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Abstract: The paper is a part of wider research based on the system approach to the problem of modelling and calculations of bolted flange connections. With this approach it is possible to independent consideration of each system’s element in order to find the best model of this element. The aim of this study is to develop a model of the single-bolted joint separated from the bolted flange connection.

An analysis is conducted for the spider bolt model which is an equivalent model corresponding to the spatial bolt model. The key problem in the case of modelling bolts with the spider bolt model is adequate distribution of the preload on the bolt head. Accuracy of modelling bolts using the spider bolt model strongly depends on the way of this distribution. The effect of preload distribution in the spider bolt model on stiffness values of the element fastened in the bolted joint has been examined. The result of actions described in the paper is proposal distribution of the preload on nodes belonging to the bolt head which guarantees the best effects of spider bolt model application.

Keywords: BOLTED JOINT, SPIDER BOLT MODEL, PRELOAD, FINITE ELEMENT METHOD

1. Introduction

The primary task of the machine modelling phase is to find a compromise between the level of model simplification and the expected accuracy of modelling. This is particularly important for modelling complex systems with many elements being in a contact such as bolted connections [2, 8].

The bolted connection can be regarded as a system consisting of subsystems (which are the elements of the connection) [3]. In the case of the bolted flange connection (Fig. 1), these subsystems include: bolts, the flange element, the rigid support and the contact layer between joined elements. With this approach, each of these subsystems can be developed and modeled separately using different methods for modelling.

![Fig. 1 Example of a bolted flange connection](image1)

The most common method of modelling complex structures is currently the finite element method (FEM) [12]. In the papers [4, 6, 7] several different FE-models of bolts are presented, which can be used for modelling bolted flange connections. These include:
- the no-bolt model,
- plane models,
- the coupled bolt model,
- the rigid body bolt model,
- the spider bolt model,
- spatial bolt models.

In the present study, the spider bolt model (SB model) is assumed as the model of the bolt. This is an equivalent model corresponding to the reference spatial bolt model (3DB model), which according to experimental tests deliver the best accurate results of modelling [11].

One of the fundamental aspects of modelling bolts with the SB model is adequate distribution of the preload on the bolt head. It has a significant impact on the accuracy of joint modelling. The aim of this study is to investigate the effect of preload distribution in the SB model on stiffness of the flange element fastened in the bolted joint. As a result of the research proposal distribution of the preload on nodes belonging to the bolt head which guarantees the best effects of SB model application is presented. All calculations were conducted with use of the Midas NFX 2014 finite element software.

2. Models of the bolted joint

The tests were performed on the example of a fragment of the bolted flange connection shown in Fig. 1. The considered joint consists of a deformable flange element mounted with a single bold M10 made in the mechanical property class 10.9 to a rigid support. Thickness of the flange element \( h \) is equal to 30 mm. The preload of the bolt \( F_p \) is equal to 17.2 kN and it was set down based on Polish Standard PN-EN 1591 [9]. The surface area of preload acting \( A_n \) is equal to 69.75π mm² and it was set down on the base of Polish Standard PN-EN ISO 7091 [10].

In the FEM-based models of the joint, occurrence of the contact layer between connected elements is omitted. For the construction of discrete models standard finite elements are used. The joined element model and the spatial bolt model are created with 3D finite elements. In contrast, in the SB model the plain part of the bolt and its head are modeled with use of beam elements but the total volume of beam elements for the head is assumed to be equal to the volume of the head of the bolt in the 3DB model. Developed discrete models of the bolted joint are shown respectively in Fig. 2 and Fig. 3.

![Fig. 2 Bolted joint with the spider bolt model](image2)

Methods of models loading are shown in Fig. 4. In this figure the following new designations are used:
There is no simple formulas for calculating stiffness of the joined flange element $k_f$. To designate it the most frequently the finite element method is used. Then stiffness of the joined flange element can be defined based on the relationship (for a review, see [5])

$$k_{f,j} = \frac{F_m}{\delta_{j_{ave}}}$$

(7)

where $\delta_{j_{ave}}$ – average normal displacement of nodes lying in the surface area $A_n$ under the action of the preload $F_m$. $j$ – symbol of the model of the joint, $j \in \{SB, 3DB\}$.

Stiffness values of the joined flange element for both models are given in Table 1.

**Table 1: Stiffness of the joined flange element as a function of the bolt load**

<table>
<thead>
<tr>
<th>$\alpha$</th>
<th>$\beta$</th>
<th>$k_{f,SB}$ [MN/mm]</th>
<th>$k_{f,3DB}$ [MN/mm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1,0</td>
<td>0</td>
<td>2,72</td>
<td></td>
</tr>
<tr>
<td>0,9</td>
<td>0,1</td>
<td>2,47</td>
<td></td>
</tr>
<tr>
<td>0,85</td>
<td>0,15</td>
<td>2,61</td>
<td></td>
</tr>
<tr>
<td>0,8</td>
<td>0,2</td>
<td>2,78</td>
<td></td>
</tr>
<tr>
<td>0,75</td>
<td>0,25</td>
<td>2,96</td>
<td></td>
</tr>
<tr>
<td>0,7</td>
<td>0,3</td>
<td>3,18</td>
<td></td>
</tr>
<tr>
<td>0,6</td>
<td>0,4</td>
<td>3,70</td>
<td></td>
</tr>
<tr>
<td>2,90</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

The relative difference between the $k_{f,SB}$ and $k_{f,3DB}$ values can be analyzed on the basis of the $W$ index

$$W = \frac{k_{f,SB} - k_{f,3DB}}{k_{f,3DB}} \times 100$$

(8)

Calculated $W$ index values are summarized in Table 2. Based on these results it can be concluded that the 3DB model in the best way may be replaced by the SB model, when

$$\frac{\alpha}{\beta} \epsilon (3+4)$$

(9)

**4. Conclusions**

In the paper an analysis of the bolted flange connection for named models created with use of the finite element method was carried out. Two types of the connection were examined: with a bolt modeled using spatial elements and with a bolt modeled as the spider bolt model. It has been shown that the spider bolt model can be successfully applied as a substitute model for the reference spatial model by adopting the appropriate preload distribution.

Investigations of bolted flange connections are often conducted in the aspect of the selected problems. Then knowledge of the distribution of stress and strain levels in all elements of the connection is not needed. In the case of FEM analysis of stiffness of bolted flange connections, it is better to use simplified models of bolts and bolted flange connections, as demonstrated in this paper.

**References**


10. PN-EN ISO 7091: 2003, Plain washers, Normal series, Product grade C.


PARAMETRIC ANALYSIS OF THE SHIP CAPSIZE PROBLEM

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Abstract: The non-linear ship capsize equation derived by Thompson et al., that incorporates both direct and parametric excitation, is examined numerically in an attempt to deepen our understanding on the influence of the parameters involved in the final ship’s response. Because our interest is focused on the binary outcome of capsize-non-capsize, no remark of the steady-state onto which a non-capsize motion may settle is made. The four-dimensional phase-control space includes the non-dimensional damping coefficient, the ratio between wave frequency and ship’s natural frequency, and the direct and parametric forcing amplitudes. All the computed boundaries between capsizing and non-capsizing regions in bi-dimensional projections of control parameter space show fractal features.

Keywords: TRANSIENT SHIP CAPSIZE, BEAM SEA, FRACTAL BOUNDARY, PARAMETRIC STUDY

1. Introduction

Capsizing or keeling over is that catastrophic situation in which a ship is turned on its side. It is responsible every year for a lot of material damages and losses of human lives. To prevent such events, a better understanding of ship stability is necessary.

The nonlinear behavior of ship motion leading to capsize has been extensively studied in the last three decades by many researchers using mathematical models in conjunction with theoretical developments in the dynamics of nonlinear systems. Thus, Thompson thought ship capsizing as an escape from a potential well and introduced innovative concepts like transient capsizability [1]. Other authors, including Sanchez and Nayfeh [2], Kan and Taguchi [3] etc., focused on understanding fundamental mechanisms of capsizing with rather simplified mathematical modelling [4, 5]. Multi-degree-of-freedom models have been proposed by Spyrou [6], Vassalos et al. [7], and Oh et al. [8]. Experimental investigations were made by Bird and Odibasi [9], Umeda et al. [10], Hamamoto et al. [11], and others.

Both theoretical and experimental studies have identified several physical mechanisms of ship capsizing, including pure resonant rolling, parametric excitation, broaching, and loss of stability at a wave crest. It was emphasized the fact that more attention should be pay to capsize under transient, rather than steady-state conditions. This situation corresponds to a short train of regular waves impinging upon the ship in otherwise relatively calm weather conditions. Dangerous large-amplitude motions and, finally, capsizing can appear when the ship is hit by no more than 8 – 10 sufficiently steep waves [12].

In the paper, we concentrate on the archetypical single degree of freedom oscillator used by Thompson et al. to model the ship capsizing under direct and parametric wave excitation. The model assumes a linear damping and a restoring moment curve represented by a second order polynomial. Besides direct forcing, derived from the rotational acceleration of the wave normal, the model equation includes the parametric excitation, generated by the fluctuating gravitational field [13]. The parameters space is four-dimensional and has as components the non-dimensional damping coefficient, the ratio between wave frequency and ship’s natural frequency, and the direct and parametric forcing amplitudes.

We content ourselves here with an extended numerical investigation on the role played by each of these parameters on transient ship capsize. We will not make any reference to the long term behavior of ship, this issue being studied in a companion paper [14].

2. Capsize equation

In present work, the following non-linear second-order differential equation derived by Thompson et al. is numerically investigated in an attempt to understand the effect on ship capsizing of every parameter involved:

\[ \ddot{x} + \beta \dot{x} + \left( x - x_0^2 \right) \left( 1 + G \cos \omega t \right) = F \sin \omega t \]  

Here, \( x \) is the ratio between roll angle and the angle of vanishing stability, \( \beta \) is the linear damping coefficient, \( \omega \) the ratio between wave frequency and ship’s natural frequency, while \( G \) and \( F \) stand for direct and parametric forcing amplitudes, respectively. A dot denotes differentiation with respect to non-dimensional time \( t \). The derivation of Eq. 1 can be found in [13].

In the paper, a special attention was paid to the range of possible magnitudes and signs for the amplitudes \( F \) and \( G \). Thus, if the wind and wave acting on the ship travel in the same direction, then \( F \) and \( G \) have the same sign, while if the wind and wave travel in opposite directions, \( G \) and \( F \) have different signs. If the wave propagates to the right then \( F \) is negative, while the sign must be considered positive if the direction of wave propagation is to the left. Finally, \( F/G \leq 1 \) if the ship sails in deep water, \( F/G < 1 \) for oblique waves, and \( F/G > 1 \) for shallow water.

In [14], the periodic solution of Eq.1, describing the long term ship behavior, were approximated by using Fast Fourier Transform and Harmonic Balance Technique. As we have already pointed out, in the next section the transient behavior will be subject to our analysis.

3. Parametric studies on capsize equation

Experimental studies and data from the reported capsizing conducted to the conclusion that the worst-case scenario and, in the same time, a more realistic representation of a sea state consists in a short sequence of no more than ten steep waves hitting the ship and not in a long pulse of regular waves. In conclusion, the transient response is essential for the analyses problem. If capsize not occur within 8 – 10 cycles of forcing than it is unlikely to appear in the following cycles.

Fig. 1 The phase trajectory of a) a non-capsizing oscillation; b) a capsizing oscillation, for \( \beta = 0.1, \omega = 0.9, G = 5F \) and \( F = 0.1 \), respective \( F = 0.15 \)

To verify this assumption, equation (1) has been numerically integrated by use of a fourth order Runge – Kutta - Gill procedure with constant step, starting with zero initial conditions, and for a time interval equal with ten cycles of the forcing (having period...
For a specified set of parameters $\beta, \omega, G$, and variable $F$, the system (1) evolves to a limit cycle for small forcing amplitude $F$ or goes out in the phase plane, like a spiral with extending amplitude, for sufficiently large values of $F$. The second case corresponds to capsizing (see Fig. 1).

### 3.1. Influence of the ratio of the forcing magnitudes on capsizing

We tested first the effect of ratio $G/F$ on the ship capsizing. The parameters $\beta$ and $G/F$ were maintained fixed, while $\omega$ and $F$ were slowly increased as follows. For a given $\omega$ between 0.4 and 2.0, $F = 0.0$ and equilibrium conditions, $x(0) = x(0) = 0$, equation (1) has been integrated for the time interval $[0, 10]$. If at the end of this process the solution does not diverge, then the pair $(\omega, F) = (\omega, 0)$ is classified as safe and a small black rectangle is drawn around it in the parameter plane $(\omega, F)$. Otherwise, the rectangle is colored in white. If the pair is safe, F is increased by a small amount $\Delta F$, and the procedure is repeated. When the solution diverges to infinity, a small amount $\Delta \omega$ is added to $\omega$ and $F$ is reset to zero. By continuing this algorithm until $\omega$ becomes 2.0, a diagram like in Fig. 2 is obtained. The pairs $(\omega, F)$ in Fig. 2 were selected from a vast set having 201 x 201 equally spaced elements.

Fig. 2 allows us to draw at least three observations. First, it should be noted that, whatever the ratio $G/F$ is considered, the most dangerous area for capsizing is that explored by Thompson et al., i.e. the range of frequencies near resonance, $0.8 \leq \omega \leq 1.0$. For frequencies far from resonance ($\omega > 1.5$), the forcing amplitudes for capsizing are ten times larger.

Second, for large $\omega$ the danger of capsizing is lower for shallow waters ($G/F < 1$) than for deep waters ($G/F \geq 1$) or oblique waves ($G/F > 1$). Third, boundaries between capsizing and non-capsizing regions show fractal shapes.

These conclusions are more clearly shown in Fig. 3, where the parameter plane $(G, F)$ is presented for fixed $\omega$ values near and far from resonance.

![Fig. 3 Parameter plane $(G, F)$ for $\beta = 0.1$ and $a) \omega = 0.9$; $b) \omega = 1.8$](image)

### 3.2. Influence of the damping on capsizing

Let now clarify how the damping affects the capsizing phenomenon. To do this, the response of the system (1) has been investigated in the parameter plane $(\omega, F)$ for the fixed ratio $G/F = 5$, and different damping coefficients $\beta$. Some of our results are displayed in Fig. 4 (see also Fig. 2(c)). As expected, as $\beta$ increases, the system resists better to the external stimuli, in that the forcing amplitudes for capsizing are moving to higher values. The most sensitive area continues to remain that near the resonance.

![Fig. 4 Examples of $(\omega, F)$ parameter control planes for $G/F = 5$ and different damping coefficients: $a) \beta = 0.04$; $b) \beta = 0.25$](image)

Another perspective on the influence of damping on capsizing is revealed in Fig. 5, where the parameter plane $(\beta, F)$ is presented instead, for fixed $\omega$, and $G/F$. It is worth to note again the fractal boundaries between capsizing and non-capsizing regions.

![Fig. 5 Examples of $(\beta, F)$ parameter control planes for $G/F = 5$ and different non-dimensional frequencies: $a) \omega = 0.9$; $b) \omega = 1.8$](image)

### 3.3. Influence of the initial conditions on capsizing

The role played by initial conditions $(x(0), x(0))$ on capsizing has been intensively studied in the literature by building the so-
called basins of attraction (formed by all initial conditions that do not lead to capsizing). Here, we consider two sets of initial conditions, namely (0, 0) and (-0.3, 0), and integrate equation (1) for ten cycles of forcing, \( \beta = 0.1 \), \( \varphi \in [0.9, 1.8] \), and different \( G \) and \( F \). If both initial conditions are safe, a black rectangle is drawn in (\( G, F \)) plane around the considered pair of forcing amplitudes. If only first set is safe, the rectangle is colored in red, while it is colored in grey if the second set is safe. For unsafe pairs, the rectangle remains white. Our findings are shown in Fig. 6. It seems that initial conditions significantly affect the safe area, especially at the boundary with unsafe area.

![Fig. 6 Influence of the initial conditions on capsizing. Examples of (G, F) parameter control planes for \( \beta = 0.1 \) and different non-dimensional frequencies: a) \( \varphi = 0.9 \); b) \( \varphi = 1.8 \).](image)

### 3.4. Influence of the transient length on capsizing

In all the numerical simulations described above we considered the transient motion of no more than ten cycles of the forcing. If capsizing not occur within this time period then it is likely that it will not occur in the next cycles. The last numerical results we report in the paper refer just to this assumption. Equation (1) was numerically integrated with zero initial conditions for the same sets of system parameters as given in the previous sub-section. This report in the paper refer just to this assumption. Equation (1) was numerically investigated in order to determine the influence of the four parameters involved in the capsizing or non-capsizing response of the system. The main conclusions of the study are as follows:

a) Whatever the damping coefficient and the ratio between the parametric and direct forcing amplitudes are, the most dangerous frequencies for capsizing are those near the resonance. Far from resonance, the probability of capsizing is higher for deep water and oblique waves than for shallow water;

b) The system behaves much better in terms of capsizing for larger damping coefficients;

c) Initial conditions and transient length affect somewhat the safe area in the parameter space but the changes only occur in the border area, which is better to be avoided;

d) The boundaries between capsizing and non-capsizing regions in bi-dimensional projections of control parameter space show fractal features.

### Literature


DEVELOPMENT OF A CAR BODY DESIGNING PROCESS

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Abstract: Car industry is one of the first industries that started using computers in the designing process. A replacement of a conventional designing process has been a long-lasting procedure, especially regarding collaboration. In addition to the necessity for designers’ training and changes in the way of thinking, considerable investments were needed as well. The ambitious goals were not accomplished after all, since the entire process had been conditioned by a large number of participants at various development levels. In this paper, the differences between the conventional designing, designing by computers and virtual designing are presented. A new model was produced, without manufacture of classic prototypes which were replaced by vehicles from the test production. There are the upcoming analyses of the effects, which will certainly influence further development of the designing process.

KEYWORDS: CAR BODY, DESIGNING, NOVELTIES.

1. Introduction
Car industry was one of the first industries that started using computers in the designing process. It has been a very long procedure, with transition phases and necessity for maintaining the conventional designing in transition period. The situation and possibilities for collaboration, especially regarding smaller suppliers, have been the additional complication. In the beginning of the car designing process, the computer was mainly used for designing of a car body, interior and exterior parts, which was the consequence of a very complicated classic designing process, so the application of the computer contributed to a considerable shortening of time of designing; using the example of a car body, this is accomplished:

• By analysis of several versions of the carrying construction in the initial designing phase. Based on calculation results, several versions are selected and the most favourable among them undergoes further optimization.
• By designing a car body on the computer, manufacture of a prototypes batch can be reduced, which considerably reduces the costs of sending a new vehicle model into the production.
• By rapid analysis of effects of new materials application.
• By a public display of activities related to this area, which will contribute to improvement of the image of the company, which is increasingly more used for marketing etc..

2. Conventional designing method
A project task is defined according to the market demands. This part of preparatory activities all the way to project task defining is the same for all designing methods. When designing a new model, i.e. car body, it is necessary to carry out a lot of preparatory activities before a project task is defined. One of the most important activities is market research, as well as analysis of production initiation. Fig.1 shows a stylistic study of the exterior and study of the interior of a vehicle, based on which analysis of market was performed.

After that, in conventional designing several plaster models are made in proportions 1:5, see fig. 2. After several proposals are observed, two models are adopted, which are then made out of gypsum in proportion 1:1, see fig. 2. Model of vehicle interior, with completely defined interior regarding style, is also realized in proportion 1:1. After presentations, considerations and acknowledgment of comments, model 1:1 is adjusted, i.e. a final version is made, with controlled dimensions and finishing paint. Model has the form of the vehicle. After that, the model undergoes final investigations in aero tunnel, see fig. 3.

Fig. 3 Investigations

After the decision is made to accept one model, designing of carrying construction of a car body starts. It is necessary to define the positions of all the supports and to propose the shape of their cross sections and inter connections applying the concepts used on previous models and used by competitors. Lately, major attention is given to the protection of space for passengers, where car body plays one of the major roles. Most decisions made at this time are based on experience acquired by participation in previous concepts and constructions, subjective grades and similarities with already existing constructions. Only after execution of such an analysis can the presented stylistic model be accepted, construction of car body initiated and, later, a certain number of prototypes made. In the following cycle, car body prototypes are subjected to laboratory investigations, first, and then to travel investigations, both individually and as parts of vehicle. One of the first investigations of a car body, performed in laboratory conditions, is bending and torsion static investigation of car body, see fig. 3. Laboratory investigations which are implemented can be related to investigation of a car body behaviour, connections of movable car body parts, connections of elements which are attached to a car body, adjustment to regulations etc. In addition to car body investigations, these parts are also investigated individually. Based on previous experience and legal regulations valid at the period of designing, i.e. production initiation, critical points are modified. After that, a new batch of prototypes is made. The investigation cycle is repeated on the new batch of modified prototypes. Construction documentation is modified, if necessary, based on results of repeated investigations. In the last part, homologation, i.e. verification of the accepted model is needed, for which a master (clay) model is made. Car body is assembled on assembly line and painted on painting line, fig. 4. The operations are mainly performed manually. At the same time, painting of plastic parts is organized, on car body painting line or parallel to this line. Assembling of vehicle is organized on assembly line. It is done.
manually, where equipment for assembling is defined on special points. Conventional concept reflects the accomplished technological level at the preparatory period and is adjusted to the size of batch and need to employ a large number of workers in the overall process.

3. Car body designing by using a computer

The process of designing of passenger car body by using a computer is very different from a classic designing method [1].

The differences in the designing method can be observed at the very beginning of the process of defining large surfaces of exterior or interior parts. Already in this phase the stylist is able to simulate the behaviour of vehicle in the aero tunnel and to eliminate the weak points which influence the definition. In computer designing, the stylist has the opportunity to facilitate the job for the constructors significantly and to facilitate the manufacture of prototypes by defining style (of the exterior or interior).

The advantage of such designing is the possibility to analyze the main and alternative concepts in a short time, in the initial designing phase. Based on results of comparative analysis, we can eliminate unfavourable concepts already in the initial phase. The results and reliability of the analysis depend on accuracy and possibilities of the selected programme, mathematical model quality and the knowledge of the analyst. In the first few analyses we can already determine the concentrated strain points and, if needed, the necessary modifications of the observed model can be performed quickly, in order to obtain the most favourable distribution of strains and movements on carrying construction elements. After the analysis, an optimized carrying construction is obtained, fig. 6. After that, the constructors define the car body, see fig. 7. In this designing phase, reduction of the total car body designing time can be influenced considerably, because manufacture of the first batch of prototypes can be avoided, which is necessary in classic designing. This is achieved by simultaneous work of stylists, calculators and constructors, the results of which is an optimized construction. After that, a batch of prototypes is made and the further procedure is the same as for conventional designing. The greatest progress is achieved in the manufacture of prototypes, which contributes to a considerable reduction of total designing time. All the models are made on numerical machines, where the results of previous designing are used, especially in manufacture of large surfaces. Quite often, with the aim of rapid comprehension of designing effects, polyurethanes of small hardness, easily formed, are applied. In that way style or construction can be tested or perceived. In addition to that, new methods for prototypes manufacture have been developed (rapid prototyping etc.), due to which testing of dimensions is easily performed, and functional testing can also be realized if better materials are applied. Due to the introduction of new technologies, there is no longer a problem of negative angles, complicated geometry and similar, due to the previous testing of their assembling. Defined shapes of parts on the computer can be used for manufacture of models, prototype parts or tools for prototype parts manufacture, and also later on, for manufacture of small batches.

After the manufacture of prototypes, investigations are performed. The investigation procedure is the same as for conventional designing, except that it is realized with improved methods and on innovated equipment. Investigation methods must be constantly improved since the valid regulations are getting stricter all the time, as well as market requirements. One part of investigations is transferred to collaboration, while the grading tests are realized at the finalist’s.

Nowadays, it is unthinkable to manufacture a large batch of car bodies without using a robot. Initially, the robots were used for spot
welding of sensitive points and points that were hard to reach. After that, the application was expanded to assembling of major connections and final assembling of a car body. At the present time, mainly the whole lines for car body assembling are robotized. For points with laser welding, robots were applied even earlier. There is a constant request for the reduction of these welding points. Even for car bodies assembled in this way, additional finishing is necessary, which is still manual. Production without errors is still only a goal.

The process of vehicles assembling can be organized in various ways, depending on the producer. Fig. 13 shows the typical assembly line, where the process has been improved in the sense of better logistics and application of modern manual assembling tools. Introduction of new technologies, such as glass bonding, requires the change of assembling process which is realized on that work place. The application of a manipulator is frequent and the same ones are used for assembling of large and heavy sets (instrument panels, doors...). Robots are also used in the assembling process, for more complex operations and in large batches.

In recent times, structural adhesives are increasingly more used for connecting of car body parts. For initiation of such a method for connecting of car body parts, it is necessary to change the existing connecting technology completely, with obligatory preservation of very strict production conditions. Regarding the improvement of production quality, the conditions for control of finished car body and for introduction of laser control on the car body assembly line itself were enhanced, see fig. 11. This control is more important from the aspect of parts installation in the assembling process in which robots are used. Painting of a car body is being given more and more attention with the purpose of getting as good appearance as possible and reducing errors to zero. Painting of a car body is mainly robotized, see fig. 12.

Big producers are leaving aggregates designing to collaborators nowadays. Conditions for installing are defined by the finalist. A collaborator is responsible for the success of his/her construction, such as fulfillment of valid regulations, elimination of all the errors in the batch and a guarantee period. The goal of each collaborator is to have his/her construction used in the first installing by which he/she acquires a reference, secure income and profit from the spare part. Along with the construction, calculations, i.e. simulations of certain processes, important for good operating of the vehicle, are realized, fig. 15. All technology departments are directly included in
the development process (manufacture of parts, safety, assembling, installing, quality, lifetime). Fig. 16 shows the simulation of the installation process, i.e. ergonomic studies at this work place.

In this phase of the project, verification of technologic properties of the car part is necessary, as well as the construction of tools needed for manufacture of the car part. In addition to that, conditions of car part manufacture are simulated, especially for plastic parts, i.e. flowing process is simulated. Due to the simulation of parts manufacture, it is possible to eliminate possible problems in this phase, i.e. to eliminate finishing works on the tool which could consume a lot of time.

Car industry strives to the production without the defective parts. For those reasons, the system of quality is getting increasingly greater attention. Advanced softwares are extremely useful in this process.

Company FIAT, as one of the leading companies worldwide, was the first to introduce a single model into a batch production (vehicle Fiat 500L), without manufacturing a batch of prototypes necessary for developmental investigations and project verification.

By the initiation of the production of model Fiat 500L, the new era in the batch car production began – introduction of the new model into the production without manufacturing a batch of prototypes.

In 2007, company FIAT started the production of model Fiat 500, and in 2012 the production of model Fiat 500L started. In 2013 a new model with seven seats was promoted, Fiat 500L trekking, a partially off-road vehicle.

The mark of both cars (500, 500L) indicates that these are two mutually connected models of vehicle, i.e. that model 500L was developed out of model 500.

For vehicle 500L, a concept of virtual designing was applied. According to information from company FAS site, the cancellation of batch of prototypes was accomplished due to:

- new architecture "Small Wide"
- thousands of hours of virtual simulations
- 200 tests of components and subsystems
- > 100 impact tests.

It was not specified which subsystem tests were carried out and which were related to a car body, and in which phase of car body assembling. Over a hundred simulations of impact tests and a good basic vehicle were not sufficient to reduce the number of impact tests, which caused prolongation of the initiation of the production of the vehicle, modifications of the vehicle and increase of costs.

In the applied concept, the main project was handled by the finalist – starting with style, 3D forming, creation of a research, all the way to calculation and process simulation. The entire process was realized on the computer, by application of modern designing packages. The process was simplified by the fact that the data from project Fiat 500 were used (modified platform, developed and tested engine compartment etc.). In addition to all these advantages, over 1000 virtual simulations necessary in this phase of the project were realized. The processes of all simulations were probably shortened considerably, since no major problems were to be expected.

A modern designing process implies a complete responsibility of the finalist for the definition of the vehicle platform.

5. Conclusions

A new era began in the designing and development of vehicles – production initiation without a classic vehicle. Vehicle prototype was replaced by a test batch vehicle. Such a procedure probably shortens the time of new model initiation and reduces the development costs. Responsibility of the finalist is partly transferred to other participants in the production process (components producers, equipment producers etc.). In the following period, greater centralization of the development can be expected for some finalists.

6. Literature

SUGGESTION OF SPARE PARTS INVENTORY THEORY MODEL IN ARMY

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Abstract: The article introduces a model which might be used for supplying spare parts of ground forces technical equipment to the Army of the Czech Republic. When designing inventory management, an Economic Order Quantity (EOQ) model is applied. It is a classical model used to determine stock order quantity. When setting the model, stock classification based on the ABC method and the EW matrix is made. Following the stock classification, for the Army of the Czech Republic we would suggest determining a signal stock level which is the same as for the Q-model of a stock management system. In this model used to optimize the stock we apply a dynamic model with absolutely determined stock movement, a dynamic multiproduct model where purchase cost is constant, and a dynamic model completed by a temporary unsatisfied demand. The important aspects which should be taken into consideration when setting an inventory model are stock purchase cost, storage costs, stock holding cost, or the cost which arise due to a short supply.

KEYWORDS: ECONOMIC ORDER QUANTITY, INVENTORY THEORY MODEL, SPARE PARTS, P AND Q – SYSTEM OF SUPPLY CONTROL.

1. Introduction

Inventory theory might be described as a set of mathematical methods used to model and optimize the processes of generated items to ensure smooth company running. When determining the strategy under certainty conditions, it is necessary to assess and offset ordering cost against stock holding cost. The strategy to order a large amount less frequently might often result in inventory holding cost increase which can be higher than ordering cost savings. The ordering cost consists of settling stock transfer, product handling cost, product storage costs, and settling an invoice cost. The inventory holding cost consists of the capital tied up in stocks cost, service cost (insurance, taxes), storage facilities cost, and also the cost of stock deterioration risk.

When setting an order strategy the aim of which is to minimize total inventory holding costs and ordering costs, it is possible to use Economic order quantity (EOQ) [1]. The model EOQ is a concept which determines optimal order quantity based on ordering costs and holding costs. The optimal order quantity is when incremental ordering costs equal incremental holding costs. Cost relations which are to be taken into account when setting economic, i.e. optimal order quantity.

One of the basic characteristics in stock management models is the nature of the observed stock unit demand. This demand might be described by different models which can be divided into two basic criteria:

1. When taking into account the way of determining an enquiry (consumption) level and lead time length, there are:
   a) deterministic models which assume the enquiry (consumption) quantity and lead time length to be precisely known,
      - Q - system of supply control – the delivery frequency changes while the delivery size is constant,
      - P - system of supply control – the delivery size changes while the delivery interval is fixed.
   b) stochastic models based on the probabilistic character of enquiry (consumption) and the length of lead time.
   c) non-deterministic models where the character of enquiry (consumption) and lead time is not known.

2. When dealing with the way of renewing supplies, there are:
   a) static models where the supply is made by one-time delivery,
   b) dynamic models where an item supply is kept in stock on a long term basis and is renewed by frequent deliveries.

The supply theory models introduced above use a basic evaluation criterion based on minimizing the overall costs of purchasing, storing and keeping supplies, and in some cases even short supply costs.

2. Results of discussion - Design of Dynamic Inventory Theory Model with Determined Movement of Spare Parts

In order to calculate inventory theory optimization within the conditions of the Army of the Czech Republic, we would suggest using a dynamic model with absolutely determined spare parts movement for the orders of separate spare parts, a dynamic multiproduct model with a constant amount of cost including supply purchase for the associated orders of spare parts, and a dynamic model completed by a temporary unsatisfied demand.

2.1 Proposal of a Model for Controlling Spare Parts Vehicles Stock

The model proposal is based on the EW matrix which includes the ABCD analysis completed by the XYZZ analysis. The ABCD analysis will be used for classifying spare parts by a material price during purchase. The group A stands for the material of the highest price and the group D on the other hand means that the material is of the lowest purchase price. Next we will classify the material into XYZZ groups by stock rotation per year. The group X means that spare parts are of the highest rotation. On the other hand the group Z stands for the spare parts with zero rotation. Along with these two criteria we select one more criterion which expresses delivery time of a spare part. The marking I stands for a short delivery time of spare parts, while the group labelled as IV means that the delivery time of spare parts is the longest.

On the basis of monitoring and then analyzing the criteria we can classify the assortment of spare parts into 64 groups. Next, the stock rotation of single spare parts and presumed term of delivery are determined. After the spare parts classification according to the described model is performed, a signal level amount will be calculated for each group at a certain risk level in terms of early delivery and determination of delivery optimum amount. This risk is to be considered in the light of purchase cost and spare parts stocking, and that is the main criterion when it comes to selecting a proper model. In the military area it is necessary to take into account the coefficient of vehicle technical availability which serves as input information for determining a safety stock level. This is most closely related to the calculation of stock provision reliability, see Chapter 2.6.
2.2 Selected methods dealing with prediction of spare parts demand

Next step when setting optimal spare parts stock control is to forecast the future consumption of items in stock. The forecast is based on the consumption history which should be representative, or, in other words, long enough. In case of spare parts we regularly work with the history of five to ten years, in case of occasional or, in other words, long enough. In case of spare parts we regularly based on the consumption history which should be representative (generally the following rule applies: the longer the history the more accurate and reliable the forecast). The methods used to calculate:

a) Moving Average Method.
b) Moving Weighted Average Method.
c) Least Squares Method.
d) Brown’s Method.
e) Holt-Winters Method.
f) Smart-Willemain Method.
g) Smart-Willemain Method.

In 2002 Smart and Willemain introduced a simulation statistical method which addresses the stochastic prediction of future consumption. With this method it is possible to set a minimum stock level (a reorder level) so that the demands could be met with a determined probability (logistics service). The Smart-Willemain method is based on random sampling from a consumption history (in statistics this procedure is called bootstrapping).

2.3 Dynamic model with absolutely determined stock movement

It is a model which assumes that the quantity of enquiry is exactly known in advance. The aim of the optimization is to determine an optimal delivery size \( x_{opt} \) for which the total costs connected with delivery acquisition, keeping and storing supplies \( N_e(x_{opt}) \) during the period of a length \( T \) will be minimal.

If it is necessary to purchase \( Q \) units during the period of a length \( T \) and the supply is regularly renewed by the delivery of \( x \) units, then the number of deliveries \( v \) per a given period might be expressed by the following formula [1]

\[
    n = \frac{Q}{x} .
\]

The costs used for acquiring all deliveries \( N_p(x) \) are given by the product of a delivery amount \( n \) and one delivery costs \( c_p \) [1]

\[
    N_p(x) = nc_p = \frac{Q}{x}c_p .
\]

The costs used for keeping and storing supplies \( N_s(x) \) during a period \( T \) depend on the quantity of average supply \( x \). An average stock level \( \bar{x} \) equals exactly the half of a delivery size \( x \). The storing costs can be put as follows [1]

\[
    N_s(x) = \frac{x}{2}c_s .
\]

where \( c_s \) is the cost which includes maintaining and storekeeping per one order.

Along with the increase in delivery size, an average carry over supply and consequently the total costs of keeping and storing supplies grow too. By summing both cost items we obtain a total cost function \( N_e(x) \) which might be expressed in the following manner [1]

\[
    N_e(x) = \frac{Q}{x}c_p + \frac{x}{2}Tc_s .
\]

Now we set the first derivative of formula (7) equal zero and we get the Harris-Wilson equation which is put this way

\[
    x_{(opt)} = \sqrt{\frac{2Qc_p}{Tc_s}} .
\]

When we put an optimum delivery size expression \( x_{opt} \) in a cost function \( N_e(x) \), we will get the formula used for calculating minimum total costs

\[
    N_e(x_{opt}) = \sqrt{2QTc_s c_p} .
\]

Optimum length of a re-order cycle \( t_{(opt)} \) might be expressed by the formula [1],

\[
    t_{(opt)} = \frac{T}{x_{(opt)}} = \sqrt{\frac{2Tc_p}{Qc_s}} .
\]

Besides total minimal so-called optimum costs it is necessary to determine the time when an order is to be raised to get a new order to the store in time. Therefore it is essential to set an optimum so-called ordering supply level \( r_0 \)

\[
    r_0 = Q_{(opt)} - mx_{(opt)} .
\]

where: \( t_p \) – the length of order lead time, \( t_c \) – the length of a delivery cycle, \( m \) – the amount of orders a route.

The value \( m \) will be calculated the following way

\[
    m = \frac{t_c}{t_p} .
\]

The calculated value \( m \) will be rounded up to whole numbers.

When following this way, the requirements listed below are to be observed [1]:

- enquiry (consumption) should be known and constant,
- supply collection cannot swing,
- supply renewal is one-time in the form of an optimum delivery size,
- acquiring and storing costs are to be stabilized,
- purchase price is independent from order quantity,
- optimum delivery size is calculated for each supply item separately.

2.4 Dynamic multiproduct model with constant stock purchase

This model assumes the customer will order more items to a store at once. On that account, however, an optimum delivery cycle as well as the optimum delivery size of single items will be abandoned, which can affect the costs of supply keeping and storing. This model presumes that supply acquisition costs do not depend on the number of ordered items. Furthermore, it is assumed that for the length \( T \) period the army needs to order \( k \) items of supplies with expected consumption (enquiry) \( Q_i \) of quantity units. The costs of the keeping and storing of single supply items are \( c_{si} \). The total costs of a group order are then expressed by formula (13), provided that the items will be delivered in equal length delivery cycles \( t \). The number of deliveries \( n \) is to be also the same [3]

\[
    N_e(t) = \frac{T}{t}c_p + \frac{1}{2}t \sum_{i=1}^{k}Q_i c_{si} .
\]
If a function $N_t(t_i)$ derivative by $t_i$ equals zero, then for an optimum delivery cycle length we will obtain the formula below [3]

$$t^* = \frac{2Tc_p}{\sum_{i=1}^{k} Q_i c_{si}}.$$  \hfill (11)

Now it is necessary to determine an optimum structure (amount) of single material items delivery [3]

$$x^* = \frac{Q_{x^*} t^*}{T}.$$  \hfill (12)

After, we will specify the total minimal costs of supply acquisition for item group ordering in the following manner [3]

$$N_c(t^*) = \frac{2Tc_p}{\sum_{i=1}^{k} Q_i c_{si}} x^*.$$  \hfill (13)

Beside the total minimal optimum cost we should also set the time when to raise an order so that the new order could arrive at the storehouse in time. Therefore it is necessary to determine an optimum stock signal level.

$$x_{opt} = Q_{opt} - mx_{opt}.$$  \hfill (14)

where: $t_p$ – the length of order lead time, $t_c$ – the length of a delivery cycle, $m$ – the amount of orders a route. The value $m$ will be calculated the following way

$$m = \frac{t_c}{t_p}.$$  \hfill (15)

2.5 Dynamic model completed by temporary waiting demand

In the dynamic model let us presume that there is a temporary lack of inventory on hand. It means that the demand for particular items might be temporarily unsatisfied. Therefore the re-order cycle splits into two intervals. During the first interval there is a withdrawal form inventory and the withdrawal time is designated as $t_1$. During the second interval there has been the lack of inventory on hand and the demands for a withdrawal form inventory which occur during this interval are not satisfied. The interval length is denoted by $t_2$. The length of a delivery cycle is then

$$t = t_1 + t_2.$$  \hfill (16)

The level of waiting demand during the interval $t_2$ is marked with $s$. This model presumes that this waiting demand will be satisfied immediately after the delivery is in stock. Out of the total supply portion $x \times s$ items will be immediately used for satisfying waiting demands and the rest in amount of $(x-s) \times s$ items will be placed in stock. The maximum inventory held level might be then only $(x-s) \times s$.

The total cost of storage and material acquisition will also include short supply cost $N_s$. It consists of inventory carrying cost $N_{s\alpha}$, acquisition cost $N_{s\beta}$ and short supply cost $N_s$. The inventory carrying cost within one delivery cycle might be expressed as a product of average stock the size of which is in each cycle $(x-s)/2$, inventory carrying cost per unit $c_s$ and the time $t_1$, during which the stock is being withdrawn [4]

$$N_s = \frac{x-s}{2} t_1.$$  \hfill (17)

In each cycle at least one delivery, which is related to acquisition cost $N_{s\alpha}$ amounting of unit cost $c_{s\alpha}$ is acquired. Short supply cost $N_s$ is calculated within one cycle as a product of mean short supply, i.e. $s/2$ of unit cost $c_{s\beta}$ and the time $t_1$ during which the stock is not available [4]

$$N_s = \frac{s}{2} t_2.$$  \hfill (18)

Within one cycle the total cost equals the sum of the three given items. During an observed period (one year for example), however, $Q/x$ cycles pass, where $Q$ is the total yearly demand size when temporary short supply occurs, and $x$ is the size of one delivery during temporary short supply. In order to calculate total cost during all period it is sufficient to multiply one cycle short supply cost by the number of cycles. The resulting cost function will be then as follows

$$N_c(x,s) = \left( c_s \frac{x-s}{2} t_1 + c_{s\alpha} + c_{s\beta} \frac{s}{2} t_2 \right) \frac{Q}{x}.$$  \hfill (19)

This is a function of two variables $x$ and $s$. In the above formula there are besides variables $x$ and $s$ also time characteristics $t_1$ and $t_2$. After some changes the resulting equation is as follows [4]

$$N_c(x,s) = c_s \frac{Q}{x} \left( x - s \right) c_{s\alpha} + c_{s\beta} \frac{Q}{x} c_{s\beta} s.$$  \hfill (20)

The extreme value of the function will be calculated so that the partial derivative by variables $x$ and $s$ equals zero. The optimum values of delivery size $x_{opt}$ and the optimum waiting demand size $s_0$ are the outcome of the above calculation:

$$x_{opt} = \frac{2Q c_p}{T c_s} \frac{c_s + c_{s\alpha}}{c_{s\beta}},$$  \hfill (21)

$$s_0 = \frac{2Q c_p}{T c_s} \frac{c_s}{c_{s\alpha} + c_{s\beta}}.$$  \hfill (22)

In the temporary short supply model providing the demand is not satisfied, the optimum order size will increase, but at the same time the mean stock size will decrease.

The optimum length of a delivery cycle $t_p$ corresponding to the optimum order size $x_{opt}$ is calculated in the following way [4]

$$t_p = \frac{2c_p T}{c_s Q} \sqrt{\frac{c_s + c_{s\alpha}}{c_{s\beta}}}.$$  \hfill (23)

Minimum achievable cost $N$ will be obtained by putting expressions (24) and (25) into the cost function (23)

$$N_{min}(x_{opt}, s_{0}) = \sqrt{2Q T c_p c_{s\alpha} \frac{c_{s\beta}}{c_s}}.$$  \hfill (24)

The optimum signal supply level $x_0$ will be calculated so that we subtract the size of en route order and the number of temporary waiting demands from the expected demand during the time $t_p$ [4]

$$x_0 = Q t_p - mx_{opt} = (x_{opt} - s_{0}).$$  \hfill (25)

2.6 Calculation of supply ensuring reliability

The reliability of ensuring supply is about the way to protect the company against short supply with safety stock. In practice this is assessed by a service level or level of stock availability. However, in both cases it remains true that the higher the level of ensuring reliability, the higher the level of safety stock which grows proportionally.

Service level $a$ expresses the probability ($t_f/t$) that within one delivery cycle no short supply occurs. Logical complement ($1-a$) expresses the probability that the customer demand will not be satisfied. The level of stock availability $\beta$ might be defined as the probability ($t_f/t$) that the order can be fully satisfied immediately after it is exercised from the stock. The complement ($1-\beta$) expresses what relative part of the total demand will not be satisfied when short supply occurs. The values $a$, $\beta$ might be expressed by the following equations [5]:

\[a = 1 - \frac{2}{t_c} \left( 1 - \frac{t_f}{t_c} \right),\]

\[\beta = 1 - \frac{2}{t_c} \left( 1 - \frac{t_f}{t_c} \right)^2,\]
\[
\beta = \frac{x_0}{x_{opt}} = \frac{c_x}{c_p + c_w},
\]
(26)
\[
\alpha = 1 - \beta = \frac{c_w}{c_x + c_w},
\]
(27)

2.7 Setting safety stock level

Determining the safety stock level is most often based on normal distribution of random variables of demand (consumption), deliveries and lead time. However, the precondition of normality is not always fulfilled in practice. Therefore it is advisable to perform a relevant test, e.g. the chi square goodness of fit test, or the Kolmogorov-Smirnov test.

To determine the safety stock level, providing the delivery and demand (consumption) level fluctuate and the length of uncertainty interval is constant, we would suggest using the following method. The fluctuation of the delivery level is expressed by a standard deviation of difference value between inventory on order and actually delivered amount using the formula below [5]

\[
x_p = K \sqrt{\frac{1}{n-1} \sum_{i=1}^{n} (p_i - p)^2},
\]
(28)
where \( K \) – safety factor, \( t_u \) – uncertainty interval, \( \sigma_p \) – standard deviation of lead time length, \( \sigma_x \) – standard deviation of difference value \( r_{xi} \) between inventory on order \( x^* \) and actually delivered amount \( x_i \) according to (37).

The safety factor \( K \) is a determined quantile of a distribution function of a standardized normal distribution. The value of a safety factor \( K \) can be found in charts commonly present in the literature about statistics or by statistic software.

The fluctuation of demand \( p \) and uncertainty interval \( t_u \), or lead time is usually measured by sample standard deviations \( \sigma_p \) and \( \sigma_x \), where \( n \) is the number of observations;

\[
\sigma_p = \sqrt{\frac{1}{n-1} \sum_{i=1}^{n} (p_i - \bar{p})^2},
\]
(29)
\[
\sigma_x = \sqrt{\frac{1}{n-1} \sum_{i=1}^{n} (x_i - \bar{x})^2},
\]
(30)
where \( p \) is the actual demand size.

In case the lead time is too long, it is possible to use an alternative approximation method, since the number of the measured values will be very low. This approximation method is based on the knowledge that the standard deviation of lead time length and thus all uncertainty interval is for different theoretical probability distributions about one-fourth of the range. Then, the following approximate equation applies

\[
\sigma_x \approx 0.25 \left( t_{max} - t_{min} \right).
\]
(31)

The fluctuation of delivery size \( x \) might be expressed either by the standard deviation \( \sigma_x \) of single deliveries size (35), or the standard deviation \( \sigma_t \) of difference value \( r_{xi} \) between inventory on order \( x^* \) and actually delivered amount \( x_i \) according to (36);

\[
\sigma_s = \sqrt{\frac{1}{n-1} \sum_{i=1}^{n} (x_i - \bar{x})^2},
\]
(32)
\[
\sigma_r = \sqrt{\frac{1}{n-1} \sum_{i=1}^{n} (r_{xi} - \bar{r})^2},
\]
(33)
\[
r_{xi} = x_i - x^*,
\]
(34)
where \( p \) – demand variability, \( t_u \) – the interval of uncertainty or lead time, \( n \) – the number of observations.

This method is suitable for determining safety stock for overhead material for which we may derive a consumption level in the following period using the facts of a previous period. In armies there are logistics information systems which enable us to perform an accurate analysis of a material consumption level during a previous period.

3. Conclusion

The aim of the inventory management is to obtain a required service level for a reasonable price. This might be achieved by finding a balance between purchase and storekeeping cost on one side and the cost of providing a required service level by customer request. Generally speaking, if the stock volume is high, the service cost is also high as this is the inventory holding cost. Besides this higher cost also the cost of the supply purchasing is higher. There is a potential of financial loss because of the possibility of investing this sum of money in banking and non-banking products. Conversely, if the amount of inventory on hand is low, inventory holding cost will be also low including supply cost. The disadvantage in this case might be a short supply which can be the cause of secondary cost as a result of looking for a stopgap solution. The cost due to a short supply is often very high especially when the production is stopped. In the army this measure is of no importance from the economical point of view. However, this process might include a technical equipment combat efficiency (dependability) requirement at a determined level.

The best solution is to have low cost of service related to spare parts stocking, and spare parts provided in time. It means that there will be no lack of spare parts because of short supply, or only a segment of spare parts will not be available during a maximally determined period of time. This selection segment is performed according to the ABCD method using EW matrix. The key aspect in the stock management is the ability to deal with the uncertainty of the stock.

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4. References

Abstract: The methods of remote vibration diagnostics realization of metal-rolling department’s equipment will be considered in the article, and also the example of the really set system is resulted. Application of this system allows not only exposing the state of workings parts of mechanism presently, but also carrying out the prognosis of equipment’s working capacity.

Keywords: MONITORING, SENSOR, VIBRATION DIAGNOSTICS, DIRECT SPECTRUM, ENVELOPE SPECTRUM

1. Introduction

Modern industry development produces the promoted requirements to rotating machinery reliability. Therefore on most industrial enterprises, possessing some park of rotating equipment the vibration monitoring and diagnostics systems are implemented. Different firms can make these systems, which could have different configuration, but task for them one is looking after a working equipment, to expose the already present defects of rotation units (basically rolling bearings and journal bearings), and also to make the equipment capacity prediction.

Machines and equipment condition monitoring is one of the most effective methods to decline accident rate and increase technical systems reliability. For the rotating equipment vibration monitoring is very important, because in the process of condition irreversible change there always is a chain of defects and even one of them really changes the equipment vibration background.

The monitoring and diagnostics systems can be both stationary and portable. Functionally these systems do not differ practically. A difference is only in mobility. Application of these systems is gives another very important advantage: due to these systems, equipment repair conducted now on the real condition only.

Thus, in order that equipment always was under supervision it is necessary to produce the equipment with the already specified sensors fastening places of the vibration monitoring and diagnostics systems. Reliable fastening of sensors, assembling and disassembling simplicity, and also outer influence defense them must be concerning the industrial rotating equipment producers.

2. Nature of rolling bearing’s vibration

Rolling bearing’s work in high-speed pump composition and at presence faults in it can influence on a vibration and modulating it processes with the followings fundamental frequencies:

- Rotation frequency of movable ring in relation to immobile: \( f_{rot} \);
  \[ f_{rot} = \frac{1}{2} \cdot f_{rot}(1 - \frac{d_{sr}}{d_{in}} \cos(\alpha)) \cdot z = f_z \cdot z; \]  
  \( f_z \) - solids of revolution number;
- Rolling frequency of solid of revolution on an outer ring:
  \[ f_{sr} = \frac{1}{2} \cdot f_{sr}(1 + \frac{d_{sr}}{d_{rot}} \cos(\alpha)) \cdot z = (f_{rot} - f_z) \cdot z; \]

Where:
- \( z \) - solids of revolution number;
- \( \alpha \) - contact angle of bodies and rolling paths;
- \( d_{sr} \) - solid of revolution diameter;
- \( d_{rot} \) - Diameter of outer ring;
- \( d_{in} \) - Diameter of inner ring;
- \( f_{rot} \) - Diameter of separator;

Expressions (Eq.1, Eq.2, Eq.3, and Eq.4) are evaluating only basic harmonics frequencies in the vibration spectrums and envelope of its high-frequency components at the different types of defects [1].

3. Industrial vibration monitoring and diagnostics systems.

In world industry is used the enormous amount the rotating equipment vibration monitoring and diagnostics systems of different firms.

2.1. Stationary systems.

The stationary monitoring system is needed above all things for a multimode strength equipment, guided an auxiliary personnel. Exactly personnel errors is more frequent than all are the defects multiplying reason of the guided equipment, which it must find out practically instantly (for a few turns of rotor) for failure timely prevention.

On Figure 1 the simplified structure of the vibration monitoring and diagnostics stationary system is shown.

Figure1 Structure of the vibration monitoring and diagnostics stationary system.

1 - The computer with special software; 2 - The Signals transformation card in digital form; 3 - Vibration sensors; 4 - Supervising equipment.
2.2. Portable systems.

Equally with the stationary systems wide application is found the portable monitoring and diagnostics systems (Fig. 2), equipped by expert or automatic troubleshooting routines. These systems can be divided into two basic classes are the extended monitoring systems, including with the expert programs, it means, that diagnostics is executed by the prepared expert, and mass diagnostic systems with the standard rotating equipment automatic condition diagnostic and prediction programs. Such system consists of:

- A portable device (devices), providing vibration measuring and analysis in heavy industrial terms;
- Computer with the program of monitoring, containing a database and fulfilling operations row of signals analysis and processing of analysis results;
- Expert or automatic diagnostic program, processing obtained diagnostic information.

![Figure 2. Vibration portable set on the vibration analyzer C/D-12M base, produced by company BACT (Russia) [2]](image)

4. Design of equipment with the monitoring system sensors fastening places.

All diagnostic equipment listed above has different sensors in the complete set, it are:

- Piezoelectric vibration acceleration transformers (accelerometer);
- Optical (lasers) vibration speed transformers;
- Eddy currents transformers of the relative movement (proximeters);
- Optical or eddy currents revolution sensors.

The more reliable the monitoring and diagnostics system sensor is set, the more precisely parameters taken off and the high-quality working capacity diagnosis for supervised units. Therefore it is very important to produce industrial equipment with the already prepared places for monitoring systems sensors fastening. It can be as screw-threads holes for the stationary system sensors permanent setting as milled places for the portable system sensors temporal mounting by magnets.

5. The system of remote vibration monitoring and diagnostics.

This current system of vibration monitoring and subsequent analysis of the signals is set in one of the steel mills. For example, the system installed on the double shaft, which is necessary for transmitting rotation from the gearbox to the working stand. The system consists of the following components (Fig. 3):

1. Controlled equipment (double shaft);
2. The point of installation of the vibration sensor №1;
3. The point of installation of the vibration sensor №2;
4. The point of installation of the vibration sensor №4;
5. The point of installation of the vibration sensor №3;
6. The connecting cables in the protective shell;
7. Block for information reading or conversion.

![Figure 3. The scheme of vibration monitoring system installed on a double shaft.](image)

Also, the system can be done in two ways:

1. Information reading made by the operator by means of a portable data collector, and then analyzed on a computer. Connecting the collector is made in block established remotely;
2. Information reading is made remotely using a wireless device for reading, conversion and sending information.

The information that comes from the sensors is processed by a special program, or an operator manually compare information obtained on different days, and concludes on the state of the equipment.

The following are comparison examples of direct spectrum (Fig. 4) and the envelope spectrum (Fig. 5) obtained on different days, from the sensor in 1 point.

![Figure 5. Direct spectrum – (dB(A)), \( F_b = 800 \text{Hz} \), Spectrum lines: 1600, Average: 8, measured at point 1 on different days.](image)

![Figure 6. Envelope spectrum – (dB(A)), \( F_b = 200 \text{Hz} \), Band-pass filter \( F_a = 3200 \text{Hz} \) Spectrum lines: 1600, Average: 8, measured at point 1 on different days.](image)

As seen from Fig.5 and Fig.6, vibration levels over time tend to increase. From this we can judge the state of the equipment and make predictions of its working capacity.

Conclusions

Vibration diagnostics of metallurgical or any rotating equipment is very important. This makes it possible to significantly reduce the costs of the equipment, to extend the term of its operation, as well as improve the quality of products. The company must have a diagnostic service that provides vibration monitoring and forecasting the state of the equipment. It is also necessary to establish the system of remote monitoring equipment. The cost of equipment and software for the diagnosis is usually recouped within a year at its regular use.

Literature

METHODOLOGY FOR ASSESSING THE SAFETY OF A GEARBOX UNDER DYNAMIC LOAD

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Abstract: The gearboxes (reducing gears) built in the rotating excavators used at surface coal mines, are exposed to dynamic loads which are frequently unpredictable (stochastic) given the exploitation conditions. These gearboxes are very frequently oversized by the manufacturer. Nevertheless, the possibility of impact loads due to unpredictable working resistances the excavators run up against, can reduce the safety of the gearbox, even to the extent of breakdown of some of the constituent elements in given period till the planned overhaul. This paper presents the methodology for assessing the safety (reliability) of most unreliable gear in specific gearbox, being ascertained by the maintenance experience of the gearbox. Concerning this gearbox, the load function has been established based on experimental investigations, being presented in this paper. Regarding the most unreliable gear, the carrying capacity function has been established by means of theoretical research being displayed in the comparative diagram together with the load function. This comparative diagram helps to infer the safety of the most unreliable gear in the gearbox, i.e. the safety of the gearbox as a whole. The loading on the gearbox is a variable course of the rotating moment with the driven shaft depending on the time span during the total exploitation life of the gearbox. The carrying capacity of the gearbox is equal to the permissible temporally constant rotating moment acting on the driven shaft within a time span encompassing a number of changes in the loading on the driving shaft. The presented methodology can be applied to investigate the safety of other similar gearboxes.

Keywords: DYNAMIC LOAD, GEARBOX, SAFETY

1. Introduction

The loading regime of the gearbox (reducing gears) at the working organ (working wheel) on an excavator under normal and specific conditions of exploitation, is defined by the distribution function of the rotating moment at the output shaft depending on the load variations within the expected exploitation lifetime of the gearbox.

The loading regime of the gearbox with the working wheel on the excavator SRs-630 used at the coal mine Suvodol-Bitola is determined based on performed experimental measurements and extensive theoretical analysis.

![Fig. 1 Scheme of the gearbox (reducing gears) with the working wheel on the excavator SRs-630](image)

The carrying capacity of the gearbox (reducing gears) at the work wheel on an excavator is determined by its mostly loaded gear, that is, by the gear whereon some weaknesses during exploitation have been noticed.

The functions of the carrying capacity for all gears within the gearbox at the work wheel on the excavator SRs-630, are derived theoretically by means of methodology set up for this purpose. The analysis showed that the most loaded is the last gear pair of the gearbox (marked with 5 on figure 1), to wit, the driven gear of the gear pair, which is mounted on the output shaft of the gearbox. The same conclusion was confirmed by the maintenance practice of the gearbox showing most interventions on this gear.

2. Load function of the gearbox

The loading of the gearbox (reducing gears) is a variation course of the rotating moment at the driven shaft depending on the time point over the entire exploitation life of the gearbox.

The loading of the gearbox of the working wheel on the excavator SRs-630 represents a variation course of the rotating moment $T_2$ at the driven shaft depending on the time point during the total exploitation life of the gearbox.

The load function of this gearbox (shown on figure 2) is obtained by processing the deformation records on the output shaft of the gearbox under characteristic working regimes by means of experimental measurements.

![Fig. 2 Load function of the gearbox at the working wheel on the rotating excavator SRs-630 under working conditions at the coal mine Suvodol-Bitola](image)
3. Carrying capacity function of the gearbox

The carrying capacity stands for the ability of the material having certain shape to withstand certain types of maximum loads entailing number of changes, under certain working conditions. The carrying capacity can be brought into proportion with the Veler’s curves of fatigue, obtained by checking the gear (separately the flexion at the tooth root, and likewise of the side surface pressure). This is achieved by theoretical figuring out of the carrying capacity function curve, whilst for practical purposes it suffices to establish only two points on the curve, and define their the driving factor value.

Thus, the carrying capacity of the gearbox-reducing gears is equal to the allowable temporally constant rotating moment acting upon the driven shaft within a period of changes of the load at the driving shaft.

The carrying capacity of the gearbox at the working wheel on the excavator SRs-630 (shown on figure 3) is equal to the allowable temporally constant rotating moment T2 acting on the driven shaft during N changes of the load upon the driving shaft, that is, the ability of the shaped material in gear to withstand certain maximum loads entailing number of changes under certain working conditions.

4. Comparing of the load and carrying capacity functions of the gearbox

Comparing the load function of the gearbox at the working wheel on the rotating excavator SRs-630 with carrying capacity functions for surface pressure and bending (shown on figure 3) yielded the following: the load function with a value $T_2 = 1.2 \times 10^9$ Nmm for $N = 10^3$ changes of the load and decreasing values for $T_2$ with increase for the number of changes of the load $N$, is significantly lower in relation to the carrying capacity functions, for surface pressure and bending. Since these curves do not intersect and are not touching, theoretically deduced, the conclusion has been reached that the gearbox is oversized and possesses greater safety under nominal working regime of the excavator.

5. Conclusion

Loading regime condition of the gearbox at the working wheel under normal and specific conditions of exploitation on the excavator determined by experimental measuring, is defined by the function of rotating moment $T_2$ distribution upon the driving shaft depending on the number of load changes within the gearbox over the expected exploitation lifetime of the gearbox.

The analysis of the results from theoretical and experimental research, as well as comparative analysis of the load function of the gearbox and carrying capacity functions, it can be concluded that the load functions with increasing number of load changes is significantly lower positioned in relation to the carrying capacity functions for the surface pressure and bending.

The general conclusion would be: the gearbox at the rotating wheel on the excavator SRs-630 (shown on figure 4) altogether possesses high safety and reliability to work under normal exploitation conditions, in terms of design and construction parameters and characteristics.

6. References

Hristovska E. Optimization of gearboxes for drive to the rotating wheel on the excavators, Skopje, 1997

Manual handling for the excavator SRs-630 and other technical documentation from the manufacturer TAKRAF-Germany

Numerous documentation for modifications and repairs of gearboxes from the coal mine Suvodol-Bitola.
Abstract: The recently issued IMO requirements for reduction of GHG emissions for shipping reinforced the attempts to increase the energy efficiency of ships.

One of the measures prescribed by IMO is Ship Energy Efficiency Management Plan (SEEMP). This, among other things, involves trim optimization.

Trim i.e. difference between the draft at the bow and the stern is controlled parameter worthy of attention with respect of fuel usage and GHG emissions while the ship is cruising. It is shown in this paper that the powering performance of vessels varies with different trim conditions.

The main objective function of the trim optimization is the powering performance (resistance and propulsion).

The objective of this paper is to analyze the physics behind the effect of varied trim on ship resistance and propulsion, to detect the origin of this effect.

In particular are examined the change of total resistance depending on the change of some parameters such as length of waterline (hence frictional resistance), submerged surface, change of the residual resistance, wave making/breaking at the bow and analyzed in terms of and viscous - pressure resistance (form-factor). Also examined are parameters influencing the powering performance as: thrust deduction, wake fraction, relative rotative efficiency, and propeller efficiency when the ship is trimmed.

This analysis has been based on experimental data of particular ships.

**KEYWORDS:** ENERGY EFFICIENCY, SEEMP, TRIM OPTIMIZATION

1. Introduction

Energy efficiency of ships has ever been an important issue for ship-owners and ship operators due to entirely economic reasons.

Lately, however, this interest has been revived and reinforced by the environmental concerns of mankind in all fields of our life.

The recently issued IMO Marine Environment Protection Committee (MEPC) requirement for reduction of GHG emissions from shipping [4] is SEEMP (Ship Energy Efficiency Management Plan). SEEMP is mandatory to exist onboard every ship but voluntary in contents. Table 1.1 shows SEEMP related measures.

**Table 1.1 SEEMP related measures**

<table>
<thead>
<tr>
<th>No.</th>
<th>Energy Efficiency Measure</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Engine tuning and monitoring</td>
</tr>
<tr>
<td>2</td>
<td>Hull condition</td>
</tr>
<tr>
<td>3</td>
<td>Propeller condition</td>
</tr>
<tr>
<td>4</td>
<td>Reduced auxiliary power</td>
</tr>
<tr>
<td>5</td>
<td>Speed reduction (operation)</td>
</tr>
<tr>
<td>6</td>
<td>Trim/draft monitoring and optimisation</td>
</tr>
<tr>
<td>7</td>
<td>Voyage execution</td>
</tr>
<tr>
<td>8</td>
<td>Weather routing</td>
</tr>
<tr>
<td>9</td>
<td>Advanced hull coating</td>
</tr>
<tr>
<td>10</td>
<td>Propeller upgrade and aft body flow devices</td>
</tr>
</tbody>
</table>

The topic of this paper is trim/draft optimization and more specifically analysis of the physics behind its effect on ship powering performance, effect of flow and resistance when draft and trim vary.

2. Physics behind the effect of varied trim on ship powering performance

FORCE Technology, Denmark [3] investigated what causes the change in propulsive power when a vessel is trimmed. The possible explanations may relate to changes in the following parameters:

wetted surface area; water line length; form-factor; residual resistance coefficient; thrust deduction; wake fraction; propeller efficiency and relative rotative efficiency.

The objective of this section is to try decompose the effects of trim on the above parameters. All discussions are based on the results of powering model tests of 320 000 DWT VLCC at one speed (The trim optimization test matrix included four practicable ballast conditions- 1 m, 2 m, 2.4 m and 3 m trim aft, without systematic variation of mean draught).

Trim is defined as the difference between the draft at aft perpendicular (Ta) and the draft at fore perpendicular (Tf):

\[ \text{TRIM} = T_a - T_f \] (2.1)

The physical effects that reduce delivered power when the ship is trimmed can relate to ship resistance and propulsive efficiency, as shown in the formula to determine the delivered power (2.2):

\[ P_d = \frac{R_T V^2}{\eta_D} \] (2.2)

i.e. the scope to reduce the power are reducing the ship resistance (R_T) and/or increasing the propulsive efficiency ( \( \eta_D \)).

2.1. Resistance

The total resistance coefficient, according to ITTC [1] is:

\[ C_R = C_{R_f} + (1 + k)C_{R_f} + C_{R_p} \] (2.3)

The correlation allowance \( C_{R_p} \) usually is assumed constant for all trim conditions, except in case of major changes in draft

Change in the friction resistance coefficient \( C_{R_f} \), according to ITTC-57 is:

\[ C_{R_f} = \frac{0.075}{(\log_{10}(Re) - 2)^2} \] (2.4)

Where Re is Reynolds number defined by:

\[ Re = \frac{V L_{ref}}{v} \] (2.5)
The kinematic viscosity of sea water (\(\nu\)) is constant for the same temperature.

From the above formulas, it follows that the frictional coefficient depends on the length on waterline.

**Table 2.1 Change in power due to waterline length at \(V_s=14kn\) for 320k VLCC**

<table>
<thead>
<tr>
<th>Trim</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cases</td>
<td>B1</td>
<td>B2</td>
<td>B3</td>
<td>B4</td>
</tr>
<tr>
<td>(\Delta C_{L,W}) [%]</td>
<td>0.6%</td>
<td>0.7%</td>
<td>0.5%</td>
<td>0.2%</td>
</tr>
<tr>
<td>(\Delta P_{D,wt}) [%]</td>
<td>-0.00%</td>
<td>-0.07%</td>
<td>-0.07%</td>
<td>-0.07%</td>
</tr>
</tbody>
</table>

From above table can be seen that the effect of waterline length changing is negligible.

When the vessel is trimmed, the wetted surface area varies as shown in Table 2.2.

**Table 2.2 Change in power due to wetted surface area at \(V_s=14kn\) of 320k VLCC**

<table>
<thead>
<tr>
<th>Trim</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cases</td>
<td>B1</td>
<td>B2</td>
<td>B3</td>
<td>B4</td>
</tr>
<tr>
<td>(\Delta S) [%]</td>
<td>0.44%</td>
<td>-0.61%</td>
<td>-0.97%</td>
<td>1.74%</td>
</tr>
<tr>
<td>(\Delta P_{D,wt}) [%]</td>
<td>0.44%</td>
<td>-0.61%</td>
<td>-0.97%</td>
<td>1.74%</td>
</tr>
</tbody>
</table>

The saving in power due to wetted surface area varies not so little as due to waterline length. The delivered power varies proportionally to the wetted surface variation.

The residual resistance coefficient (\(C_R\)) is often claimed to be the quantity most affected by trim.

In Table 2.3 below are shown the savings due to changes in \(C_R\).

**Table 2.3 Change in power due to \(C_R\) at \(V_s=14kn\) of 320k VLCC**

<table>
<thead>
<tr>
<th>Trim</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cases</td>
<td>B1</td>
<td>B2</td>
<td>B3</td>
<td>B4</td>
</tr>
<tr>
<td>(\Delta C_{R}) [%]</td>
<td>107.9%</td>
<td>-47.1%</td>
<td>-13.6%</td>
<td>-48.3%</td>
</tr>
<tr>
<td>(\Delta P_{D,wt}) [%]</td>
<td>3.28%</td>
<td>-1.43%</td>
<td>-1.43%</td>
<td>-1.43%</td>
</tr>
</tbody>
</table>

From table 2.3 it can be seen that the \(C_R\) changing in all cases is high. Its effect on the delivered power is more than this due to previously mentioned parameters.

The summing up of all the saving due to resistance components to propulsive power gives the following results:

**Table 2.4 Change in propulsive power due to total resistance at \(V_s=14kn\) of 320k VLCC**

<table>
<thead>
<tr>
<th>Trim</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cases</td>
<td>B1</td>
<td>B2</td>
<td>B3</td>
<td>B4</td>
</tr>
<tr>
<td>(\Delta P_{D,wt}) [%]</td>
<td>-0.06%</td>
<td>-0.07%</td>
<td>-0.07%</td>
<td>0%</td>
</tr>
<tr>
<td>(\Delta P_{D,Re}) [%]</td>
<td>0.44%</td>
<td>-0.61%</td>
<td>-0.97%</td>
<td>1.74%</td>
</tr>
<tr>
<td>(\Delta P_{D,Re}) [%]</td>
<td>3.28%</td>
<td>-1.43%</td>
<td>-1.43%</td>
<td>-1.43%</td>
</tr>
<tr>
<td>(\Delta P_{D,Re}) [%]</td>
<td>3.66%</td>
<td>-2.11%</td>
<td>-2.47%</td>
<td>0.24%</td>
</tr>
</tbody>
</table>

2.2. Propulsion

The propulsive efficiency is also affected when the vessel is trimmed. It is composed of the hull efficiency:

\[
\eta_h = \frac{1-\tau}{1-\omega} \tag{2.6}
\]

The relative rotative efficiency (\(\eta_R\)) and the propeller efficiency (\(\eta_P\)) as shown in the formula below, according to ITTC [2]:

\[
\eta_R = \eta_h \cdot \eta_r \cdot \eta_P \tag{2.7}
\]

The hull efficiency is a function of thrust deduction (\(t\)) and wake fraction (\(\omega\)). The thrust deduction is a function of total resistance and propeller thrust:

\[
t = \frac{T - R_p}{T} \tag{2.8}
\]

It has already been said that the total resistance changes when ship is trimmed. Propeller thrust will change also.

The wake fraction is defined as a ratio of effective wake velocity (\(V-V_a\)) and ship speed (\(V\)), where (\(V_a\)) is propeller inflow velocity:

\[
w = \frac{(V-V_a)}{V} \tag{2.9}
\]

When the ship speed is kept constant, the wake fraction changes due only to propeller inflow velocity.

Propeller efficiency is defined from open water test, not in the wake of the vessel. The open water curve is a function of the advance ratio (\(J\)), which is a function of propeller inflow velocity, propeller revolutions and diameter:

\[
J = \frac{V_a}{nD} \tag{2.10}
\]

So the propeller efficiency is affected by trim also due to \(V_a\).

The relative rotative efficiency is defined as a ratio of propeller torque in open water (\(Q_{ow}\)) and behind the ship (\(Q_{shp}\)):

\[
\eta_r = \frac{Q_{ow}}{Q_{shp}} \tag{2.11}
\]

The summing up all power saving from propulsive efficiency components to propulsive power gives the following results:

**Table 2.5 Change in propulsive power due to propulsive efficiency at \(V_s=14kn\) of 320k VLCC**

<table>
<thead>
<tr>
<th>Trim</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cases</td>
<td>B1</td>
<td>B2</td>
<td>B3</td>
<td>B4</td>
</tr>
<tr>
<td>(\Delta P_{D,wt}) [%]</td>
<td>-0.64%</td>
<td>-2.89%</td>
<td>0.51%</td>
<td>0.13%</td>
</tr>
<tr>
<td>(\Delta P_{D,Re}) [%]</td>
<td>-0.91%</td>
<td>-3.84%</td>
<td>-0.91%</td>
<td>-1.10%</td>
</tr>
<tr>
<td>(\Delta P_{D,Re}) [%]</td>
<td>0.72%</td>
<td>-0.31%</td>
<td>0.51%</td>
<td>0.72%</td>
</tr>
<tr>
<td>(\Delta P_{D,Re}) [%]</td>
<td>0.68%</td>
<td>1.73%</td>
<td>0.00%</td>
<td>0.68%</td>
</tr>
<tr>
<td>(\Delta P_{D,Re}) [%]</td>
<td>-0.14%</td>
<td>1.09%</td>
<td>0.11%</td>
<td>0.44%</td>
</tr>
<tr>
<td>(\Delta P_{D,Re}) [%]</td>
<td>3.28%</td>
<td>-2.11%</td>
<td>-2.47%</td>
<td>0.24%</td>
</tr>
</tbody>
</table>

The saving from changes in total resistance and propulsive coefficients are shown in Table 2.6.

**Table 2.6 Change in propulsive power due to trim at \(V_s=14kn\) of 320k VLCC**

<table>
<thead>
<tr>
<th>Trim</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cases</td>
<td>B1</td>
<td>B2</td>
<td>B3</td>
<td>B4</td>
</tr>
<tr>
<td>(\Delta P_{D,Re}) [%]</td>
<td>3.7%</td>
<td>-2.1%</td>
<td>-2.5%</td>
<td>0.2%</td>
</tr>
<tr>
<td>(\Delta P_{D,Re}) [%]</td>
<td>0.14%</td>
<td>1.09%</td>
<td>0.11%</td>
<td>0.44%</td>
</tr>
<tr>
<td>(\Delta P_{D,Re}) [%]</td>
<td>3.52%</td>
<td>-1.02%</td>
<td>-2.35%</td>
<td>0.68%</td>
</tr>
<tr>
<td>(\Delta P_{D,Re}) [%]</td>
<td>1.79%</td>
<td>-1.93%</td>
<td>-3.35%</td>
<td>3.62%</td>
</tr>
<tr>
<td>(\Delta P_{D,Re}) [%]</td>
<td>1.73%</td>
<td>0.91%</td>
<td>1.00%</td>
<td>-2.94%</td>
</tr>
</tbody>
</table>

It can be seen that in some of the cases the difference is high (e.g. case B4), but the direction of the change due to trim is the same compared to the values from model tests.

From the results shown in Table 2.6 it can be concluded that the most affected factor due to trim is the total resistance. Still, the propulsive coefficients change effects to the total sum of change due to trim are significant and they should not be neglected.

### 3. Effect of bow bulb close to or intersecting the free surface

All discussions in this section are based on observations of model tests with 19 000 DWT tanker (a matrix of three drafts, three trims each, has been tested), 320 000 DWT VLCC and 50 000 DWT Product Carrier (three trims of 50k Product Carrier have been tested with one mean draft and displacement, i.e. systematically varied).

Fig. 3.1 and Fig.3.2 shows a sample photo of the wave pattern related to the wave resistance coefficient curve for one of the cases of model tests with 19 000 DWT tanker (Fig.3.1) and one of the cases of model tests of 50k Product Carrier (Fig.3.2).
Fig. 3.1 Photo of wave pattern related to the wave resistance coefficient curve of 19000 DWT tanker

50k Product Carrier, T4

Fig. 3.2 Photo of wave pattern related to the wave resistance coefficient curve of 50k Product Carrier

It can be seen that when the bulb is intersecting the free surface at still water with increasing Froude number there occurs wave breaking in front of the blunt part of the bulb and a single overturning wave immediately behind the bulb where flow meets the main hull. At certain Fn these events are most intensive that corresponds to a local maximum of the wave resistance coefficient. Only when the dynamic pressure gets high enough to raise the water over the bulb and smoothly flow downstream there is a local minimum of the wave resistance coefficient. This phenomenon is more obvious for the case of 19000 DWT tanker.

The previous assertion is not valid for 320 000 DWT VLCC due to the fact that water does not cover the bow bulb, in all trim cases it is not submerged and there are no peaks in wave resistance coefficient curve (Fig.3.3).

Using 4th order polynomial curve-fit the local maximum and minimum of the effective wave resistance $C_w=C_T-(1+k)C_F$ and their location on the Fn axis have been determined.

From physical considerations and common sense, the following assumption was made:

**Assumption** The observations illustrated with Fig. 3.1 and Fig. 3.2 suggest that the local minimum of the wave resistance curve occurs when the flow covers completely the bow and smoothly continues downstream. Theoretically, by virtue of Bernoulli’s equation, the free surface elevation at the stagnation point is:

$$z = \frac{1}{2} \frac{V^2}{g} \quad (3.1) \quad \text{or non-dimensionally} \quad \frac{z}{L} = \frac{1}{2} \frac{F_{n_{min}}^2}{L} \quad (3.2)$$

That means that if the height needed to cover the bow bulb is $Z_{up}$ this will happen at

$$F_{n_{min}} \sim \sqrt{\frac{2Z_{up}}{L}} \quad (3.3)$$

As a representative height $Z_{up}$, was accepted the distance between the crossing point of the still waterline with the bulb profile and the level of the design waterline.

Fig. 2.4 and Fig 2.5 shows the relation $F_{n_{min}} (z/L_{wl})$

**Fig. 3.4 Relation between $F_{n_{min}}$ of local minimum $C_w$ and the height to the top of the bulb of 19k tanker**

A trend line, approximating the observations with satisfactory determination coefficient, really has the behavior of the square root function (3.3).

**Fig. 3.5 Relation between $F_{n_{min}}$ of local minimum $C_w$ and the height to the top of the bulb of 50k Product Carrier**

The location and magnitude of these local extrema probably depend on the height from the still waterline to the level above the bulb and the bluntness of the bulb shape.
4. Effect of trim on viscous resistance

Viscous resistance is affected by trim in two ways. First, wetted length (hence Reynolds number) and wetted surface change with trim but this is a minor effect. Equally minor is the effect of trim on windage area and air resistance.

The more important effect of trim is on viscous-pressure resistance, i.e. the form-factor.

The data analyzed in this section are based on two powering model tests of a 320 000 DWT VLCC and 50 000 DWT Product Carrier.

After trying to relate different geometrical parameters to form-factor values, the best try was the curvature of the run shoulder of the sectional area curve.

Fig. 4.1. and Fig. 4.2 shows the prismatic curves for the two tested models.

![Fig. 4.1 Prismatic curves for investigated trim condition of 320k VLCC](image)

![Fig. 4.2 Prismatic curves for investigated trim condition of 50k Product Carrier](image)

The curvature of the prismatic curve in the area outlined in Fig. 4.1 and Fig. 4.2 was calculated for the same x-values in all cases for model according to the classical formula:

\[ k = \frac{y^2}{(1+y^2)^{3/2}} \]  

(4.1)

These values were related to the corresponding values of 1+k (Fig. 4.3).

![Fig. 4.3 Relation of form-factor vs. run-shoulder curvature for 320 000 DWT VLCC and 50k Product Carrier](image)

5. Summary and conclusion

On the basis of specific model test results the physics behind trim effect on ship powering performance, the flow around the hull and ship resistance components have been analyzed to find that:

- The most affected quantity by trim is the total resistance.
- The propulsive coefficients change effects to the total sum of change due to trim are significant and they should not be neglected.
- With bow bulbs intersecting the free surface the flow features wave breaking followed by a smooth flow around the bulb which result in a hump and hollow of the wave resistance curve. It has been established that they and their Fn position depend on the depth of the bulb of the still-water line. The viscous resistance, specifically the form-factor, of an investigated case relates quite definitively to the curvature of the stern run shoulder of the prismatic curve.

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ANALYSIS OF THE VEHICLE UNDERCARRIAGE

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Abstract: Parameters of the vehicle undercarriage affect the vehicle movement and its behaviour on the road or on the terrain. Basic geometry of the vehicle undercarriage can be expressed by mathematical methods, yet undercarriage analysis of the moving vehicle on the terrain is very complex procedure. We have to take into account wheels vertical movement and deflection (which influences the suspension system). Utilization of the simulating technologies can be very beneficial for solution of this issue. The paper is focus on mathematic model of the vehicle undercarriage and its application for analysis of the vehicle undercarriage.

Keywords: SIMULATIONS, TRUCK, STEERING, SUSPENSION

1. Introduction

Lethality, protection, mobility and communication are the general capabilities of the combat vehicle. Nowadays conflicts and military missions require different types of vehicles. Improving of the lethality and protection is current trend. Both these capabilities affect vehicle weight and its mobility. Finding balance between mobility, protection and fire power can be very difficult process and very often must be chosen a compromise, between fire power and protection on the one side and mobility and weight on the other side. Simulating technologies are suitable manner for analysis of the vehicle mobility and its change during the process of vehicle improvement.

Simulating technologies are very efficient manner in construction of the mechanical systems and nowadays they are utilized in design and development stage of life cycle of vehicles. Mathematic simulations can be used for assessing, evaluating and comparing of basic capabilities amongst vehicles. The main advantages are:

• possibility of comparing different vehicles in the same operational environment (e.g. on the same obstacles)
• simulation of various critical operational states and in-service behaviour of vehicle (e.g. destruction of wheel)
• evaluation of different modifications (modernization) influence on vehicle mobility
• development of capabilities – mobility, survivability and reliability

Simulating technologies have some disadvantages, too. The main key point is input data – either insufficient or lack of them.

The purpose of this paper is to demonstrate utilization of simulating technologies (mathematic model development) for the analysis of the vehicle undercarriage.

2. Description of the mathematic model

For vehicle analysis and mathematic model development we use Multibody Dynamics software ADAMS (Automatic Dynamic Analysis Mechanical Systems) of MSC software company. Adams improves engineering efficiency and reduces product development costs by enabling easy system-level design validation.

Adams is optimized for large-scale problems, taking advantage of high performance computing environments. For dynamics vehicle development and testing MSC offers module ADAMS/Car. With Adams/Car engineering teams can build and test functional virtual prototypes of complete vehicles and vehicle subsystems. Working in the Adams vehicle environment, automotive engineering teams can exercise their vehicle designs under various road conditions, performing the same tests they normally run in a test lab or on a test track, but in a fraction of time.

Czech military vehicle T-810 has been selected for application of simulating technologies. The vehicle T-810 is medium off-road truck which meets requirements of high terrain throughput, robustness, endurance, airway, seaway and railway transportability. Picture of this vehicle is shown in the Fig. 1.

Fig. 1 The medium off-road truck T-810 [7].

Basic tactical-technical parameters of the truck T-810 are presented in the table 1.

Table 1: Basic tactical-technical parameters of the vehicle T-810 [3, 9].

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Engine</td>
<td>water cooled diesel, in-line 6 cylinders</td>
</tr>
<tr>
<td>Engine power</td>
<td>177 kW</td>
</tr>
<tr>
<td>Engine torque</td>
<td>920 Nm</td>
</tr>
<tr>
<td>Max. speed</td>
<td>106 km/h</td>
</tr>
<tr>
<td>Min. speed</td>
<td>3.2 km/h</td>
</tr>
<tr>
<td>Tank range</td>
<td>800 km</td>
</tr>
<tr>
<td>Max. weight</td>
<td>13 000 kg</td>
</tr>
<tr>
<td>Standby weight</td>
<td>8 500 kg</td>
</tr>
<tr>
<td>Length</td>
<td>7 490 mm</td>
</tr>
<tr>
<td>Width</td>
<td>2 550 mm</td>
</tr>
<tr>
<td>Height</td>
<td>3 280 mm</td>
</tr>
<tr>
<td>Gauge of the front axle</td>
<td>2 020 mm</td>
</tr>
<tr>
<td>Gauge of the rear axles</td>
<td>2 100 mm</td>
</tr>
<tr>
<td>Wheelbase</td>
<td>3 150 + 1 200 mm</td>
</tr>
<tr>
<td>Step obstacle</td>
<td>600 mm</td>
</tr>
<tr>
<td>Ditch obstacle</td>
<td>900 mm</td>
</tr>
<tr>
<td>Ride height</td>
<td>460 mm</td>
</tr>
<tr>
<td>Wheels</td>
<td>362/80 R20</td>
</tr>
<tr>
<td>Front suspension</td>
<td>wound springs</td>
</tr>
<tr>
<td>Rear suspension</td>
<td>leaf springs</td>
</tr>
<tr>
<td>Steering gear ratio</td>
<td>18,1 – 21,4</td>
</tr>
</tbody>
</table>

The objective of the work is to create complex model of the vehicle T-810 which can be used for general simulations (e.g. turning, gear shifting, vehicle manoeuvres). The general model (assembly) has to include minimally next subsystems: front/rear suspension; steering; powertrain; transmission; brakes; front/rear wheel; body. Chassis of the vehicle T-810 is shown in the Fig. 2.
In the paper will be presented assembly of the front axle. Created model of front suspension system with the steering system is shown in the Fig. 4. Assembly of the front axle consists of the two subsystems – front suspension subsystem and steering subsystem. Main parts of the suspension subsystem are: tube, leading rods, V rod, propelling shafts, hook joints, steering pins, springs, dumpers, bumpstops and bushings. The steering subsystem consists of: steering wheel, shafts, hook joints, steering box, steering bars and levers. Testing workbenches and tires are the main parts of this assembly, too. This assembly enables analysis of the steering systems in dependency on wheels vertical position or their vibrations. Characteristics of the wheel can be changed as like as other parameters of the main parts.

Fig. 2 Chassis of vehicle T-810 [6].

Fig. 3 Front axle of the vehicle T-810 [9].

Fig. 4 Created assembly of the front suspension system with steering.

Fig. 5 Characteristics of the front springs.

Fig. 6 Characteristics of the front dumpers.

Fig. 7 Selected parameters of the model tires – part 1.
3. Simulations

Pasted simulations were focus on wheel swerve in different situations: flat road, raised one wheel and bumpy terrain (profile of the left and right traces are shown in the next graphs – Fig. 9 and Fig. 10). The length of the simulations was 10s. Rotation of the steering wheel was defined by the next graph (Fig. 11).

4. Outcomes

In the Fig. 11 there is also presented rotation of the steering bar. Steering angle of the wheel depends on steering wheel position, geometry of the steering mechanism and vertical position of the wheels. Simulations were focus on the analysis of the change of the steering wheel angle during the vertical wheel movement. The basic simulation was done with no wheels movement – these angles (wheel rotational movements) were reference. 2nd simulation was passed with left wheel up (150 mm) and right wheel down (150 mm). 3rd simulation was inverted left wheel down, right wheel up. 4th simulation was passed with wheels movement defined in the Fig. 9 and Fig. 10 – model speed during the simulation was set up to 40 km/h. Outcomes are presented in the next graphs (Fig. 12 – Fig. 16). In the left side there are the wheel steering angles of the left and right wheel. In the right side there are presented differentials of the each simulation (dif.1 = outcomes of sim.1 – outcomes of sim.2; dif.2 = outcomes of sim.1 – outcomes of sim.3; dif.3 = outcomes of sim.1 – outcomes of sim.4).

Steering force was the next analysed parameter, because vehicle movement produces forces which are transferred though the steering mechanism and burden the driver. Courses of the steering wheel input torque are presented in the next graphs (Figure 16).

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**Fig. 8** Selected parameters of the model tires – part 2.

**Fig. 9** Profile of the left wheel track – simulation of bumpy road.

**Fig. 10** Profile of the right wheel track – simulation of bumpy road.

**Fig. 11** Position of the steering wheel and steering bar during the simulations.

**Fig. 12** Wheels angles on the flat road.

**Fig. 13** Wheels angles on the flat road – left wheel up.

**Fig. 14** Wheels angles on the flat road – right wheel up.
Outcomes of the developed mathematic model show the behaviour of the front suspension system with the steering mechanism. Wheel steering angle depends on the geometry of the steering mechanism and vertical position of the wheels. Steering input torque was the next analysed value. From the simulation we can find that magnitude of this torque depends on vertical movement of the wheels and turning side. In the graph (Figure 16) we can see that (in case with the right wheel ascended) the torque during the right turning is higher than during the left turning (blue line). Course of this torque has highly dynamic character and the highest values are during the ride on the bumpy road (grey line).

Acknowledgement

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References

THE CONCEPT OF TECHNICAL STATE CONTROL OF VEHICLE BRAKING SYSTEM WITH ABS IN OPERATING PROCESS

Abstract: Possibilities of diagnosing of a technical condition of brake system of the car with ABS while in service are considered. The concept of sharing of onboard means and stands for diagnosing is presented.

KEYWORDS: VEHICLE, ANTIBLOCK SYSTEM (ABS), DIAGNOSTICS, BENCH.

Automobile active safety is defined not only by its technical equipment, but also by serviceability of its elements technical state in operating process. It is thus obvious that a lot of things depend on effective work of brake system of the vehicle. Number of automobiles equipped by anti-lock braking systems (ABS) exceeded 80% for the ones released in Europe. However, a lot of things are still unclear with the ideas of technical state control of vehicle brake system with ABS. We will try to sort the problem out. First of all it is absence of traditional serviceability signals of brake system for the driver. If a vehicle is skidding these signals are often the traces left on asphalt surface by slowed-down wheels which testify, first, about efficiency of functioning of the brake drive (the brake moment on wheels exceeded (or not) greatest possible in these conditions the brake torque on tractive contact). The quantity of the last is defined on the known formula:

$$M_{\text{max}} = R_z \, \varphi_{\text{max}} \, r_d , \ [1]$$

If: $R_z$ – normal loading on a wheel, $r_d$ – the dynamic radius of a wheel, $\varphi_{\text{max}}$ - the maximum size of coefficient of traction at optimal wheel spin.

Secondly, by the traces left while skidding it is possible to judge indirectly about unevenness of operation of brake gears (u. o. b. g.) . The essential traces discrepancy is incentive motive for maintenance of brakes or bench preliminary treatment of the vehicle.

If a vehicle is provided with ABS brake system skidding traces absence on the road surface is regarded in two ways. It can testify effective work of ABS or insufficient efficiency of operating of the brake drive units and its general inability to provide the quantity of the maximum moment on traction that enables braking a wheel in subcritical skidding area $\varphi$ (S) - charts.

However, even in the first case not everything is so obvious. The matter is that while ABS is operating it is almost impossible for the driver to estimate degree of use of the maximum coefficient of traction and its compliance to standard requirements basing only on his feelings.

It is known that technical state control of an operating vehicle is carried out regularly during annual maintenance or a planned one after a certain mileage. At the same time malfunction of vehicle brake system units, especially ABS, involves serious consequences. So, according to European countries traffic police sudden failure of ABS caused more serious road accidents’ consequences than while skidding.

Producers equip ABS with self-control system which carries out testing serviceability of electric chains and signal level. So, break of a power-supply circuit of the modulator or the sensor will immediately lead to shutdown of ABS and giving a warning signal to the driver.

It is thus obvious that this system doesn't settle possible malfunctions of its units. Besides malfunctions listed above it is possible to add, for example, such ones as change of the modulator channels section owing to their contamination, a delay at valves operation, an angular pliability of a stator of the ABS sensor, weakening of springs pulling together brake shoes, etc. In cases of all listed above failures ABS system of self-diagnostics doesn't give a signal of malfunction.

The listed facts dictate an urgent need to improve onboard diagnostic tools of the brake system elements’ technical state to make them become capable to provide, at least as at a first approximation, objective information for the driver about a condition of brake system as a whole and quality of carried out operating process during the inter-control period of operation. At this stage diagnosis has to be carried out first of all in parameters of efficiency and give the general "integrated" assessment, thereby, filling for the driver absence of objective visual criteria of skidding traces. Therefore, brake dynamism based on implemented slowdown has to be the basis for such onboard diagnostic tools.

Further development of means of onboard diagnostics is seen in creating intellectual systems constructed on the basis of developed structural and investigative schemes. As an example in figure 1 the example of the structural and investigative scheme developed for the pneumatic drive of brakes with ABS is presented. Such approach will allow seeing codes of estimated malfunctions or failures of system elements. At the same time, it is necessary to remember that in actual practice operation braking process proceeds at an essential deviation of characteristics of external conditions, in particular, fluctuations of coefficient of traction both in longitudinal and cross directions that will affect diagnosis accuracy.

The last circumstance causes need of monitoring technical state of braking system units under the fixed and reproduced external conditions that is reached at bench preliminary treatment.

Existing benches at which diagnosis of a car with ABS is possible can be divided into two large groups: the roller power and drum inertia. We do not mention benches of areal type considering necessity of providing rather high speed of movement. The benches of the first group provide the greatest possible traction of the tire with a roller surface due to its longitudinal corrugation. The main objective thus is testing efficiency of brake gears functioning and the system as a whole on the basis of assessment of the implemented braking torques.

Therefore testing the braking system of the car with ABS at such benches is possible only at implementing additional devices (often controlled by a computer), allowing to provide change of a relative wheel slip concerning a roller or the last relatively a drive gear.

During testing quality of functioning of ABS in vitro the important circumstance is the greatest possible reconstruction of real conditions of interaction of the tire with a road surface. The last is reached, on the one hand, by distribution of normal loadings adequate to real process in a spot piece of contact, and on the second hand by a reconstruction of coefficient of traction. The first condition causes at least an one and a half-multiple ratio of diameters of a chassis dynamometer and a car wheel, and the second causes implementing drum cleaning system in a contact piece spot from products of tire wear (for example by means of adding kaolin powder into a spot of contact piece as it is implemented at the bench for tire wear testing on the Volzhsk Tire Plant), and also adding liquid for imitation of a wet surface on cement sectors of a drum, etc.

All the items mentioned above significantly complicate the design of the bench and increase its dimensions, transferring it into the category of the research ones that limits use of such benches in service centres, workshops and MTE. The widespread roller power
brake bench has rollers with longitudinal corrugation made for ensuring the maximum traction with the tire. Practically it excludes opportunity to reproduce road conditions for tire traction close to ensuring the maximum traction with the tire. Practically it excludes the reality. However, in the power bench this point is not required as the main objective is the assessment of the greatest possible implementation of braking torques on wheels, and also distinctions of brake forces regarding their compliance to admissible values according to the standards. Due to the relative compactness and rather low cost this type of benches gained the greatest distribution in service centres, workshops and MTE. At the first glance such design does not allow carrying out diagnosing the braking system of the car with ABS. However, possible way out can be diagnosing with implementing a special set point adjuster of modes which imitates signals of the sensor of angular speed of a wheel or the commands given by the logical block on the ABS modulator. At imitation of the sensor’s signals of a wheel angular speed an input of the ABS logical block the sinusoidal signal of variable frequency from the generator is given. At a certain value of frequency the ABS logical block has to form a controlling signal on the modulator for a wheel brake release that is reflected in the stand oscillogram.

When testing modulator on its inputs (solenoid coils) the test controlling signal "braking – cut-off – brake release - a cut-off" is given that will be reflected in the measure of the produced braking torque. In the both cases, if the driver presses out the brake pedal completely and the rollers of the bench are rotating simultaneously, on the received oscillogram it is possible to track the value of valves controlling signal "braking – cut-off – brake release - a cut-off" is given. From all the described above it is possible to draw the following conclusions:

• taking into account features of functioning braking system of the vehicle with ABS diagnosing has to be made in a complex: as means of onboard preliminary treatment during the intercontrol period, so at power roller benches equipped with a set point adjuster of modes at state technical inspection;
• implementing ABS demands further development of onboard means of diagnosing on the basis of development of new methods of diagnosing with use of multilevel structural and investigative schemes, first of all in efficiency parameters;
• it is expedient to carry out profound unit-by-unit preliminary treatment in bench conditions, with the greatest possible exception of random factors influencing the process. For widely spread implementing the process of diagnosing in practice maintenance of cars with ABS is expedient to use the widespread approbated bench equipment of a domestic production with rather small cost and outline dimensions, for example, the power roller benches produced by ZAO "GARO-Trade", at their corresponding development [2].

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Abstract: The effect of changing coating length on the cohesive and adhesive failure of the coating on metallic flat substrates is analyzed by uniaxial loading. The criterion for determining the fragmentation length of the coating with mixed (adhesive-cohesive) failure was developed. Calculating the critical length as the intersection point of the two lines (interfacial failure shear stress-length curve and the normal failure stress-length curve) does not always give correct value. The critical length of coating should be evaluated as maximum value within the annular trapezoid. It is established that the kind of probability density function both the adhesion strength and the cohesion one should be considered when determining the critical length of the coating. Verification of the developed criterion was performed on the example of plasma-sprayed coatings. It was noted, the formation of multiple cracks in the plasma-sprayed coating has random character due to microstructural defects, such as pores and microcracks.

Keywords: COATING, DETACHMENT, MULTIPLE CRACKING, FRACTURE CRITERION, INTERFACE SHEAR STRESS, NORMAL STRESS, PLASMA SPRAYING, TENSILE TESTING

1. Introduction

Coatings are subjected to various kinds of mechanical loading in service. It can cause their destruction. Researchers use various models that allow studying how the load is transferred from the substrate to the coating through the interface for understanding of the destruction processes in the coated materials [1]. Pioneering study of the load transfer through the interface contact has been performed in shear lag model by Cox [2].

The stress state of the substrate–coating system is inhomogeneous due to difference of the elastic properties of the substrate and the coating. Adhesion and cohesion failure occurs in areas where the stress reaches a critical value. The true data on the adhesion and cohesion strength can be obtained only by taking into account the inhomogeneity of the stress state in the coated materials or when we using specimens with coatings, in which the inhomogeneity of stress state can be neglected. However, both in the coated specimens and in the structural elements with coatings there is considerable heterogeneity of the stress state. The feature of the substrate–coating system is the ability to transmit the load both in the coating and in the substrate through the interface. Consider a case where the stress arises in the coating due to load applied to the substrate (Fig. 1).

There are a number of models that are used to study the stress distributions in the coatings and in the interface area. The basis of the classical Cox shear-lag model is the assumption that stresses in the fiber-matrix system are proportional to the difference of the displacement components of this system [2]. The hypothesis regarding that the stresses in the coating–substrate system are proportional to the difference of substrate and coating displacements, led to the development of another models [3–8].

Usually, in the case of the uniaxial loading the periodical multiply cracking in the coating occurs in the direction perpendicular to the load [9]. The cracking space depends on the magnitude of the applied loads and the coating strength.

Some kinds of coatings continue to fulfill their functions even after the cracking, such as: wear-resistant [10], tribological “chameleon” type [11] and thermal barrier [12]. Positive effect from the cracking of the thermal barrier coatings is reported in [13]. Coatings with the previously created cracks have a higher thermal shock resistance than coatings without cracks [14]. Therefore, this paper deals with development of the criterion for defining of the critical coating length at which the delamination and the cracking occur simultaneously. The aim of our study was to perform a consideration on the fracture of plasma-sprayed coating during application of uniaxial load, taking into account the processes of cracking and delamination.

2. Theory

In the theoretical coating–substrate model the maximum shear stress occurs at the coating end [4]. In accordance with the equations for determining the distribution of normal stresses in the coating and shear stresses in the interface, given in [4], the strains at which the cracking \( \varepsilon_{coh}(l) \) and delamination \( \varepsilon_{adh}(l) \) of the coating are occurred, is given by the model as
The crack of the coating stops if the coating will detach from the substrate. In case \( \psi < 1 \), at \( \psi = 1 \), the delamination occurs detaches without cracking, i.e., \( l < l_{cr} \). When \( \psi = 1 \) delamination and cracking of the coating occur simultaneously.

Strength criteria are usually formulated for micro-homogeneous environments, metals in particular. Strength of metals is determined by averaging over the volume characteristics. At the same time the criteria that are formulated on the basis of hypotheses about the continuity and micro-homogeneity of the environment cannot be used in the case of structurally inhomogeneous coatings. In these cases, the destruction process is dependent on the local defects (micro-cracks, pores, phase inhomogeneities) in the coating structure, and it is not dependent by averaged characteristics. Structure defects become sources of the fracture. In addition, the structural heterogeneities of coatings have a random nature. That is why, the probabilistic methods that describe the fracture of the substrate–coating system, are used to develop the adhesion-cohesion fracture criterion.

Experimental data indicate that tensile strength, yield strength, elastic modulus and other mechanical characteristics of the materials have very appreciable scatter. Coating can crack randomly [15–19] and the critical length for the same coating varies.

### 3. Equipment and materials

We used plasma-sprayed coatings on stainless steel substrates as model system. Details of the test technique were as follows. Static tensile testing of coated specimens is used in the method. Metal substrate 2.0 mm in thickness and 6.0 mm in width was made according to standards for definition of mechanical properties of metals without coatings. The coating is sprayed up to the full working length of the specimen (Fig. 3a) or over the half of the working length of the specimen (Fig. 3b). Tensile testing was conducted on an FM-1000 mechanical testing system.

The plasma-sprayed NiAl coating was applied using a plasma torch, with partially imposition of arc and additional cooling of plasma jet by concentric flow of protective gas. Argon was used as the plasma and protective gas. The following plasma spraying conditions were for all powders: current 80–100 A, voltage 50–60 V, gas flow rate 2–3 l/min, powder feed rate 2 kg/h and spray distance 120 mm at 2 mm diameter nozzle. Plasma jet has laminar character of expiration and more extended high temperature area due to equipment design and operating conditions. Before coating deposition substrates were sand blasted. Coatings with thickness of 0.15 and 0.3 mm were sprayed onto substrates.

### 4. Experimental

In order to study the failure due to applied tensile strain, five tests were performed for different thickness. The test was ended when the coatings failed by spallation. Experimental studies of tensile specimens with plasma-sprayed coatings show that the crack divides the coating into two equal length segments (Fig. 4). The thicker coatings had a different path of failure than the thinner ones (Fig. 5). Features of the coating's cracking and delamination occur simultaneously.
define four curves characterizing the maximum $\epsilon_{\text{coh}}^\text{max}(l)$ and minimum $\epsilon_{\text{coh}}^\text{min}(l)$ strains at which cohesion failure occurs as well as the maximum $\epsilon_{\text{adh}}^\text{max}(l)$ and minimum $\epsilon_{\text{adh}}^\text{min}(l)$ strains at which adhesion failure occurs (Fig. 6.). Area containing the value of the coating critical length is limited to a curved figure which can be approximately represented as a tetragon. Thus, the critical length of the coating is not determined by a single value, but its values form the quadrangular region.

![Figure 3](image)

**Fig. 3.** Schematic drawing test tensile specimen (dog-bone shape) with coating on two side (a–short length of coating, b–long length of coating)

![Figure 4](image)

**Fig. 4.** Multiple cracking of the plasma-sprayed coating with thickness of 150 µm after testing of the tensile specimen. There are a lot of transverse cracks

![Figure 5](image)

**Fig. 5.** Delamination of the plasma-sprayed coating with the thickness of 300 µm

The critical length of the coating for the normal probability density function coincides with the intersection point of the curves on the Fig. 6.

Figure 7 shows a plot for determining the critical length assuming a uniform probability distribution function for the adhesion and cohesion strength of the coating with thickness of 150 µm. The dashed curves in Fig. 7 show the relations calculated with Eqs. (1) and (2). Figure 7 shows the calculated change of $\epsilon_{\text{coh}}(l)$ and $\epsilon_{\text{adh}}(l)$ as a function of the length of the coating $l$ based on Eqs. (1) and (2) for the assumed values of adhesion strength ($\tau_{\text{max}} = 122$ and 128 MPa) and cohesion strength ($\sigma_{\text{max}} = 258$ and 268 MPa) in comparison with measured ones.

![Figure 6](image)

**Fig. 6.** The area of the simultaneous delamination and cracking of the coating

The tensile test technique also can be applied to measure the elastic modulus of the coating. The elastic properties of the coatings were extracted from the tensile response of the coating–substrate system by difference between the pure substrate and coated substrate curves. Elastic properties of the plasma-sprayed coating can be calculated after the stretching of the coated specimen (Fig.3a) [20]. Thus, calculated values of $E_c$ and $E_s$ are 141 and 210 GPa, accordingly. The values of Poisson’s ratio of the coating and the substrate used in the calculation are based on values reported in the literature for similar materials ($\mu_c = 0.3$, $\mu_s = 0.28$).

![Figure 7](image)

**Fig. 7.** Experimental data and theoretical fits of change of $\epsilon_{\text{coh}}(l)$ and $\epsilon_{\text{adh}}(l)$ for a uniform probability density function of the strength as a function of the length of the coating with thickness of 150 µm

For a uniform probability distribution function of the coating strength, the value of the critical length is shifted towards higher values of the coating length (red area in Fig. 7) in comparison with the value calculated by the formula (8) (yellow
area in Fig. 7). Displacement will increase with growing values of the standard deviations for the distributions of strength. This displacement may reach 10% or more of the value of $L_{cr}$, which is defined by the formula (8), hence the form of the probability density function of strength has to be considered when determining the critical length of the coating.

5. Conclusion

The model describing the formation of the cracks and delamination was presented. It was shown that the crack space can have both constant and random values. The statistical criterion for determining the fragmentation length of the coating with mixed (adhesion-cohesion) failure was developed. Calculating the critical length as the intersection point of the two lines (interfacial failure shear stress-coating length curve and the normal failure stress-coating length curve) does not always give correct value. The critical length of coating should be evaluated as maximum value within the annular trapezoid. It is established that the kind of probability density function both the adhesion strength and the cohesion one should be considered when determining the critical length of the coating.

Literature

DEFINING THE CONTROL LAW FOR YAW MECHANISM CONTROL OF A TRICOPTER

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Abstract: Behavior of a specific constructive scheme of a tricopter is modeled. On the base of real characteristics of propeller-motor system, a model of the motion of the yaw mechanism is worked out. A flight stabilization method is proposed.

Keywords: TRICOPTER, YAW MECHANISM, FLIGHT STABILIZATION

1. Introduction

Copters are drones, which are capable of flying due to the lift created by the propellers. There are different design solutions such as tricopters, quadcopters, etc. Some of the possible design solutions are shown on fig.1.

A tricopter with an odd number of engines – three, was created. There are two possible design solutions - with unidirectional and mixed rotation of the front propeller. However, the problem with the compensation of torque remains. In order to keep the design at a low cost, the one with the unidirectional rotation of the propellers was chosen.(fig. 3)

2. Theoretical and experimental researches

The syntheses of the tricopter control algorithms, requires a mathematical model of the vehicle.

In order to derive the expressions for the forces and moments acting on the aerial vehicle, the following assumptions were made:

• The aircraft is a rigid body.
• The mass of the aircraft is symmetrically distributed.

From the definition of body frame it follows that the axes coincide with the major inertia axes of the aircraft (fig.4). Following these assumptions, the movement of the aircraft is described as a movement of a rigid body with 6 DOF.

To accurately describe the motion of a solid body with 6 DOF the Euler equations are used. They can be derived from the famous amount of movement alteration and kinetic moment alteration theorems. Thus, the free movement of the rigid body may be split into two movements:

• translational movement of the center of mass.
• rotation around the center of mass.
Fig. 4 Modeling the movement of a tricopter.

The equations for translational movement are derived from the amount of movement alteration theorem.

\[ m \frac{d \vec{V}}{dt} = \sum \vec{F} \]  

(1)

where: \( \frac{d \vec{V}}{dt} \) is the absolute acceleration;

\( \sum \vec{F} = \vec{R} + \vec{G} \) is the resultant of the external forces;

\( m \) is the mass of the tricopter.

Since the relevant forces and moments are presented in a coordinate system Oxzy (fig. 4), it is convenient for the absolute acceleration \( \frac{d \vec{V}}{dt} \) to be expressed through its projections in Oxzy:

\[ \frac{d \vec{V}}{dt} = \frac{d \vec{V}}{dt} + \vec{\omega} \times \vec{V} \]  

(2)

 ROTATION AROUND THE CENTER OF MASS IS DESCRIBED BY THE KINETIC MOMENT ALTERATION THEOREM.

\[ \frac{d \vec{K}}{dt} = \sum \vec{M} \]  

(3)

where: \( \sum \vec{M} \) is the resultant vector of the external moments;

\( \vec{K} \) is a vector of the kinetic moment, that is determined by the ratio:

\[ \vec{K} = \sum \left( \vec{r}_i \times m_i \vec{V}_i \right) \]  

(4)

where \( \vec{r}_i \) is the distance from the center of gravity to each point of the body; \( m_i \) is the mass of each point of the body; \( \vec{V}_i \) is the linear velocity of any point of the body.

In reference to fig. 4 the following equations describing the motion of the tricopter are derived:

1. For the balance of the forces:

\[ \sum F_x = -G_x \sin \theta \]  

\[ \sum F_y = F_A + F_B + F_C, c \quad \sin \theta = -G_y \]  

\[ \sum F_z = F_{F_z} - G_z \]  

(5)

2. For the balance of the moments:

\[ \sum M_x = (F_A - F_B) d \]  

\[ \sum M_y = F_C \sin \alpha d - \left( M_{A_y} f(F_A) + M_{B_y} f(F_B) + M_{C_y} f(F_C, \cos \alpha) \right) \]  

\[ \sum M_z = (F_A + F_B) d \cos \left( \frac{\theta}{2} \right) - F_C \cos \alpha d + M_{C_z} f(F_C, \sin \alpha) \]  

(6)

Taking into account expressions (5) and (6) and substituting them into expressions (1) and (3) we can obtain the differential equations of the motion of the tricopter.

To determine the position of tricopter in relation to the stationary coordinate system, the three Euler equations defining the relationship between angular velocities and angles of roll (\( \gamma \)), pitch (\( \upsilon \)) and sliding (\( \psi \)) are used. The coordinates of its center of mass are obtained using the kinematic equations linking the derivative of the coordinates of this center with the projections of its speed and angles of roll, pitch and sliding.

In solving the equations of motion of the tricopter, for the forces of thrust \( F_A, F_B, F_C \) and the reactive moment of the propellers \( M_x, M_y, M_z \) are used the results obtained from [2]. On fig. 5 and fig. 6 are presented the graphics illustrating the character of the change of the thrust and the torque for a single motor-propeller system.

The displayed graphs of the thrust and torque of an engine-propeller system are dependent on the rotational speed of the propeller.
According to expression (6) for balancing the tricopter in the horizontal plane $M_z = 0$. To offset the resultant reactive moment of the three blades, it is necessary for the tail mechanism of the tricopter to be rotated on a certain angle ($\alpha$) depending on the current rotational speed of the blades. The balancing curve of the variation of the angle of the tail plane when the tricopter is hovering is shown on fig. 7.

![Changing angle alpha](image)

**Fig. 7** Modification of the angle of the tail assembly as a function of the angular speed of rotation of the blades.

Using the complete mathematical model for such a body, a case is studied in which the tricopter hovers in a selected point above the surface of the Earth. The tricopter control is carried out by tilting the tail rotor so as to achieve the desired angle $\Psi_{\text{null}}$ in the horizontal plane.

In this case, the nonlinear mathematical model in expressions (1) and (3) becomes (7).

\[
\begin{align*}
\frac{d\omega_y}{dt} &= \frac{\sum M_y}{I_y}, \\
\frac{d\omega_z}{dt} &= \frac{\sum M_z}{I_z}, \\
\frac{d\psi}{dt} &= \omega_y.
\end{align*}
\]

(7)

So the nonlinear mathematical model expressed with differential equations (7) is linearized using the method of partial derivatives in the previously selected state vector $x = [\omega_y, \omega_z, \psi]$ and vector of control $u = [\omega_d, n_\omega_d, \alpha]$. The vector of control incorporates the angular rotational speeds of the front propellers, the rear propeller and the angular deviation of the tail mechanism of the tricopter. The output $y$ is drawn from the angular velocities along the axes Y and Z, as well as the angle of rotation in the horizontal plane $\psi$.

After linearization of the nonlinear model in expression (7), a model of the system in the state space is obtained—expression (8).

\[
\begin{align*}
\dot{x} &= Ax + Bu; \\
y &= Cx + Du.
\end{align*}
\]

(8)

The open system obtained following the model is on the border of stability.

Of the many options for setting the coefficients of the matrix of feedback $K$ a linear quadratic regulator (LQR) is selected. From the theory it is known that LQR indirectly gives account of the response of the system, the limits of the amplitude of controllable variables and the controlling impacts.

Determining a reference point for the linearization of the model is carried out based on the selected mode of hovering. In this case the following requirements have to be fulfilled - $\sum F_y = 0, \sum M_y = 0$ and $\sum M_z = 0$. From here shall be identified base values of the vector of control.

A specific feature of the use of LQR is determination of the weight matrices $Q$ and $R$.

\[
J = \int_0^\infty \left[ x^T(t)Q + u^T(t)R \left( \begin{array}{c} x^T(t) \\ u^T(t) \end{array} \right) \right] dt
\]

(9)

These matrices need to be positively determined. One way to initially set these matrices is as diagonal matrices with elements in the main diagonal with values equal to $1/(x_{\text{max}})^2$ and $1/(u_{\text{max}})^2$. In this case $x_{\text{max}}$ is the maximum variation of the corresponding element in the state vector, and $u_{\text{max}}$ is the maximum variation of the corresponding element of the vector of control.

After closing the system with a regulator-defined optimal matrix $K$, the results shown in fig.8, fig.9 and fig.10 are obtained.

**Fig. 8** Alteration of the angular velocity $\omega_y$.

![Angular speed W_y](image)

Fig.8 shows alteration of angular velocity $\omega_y$ depending on the control applied. A family of characteristics is obtained at different values of the weight factor $p$, expression (10).

\[
J = \int_0^\infty \left[ x^T(t)Q + u^T(t)R \left( \begin{array}{c} x^T(t) \\ u^T(t) \end{array} \right) \right] dt
\]

(10)

The next figure (fig.9) shows the change in the angular velocity of the tricopter along axis Z.
Фиг. 9 Alteration of the angular velocity $\omega_z$.

Again, there are obtained a family of characteristics with different weight factor $p$.

Фиг. 10 Altering the angle $\psi$ depending on the control applied.

In fig.10 is shown a family of characteristics, obtained at different weight factor $p$, of the angle of rotation in the horizontal plane of the tricopter. In the simulation, the desired control is supplied in the form of a step function (fig.10), a transitional process of the system is represented through a family of baseline characteristics.

A change of the control signal ($\psi_{\text{act}}$) in both directions ($\uparrow\Downarrow$), leads to a change in angular velocities along the axes $\text{Y}$ and $\text{Z}$, which is compensated by the regulator. The tendencies in the change of the variables in the simulations of the mathematical model correspond to the behavior of the real tricopter (fig.11).

3. Conclusions and results

1. The selected tricopter design requires a greater angle of alteration of the tail rotor to offset the resultant reactive moment;
2. The balancing angle of alteration of the tail rotor ($\alpha$) was calculated to provide zero angular velocities;
3. The resulting matrix $K$ is optimal, but due to the fact that a consistent theory for the initial determination of the weight matrices $Q$ and $R$ is not available, the behavior of the system with three different weight factors is simulated;
4. With the increase of $p$, the alteration of the angle $\psi$ approaches the desired values, but this results in increased $\omega_z$. Theoretically, it was found that when $p=10$ resulting angle speeds are safe for the tricopter;
5. An optimal control system of the rotation angle in the horizontal plane was proposed.

Fig. 11 Flight of the created tricopter.

4. Bibliography

NONMANEUVER AIRCRAFT WING DURABILITY DEPENDENCE ON DIRECTIVE STRESSES ESTABLISHED IN THE DESIGN

Abstract: The paper presents a method for calculating the durability of structure regular zones in view of working stresses and aircraft routine flight profile. Application of the method is advisable at the early design stages. The method is based on the discrete atmospheric turbulence model and dependence model for durability calculation by nominal stresses. Fatigue damage for a routine flight has been determined as the amount of damage due to accidental loads and the damage resulting from enveloping ground-air-ground cycle, naturally repeated during each flight. It has been noted that the consideration of the fatigue curve break point in the low-stress area leads to an increase in the calculated durability by 10-15%. A comparison has been made of the calculated durability values for two airplanes with the experimental data of TsAGI. The correlation is satisfactory enough.

KEYWORDS: WING, DURABILITY, ROUTINE FLIGHT PROFILE, WORKING STRESS, FATIGUE

1. Introduction

Durability requirements should be considered at the earliest stages of design by selecting the working stresses in structural elements. During calculations load-bearing structure is considered as consisting of the so-called regular zones and structural irregular zones. Regular zones include parts of the construction containing the unavoidable stress concentrators such as holes for rivets or bolts in assembled units. Structure durability is on the one hand limited by the durability of regular construction zones.

In some cases, working stresses are selected on the basis of statistical data on the flying airplanes of this class. In so doing, aircraft characteristics, routine flight profiles and the materials used have to be close or the same.

More promising is the approach based on atmospheric turbulence models. In this paper a discrete turbulence model has been adopted. Application of this model is efficient at the early design stages. The model prescribes the discrete assignment of cumulative frequency for vertical wind gust speeds per kilometer on the flight altitude.

Initial data for the calculation of load factors in the aircraft gravity center, and, consequently, the load on the wing include routine flight profile and atmospheric turbulence characteristics. Routine flight profile determines the changes in flight altitude, speed and weight of the aircraft, depending on the flight duration. At the design stage, this profile is determined by the planned data on weights, speeds and altitudes, and it is further refined using the results of flight measurements. When calculating the durability of long-haul airplanes the enveloping cycle is determined within the load program. This cycle determines largely the structure durability, being called the ground-air-ground cycle (GAG). For high aspect ratio wings external loads are further applied as shear forces, torsional and bending moments diagrams, because normal stresses determine the durability of these structures.

Experimental studies of load factors in flight suggest that maneuver load factor magnitudes and frequency to are too little to maintain flight modes of passenger and cargo planes [1]. The main fatigue damage result from load factors during flight in turbulent air. It is important to take into account the fatigue damage caused by both accidental loads and the enveloping ground-air-ground cycle (GAG).

2. Background for solving the problem of wing load calculation using the discrete gust scheme

The industry standard "Atmospheric turbulence model" OST 1 02514-84 determines the discrete vertical wind gust cumulative frequency per 1 kilometer of aircraft flight depending on the flight altitude. Cumulative frequency characterizes the number of gusts per 1 km of the flight greater than the given vertical speed

\[ F(w) = F_0 e^{-\frac{w}{C_w}}, \]

where \( F_0 \) and \( C_w \) are the parameters of the gust cumulative frequency, depending on altitude, \( F_0 \) is the total number of gusts per 1 km of the flight, \( F(w) \) is the number of gusts per 1 km of the flight with vertical speed exceeding \( w \). Vertical load factor increment \( \Delta n_y \) at the aircraft gravity center depends on the vertical gust speed

\[ \Delta n_y = \frac{C_{Qy} \rho_q V_{YW} W}{2p} \cdot K', \]

where \( K \) is the gust reduction factor determined as

\[ K = 0.8 \left(1 - e^{-\frac{\chi}{2}}\right) \cdot \frac{C_w}{2p} = \frac{\mu q}{\chi}, \]

where \( p \) is the wing specific load at the given aircraft weight \( m, N/m^2; \ q \) is gravitational acceleration, \( m/s^2; \ S_w \) is wing area, \( m^2; \ C_{Qy} \) is a derivative of the lift coefficient against the attack angle, \( 1/rad; \ H \) is the length of the trapezoidal gust transition section equal to 30 m; \( \rho_0 \) and \( \rho_1 \) are air density according to the table of standard atmosphere at sea level, and at height \( H \), respectively, \( kg/m^3; \ V_{YW} \) is the indicated airspeed corresponding to the considered altitude, \( m/s \).

3. Calculation of structure fatigue damage for routine flight

Each of three main flight phases, including climb, cruise flight and descend will be split into several modes in which aircraft speed, height and aircraft weight are constant. The total number of modes for the entire routine flight will be denoted by \( r \). Height \( H_j \), velocity \( V_j \), weight \( G_j \), the covered flight distance \( L_j \) are to be calculated for every \( j \)-th flight mode.

Formula (1) determines the number of gusts per 1 km of the flight in accordance with OST 1 02514-84. Considering \( L_j \) covered within the \( j \)-th mode, the number of gusts exceeding speed \( w \) will be:
Let us substitute the dependence (2) into (3) and introduce the following notation:

\[ C_{nj} = \frac{c_{nj}}{\varphi_{nj}^{2} p_{nj}} \cdot K_{j}. \]

Then

\[ F_j(\Delta n_y) = L_j F_{oj} e^{-\frac{\Delta n_y}{C_{nj}}}. \]

The probable exceeding of the load factor \( \Delta n_y \) in the j-th mode

\[ P_j(\Delta n_y) = \frac{F_j(\Delta n_y)}{L_j F_{oj}} = e^{-\frac{\Delta n_y}{C_{nj}}}. \]

The probability density of incremental load factor distribution can be determined as

\[ \phi_j(\Delta n_y) = \frac{dP_j(\Delta n_y)}{d\Delta n_y} = \frac{I}{C_{nj}} e^{\frac{-\Delta n_y}{C_{nj}}}. \]

The probability of falling into the interval \( d\Delta n_y \) is

\[ \phi_j(\Delta n_y) d\Delta n_y. \]

Since the total load factors number in the mode is \( L_j F_{oj} \), the increment of load cycle number equals

\[ dn = L_j F_{oj} \phi_j(\Delta n_y) d\Delta n_y, \]

or

\[ dn = \frac{L_j F_{oj}}{C_{nj}} e^{-\frac{\Delta n_y}{C_{nj}}} d\Delta n_y. \]

The accumulated fatigue damage in the j-th mode according to the linear hypothesis of a fatigue damage summation will be equal to

\[ D_j = \int \frac{dn}{N} = \frac{L_j F_{oj}}{C_{nj}} \left[ \frac{1}{N} e^{-\frac{\Delta n_y}{C_{nj}}} \right] d\Delta n_y. \]


to determine the number of cycles before failure \( N \) occurs at regular loads on structural element and at stresses corresponding to the incremental load factor \( d\Delta n_y \), we need to consider that positive values of vertical gust speeds correspond to the equivalent negative values. During the level flight maximum and minimum loads cycle load factors are as follows

\[ n_{y_{\text{max}}} = I + \Delta n_y, \quad n_{y_{\text{min}}} = I - \Delta n_y. \]

In accordance with the Oding formula [2] the equivalent incremental load factor corresponding to zero-to-tension loads cycle, can be determined as:

\[ \Delta n_{y_{\text{eq}}} = \sqrt{2 \Delta n_y (1 + \Delta n_y)}. \]

The equivalent stress of the zero-to-tension cycle will be equal to

\[ \sigma_{eq} = \sigma_{n_{y_{1}}} \cdot \Delta n_{y_{\text{eq}}}. \]

or

\[ \sigma_{eq} = \sigma_{n_{y_{1}}} \cdot \sqrt{2 \Delta n_y (1 + \Delta n_y)}. \]

During the aircraft flight in turbulent atmosphere high speed gusts are rare, but there is a great number of gusts with relatively low speed. These gusts result in the main fatigue damage. At low speeds of vertical gusts the incremental load factor and, consequently, the equivalent stress are also small. In order not to overestimate the accumulated fatigue damage from the effects of small gusts during the calculation, fatigue curve equation is thought to be a broken line with logarithmic coordinates, then

\[ N = N_o \left( \frac{\sigma_o}{\sigma} \right)^m, \]

where \( N_o \) and \( \sigma_o \) are coordinates of the structure element fatigue curve breaking point. The exponent of power \( m \) depends on the actual stress at \( \sigma > \sigma_o \) \( m = m_j \), if \( \sigma < \sigma_o \) then \( m = 2m_j - 1 \).

After substituting (6) into equation (7)

\[ N = N_o \left( \frac{\sigma_o}{\sigma} \right)^m \left( 2 \Delta n_y (1 + \Delta n_y) \right)^{\frac{m}{2}}. \]

The dependence used for calculation of the cumulative damage in the j-th mode (5) considering (8) is as follows:

\[ D_j = \frac{L_j F_{oj}}{N_o C_{nj}} \left( \frac{\sigma_{n_{y_{1}}}}{\sigma_{o}} \right)^m (2 \Delta n_y (1 + \Delta n_y))^{\frac{m}{2}} e^{-\frac{\Delta n_y}{C_{nj}}} d\Delta n_y. \]

The integral in equation (9) must be determined numerically. The upper limit of integration can be set to operating load factor. In numerical integration process the equivalent stress must be found according to (6) and the exponent value is to be set depending on whether the equivalent stress exceeds the stress value corresponding to the breaking point of structural element fatigue curve.

Fatigue damage caused by accidental loads in all modes of the routine flight can be determined as

\[ D_{\text{rand}} = \sum_{j=1}^{r} D_j. \]

Let’s consider the definition of fatigue damage within the GAG cycle. Equation (4) sets the number of exceedings of the incremental load factor \( \Delta n_y \) for the j-th mode of the routine flight. The total number of \( \Delta n_y \) value exceedings for the routine flight would be equal to
\[ F_{\text{rand}}(\Delta n_y) = F(\Delta n_y) \]
onumber

or
\[ F_{\text{rand}}(\Delta n_y) = \sum_{j=1}^{r} L_j F_{eq} \exp\left(-\frac{\Delta n_y}{C_{nj}}\right). \]

In accordance with the TsAGI regulations, to determine the maximum increment of load factor corresponding to the GAG cycle, the following parameter must be considered:
\[ (11) \quad F_{\text{rand}}(\Delta n_{y,\text{max}}) = 0.694. \]

Dependence (11) with the consideration of (10) provides a condition needed to determine the GAG cycle incremental load factor
\[ (12) \quad \sum_{j=1}^{r} L_j F_{eq} \exp\left(-\frac{\Delta n_{y,\text{max}}}{C_{nj}}\right) = 0.694. \]

Equation (12) can be easily solved numerically with respect to \( \Delta n_{y,\text{max}} \). Maximum GAG cycle load factor will be
\[ n_{y,\text{max}}^{\text{GAG}} = 1 + \Delta n_{y,\text{max}}. \]

The maximum stress in the structural element is calculated as
\[ \sigma_{\text{max}}^{\text{GAG}} = \sigma_{n_{y,\text{max}}^{\text{GAG}}} \cdot n_{y,\text{max}}^{\text{GAG}}. \]

The average value is usually taken as minimum GAG cycle stress
\[ \sigma_{\text{min}}^{\text{GAG}} = -L \cdot \sigma_{n_{y,\text{max}}^{\text{GAG}}}. \]

Coefficient \( L \) depends on the aircraft construction-load diagram and the given construction zone with respect to the landing gear arrangement. The average value of \( L \) is within the range of 0.5 - 0.8.

The equivalent stress of the zero-to-tension loads cycle corresponding to the GAG cycle can be determined as follows
\[ \sigma_{\text{eq}}^{\text{GAG}} = \sigma_{n_{y,\text{max}}^{\text{GAG}}} \cdot \Delta n_{y,\text{max}}^{\text{GAG}} + L. \]

The number of cycles before failure occurs at regular loads with these stresses can be derived from formula (7). Usually \( \sigma_{\text{eq}}^{\text{GAG}} \geq \sigma_{\text{d}} \) and \( m = m_1 \).

GAG cycle fatigue damage within one routine flight is
\[ D_{\text{GAG}} = 1 / N_{\text{GAG}}. \]

The total damage for a routine flight is
\[ D_{\text{total}} = D_{\text{GAG}} + D_{\text{rand}}. \]

The number of routine flights before failure of the structural element occurs can be determined as
\[ \lambda = 1 / D_{\text{total}}. \]

The structure durability expressed by the amount of routine flights will be
\[ T_{\lambda} = \frac{\lambda}{\eta_T}, \]

where \( \eta_T \) is the total safety factor.

The directive stresses (considered in the calculation of the ultimate load factor for structural element), which corresponds to the durability, can be determined as
\[ \sigma_{\text{dir}} = \sigma_{n_{y,\text{max}}^{\text{GAG}}} \cdot f. \]

where \( f \) and \( n_{y,\text{max}}^{\text{GAG}} \) are safety factor and operating load factor established in the structure design.

Knowing the required structure durability, the required value of \( \sigma_{n_{y,\text{max}}^{\text{GAG}}} \) can be selected.

4. Findings

The aircraft receives the greatest fatigue damage during and its climb and descend phases. In cruising mode, despite its long duration, it is damaged less. In this regard, the aircraft durability should be rather expressed in the number of flights instead of flight operation hours.

When selecting routine flight parameters it is rational to limit the flight speed at low altitudes, focusing not only on the optimal speed for fuel consumption, but also for construction fatigue damage.

It is challenging to compare the results of durability calculations depending on the working stresses with experimental data already determined.

Figure 1 shows the dependence between the regular zones durability of the lower wing panels made of D16T and the calculated working stresses for Tu-134 and Il-76 routine flight profiles. Markers denote the TsAGI experimental data given in [3] for the lower wing panels made of D16T on modern passenger airplanes with an average reliability coefficient equal to three.

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**Fig. 1. Structure durability dependence on the working stresses.**

Markers denote the experimental data. The lines show the calculated dependences for two airplanes.

The correlation of the calculated structure durability depending on the working stresses is satisfactory enough.
It should be noted that consideration of structure material fatigue curve breaking point in the low-stressed area results in the increase of durability by 10-15%.

The weight of the aircraft decreases in the routine flight due to the fuel consumption leading to both a greater load factor caused by the gusts, and to smaller stresses on structural elements with a single load factor because of the wing load relief through fuel consumption. The lifting force of the wing, including the bending moments and normal stresses, depends on the load factor value in the aircraft gravity center and the value of balancing load on the horizontal tail. In the early design stages the balancing load may be neglected, whereas the stress caused by the single load factor can be represented as the stress (with the aircraft weight calculated only) multiplied by the ratio of the actual aircraft weight to the calculated one.

If the changing stress is not considered at the single load factor, it can lead to errors in structure durability calculation. The greater the change in aircraft weight due to fuel consumption is, the greater might be the error.

The aircraft designers tend to obtain equal structure strengths and, if possible, the same durability for regular and irregular structural zones. Irregular structure zones include panel transverse joints, stringer joints and end zones, cutouts and skin joints etc.

Since the stresses caused by the single load factor and the effective stress concentration factor in irregular structure zone [2] are included in the equation for calculation of fatigue damage (due to accidental loads and the GAG cycle) as a product, one should follow the recommendations below to design irregular structure zones:

- one should provide the lowest possible value of the effective stress concentration factor by soft start of the force elements, with the bearing stress on fasteners reduced;
- taking into account the provided value of the effective stress concentration factor, one should tend to decrease the working stresses in irregular structural zone.

5. Conclusion

1. The calculated working stresses providing the required structure durability well correlate with the TsAGI experimental data. It is necessary to take into account the fatigue curve breaking point and the change in stresses caused by the single load factor due to fuel consumption.

2. The durability calculation method based on the discrete gust scheme, allows setting the dependence between the working stresses and the required structure durability for a particular structure material using its fatigue curve and the aircraft routine flight profile.

3. The structure gets the greatest fatigue damage at low flight altitudes, in this connection, it is necessary to establish the optimal climb and descend speeds by both fuel consumption and the durability limits.

6. References


DETERMINATION OF AIRCRAFT WING LOADS ON THE ROUTINE FLIGHT MODES WITHIN STRUCTURAL ELASTIC VIBRATIONS

ОПРЕДЕЛЕНИЕ НАГРУЗОК НА КРЫЛО САМОЛЕТА НА ЭТАПАХ ТИПОВОГО ПОЛЕТА С УЧЕТОМ УПРУГИХ КОЛЕБАНИЙ КОНСТРУКЦИИ

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Abstract: The article deals with a problem of fatigue load calculations when designing the aircraft wing. Much of fatigue damage is done to the wing by occasional air gusts in turbulent atmosphere. Therefore, the author considers the pattern of continuous atmospheric turbulence that gives an idea of low-altitude gusts. It is at low altitudes that nonstandard flights are performed for which it is wrong to use the statistical data on fatigue damage available from the previously operated airplanes of the same class. The method is offered to determine the equivalent bending moments along the wing span in discrete flight modes and within the entire routine flight. The above method takes into account the profile parameters of routine flights, the dynamic vibrations of the wing and fatigue characteristics of the structural material. To prove the reliability of the method the calculation results have been compared to those of medium-range aircraft testing flights.

KEYWORDS: WING, DURABILITY, DYNAMICS, VIBRATIONS, TURBULENCE, LOAD FACTOR, EQUIVALENT MOMENT

1. Introduction

Requirements for modern aircraft frame durability become ever more sophisticated. Improved calculation techniques for load-bearing structure durability require determining of the loads acting in various aircraft flight modes.

When manufacturing an airplane, the operating conditions must be determined such as routine flight profiles specified by dependences between weight, velocity, altitude and flying hours. The main parameter to determine the intensity of structural load is a load factor in the aircraft center of gravity in various flight modes. The use of statistical data on this parameter is problematic, because they depend both on the aircraft performance and routine flight profiles. Along with the standard flight profiles the aircraft can perform training flights, firefighting fights, etc. These profiles cause fatigue damages which significantly differ from the typical fatigue damages of structural elements.

2. Problem solution

When the aircraft is being designed, consideration must be given to the routine flight profile that provides initial data to determine the structure durability by the airframe fatigue behavior. In flight, the aircraft is continuously exposed to random wind gusts. For non-maneuverable aircrafts, the worst damage to structural elements is done by vertical air gusts. To calculate the fatigue damage, a model of continuous atmosphere turbulence has been adopted under the Industry Standard [1]. The spectral density of intensity of wind gust vertical speed is determined as follows:

\[ \Phi_w(\Omega) = \frac{L \cdot \sigma_{w}^2}{\pi} \frac{1 + \frac{8}{3} \left(1.339 \cdot L \cdot \Omega \right)^2}{\left[1 + \left(1.339 \cdot L \cdot \Omega \right)^2\right]} \]

where \( \Omega \) is space frequency, \( L \) is integral turbulence scale, \( \sigma_{w} \) - intensity of wind gust vertical speed.

The density function of standard deviation for the vertical component of wind gust speed during the aircraft flight in turbulent atmosphere is determined as follows:

\[ f(\sigma_w) = \frac{1}{\sqrt{2\pi}} \frac{P_1}{b_1} \exp \left(-\frac{\sigma_{w}^2}{2b_1^2}\right) + \frac{1}{\sqrt{2\pi}} \frac{P_2}{b_2} \exp \left(-\frac{\sigma_{w}^2}{2b_2^2}\right), \]

where \( P_1 \) and \( P_2 \) are probabilities of flight in mild and intensive turbulence zones, \( b_1 \) and \( b_2 \) are coefficients of mild and intensive turbulences.

Standard methods [2, 3] in aviation rely on linear hypothesis for calculating the durability as a sum of the fatigue damages caused by random loading

\[ \int_{0}^{\sigma N(\sigma)} \frac{dn}{\sigma} = 1, \]

where \( dn \) is an increment of the number of load cycles

\[ dn = N_\Sigma \cdot \phi(\sigma)d\sigma, \]

where \( \phi(\sigma) \) is the stress distribution density, \( N_\Sigma \) is the total number of cycles before the failure

\[ N_\Sigma = N_{0j} \cdot T_i; \]

where \( T_i \) is the durability for flying in \( \sigma_{w} \) which is the constant intensity zone, \( N_{0j} \) is the number of mean loads in the j-th flight mode. According to Rice’s formula it is determined as

\[ N_{0j} = \frac{1}{2\pi} \int_{\Omega} \frac{\Phi_M(\Omega) \cdot \Omega^2 d\Omega}{\int_{\Omega} \Phi_M(\Omega) d\Omega}, \]

where: \( \Phi_M(\Omega) \) is the spectral density of bending moment power.

In formula (4), \( \phi(\sigma) \) shall be determined according to Rayleigh’s distribution law

\[ \phi(\sigma) = \frac{\sigma}{S_\sigma^2} \cdot e^{-\frac{\sigma^2}{2S_\sigma^2}}, \]

where \( S_\sigma^2 \) is the power factor dispersion.

Spectral power density of the bending moment is equal to

\[ \Phi_M(\Omega) = \Phi_w(\Omega) \cdot H_M^2(\Omega), \]
where: $H_M(\Omega)$ is the transfer function of the bending moment under the action of vertical wind gust with sinusoidal speed variation in the $j$-th mode of routine flight. We offer to determine it as follows [5]:

\[
H_M(\Omega) = M_j \frac{h_j}{g_j} \left[ \frac{1}{\Omega^2 + \left( \frac{h_j}{V_uj} \right)^2} \right]^{1/2} \left[ \frac{1}{1 + \pi \cdot b \cdot \Omega} \right],
\]

where $M_j$ is the bending moment value of the $j$-th mode of routine flight under vertical load factor $n_v = 1$;

\[
h_j = \frac{\rho_j \cdot V_uj}{2 \cdot g_j} \cdot C_y \cdot S_w,
\]

where $\rho_j$ is the air density according to table of standard atmosphere at altitude $H$; $g_j$ is the free-fall acceleration at altitude $H$; $V_uj$ is indicated flight speed at the considered altitude, modified to the air density near the ground; $C_y$ is the derivative of the lift force coefficient on the attack angle; $S_w$ is the wing area, $b$ is the mean geometric chord of the wing.

Integrating the diagram of load per unit of length in [6] we obtained formula [10] of simplified dependence between the aircraft weight, filled wing tank weight and the bending moment in various cross-sections of the wing.

\[
M_j = \frac{Y_{\omega j}}{G_{y0}} \cdot a(z) - \frac{G_{Ej}}{G_{y0}} \cdot b(z) - c(z),
\]

where $Y_{\omega}$, $G_{E}$ are the wing lift force and the fuel weight in the routine flight mode; $G_{y0}$, $G_{y0}$ are the aircraft and fuel weights, which have determined the functions $a(z)$, $b(z)$, $c(z)$. Functions $a(z)$, $b(z)$, $c(z)$ are the bending moments due to air and mass loads by the weights of fuel, wing and wing concentrated loads.

Substituting expressions (1, 2, 4-12) in formula (3) and making some transformation, we obtain the dependence for calculation of the SD of the bending moment in various flight modes, characterized by atmospheric turbulence in the routine flight mode under consideration

\[
d_j = \frac{N_0 \cdot 2^m \cdot C_{\sigma w}^m \cdot I_y \cdot I_W \cdot l_j}{C_{\sigma w}},
\]

where

\[
I_y = \int_y^{\frac{m}{2}} e^{-\gamma \cdot y^2} dy; \quad I_W = \int_0^{\infty} f(\sigma_w) d\sigma_w;
\]

$l_j$ is the length of the $j$-th routine flight mode; $C_{\sigma w}^m$ is the coefficient that characterizes the dependence between the SD of the vertical gust speeds and the SD of the bending moment

\[
C_{\sigma w} = \left[ \int_0^{\infty} \sigma_{\omega}^m \cdot f(\sigma_w) d\sigma_w \right]^{1/2}.
\]

This particular damage can be added by the application of one non-zero loading cycle with the maximal value of $M_{\text{eq}}$

\[
d_j = \left[ \frac{\sigma(M_{\text{eq}})}{C} \right]^m.
\]

As the left parts of equations (13) and (15) are equal, we can make their right parts equal too, where the equivalent bending moment in the $j$-th mode of the routine flight will be equal to

\[
M_{\text{eq},j} = \sqrt{\frac{N_0 \cdot 2^m \cdot C_{\sigma w}^m \cdot I_y \cdot I_W \cdot l_j}{C_{\sigma w}}}.
\]

To determine the equivalent loads during the aircraft operation in combined modes, it is necessary to know the value of equivalent bending moment during the routine flight. It can be determined similarly to (16), taking into account that the total damage in (13), which occurs due to the turbulent atmosphere in the routine flight, is equal to

\[
d = d_{\text{gag}} + \sum_j d_j,
\]

where: $d_{\text{gag}}$ is the damage resulting from the ground-air-ground (GAG) cycle. To calculate the loads of the GAG-cycle, the value of bending moment in level flight is determined as

\[
M_{\text{lev}} = \sum_j M_j \cdot d_j.
\]

The regularity of the bending moment increments in separate routine flight modes is determined according to [1]

\[
F_j(\Delta M_{\text{gag}}) = N_0 \cdot \tau_f \cdot \left[ F_{l_1} \cdot e^{-b_1^j \cdot \tau_f} + F_{l_2} \cdot e^{-b_2^j \cdot \tau_f} \right],
\]

where $A_{w_1}$ is the ratio between the vertical gust transfer function and the bending moment increment for every $j$-th mode of the routine flight

\[
A_w = \left[ \frac{\Phi_{\omega}(\Omega) d\Omega}{\int_0^{\infty} \Phi_{\omega}(\Omega) d\Omega} \right].
\]

The total number of exceeded $\Delta M_{\text{gag}}$ in the routine flight is

\[
F_c(\Delta M_{\text{gag}}) = \sum_j F_j(\Delta M_{\text{gag}}).
\]

According to TsAGI regulations to determine the maximal GAG-cycle bending moment increment, the below parameter must be

\[
F_{\text{rand}}(\Delta M_{\text{gag}}) = 0.694.
\]

Maximal and minimal GAG-cycle bending moments in the flight are equal to

\[
M_{\text{max}} = M_{\text{lev}} + \Delta M_{\text{gag}}; \quad M_{\text{min}} = -0.5 \cdot M_{\text{lev}}.
\]

The GAG-cycle loads need to be modified to the non-zero cycle maximum by Odling’s formula

\[
M_{\text{gag},0} = \sqrt{M_{\text{max}} \cdot (M_{\text{max}} - M_{\text{min}})}.
\]

3. Findings and discussion

The method described has enabled the calculation of the bending moment along the wing of medium-range aircraft with the take-off mass of 42 t, wing span of 32 m and with 30 ribs in the wing. Fig 1. shows the typical firefighting flight profile including 7 water lifts:
The bending moment calculations are done for the next modes of routine flight (see fig. 1): 0-1 – taxiing, 1-2 – takeoff, 2-3 – climbing up to 2000 m, 3-4 – level flight at 2000 m, 4-5 – descending from 2000 m, 5-6 – level flight at 400 m, 6-7 – descending from 400 m, 7-8 – water lift, 8-9 – climbing up to 400 m, 9-10 – level flight at 400 m, 10-11 – descending from 400 m, 11-12 – water release, 12-13 – climbing up to 400 m, 13-14 – climbing up to 2000 m, 14-15 – level flight 2000 m, 15-16 – descending from 2000 m, 16-17 – circling and landing.

The power equation for fatigue behaviour curve has been accepted for maximal "brutto" stresses, MPa within non-zero loading cycle

\[ N \cdot \sigma^m = C, \]

where \( m, C \) are experimentally determined parameters. For the most conventional wing alloy D16T, \( m = 4, C = 1.767 \times 10^9 \).

Figures 2 – 5 show the dependence between the calculated equivalent bending moment and \( M_{eq} \) which is the value obtained after processing the flight test data from spanwise cross-sections of the wing in different flight modes. The line shows the equality between the calculated \( M_{eq} \) and the experimental \( M_{eq} \).

Figure 6 shows the dependence between the calculated equivalent bending moment and \( M_{eq} \) in different wing cross-sections within the entire routine firefighting flight.

4. Conclusion

The equivalent bending moment for the entire flight calculated by the offered method is 10% different from the equivalent bending moment obtained from the flight test. The offered calculation method for wing loads in routine flights can be used when the durability must be estimated for regular zones of the wing.
Fig. 6. Dependence between the calculated equivalent bending moment and $M_{eq}$ obtained from the flight test data in various wing cross-sections within the entire routine firefighting flight.

The divergence of the results can be explained by the fact that to calculate the transfer function of the bending moment affected by the wind gust, only the aircraft gravity centre vibrations have been taken into account. According to the preliminary research [8], the consideration of the wing vibration against the aircraft gravity centre can increase the wing spanwise bending moments by 20% which would be the object of our future research.

5. References

1. Introduction

Stress concentration take place when axial forces are loaded on threaded joints. Under ideal conditions, the tension in the screw (bolt) and the compression in the nut (such scheme of loading is called Bolt-Nut I) should be reduced uniformly starting from full load at the first contact between screw and nut. The same condition is required for other schemes of loading when the screw is compressed and the nut is tensioned (Bolt-Nut II) or both screw and nut are tensioned (Tightener) or compressed (Post). However, the pitches of the screw and the nut increase or decrease so that correct compliance between the loaded threaded portions is not maintained. The major portion of the load is transferred at the pair of contacting threads near the bearing nut face, a large stress concentration is present here, and most screw failures occur at this cross section. So, it is very important to know the load on the most loaded threads. Despite the fact that for the first time the load distribution on the thread was considered by Joukovsky, N.E. [1] many researchers developed this subject up to date.

Considerably later Birger, I.A. [2], unlike Joukovsky which has given the solution of this problem for discrete model of loading, has obtained the solution for continuous threads in differential form. This solution is widely used for calculation of the thread strength.

Considering that the intensity \( q(z) = \frac{dF(z)}{dz} \) of axial force \( F(z) \) on a unit of the length of the threaded joint in any cross section \( z \) is the force \( F(z) = \int_0^z q(z)\,dz \) and then a load \( F(i) \) on any thread located between cross sections \( z_i \) and \( z+iP \) (here \( P \) is pitch thread) will be equal to

\[
F(i) = \int_{z_i}^{z_i+P} q(z)\,dz = \int_{0}^{z} q(z)\,dz \quad (1)
\]

Unlike Birger’s solution for two schemes of loading Bolt-Nut I (Fig.1a) and Tightener (Fig.1c) in work [3] the schemes of loading Bolt-Nut II (Fig-1b) and Post (Fig.1d) were considered. In these solutions the friction forces on the thread flanks and radial deformations of the screw shank and nut body in conformity with Poisson’s factor were taken into consideration.

2. Load distribution on the threads for different schemes of loading

For the scheme of loading on Fig.1a a condition of joint deformations will be as follows

\[
\Delta_1 + \Delta_2 = \delta_1(z) + \delta_2(z) = \Delta_1(0) + \Delta_2(0) + \delta_1(z) + \delta_2(z)
\]

where \( \Delta_1 = \int_0^z \frac{\sigma_1(z)}{E_1} dz \) and \( \Delta_2 = \int_0^z \frac{\sigma_2(z)}{E_2} dz \) are screw elongation and nut contraction, accordingly. \( \Delta_1(0) + \Delta_2(0) \) is the axial displacements of the screw and nut threads along the pitch diameter \( d_2 \) in cross sections \( z = 0 \) (point 0) and \( z \) (point A). \( \sigma_1(z) \) and \( \sigma_2(z) \) are stresses in the cross section \( z \) of a screw shank and a nut body. \( E_1 \) and \( E_2 \) are the moduli of elasticity of screw and nut materials.

![Fig. 1. Calculation scheme and schemes of loading of threaded joints.](image)

The axial displacements of the threads along the pitch diameter \( d_2 \) are

\[
\delta = \delta^s + \delta^r \quad \text{and} \quad \delta' = \delta_b + \delta_{shb} + \delta_f
\]

where \( \delta_b \) and \( \delta_{shb} \) are displacements due to the thread bending and shearing; \( \delta_f \) is displacement due to the radial deformation from forces on the thread flank; \( \delta^r \) is axial deformation in consequence of the Poisson’s factor \( \mu \). Assuming that a unit pressure on thread flanks in cross section \( z \) to be \( p(z) \), the values of \( \delta^s_1(z) \) and \( \delta^s_2(z) \) can be expressed as follows

\[
\delta_1(z) = \frac{p(z)P}{E_1} \lambda_1 \quad \text{and} \quad \delta_2(z) = \frac{p(z)P}{E_2} \lambda_2
\]

where \( \lambda_1 \) and \( \lambda_2 \) are non-dimensional coefficients depending on the thread profile.

As to the values of \( \delta^s_1(z) \) and \( \delta^s_2(z) \) they can be found from equations

\[
\delta^s_1(z) = \mu_1 \cdot \frac{\sigma_1(z)dz}{2E_1} \tan \frac{\alpha}{2} \quad \text{and} \quad \delta^s_2(z) = \mu_2 \cdot \frac{\sigma_2(z)dz}{2E_2} \tan \frac{\alpha}{2}
\]

where \( \alpha/2 \) is loaded flank angle.

The values of \( \delta^s_1(z) \) and \( \delta^s_2(z) \) are positive if a clearance between the engaged threads to be increased and are negative if the clearance between the engaged threads to be decreased. Taking into consideration these facts and that the stress conditions depend on the schemes of loading of the threaded joints, the axial loads \( F(z) \) from Eqs.(1)-(5) will be as follows:

\[
F(z) = F_0 \frac{\alpha(z)Hz}{\sinh \beta z} \frac{\sinh \beta z}{\sinh \beta H}
\]
and for schemes of loading Tightener and Post

\[ F(z) = \frac{F}{\beta} \left[ e^{\alpha z} \frac{\sinh \beta z}{\sinh \beta h} \frac{1}{E_1 A_1} + \frac{e^{\beta z}}{E_2 A_2} \right] \]

where \( e = 2.71 \) is the basis of natural logarithm; \( a = \xi / (2\gamma) \);
\( b = \sqrt{a^2 + m} \); \( m = \beta / \gamma \); \( \beta = 1 + \frac{1}{E_1 A_1} \);
\( \xi = \left( \frac{\mu_1}{E_1 A_1} + \frac{\mu_2}{E_2 A_2} \right) d_2 \frac{\alpha}{2} \); \( \gamma = \frac{P^2}{A_1} \left( \frac{\lambda_1}{E_1} + \frac{\lambda_2}{E_2} \right) \); \( A_1 \) and \( A_2 \) are screw and nut cross-sectional areas; \( A_t = \pi d_2 \); \( t_2 \) is thread overlap. In Eqs. (6) and (7) upper signs correspond to the schemes of loading Bolt-Nut I and Tightener.

For threads having asymmetric thread profile in Eqs. (5) - (7) instead of \( \alpha / 2 \) it is necessary to take the loaded flank angle.

As to the non-dimensional coefficients \( \lambda_1 \) and \( \lambda_2 \), they are to be determined for different thread profiles.

3. Determination of coefficients \( \lambda_1 \) and \( \lambda_2 \)

3.1. Triangular threads

Considering the screw shank and the nut body as tubes, loaded with external and internal pressures, the axial thread displacements along the pitch diameters due to the radial deformations in Eq.(3) are equal to

\[ \delta_{1f} = \frac{d_2}{2E_1} \left( \frac{\frac{d_2^2}{2} + \frac{d_2^2}{2} - \frac{d_0^2}{2}}{\frac{d_2^2}{2} - \frac{d_0^2}{2}} - \frac{\mu_1}{P} \right) \tan \alpha \frac{\alpha}{2} - \tan \rho \frac{\alpha}{2} \]

\[ \delta_{2f} = \frac{d_2}{2E_2} \left( \frac{\frac{d_2^2}{2} + \frac{d_2^2}{2} + \frac{d_2^2}{2}}{\frac{d_2^2}{2} - \frac{d_0^2}{2}} + \frac{\mu_2}{P} \right) \tan \alpha \frac{\alpha}{2} - \tan \rho \frac{\alpha}{2} \]

where \( d_0 \) is diameter of a hole in the screw shank; \( d_2 \) is equivalent external diameter of the nut; \( \rho \) is friction angle. The friction forces on the thread flank prevent radial displacements of the screw and nut bodies and increase the axial displacements of the threads.

![Fig. 2. Calculation scheme for triangular thread profiles.](image)

To determine the values \( \lambda_1 \) and \( \lambda_2 \) for Eq.(4) it is necessary to find the values of \( \delta_{1f}^* \) and \( \delta_{2f}^* \).

In works [2] and [3] these values for ideal theoretical (or nominal) thread profiles were obtained. However, at thread manufacturing, the main thread dimensions have allowances depending on dimensions of thread, accuracy class and fit.

If the allowances are small, the determination of the thread deflection can be calculated for the threads with nominal thread dimensions, but it is necessary, unlike Birger [2], to take into account the friction forces \( F_I \) acting on the thread flanks. When the allowances are great, the value of the thread overlap \( t_2 \) decreases and the pressure on the thread is displaced to the thread crest.

Analysis of the coefficients for Eqs. (4), (6) and (7) shows, on the one hand, that increasing of value \( t_2 \) (Figs.2b and 2c) increases the values \( a \) and \( b \) and, on the other hand, the decreasing of thread deflection decreases values of \( \lambda_1 \) and \( \lambda_2 \).

Owing to these facts for real threads, being manufactured with allowances, the load on the most loaded thread will be increased.

It need to be noted that the standard threads having different thread profiles have definite relationships of thread profile dimensions to the thread pitch \( P \). Therefore, to determine the values of \( \lambda_1 \) and \( \lambda_2 \) in Eq. (4) it is necessary, in calculations, to take the thread profile dimensions to be expressed as a part of \( P \). Hence, on calculation of values \( \lambda_1 \) and \( \lambda_2 \) for threads with allowances, it is necessary to use real thread dimensions and received values of the deflections \( \delta \) to divide by \( P \). Calculation scheme is shown in Fig.2a.

Vertical displacement of point \( M \) from bending, \( \delta_M \), and shearing, \( \delta_s \), can be found by Maxwell-Mohr’s reciprocal theorem for flexibilities based on Clapeyron’s second theorem. The thread is assumed to be as a cantilever beam with thickness equal to unit.

For the screw thread

\[ \delta_{sh} = \frac{1}{E_1 b} \int_a^b M(x) \frac{M_0(x) dx}{I_1(x)} = \frac{1}{E_1 b} \left[ \frac{M_p(x)}{I_1(x)} + \frac{M_f(x)}{I_1(x)} \right] \]

where \( M_p(x) = \frac{1}{2} \rho (x-a)^2 - \left( x^2 - a^2 \right)^2 \) is the bending moment from \( p \); \( M_f(x) = p \left( \frac{\alpha}{2} - \tan \rho x \right) \) is the bending moment from friction forces; \( M_0(x) = F_0(x-b)=1-(x-b) \)
\( F_0 = 1 \) is the auxiliary unit force; \( I_1(x) = \frac{3}{2} \left( x^3 \tan \frac{\alpha}{2} \right) \) is the moment of inertia of the cross section. Solution of the integral gives

\[ \delta_{sh} = \frac{3p}{4E_1 \tan^2 \frac{\alpha}{2}} \left[ \left( 1 - \tan^2 \frac{\alpha}{2} \right) \left( c - b \right) + \left( \tan^2 \frac{\alpha}{2} - 1 \right) - 2a \right] \ln \frac{c}{b} \]

\[ + \left( a^2 - 1 + \tan^2 \frac{\alpha}{2} \right) + 2ab \]

\[ + 2a b \left( \frac{c - b}{c - b - a + \ln \frac{c}{b}} \right) \frac{\tan^2 \rho}{c} \]

\[ + 2a b \left( \frac{\tan^2 \rho}{c} \right) \frac{\tan^2 \rho}{c} \frac{\tan^2 \rho}{c} \]

For the nut threads the value of \( \delta_{no} \) can be calculated by Eqs.(9) and (10) but with values of \( b, c, a \) and \( E_2 \) for a nut and \( I_2(x) = I_1(x) \frac{d_2}{d_1} \), where \( D \) is the external diameter of nut thread

\[ \delta_{sh} = \frac{1}{G_1 t_1 A_{sh}(x)} \int_a^b \frac{F_{sh}(x) dx}{A_{sh}(x)} \int_a^b \frac{F_{sh}(x) + F_f(x) dx}{A_{sh}(x)} \]

where \( F_p = p(x-a) \) and \( F_f = p(x-a) \tan \alpha / 2 \) are vertical forces; \( A_{sh} = 2x \tan \alpha / 2 \) is cross-sectional area in shearing;
\( G_1 = \frac{E_1}{2(1 + \mu_1)} \) is the modulus of elasticity in shear; \( K = 1.2 \) for short cantilever beam. Solution of the integral gives

\[ \delta_{sh} = \frac{1.2}{E_1 \tan(\alpha/2)} \left( 1 + \tan \frac{\alpha}{2} \right) \left( c - b - a \ln \frac{c}{b} \right) \]
For the nut threads it is necessary to take into account the values of $b$, $c$, $a$, $\mu_2$ and $E_2$ for nut and $A_{2sh} = A_{sh} D/d_1$.

### 3.1.1 Determination of values of $\delta_\beta$ and $\delta_{sh}$ for real threads

The calculation scheme for the thread having dimensions with allowances is shown on Fig. 2b. For this scheme $e = c - C'$ and $b = a + (t_3 - C')/2$.

The values $C'$ must be determined for definite thread fit and class of accuracy.

$$\delta_{sh} = \frac{1}{E_1} \int \frac{M(x)M_0(x)}{I_1(x)} \frac{dx}{E_1} + \frac{1}{E_1} \int \frac{M(x)M_0(x)}{I_1(x)} \frac{dx}{E_1} \tag{13}$$

Solution for the first term of Eq.(13) gives the same result as for Eq.(9) after substitution of $c$ by $e$. For the second term of Eq.(13)

$$M(x) = p(e-a) \left( x-b-b \tan \frac{\alpha}{2} + x \tan \frac{\alpha}{2} \tan \rho \right)$$

and $M_0(x) = x-b$

Then the displacement of point $M$ for screw thread will be as follows:

$$\delta_{sh} = \frac{3p}{4E_1 \tan(\alpha/2)} \left[ \frac{1}{2} \tan \frac{\alpha}{2} (e-b) + \frac{1}{2} \tan \frac{\alpha}{2} - 2a \right]$$

$$+ \frac{e-b}{2a^2} b + \frac{1}{2} \tan \frac{\alpha}{2} \left[ e - b - (a + b) \ln \frac{e}{b} + a \left( 1 - \frac{b}{c} \right) \right]$$

$$+ \frac{2(e-a)}{\tan \rho} \left[ \frac{1}{2} \tan \frac{\alpha}{2} \ln \frac{c}{e} - 2b(e-a) \frac{c-e}{ce} \right]$$

$$+ \left[ 2 + \tan \frac{\alpha}{2} + \tan \frac{\alpha}{2} \tan \rho - h \left( 1 + \tan \frac{\alpha}{2} \frac{c+e}{ce} \right) \right]$$

For the nut the value of $\delta_{sh}$ in this case is determined with remarks given above.

$$\delta_{sh} = \frac{K}{G_1} \int \frac{F_{sh}(x)}{A_{sh}(x)} \frac{dx}{A_{sh}(x)} \tag{15}$$

Solution for the first term of Eq.(15) gives the result as in Eq.(12) after substitution of $c$ by $e$.

For the second term

$$F_{sh}(x) = F_p + F_f = p(e-a) \left[ 1 + \tan \rho \tan(\alpha/2) \right]$$

then

$$\delta_{sh} = \frac{1.2p(1+\mu_1)}{E_1 \tan(a/2)} \left[ 1 + \tan \frac{\alpha}{2} \tan \rho \left[ e - b - a \ln \frac{e}{b} + (e-a) \ln \frac{c}{e} \right] \right]$$

For the nut thread must be observed the same remarks as it were given above.

From Eqs.(3), (4), (8) and the results, received for $\delta_\beta$ and $\delta_{sh}$, the following equations will be:

$$\lambda_1 = A_1 + B_1 \tan \rho + \frac{d_1^2+\mu_1}{2P^2} \left( \frac{d_2^2 + d_0^2}{d_2^2 - d_0^2} - \mu_1 \right) \tan \frac{\alpha}{2} \left( \tan \frac{\alpha}{2} - \tan \rho \right)$$

and for nut

$$\lambda_2 = A_2 + B_2 \tan \rho + \frac{d_2^2+\mu_2}{2P^2} \left( \frac{d_1^2 + d_2^2}{d_1^2 - d_2^2} + \mu_2 \right) \tan \frac{\alpha}{2} \left( \tan \frac{\alpha}{2} - \tan \rho \right) \tag{17}$$

To show the influence of the allowances the values for $A$ and $B$ were obtained next results:

<table>
<thead>
<tr>
<th>Kind of thread (on $\mu_1 = \mu_2 = 0.3$; $f = 0.1$)</th>
<th>Screw thread</th>
<th>Nut thread</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bolt-Nut I</td>
<td>A</td>
<td>B</td>
</tr>
<tr>
<td>All metric threads with any pitch when allowances are equal to zero</td>
<td>0.8328</td>
<td>0.7203</td>
</tr>
<tr>
<td>M24x1-7G/7e6e</td>
<td>1.2203</td>
<td>0.7565</td>
</tr>
<tr>
<td>M24x2-7G/7e6e</td>
<td>1.3598</td>
<td>0.9124</td>
</tr>
<tr>
<td>M24x3-7G/7e6e</td>
<td>1.5350</td>
<td>0.9830</td>
</tr>
</tbody>
</table>

These values of $A$ and $B$ illustrate that the less the thread pitch for the same nominal thread dimension the greater the influence of the allowances on the axial thread displacements.

The loads on the most loaded thread are shown in Table 1.

<table>
<thead>
<tr>
<th>Thread pitch, $P$, mm; number of threads in joint, $n$</th>
<th>Scheme of loading</th>
</tr>
</thead>
<tbody>
<tr>
<td>[Tightener, Post]</td>
<td>Bolt-Nut I</td>
</tr>
<tr>
<td>$P=1$, $n=18$</td>
<td>12.24</td>
</tr>
<tr>
<td>$P=1$, $n=18$</td>
<td>11.19</td>
</tr>
<tr>
<td>$P=2$, $n=9$</td>
<td>22.42</td>
</tr>
<tr>
<td>$P=2$, $n=9$</td>
<td>20.32</td>
</tr>
<tr>
<td>$P=3$, $n=6$</td>
<td>31.19</td>
</tr>
<tr>
<td>$P=3$, $n=6$</td>
<td>28.35</td>
</tr>
</tbody>
</table>

Allowances are equal to zero.

Table 2 illustrates that the less the thread pitch the less the load on the most loaded thread and when the allowances are taken into account the load on the thread may be considerably decreased. For Tightener and Post schemes of loading the axial load $F$ is distributed more uniformly.

### 3.2 Threads with rectangular thread profiles

Threads with rectangular thread profiles are widely used in various devices with lead screws. In addition, when the screw and nut are made of different materials or plastics.

The calculation schemes for such thread are shown on Fig. 3. These threads as compared with the triangle thread have no standard dimensions and relationships to the thread pitch, the thread flank angle is equal to zero, and a cross-sectional area of the thread is not changed within the range from top to bottom of the thread, $\xi = 0$, and in Eq.(8) $\alpha/2 = 0$.

Analogously as for the triangular threads by Eqs.(9) and (11) for fundamental thread profile without allowances (Fig. 3a)
\[
\delta_{th} = \frac{6p}{E_1a^3} \left[ \frac{1}{4} (e^4 - b^4) + \frac{1}{3} (e^3 - b^3) (a \tan \rho - b) \right] - \frac{1}{2} \left( e^2 - b^2 \right) ab \tan \rho
\]

(18)

\[
\delta_{th} = \frac{1.2p(1 + \mu_1)}{E_1a} (e^2 - b^2)
\]

(19)

\[
\text{(18) after substituting of the values } a, b, e, \text{ and } c \text{ by } a_1 = a / \cos \psi, b_1 = b / \cos \psi, e_1 = e / \cos \psi, \text{ and } c_1 = c / \cos \psi \text{ where } \psi = (\beta - \gamma)/2 \text{. When the calculations for } \delta_b \text{ and } \delta_{th} \text{ in direction parallel to axis } y_1 \text{ are fulfilled the received results must be multiplied by } \cos \psi \text{ to obtain the displacements of point } B \text{ (Fig.4) along the pitch diameter parallel to axis } y.
\]

For example, the loads \(F(i)\) on the threads for buttress thread are shown in Table 2.

\[
\begin{array}{|c|c|c|c|}
\hline
\text{Number of thread} & \text{Bolt-Nut I} & \text{Bolt-Nut II} & \text{Tightener} & \text{Post} \\
\hline
1 & 4.53/5.98 & 4.17/5.61 & 17.73/17.24 & 17.15/16.72 \\
2 & 5.83/7.25 & 5.52/9.96 & 12.34/13.06 & 12.08/12.81 \\
3 & 8.87/10.14 & 8.59/9.92 & 10.77/11.89 & 10.65/11.78 \\
4 & 14.54/15.28 & 14.36/15.18 & 12.48/13.41 & 12.46/13.43 \\
5 & 24.51/23.79 & 24.65/23.98 & 17.94/17.94 & 18.15/18.16 \\
6 & 41.73/37.56 & 42.71/38.34 & 28.75/26.45 & 29.51/27.10 \\
\hline
\end{array}
\]

Here in nominator are the loads on the threads received for fundamental thread profile and in denominator are the loads on the thread received for thread dimensions with taking into account the allowances.

It is well shown from Table 2 that the load distribution on the threads for buttress threads considerably worse as compared with the metric threads (see Table 1), and for the joint with real thread, the load on the thread (in denominator) is less as compared with the load for joint with fundamental thread profile (in nominator).

So for joint loaded by the scheme of Bolt-Nut I the load on the sixth thread for the real threads 10% less as compared with the load for fundamental threads.

It will be noted that when the threaded joints are loaded by Tightening or Post types the distribution of the axial load on the thread is more uniform as compared with loading by Bolt-Nut I or Bolt-Nut II types.

### 3.4 Round-profiled thread

The thread having a round profile of threads known as a round thread like a trapezoidal thread but with rounded crests and roots. The profiles and basic dimensions of the round-profiled screw thread in accordance with the standards ST SEV 3293-81 and DIN405 are covered. The included angle of the thread \(\alpha = 30^\circ\), the flank angle \(\alpha'/2 = 15^\circ\). These threads have only small straight contact zones and a thread overlap is equal to 0.0835P. Round-profiled threads for nominal diameters from 8 to 200 mm have only four pitches: 2.540, 3.175, 4.233 and 6.350 mm. For these pitches the thread overlaps are: 0.212, 0.265, 0.353, and 0.530 mm, four pitches: 2.540, 3.175, 4.233 and 6.350 mm. For these pitches the thread overlaps are: 0.212, 0.265, 0.353, and 0.530 mm, accordingly. Due to the radial displacement of the contact zones on taking into account of allowances the contact zones are a points and pressure angle of the force acting on the thread changes from \(\alpha'/2\) to \(\alpha'/2 > \alpha'/2\) (Fig.5) on that \(\alpha'/2 > \alpha'/2\).

Pressure angles for different fits are shown in Table 3.

As Table 4 represents for round-profiled thread, it is necessary to take into account not only the real thread dimensions but also the pressure angle for each thread pitch.

Calculation scheme to determine the values of the \(\delta_b\) and \(\delta_{th}\) is shown on Fig.6.
where

\[ \delta_{sh} = \frac{1.2\rho(1 + \mu)}{E_1 \tan \frac{\alpha}{2}} \left( \frac{\cos \frac{\alpha}{2} + \sin \frac{\alpha}{2} \tan \rho}{c} \right) \ln \frac{c}{b} \quad (23) \]

Equation (8) can be as follows

\[ \delta_f = \frac{pd_2}{2E_1 \rho \cos \rho} \sin \left( \frac{\alpha}{2} - \rho \right) \tan \frac{\alpha}{2} \]

\[ \delta_{2f} = \frac{pd_2}{2E_2 \rho \cos \rho} \sin \left( \frac{\alpha}{2} - \rho \right) \tan \frac{\alpha}{2} \]

(24)

and \( \xi = \left( \frac{\mu}{E_1 A_1} + \frac{\mu}{E_2 A_2} \right) \frac{d_2}{2} \tan \frac{\alpha}{2} \).

**Conclusions**

The method of calculation the load distribution for threaded joints with any thread profiles had been considered. It is possible to determine the load distribution on threads practically for any combinations of nut and bolt profiles. The method of determination of deflection of \( \delta_l \) and \( \delta_{sb} \) can be applied to other machine parts. The method of determination of deflection of \( \delta_1 \) and \( \delta_2 \) can be applied to other machine parts. A significant portion of my research work was done with him. Considered methods of calculation of loads on the threads give results more closely approaches to the real conditions.

**Acknowledgments**

I would like to express my deep gratitude to my colleague and supervisor Viktor Strizhak who died 6.01.2015. A significant portion of my research work was done with him. Theoretical analysis of different thread profiles, load distribution between threads and influencing of different parameters on thread reliability was a large portion of scientific interests of Dr. V. Strizhak.

I also express my gratitude to the editorial board for the opportunity to publish this work.

**References**

METHODS OF ASSESSING THE TECHNICAL CONDITION OF AUTOMOTIVE SHOCK ABSORBERS IN THE ROAD VEHICLES OPERATION

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Abstract: Automotive shock absorbers are an important part of the vehicle suspension, which directly affect the driving comfort and maneuverability. Therefore, they direct impact on the safety of their operations. Wrong technical condition of the car shock absorbers can cause extension of braking distance by up to 20%. In actual service station practice, the automotive shock absorbers are diagnosed by several methods. This article presents the results of measurements of automotive shock absorbers on various diagnostic devices. During testing, on the vehicle there were replaced shock absorbers and they were compared with the results of the technical condition before and after the exchange.

Keywords: SHOCK ABSORBERS,

1. Introduction

In this time is in automobile industry given the increasing emphasis on the safety of vehicles, so passive (a stronger constructions, more durable materials, different crumpling zones), as well as active (different systems and sensors actively involving to the driving the vehicle, for example: ABS, ESP, BAS etc.). Among the basic elements of security is necessary to ensure permanent contact wheels of the vehicle on the roadway. This is in charge of overall construction of the chassis vehicle within the frame of fulfill the role the suspension damping — automobile shock absorber. Their task is to muffle the oscillating movement of the suspension of car and keep wheel in constant contact with the roadway.

It is important to check the aspect of the shock absorbers, because even these components will gradually wear out and reduces their ability to dampen the suspension. In the large wearing the damping’s lose effect and already in the larger health the wheel loses contact with the roadway. It is undesirable in terms of security.

There are two options to verify the status of automobile shock absorber. It is a test of the damper in the disassembled and undisassembled condition. In the testing disassembled damper is measure character alone damper, what is however more time consuming. In the testing undisassembled damper is not measure alone damper but road grip – adhesion all axles of the vehicle. It is only an approximate state of shock absorber. We can say, that the testing undisassembled damper verification the overall condition of the axe.

In this report we are going to continue to deal only with the method of diagnosis of the shock absorbers mounted on the vehicle.

2. Methods of diagnostic of automobile shock absorbers

In normal car service practice we meet with the following methods of diagnosis of automobile shock absorbers, which were also used in practical measurements.

1 Principle EUSAMA

Resonance method composed on the principle EUSAMA (EUropean Shock Absorber Manufactures Association) is resonance adhesion method, where is measured the static strength of the wheel to the bearing surface. Its advantage is, that the result is a percentage, which indicates the status of the shock absorber, as well as the overall suspension and there is no need to compare it with the values from the manufacturer. Disadvantage is the inaccurate measurement of lighter vehicles that is due to the fact, the engine with an eccentric transfer the oscillating movement directly on the plate, on which the vehicle rests.

The measurement shall be carried out on both sides of the axle at the same time. First measure the vertical force applied to each side of the axle – the overall axle weight. Then using the electric motors start vibrating platforms for prescribed amplitude with a frequency 25 Hz. After reaching the maximum frequency, the electric motor is turned off and the platform vibrates to complete stop. Measured instant contact force on the platform. The percentage will then be evaluated the smallest force of thrust to force in stationary. [19]

<table>
<thead>
<tr>
<th>Condition of damper</th>
<th>Value of weight in %</th>
</tr>
</thead>
<tbody>
<tr>
<td>excellent</td>
<td>61% and more</td>
</tr>
<tr>
<td>good</td>
<td>41% - 60%</td>
</tr>
<tr>
<td>fair</td>
<td>21% - 40%</td>
</tr>
<tr>
<td>bad</td>
<td>20% and less</td>
</tr>
</tbody>
</table>

2 Principle THETA

Resonance method based on the principle of THETA is the youngest method, launched in 2009. When measuring the unsprung mass holding in oscillating movement constantly and the computer controls the frequency of oscillation by measuring procedure. The result is a dimensionless number THETA, which are determined by the overall condition of the suspension and shock absorbers. The advantage is a measure even lighter vehicles, because oscillation is not transmitted directly, but through a special spring, that is transmitted oscillation to the platform, on which the vehicle rests. The frequency of oscillation is managed by the computer, and as with the principle of EUSAMA the resulting value is nothing compared.

The measurement shall be carried out on both sides of the axle at the same time. With using the electric motors will start vibrating platforms into forced oscillation with frequency 10 Hz. Gradually decreases to almost frequency 0 Hz. Frequency in doing so, passes through a zone resonant frequency of the unsprung masses. The maximum amplitude of the frequency is measured and compared with the time course absorption oscillation.

<table>
<thead>
<tr>
<th>Condition of damper</th>
<th>Value of THETA</th>
</tr>
</thead>
<tbody>
<tr>
<td>bad</td>
<td>0,00 - 0,09</td>
</tr>
<tr>
<td>on the border of life</td>
<td>0,10 - 0,13</td>
</tr>
<tr>
<td>good</td>
<td>0,14 - 0,30</td>
</tr>
</tbody>
</table>

3. Practical measurement for the diagnosis of automobile shock absorbers

This chapter is dedicated to undisassembled testing of shock absorber on several devices using a variety of methods. It will be used for the measurement of personal vehicle VW Golf III 1.9 TDI. On the vehicle have not been altered dampers at least 60 000 km.

Table 1: The percentage evaluation according of the principle EUSAMA

Table 2: Evaluation by dimensionless number THETA
In one of the last measurement before changing of the rear shock absorbers on the device working on the principle of EUSAMA, where it was carried out by measurement with overinflated tire and underinflated tire, spilted oil filling of the left rear shock absorber. The measurements have been completed and have been achieved more interesting results. For these purposes, there were used three different diagnostic equipment to enhance the objectivity of measurements.

<table>
<thead>
<tr>
<th>Device</th>
<th>Principle</th>
<th>Method of measurement</th>
</tr>
</thead>
<tbody>
<tr>
<td>MAHA MSD 3000</td>
<td>THETA</td>
<td>the entire axle at once</td>
</tr>
<tr>
<td>MAHA SA – 2 – D</td>
<td>EUSAMA</td>
<td>each wheel separately</td>
</tr>
<tr>
<td>BOSCH SDL 260 S 10</td>
<td>EUSAMA</td>
<td>each wheel separately</td>
</tr>
</tbody>
</table>

In the following sections are presented the results that describe above:

a) State of shock absorbers in the ordinary course of vehicle load

The vehicle was subjected to measurements of three diagnostic equipment’s. From the results is evident that the front axle is on average the same results under the principle of EUSAMA and under the principle of THETA.

Rear axle is under principle THETA in 100% condition. According to the principle EUSAMA the rear axle is at the level of about 63%. The results of measurements on devices MAHA and BOSCH the same principle of EUSAMA the difference showed on the right rear damper 13%.

b) State of shock absorber in the overinflated tire or underinflated tire

The overinflated tire and underinflated tire is a relatively common occurrence during operation of the vehicle. This can distort the measured values the positive and negative direction. It was conducted diagnostics of the shock absorbers, where on each axle has increased and then decreased the prescribed tire pressure about 0,5 atmosphere. Afore this test was a leakage of oil filling the left rear shock absorber. The measured values are recorded in the table number 5. The measurement is carried out on the device BOSCH SDL 260 S10 principle of EUSAMA.

<table>
<thead>
<tr>
<th>Device</th>
<th>Left front</th>
<th>Right front</th>
<th>Weight front axle</th>
</tr>
</thead>
<tbody>
<tr>
<td>MAHA MSD 3000</td>
<td>0,23</td>
<td>0,20</td>
<td>729 kg</td>
</tr>
<tr>
<td>MAHA SA – 2 – D</td>
<td>71%</td>
<td>69%</td>
<td>706 kg</td>
</tr>
<tr>
<td>BOSCH SDL 260 S 10</td>
<td>71%</td>
<td>68%</td>
<td>797 kg</td>
</tr>
</tbody>
</table>

According to the results of the overinflated tire about 0,5 atmosphere resulted in a negative. On the front axle dropped the value about 3% - 7%, on the rear axle dropped about 8% - 9%. Underinflated tire showed positive, where values grew on the front axle about 5% - 6% and on the rear axle about 5% - 8%. This phenomenon could affect changing the radial stiffness of tire, so when a larger tire pressure radial stiffness is growing, and thus increases the resonance amplitude of axle. By contrast, less pressure decreases radial stiffness of tire, and thereby reducing the resonance amplitude. Therefore the results were at overinflated tire in the negative direction and at underinflated in the positive direction opposite result when prescribed inflated.

c) State of shock absorber in different load axles

Principle of EUSAMA shows a distorted results in lighter vehicles measurements. When peak to peak a vehicle axle with less weight on the axle load there may be a break in contact with the tires and platform, and it causes the measurement uncertainty. The first testing of all primal shock absorbers we moved weight out of the front axle to the rear axle. After changing the rear shock absorbers we test with moved weights reiterated. The measurement was carried out on the device BOSCH SDL 260 S10 principle of EUSAMA.

<table>
<thead>
<tr>
<th>Device</th>
<th>Left rear</th>
<th>Right rear</th>
<th>Weight rear axle</th>
</tr>
</thead>
<tbody>
<tr>
<td>MAHA MSD 3000</td>
<td>&gt;0,30</td>
<td>&gt;0,30</td>
<td>468 kg</td>
</tr>
<tr>
<td>MAHA SA – 2 – D</td>
<td>63%</td>
<td>70%</td>
<td>445 kg</td>
</tr>
<tr>
<td>BOSCH SDL 260 S 10</td>
<td>60%</td>
<td>57%</td>
<td>446 kg</td>
</tr>
</tbody>
</table>

It is obvious that will relieve the already heavy front axle about 84 kg due to the difference in the negative direction only about 3% - 4%. Rear axle has been overcast about 78 kg, what caused the slightly worn shock absorber difference 5%. When a damper with leaking oil filling it was 16%.
Table 7: Comparison dampers when transferring weight between the axles after the exchanging rear shock absorbers

<table>
<thead>
<tr>
<th></th>
<th>Left front</th>
<th>Right front</th>
<th>Weight front axle</th>
</tr>
</thead>
<tbody>
<tr>
<td>Standard measurement</td>
<td>74%</td>
<td>73%</td>
<td>817 kg</td>
</tr>
<tr>
<td>Weight transfer to the rear axle</td>
<td>72%</td>
<td>69%</td>
<td>743 kg</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th></th>
<th>Left rear</th>
<th>Right rear</th>
<th>Weight rear axle</th>
</tr>
</thead>
<tbody>
<tr>
<td>Standard measurement</td>
<td>73%</td>
<td>65%</td>
<td>516 kg</td>
</tr>
<tr>
<td>Weight transfer to the rear axle</td>
<td>72%</td>
<td>63%</td>
<td>591 kg</td>
</tr>
</tbody>
</table>

The removal weight of the front axle about 74 kg was the difference in a negative direction about 2% - 4%, what is approximately the same as in the previous measurements of the same shock absorbers in the same device. Rear axle has been charged about 75 kg. Surprisingly, this result at this device has moved towards the negative, where to test the shock absorbers at the load caused the deterioration about 1% - 2%.

d) State of shock absorber after exchange on the rear axle

In this test are compared a shock absorbers on a vehicle before and after changing the rear shock absorbers for a new (SACHS), including the imposition of the shock absorbers (KYB). This test should show what kind of improvement occurs after changing the worn rear shock absorbers under the absolutely new shock absorber in the context of the entire axle testing.

Table 8: State of shock absorber before and after exchange on the rear axle

<table>
<thead>
<tr>
<th>Device</th>
<th>Left front</th>
<th>Right front</th>
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</tr>
</thead>
<tbody>
<tr>
<td>MAHA MSD 3000</td>
<td>0,23</td>
<td>0,20</td>
<td>729 kg</td>
</tr>
<tr>
<td>Before exchange</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>MAHA MSD 3000</td>
<td>0,22</td>
<td>0,21</td>
<td>764 kg</td>
</tr>
<tr>
<td>After exchange</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>MAHA SA – 2 – D</td>
<td>71%</td>
<td>69%</td>
<td>706 kg</td>
</tr>
<tr>
<td>Before exchange</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>MAHA SA – 2 – D</td>
<td>72%</td>
<td>67%</td>
<td>715 kg</td>
</tr>
<tr>
<td>After exchange</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>BOSCH SDL 260 S10</td>
<td>71%</td>
<td>68%</td>
<td>797 kg</td>
</tr>
<tr>
<td>Before exchange</td>
<td></td>
<td></td>
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</tr>
<tr>
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<tr>
<td>After exchange</td>
<td></td>
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</tr>
<tr>
<td>Before exchange</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>MAHA MSD 3000</td>
<td>0,22</td>
<td>0,21</td>
<td>457 kg</td>
</tr>
<tr>
<td>After exchange</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>MAHA SA – 2 – D</td>
<td>63%</td>
<td>70%</td>
<td>445 kg</td>
</tr>
<tr>
<td>Before exchange</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>MAHA SA – 2 – D</td>
<td>73%</td>
<td>76%</td>
<td>454 kg</td>
</tr>
<tr>
<td>After exchange</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>BOSCH SDL 260 S10</td>
<td>60%</td>
<td>57%</td>
<td>446 kg</td>
</tr>
<tr>
<td>Before exchange</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>BOSCH SDL 260 S10</td>
<td>73%</td>
<td>65%</td>
<td>516 kg</td>
</tr>
</tbody>
</table>

4. Conclusion

All those measurements have shown, what impact have different factors to the overall result of the diagnosis of automobile shock absorbers. Even small changes in the inflated tire, or loaded the vehicle when measurements can distort the result and affect the assessment of the status of the shock absorber, what can adversely affect the safety of the vehicle not just for his crew, but also for other road users.

References


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VEHICLE DYNAMOMETER USING FOR THE DIAGNOSING OF VEHICLE DRIVE TRAIN STATE

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tomas.skrucany@fpedas.uniza.sk¹

Abstract: The article is about the possibilities to diagnose the technical state of transfer mechanism of road motor vehicles. The diagnostic is done on a performance dynamometer, which measures the resistance of transfer mechanism during the coasting of rolling mass. The resistance makes itself felt like power losses by the movement of vehicle, which affects directly dynamical characteristics and fuel consumption. By the measurement we can quantify its value, what is the sign of the technical state of the mechanism.

Keywords: POWER LOSSES, OUTPUT POWER, DRIVE TRAIN, VEHICLE DYNAMOMETER

1. Introduction

During the vehicle moving, it is engaged by the various types of resistances. In addition to the external physical such as rolling resistance or drag it is also the vehicle's own resistance resulting from its structural arrangement and technical condition. With increasing value of these resistances, the engine output power is also increasing to overcome them. This fact increases the fuel consumption, shrinking the dynamic characteristics of the vehicle and reduces the transport of passenger comfort (noise, vibration). When it is not possible to influence the structural layout of the operational by the user, it is important to keep the various constituent parts of the vehicle not only capable of operation, but in the best condition. This will achieve the desired properties vehicle operation.

In this paper there we diagnose the state of the transmission using a cylindrical test laboratory in the laboratories of the Department of Road and Urban Transport at the University of Žilina.

2. Effectiveness of the drive train

Power and torque obtained from the crankshaft, must be transmitted to the wheels on the driving axle of the vehicle, which they passed on the road in the form of motive power, thanks to which the vehicle is moving. This is the role of the gear system, which provides two ways:

a) power transmission at speeds unchanged
b) power transmission speed changes

For physical law that power can not be changed during its transmission. Thus, the power input (to the whole shifting assembly or in parts) equals that of the output. As previously written, the influence of physical resistance can not be transferred without any loss of performance. So the output power is actually power at the reduced losses. This can be expressed in the form of efficiency gear system.

\[ \eta_T = \frac{\eta_D \eta_S \eta_G}{\eta_i} \]

f.e.: \( \eta_T = \eta_G \cdot \eta_S \cdot \eta_D \)

GB – gear box, SF – shafts, D – differential

The losses are mainly formed:

a) friction in bearings and moving parts,

b) mechanical transfers to a rigid kinematic linkage (teeth),

c) the hydrodynamic transfers,

d) splash of individual components in the oil filling.

The effectiveness results from the shifting assembly of these components, their design. The more parts, the greater the losses and lower efficiency. The same is true of the number of images of gear, shafts, joints, deposit. Thus, more complex shifting assembly, the lower its efficiency (the greater losses).

3. Vehicle dynamometer Maha LPS 2000

We use a performance dynamometer to determine the size of the loss power in the laboratories of the Department of Road and Urban Transport.

It consists of:
- Roller set,
- communication panel screen, keyboard and remote control,
- accessories (box interface for connecting sensors, engine cooling fan, fastening straps, suction device exhaust flue, printer).

Performance dynamometers can be supplied with different sets of rollers depending on the requirements of measurements being made. In general, can be measured trucks, cars and motorcycles.

Communication counter screen, keyboard and remote control used to control themselves from all power stations. Simply by moving the power station menu you can select the desired measurement.

The box interface is an inbuilt barometer and temperature sensor. Use the box interface can connect to the laboratory in a variety of external devices that are required for measurement.
3.1 The principle of the measurement

The dynamometer measures directly force at the periphery of the drive wheel of the car. It is transmitted by rolling the drive wheel on the measuring cylinder placed on the rotor, which is currently reading its speed. The cylinder transfers the power to the lever arm located on the stator. The strength of the arm picks up electromagnetic sensor. On the basis of the force measured on the arm we can determine the strength of the wheel. By the multiplying its speed (velocity) and force we can determine the wheel performance.

![Fig. 1 Measuring principle of the dynamometer](image1)


4. Methodology of the power losses measurement

The procedure is continuous acceleration on the selected gear. The measurement is started after a speed 50km.h-1. At this point, it is necessary to fully depressing the accelerator and the following screen communications console. Once the maximum power operator turns off the clutch and the throttle is released. Now is the deceleration phase, in which the loss power is measured and calculated. After completion of measuring the output appears on the screen graphical representation of the engine according to the selected standard.

After the successful measurement are displayed on three line chart (Figure 6):

- Curve A shows the engine performance calculated in accordance with standards = the corrected power given the current pressure and temperature
- Curve B represents the measured power at the wheels
- Curve C represents the power dissipation
- Curve D represents the torque

Engine power (A) are summed to power on the wheels, (B) and power losses(C).

![Fig. 2 Graphical evaluation of the results](image2)

Measured power loss is not only the loss of the transmission, but also rolling resistance, the driving wheel, rolling friction tires.

Even two identical vehicles may have a different value of power loss, for example due to different tires. But always, the rolling resistance is the largest resistance in the case of failure-free vehicle state. It depends mainly on:

1. wheel size
2. type of the tire
3. actual tire pressure
4. axle load

From these factor, the axle load affects the rolling resistance the most, thus heavier vehicles generally have greater losses than light vehicles. Even when comparing identical vehicles must meet their current weight and its distribution between the axles (persons and cargo on board).

A significant impact on rolling resistance has also the inflation pressure. Before the measurement, therefore, always be tire inflated to the pressure rewritten by the manufacturer and maintain for all measurements. This eliminates variations in measurements. The differences between tire pressures in each measurement can guide the diagnosis of some false failure of the transmission.

4.1 Measurement abilities

In view of the above, it is possible to diagnose the condition of the vehicle as a comparative diagnosis of the same vehicle or two identical vehicles. We have the highest explanatory power measurement results if we have the opportunity to compare them with measurements carried out under fault conditions of a particular vehicle with the same tires and inflation pressure. The difference of the measured values is a fault condition. But there should be observed vehicles parameters affecting rolling resistance, so that their value must be the same in both vehicles.

5. Measurement examples

The following measurements were carried out on the vehicle Citroen Berlingo 2.0 HDi. It is a vehicle with front-wheel drive.

Two fact should be evaluated in the diagnosis of graphical output:

1. The value of teh power loss (the rate to the output engine power)
2. The shape of the power loss curve
Not only value of the power loss points to a fault. Also, the shape of its values, the curve diagnosed condition. The shape of the curve at fault conditions shall be similar to the FIG. 4. Parabolic continuous growth is responsible for rolling resistance. If the curve will at some point fall or reduce the angle of the climb, there is a fault, also climb steep curve or purely linear progress.

![Fig. 4 Power loss curve at failure-free state](image)

**Fig. 4 Power loss curve at failure-free state**

\[ P_e - \text{engine power, } P_W - \text{wheel power, } P_L - \text{power losses} \]

![Fig. 5 Power loss at filre state (stuck brake pad)](image)

**Fig. 5 Power loss at filre state (stuck brake pad)**

6. Conclusion

From the results it is observable that in the diagnosis by dynamometer it is not a detailed diagnostics which help to immediately identify a particular failure. Rather initial diagnosis, which provides insight into the overall state of the examined organs and thus induces a further fault-finding. In most cases, it is still necessary to use other methods, or look for the problem of removing individual parts to diagnose the failure. It is not able to compare the measured values of the power loss with standard table number. It is therefore necessary always to perform a comparative diagnosis of the same vehicle or two identical vehicles and in view of the deviation can be diagnosed whether or not a fault.

7. References


[4] Instructions for operation of the vehicle dynamometer MAHA LPS 2000


EFFECT OF SHEAR FLEXIBILITY IN BUCKLING ANALYSIS OF BEAM STRUCTURES

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Abstract: Paper deals with finite element buckling analysis of shear deformable beam-type structures. Displacements and rotations are allowed to be large but strains are assumed to be small. The corresponding equilibrium equations are formulated in the framework of co-rotational description, using the virtual work principle. Displacements and rotations are allowed to be large while strains are assumed to be small. Linear shape functions are used for the axial displacement, while cubic shape functions are employed for transverse displacements and angle of twist. The algorithm is validated on test examples.

Keywords: FINITE ELEMENT, SHEAR FLEXIBILITY, BUCKLING, BEAM, STABILITY

1. Introduction

Load-carrying structures composed of thin-walled sections are extensively used in engineering practice, both in stand-alone forms and as stiffeners for plate-like or shell-like structures. Unfortunately, such weight-optimised structural components, especially those with open profiles, are commonly weak in torsion and very susceptible to instability or buckling failure [1, 2]. Results of classical Euler-Bernoulli theory for bending and torsion of thin-walled beams do not always agree with experimental results and that is mainly caused by neglecting the shear deformations in cross sectional plane as results of transversal forces and restricted warping [1]. Shear-flexible beam finite element developed in this paper takes into account shear deformation effects during the bending and torsion.

This work presents a one-dimensional shear flexible finite element formulation for non-linear analysis of beam structures. Beam members are supposed to be straight and prismatic. Spatial displacements and rotations are allowed to be large but strains are assumed to be small. Using virtual work principle and assuming isotropic and linear-elastic material a tangent stiffness matrix of a two-node space beam element is developed in local Eulerian coordinate system. This system follows the element chord during deformation so this approach is linear in local system.

Numerical results are obtained for cantilever as well as simply-supported beam. Beam high to span ratios influence on the critical buckling loads are also considered. Numerical results show that the shear effects play very important role on the buckling analysis of beams.

2. Basic consideration

Cross sectional displacements are defined with rigid-body translational components $w_{z}$, $u_{y}$, $v_{x}$, rigid-body rotational components $\phi_{z}$, $\phi_{y}$, $\phi_{x}$ and $\theta$ component defining warping of cross-section.

In used right handed Cartesian coordinate system $(z, x, y)$, axis $z$ coincident with beam axis passing through the centroids O of cross sections. Coordinate axes x and y are cross sectional principal axes of inertia. Cross sectional points are defined with x and y coordinates and warping function $\omega = \omega(x, y)$.

Assuming that displacement and rotations are small in local Eulerian coordinate system follows the displacement field:

$$
\begin{align*}
  \omega(z, x, y) &= w_{z}(z) - y \phi_{y}(z) - x \phi_{x}(z) - \omega(x, y) \theta(z) \\
  u(z, x, y) &= u_{y}(z) - y \phi_{y}(z) \\
  v(z, x, y) &= v_{x}(z) + x \phi_{x}(z)
\end{align*}
$$

Strain tensor can be written as:

$$
\varepsilon_{y} = 0.5 \left( u_{,,y} + u_{,,y} \right)
$$

The stress resultant of cross section consist of following components [3]: axial force $F_{x}$; shearforces $F_{y}, F_{x}$; bending moments $M_{z}, M_{x}$; torsional moment $M_{\theta}$; bimoment $M_{u}$. In equation for torsional moment $T_{w}$ represents St.Venant or uniform torsional moment while $T_{u}$ is the warping or non-uniform torsional moment.

Considering the shear deformation due to $F_{y}$, $F_{x}$ and $T_{u}$ we have:

$$
\begin{align*}
  \phi_{y} &\neq - \frac{dv_{y}}{dz} \Rightarrow \frac{dv_{y}}{dz} + \phi_{y} = \gamma_{y}, \\
  \phi_{x} &\neq - \frac{du_{x}}{dz} \Rightarrow \frac{du_{x}}{dz} - \phi_{x} = \gamma_{x}, \\
  \theta &\neq - \frac{d\phi_{x}}{dz} \Rightarrow \frac{d\phi_{x}}{dz} + \theta = \gamma_{\theta}
\end{align*}
$$

In the plane y-z, according to (Fig. 1), we have:

$$
\begin{align*}
  \frac{dv_{y}}{dz} + \beta_{y} = \frac{dv_{y}}{dz} + \phi_{y} = \gamma_{y} \\
  M_{y} = -E1 \frac{d\beta_{y}}{dz} = E1 \frac{d\phi_{y}}{dz} \\
  F_{y} = \tau_{y} \cdot A_{y} = G \cdot \gamma_{y} \cdot A_{y} = \frac{GA_{y}}{k_{s}} \left( \phi_{y} + \frac{dv_{y}}{dz} \right)
\end{align*}
$$

and the same, for bending in x-y plane follows:
\[
\begin{align*}
\frac{du}{dz} - \beta \frac{du}{dz} - \frac{dv}{dz} - \varphi_y &= \tilde{\tau}_y, \\
M_y &= EI_y \frac{d^2v}{dz^2} + EI_y \frac{d\varphi_y}{dz}, \\
F_y &= \tau_m \cdot A_y + G \cdot \gamma_y \cdot A_y = \frac{GAT}{k_y} \left( \frac{dv}{dz} - \varphi_y \right),
\end{align*}
\]

in equations above \(\tilde{\tau}_y\), \(\tau_m\) are average values of shear deformations, \(\gamma_y\), \(\tau_m\) are average values of shear stresses, \(A_y\), \(A_m\) are shear areas and \(k_y\) are shear coefficients.

On an analogous way for torsion follows:

\[
\frac{d\varphi}{dz} + \theta = \tilde{\tau}_y,
\]

\(M_y = EI_y \frac{d^2\varphi}{dz^2}, \quad T_{ST} = GJ \frac{d\varphi}{dz}, \quad T_y = G\gamma \frac{d\varphi}{dz} + \theta\)

where \(J_y\) is shear area with respect to \(\omega\) and \(k_y\) is shear factor due to restricted warping.

### 3. Finite element formulation

The nodal displacement vector of the e-th beam finite element from (Fig. 2) is:

\[
\{u\}^T = \{w_e, \varphi_{eA}, \varphi_{eB}, \varphi_{eA} B, \varphi_{eB}, \omega_{eA}, \omega_{eB}\}
\]

and appropriate nodal force vector is:

\[
\{F\}^T = \{F_{eA}, M_{eA}, M_{eB}, M_{eA}, M_{eB}, M_{eA}, M_{eB}\}
\]

Considering for finite element in y-z plane:

\[
F_y = \frac{dM_y}{dz}, \quad M_y = EI_y \frac{d^2v}{dz^2}, \quad F_y = \frac{GAT}{k_y} \left( \frac{dv}{dz} - \varphi_y \right) - \frac{dv}{dz} = 0
\]

With: \(z = 0 \to \varphi_y = \varphi_{eA}, \quad z = l \to \varphi_y = \varphi_{eB}\) and substitution \(\zeta = z/l\), follows:

\[
v_e = N, v, \quad \varphi_e = N, \varphi, \quad v^T = \{\varphi_{eA}, \varphi_{eB}\},
\]

where \(N, v\) and \(N, \varphi\) are matrices of interpolation functions:

\[
N_y = \begin{bmatrix} 1 & -1 & 1 & -1 & 1 & -1 & 1 & -1 \\ 1 & -1 & 1 & -1 & 1 & -1 & 1 & -1 \end{bmatrix}, \quad \Omega = \begin{bmatrix} 0 & 1 \\ 1 & 0 \end{bmatrix}
\]

and for torsion:

\[
N_y = \begin{bmatrix} 1 & -1 & 1 & -1 & 1 & -1 & 1 & -1 \\ 1 & -1 & 1 & -1 & 1 & -1 & 1 & -1 \end{bmatrix}, \quad \Omega = \begin{bmatrix} 0 & 1 \\ 1 & 0 \end{bmatrix}
\]

so displacements are:

\[
u_e = N, u, \quad \varphi_e = N, \varphi, \quad u^T = \{\varphi_{eA}, \varphi_{eB}\}
\]

\[
\varphi_e = N, \varphi, \quad \theta = N, \theta, \quad \varphi^T = \{\varphi_{eA}, \varphi_{eB}\}
\]

For axial displacement linear interpolation function is used:

\[
w_e = N, w, \quad w^T = \{w_e\}, \quad N_e = \left[ \begin{array}{c} \zeta \end{array} \right]
\]

Finite element stiffness matrix results from integral (10) in form of \(8 \times 8\) with non-zero components:

\[
k_{ij} = \begin{cases} AE/\ell & \text{for } i,j \in \{1,2\} \\
6EI/12(\ell)^2 & \text{for } i,j \in \{3,4\} \\
6EI/10(\ell)^2 & \text{for } i,j \in \{5,6\} \\
2EI/15(\ell)^2 & \text{for } i,j \in \{7,8\} \\
6EI/30(\ell)^2 & \text{for } i,j \in \{9,10\} \\end{cases}
\]

with:

\[
\Omega = 1 + 12\psi; \quad \psi_y = k_1\psi / GAl; \quad \Omega = 1 + 12\psi; \quad \psi_y = k_1\psi / GAl^2
\]
Element force vector transformed from local to global coordinate system is:
\[ \mathbf{f}^e = \mathbf{t}_1^e f^e \] (15)
The element global stiffness matrix \( \mathbf{k}^e \) can be obtained as follows:
\[ \mathbf{k}^e_{ij} = \sum_k \left( \mathbf{t}^e_{i,k} \cdot \mathbf{t}^e_{j,k} \right) + \mathbf{t}^e_{i,1} \cdot f^e \] (16)
\[ \mathbf{k}^e = \left[ \begin{array}{cccc}
\mathbf{k}^e_{11} & \cdots & \mathbf{k}^e_{1n} \\
\vdots & \ddots & \vdots \\
\mathbf{k}^e_{nj} & \cdots & \mathbf{k}^e_{nn}
\end{array} \right] \] (17)

In equations above, \( \mathbf{t}_1^e \) is a transformation matrix of first derivations while \( \mathbf{t}_2^e \) is a matrix of second derivations of the local with respect to global displacements. Matrix \( \mathbf{k}^e \) presents geometric stiffness contribution because it contains effects on global forces caused with change in geometry. Evaluations of matrices \( \mathbf{t}_1^e \) and \( \mathbf{t}_2^e \) are explained in [5, 6].

Performing the standard assembly procedure, the overall incremental equilibrium equations can be obtained as:
\[ \mathbf{K}_c \mathbf{U}^e + \mathbf{P} = \mathbf{F} \] (18)
where \( \mathbf{K}_c \) represent the tangential stiffness matrix of a structure, obtained from their counterparts on the element level, while \( \mathbf{U}^e \) and \( \mathbf{P} \) are the incremental displacement vector and the incremental external loads of the structure, respectively.

### 4. Examples

A first example presents a simple supported I-cross section beam. The lengths of beam varieties to be \( L = 100; 200; 400; 800 \) cm. The beam is loaded axially by the compression force \( F \) as it is shown on (Fig. 3).

![Fig. 3 Axially compressed simple beam.](image)

Material moduli are: \( E = 2,1 \cdot 10^7 \) Ncm\(^{-2}\) and \( G = 80,77 \cdot 10^5 \) Ncm\(^{-2}\). Cross sectional properties are: moments of inertia \( I_x = 3830,52 \) cm\(^4\); \( I_y = 3549,72 \) cm\(^4\) and the cross sectional area \( A = 96,25 \) cm\(^2\), \( k_x = 11,67 \), \( k_y = 1,3125 \). To initiate buckling, a small lateral perturbation force \( \Delta F = 10^{-5} F \) is performed at the mid-span point of beam in \( X \)-direction.

![Fig. 4 Load displacement curves for beam length \( L = 100; 200; 400; 800 \) cm](image)

The second example presents an axially loaded cantilever with doubly symmetric cross section, (Fig. 5). The lengths of cantilever is \( L = 200 \) cm while the material moduli are the same as in the previous example: \( E = 2,1 \cdot 10^7 \) Ncm\(^{-2}\) and \( G = 80,77 \cdot 10^5 \) Ncm\(^{-2}\). The cross sectional properties are: moments of inertia \( I_x = I_y = 50 \) cm\(^4\) and the cross sectional area \( A = 20 \) cm\(^2\), \( k_x = k_y \). The perturbation force \( \Delta F = 10^{-4} F \) is performed at point B in \( X \)-direction to initiate buckling. This example shows the influence of shear factor on critical buckling load.

![Fig. 5 Axially loaded cantilever.](image)

On (Fig. 6), the load vs tip deflection curves for different values of shear factor \( k \) are given as well as the Timoshenko analytical results for comparison.

![Fig. 6 Load vs displacement curves for cantilever for different values of shear factor \( k \)](image)

### 4. Concluding remarks

Finite element numerical algorithm based shear flexible theory is developed. The present model is found to be appropriate and efficient in the field of the stability analysis of beam structures. Presented finite element includes both shear flexural deformation and torsional warping. Shear flexible deformations of cross section taken into account during analysis, guarantees results very precise.
and close to theoretical values. Results of computer program comparing with analytic results, guarantee his successful applications.

5. Acknowledgement

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6. References


1. Introduction

Tests of reliability in the field of power transmissions are widely used and include both specific components as assemblies (transmission, drive axle, hub drive ...), as well as individual elements that are critical in terms of lifetime (gears, bearings, shafts, cardan shafts ...).

Given the complexity of the power transmission system on vehicles as a technical system, it is quite normal to test the reliability of their assemblies and elements for a whole range of test installations and different methodologies. These are often simple solutions that were realized with greater or lesser use of parts and vehicle assemblies, but there is also an increasing application of specially designed test benches, with a relatively high degree of universality.

The impact on the total cost of reliability testing comes from several factors. One of the most important concerns the test installation alone. It is therefore understandable the intention to create a test installation in such a manner to provide a wide application for different types of tests.

Basically, there are two methods of reliability investigations: transmission load applied by use of parasitic power or active power. In the first case the transmission is statically loaded with a certain torque, of constant or variable values, and thus the motor that drives the system loaded in such a manner, only overcomes the internal resistance (typically 20-30% of the total circulating power), while the parasitic power circulates through the system determined by the torque and speed. Such installations are usually called test benches with circulating power, or "closed contour" test benches ("4-Square-Test-System", "closed loop"). A general characteristic of the test benches with the circulation of parasitic power is their simplicity, and in particular, the relatively low cost of manufacture.

In the second case, the transmission is driven by a motor and loaded by a braking device situated on its output shaft. Thus the brake reduces total power applied on the transmission (minus the degree of efficiency), that leads to a significant increase of the direct costs of the test. These test benches are called in-line or "open contour".

The installations aimed for reliability testing of the front driving axle are somewhat more complex. In this case, it is necessary to apply on driving half-shafts not only torque but also a motion that simulates the control of front steering wheels.

2. Review of the applied solutions

The complexity of the power transmission system on vehicle requires the application of numerous methods and test installations for testing the reliability of their assemblies and components. These are often simple solutions that were realized with greater or lesser use of vehicle parts and assemblies, but there is also the increasing application of specially designed test benches, of a relatively high degree of universality.

Figure 1 shows a common base configuration of the closed loop test installation applied for testing of coupled gears, reduction gearboxes, etc. Basically, it consists of two reduction gearboxes linked with shafts, while one of them is equipped with the torque input system. The shown test bench T12-U (Tribology Department of ITeE-PiB in Radom) [3] is equipped with a control-measuring system, consisting of sensors (thermocouples, an indicator of the number of revolutions) and controllers.

As already mentioned, this concept of means that the electric motor must overcome only the friction between gears, rolling bearings, and several smaller resistance components (seals, internal friction in oil, ...). The whole structure is very simple and compact.

Figure 2 shows the "closed loop" test bench for transmission testing, which is designed for testing of the gearboxes [2] of the
well-known car manufacturer Škoda (Laboratory at Juliska at CTU in Prague). A special feature of the test bench is that the way the gearbox is installed fully corresponds to its position in the vehicle. Entering load in the contour, which simultaneously examines two transmissions, is performed by a mechanism with the screw wheel.

Figure 3 shows a test bench for testing of the articulated shaft [4] of the firm Blum-Novotest, Willich, Germany, which examines simultaneously four articulated half-shafts in a "closed contour" configuration. Swivel stand allows variable rotation angle for simulating real operating conditions. The test bench measures and controls the following values: speed, torque and angle of rotation; safety parameters are simultaneously monitored, such as leakage of the lubricant, the temperature of the joints, etc. In case of failure detection, test bench stops automatically. Implemented cooling system allows for the maintenance of operating regime in real conditions.

3. Description of the transmission test bench

Test bench of "closed contour" type has indisputable advantages when it comes to the necessary resources for the implementation of the test. In this case, the transmission is statically loaded with the certain torque, of constant or variable value, and the driving motor loads the system only to overcome internal resistance, while the parasitic power circulates through the system determined by the torque and rotational speed. The general characteristic of test benches with the circulation of the parasitic power is their simplicity, and in particular, the relatively low cost of manufacture [1]. It is also of great importance that the costs of tests on this type of test benches are significantly lower.

When selecting a concept of the test bench it is useful to have the possibility of realization of parallel testing of various solutions of elements and components of the transmission, and therefore the selected solution has the ability to simultaneously test at least two sets of the same type (gearbox, differential, shaft ...). Figure 4 shows the scheme of the realized test bench that meets the above criteria and allows testing of complete transmissions of passenger cars and light commercial vehicles.

Input the number of revolutions of the contour is defined by the required testing regime and is realized by setting the appropriate belt drive applied on the electric motor driving shaft. The existing solution is intended for testing the transmissions of high-speed IC engines, but the simple manipulation can easily lead to the application of the other operating regimes.

Torsional load of the closed contour is entered via a screw mechanism integrated in the gear train (Fig. 6) by use of specially designed key. The value of the torque can be read via the built-in sensor (11).

When the braking mechanism is actuated (Fig. 5), which consists of a disc brake steered by a hydro pneumatic system controlled via a programmable electronic control unit, one of the articulated half-shafts on the transmission output stops and in this way activates the differential mechanism in the transmission. Operation of the control unit is monitored by speed gauge (11) on the output shaft of the transmission.

The examination of the articulated drive-shafts is realized by moving the gearbox (item 9, Fig. 4) so as to change the angle of rotation for larger or smaller value, depending on the length of the shaft (eight joints).
• Basic technical characteristics of the device

Number of test objects (transmissions)  2

Rated speed (input of the transmission)  2580 min⁻¹

Maximum torque (output of the transmission)  2000 Nm

High power electric motor (50 kW) provides continuous operating regime in a closed contour, but keeping in mind that driving power accounts for about 20-30% of the power circulating in the contour, it can be expected that the capacity of the drive group meets the requirements for the most of the transmissions of passenger cars and light commercial vehicles.

Coupled gear pairs (item 2,4,7,8,9, Fig. 4) in the gearboxes are designed for multiple higher loads in comparison with the experimental transmission for the sake of reliable operation over an extended period of time.

Measuring equipment on the test bench (Fig. 8) consists of HBM T2 torque sensor, range 0-2000 Nm, with associated HPSC amplifier bridge with the carrier frequency 5kHz (measuring counter, Figure 11), then the speed gauge at the exit of the transmission and thermo-sensor that measures transmission oil temperature (Figs. 9, 10) and are connected to the control desk so as to activate the operation break in case of exceeding the pre-set temperature.

Complete look of the test bench for transmission in operating condition with mounted safety cover can be seen in Fig. 11. If necessary, the measuring system can be extended with the existing data acquisition system based on the AD converter. In order to maintain the temperature of transmission elements within the permissible limits, the system is supplemented with two high capacity fans.

4. Conclusions

Application of the configuration with two gearbox transmissions and a larger number of shafts and cardanshafts gives the possibility of parallel testing of various solutions in the early stages of transmission development, which significantly speeds up the process of finding the optimal solution.

Implementation of the brake mechanism in the contour allows testing of differential mechanisms by changing regimes without stopping the test bench. Simple procedure enables variation of driveshaft’s angles to a considerable extent, covering most of the real operating regimes.

References


RESEARCH OF THE CHARACTERISTICS OF A STEGANOGRAPHY ALGORITHM BASED ON LSB METHOD OF EMBEDDING INFORMATION IN IMAGES

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Abstract: The article deals with the steganography system which hides text inside images without losing data. The secret message is hidden in the cover image using Least Significant Bit (LSB) algorithm. To evaluate steganography system properties the measures like Signal-to-Noise Ratio (SNR), Peak Signal-to-Noise Ratio (PSNR), Mean Squared Error (MSE) and Structural Similarity Index for measuring (SSIM) are used. Experimental results show the advantages of the described steganography system.

Keywords: STEGANOGRAPHY, LSB, SNR, PSNR, MSE

1. Introduction

Modern world communication between companies and consumers needs a more protected and secure exchange of confidential information via Internet. One way to achieve this is, through the use of stenography. It ensures secure data transfer, due to the very fact of hiding the implementation of secret communication. Two steganographic categories are defined in [1]. The first, called Fragile includes embedding information in a file that will be destroyed if it is changed. The other type of steganography is considered to be Robust. Its purpose is to embed information in a file that cannot be easily destroyed. Steganography algorithms can also be classified according to other criteria. According to the object in which the secret message will be embedded, steganography can be achieved through images, audio and video files, text or web protocols.

To be able to select a steganography algorithm for a particular application, it is necessary to clearly distinguish between the pros and cons of each one of them. A major factor is the identification of specific indicators by which to evaluate algorithms. The criteria which one steganography algorithm must meet are invisibility, amount of embedded data, type of files, sustainability of stego-analysis and sustainability of modification.

All criteria of a method cannot be optimized simultaneously. For example, if you need to embed a large amount of data, the image quality will deteriorate and/or a compromise on any of the properties of the method will have to be done. Adding extra property to the method of hiding data (such as detecting illegally altered areas in the image) reduces the maximum amount of embedded data and/or reduces the quality of the image. When evaluating methods for hiding data, these compromises should be taken into account.

2. Basic characteristics of steganography algorithms evaluation

When comparing two images four major statistical properties which describe the degree of similarity between the images are calculated: MSE (Mean Squared Error), PSNR (Peak Signal-to-Noise Ratio), SSIM (Structural Similarity Index for Measuring), and images entropy.

The calculation of the MSE (Mean Squared Error) is a standard statistical approach to objectively measuring the degree of difference between two images. A small value of the MSE means that the average level of difference between them is little. In the case of two identical images, the MSE value is zero. Unlike the MSE, a greater value of PSNR means better image quality. When there is uniformity of two images, PSNR has a value tending towards infinity. The main purpose of all steganography methods is to minimize the value of the MSE, and accordingly maximizing the value of the PSNR.

The characteristics studied are represented by formulas (1) and (2), the PSNR is based on values obtained for the MSE:

$$\text{MSE} = \frac{1}{mn} \sum_{i=0}^{m-1} \sum_{j=0}^{n-1} [I(i, j) - K(i, j)]^2,$$

Where $m$ and $n$ are the width and height of the image; $I(i, j)$ and $K(i, j)$ are relevant pixels with coordinates $(i, j)$ in the original stego-image.

$$\text{PSNR} = 10 \log_{10} \left( \frac{\text{max}^2}{\text{MSE}} \right) = 10 \log_{10} \left( \frac{\text{max}}{\sqrt{\text{MSE}}} \right),$$

where max = 255 for 8 bit images.

The degree of similarity of the images before and after the embedding of the data, measured by the MSE and the PSNR, determines the quality the stego-image [5]. When the similarity between the images studied is small it is assumed that the quality of the stego-image is lower.

Figure 1 is a block diagram visualizing the steps of calculating the basic functions of the images quality. From the figure it becomes clear that they are interrelated and derived from one another.
where $C_1$, $C_2$ and $C_3$ are constants obtained by the equations.

\begin{align}
(3) \quad \ell(x, y) &= \frac{2\mu_x\mu_y + C_1}{\mu_x^2 + \mu_y^2 + C_1} \\
(4) \quad c(x, y) &= \frac{2\sigma_x\sigma_y + C_2}{\sigma_x^2 + \sigma_y^2 + C_2} \\
(5) \quad s(x, y) &= \frac{2\sigma_x\sigma_y + C_3}{\sigma_x^2 + \sigma_y^2 + C_3}
\end{align}

In (6) $L = 255$ for an image with 8 bits/pixel, and $K_1 << 1$ and $K_2 << 1$ are very small constants.

The values obtained in (3), (4) and (5) are combined into a final average value for the SSIM index

\begin{equation}
SSIM(x, y) = [\ell(x, y)]^\alpha \cdot [c(x, y)]^\beta \cdot [s(x, y)]^\gamma,
\end{equation}

Where $\alpha > 0$, $\beta > 0$ and $\gamma > 0$ are parameters which determine the relative importance of the three components in the value of the $SSIM(x, y)$. In the article was accepted equal importance of brightness, contrast and value of structural similarity, i.e. $\alpha = \beta = \gamma = 1$ and $C_3 = C_2/2$.

From the above-mentioned equation (7) follows in (8), which represents the final form of the SSIM index between two images of $x$ and $y$:

\begin{equation}
SSIM(x, y) = \frac{(2\mu_x\mu_y + C_1)(2\sigma_x\sigma_y + C_2)}{\mu_x^2 + \mu_y^2 + C_1(\sigma_x^2 + \sigma_y^2 + C_2)}
\end{equation}

SSIM($x, y$) takes values in the range $[0; 1]$, and when $x = y$, the value of SSIM ($x, y$) = 1.

In the article of [10] is proposed constraints analysis of the MSE and the PSNR and they are compared with the SSIM.

In [9] a special case of (8) is discussed, in which $C_2 = C_3 = 0$. A disadvantage of these parameters is that when the denominator of (8) tends to 0, the resulting measurement becomes unstable. This problem has been successfully solved in [8] by adding two constants $C_1$, $C_2$, for which is calculated by (6) that are $K_1 = 0.01$ and $K_2 = 0.03$.

According to Shannon Entropy $E$ is a measure of uncertainty that exists in a source of information with respect to some variable or event.

3. Steganography algorithm based on the LSB method of embedding information in images

Stego-image 24-bit format file storage using RGB (Red, Green, Blue) color model [3] is a prerequisite for the presence of a large excess of information that can be used for the purpose of steganography. Steganography synthesized algorithm for embedding information in images uses up to three of the last significant bits in each color channel pixels.

Additionally, the opportunity for selecting a color-channel and pixels in which the information is embedded is achieved.

Steganography modification principle [2] is used to implement the functions of embedding and retrieving data in the cover image, where the pre-existing cover images are changed during the process of embedding. An algorithm is used for insertion into the LSB (Last Significant Bit) for hiding information in image [2], [7]. Through the LSB method is obtained embedding of the message bits in the last significant bits of the color components of individual pixels of the image. It is symmetrical, which means that for embedding and extracting a message, identical operations are performed in the same order. The effect of the algorithm is based on the fact that the secret information is stored in the last significant bits of the pixels of an image, without causing visible differences in its view.

Figure 2 is a general block diagram of the algorithm, in which is checked whether there is a data entry or retrieval. Thus is checked what operation will be realized and is proceeded to the incorporation or extraction of the confidential information

![Fig. 2. Block diagram of the software system](image)

The proposed algorithm can work with BMP, PNG and TIFF file formats of images without size limitations. Since it does not have a block for pre-compression, the maximum amount of information that will be embedded in the image is determined depending on the size of the carrier file minus the information in the header file (service information), while also allocating bytes for the generated code of the password.

The amount of the stego-file is the same as that of the carrier file. A stego-key can be typed in the algorithm. The stego-key is used to control the process of embedding confidential information and limit the ability to find and recover the secret message [6]. Rijndael symmetric encryption algorithm is used in the proposed algorithm for protecting the stego-key.

4. Experimental set of images

By implementing a program system for embedding/extracting text messages many tests with different size messages and images have been carried out. The studied algorithm is based on the LSB method applied and tested on BMP image formats. Test results of the qualitative characteristics MSE, SNR, PSNR, SSIM and $E$ are analyzed. Visual analysis of the compared images shows lack of visual differences in visual control. Histogram analysis and the results of the qualitative characteristics are obtained by MATLAB. Table 1 presents the results of the qualitative characteristics of embedded text files in English with a size of 170 to 240 kB and cover digital image parrot.bmp is used.

<table>
<thead>
<tr>
<th>Size</th>
<th>$MSE_{av}$</th>
<th>SNR</th>
<th>PSNR</th>
<th>SSIM</th>
<th>$E$</th>
</tr>
</thead>
<tbody>
<tr>
<td>170B</td>
<td>2,6809e5</td>
<td>79,4718</td>
<td>86,2179</td>
<td>7,6200</td>
<td></td>
</tr>
<tr>
<td>300B</td>
<td>4,471e5</td>
<td>77,2806</td>
<td>84,0267</td>
<td>7,6200</td>
<td></td>
</tr>
<tr>
<td>600B</td>
<td>9,081e5</td>
<td>74,2614</td>
<td>81,0075</td>
<td>7,6200</td>
<td></td>
</tr>
<tr>
<td>10kB</td>
<td>0,0015</td>
<td>61,9987</td>
<td>68,7448</td>
<td>7,6202</td>
<td></td>
</tr>
<tr>
<td>30kB</td>
<td>0,00455</td>
<td>57,2162</td>
<td>63,9623</td>
<td>7,6204</td>
<td></td>
</tr>
<tr>
<td>40kB</td>
<td>0,00485</td>
<td>56,0551</td>
<td>62,8012</td>
<td>7,6204</td>
<td></td>
</tr>
<tr>
<td>50kB</td>
<td>0,0076</td>
<td>55,0116</td>
<td>61,7577</td>
<td>7,6207</td>
<td></td>
</tr>
<tr>
<td>60kB</td>
<td>0,0089</td>
<td>54,2661</td>
<td>61,0122</td>
<td>7,6208</td>
<td></td>
</tr>
<tr>
<td>70kB</td>
<td>0,0125</td>
<td>52,7076</td>
<td>59,7072</td>
<td>7,6209</td>
<td></td>
</tr>
<tr>
<td>80kB</td>
<td>0,0178</td>
<td>51,2512</td>
<td>57,9974</td>
<td>7,6215</td>
<td></td>
</tr>
<tr>
<td>120kB</td>
<td>0,0355</td>
<td>48,2430</td>
<td>54,9891</td>
<td>7,6224</td>
<td></td>
</tr>
<tr>
<td>240kB</td>
<td>0,0355</td>
<td>48,2429</td>
<td>54,9890</td>
<td>7,6224</td>
<td></td>
</tr>
</tbody>
</table>
The difference between the minimum and maximum value of $MSE$ is difficult to establish because the change is only in the thousands. The average $MSE_{av}$ was obtained as the average of the minimum and maximum $MSE$ and is visualized in the graph of Figure 3(b).

With the increasing amount of embedded data, $PSNR$ values decrease and the values of $MSE$ grow. At 10 kB embedded information is observed value of $PSNR$, which [4] proposed as excellent and their value is 43,6396 dB.

Table 2 presents the results for the tested qualitative characteristics in the embedding of 60 kB information. When embedding the information one, two or three last significant bits in the three color components of individual pixels of the parrots.bmp image are used respectively. From the results presented in Table 2, it can be concluded that the basic statistical characteristics of the image are deteriorated by an increase in the number of the used last significant bits of embedded information.

**Table 2.** Values of the studied statistical characteristics of the parrots.bmp image with integrated 60 kB text in English with increasing the number of the used last significant bits from 1 to 3 inclusive

<table>
<thead>
<tr>
<th>Number of last significant bits of embedded information</th>
<th>$MSE_{av}$</th>
<th>SNR</th>
<th>$PSNR$</th>
<th>$SSIM$</th>
<th>$E$</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.00089</td>
<td>54,2661</td>
<td>61,0122</td>
<td>0.998</td>
<td>7,6208</td>
</tr>
<tr>
<td>2</td>
<td>0.0242</td>
<td>50,2316</td>
<td>56,9777</td>
<td>0.998</td>
<td>7,6208</td>
</tr>
<tr>
<td>3</td>
<td>0.0620</td>
<td>45,9034</td>
<td>52,6495</td>
<td>0.998</td>
<td>7,6210</td>
</tr>
</tbody>
</table>

Table 2 corresponds with the graphical results in Figure 4.

Graphical visualization of the results is presented in Figure 5.

**Fig. 3.** Dependence of (a) SNR and $PSNR$ and (b) $MSE_{av}$ for parrots.bmp image for values of 70 to 240 kB

**Fig. 5.** Dependency in embedding of 4 kB in the last significant bit of information in Bulgarian and English with indicators (a) $MSE_{av}$ and (b) SNR, $PSNR$ parrots.bmp image

Table 3 presents the results of the studied qualitative characteristics in embedding of 4 kB information in English and the same amount of information in Bulgarian at the same other input parameters of the software system. Various images, whose dimension is different, are used.

In Figure 6 can be seen histograms of original and stego-image obtained by embedding of 10 kB information at base settings of the steganography algorithm, i.e. successively embedding in the three color components of pixels without using protection by stego-key in a pepper.bmp cover image.

**Table 3.** Embedding of 4 kB text in Bulgarian and English in different images

<table>
<thead>
<tr>
<th>Original image</th>
<th>Type of text</th>
<th>$MSE_{av}$</th>
<th>SNR</th>
<th>$PSNR$</th>
<th>$SSIM$</th>
<th>$E$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Alps</td>
<td>engl.</td>
<td>0.0014</td>
<td>62,265</td>
<td>69,8157</td>
<td>7,5557</td>
<td></td>
</tr>
<tr>
<td>Alps</td>
<td>bg</td>
<td>0.0013</td>
<td>62,4801</td>
<td>70,0309</td>
<td>7,5549</td>
<td></td>
</tr>
<tr>
<td>Paradise</td>
<td>engl.</td>
<td>0.0011</td>
<td>60,1414</td>
<td>69,8151</td>
<td>6,456</td>
<td></td>
</tr>
<tr>
<td>Paradise</td>
<td>bg</td>
<td>0.00098</td>
<td>60,1312</td>
<td>69,8050</td>
<td>6,466</td>
<td></td>
</tr>
<tr>
<td>change</td>
<td>engl.</td>
<td>0.0012</td>
<td>64,1293</td>
<td>69,9402</td>
<td>6,9676</td>
<td></td>
</tr>
<tr>
<td>change</td>
<td>bg</td>
<td>0.0015</td>
<td>83,9895</td>
<td>70,3094</td>
<td>6,9698</td>
<td></td>
</tr>
<tr>
<td>Marbles</td>
<td>engl.</td>
<td>0.0013</td>
<td>66,9616</td>
<td>72,1563</td>
<td>6,9900</td>
<td></td>
</tr>
<tr>
<td>marbles</td>
<td>bg</td>
<td>0.0012</td>
<td>67,6274</td>
<td>72,8221</td>
<td>6,9874</td>
<td></td>
</tr>
<tr>
<td>ice</td>
<td>engl.</td>
<td>0.0012</td>
<td>61,538</td>
<td>69,7777</td>
<td>6,5832</td>
<td></td>
</tr>
<tr>
<td>ice</td>
<td>bg</td>
<td>0.0013</td>
<td>61,5217</td>
<td>69,7613</td>
<td>6,5833</td>
<td></td>
</tr>
<tr>
<td>snow</td>
<td>engl.</td>
<td>0.0012</td>
<td>63,5252</td>
<td>69,7424</td>
<td>7,2032</td>
<td></td>
</tr>
<tr>
<td>snow</td>
<td>bg</td>
<td>0.0011</td>
<td>83,5473</td>
<td>69,7664</td>
<td>7,2032</td>
<td></td>
</tr>
<tr>
<td>tahaa</td>
<td>engl.</td>
<td>0.0013</td>
<td>64,3828</td>
<td>69,7678</td>
<td>7,7233</td>
<td></td>
</tr>
<tr>
<td>tahaa</td>
<td>bg</td>
<td>0.0011</td>
<td>64,4358</td>
<td>69,8212</td>
<td>7,7232</td>
<td></td>
</tr>
</tbody>
</table>

**Fig. 6.** Histogram of original and stego-image, which has 10 kB of embedded information

Difference in the histograms is hardly observed. This result can be attributed to the fact that the embedded message is not particularly large.

**5. Conclusion**

The results from the research can be summarized in the following conclusions about the characteristics of the studied steganography algorithm:

- By increasing the size of the embedding data the statistical characteristics of the images deteriorate,
Although, the visual quality of the images processed with steganography system remains excellent.

- Using more than one of the last significant bits in the bytes representing each pixel increases the ability to embed a larger amount of data.
- The used cryptographic algorithm ensures additional security during transmission and recovery of the stego-message.
- In embedding in the same image of equal length messages in Cyrillic and Latin, the stego-images obtained have approximately 0.033% difference in the values of the parameters examined.
- This regularity is sometimes in favor of images containing text in Bulgarian, as the percentage is almost the same, i.e. less than 1%.
- The modification of the PSNR in embedding of the same amount of information using one and three bits from the last significant bits is about 15%.

6. References:
https://ece.uwaterloo.ca/~z70wang/publications/SPM09.pdf
INFLUENCE OF LOAD SWINGING ON DYNAMIC BEHAVIOR OF L-TYPE PORTAL CRANES DURING FORWARD TRAVELLING

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Section: MECHANICS, DYNAMICS, STRENGTH AND RELIABILITY, THEORY OF MACHINES AND MECHANISMS

Abstract: Portal Cranes are used for load carrying in industrial and transportation sites. They have complex structure with big dimensions and many mechanisms with high security requests. In this work, we will study the influence of load swinging in dynamic behavior of Single girder L- type portal crane, in case of forward motion- travelling with full loading using computer modelling and simulations. Studying the behavior of portal cranes proves to be difficult using physical experimentation and measurement devices. Creating the crane’s computer model and applying simulations is useful method to study dynamic occurrences, which helps explaining the reasons of oscillations, failures and accidents of cranes, and gives conclusions that can be useful for design considerations and safety. The analysis will be concentrated in finding the nature of forces, moments and stresses that acts on crane’s construction and effects the stability, particularly at the start and end of travelling motion. Also, the study will look to find main parameters that contribute on the negative effects of load swinging. For this purpose, we designed “virtual portal crane” using model design and simulation applications. Crane is modeled from standard manufacturer, as a common model of L type portal cranes.

Keywords: PORTAL CRANE, LOAD SWINGING, TRANSLATIONAL MOTION, OSCILLATIONS, MODELING, SIMULATIONS

1. Introduction

Main cycles of work of Portal Cranes are: lifting and lowering the load, Travell of trolley, Travelling of crane- translational movement forward and backwards, and sometimes rotation of lifting mechanism with load. Their working usage is high, sometimes without break during the working times. The work of portal crane while travelling with weight (load) is considered difficult process with high oscillations, high dynamic strains, concerns of stability, particularly when it carries maximum load. This is mainly caused by swinging of load during work of crane. The importance of study is to find the intensity, type and nature of impact of load swinging on entire crane.

Study will be done for the Crane of type Mostovna [2] shown in Fig. 1, which was available for testing in the factory of local company. It is a single girder Gantry Crane L Type, with Overhead Trolley and Electric Hoist system. It lays with its wheels on rails, which are mounted on Basement. (Fig.2). Results of research will be required for some main parts of portal crane. Speed is considered main parameter that influences swinging of load [2]. In order to prove this, study will be done for two travel speeds, vmin = 8 m/min and v_max = 18 m/min, to determine the impact of speed in load swinging and dynamic behavior of crane. Results will be represented with graphs and tables and compared.

2. Modeling of portal crane and simulations

Model of Crane with its main parts is created with software [3]. Results will be achieved using Numerical methods (Kutta-Merson) and Finite Elements Method (FEM), supported by software in order to achieve best results. Working load has prismatic form with dimensions (1.5*1.5*1.5 m), with mass Q = 5000 kg connected on 4 carrying cables. Carrying cables are connected with the Hook and above with pulley system that connects to Drum with 2 lifting cables. (Fig.2). The load height from basement is 1.5 m. It is positioned on the center of crane’s work space. We consider that best results will be achieved if the study is done with max carrying load Q = 5000 kg, as given by manufacturer (Table 1).

Fig. 1. Portal Crane in working environment

Table 1. Technical features of Portal Crane [2]

<table>
<thead>
<tr>
<th>Crane Parameters</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Load Capacity</td>
<td>5000 kg</td>
</tr>
<tr>
<td>Columns height</td>
<td>11 m</td>
</tr>
<tr>
<td>Columns distance between</td>
<td>30 m</td>
</tr>
<tr>
<td>Girders dimensions</td>
<td>2<em>1.5</em>47.6 m</td>
</tr>
<tr>
<td>Travel speed forward</td>
<td>8÷18 m/min</td>
</tr>
<tr>
<td>Travel Length</td>
<td>147 m</td>
</tr>
<tr>
<td>Total Weight</td>
<td>510 kN (52000 kg)</td>
</tr>
<tr>
<td>Weight of one column</td>
<td>80 kN</td>
</tr>
<tr>
<td>Weight of Trolley</td>
<td>19.3 kN</td>
</tr>
<tr>
<td>Power of motors, kW</td>
<td>2*7.41 kW</td>
</tr>
<tr>
<td>Crane Material</td>
<td>St.37</td>
</tr>
<tr>
<td>Crane Rails Dimensions</td>
<td>40x30 mm x 30 m</td>
</tr>
</tbody>
</table>

Fig. 2. Model of Portal Crane with main parts
For the proper scenario of travelling process, simulation will start without travelling until time $t = 0.5$ s. This in order to have the static stability of load hanging on cables. After $t = 1$ s, will start the travel of crane with given speed. Travelling will end on time $t = 15$ s, at length of travel $L_{\min} = 2$ m or $L_{\max} = 4.5$ m. For the purpose of analysis of the occurrences after travel stoppage, simulation will continue until time $t = 20$ s. We consider that analysing the occurrences before, during the process and after travel stoppage is the best way to simulate travel motion [4],[5].

3. Results

3.1. Force on lifting cables

This is the force acting on lifting cables - $F_c$, resulting from load hanging and swinging during crane travel. Nature of this force is axial force – tension. There are 2 branches of lifting cables, which lifts and lowers the load. Maximal load in one branch of hanging cables is [1]:

$$F_{c_{\max}} = \frac{Q}{m \cdot \eta_{ho}} = \frac{50}{2 \cdot 0.99} = 25.25 \text{[kN]}$$

(3.1)

$Q = 5100 \text{ kg} = 5100 \cdot 9.81 \text{ N} = 50 \text{ kN}$ (Mass of load+approx.mass of lifting devices):

$\eta_{ho} = 0.99$ - working coefficient of hoist;

$m = 2$ – number of cable branches participating in the lifting/hanging of weight.

Results of simulation are shown in Fig.3 and Fig.4, and in Table 2. Based on Fig.3 and Fig.4, between time $0 \leq t \leq 0.5$ s there is no lifting, and no force on cables. After time $t = 0.5$ s, tension force will increase. On the intervals $0.5 \leq t \leq 15$ s is travel of crane. Based on Fig. 3 and Fig.4, travelling of crane is followed by amplitudes of force-tension in cables at almost entire process from start to end of travel. After travel stop, $t = 15$ s, amplitudes are lower, but frequencies of oscillations remain high, up to $\nu = 15$ Hz. After $t = 15$ s oscillations of cables continue for long time due to load swinging. Conclusion is that cables are heavily loaded with oscillations that result in high amplitudes and high number of frequencies. Maximum value of force is achieved after travel stoppage $t = 10.3$ s.

![Fig. 3. Tension force on lifting cable – speed $v_{\max} = 18$ m/min](image)

On Table 2 are given results and comparison of force in hanging-lifting cables. It can be concluded that difference in speed gives difference in all parameters. It is important to emphasize the Average amplitude of force which is 247% higher for travel with $v_{\max} = 18$ m/min compared to $v_{\min} = 8$ m/min.

![Fig. 4. Tension force on lifting cable – speed $v_{\min} = 8$ m/min](image)

Table 2. Results of tension force on lifting cables

<table>
<thead>
<tr>
<th>Parameters of cables</th>
<th>speed $v_{\min} = 8$ m/min</th>
<th>speed $v_{\max} = 18$ m/min</th>
<th>Difference in %</th>
</tr>
</thead>
<tbody>
<tr>
<td>Average tension Force</td>
<td>$\approx 25000$ N</td>
<td>$\approx 25000$ N</td>
<td>0</td>
</tr>
<tr>
<td>Max tension force</td>
<td>$34900$ N</td>
<td>$36400$ N</td>
<td>4.2%</td>
</tr>
<tr>
<td>Time and height of occurrence</td>
<td>$t = 15.1$ s</td>
<td>$t = 15.3$ s</td>
<td>+4.2%</td>
</tr>
<tr>
<td>Dynamic coefficient: $\psi = F_{c_{\max}} / F_{c_{\text{ave}}}$</td>
<td>1.39</td>
<td>1.45</td>
<td>+4.2%</td>
</tr>
<tr>
<td>Max amplitude of tension force, (Peak-to-peak)</td>
<td>$F_{c_{\max}} = 34900$ N</td>
<td>$F_{c_{\max}} = 36400$ N</td>
<td>+112.6%</td>
</tr>
<tr>
<td>Time and height of occurrence</td>
<td>$\lambda_{F} = 17200$ N</td>
<td>$\lambda_{F} = 36400$ N</td>
<td>+4.2%</td>
</tr>
<tr>
<td>$t = 15 \pm 15.1$ s</td>
<td>$t = 15 \pm 15.3$ s</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Average amplitude of tension force</td>
<td>$\lambda_{\text{ave}} = 3400$ N</td>
<td>$\lambda = 8400$ N</td>
<td>+147%</td>
</tr>
<tr>
<td>Frequency of oscillations of Force (Average)</td>
<td>15 Hz</td>
<td>13 Hz</td>
<td>-13.33%</td>
</tr>
</tbody>
</table>

3.2. Resultant force in the connection of girder and column

Crane main parts consists of one girder and two columns. Forces in the connections between these parts are important for determining dynamic effects on crane. Parameter for studying is resultant force in constraints of connection. Results are shown in Fig.5 for right girder-column connection, and Fig.6 for left girder-column connection (as seen in Fig.2). On Tab.3 are shown results of main parameters and comparison for $v_{\min} = 18$ m/min and $v_{\max} = 8$ m/min.

![Fig.5. Resultant force in right constraint - $v_{\max} = 18$ m/min](image)
Based on Graphs of resultant force in constraints, for both cases of travel speed, it can be concluded that for minimum speed $v_{\text{min}} = 8$ m/min, constraints are far less strained than for maximum speed. This can be concluded by results on Table.3, particularly for parameter of Average amplitude. But, in both cases, maximum value of resultant force and Dynamic coefficient don't change much. Main resulting parameters and difference in % are shown in Table.3.

Table 3. Results of parameters of left constraint, based on Fig.6 and Fig.7

<table>
<thead>
<tr>
<th>Parameters of constraints resultant force – Left Constraint</th>
<th>speed $v_{\text{min}} = 8$ m/min</th>
<th>speed $v_{\text{max}} = 18$ m/min</th>
<th>Difference in %</th>
</tr>
</thead>
<tbody>
<tr>
<td>Average resultant Force</td>
<td>$\approx 230000$ N</td>
<td>$\approx 230000$ N</td>
<td></td>
</tr>
<tr>
<td>Max resultant force</td>
<td>$\approx 375000$ N ($t = 15.2s$)</td>
<td>$\approx 449100$ N ($t = 15.2s$)</td>
<td>$\approx +19.7%$</td>
</tr>
<tr>
<td>Dynamic coeff: $\psi = F_{\text{max}} / F_{\text{avg}}$</td>
<td>1.63</td>
<td>1.95</td>
<td>$\approx + 19.7%$</td>
</tr>
<tr>
<td>Max amplitude of resultant force, (Peak-to-peak) Time of occurrence</td>
<td>$F_{\text{max}} = 290000$ N, $\lambda_{\text{Fc}} = 178000$ N, $t = 1.2$ s</td>
<td>$F_{\text{max}} = 370000$ N, $\lambda_{\text{Fc}} = 326300$ N, $t = 9.3$ s</td>
<td>$+53.19%$</td>
</tr>
<tr>
<td>Average amplitude $\lambda_{\text{avg}}$</td>
<td>$\approx 30000$ N</td>
<td>$\approx 131000$ N</td>
<td>$+336%$</td>
</tr>
<tr>
<td>Frequency of oscillations (Average)</td>
<td>12 Hz</td>
<td>/</td>
<td></td>
</tr>
</tbody>
</table>

3.3. Stresses on girder

Stresses (von Mises) are gained using Finite Elements Methods (FEM) by meshing parts of Portal Crane. Girders are important part of cranes for dynamic analysis while they carry most of loading and strains. Results of stress are shown graphically in Fig.9 and Fig.10. Comparison of results is given in Table.4.
3.4. Swinging and oscillations of workload

Pulley system with cables and working load is lifting/carrying system that passes oscillations and swinging to the metal construction of crane. On fig.12 and fig.13 are given oscillations and swinging of workload for both speeds. They are measured in degrees of swinging towards local coordinate system (º) (fig.11). Oscillations between time 0<t< 15 s are almost the same for both speeds. After travel stop t ≥ 15 s, oscillations and swinging shows higher change (Table 5). Max amplitude of oscillations increases due to the swinging of load and pulley.

![Fig.11. Workload Load and pulley with Local coordinate systems](image)

![Fig.12. Swinging and oscillations of workload around Y axes and max amplitude λdeg – for Vmax = 18 m/min](image)

![Fig.13. Swinging and oscillations of workload around Y axes and amplitude λdeg – for Vmin = 8 m/min](image)

On Table 5 are given results and comparison for swinging and oscillations of workload.

<table>
<thead>
<tr>
<th>Results of Oscillations and swinging of workload</th>
<th>speed v&lt;sub&gt;min&lt;/sub&gt; = 8 m/min</th>
<th>speed v&lt;sub&gt;max&lt;/sub&gt; = 18 m/min</th>
<th>Difference in %</th>
</tr>
</thead>
<tbody>
<tr>
<td>Max Amplitude of swinging λ&lt;sub&gt;deg&lt;/sub&gt; (º) t&lt; 15 s</td>
<td>1.4</td>
<td>1.6</td>
<td>+14.2%</td>
</tr>
<tr>
<td>Max Amplitude of swinging λ&lt;sub&gt;deg&lt;/sub&gt; (º) t ≥ 15 s</td>
<td>3</td>
<td>3.7</td>
<td>+ 23.3%</td>
</tr>
<tr>
<td>Frequency of oscillations</td>
<td>1.4 Hz</td>
<td>1.4 Hz</td>
<td>≈ 0</td>
</tr>
</tbody>
</table>

4. Conclusions

Main issues of crane travelling are intensive oscillations with high frequency and big amplitudes, and mostly with irregular occurrence. Most complex work periods are motion start and stop, which gives maximum values for all parameters, and these are the periods when load swinging has highest influence. Main parameter of influence is speed of motion that affects load swinging and other dynamic occurrences. Speed must remain in optimal value, as lower as possible to minimize negative effects of load swinging, strain on parts of crane and safety considerations. Oscillations in cranes are difficult to measure with instruments, and they can cause parts failure, materials fatigue and stability problems. They are mainly induced by load and pulley swinging that induces forces in cables, and further forces, moments and stress in other parts.

5. References

[1] Bajraktari, Musli, Mjetet Transportuese, Fakulteti Teknik, Prishtinë, 1986