal and serious injury accidents occurring on different parts of the British primary road network to be compared. As a second stage of risk comparison, a risk rating system was developed which ranked road sections according to their road design features. This rating process was first used extensively in Sweden and Germany, but was subsequently applied in other European countries including the United Kingdom, /8/.

As critical zones were defined as an area or location where there is a higher likelihood for an accident to occur based on historical data and spatial dependency, /9/, within this area it is possible to determine micro location – an accident-prone spot (black spot) – place with a higher number of accidents compared to other similar spots due to local risk factors. The definition of black spot is different from country to country. Many definitions of accident-prone spots are available through research emphasizes that there is no comprehensive definition of what is accepted as a hazard, /10/.

Montenegro has started a Black spot management with risk mapping in 2007 under the responsibility of Road Directorate and there are proposals under discussion to add a special department for road safety issues (road safety unit) in the level of the Ministry of Transport. Neighbor countries, Serbia and Croatia, are also adopting programs for black spot management.
In road accidents statistics in Montenegro, accidents were categorized in: fatal accidents, serious injury accidents, slight injury accidents and damage only accidents, /15/. Both, number of accidents and injuries are weighted with following coefficients recommended by British ministry of transport:

- **Accidents:**
  - $I = \text{accidents only with damage}$
  - $20:50 = \text{injury accidents}$
  - $150 = \text{fatal accidents}$

- **Injuries:**
  - $I = \text{slight injuries}$
  - $5 = \text{serious injuries}$
  - $50 = \text{fatal injuries}$.

The road safety ranking is based on a methodology used in the EuroRAP Programme, /7/. In order to show the varying levels of risk across municipalities, individual sections are allocated into one of five colour coded risk bandings, which represent standard RAP Road Risk Mapping colour palette. The colour palette has been designed to meet the following criteria:

- Achieves international consensus;
- Is based on the significance of particular colors (such as black and red to signify danger) in different countries and is therefore meaningful to a wide audience;
- Adjoining colors are easily distinguishable from one another;
- Ensures that the information is clear and distinguishable when presented in a variety of media (online, print, high resolution, low resolution).

Additionally, in this paper, because of wide dispersion of extreme data, median (Me) as a measure of central tendency was used to obtain the thresholds for each risk banding.

### 4. Results and discussion

Table 1 shows basic data for Montenegro for 2013 and 2014. Number of inhabitant is based on population census from 2011.

<table>
<thead>
<tr>
<th>Municipalities</th>
<th>Inhabitants</th>
<th>Registered vehicles</th>
<th>2013</th>
<th>2014</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td>I</td>
<td>II</td>
</tr>
<tr>
<td>Podgorica</td>
<td>186290</td>
<td>69318</td>
<td>1455</td>
<td>434</td>
</tr>
<tr>
<td>Cetinje</td>
<td>16689</td>
<td>6283</td>
<td>83</td>
<td>47</td>
</tr>
<tr>
<td>Kolosin</td>
<td>8396</td>
<td>1815</td>
<td>74</td>
<td>15</td>
</tr>
<tr>
<td>Danilovgrad</td>
<td>18507</td>
<td>4881</td>
<td>111</td>
<td>24</td>
</tr>
<tr>
<td>Niksic</td>
<td>72581</td>
<td>19589</td>
<td>486</td>
<td>151</td>
</tr>
<tr>
<td>Pluzine</td>
<td>3252</td>
<td>512</td>
<td>12</td>
<td>5</td>
</tr>
<tr>
<td>Savnik</td>
<td>2074</td>
<td>332</td>
<td>15</td>
<td>2</td>
</tr>
<tr>
<td>Bijelo Polje</td>
<td>46138</td>
<td>9232</td>
<td>101</td>
<td>66</td>
</tr>
<tr>
<td>Mojkovac</td>
<td>8638</td>
<td>1686</td>
<td>43</td>
<td>24</td>
</tr>
<tr>
<td>Berane</td>
<td>34035</td>
<td>6713</td>
<td>29</td>
<td>16</td>
</tr>
<tr>
<td>Rozaje</td>
<td>23008</td>
<td>5092</td>
<td>39</td>
<td>26</td>
</tr>
<tr>
<td>Plav</td>
<td>13133</td>
<td>2378</td>
<td>10</td>
<td>5</td>
</tr>
<tr>
<td>Andrijevica</td>
<td>5081</td>
<td>1021</td>
<td>5</td>
<td>2</td>
</tr>
<tr>
<td>Pljevlja</td>
<td>30844</td>
<td>7186</td>
<td>63</td>
<td>27</td>
</tr>
<tr>
<td>Zabljak</td>
<td>3576</td>
<td>772</td>
<td>13</td>
<td>6</td>
</tr>
<tr>
<td>Herceg Novi</td>
<td>30923</td>
<td>12746</td>
<td>232</td>
<td>62</td>
</tr>
<tr>
<td>Kotor</td>
<td>22644</td>
<td>10647</td>
<td>297</td>
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<td>Titov</td>
<td>14058</td>
<td>6592</td>
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<td>24</td>
</tr>
<tr>
<td>Budva</td>
<td>19255</td>
<td>11283</td>
<td>249</td>
<td>69</td>
</tr>
<tr>
<td>Bar</td>
<td>42128</td>
<td>17566</td>
<td>366</td>
<td>100</td>
</tr>
<tr>
<td>Ulcinj</td>
<td>19959</td>
<td>7399</td>
<td>177</td>
<td>67</td>
</tr>
</tbody>
</table>

1 = accidents only with damage; 2 = injury accidents; 3 = fatal accidents; I = slight injuries; II = serious injuries; III = fatal injuries
Annual average values for demographic risk range thresholds (Me for accidents and injuries retrospectively: Me_ac=619.91; Me_in=103.94) and traffic risk range (Me_ac=195.18; Me_in=37.64) for two-years data period is presented in Table 2. Demographic risk was shown as weighted number of accidents and injuries per 100,000 inhabitants; while traffic risk was shown as weighted accidents and injuries per 10,000 registered vehicles. Risk rate maps for those are presented on Fig. 1 and Fig. 2.

Critical zones were north and central municipalities, especially 13 (Kolasin) and 16 (Pluzine) where both, demographic and traffic risk were twice as high as average. High risk rates are also recorded in municipalities: 8, 14, 15, 16, 17, and 21. The lowest risk of being involved in accidents or injured in it, were in municipalities 4 (Pljevlja) and 18 (Berane). It can be observed that there are a large number of municipalities defined as ‘high risk’. Therefore, it’s not surprising that on Montenegro roads 75 black spots have been identified during Black Spot Management programme, where 18 have been alarming, mostly in municipalities identified as very risky. To understand picture of road safety in Montenegro it should be noted that the large number of tourists’ vehicles were involved in traffic flow during summer season. That’s reason why smaller municipalities (with small number of registered vehicles: municipalities N° 14, 15, 20, 21 on figs) were with ‘high’ range when observing traffic risk. Also, those were municipalities with main and regional roads. One of the reasons for bad road safety situation in Montenegro is inapproachable (rough) terrain. Main road, from north border (with Serbia) to capital city (Podgorica) and seaside, is through canyon. The quality of road network in Montenegro, as well as transport safety will be enhanced with the construction of two highways Bar-Boljare and Jadranosko–Jonski highway, in accordance with Transport Strategy of Montenegro.

Figure 3 and figure 4 presents change of demographic and traffic risk from 2013 to 2014. Risk for 2013 is presented on the left side of contour of municipality and risk for 2014 on the right side of municipality. It can be observed that in most cases there was impairment of road safety. Like in following municipalities: 5, 7, 12, 15, 19, and 20. In few municipalities improvement of road safety has been recorded: 6, 10, and 11. However, this was more a coincidence rather than progress due to undertaken measures. The most critical zones stayed critical during both years.
Figure 2 Traffic risk rate for assessment period

Figure 3 Demographic risks changing in Montenegro from 2013 (left side of contours) to 2014 (right side of contours)
Figure 4 Traffic risks changing in Montenegro from 2013 (left side of contours) to 2014 (right side of contours)

5. Conclusion

In the last few decades, road safety has become a high priority in many governments’ policy-making, and is now a critical item on the political agenda. On the developing countries, such as Montenegro, the problem is even greater due to the growing number of cars, the poor quality of road projects and the lack of legislation related to driving and safety. As the absolute number of accidents cannot well reflect the road safety, the concept of relative risk indicator is proposed. Risk mapping, by its very nature, relies on the use of historic crash and/or traffic flow data, and shows the combined influence of behavior, road and vehicle. Here, analyses have been made of fatal and serious accidents occurring on the Montenegro road network during 2013 and 2014. Mapping the risk occurs there is high difference in risk values between roads/municipalities which may reproduce false road safety picture. For the individual road user, it is important to understand both the role of the infrastructure and the role of their own behaviour in determining the risk they face on the road. It is important to recognize that not all roads can be managed to the same risk level, but at the same time to seek to keep the risk within acceptable bounds for those roads with the highest levels of risk. For the road provider, the most immediate need is to decide how best to improve the collective risk across the network, primarily in terms of how to spend available budgets most effectively.

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Abstract: In the article on the basis of the developed mathematical model the dynamics of the highly nonlinear system is investigated, consistent patterns of resonant mode occurrences are established, which is experimentally confirmed. It is established that one of the main reasons for high dynamic loading and HMT elements durability reduction is subharmonic resonances caused by close agreement of frequencies of free nonlinear system fluctuations with multiple value of frequencies of diesel engine disturbances. On the basis of the new consistent patterns, the area of stability and parameter move direction is defined for its provision. The offered damper design is synthesized for significantly reducing HMT dynamic loading.

KEYWORDS: OSCILLATIONS, NATURAL FREQUENCY, SUBHARMONIC RESONANCE, DYNAMIC LOADING HYDROMECHANICAL TRANSMISSION

1. Introduction

The Multiple Wheel Chassis are widely applied both as vehicles and as processing equipment in the Russia’s oil and gas sector in extreme operational conditions of Far North, the Polar Urals and West Siberia. The specified chassis is equipped with single-type hydromechanical transmissions which reliability is limited in many respects, in particular, in durability of matching reduction gearboxes [1].

Allison Company offers a project for modernization of MWC by installation of the engine and transmission assembly. However, the cost of the project makes 14 million RUB, which almost four times exceeds the cost of MWC complete overhaul. An increase in durability of matching reduction gearboxes, design development, their implementation, i.e. transmission modernization in the course of chassis maintenance is an effective way of increasing its reliability.

Application of a matching reduction gearbox with several tooth gears providing kinematic match of engine and hydraulic torque converter characteristics is a peculiar feature of the transmission design under study. Strong nonlinearity of the system is caused by gear backlash opening in tooth gears and existing resonance elimination methods in this case are inefficient. In this regard, the proposed work devoted to analyzing the conditions of subharmonic resonant mode occurrences in the strongly nonlinear system and to substantiating the method of their detuning is topical.

The objective of this work is to define the ways of increasing durability of matching reduction gearboxes of MWC hydromechanical transmissions, to develop and implement design solutions providing detuning from subharmonic resonant oscillations on the basis of a new type torque vibration damper synthesis.

Scientific novelty of the work lies in studying consistent patterns of generation and provision of rationale for a method of eliminating subharmonic resonant modes in the system "Diesel Engine – Hydromechanical Transmission" on the basis of studying the dynamics of the strong nonlinear system with polyharmonic disturbances from the diesel engine.

2. The background of the problem and rationale for the key questions to be explored

Transmission dynamic loading is determined by influence of considerable sign-variable torques occurring in the steady-state regime under resonances, as well as under transient regime of engine start-up and dying out, acceleration and braking operation, gear shifting and torque converter lockup.

The analysis of the failure rate shows that the greatest amount of element breakage of the mechanical system "Engine – Transmission - Transport Vehicle" deals with the part between the diesel (the activator of mechanical oscillations) and a pump wheel of the torque converter.

The solution of specific problems is hindered due to absence of an objective method for choosing a type of damper design and its parameter identification. It predetermines a large volume of experimental and test and evaluation work at the stage of developing the design when modification demands essential time, work and material expenditures.

The background for the research is scientific development of many domestic and foreign experts. Most fully the technique of tuning resonant modes on the basis of oscillation damper synthesis is stated in the works of Grishkevich A. [1]. The results of researching the dynamic behavior of nonlinear systems leading to generation of such phenomena as clunk noise and shuffle are given in works of A. Crowther, R. Singh, Stahl, K., Pflaum, H., Meingassner, G. J., Lohmann, B. et al. [2,3,4,5]. The solution of this problem is stated in practical methods and reference books of Centa Company (Germany). The results of the company's research and its development are widely applied in mechanical engineering around the world, including Russia.

However, the existing works don't allow for considering high nonlinearity of the elastic characteristics peculiar to the system under study.

On the basis of the analysis of the scientific works devoted to research and design of oscillation dampers, the conclusion is made that reduction in dynamic loading and damper synthesis, which consider real nonlinear properties, does not seem possible due to a complex interrelation of the system elements. The article substantiates the necessity of carrying out theoretical and pilot studies with application of up-to-date methods of nonlinear mechanics, of parametrical oscillation stability assessment, of simulation modeling and of processing of experimental data.

3. Theoretical research of dynamic loading

Hydromechanical transmission is a complex multiunit mechanical nonlinear system of a variable structure containing ring elements.

The detailed analysis of this model has allowed for drawing a conclusion that the dynamic torque on transmission shafts is formed by:

- a periodic component of the engine torque, both in steady-state and non-steady-state operating regimes;
- the dynamics of the mechanical system under transients of standing start, acceleration, gear shifting and torque converter lockup.

The dynamic loading of the matching reduction gearbox design elements (the kinematic scheme is given in Fig.1) is a defining factor limiting durability of engine and transmission assembly for the MWC family under study.
The analysis has shown that the dynamic loading of the matching reduction gearbox design elements can be effectively defined on the basis of studying the dynamics of a two-mass system – the so-called pre-torque-converter zone [1]. This system (the computational scheme is given in Fig. 2) comprises an engine flywheel (the first mass), a torque converter impeller wheel (the second mass) with the inertial masses of a matching reduction gearbox (cooling system fans, compressors, etc.) connected to it.

The nature of interrelation between these two masses is built up by elastic and dissipative characteristics of the torsion damper (torsion, see Fig. 2) and tooth gears of a matching reduction gearbox, which backlash opening forms nonlinearity of an elastic interaction. The analysis of the dynamic process in the system under study and predetermination of the ways for reduction in dynamic loading are made on the basis of a mathematical model of the pre-torque-converter zone of the hydromechanical transmission as a nonlinear two-mass system (1).

\[
\begin{align*}
J_1 \ddot{\varphi}_1 + M(\varphi) &= M(t) \\
J_2 \ddot{\varphi}_2 - M(\varphi) &= 0
\end{align*}
\]

where \( J_1, J_2 \) are inertia moments of the engine and the impeller respectively; \( M(\varphi) \) is the nonlinear elastic interaction function (Fig. 3); \( M(t) \) - time function of engine and impeller wheel torque.

Nonlinear function - dependence of the moment on relative angular coordinate – is schematized (Fig. 3) and is accepted as symmetric with a dead zone without saturation [7], i.e.

\[
M(\varphi) = \begin{cases} 
0 & \text{at } |\varphi| \leq \delta \\
\frac{c}{|\varphi| - \delta} \cdot \text{sign}(\varphi) & \text{at } |\varphi| > \delta,
\end{cases}
\]

the parameters of which \( \delta \) and \( c \) are defined experimentally.

The time function of the engine torque \( M(t) \) is accepted in Fourier’s series form, the parameters of which are defined according to the data of the manufacturing factory.

\[
M(t) = M_0 + \sum_{m=1}^{\infty} M_m \cos(m\omega t + \beta_m)
\]

The harmonious analysis of the engine torque shows that the most dangerous harmonics of the engines YaMZ-8401, YaMZ-240, D-12 (according to the manufacturer, JSC Avtodiesel) is the sixth, i.e. in the engine rpm range, the exciting frequency makes from 70 to 85 Hz that considerably (3 ... 4 times) exceeds the frequencies of natural oscillations of the system under study (12 ... 17 Hz).

The results of the numerical solution (1) at values of the parameters corresponding to the object of the pilot study are given in Fig. 4. As comes from this Fig. 4 oscillations of the torque occur regarding the established value with amplitude 3 ... 4 times exceeding the established value.

The characteristics of the periodic process under study (moment amplitude, natural oscillations) significantly depend on the initial conditions which is characteristic for nonlinear systems. In fig. 4, two various types of movement characterized by the size of backlash opening are shown. It leads both to an amplitude change and frequency of periodic oscillations of the system under study.

The results of numerical modeling are confirmed experimentally at running trials of the wheel chassis KZKT-7428, equipped with YaMZ-8401.10. Similar results were obtained at trials of wheeled chassis KZKT-74286 with engine YaMZ-240 and MWC MAZ-537 with the engine D-12 (the trial results of KZKT-74286 and MAZ-537 are not given).

The fragment of the oscillogram of the torque changing in the steady-state mode, at the engine rpm range of 700 ... 850 rpm, is given in Fig. 5a. From the provided data, it follows that the change of the torque has an oscillatory character with an amplitude up to 2,500 Newton meters, and the process frequency makes from 10 to 19 Hz with a disturbing frequency of the engine from 70 to 85 Hz (the 6th main engine harmonics of the engine YaMZ-8401). This alternating character of the torque within the time frame (Fig. 4, Fig. 5), corresponds to subharmonic resonant oscillations in the nonlinear mechanical system with a dead zone.


\[
\phi + [a - 2h \cos(2\tau)]\phi = 0, \quad (5)
\]

where 
\[a = \left(\frac{2\omega}{p}\right)^2; \quad h = a\mu; \quad 2\tau = pt.\]  

Ince-Strutt diagram with the coordinates \(h(a)\) is given in Fig. 6 (stability area is shaded).  

The parameters of the system under study are represented by point A, being in the unstable zone. The coordinates of the point are determined as follows. The abscissa of the point A is determined according to the formula \(a = \left(\frac{2\omega}{p}\right)^2\), where \(\omega\) is the frequency of natural oscillations of the nonlinear system, determined according to the formula (4) and corresponding to the amplitude of the moment of the main motor harmonics with a frequency \(p = 6\), \(M = 1,200\) Nm).  

To determine the parameter \(h\), it is necessary to evaluate modulation depth \(\mu\), characterizing the amplitude of a variable parameter of the system (relation \(q_{dyn}/q_{st}\) in formula (3)) being determined for this system according to the formula (6).

\[
h = \frac{\arcsin \left(\frac{\pi}{\tau} \left(\frac{\beta + x}{\sqrt{\beta - x} + \sqrt{\beta + x}}\right)\right)}{\pi} \left(\frac{\pi}{\tau} \left(\frac{\beta + x}{\sqrt{\beta - x} + \sqrt{\beta + x}}\right)\right), \quad (6)
\]

where \(\epsilon\) is rigidity of an elastic element (torsion), \(\epsilon = 72,000\) Nm/rad; \(q_{dyn}\) is the amplitude of the sixth engine harmonics; \(q_{st}\) is the amplitude corresponding to the engine load moment (loss torque); \(\delta\) is a dead zone defined by backlashes in the gear of the matching reduction becogear (\(\delta = 0.026\) rad); \(x\) is a shift of the zero line \((x = 0.029)\).

For the given parameters at \(rpm\) engine speed of 850 rev/min, Ince-Strutt diagram parameters makes \(a = 0.07\) \(h = 1.033\).

**Figure 5.** The fragment of the oscillogram of the torque changing at the torsion shaft (a) and its spectrum (b) in an idle mode (NICE=740 rev/min.)

The spectral analysis of the obtained numerical solution verifies the abovementioned conclusion about generation of oscillations at subharmonic frequencies multiple times smaller than the main harmonics (the 6th) of the YaMZ-8401 engine. In this case, the subharmonic resonance oscillations in nonlinear systems coexist with principal forced oscillations.

Thus, the solution of the system of nonlinear differential equations is not unambiguous. In addition, existence of several steady-state modes with various amplitudes is possible \(\phi_1 = \phi_2(t)\), including an unstable mode. Research of oscillation process stability and provision of a rationale for the direction of resonant modes elimination is carried out on the basis of the device of parametrical oscillations [8].

To implement this approach, an equation system of (1) by introduction of a coordinate of relative angular movement \(\phi = \phi_1 - \phi_2\) is reduced to the form of nonlinear differential equation (2)

\[
J_{\text{reduced}} \ddot{\phi} + M(\phi) = M(t), \quad (2)
\]

where \(J_{\text{reduced}}\) — reduced moment of inertia. Other functions introduced into the equation (2) are given above. The complexity of the analytical determination of the reduced inertia moment is that it is necessary along with the moment of flywheel inertia to consider response time of the torque converter impeller filled with hydraulic fluid. Besides, in the design of MWC transmission, it is from the pump wheel that many mechanical devices are actuated (fans, pumps, compressors, etc.). In this regard this size is determined experimentally [6] \((J_{ICE} = 8.5\) kgm\(^2\)).

To analyze stability of the system, the mathematical (2) model is modified to Mathieu equation [8]:

\[
\ddot{\phi} + 2\epsilon \dot{\phi} + \omega^2 \left(1 - \frac{\text{\phi}_{\text{dyn}} \cos(\omega t)}{\text{\phi}_{\text{st}}}\right) \phi = 0 \quad (3)
\]

In this equation \(\epsilon = \frac{b}{2}\) is a dissipation parameter; \(\omega^2\) is a square of natural frequencies of the nonlinear system, determined considering the constant component of the engine moment. The value of this moment is determined by losses in the matching reduction gearbox, by losses on overcoming resistance in the compressor drive, the drive of fans, etc. The frequency of natural oscillations is defined by the following response characteristics:

\[
\omega = \sqrt{\frac{\pi}{\tau} \left(\frac{\beta + x}{\sqrt{\beta - x} + \sqrt{\beta + x}}\right) \left(\frac{\pi}{\tau} \left(\frac{\beta + x}{\sqrt{\beta - x} + \sqrt{\beta + x}}\right)\right)}, \quad (4)
\]

where \(\phi\) is angular amplitude at the neutral point in the transmission and at the shaft speed matching to idle speed; \(x\) is the shift of the zero line determined by a constant component of loading. Introducing the parameters of depth of disturbance \(\mu\) and its alternation frequency \(p\), Mathieu equation without considering dissipation is modified to

\[
\ddot{\phi} + [a - 2h \cos(2\tau)]\phi = 0, \quad (5)
\]

**Figure 6.** Ince-Strutt Diagram.

Analyzing the results of numerous calculations, corresponding to the diagram it appears that at the given parameters of the dynamic system in the range of parameter values from 0 to 0.111 (frequency range of the disturbance we are interested which is determined by the detuning factor \(b = 6\) at the minimum allowable speed ICE – 600 rev/min) the parameter \(h\) exceeds the value of 0.9. Therefore, all estimated values of set points in h-a coordinates reside in an instability zone \(A\) (shaded) to the left from the red line. Such a situation remains unchanged up to 900 rev/min when periodic oscillations of the dynamical system have an unstable (resonance) character.

Solution of the inverse problem of choosing optimal parameters of a dynamical system, being unable to control the modulation depth \(\mu\) and to change the characteristics of the nonlinear elastic relationship (dead zone value), is possible only if to change the setting of detuning of the system \(P/\omega\) (relation \(P/\omega\) is...
Ince-Strutt diagram parameters of the elastic element of 45,000 … 50,000 Nm/rad. Furthermore, the dynamical system, the desired effect is achieved at the rigidity of stable one, i.e. from zone $A$ to zone $B$. With respect to the dynamical system, the desired effect is achieved at the rigidity of the elastic element of 45,000 … 50,000 Nm/rad. Furthermore, the Ince-Strutt diagram parameters made $a=0.089$ $h=0.684$ at $c=45000$ Nm/rad and $a=0.098$ $k=0.854$ at $c=50000$ Nm/rad.

When the reliability of the results obtained by comparison with experimental data, they can become the basis for the method of limiting torque oscillation amplitudes and, respectively, of dynamic loading.

4. Analysis of the research results

Qualitative and quantitative comparison of the research results was carried out on the base of analysis for amplitudes and frequencies of high frequency oscillations in the pre-torque-converter zone (see. Fig 3 and Fig 4). Some differences of amplitude and frequency might be explained by deviation of initial conditions when modeling.

On the basis of the research results the method of elimination of resonant modes in the nonlinear system is developed. According to nonlinear elastic characteristic parameters on the basis of the developed mathematical model, the zone of stability and probability of subharmonic resonant mode generation is defined. For their eliminating, it is necessary to vary parameters of Ince-Strutt diagram ($h$ and $a$) from a condition of getting into the stability zone. The most effective and easy way is tuning out the natural frequency of the system by changing angular rigidity of matching reduction gearbox. In particular, for reduction in dynamic loading of the HMT under study by elimination of a resonant mode, it is necessary that torsion damper angular rigidity was not higher than 50,000 Newton meters per radian.

Accordingly, a design of a matching reduction gearbox (Fig. 7) with a new type of torsion vibration damper with the rigidity of 21,000 Nm/rad. This silicone elastic element comes from the line of the Centa Company dampers and fit well into the existing design of the matching gearbox.

The proposed design of the matching gearbox (Fig. 7) is produced as a three-shaft transmission with overdrive skew gears. In the case 1 on the bearing supports, there is a drive shaft-mounted gear 2, intermediate 3 and driven 4 shafts. The case 1 is fastened with the bolts to the housing 5 of the engine flywheel 6. The support 9 is connected with the engine flywheel 6. The disk 10 is connected to the support 9 with female splines. The elastic element, made as a silicone ring 11, with the help of the splines on the outside surface is connected with the disk 10, and the splines on the internal surface with multiple piece hub group 12, carried by bearings 15, installed in the support 9. The transmission shutdown mechanism made in the form of a two-stage gear coupling located in the cavity formed by the front part of the case. The gear clutch consists of a driven half coupling 8, connected by internal splines with the left end of the extended drive-shaft of the gear 2. The half-coupling 8 is connected by internal splines with the sleeve 7. Thus, the sleeve 7 provides for connection of the driven half coupling 8 with multiple piece hub group 12 on the outside surface where there are splines. At that, the sleeve 7 is fixed at two positions with ball locking mechanism 16. The driven shaft 4 through cardan shaft rotates a pump wheel of the torque convertor of hydromechanical transmission of the transport vehicle (fig. 7 does not show it).

The proposed design of the matching reduction boxgear works in the following way. With steady rotation of the shaft, the flywheel 6, the support 9, the disk 10, the elastic ring 11, the multiple hub group 12 due to the sleeve connection 7 in an operating condition, the drive shaft-mounted gear 2 rotate as one unit. As the tooth rim of the drive shaft-mounted gear 2 by means of the gear 13 of the intermediate shaft 3 is connected with the gear 14 of the driven shaft 4, the latter rotates taking into account the gear ratio transmission with an angular velocity greater than the drive shaft speed.

With unsteady rotation of the shaft, the flywheel 6, the support 9, the disk 10, can oscillate in regard to the multiple hub group 12, and respectively, to the drive shaft-mounted gear 2 within the angular allowable deformation of the elastic ring 11.

As it is shown above, the parameters of the elastic ring 11 are chosen to provide the dynamic system stability, decrease the frequency of natural oscillations of the mechanical system «Engine-Damper-Impeller Wheel» take resonant modes out of the limits of an engine rpm speed operating range. This provides the desired resource of the matching reduction gearbox.

The efficiency of research results is defined on the basis of experimental determination of torque convertor loading at trial runs. Dynamic loading of the hydromechanical transmission with an improved design of a matching reduction gearbox is evaluated at road test simulation. Efficiency of the proposed solutions was evaluated by a value of the dynamic torque on the drive shaft according to the amplitude and reverse characteristic (see Fig. 8).
Comparison of the amplitude and reverse characteristics (lines 1, 2 in Fig. 5) shows that the proposed design of a matching reduction gearbox allows for tuning resonance out of engine rpm working range and reduce the dynamic torque in 5... 6 times. Thus, durability of matching reduction gearbox elements increased significantly.

Conclusions

1. The mathematical model applied in the research and a package of computer programs give a possibility to investigate dynamics of highly nonlinear system, to determine consistent patterns of generating resonant modes, verified experimentally.

2. It is substantiated that one of the main reasons for high dynamic loading and HMT element durability restriction is subharmonic resonance caused by close coincidence of natural frequencies of the nonlinear system with multiple value of diesel engine disturbing frequencies. On the basis of the determined consistent patterns the area of stability and the direction of parameter variation for its maintenance is defined.

3. The conducted pilot researches of dynamic loading of the hydromechanical transmission of three MWC models, statistical data processing confirms validity of the developed mathematical model reflecting physical processes in resonant modes and of correctness of the basic assumptions.

4. On the basis of the obtained results, the engineering solutions, allowing for reducing dynamic loading of the transmission are proposed and designs of new type dampers are developed for matching reduction gearboxes of the HMT of various models of MWC. It allows for tuning resonant modes out of engine rpm working range and for reducing dynamic loading of the transmission up to 5 ... 6 times, thus predetermining an increase of matching reduction gearbox elements durability.

References

Abstract – Micro aerial vehicles design, due to their low Reynolds number aerodynamics, motivated the aerospace engineering community a lot the last year, and especially the flapping wing operated MAVs.

The simulation of the mechanical equations of motion for a flapping wing MAV is presented in this work to have an approximate the behavior and the condition of flight of the vehicle and to present a control model that can be implemented to auto control the vehicle.

The spherical coordinates system is used to develop the equations in this work, and Mathcad software is mainly used for the solution, simulation and graphing of the results, constants related to the size of the vehicle are changed to match different a range of existing flying insects or birds.

Upstroke and down stroke of the flapping wing were modeled using two different drag coefficient.

The study resulted in excluding the smaller sizes and higher flapping frequency from use of this model without any rectifications that account for the accentuated mow Reynolds unsteady effect, but bigger vehicles (bird sized) were modeled with a good accuracy and the model can be used as well for auto-control by predefined flying path.

Keywords: Aerodynamic, control, feather, flapping, MAV, wings.

Introduction

Birds, insect and bats was a subject of fascination to the human since the dawn of humanity, and he did not just content by envying their ability, but he invested his energy and intellect in order to mimic them and conquer the sky using the ability of those great flyers.

However it was only recently with the advent of simulation, analysis and high-speed videography that we could decipher some of their complex mechanisms that make them that good.

Such a way of generating lift and thrust was examined closely from pioneers from Abbas Ibn Firnas to Leonardo di Vinci; however lately researchers are more and more attracted by this kind of propulsion for the application on small scale flapping wings vehicles, because of the need for a smaller, maneuverable aerial vehicle.

1. State of art in micro aerial vehicle design

Flapping wings:

Nature makes it clear that this propulsion mechanism is very efficient and effective in an aerodynamics view at a low Reynolds number and in the same time allows high degrees of maneuverability, humming birds are the best example, while despite requiring a lift coefficient, in the quasi-steady sense, over twice that of any aircraft. Bumblebees are able to fly

However, a distinction should be made when focusing on this class of vehicles between bird-like vehicles called Ornithopter and Insect-like vehicles called Entemopters [1].

The last ones suits more MAV tasks because hovering and this class provides maneuver in tight spaces.

1. Numerical investigation with Mathcad

Theoretical investigation:

the flight of the vehicle is well described my Newtonian mechanics, the equations of equilibrium which result from Newton’s second law are used to describe the flapping wing vehicle, and other systems of coordinates than the rectangular is used because it describes better this motion in space [9].

The resultant of all the forces acting on a particle is proportional to the acceleration of the particle:

\[ \sum F = ma \]  

By applying this basic principal to a body in motion in a plan, the equations in a rectangular basis would be:

\[ \sum F_x = ma_x \]

\[ \sum F_y = ma_y \]
The original idea of this work [9], is the use of two drag coefficient in the expression of the resultant force; as we deduct from nature, when a wing is in the upstroke the muscle is used to alter the shape of the wing to cause less drag therefore a smaller drag coefficient $C_d_1$ in contrast in the down stroke the shape of the wing is optimal for a large lift force which is the opposite of the drag force, if we change the direction of the axis, therefore a biggest drag coefficient $C_d_2$.

With the introduction of this two-drag coefficient approach, we will have two forces and knowing that the lift force expression is:

$$2\rhoAVC FL$$

(2)

Where:
- $\rho$: density of the air.
- V: the velocity.
- S: the lifting area (both wings).
- $CL$: lift coefficient

By introducing both drag coefficients:

$$2\rhoAVC F_d$$

(3)

$$S, \rho(Ao\cos(\alpha))^2 \sin(\cos(\alpha))][\cos(\beta \sin(\frac{\pi}{2}) + \phi - \alpha)] - h_r \cos(\phi) \frac{d\theta}{dt}$$

(4)

The switch between the two different drag coefficients in the upstroke and the down stroke is done by the introduction of a formulation that uses the direction (therefore sign) of the velocity to deactivate one or the other coefficient such as:

$$0.5 - 0.5\sin(\cos(\alpha))$$

The velocity approximated from the sinusoidal motion of the wing, the path would be $A \sin(\omega t)$ therefore the velocity can be expressed as the first derivative: $A \omega \cos(\alpha t)$ another sign related expression is needed so the expression of the velocity is:

$$v^2 = (A\omega \cos(\alpha t))^2 \sin(\cos(\alpha))$$

(5)

Where:
- $\omega$: Angular frequency.
- A: Amplitude.

A projection of all those expressions on their respective axis is performed the find the 3D equations, the gives the explanation of the forces acting on the center of gravity of the MAV and the angles between the projections.

The accelerations equations for each axis are:

**For The radial, coordinate axis:**

$$\ddot{r} = \frac{1}{m} (-mg \sin \phi + mr\dot{\omega}^2 \cos^2 \phi + m\phi^2 +$$

$$\frac{1}{2}(0.5 - 0.5\sin(\cos(\alpha)))C_{d_1} + 0.5 + 0.5\sin(\cos(\alpha))C_{d_2}).$$

(6)

For The polar axis:

$$\dot{\theta} = \frac{1}{m r \cos \phi} (-2mr \phi \cos \phi + 2m \dot{\phi} \sin \phi) +$$

$$\frac{1}{2}(0.5 - 0.5\sin(\cos(\alpha)))C_{d_1} + 0.5 + 0.5\sin(\cos(\alpha))C_{d_2})$$

$$S, \rho(Ao\cos(\alpha))^2 \sin(\cos(\alpha))][\cos(\beta \sin(\frac{\pi}{2}) + \phi - \alpha)] - h_r \cos(\phi) \frac{d\theta}{dt}$$

(7)

**The Azimuth coordinate axis:**

$$\dot{\phi} = \frac{1}{m} (-mg \cos \phi - 2mr \dot{\phi} - m\dot{\phi}^2 \sin \phi \cos \phi +$$

$$\frac{1}{2}(0.5 - 0.5\sin(\cos(\alpha)))C_{d_1} + 0.5 + 0.5\sin(\cos(\alpha))C_{d_2})$$

$$A, \rho(Ao\cos(\alpha))^2 \sin(\beta) - h_r \frac{d\theta}{dt}$$

(8)

A certain criteria was added the last equation in order to control the altitude of the MAV, [9]:

$$0.5 - 0.5\sin(\cos(\alpha - H_m))$$

(9)

Where $H_m$ is the maximum height that the MAV is supposed to reach.

### 2. Mathcad implementation:

We had a set of parameters, some of them remained constant, and others were modified.

Two different range of dimension were used, the bird-like and insect-like, in the literature we found the parameters needed and a formula developed by PENNYCUICK, that link the flapping frequency of the birds wing to the other parameter ($m, S, \rho, g, b$ (wing span)) [6] was used, which is written as:

$$f = m^{3/8} g^{-1/2} b^{-23/24} S^{-1/3} \rho^{-3/8}$$

(10)

For the bird-like configuration two different sizes were used, the first ones from DARPA [9] definition the second one from an actual bird which is the Dove prion.

For the insect-like configuration, dimensions of a dragonfly were used.

### Drag coefficient:

It is mentioned in the literature that the lift to drag ratio is between 3 and 17 which gives a drag coefficient for birds between 0.8 and 1.2 however the motion of the wing affect significantly the coefficient therefore a set of drag coefficient around the real value were used.

For insect like MAVs insects wing have very small drag coefficient around 0.06, so we used numbers around it.

### 3. Mathcad manipulations:

The Cartesian coordinates where obtained by the transformations:

**Fig.1: spherical Coordinate**
We found out that the MAV goes very high up to 40m, which means that the force is very strong; therefore, changes in the damping coefficient were done for the radial coordinates, by steps of 0.05. [1]

We performed more manipulations on different parameters [7] the results are illustrated in graphs (1), (2), (3)

It was obvious the instability occurred in the z-axis so the damping coefficient was changed from b to b1 and the later was varied. the increase of the damping coefficient (b1=0.1 and 0.15) the results were close but far more stable, as it’s illustrated in the graphs (4) where we can see the path of flying in the \((r,\phi)\).

3.1. Steering:

We steered the MAV by directly implementing and fixing the angles \(\alpha\) and \(\beta\) or by making one or both of the angles time dependent.

We used as an example [7], the change in the angles value by a time depending functions:

This do not affect the vertical coordinates; however, we can see the change of path in the other plans

\[
\alpha = \frac{\pi}{2.5} (e^{-0.01t})
\]

\[
\beta = 1.5 \frac{\pi}{2}(e^{-0.02t})
\]

(12)

In the Graph 5 we see that after the stabilization the MAV stop turning in circles and take a fixed direction.

3.2. Dimensions of an insect:

We tried the characteristics of a dragonfly, we did some manipulation in all the parameter but the MAV only flies for very high frequency, amplitude, or drag, so this model does not work for very small dimensions.

4. Results:

We can summarize the results in the next points:

1. The major parameter that influences the MAV flying is the damping coefficients.

2. The flapping frequencies have to be defined and limited under certain value to keep up with the reality, and the available force for flapping.

3. The difference between the two drags coefficients plays a central role in making the MAV fly.

4. And finally, we had a confirmation that due to the unsteady aerodynamic phenomenon that are accentuated in the insect case, the approximation used cannot be applied.

5. References:


THE USAGE OF BITUMINOUS SANDS IN ROAD PAVEMENT ASPHALT MIXTURES COMPOSITION

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Abstract: In order to obtain significant costs reductions associated with the use of road bitumen in asphalt mixes, this product currently being only obtained through imports, an ample research have been undertaken regarding the use of bituminous sands or oil sands, which present great advantages, especially in areas near deposits located in Bihor County. These raw materials present the advantage of having a percentage of natural bitumen in their mineral composition and thus their usage for asphalt mixtures preparation leads to substantial reductions of bitumen quantities required in the mix. Specific technological recipes have been researched in road laboratory, based on the department norm CD 42-85, concerning the direct use of oil sands, with and without adding hard paving grade bitumen, for warm asphaltic road pavement construction.

Keywords: AGGREGATES, BITUMINOUS SAND, HARD PAVING GRADE BITUMEN, ASPHALT MIXTURES, ROAD PAVEMENT

1. Introduction

Bituminous binders, according with specialized literature, are complex mixtures of animal origin hydrocarbons or are being obtained through a pyrogenic reaction, often accompanied by their combinations with oxygen, nitrogen, sulfur, etc. They are in the form of liquid, viscous or solid, having a dark brownish and black color and being completely soluble in carbon disulfide (Mătăsaru et al., 1966).

In order to fulfill their role as road binders, bituminous materials must have a bond strength or agglomeration. This condition is satisfied only by two categories of compounds, namely natural bitumen and tar. Bituminous binders, which occur in nature, are most often associated with minerals and also are used from ancient times (Les ENROBES bitumineux - Tome 1 + 2, 2005).

The largest oil field is located near Athabasca, Alberta Canada, but there are significant oil reserves located in Venezuela (Lake Bermudes), Cuba, Madagascar, Syria, Albania (SELENE) (Joseph K. Anocha - Boateng, Erol Tutumluer, 2012).

In Romania, the bitumen is found naturally in the form of sand impregnations in two geographical areas, namely: in Bihor County, in the Derna-Tatarus-Budoi basin, near Oradea, Prahova County, Mattita and Pacureti. The impregnated substance is a very soft asphalt that must be processed in order to obtain the bitumen. The sands contain on average 10-20% pure bitumen, the separation of bitumen from sand being performed with warm water having weak alkaline proprieties. An extracting attempt of bituminous material has been performed using kerosene, but the results haven’t been favorable (Nicoară et al., 1985). The structure and the composition is very similar to the natural petroleum bitumen; the superiority in behavior is given occasionally by the higher content of anhydrides.

In situ, the bituminous sand deposits are mainly composed of quartz sand, which have a thin film of water and fine particles and the bitumen fills the pore spaces between grains of sand. Quartz sand, silt and clay, meaning the inorganic materials of the oil sand composition, normally constitutes about 80 % by bitumen weight and water is about 15% and 5% (Ronald F. Probstein, Edwin E. Hicks, 2006).

Fig.1 Bituminous sand (Dayna Linley, 2010)

2. The research methodology

In roads laboratory have been conducted four technological recipes complying with the directions of departmental normative CD 42-85 concerning the direct use of bituminous sands with and without adding hard paving grade bitumen for the execution of warm asphalt mixtures for road pavements.

The first recipe conducted in the laboratory is for asphalt base associated with road pavements for light traffic (A.31.nb.), the asphalt mixture, containing bituminous sands and having added hard paving grade bitumen, mostly being used for roads with a technical class III. The other three warm asphalt mix recipes have in their composition bituminous sand without adding additional hard paving grade bitumen, as follows: asphalt concrete (open graded asphalt concrete) with crushed gravel for the binder layer (B.a.31.nb.f), asphalt base concrete for bituminous base course (A.31.nbf) and asphalt base concrete (A.31.nb.f) designed for light traffic, which are used for roads with a technical class of IV-V. All these asphalt mixes have in their composition bituminous sands from Derna-Tătăruș, Bihor County and cylindrical samples have been realized in order to study their behavior.

It has been used crushed sand and gravel from Cristesti (Iasi), natural sand from Boureni (Iasi) quarry, filler originated from Bicaz (Neamt County), bituminous sand from Derna Tatarus (Bihor County) career and the hard paving grade bitumen have been supplied by OMV Refining & Marketing GmbH (Austria). The preparation of asphalt mixtures in the laboratory is carried out by heating and mixing the mixture components for 10-15 minutes at 170-180°C in the case of bituminous mixtures with additional hard paving grade bitumen and temperatures of 175-190°C for mixes without hard paving grade bitumen (Indicativ 605-2014, 2014).

The bitumen content specific to bituminous sands from Derna-Tătăruș in Bihor County is 19.39%.

Table 1 presents the particle size distribution curves of the used aggregates.

Table 1: Particle size distribution of natural aggregates (gradation)

<table>
<thead>
<tr>
<th>Aggregate size, mm</th>
<th>Sieve size, mm</th>
<th>Percentage of weight passing sieve, [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Crushed gravel 16 – 31.5</td>
<td>100.0</td>
<td>93.4</td>
</tr>
<tr>
<td>Crushed gravel 8 – 16</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Crushed gravel 4-8</td>
<td>-</td>
<td>-</td>
</tr>
</tbody>
</table>
Table 2: Hard paving grade bitumen characteristics

<table>
<thead>
<tr>
<th>No.</th>
<th>Characteristics</th>
<th>U.M</th>
<th>Values obtained</th>
<th>Reference values EN 13304 - 2009</th>
<th>Test method</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Penetration at 25°C</td>
<td>1/10  mm</td>
<td>10</td>
<td>5-15</td>
<td>SR EN 14267-07</td>
</tr>
<tr>
<td>2</td>
<td>Softening point</td>
<td>°C</td>
<td>91.5</td>
<td>85-95</td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>Solubility in organic solvents</td>
<td>%</td>
<td>99.62</td>
<td>Min.99</td>
<td>SR EN 12592-03</td>
</tr>
<tr>
<td>4</td>
<td>Loss of bitumen mass by heating at 163 °C</td>
<td>%</td>
<td>0.24</td>
<td>Max. 0.5</td>
<td>SR EN 12607/1-07</td>
</tr>
<tr>
<td>5</td>
<td>Marcusson flash point</td>
<td>°C</td>
<td>267</td>
<td>Min. 250</td>
<td>EN ISO 2592</td>
</tr>
</tbody>
</table>

The dosage of natural aggregates and aggregate mixture distribution curve for each type of mixture is represented in the tables below, as follows: Table 3 for A.31.nb, Table 4 for B.a.31.nb.f, Table 5 for A.b.31.nb.f and Table 6 for A.31 nb.f.

The calculation for the addition value of hard paving grade bitumen in the case of asphalt concrete base with crushed gravel, designed for road pavements with light traffic A.31 nb, has been done as recommended by appropriate standard CD 42-85, as follows: natural bitumen content in the mixture is required to be 70% and the hard paving grade bitumen 30%.

For each type of mixture performed in the laboratory have been calculated 5 dosages binder as presented below:

- natural bitumen percentage:
  - binder percentage x 0.7 = a% ;
  - hard paving grade bitumen percentage:
  - binder percentage x 0.3 = b% ;

- dosage of bituminous sand: a - 100 = c% ;

- dosage of natural sand contained in the bituminous sand:
  - c / a = d %.

For the asphalt base concrete with crushed gravel, designed for road pavements with light traffic (A.31 nb) have been used bitumen percent of: 5.00%, 5.20%, 5.40%, 5.60%, 5.80% (the recommended values ranging between 5.00% and 6.00%); for asphalt concrete with crushed gravel for the binder layer (B.a.31 nb.f) have been used bitumen percent of: 4.00%, 4.20%, 4.40%, 4.60%, 4.80% (the recommended values ranging between 4.00% and 5.00%); for the asphalt base concrete for bituminous base course (A.b.31 nb.f) the used percentages were: 3.80%, 4.00%, 4.20%, 4.40%, 4.60% (the recommended range being between 3.80% and 4.60%); for asphalt base designed for light traffic (A.31 nb.f), the bitumen percent is: 4.50%, 4.70%, 4.90%, 5.10%, 5.30% (the recommended values ranging between 4.50% and 6.00%).

Table 3: The natural aggregates dosage for asphalt base concrete with crushed gravel, designed for road pavements with light traffic A.31 nb

<table>
<thead>
<tr>
<th>Aggregate</th>
<th>Percentage of weight passing sieve, [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>31.5</td>
<td>25 20 16 12.5 8 4 2 1 0.12 0.06 0.03</td>
</tr>
<tr>
<td>Crushed gravel 16 – 31.5</td>
<td>25.3 7 25.3 7 23.7 2 17.8 2 7.7 6 1.65 0.14 0.05 - - - -</td>
</tr>
<tr>
<td>Crushed gravel 8 – 16</td>
<td>10.5 7 10.5 7 10.5 7 10.5 7 8.5 1 2.76 0.44 0.16 0.09 0.04 0.02</td>
</tr>
<tr>
<td>Natural sand 0 - 4</td>
<td>3.9 4 3.9 4 3.9 4 3.9 4 3.9 4 3.9 4 3.9 4 3.9 4 3.9 4 3.9 4</td>
</tr>
<tr>
<td>Bituminous sand</td>
<td>17.8 7 17.8 7 17.8 7 17.8 7 17.8 7 17.8 7 17.8 7 17.8 7</td>
</tr>
<tr>
<td>Total</td>
<td>100 100 98.3 92.4 82.3 74.2 62.9 56.4 52.8 49.8 46.8 43.8 40.8 37.8 34.8 31.8 28.8 25.8 22.8 19.8 16.8 13.8 10.8 7.8</td>
</tr>
<tr>
<td>Limits particle size area</td>
<td>90 100 85 100 77 94 70 63 55 46 39 36 26</td>
</tr>
</tbody>
</table>

Table 4: The natural aggregates dosage for asphalt concrete (open graded asphalt concrete) with crushed gravel for the binder layer B.a.31 nb.f

<table>
<thead>
<tr>
<th>Aggregate</th>
<th>Percentage of weight passing sieve, [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>31.5</td>
<td>25 20 16 12.5 8 4 2 1 0.12 0.06 0.03</td>
</tr>
<tr>
<td>Crushed gravel 16 – 31.5</td>
<td>26.15 5 26.15 5 24.4 8 18.4 5 1 8.03 1 1.70 0.15 0.05 - - - -</td>
</tr>
<tr>
<td>Crushed gravel 8 – 16</td>
<td>15.69 9 15.6 9 15.6 9 15.6 9 15.6 9 15.6 9 15.6 9 15.6 9 15.6 9</td>
</tr>
<tr>
<td>Natural sand 0 - 4</td>
<td>55.89 9 55.8 9 55.8 9 55.8 9 55.8 9 55.8 9 55.8 9 55.8 9</td>
</tr>
<tr>
<td>Total</td>
<td>100 100 99.3 92.2 81.8 72.5 62.4 58.8 56.0 49.9 44.8 40.6 37.8 34.8 31.8 28.8 25.8 22.8 19.8 16.8 13.8 10.8 7.8</td>
</tr>
<tr>
<td>Limits particle size area</td>
<td>90 100 85 100 77 94 90 81 70 61 56 52 5</td>
</tr>
</tbody>
</table>

Table 5: The natural aggregates dosage for the asphalt base for bituminous base layer A.b.31 nb.f

<table>
<thead>
<tr>
<th>Aggregate</th>
<th>Percentage of weight passing sieve, [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>31.5</td>
<td>25 20 16 12.5 8 4 2 1 0.12 0.06 0.03</td>
</tr>
<tr>
<td>Crushed gravel 16 – 31.5</td>
<td>31.31 7 31.3 7 29.2 7 22.9 7</td>
</tr>
<tr>
<td>Crushed gravel 8 – 16</td>
<td>15.66 6 15.6 6 15.6 6 15.6 6 15.6 6</td>
</tr>
<tr>
<td>Natural sand 0 - 4</td>
<td>31.67 7 31.6 7 31.6 7 31.6 7 31.6 7 31.6 7 31.6 7 31.6 7 31.6 7</td>
</tr>
<tr>
<td>Bituminous sand</td>
<td>18.23 3 18.2 3 18.2 3 18.2 3 18.2 3 18.2 3</td>
</tr>
<tr>
<td>Total</td>
<td>100 100 97.9 90.7 78.2 67.6 57.3 45.6 34.1 23.7 12.5 0.14 0.06 0.03</td>
</tr>
<tr>
<td>Limits particle size area</td>
<td>90 100 85 100 77 94 90 81 70 61 56 52 5</td>
</tr>
</tbody>
</table>
3. Results and discussions

For all the 4 types of mixtures carried out in the roads laboratory have been determined the following physical-mechanical (only in static regime): apparent (bulk) density; water absorption; swelling over time; stability and creep Marshall.

Table 7: Physical-mechanical properties of the asphalt mixture test samples A.31

<table>
<thead>
<tr>
<th>Characteristic name</th>
<th>UM</th>
<th>Bitumen percentage [%]</th>
<th>Relative to normative AND 605/2014</th>
</tr>
</thead>
<tbody>
<tr>
<td>Apparent density</td>
<td>g/cm³</td>
<td>2.232</td>
<td>2.237</td>
</tr>
<tr>
<td>Water absorption</td>
<td>% vol</td>
<td>6.766</td>
<td>7.179</td>
</tr>
<tr>
<td>Stability at 60° C</td>
<td>kN</td>
<td>5.2</td>
<td>4.4</td>
</tr>
<tr>
<td>Flowing index</td>
<td>mm</td>
<td>1.7</td>
<td>1.4</td>
</tr>
<tr>
<td>Swelling at 28 days</td>
<td>%</td>
<td>0.544</td>
<td>0.450</td>
</tr>
</tbody>
</table>

Table 8: Physical-mechanical properties of the asphalt mixture test samples B.a.31

<table>
<thead>
<tr>
<th>Characteristic name</th>
<th>UM</th>
<th>Bitumen percentage [%]</th>
<th>Relative to normative CD 42-85</th>
</tr>
</thead>
<tbody>
<tr>
<td>Apparent density</td>
<td>g/cm³</td>
<td>2.239</td>
<td>2.214</td>
</tr>
<tr>
<td>Water absorption</td>
<td>% vol</td>
<td>10.108</td>
<td>10.035</td>
</tr>
<tr>
<td>Stability at 60° C</td>
<td>kN</td>
<td>5.1</td>
<td>3.0</td>
</tr>
<tr>
<td>Flowing index</td>
<td>mm</td>
<td>1.2</td>
<td>1.5</td>
</tr>
<tr>
<td>Swelling at 28 days</td>
<td>%</td>
<td>0.325</td>
<td>0.000</td>
</tr>
</tbody>
</table>

Table 9: Physical-mechanical properties of the asphalt mixture test samples A.b.31

<table>
<thead>
<tr>
<th>Characteristic name</th>
<th>UM</th>
<th>Bitumen percentage [%]</th>
<th>Relative to normative CD 42-85</th>
</tr>
</thead>
<tbody>
<tr>
<td>Apparent density</td>
<td>g/cm³</td>
<td>2.211</td>
<td>2.227</td>
</tr>
<tr>
<td>Water absorption</td>
<td>% vol</td>
<td>10.284</td>
<td>9.915</td>
</tr>
</tbody>
</table>

4. Conclusions

Analyzing the results values obtained from the laboratory research and in particular the behavior in time to water action (swelling), due to a good adhesion to natural aggregates of natural bitumen from bituminous sands, the four types of mixtures with and without adding hard paving grade bitumen: the first warm asphalt mix is an asphalt base concrete with crushed gravel, designed for road pavements with light traffic (A.31.nb), the mixture containing added hard paving grade bitumen and the other three bituminous mixture without hard paving grade bitumen in their composition, namely asphalt concrete (open graded asphalt concrete) with crushed gravel for the binder layer (B.a.31.nb.f), asphalt base for bituminous base course (A.b.31.nb.f) and asphalt base designed for light traffic (A.31.nb.f), it can be affirmed that the physical and mechanical characteristics are within the limits imposed by normative CD 42-85. However, given the conditions and the necessity to complete the results with dynamic tests imposed by normative AND 605/2014, it is required to develop an update of the norm CD 42-85.

5. References

1. ***, Normativ Departamental pentru folosirea directa a nisipurilor bituminoua cu si fara adaoas de bitum dur la executarea a cald a straturilor bituminoua rutiere [Departmental normative for the direct use of bituminous sands with and without adding hard paving grade bitumen for the execution of warm asphaltic courses for pavements], CD 42-85, Bucuresti, 1985;
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5. Matasaru Tr., Craus I., Dorobantu St., Drumuri [Roads], Bucuresti, Editura Tehnica Bucuresti, 1966;