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Abstract: A bulldozer, like many building, road, agricultural and other machines and mechanisms has a working unit supporting wheels, tracks or skis. All of them have a late feedback in kinematics scheme. In contrast other the bulldozer in kinematics chart has a late feedback with amplification factor of \( k > 1 \). Thus insignificant change of position of supporting units or shovel, cause a considerable change in the depth of treatment. Number of research works revealed that this transitions period described by an arc, and a circumference.

KEYWORDS: MECHANISM, BULLDOZER, LATE FEEDBACK, TRANSITIONS PROCESS, WORKING UNIT, TRAJECTORY OF KNIFE-BLADE.

Introduction

In the work process operators (bulldozer drivers) very often, sometimes every few seconds, should adjust the shovels (working unit). According to results of the research conducted by us, bulldozers are machines and mechanisms with late feedback and with a coefficient greater than 1. Such mechanisms and machines have closed loops of transfer effects in their kinematic schemes: vertical deflection bearings located behind a working unit is transmitted with a coefficient of more than one back through the frame to the working unit. The transition of effects on the chain bearing-frame-working unit forms a direct bond, and from the working unit through the surface formed by them again to bearing-back. The transition process of bulldozer at a single push is given below. Single push (as a perturbation) due to feedback with more than one coefficient causes further expansion of the roughness formed by vehicle shovel. To eliminate the expansion of roughness the operator intervenes in the course of the transition process, by positive change in the position of shovels. Interference frequency is determined by speed, perturbation size and surface properties of speed and bulldozer bearings.

2. The transition process of the bulldozer as a tool with delayed feedback

Among the reclamation, road construction and agricultural machinery, tools and separate mechanisms there are those in which different supports (wheels, compactors, skis, heels, etc.) are located in the working units and move on surfaces generated by these working units during of technological processes performance. Such machines are bulldozers, which have wheel 1 (Figure 1a) or crawler 1 (Figure 1, b) moving the surface formed by the working units - with blade 2.

These bulldozers as machines and mechanisms, have in their kinematic schemes closed contours transition effects Yaw bearings - 1, located behind the working unit - 2 is transmitted through the frame of bulldozer 3 to working unit 2 (dump), and from the latter formed by them through movement to the support surface, Fig. 2.

Figure 2. Scheme of direct 1 and feedback 2 of bulldozer.

There are direct 1 and feedback 2 connections in these specified machines. Feedback 2 of the bulldozer is delayed. We will assume the transition of effects in the bearing-frame-working unit chain as a direct connection, and from the working unit through the surface formed by them again to support – feedback. Deviations of bearings located behind the working unit are caused by a shift of the latter, but in relation to them are occurred with delay time, which depends on the distance from the working unit to bearing and travel speed of the machine. Of course, bulldozer bearings fluctuations are not literal repetition of blade vibrations, since not all operating conditions can be identified offset edge of the working unit and profile surface formed by it. Surface of bearings movement is not absolutely rigid, also like bearings, for example pneumatic-tire wheels, and, finally, by virtue of the geometry of the bearings themselves, which are of great structural diversity and non-ideal cams of irregularities of its profile movement.

It is necessary to enter certain simplifications. In this regard, it is assumed that the entire working period of machine movement the working unit stays on the treated layer of soil. The blade tip (of working unit) is acute. Profile of to be formed surface is either the trajectory of the blade edge (of working unit) or functionally connected with it. Surface motion bearings (wheels or crawlers) and created by a working unit, is tough, and machine bearings have a view of either rigid struts or elastic suspension (springs) with linear characteristics, perfectly gauging the surface profile of its movement. In accordance with widespread assumptions in the theory of transport machines and such assumptions are introduced for the possibility of developing a theory of fluctuations in the elastic (pneumatic) wheels [1,2,3].

Bulldozer models can be divided into two classes, the difference between them will be considered by the example of flat models, shown in Figure 3. In the first case we can neglect the deformations of bearings and surfaces of their movements and consider that bearings have rigid props, perfectly gauging rigid profiles of their movement, Fig.3a. In this case the position of the machine as a planar rigid body at any time is determined by the position of its two contact points B and C of bearings with surfaces (tracks) of its movement. Consequently, in this case, movement of the machine is studied purely in kinematic aspect as copying the points B and C of some profiles and transmission through deviations frame of these points on the edge of the working unit D. Bulldozer movement, particularly point D displacement vertically defined by movement of wheels or
crawlers according to created profile of the cutting edge D. In the case of the spatial model which has multiple bearings, moving along the surface formed by the working unit, the problem is complicated.

Bulldozer models shown in Fig. 3b and 3c have one or two elastic and deformable bearings, and therefore, to describe their movement it is necessary to model the relevant differential equations. The model shown in Fig. 3a and others like this, can be called a kinematic or programmed movement system, and in Fig. 3b and 3c dynamic.

Two ways of tooling feedback can be mentioned by Fig. 3a and 3b compared to Fig. 3v. In the first two cases, we have rigid bearings, located behind the working unit, moving along the absolutely rigid surface formed by them. This surface, limiting the number of degrees of freedom of the machine, as a solid body, is a constraint and has all the properties relevant existing classification. It can be holonomic and nonholonomic (recently, for example, wheel type bearings), scleronomic and rheonomic, unilateral and bilateral. Along with this the movement of bearings surface C in Fig. 3a and 3b also has the property that is formed by the machine itself, its working unit in the movement process. The equation of this surface (or trajectory in the plane case) is not specified in advance and must be based on the analysis of the machine movement itself.

Forming the surface of bearings movement occurs on segments, Fig. 4. In the start position of machine it is necessary to assume some surface between the bearing C and working unit D. We call this initial. Front bearing moves according to a predetermined surface. When moving of the machine bearings C and B move in the initial segment, and working unit D creates a new segment in this period of time, the form of which is determined by the state of the bulldozer movement. This equation must be calculated based on the observed laws of the machine movement, and in particular, its point D. In the following scheme for the plane scheme of the machine shown in Fig. 4, the trajectory of the point D is accepted as derivative trajectory of the points C and B movement. Thus, the trajectory of the points C and B consists of a number of segments consistently forming by the working unit while gauging bearings C and B of each previous one. From the standpoint of mechanics the case shown in Figure 4 can be represented as the plane movement of a rigid body, some points C and B which are assigned along the trajectory of D.

For spatial machine schemes the working unit generates movement surface which is also formed by segments, and by the points C and B of bearings the location on these surfaces is assigned.

A characteristic feature of machines and mechanisms work with delayed feedback is their compliance with the automatic system, using the exception principle in the presence of feedback. In the automatic system, feedbacks are divided into rigid and flexible, positive and negative, major and local.

The feedback action study involves examining the changes of the controlled quantity (in this case the surface profile field formed by a bulldozer) from appearing on the bulldozer shovel controlling the exposure. With the help of tachogenerator TG the output of the system is connected to its input. Thus, the feedback is a relationship, in which the information about the managed object state (surface profile of field) is transmitted from the output of the system to its entrance.

If the sign of the feedback action being supplied to the input of the system coincides with the sign of the reference action, so it is called positive. Otherwise, it is called negative feedback. In our case of the bulldozer work, the sign of the feedback action supplied to the input of the system coincides with the sign of the reference action, so it is called positive.

When transmitted feedback action depends only on the output quantity and does not depend on time, the constraint is considered rigid. Practically the rigid feedback acts as steady and transient conditions. Feedback that affects the operation of the system only in a transient condition is called flexible. Such constraints react to increment actions applied to their input. And those that respond to derivative actions are called differentiating, to integrals from actions are called integrating flexible feedbacks.

If feedback connects the output of the system with its input, it is called major. The remaining feedbacks are considered local. They serve to improve the adjustment features of individual elements or their groups, connecting the element or group of elements output to the corresponding input. They are also called correcting. Local constraints like a major can be flexible or rigid.

Transition process of the bulldozer will be made on the assumption that the model (Figure 5) is delay less, the deviation quantity of suspension points O1 and bearing are small compared with the length of the OS frame $OO_1 \leq CC_1$ velocity $v = \text{const}$ and bearing $OO_1$ has a rigid type of the prop.

Let us consider the model behavior (Figure 5) when the bearing O hits on a step with a deviation point D height of $h_0$ of the working unit on the quantity of $h_1$ as a single push. Because of the rotation model with respect to the bearing point C the working unit forms a first step. Changes in the depth of processing with height of $\Delta Y_1$
\[ \Delta Y_1 = h_0 \cdot k \]

where \( k = \frac{l}{L} \).

When copying the first stage by bearing \( C \) the working unit forms the next step with height of \( \Delta Y_2 \) due to model rotation relative to point \( O_1 \).

\[ \Delta Y_2 = h_1 (1 - k) \]

Figure 6. Transition processes

In the formation of \( n \)-the step the levelled ground change is equal to

\[ Y_n = \Delta Y_1 + \Delta Y_2 + \Delta Y_3 + \Delta Y_4 + \cdots + \Delta Y_n \]  (1)

The transition process calculated by equation (1) is shown in Figure 6.

Movement trajectory behavior (Figure 6) is described by a circular arc, parameters of which depend on the unit step height \( h_0 \), the lengths \( l \) and \( L \), \( k \), etc. This dependence can be determined by examining \( r \) at different values of \( l \) and \( L \), \( k \).

Figure 7. Scheme of determining the radius of circular arc curvature in the transition process of physical models of machines.

According to this experiment data the graph of \( r = r(k) \) (Figure 9) can be made. The resulting curve gives you the opportunity to judge a function \( r(k) \). However, the constant coefficients, which are included in this function, remain unknown. The method of least squares allows to determine them.

Radius \( r \) of the circular arc is defined by

\[ r = \frac{h_0^2 l^2 + l^2 (L + L) + h_0^4 + 2l(h_0 - Ll + h_0L)}{4h_0^2} \]

The analysis is valid for bearing in the form of a rigid prop and instant copying of vertical front steps. In real conditions various deviations are possible in the course of transition processes due to the influence of various factors.

Figure 8 shows that when the unit step \( h_0 \) the theoretical transition tool is described by a circle radius \( r \). Therefore, with little \( h_0 \), through a small period of time, bulldozers are fully lowered or deepened. Therefore, operators very often steer the position of the working unit - bulldozer shovel.

We have determined the quantity of \( r \) (Figure 8) at different values of the coefficient \( k \), from 1.05 to 1.25. Circle radius \( r \) range of the bulldozers transition process is 50-250m.

Results and discussion

The conducted studies of the operators (bulldozer drivers) work showed that in the process very often, sometimes every few seconds, they should adjust the shovels (working unit).

Transition processes of plane and spatial models machines and mechanisms with late feedback and with a coefficient greater than 1 under the influence of unit step of the bearing surface relief have been studied.

Research results have shown that bulldozers are machines and mechanisms with late feedback and with a coefficient greater than 1. For example, vertical deflection of bearings located behind the working unit is transmitted with a coefficient of more than one back through the frame to the working unit (shovel), causing expansion of the roughness formed by the machine as a radial arc.

To eliminate the expansion of roughness the operator intervenes in the course of the transmission process, by positive change in the position of shovels. Interference frequency is determined by work quality requirement, speed, primary roughness and surface properties of speed and bulldozer bearings.

The transition process at model coefficient \( k > 1 \) from a single push – \( h_0 \) has the form of a circle with radius \( r \).

Suggested theoretical research methods of machines and mechanisms models allow us to study their properties, parameters and modes.

Conclusion

The foundations of the theory of the machines and mechanisms with late feedback and with a coefficient greater than 1 study have been set, different from the existing approaches to the study of the problem. The application of this theory in calculations enables to prove optimal parameters and real machines and tools modes with late feedback.

We conducted laboratory research of transition processes of machines and mechanisms physical models with late feedback from a single push.
Primary cause of bulldozer operator’s frequent intervention in controlling the shovel position is that transition process of tool increases in geometric progression at a slight push. This process can be described by a radial arc of radius $r$. In the absence of operator’s intervention the bulldozer shovel fully lowers or deepens.

**Bibliography**

THE DESIGN PROCEDURE OF RECUPERATIVE HEAT-EXCHANGER FOR HEATING OIL MOVING IN A PIPELINE

МЕТОДИКА РАСЧЕТА РЕКУПЕРАЦИОННОГО ТЕПЛООБМЕННИКА ДЛЯ НАГРЕВА ДВИЖУЩЕЙ НЕФТИ В ТРУБОПРОВОДЕ

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Abstract: Transportation of oil and oil products through the pipeline, due to their high viscosity and the paraffin content is technically and technologically complex process. At present, for the purpose of improving rheological parameters of oil, one of the most effective methods consists in its heating in heat-exchange installations to a certain temperature. The degree of improvement of rheological parameters of oil depends on its heating temperature and conditions of subsequent cooling. Heat treatment improves the inlet capabilities of pumps and allows improving the oil transportation process through the main pipes and pipelines.

The given paper dwells on the design procedure of the heat-exchange installation “the pipe in pipe”. Such type of heat-exchanger is characterized by a simple design and low hydraulic resistance, in which the hot heat transmitter is ecologically clean water.

The process of oil heating in heat-exchanger is the process of convective heat exchange, the intensity of which depends on a number of parameters, including the oil density and viscosity, which in turn are the values depending on temperature. On that basis, the paper dwells on the design procedure of the heat-exchange installation (“the pipe in pipe”) that should enable us to ensure the optimal heating temperature and calculation of geometrical sizes of heat-exchanger through mathematical way.

KEY WORDS: OIL; HEAT-EXCHANGER; RHEOLOGICAL PARAMETERS; TRANSPORTATION.

1. Introduction

Production and pumping of high-viscosity oil still remain a tough technical and technological challenge. The most popular means of pipeline transport of high-viscosity oil is its pumping by heating.

The thermal treatment of oil is referred to as its thermal treatment intended for improving rheological parameters. The thermal treatment (2) of oil allows reducing its viscosity and lowering the freezing point, that ensures the pump inlet capability and improving the oil transportation process through the main pipes and pipelines.

The degree of improvement of rheological parameters of oil depends on its heating temperature and conditions of subsequent cooling. For the paraffin-base oil, there exists the optimal heating temperature at which there is obtained the highest heat treatment effect. This temperature is always higher than melting temperature of paraffin contained in oil.

Since the different types of oil have the various contents of paraffin, the optimal temperature of thermal treatment is determined experimentally for each type of oil.

2. Preconditions and means for resolving the problem

The given paper dwells on the heating design procedure of the heat-exchange installation “the pipe in pipe” (Fig. 1) for the purpose of further experimental investigation of rheological parameters of heated oil. The installation is comprised of continuously acting heat-exchanger, into which the water heating the heat carrier moves through the inner steel pipe, but the heating up oil moves in counter-flow by annular channel between pipes.

When doing designing of continuously acting heat-exchangers it is possible to face the following tasks:
1. Determination of the heating area F, which ensures the passage of the preset amount of heat from the hot heat carrier to the cold one.
2. Determination of the amount of heat Q, which can be transferred from the hot liquid to the cold one at the known heating area.
3. Determination of the final temperatures of heat carriers at the known values of F and Q.

The main design equations for solution of set problems are the heat-transfer and heat-balance equations (1)

\[ Q = G_1 C_1 (t_1^i - t_1^f) = G_2 C_2 (t_2^f - t_1^i) \]

where, \( G_1, G_2 \) are the flow rates of the hot heat carriers; \( t_1^i, t_1^f \) as well as \( t_2^f, t_1^i \) - are the initial and final temperatures of the hot and cold heat carriers.

In the heat-balance equation, the value G is usually determined by product of W of (where, \( W \) – is the heat carrier’s velocity, \( f \) – is the cross section area of the channel, \( \rho \) – the density), and the heat-balance equation takes the following form:

\[ W_1 f_1 \rho_1 C_1 (t_1^i - t_1^f) = W_2 f_2 \rho_2 C_2 (t_2^f - t_1^i) \]
In the heat-transfer equations, the $\Delta T$ is an average temperature drop determined by the nature of changes in temperatures of working liquids along the heating surface. If the heating and heated liquids move along the heating surface in a similar direction, then such flow of movement is called counter-flow. Actually, an average temperature drop at counter-flow is obtained larger (Fig. 2), and therefore the heat-exchanger itself will be more compact than at counter-flow.

![Fig. 2 Changes in temperatures of heat carriers at counter-flow (a) and (b).](image)

An average logarithmic temperature drop at counter-flow is determined as follows:

$$\Delta t = \frac{(t_1^I - t_1^{II}) - (t_2^I - t_2^{II})}{\ln\left(\frac{(t_1^I - t_2^{II})}{(t_1^{II} - t_2^I)}\right)}$$

The process of oil heating in heat-exchanger is the process of convective heat exchange, and the heat-transfer coefficient is directly influenced by moving modes and physical parameters of liquids: specific heat, density, viscosity and thermal conductivity. The heat-transfer coefficient is equal to

$$K = \frac{1}{\alpha_1 + \frac{d_3}{\lambda_c} + \frac{1}{\alpha_2}}$$

The process of oil heating in heat-exchanger is the process of convective heat exchange, and the heat-transfer coefficient is directly influenced by moving modes and physical parameters of liquids: specific heat, density, viscosity and thermal conductivity. The heat-transfer coefficient is equal to

$$K = \frac{1}{\alpha_1 + \frac{d_3}{\lambda_c} + \frac{1}{\alpha_2}}$$

The values of the heat-transfer coefficients $\alpha_1$ and $\alpha_2$ are determined from the appropriate criteria equations at an average temperature of working liquids.

In solution of the problem, we determine the amount of transferred heat and temperature of heating liquid at the output

$$Q = G_2 c_2 (t_2^{II} - t_1^I). t_1^I = t_1^I - \frac{Q}{G_2 c_1}$$

By arithmetic mean values $t_1 = 0.5(t_1^I + t_1^{II})$, $t_2 = 0.5(t_2^I + t_2^{II})$ we determine the values of physical properties of the heat carriers at these specified temperatures: the densities $\rho_1$, $\rho_2$; the heat-transfer coefficients $\lambda_1$, $\lambda_2$; viscosities $\nu_1$, $\nu_2$; Prandtl numbers $Pr_1$, $Pr_2$.

The movement velocities of the heat carriers are determined as follows:

$$\alpha_1 = N_{u1} \frac{\lambda_1}{d_1}$$

$$\alpha_2 = N_{u2} \frac{\lambda_2}{d_2}$$

where, $d_3$ is an equivalent diameter for the annular channel $d_3 = D - d_2$

By values $K$ and $\Delta t$ the density of the thermal flow is equal to $q = K \cdot \Delta t$, but the heating surface and the number of sections of the heat-exchanger is determined by formulas $F = Q / q$ and $n = F / \pi d_1 l$.

3. Conclusion

Thus and so, the proposed design procedure enables us to define the optimal design sizes of the heat-exchange installation (“the pipe in pipe”) by known heating value and heat-transfer temperature, or by using of this design procedure it is also possible to determine the final temperatures of the heat-transmitters on the basis of known area of heat-exchanger and initial temperatures of temperature and calculation of geometrical sizes of heat-exchanger through mathematical heat-transmitters.

4. Literature

1. AIM AND CONTENT

Authors took up the issue of engine simulation and tuning. Engine taken into consideration is developed since 1983 and nowadays is one of the best engine used in speedway. That makes it hard to improve. Therefore the author decide to use modern computer design technics. In racing, where the few Nm is the difference between winning and losing, every improve is important. Aim of this description is show nowadays possibilities of improving engine parameters by changing characteristic of flow.

2. PRINCIPLE OF ENGINE TUNING

Every engine consists of block and cylinders, each cylinder has a bore and stroke. On this data it is possible to define displacement of the engine, bore to stroke ratio and mean piston speed (Cp). Cp parameter is very high for a racing used engines. Nowadays Moto GP engines has mean piston equal 25m/s.

Bore to stroke ratio: $K_{bs} = \frac{B}{S}$

Engine displacement $V_e = \frac{\pi}{4} \times \frac{B_{mm}^2}{100} \times \frac{S_{mm}}{10}$

Mean piston speed $[\text{m/s}]$ $C_p = 2 \times \frac{S_{mm}}{10^3} \times \frac{N_{rpm}}{60}$

Important factor, especially in racing engines is power and torque. Torque is taken from the engine dynamometers measurements or could be calculated from brake mean effective pressure (BMEP). Power is a value which results from torque and rotational speed of the engine.

Power $[\text{kW}]$: $N = \frac{\pi}{10^3} \times \frac{TORQUE_{Nm} \times N_{rpm}}{60}$

or: $N = \frac{BMEP_{bar} \times 10^5 \times V_e \times N_{rpm}}{10^3 \times 60 \times 2 \text{ revs per cycle}}$

BMEP is proportional to the torque of the engine. Typical BMEP value in MotoGP engine is a 14 bar at 16,100 rpm. IMEP is 18,52 bar, PMEP is 1,26 bar and FMEP is a 3,26 bar. BMEP is rarely higher than 14 bar. For example racing motorcycle engine from 1955 has BMEP equal almost 14 bar. Nowadays MotoGP engines has BMEP also about 14 bar, but the power is taken from the rpm's. Todays that BMEP is achieved in about 17 000 rpm and in old engines it was only 7000 rpm. Mechanical efficiency of that kind of modern engine is a 75,6 %.

Another important factor is delivery ratio (DR). It is talking about how many air is taken by the engine.

$DR = \frac{\rho_{air} \times V_e \times n_{cyl}}{\rho_{ref} \times V_e}$

$\rho_{air}$ - air density
$C_d$ - discharge coefficient
$c$ - particle velocity
$A_t$ - specific time area
$\theta$ - crank angle

It is different than volumetric efficiency because it takes into account the air density which for example varies with altitude. Increase of DR gives increase of BMEP, torque and power. It have to be explained that $A_t$ is a side area of cone in geometry meaning. It is characterized by valve lift, valve diameter, valve and valve seating shape. Flow area vary with crankangle. Specific time area ($A_{st}$) has units of s/m. It is area under the intake (or exhaust) line from tdc to bdc like it is shown on figure 2.1.

Fig. 2.1 Intake specific time area [3]

On this equation it could be assumed that the best engine is engine with high $A_t$ and $C_d$.

Engines are always tuned with some designation. In daily used vehicles it is tuned to high torque on low part of rotational speed and connected with that low fuel consumption and relative low cost. In sport vehicle it is high brake power. Fig.2.2 shows different characteristic of volumetric efficiency in regular passenger vehicle and in racing vehicle. That characteristic is mainly regulated by length of the ducts, cross section area of the ducts, and camshaft shape which has been shown in the next part of description. Fig. 2.3 shows particle velocity in ducts with three ducts size. The goal is to have maximum particle velocity about 0.5 mach number. [3
In racing there are used bell ended ducts which are called bellmouths. There are used to improve discharge coefficient or because of regulation -before area reducer. Optimal bellmouth is a bellmouth which gives minimal discharge coefficient.

On the numerical discharge coefficient (CD) value strongly affects pressure ratio($P_o/P_s$). However, finally the value of that coefficient is a ratio of measured the mass flow rate to the theoretical mass flow rate, or ‘vena contracta’ area to real pipe area.

In [2] bellmouth measurement was done by attaching them to the tank with lower pressure. The obtained results are similar to those which were computed (fig. 2.4). They are similar in both numerical and as a trend way. It is worth to sad that authors noted that pressure ratio in real engine is rarely higher than 1,1.
The investigation shows that CD value is the best for short and fat shaped bellmouth. The entry diameter should be equal to the exit diameter multiplied by 2.13 and corner radius should be equal to 8% of intake diameter. It is important to mentioned that aero profile gives lower results than the elliptical profile.

3. INVESTIGATED ENGINE
Engine under investigation is GM 500. That is engine designed by Giuseppe Marzotto. It was built in 1983 and is still in use. It has 4 valve engine head with camshaft inside. Maximal brake power is about 50kW. Weight of that engine is about 25kg. There are many available parts to this engine. It is possible to assemble many engines with completely different characteristic. There are three main types of that engine characterized by bore and stroke: standard, offset and baby offset. Figure 3.1 shows more detailed information about engine under investigation which was taken from [9] and measured.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
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<td>No. Of cylinders</td>
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<tr>
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<tr>
<td>Displacement</td>
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<tr>
<td>Connecting rod lenght</td>
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</tr>
<tr>
<td>Compression Ratio</td>
<td>17</td>
</tr>
<tr>
<td>Engine head</td>
<td>OHC / 4 valve</td>
</tr>
</tbody>
</table>

![Fig. 3.1 Investigated GM 500 parameters.](image)
4. GEOMETRY

![Fig. 4.1 Flow contours of bellmouth and carburettor.](image)

Taken into consideration part of engine is bellmouth with carburetor. General dimensions important from the flow point of view (interior volume) of assembled bellmouth with carburetor has been measured with caliper (Fig. 4.1).

More detailed measurement of bellmouth has been done on university special equipment. Figure 4.2 shows results.

![Fig. 4.2 Coordinates of area reducer chart](image)

5. FLOW MEASUREMENTS

Measurement device is designated to flow measurement of any ducts, mainly engine cylinder heads, manifolds and other exhaust or intake system elements. Schema of the device shows fig. 5.1. The flow could be enforced by under and over pressure measured by test pressure meter. The pressure could be regulated by flow control knob. The value of flow is read from the flow meter.

The area reducer is mounted and sealed through the adapter. Device has two thermometers to better control of boundary conditions. On flow measurement device was being measured flow through bellmouth. Maximum under pressure was 800mm H₂O because higher values was unsteady. Obtained characteristic of mass flow seems to be correct (fig. 5.2).
Table 5.1 Flow through area reducer measurement results

<table>
<thead>
<tr>
<th>p1/p2</th>
<th>p1 [Pa]</th>
<th>p2 [Pa]</th>
<th>γ</th>
<th>R [J/(kg·K)]</th>
<th>T [K]</th>
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</table>

Fig. 5.2 Measured mass flow through area reducer versus pressure ratio

6. CFD SOFTWARE

CFD stands for computational fluid dynamics. It is part of fluid dynamics science which use to the calculation digital computers. It shows fluid behavior based on conservation of mass, momentum and energy. This method has a lot of advantage. It is quick and cheap and parameters can be easy changed.

Even though this methods are being developed for several decades they are still not 100 % accurate. On final result influent few errors like: discretization error, input data errors, initial and boundary condition error. [4]

Ansys Fluent software allow to model 3D or 2D flow, turbulence, heat transfer and reactions. It is based on final volume method. Ansys postprocessor shows high quality image, videos and charts. [5] In this article fluent is used to model flow through area reducer, which has to complicated shape to Ricardo Wave modelling. Geometry was provided by Autodesk Inventor. [8]

Very helpful in engine improvements is 1D simulation done for example by Ricardo Wave. That is computational fluid dynamic software designed to support engine design and analysis.

It could simulate:
- Engine performance
• Acoustic and noise
• Combustion and emissions
• Thermal phenomena

In this project especially important is performance part. It allows to choose the best intake, combustion and exhaust system configuration.

Computer modelling of engine has a big advantage that it easy allows to show how engine works with many variations of geometry of engine head, inlet, exhaust system and many others details. Fig. 6.1 shows computed and measured power and torque of the engine G50. It is a little similar engine to engine under investigation. G50 is a four stroke, single-cylinder, air-cooled 499ccm motorcycle engine [1].

![Fig. 6.1 Power and torque of the G50 engine [1]](image)

7. SIMULATION

The first step is to model the geometry which was done in Autodesk Inventor software and next modelling the mesh.

Mesh presented on fig. 6.1 was created at Gambit software. It is 882101 nodes mesh with 1383645 elements. Mesh has more element in more important or small parts. In this case more important part is the outer surface of area reducer and volume designated to throttle. Model has additional cylinder volume in front of bellmouth.

Simulation was computed with k-epsilon RNG viscous model with enabled energy equations.

Boundary Conditions are shown in table 6.2. Inlet and outlet are determined as pressure inlet and pressure outlet. In this project have been calculated 5 different pressure ratios by changing pressure at outlet.

Calculation has been stopped at $10^5$ because continuity curve does not show tendency to further decreasing.

Figure 6.2 shows general view of velocity vectors for geometry with pressure ratio 1,2.
As it can be seen on fig. 6.2 and 6.3 throttle volume provide a little restriction. It makes vortex which make flow velocity slower. On fig. 6.4 it is clearly visible how air mass is intake in the pipe and how this shape helps air to flow in.
Table 6.2 and figure 6.5 presents results from discharge coefficient calculations.

**Table 6.2 Calculations results.**

<table>
<thead>
<tr>
<th>p1/p2</th>
<th>m</th>
<th>γ</th>
<th>R</th>
<th>T</th>
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<th>p2</th>
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</table>

Because obtained discharge coefficient is too high it was checked also different viscosity model – k-omega. K-omega result is even worse.

**Fig. 6.5 Computed discharge coefficient characteristic of bellmouth**

**Fig. 6.6 Computed flow characteristic of bellmouth**

Chart tendency is similar to chart in fig. 2.4. Mass flow is 10% higher than mass flow obtained in real measurements (fig. 5.2).

8. SUMMARY

Summing up results obtained in real test and computer model are similar and highly close to the results presents in [2]. Both characteristics of CD, in [2] and computed are increasing. Characteristics of measured mass flow and obtained by computer simulation are also the same. Mass flow values cannot be compared with [2] because of different diameters. Computed mass flow values are about 10% higher than measured. As proper values are assumed values which was measured because decrease them by 10% decrease also CD to proper values with physical sense. Further investigation should be done with cooperation of one dimensional model of engine work.

**BIBLIOGRAPHY**


Abstract: It is evident from the experience of operating the construction machinery (excavators, bulldozers, loaders etc.) that quite often the machinery and its operators are working under severe conditions. When the operators are at risk, it is advisable to deploy remote process control technologies on the work sites. This is what brings research and development of modern remote control systems to the top of the chart in order to improve the productivity of machinery, enhance the safety and quality of the jobs carried out.

KEYWORDS: AGGRESSIVE ENVIROMENT, SAFETY,CONTROL SYSTEM, AVAILABILITY, VISIBILITY

Introduction.

The designers developing modern construction and road-making machinery have to take into account a variety of functionality indices determined by psychophysical capabilities of operators, to ensure efficient and safe operation within the man-machine-environment system.

It is due to the fact that, as long the industry development intensifies and new machinery is conceived, the operator’s activity is becoming increasingly complex and subject to stresses. The functional specifications of construction and road-making machinery do not at all times meet the requirements of the sites where it is operated. This results in deployment of substandard process layouts, which, in turn, leads to impaired efficiency and increased labor consumption [1, 2]. Such production sites feature specific operational conditions, whether natural and climatic or due to the highest level of man-made impact. Ensuring quality and safety of the operational tasks under such conditions becomes a real challenge, and the environment conditions set thresholds for the operator’s work. On great many occasions, when the personnel’s actions prove to be wrong, it is not due to poor skill level (though the there are many problems on this side as well) but to the mismatch between the machinery’s design features and human capabilities [3]. The physical environment of the production site has to correlate with the human performance features, and only then one can expect high productivity from him/her.

A wide range of operational tasks require to limit the presence of operators on the production site [6]. Listed below are examples of construction and road-making machinery operation involving increased adverse effects:

- mining and concentrating mills, mines, ore dressing mills, open pits for commonly occurring minerals, construction material works;
- waste dumps, sanitary landfills for solid household waste and toxic chemicals;
- demolition of buildings and debris handling;
- mitigation of radiation accidents and incidents involving detection of uncontrolled radiation sources.

Various transport, digging, filling, compacting, and crushing machinery is used at the above sites, all subjected to aggressive environment and quickly going out of service.

If we consider a specific machine as a complex unit of equipment, the principal natural, climatic, and environmental factors produce the following impact on it:

- **High temperature**: reduced viscosity and modified structure of diesel fuel, lubricants, pressure and process fluids, impaired cooling of internal combustion engines, accelerated ageing of rubber seals and other insulating materials.
- **Low temperature**: increased viscosity of diesel fuel, congealed lubrication oils and solid greases, frozen condensate in pneumatic systems, reduced toughness of steels, hardened and embrittled rubber seals.
- **Increased humidity**: accelerated corrosion of steel parts, reduced insulation resistance, water intrusion into fuel and process liquids, mold build-ups.
- **Reduced humidity**: thickening lubrication oils, drying out seals, fracturing insulation materials.
- **Sun radiation**: changing coefficient of friction for friction materials, accelerated ageing of polymer coatings.
- **Wind**: drying of materials, increased heat output of machine parts and extra strain on them.
- **Dust**: changing coefficient of friction for friction materials, clogging of ducts and reduction of air flows, impaired cooling and ventilation, build-up on heated surfaces reduces heat exchange, and intensely heated items may become a source of ignition.

- **Aggressive environment**: accelerated deterioration of materials. The following groups of environments is among the most widespread: potent oxidizers (nitric, chromic acids etc.); mineral and organic acids (phosphoric, acetic acids, etc.); alkali; organic compounds (petroleum products, etc.); halogen compounds. Aggressive environments can produce chemical transformation, deterioration, cracking, stiffening, etc.

Therefore, research of machinery operation and operators’ activities within the single man-machine-environment system emphasizes the importance of finding new ways to reduce the risk of human errors and enhanced operational safety of construction and road-making machinery.

**Method.**

Working under complicated operational conditions relies on operation, upkeep, and maintenance costs of machinery. If insufficient capacity, inappropriate or unreliable equipment is selected, early failures may occur which, under urgent work pressure, may prove to be critical.

Special operational conditions for machinery are accounted for at the stage of design and manufacture. Modern equipment is manufactured in various climatic options as regards their fitness for operation in various macroclimatic zones: for cold, moderate, arid, or humid tropical climate. Standard machinery greatly outnumbers the specialized options, which is due to overwhelming proportion of **brown field areas** with moderate climate, as well as to manufacturing industrial facilities, production cost of machinery etc. In this connection, it is necessary to adapt standard machinery to special operational conditions by means of special refit and by changing their modes of operation. Such necessity arises during operation of standard machinery in climatic areas with high temperature fluctuations or when it is required to operate such machinery in a variety of meteorologic conditions.

To adapt process systems to their operational conditions, proven methods are used to refit such systems, thus obtaining high efficiency of standard equipment under special conditions. The solutions improving the efficiency of machinery have to be coordinated against each factor affecting the productivity (purpose, operational environment, operating mode, technical condition, technologies deployed) and the duration of the machinery operation, as well as any possible variations of all these factors. Therefore, the aggregate range of impact for each specific factor builds into the set of positive/adverse factors affecting the productivity of process systems.

Animated graphic modeling of the man-machine-environment system enables to suggest a way to reduce the impact of aggressive environment on such system:

- Develop a set of activities which have to include the selection and setup of specialized equipment;
- Remove the operator from the potential hazard area where the operations are carried out by implementing remote control;
- Comprehensive integration of technology to improve efficiency, safety, and enhance quality control of the processes and remaking them into a single high added value production line.

This is what brings research and development of deploying modern remote control systems in construction and road making machinery to the top of the chart in order to improve the productivity of machinery, enhance the safety and quality control of the jobs carried out. There is a widespread solution for such tasks: a team of equipment enabling two operation modes, direct or remote, depending on the operational conditions.

**Informative Part.**

1) Currently the electronic control systems of construction and road making machinery are monitoring and optimizing the operation of the engine, hydraulics, all sensors and operating controls, and ensure that information is shown on the display. The consistent operation of such electronic control systems is due to digital communication and control features applied. Operators may use the electronic control system to adjust the operating force and receive feedback about the condition of and load on such machinery resulting from interaction with the objects [7]. Reliable feedback is ensured between the operator and equipment, to monitor the reaction force when actuators contact the working surface. The existing level of construction and road making machinery and the capabilities of radio electronic features enable creation of a set of radio devices which can be applied to provide remote control over operation of specialized machinery under a variety of conditions.

The capability to handle the necessary process operations is the key functionality for construction machinery remote control systems. It can only become possible subject to a fail-free control of the actuators of such machinery, which requires a homogeneity and optimization of operation for all units and modules. The construction machinery control system operation can divide into the following tasks:

1) Principal: a set of control features to carry out the machinery operational cycle (its core function).
2) Auxiliary: a set of auxiliary features enabling control between the operational cycles.
3) Visual and spatial control of process operations. A system of cameras, microphones, positioning sensors, and data from the electronic control console enables visualization of parameters and positioning of the machinery and of the working members of its actuators.

The remote control system must be capable to handle the above tasks. At the same time, it must ensure feedback regarding the force impacting each of the actuators.

The coordination between the operations of such machinery can be ensured using the Master-Slave system already widely spread in modern construction equipment [7].

A typical Master-Slave system is a team of coordinated devices consisting of the following systems:
- Master: controls one or several other devices (servo units).
- Slave: is set up to operate under control, ensuring that the operational forces are applied consistently with the gear diagrams of the operational equipment and using the installed feedback system, and features a set of information features to ensure detection of mechanical loads during the operations.

Master-Slave system enables control of motions of working members and sends signals describing such motions of each working member, enabling their realtime positioning [7]. Feedback ensures adequate effect, based on the force applied by the operational equipment to the working surfaces. Such interaction ensures the degree of “sensitivity” between the components of the man-machine-environment system, thus promoting the process quality.

2) Generally speaking, the proposed modern single-purpose remote control systems for construction machinery operate as follows:
- The sensor gathers (detects) data from outside sources and information about the condition of the controlled equipment (taking into consideration the feedback channel data) and generates the control commands based on the inputs and the source (apriori) information.

A sequence of stimuli is generated for the machinery controls, ensuring the consistency between its operational mode and movement trajectory with the operator’s purposes.

- Further on the commands are sent via the control channel to the servo unit. As a result of inherent distortion, the commands received by the servo units may be somewhat different from the transmitted commands.
- The feedback line detects adequate effect, based on the force applied by the operational equipment to the working surfaces, and returns the operational data to the electronic control console.

The necessity to use a wide radio frequency range to ensure a reliable communication channel without cross distortion or
jamming, is a serious drawback of remote control systems. The operational conditions may impose limitations on the radio channel if side electronic noises are present.

**Conclusion.** Operational and design are the two types of requirements applied to the construction machinery remote control systems.

Operational requirements consist in fail-safe and reliable operation of all remote control systems under the given weather and climatic conditions. The importance of this condition is due to the fact that the current development of equipment mostly targets its improved precision and implementation into control systems of high speed computers assuming an increasing amount of the operators' functions. Such control systems are complex and contain many different components. Whereas a failure of any single component may disturb the operation of the entire system, it is, therefore, of utmost importance that all components and the system as a whole should be highly reliable [8].

Design requirements consist in quality of the installed features’ operation. It must have minimal dimensions and weight, resist overloads, and be immune to vibration. These features should be operable under a wide range of temperatures, humidity, and pressure.

Remote control features fitted on the construction machinery will increase its base cost up to 30%. Taking into account the process operations carried out by the machinery and the conditions of such operation, the development of remote control sets for such machinery must be based on the value added. This consideration is viable both to design new machinery and to retrofit the existing equipment. For the latter, to avoid excessive costs, onboard equipment may be installed without any material redesign of the machinery. The consideration of extra costs is overshadowed by the totally different level of safety and comfort offered by the remote control systems. There are modern examples of successful deployment of remote control over machinery. In 2010 Brodrene Gjermundshaug Anlegg AS. was busy reclaiming the territory of a former military firing range on the territory of the actual Dovre National Park (Norway). The hazard consisted in the occurrence of many unexploded shells in the ground. One of the operators, Havard Thoressen, said: “It was quite a strange bit of experience, learning to do my normal job sitting in a steel box miles away from the place I am actually working at. It took me about two weeks to get used to the new way of working. First we had some difficulties to retain control over everything, but now there are no more problems” [9].

**Literature:**
A MATHEMATICAL MODEL OF OIL SPILL DIFFUSION IN NEAR-SHORE ZONE OF GEORGIA

МАТЕМАТИЧЕСКАЯ МОДЕЛЬ РАСПРОСТРАНЕНИЯ НЕФТЯННОГО ПЯТНА В ПРИБРЕЖНОЙ ЗОНЕ ГРУЗИИ

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Abstract: The oil products remain the main contaminant of the Black Sea, as well as of the entire world's oceans. A main amount of oil is transported by tankers and reloaded in port terminals. 12 per cent of overall marine pollution is the share of transportation, oil congestion and accidents losses. At the Caspian-Black Sea-Mediterranean area there are actively used the Black Sea ports of Batumi, Poti and Kulevi.

A huge damage to the nature is doing by oil spills on the surface of the oceans and seas owing to transportation oil by tankers. These spills are associated with loading and unloading operations as well as with tankers accidents. In this case, the oil spreads as a spot on the surface of the sea.

Problems associated with environmental pollution from the oil spill in the sea, made necessity to develop mathematical models that describe the transportation and transformation processes of oil spills. The proposed mathematical model can be used for prediction of the spread of an oil spill during the process of spreading of oil products on the water surface, and it also enables to take into account the reduction of the area of spill due to the action of sea-surface winds and turbulence of the water surface.

KEY WORDS: BLACK SEA, OIL SPILL, MATEMATIKAL MODEL, TRANSFORMATION PROCESSES.

1. Introduction

The oil products remain the main contaminant of the Black Sea, as well as of the entire world's oceans. According to International Energy Agency, oil production is 91.3 million barrels per day, from which 3/5 – is transported by tankers, 2/5 – through pipelines, i.e., a main amount of oil is transported by tankers and reloaded in port terminals; 12 per cent of overall marine pollution is the share of transportation, oil congestion and accidents losses. Experts estimate that by 2020, the world consumption of crude oil will be increased by 1.2-1.5. Besides, the demand for oil in developing countries will be increased by 2.5-2.8, and in advanced countries – by 30-35%.

In the near future, the annual volume of oil transportation in the Black Sea may be increased to 220-250 million ton. Besides, it is expected to transport annually: 50 mln through the ports of Ukraine; about 60 mln – through the ports of Russia; about 30 mln – through the ports of Georgia; about 25 mln – through the ports of Bulgaria; about 35 mln – through the ports of Turkey. Without regard to accidents, only with technological loss at 0.01% of the volume of transported oil products, about 20 thousand ton of oil products may be discharged into the marine environment. During accidents, these losses may be increased tenfold.

At the Caspian-Black Sea-Mediterranean area there are actively used the Black Sea ports of Batumi, Poti and Kulevi. Navigation and the maritime transport objects of Georgia have significant impact on the Black Sea ecosystem.

A huge damage to the nature is doing by oil spills on the surface of the oceans and seas owing to transportation oil by tankers. These spills are associated with loading and unloading operations as well as with tankers accidents. In this case, the oil spreads as a spot on the surface of the sea. Just one liter of oil is enough for forming of spill of almost 1 hectare. Besides, depending on the quantity of oil and the velocity of spreading the spill can be appeared either in the form of “peak” emission or as a continuous ingress of oil within a certain period of time.

2. Preconditions and means of resolving the problem

Problems associated with environmental pollution from the oil spill in the sea, made necessity to develop mathematical models that describe the transportation and transformation processes of oil spills. Such models are used for prediction of the spread of an oil spill and for estimate of its characteristics required for planning and carrying out activities for the liquidation of spills in the events of accidents, as well as for assessment of environmental impact.

The process of spreading the oil spills in the sea is a fairly complex process, which depends on a large number of factors defining as the state of the environment so the properties of oil itself. Thus, the solution of this multifunctional problem requires a comprehensive and integrated approach.

When setting the problem on oil pollution transfer into the sea, it is necessary to adequately describe not only physical-chemical properties of oil itself and the character of the source of pollution, but also such characteristic as the diameter of oil spill.

The realized complex takes account for the following processes that occur with oil as the object under study: 1. Oil spread; 2. Displacement caused by the sea water flow and wind.

For the first process of spilling of liquid lighter than oil over the water surface, it is necessary to emphasize several merging one into another stages, from which the most important for spills less than 2000 m³ is the phase of spreading under the action of forces of the superficial tension of oil so far as the spill remains a single whole.

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The oil spill diameter against the wind direction Ry is determined as follows:

$$
\sigma = \left[ \rho_{Wy} - \rho_o \right]^{a} \rho_o^{1-a}
$$

Here,

- $\rho_{Wy}$ - is the sea water density;
- $\rho_o$ - oil density; 
- $M$ – the volume of the initial spill of oil;
- $t$ – time of spill;
- $a, b, c$ – spreading coefficients of layer ($a=42.5; \ b=1/3; \ c=1/4$)

The oil spill diameter in the direction of wind Rx:

where, $W$ – the wind velocity, m/sec;

$\beta = 1.82; \ d=4/3; \ e=3/4$.

The process of displacement of oil pill under the action of sea current and wind.

It is assumed that the oil products move by means of the following factors:
1. The wave action arisen near the margin of coast of water area.

The process of displacement is described by the expression:

\[ V_I = K_W W_I + K_U U_I + K_S (L) F_I \]

where,
- \( V_I \) - the vector of velocity of the shift of the center of “micro-spill”;
- \( W_I \) - the vector of wind velocity at a height of 10 m from the surface of coast;
- \( U_I \) - the vector of the total wind and flood currents;
- \( F_I \) - the influence vector of the coast and its configuration;
- \( K_W \) - the influence coefficient of wind – in the calculations it is assumed that each spill has its own coefficient of permissible range 2-4%;
- \( K_U \) - the influence coefficient of the sea water current. In calculations it is taken as equal to 100%.
- \( K_S \) - the influence coefficient of the coast by the distance from spillage.

2. The wave action arisen near the coast-line.

The coast-line has a significant influence on “micro-spills”, since with decreasing water depth the waves turn toward the coast. This factor is especially evident when the angle between the coast and vector is under 90°. When the distance to the coast is reduced this influence is more violent. The influence coefficients of the coast are the empirical values and they are determined during the experiments.

As the input data the following information is used:
1. The sea-coast configuration;
2. Meteorological situation for the entire period of modeling;
3. The real or calculated currents;
4. The place and dynamics of spillage of oil products.

The air temperature on the Georgian coastline during the most cold months – in January and February – is an average minimum of 4,3°C, 3,8°C, but some days it can be lowered to -5,3°C, -7,5°C. During the most warm months – in July and August – the mean temperature of air is 22,3 – 23,1°C, the absolute maximum is 37,6 – 41,0°C.

For most of the year, the north-east winds blow in near-shore zone of Georgia, which are characterized by considerable velocity and duration, but in the winter they bring cooling. In the summer, the south winds are not unusual. The violent winds occur mostly in winter and autumn. The currents mostly depend on winds.

The place and dynamics of spillage of oil products.

The process of merging of oil into the “micro-spills” occurs as follows:

- by gravitational viscose forces and by surface tension forces, which move under the action of water and wind current during a certain period of time;
- by the division of spill into individual particles under the action of wind and current (there is applied the Lagrangian method by using of the stochastic nature of the formation of spills by particles).

It turned out that right after the spillage, there begins the spreading and displacement of oil over the water area of reservoir; when reaching the specified thickness of lash, the spill is divided into individual N particles, each of which has a certain mass and henceforward is considered as the «micro-spill». Then for each particle there begins the iteration (repetitive process of calculating its displacement trajectory. This process proceeds for each particle:

a) until it reaches the coast; b) until it goes beyond the boundaries of computational region.

The results of theoretical calculations have shown that the diameters of petrol and diesel fuel spill are not very different from each other. This is explained by their densities, the values of which are almost the same. Similarly, the diameters of the mazut and oil spill are also not different from each other, but their values are by 23% lower than the diameters of petrol and diesel fuel spill.

Fig. 1 shows the change of the oil spill diameter with the dependence on time of spreading. As is seen from the diagrams (Fig. 1), the diameter of spreading of oil spill depends significantly on the wind direction.

According to the results of studies (Fig. 2), there are constructed the diagrams of distribution of mazut both in the water area and in near-shore zone of Georgia.

The movement of the layer of mazut spill in near-shore zone of Georgia and in the water area is determined by the field of wind, the state of the sea surface and physical-mechanical properties of the mazut itself. The use of the method of spill division into elementary particles allows assessing the quantity of distributed oil products within particular territories, as well as determining the trajectory of displacement of these particles.

3. Conclusion

It has been established by the calculations that the diameter of spill increases with increasing quantity of the spill oil. The maximum diameter, which can be reached by oil spill also depends on the quantity of the spilt oil and oil products over the water surface.

The wind direction as well as its rate has more significant influence on the oil spill diameter than the density of oil and oil products.
4. Literature


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BALLAST WATER IN THE BLACK SEA GEORGIAN COASTAL ZONE

Abstract: Today global shipping transports over 90% of the world’s commodities in intercontinental traffic. Within the EU, waterborne traffic accounts for more than 90% of foreign and approximately 40% of domestic trade, transfers around 12 billion tons of ballast water across the planet each year. While ballast water is essential for safe and efficient modern shipping operations, it may pose serious ecological, economic and health threats. Trends anticipate an increasing role for global and local shipping in the future. Apart from harmful effects such as consequences of shipping disasters, shipping activity exerts other negative influences on the environment: e.g. sea pollution through the discharges of oily water and sewage water from vessels, air pollution from exhaust gases emitted from the vessel’s machinery, pollution of water and marine organisms from toxic protective underwater hull coatings (antifouling paints), and one of the most recent water born concerns – the translocation of harmful organisms and pathogens via ballast water and sediments inside ballast water tanks.

Ballast water is absolutely essential to the safe and efficient operation of modern shipping, providing balance and stability to unladen ships. However, it may also pose a serious ecological, economic and health threat for sea nature life. The introduction of invasive marine species into new environments by ships ballast water can be an issue that does not always receive the public exposure that it merits.

Despite the serious degradation that has already occurred in the Black Sea, studies have indicated that concerted action can both restore and protect the environment. But the problem clearly requires a multi-lateral approach. The introduction of invasive marine species into new environments by ships’ ballast water attached to ships’ hulls and via other vectors has been identified as one of the four greatest threats to the world’s oceans. The other three are land-based sources of marine pollution, over exploitation of living marine resources and physical alteration/ destruction of marine habitat. Quantity of ballast water is change depend of ship’s type. The release of ballast water may introduce non-native organisms into the port of discharge.

Black Sea geography contributes to the manifestation of additional environmental risks: these sea or do not have access or have a very limited relationship with the oceans. Pollution from land-based sources is one of the main causes of environmental degradation of rivers and, as a consequence, seas and coastal areas. Pollution from ships and other activities in the seas are another factor contributing to the deterioration of the environmental status of marine waters and coasts. Much of Georgia’s coastal zone is subject to significant anthropogenic pressures, that could be reason of causing the pollution of marine environment.

KEY WORDS: COASTAL ZONE, MARITIME TRANSPORT, BALLAST WATER MANAGEMENT, MARINE POLLUTION.

1. Introduction

Today global shipping transports over 90% of the world’s commodities in intercontinental traffic. Within the EU, waterborne traffic accounts for more than 90% of foreign and approximately 40% of domestic trade, transfers around 12 billion tons of ballast water across the planet each year. While ballast water is essential for safe and efficient modern shipping operations, it may pose serious ecological, economic and health threat. Trends anticipate an increasing role for global and local shipping in the future. What is ballast water?

Figure 1.
Ballast water provides stability and manoeuvrability to a ship. Usually ballast water is pumped into ballast tanks when a ship has delivered cargo to a port and is departing with less or no cargo. Large ships can carry millions of gallons of ballast water. Ballast water discharged by ships can have a negative impact on the marine environment. Ships use a huge amount of ballast water, which is often taken on in the coastal waters in one region after ships discharge wastewater or unload cargo, and discharged at the next port of call, wherever more cargo is loaded. Ballast water discharge typically contains a variety of biological materials, including plants, animals, viruses, and bacteria. These materials often include non-native, nuisance, exotic species that can cause extensive ecological and economic damage to aquatic ecosystems, along with serious human health issues including death.

The ballast water inside a ship can be seen as an onboard aquarium full of microscopic life forms. That’s because small organisms living in the sea water are pumped into ballast tanks along with the water. Moreover, coastal sediments and any associated organisms may be pumped into ballast tanks.

The ballast water is taken from coastal port areas and transported inside the ship to the next port of call where the water may be discharged, along with all the surviving organisms. This way, ballast water may introduce organisms into the port of discharge that do not naturally belong there. These introduced species are also called exotic species. Populations of exotic species may grow very quickly in the absence of natural predators. In that case they are called ‘invasive’.

Only few species are successful invaders, because most species are not able to survive in new surroundings, because temperature, food, and salinity are less than optimal. However, the species that do survive and establish a population are very hardy species that have the potential to cause major harm (to ecology, economy or human health).

Ballast is defined; “ballast is any material used to weight and balance an object. It is the additional weight necessary to bring the vessel to a suitable draft and trim and reduce stresses and improve stability.” In the ship’s terminology ballast is divided two types: clean ballast and dirty ballast. Clean ballast, if discharged from vessel that is stationary into clean, calm water on a clear day would...
not produce visible traces of oil on the surface of the water or on adjoining shore lines. Dirty ballast, to seawater introduced into cargo tanks upon completion of cargo discharge (Huge, 2001).

Ships have carried solid ballast, in the form of rocks, sand or metal, for thousands of years. In modern times, ships use water as ballast. It is much easier to load on and off a ship, and is therefore more efficient and economical than solid ballast. When a ship is empty of cargo, it fills with ballast water. When it loads cargo, the ballast water is discharged. Shipping moves over 80% of the world’s commodities and transfers approximately 3 to 5 billion tones of ballast water internationally each year.

Figure 2. Ballast exchange between ports (IMO GloBallast)

There are thousands of marine species that may be carried in ships’ ballast water; basically anything that is small enough to pass through a ship’ ballast water intake ports and pumps. These include bacteria and other microbes, small invertebrates and the eggs, cysts and larvae of various species. The problem is compounded by the fact that virtually all marine species have life cycles that include a planktonic stage or stages.

2. Preconditions and means for resolving the problem

The release of ballast water may introduce non-native organisms into the port of discharge. These introduced species, or bioinvaders, are also referred to as exotic species, alien species and no indigenous species. Typically, very few organisms are able to survive in new surroundings because temperature, food, and salinity are less than optimal; however, the few that do survive and establish a population have the potential to cause ecological and economic harm. Populations of bioinvaders may grow very quickly in the absence of natural predators. In turn bioinvaders may displace native organisms by preying on them or out competing native species for food and habitat space. Economic damage may occur when a bioinvader displaces species that are harvested for food or other goods, or when bioinvaders damage structures.

Apart from harmful effects such as consequences of shipping disasters, shipping activity exerts other negative influences on the environment; e.g. sea pollution through the discharges of oily water and sewage water from vessels, air pollution from exhaust gases emitted from the vessel’s machinery, pollution of water and marine organisms from toxic protective underwater hull coatings (antifouling paints), and one of the most recent water born concerns – the translocation of harmful organisms and pathogens via ballast water and sediments inside ballast water tanks.

Ballast water is absolutely essential to the safe and efficient operation of modern shipping, providing balance and stability to laden ships. However, it may also pose a serious ecological, economic and health threat for sea nature life. The introduction of invasive marine species into new environments by ships ballast water is an issue that does not always receive the public exposure that it merits.

But the problem clearly requires a multi-lateral approach. The introduction of invasive marine species into new environments by ships’ ballast water attached to ships’ hulls and via other vectors has been identified as one of the four greatest threats to the world’s oceans. The other three are land-based sources of marine pollution, over exploitation of living marine resources and physical alteration/destoyion of marine habitat. Quantity of ballast water is change depend of ship’s type. The release of ballast water may introduce non-native organisms into the port of discharge.

Black Sea geography contributes to the manifestation of additional environmental risks: these sea or do not have access or have a very limited relationship with the oceans. The Black Sea region presents a most unusual environmental problem. Of all the world is inland seas, it is the most isolated from the world is oceans. Its only link with other seas is with the Mediterranean, through the narrow channel so for the Bosphorus strait, the sea of Marmora and the Dardanelles. Relative to its size, this is indeed a tenuous link. Yet almost a third of Europe and huge areas of Asia drain into the Black Sea and more than 160 million people live in the overall Black Sea catchment area. The Black Sea coastal zone is densely populated. In the summer season, the permanent population of around 16 million swells to around 20 million with the influx of tourists.

During the last 30 years, the Black Sea environment has been transformed by the harmful effects of modern industry, agriculture and fishing. The add aditional damage caused by exotic marine species and pathogens in ships ballast water is another major contributor to the degradation of the environment. Pollution from land-based sources is one of the main causes of environmental degradation of rivers and, as a consequence, seas and coastal areas. Pollution from ships and other activities in the seas are another factor contributing to the deterioration of the environmental status of marine waters and coasts. Much of Georgia’s coastal zone is subject to significant anthropogenic pressures, that could be reason of causing the pollution of marine environment, and should be adopted the measures that will represent a significant step forward in the battle to reverse those harmful effects.

The introduction process of alien species is still ongoing in the Black Sea and it needs to be monitored at the national, regional and international level. A special monitoring programme is requested for key areas, in order to understand better the dispersion patterns of alien species. The impact of the alien species is complex and most of the time unpredictable due to lack of monitoring and the lack of scientific knowledge about those species. Experts on alien species, such as taxonomists, should be trained and encouraged. Capacity building for riparian countries is essential for the monitoring of alien species. Initiatives for the database management on Mnemiopsis and other jellyfish should be continued by an international organization like the Black Sea Commission. The International Convention for the Control and Management of Ship’s Ballast Water and Sediment (BWM Convention) within the International Maritime Organization (IMO) system was adopted in 2004. This convention has not come into force yet but in some countries like, the Russian Federation, Turkey and Ukraine the port authorities request the reporting of ballast water and follow ships to their ports. In a port Novorossiisk, ballast water is monitored for chemical contamination. Ukrainian authorities sample ballast water to assess possible chemical contamination (Matej and Gollash, 2008). Turkish authorities conduct a project for the impacts of the ship ballast waters on the
Turkish Seas. This kind of implementation should be encouraged to prevent alien species to enter local seas. To control alien species via incoming ships, a defined concerted area for discharging ballast water should be established in the Black Sea. Some of these national or local Ballast Water Management legislations are generally consistent with the IMO Convention but others impose different and often more stringent requirements on ships. Inevitably this leads to confusion amongst owners, operators and seafarers.

There may be conflicting requirements at different parts of a voyage which inevitably increase the risk of regulations being breached. Most introductions of non-indigenous species result from ballast discharge and sediment from vessels after ocean crossings. Georgia is an exporting country. Most vessels arriving in Georgian ports discharge ballast and then load oil. According to the Convention on the Protection of the Black Sea emptying segregated, un-contaminated ballast water is allowed. But different countries of the region enforce the Convention differently. For example, vessels calling for the port of Odessa have to change their ballast water immediately upon entry into the Black Sea area. This has to be recorded in the ship logbook. This policy is not a viable solution, since ballast waters are emptied upon arrival in the Black Sea.

A synopsis of known national and local ballast water management regulations for Georgia.

### Figure 3.

According to the National ballast water management requirements (January 2014), Lloyd’s Register Marine

2.8 Georgia.

- Authority: Georgian Environmental Protection Ministry;
- Ports affected: All Georgian ports;
- Ships affected: All;
- Implementation: Mandatory;
- Start date: No information;
- Acceptable methods: Ballast water exchange (BWE): BWE must be conducted in the Black Sea;
- Unwanted organisms and pathogens: No information;
- Uptake control: No information;
- Sampling: No information;
- Ballast Water Management Plan: Required;
- Records and reporting: No information;
- Alternatives to en route management procedures: No information;
- Procedure for unacceptable ballast water: No information;
- Notes: No information.

### 3. Conclusion

Despite the serious degradation that has already occurred in the Black Sea, studies have indicated that concerted action can both restore and protect the environment, and should be adopted the measures that will represent a significant step forward in the battle to reverse those harmful effects.

### 4. Literature

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5. GLOBALLAST MONOGRAPH SERIES NO.3, 4
METHODS AND INSTRUMENTS FOR MEASURING TORQUE AND SPEED OF MARINE DIESEL ENGINES

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Abstract: Torque and speed measurement has always been a great challenge for many industries such as aviation, shipbuilding and automotive industry. These forces are of substantial importance for the research of the deformation processes developing in the modern marine diesel engines. The values of these forces provide the input data for calculating the overall strength of ship power plants. Continuous monitoring of these parameters while the ship is in service ensures safe operation of all machinery, reduces the risk of unplanned repair works and improves the engine performance in view of low fuel consumption and reduction of CO2 and NOx emissions.

Keywords: SHAFT TORQUE METER, TORQUE TRANSDUCER, MEASUREMENT, TORQUE MEASUREMENT SYSTEM, SPEED MEASUREMENT SYSTEM.

1. Introduction

Ship’s shaft line is an elastic system loaded with driving and reaction moments that have variable magnitude and direction. The moment generated by the pressure of the gases in the cylinder and the moment created by the mass of the pistons and connecting rods are examples of driving moments whereas the propeller moment and the frictional moment in the cylinders and bearings are classified as reaction moments. The inertia moments of the connecting rod assembly, of the flywheel and the propeller have a specific function. Upon accelerating they serve as reaction moments because they counteract the speed changes as opposed to slowing down when they have the same direction of the driving moments. As a result of the combined action of the driving, reaction and inertia moments, the shaft line is twisted, this twist being different for each shaft section.

2. Problem discussion.

The magnitude of these forces is determined in a different way:
- the torque is determined by means of torque meters which can be two types – mechanical and electrical;
- speed is measured by direct and indirect methods by means of instruments called speedometers.

All measuring instruments shall meet certain requirements in compliance with the international standards and the Bulgarian Register of Shipping (BRS). BRS has ISO 9000 certification.

3. Objective and research methodologies.

3.1 Indirect measuring with sensors

Measurement of shaft torque assists in determining the fuel consumption of ships. The company KYMA aims at ensuring that the newly-built ships are as energy efficient as possible. For this purpose KYMA [1] Norway uses indirect measurement of torque by means of sensors (strain gauges).

Generally the active part of the sensors is about 2-10 mm². They have elastic insulation base on which metal foil is laid down. The sensor is attached to the shaft with the help of suitable adhesive, for example, cyanoacrylate glue. Strain gauges are mounted in pairs on the shaft, one of them measures the increase in length (in the direction in which the surface is under tension) and the other measures the decrease in length in the opposite direction.

The elements are located on the shaft axis in such a way that the resistance of the elements increases, if the axis is subjected to tensile forces.

Modern electronics for torque measurement is based on resistance changes in Wheatstone bridge.

3.2 Direct methods for shaft twist measurement

Here follows an overview of some of the leading companies and their modern methods and instruments:

3.2.1. LEMAG Marine Instruments [2]

LEMAG Marine specializes in the field of fuel systems for reduction of emissions and reduction of fuel consumption. LEMAG offers shaft twist measurement by means of induction sensors that measure displacement.

In order to measure the twist of rotating shafts:
- 2 rings are mounted on the shaft, spaced 500 mm apart
- 2 precision sensors are mounted opposite to the electrical arms
- The data measured are transmitted by radio waves from the shaft to the stationary unit.

Advantages of LEMAG method:
- no external (shore) maintenance is required
- the crew can calibrate the system when the engine is stopped by means of turning device
- stability and long life of the system

The following parameters are displayed on the touch screen:
- Torque (kNm)
- Shaft speed (rpm)
- Output power (hp)
- Mean output power (hp)
- Comparison of the limit curves of the nominal power and of the propeller.

3.2.2 VAF INSTRUMENTS – Holland [3]
VAF Instruments is a leading company in the design, production and sales of measurement and control systems worldwide.
For measurement of the twist of rotating shafts the company offers the following:
- two rings mounted on the shaft, spaced 250 mm apart
- the displacement between both rings is determined on the principle of optical measurement
- extreme accuracy of the optical sensor (within nanometer range)
- possibility for analysis of torsional vibration
- 4 GHz wireless transmission of data
- induction control by a transmitter
- difference of 3 mm between the fixed and running parts

Advantages:
- mounting of sensors is not time-consuming
- easy mounting and starting-up
- total costs for mounting are not big
- no wear, no need for maintenance and re-calibration
- no maintenance fees, low costs throughout the whole service life.

3.2.3. HOPPE Marine – Germany [4]
One solution that the company offers is as follows: two identical toothed wheels are mounted on the shaft at a certain distance and measure the angular displacement – fig.5 and fig.6. They can be mounted on shafts rotating up to 150 rev/min.
HOPPE POWER METER
- measures shaft twist
- two pairs of toothed wheels are mounted at a distance of 7-10 m; the number of teeth $z_1 = 480$ and $z_2 = 48$.
The calculation of torque is proportional to the phase displacement of the electrical impulses. The bigger the phase difference, the bigger the torque.

3.2.4. KONGSBERG MARITIME – Holland [5]
The company has developed another interesting method. Two identical disks with openings (code wheels) are mounted on the shaft at a certain spacing apart – fig.7 and fig.8. A photocell is exposed to LED light passing through both wheels. As torque occurs, one of the wheels overlaps the other, the light flow is changed and the photocell detects this change.
- measures the twist of rotating shafts
- light is transmitted by LEDs through the openings of the code wheels mounted at a distance of 1 to 4 m.
- digital electronic signal is transmitted for processing and calculating of torque.

Advantages in comparison with the conventional methods for torque measurement:
- no sensitive electronics
- no mechanical wear
- no zero point displaced in the course of time
- not affected by the ambient temperature
- easy re-fitting after inspection of the shafting (no replacement of torque sensors)
- not affected by centrifugal forces
- insensitive to electrical fields

Maintenance
- no wear, no tear of LEDs
- maintenance made by crew
- lenses and rings are cleaned by compressed air

3.2.5 GREX – Bulgaria [6]
The firm GREX with president Vladimir Grigorov from the city of Varna offers another method for torque measurement: two identical magnetic bands are mounted on the shaft at a certain length and they measure the phase displacement as shaft torque occurs – fig.9.
Non-contact, digital measurement of torque by means of
- two metal bands with magnetic properties located at a distance of 0,8 -1,2 m
- two magnetic sensors at 0,8 – 1,6 mm from the shaft
The magnetic sensors are easy to install on three-dimensional adjustable arms. They generate 128-628 impulses per revolution of shafts with $D = 0.2 – 1.0$ m. The microprocessor in the transmitter measures the delay $s_0, s_1, s_2, \ldots, s_n$ between the impulses caused by the shaft twist. The accuracy of the digital measurement is very high with a resolution of 100 nanoseconds (ns).

Accuracy
The torque is measured digitally by the phase displacement of the magnetic impulses of the band at periods of 100 ns.
Advantages:
- digital method for measurement of shaft twist, not affected by ambient temperature changes, thrust bearing or power supply
- connected with non-contact digital magnetic sensors, no sensitive electronics
- no need for provision of power supply for shaft elements
- no need for transmitting of the rotating parts data by telemetric signals
- not affected by centrifugal forces
- no mechanical wear
- high accuracy, resolution time – 100 ns
- no zero point displaced in the course of time
- simple and easy installation
- low installation costs, no costs for maintenance
- easy re-fitting after inspection of the shafting
- zero calibration by pressing a button
- no maintenance and easy to operate

4. Conclusion.

International practice of leading companies dealing with measuring torque and speed, includes two primary ways to solve:

- indirect measurement using strain gauges;
- direct measurement based on the measurement of the twist shaft.

4. Literature.

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A STUDY OF PRESSURES IN PNEUMATIC TYRE INFLUENCE ON VEHICLES BRAKING DECELERATION

ИЗСЛЕДВАНЕ НА ВЛИЯНИЕТО НА НАЛЯГАНЕТО В ПНЕВМАТИЧНИТЕ ГУМИ ВЪРХУ СПИРАЧНОТО ЗАКЪСНЕНИЕ НА АВТОМОБИЛИ

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Abstract: In this article has been considered the influence of various pressures in pneumatic tires of passenger cars with anti-lock braking system and without it on chosen parameter of braking process - braking deceleration. The experiments under snow road conditions are carried out. Used tires for the experiments are for winter conditions.

Keywords: PRESSURES IN PNEUMATIC TYRE, BRAKING DECELERATION, SNOW ROAD CONDITIONS.

1. Introduction

Maintaining correct inflation pressure in tires helps to keep vehicle handling and braking at its best, as well as improving fuel efficiency and tyre life. In addition it can prevent such events as tread separations and tyre blowouts which may cause loss of control of a vehicle and severe crashes such as rollovers.

Under-inflated tires can potentially result in:
- reduced vehicle handling;
- increased braking distance;
- increased likelihood of blowouts;
- increased tyre wear;
- increased fuel consumption.

Tyre pressure affects the handling of a vehicle particularly during an emergency maneuver. For loss of control crashes, inappropriate speed is usually the most critical factor. Excessive speed alone can cause a loss of control in a curve or in a lane change maneuver. Tread depth, inflation pressure of the tires, and road surface condition are the most notable of a long list of factors including vehicle steering characteristics and tyre cornering capabilities that affect the vehicle/tyre interface with the road [3, 4, 5, 6, 8].

Tires are designed to maximize their performance capabilities at a specific inflation pressure [1, 2]. The relationship of tyre inflation to stopping distance is influenced by the road conditions (wet versus dry), as well as by the road surface composition. Decreasing stopping distance is beneficial in several ways. First, some crashes can be completely avoided. Second, some crashes will still occur, but they occur at a lower impact speed and so reduce the severity of the crash and the injuries suffered.

In winter conditions deceleration decreased twice, and in some cases even more [6, 7]. It often happens that the roads are covered with snow trampled during the braking process of cars to be carried out on it.

The aim of this article is to establish the influence of tire pressure on deceleration when braking on asphalt covered with trampled snow.

2. Experimental procedure and equipment

A study of the pressure in tires influence on vehicles deceleration VBOX 3i Data Logger and IMU02 (fig. 1), of a Racelogic Ltd UK company are carried out [9]. VBOX 3i is highly valued test instruments for non-contact speed and distance measurement. It has a very powerful processor ensuring low latency with updates of speed, position and acceleration, 100 times a second. All data is logged to a compact flash card [10].

![Fig. 1. Acceleration sensors IMU02 (1) and VBOX 3i 100Hz GPS Data Logger (2) ](image)

The IMU02/YAW03 channel data will be recorded along with the existing GPS data in VBOX 3i 100Hz GPS Data Logger on the SD card. Accuracy of acceleration measurement is 0,5% [9].

The IMU02 from Racelogic is a full Inertial Measurement Unit that can measure Z, Y and X axis rotational rate (yaw, pitch and roll) as well as X, Y and Z axis acceleration. For this study a lateral accelerations (X axis) are used.

The IMU02 is mounted as close as possible to the centre of the vehicle. It is also important to mount the sensor so that it is level with the ground.

Method of the study involves a series of tests to determine the cars deceleration (tire–road coefficient of friction) and tire pressure influence on this deceleration for covered with trampled snow during emergency braking.
The studies were conducted by Ford Focus, Citroen Xsara Picasso and Opel Astra with recording equipment VBOX 3i Data Logger and IMU02 (fig. 1), tests were carried out with ABS and ABS off (fig. 2).

![Ford Focus, Citroen Xsara Picasso and Opel Astra](image)

**Fig. 3. Winter conditions**

Tests were conducted with three different cars with tires for winter conditions (fig. 3). The description of tires for various cars is shown in Table 1.

### Table 1

<table>
<thead>
<tr>
<th>Car</th>
<th>Dimensions of tires</th>
<th>Dot tires</th>
<th>Type of tires</th>
<th>Tire’s company</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ford Focus</td>
<td>195/65 R15</td>
<td>4112</td>
<td>winter conditions</td>
<td>Hankook</td>
</tr>
<tr>
<td>Citroen Xsara Picasso</td>
<td>195/60 R15</td>
<td>4011</td>
<td>winter conditions</td>
<td>Michelin</td>
</tr>
<tr>
<td>Opel Astra</td>
<td>195/70 R14</td>
<td>3808</td>
<td>winter conditions</td>
<td>Continental</td>
</tr>
</tbody>
</table>

During measurement the vehicle was loaded with two adult sitting on the front seats and one on rear (the driver and two passengers). The cars were tested at about 50 km/h initial speed measured by the speedometer of the car (speed is more precisely defined by the system of data collection VBOX 3i Data Logger). In every case, when the desired speed was reached, sudden braking was realized by the service brake (wheels of cars without ABS were locked during braking), the brake pedal was held pressed down till the car full stop. Every measurement was repeated three times in driving in the opposite direction.

Proper-inflated tire pressure is selected the recommended pressure written on plates of cars, namely 0,22MPa. There is no universal definition of what constitutes an "under-inflated tyre". The US Federal Motor Vehicle Safety Standard 138 requires a warning if tires are under-inflated by more than 25% [8]. Therefore, in under-inflated pressure is selected 0,15MPa.

**3. Results and Discussion**

The results of the study were obtained with VBOX Tools Software. VBOX Tools software is a very powerful software analysis package. VBOX Tools has been designed to easy to setup and use, yet very flexible allowing many custom styles of test to be performed, as well as providing templates of common test setups. Processing mode, a “vbo” file were taken from a compact flash card and loaded into the software, allowing graphing, analyzing and replaying the data.

The results are processing and presented determining of the maximum car deceleration (fig. 4). Average deceleration reported data after the increase in the values and their establishment to start reducing deceleration.

![VBOX Tools Software chart](image)

**Fig. 4. VBOX Tools Software chart**

The Report Generator is the main “numbers engine” in the VBOX Tools software, and is designed to create a highly configurable table of results (fig. 5).

![VBOX Tools software table of results](image)

**Fig. 5. VBOX Tools software table of results**

The columns of the table can contain any parameter logged by the VBOX, and may also shows maximums, minimums, and averages. The scale and offset of any channel is also configurable [11].

To evaluate the dissipation of the values of each test the coefficient of variation is used. It is calculated by the following equation:

\[
V = \frac{S}{\bar{x}} \times 100\% \quad (1)
\]

\[
S = \sqrt{\frac{\sum (x_i - \bar{x})^2}{n}} \quad (2)
\]

Considering that the coefficient of variation \(V\) for all series of tests was in the range of (3 – 8)% that the scattering of the results obtained is not significant.

The figure 6 shows average values of maximum deceleration in depending on pressure in tires and ABS on or ABS off in winter conditions on trampled snow for Ford Focus.

The biggest average deceleration was registered for braking with the nominal level of the pressure in tires (0,22MPa) of the vehicle, irrespective of whether ABS is on (about 2,84 m/s² for braking with ABS and about 3,02 m/s² when ABS is off). Braking with other pressure than nominal one (0,15MPa), according to processing (fig. 6) shows that the deceleration for ABS and ABS off are respectively 2,55 m/s² and 2,72 m/s².

Comparing the under-inflated (0,15MPa) tyre deceleration with proper-inflated (0,22MPa) tyre deceleration for Ford Focus (fig. 6) follows: for braking with ABS – 11,37% decrease; for braking with ABS off – 11,03% decrease.
The figure 7 shows average values of maximum deceleration in depending on pressure in tires and ABS on or ABS off in winter conditions on trampled snow for Citroen Xsara Picasso.

The biggest average deceleration was registered for braking with the nominal level of the pressure in tyres (0.22MPa) of the vehicle, irrespective of whether ABS is on (about 3.04 m/s² for braking with ABS and about 3.27 m/s² when ABS is off). Braking with other pressure than nominal one (0.15MPa), according to processing (fig. 7) shows that the deceleration for ABS and ABS off are respectively 2.81 m/s² and 3.07 m/s².

Comparing the under-inflated (0.15MPa) tyre deceleration with proper-inflated (0.22MPa) tyre deceleration for Citroen Xsara Picasso (fig. 7) follows: for braking with ABS – 8.18% decrease; for braking with ABS off – 6.51% decrease.

The biggest average deceleration was registered for braking with the nominal level of the pressure in tyres (0.22MPa) of the vehicle, irrespective of whether ABS is on (about 3.04 m/s² for braking with ABS and about 3.27 m/s² when ABS is off). Braking with other pressure than nominal one (0.15MPa), according to processing (fig. 8) shows that the deceleration is respectively 2.23 m/s².

Comparing the under-inflated (0.15MPa) tyre deceleration with proper-inflated (0.22MPa) tyre deceleration for Opel Astra (fig. 8) follows: for braking without ABS – 8.97% decrease; for braking with ABS – 8.18% decrease.

The car has no factory ABS. The results of the comparison of the two decelerations in the other vehicle equipped with a newer tire indicate that the deceleration in the Opel Astra is lower than with (20-35)%.

The results of this study indicate that the longitudinal deceleration influence of the pressure in tires of these cars is greatest for proper-inflated tires. It is in the range of from 6.51% to 11.37%. For trampled snow-covered surface for two cars ABS off longitudinal deceleration is greater. The results indicate that further research is needed to determine the ABS longitudinal deceleration influence for snow-covered surface.

Lower values of the deceleration when tires are under-inflated probably due, the shape of the tyre’s footprint and the pressure it exerts on the road surface are both altered (fig. 9). This degrades the tyre’s ability to transmit braking force to the road surface.

Fig. 9. The reduced pressure tire exerts on the road surface when tires are under-inflated

4. Conclusion

On the basis of investigation results observed that:

- For Ford Focus the under-inflated (0.15MPa) tyre deceleration with ABS - 11.37% decrease, with ABS off – 11.03% decrease, to proper-inflated (0.22MPa) tyre deceleration.

- For Citroen Xsara Picasso the under-inflated (0.15MPa) tyre deceleration with ABS – 8.18% decrease, with ABS off – 6.51% decrease, to proper-inflated (0.22MPa) tyre deceleration.

- For Opel Astra the under-inflated (0.15MPa) tyre deceleration – 8.97% decrease, to proper-inflated (0.22MPa) tyre deceleration.

5. References


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SIMULATION ON SINGLE CYLINDER DIESEL ENGINE AND ESTIMATION OF ENGINE PERFORMANCE USING AVL BOOST SOFTWARE

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Abstract: The simulation and computational development of modelling for the research is use the commercial Computational Fluid Dynamics (CFD) of AVL Boost software. In this research, the one dimensional (1D) CFD modelling of four-stroke direct injection diesel engine is developed. The analysis of the model is fluid flow and combustion performance process in the engine cylinder. In this model it can to know the diesel engine performance effect with simulation and modelling in any speed (rpm) parameters before to do the physically development, so it can do the new engine design components with the economic material and time. The model simulation covers the full engine cycle consisting of intake, compression, power and exhaust. The simulation result highlighted energetically and economic performance of the engine.

Keywords: SIMULATION, DIESEL ENGINE, MODEL, AVL BOOST, PERFORMANCE, SINGLE CYLINDER

1. Introduction

In the last decades, the legislation on internal combustion engines (ICEs) has severely reduced the limits for pollutant and noise emissions. These requirements have established the research activity at design phase as a key stage in the engine production process. Therefore, an intensive investigation on ICEs has been carried out, focusing on the optimization of performances and fuel consumption. In particular, an important effort has been done seeking the improvement of the combustion and gas exchange processes, using tools such as Computational Fluid Dynamics (CFD).

Diesel engines are typically characterized by low fuel consumption and very low CO emissions. However, the NOx emissions from diesel engines still remain high. Hence, in order to meet the environmental regulations, it is highly desirable to reduce the amount of NOx in the exhaust gas.

Simulating an intake or exhaust system is just a great exponent of this sort of problems. These systems are mainly composed of ducts, which can be accurately simulated by means of one-dimensional, non-viscous codes. However, there are several components that manifest a complex three-dimensional flow behavior, such as turbo machinery or manifolds, therefore being unable to be simulated properly by 1D codes, and thus requiring viscous, 3D codes.

Hence, it is a right choice to save computational time by simulating the complex components by means of a 3D code and modeling with a 1D code the rest of the system, i.e. the ducts. In this way, a coupling methodology between the 1D and the 3D code in the respective interfaces is required, being the objective of numerous authors [1–4].

AVL Boost is based on 1D gas dynamics which account for fluid flows and heat transfer. Each component in a AVL Boost model is discretized or separated in many smaller components. These components have very small volumes and the fluid’s scalar properties in these volumes are assumed to be constant. The scalar properties of a fluid include pressure, temperature, density and internal energy. Each volume also have vector properties that can be transferred across it’s boundaries. This properties include mass flux and fluid velocity.

Heywood [5] written that the engine ratings usually indicate the highest power at which manufacturer expect their products to give satisfactory of power, economy, reliability and durability under service conditions. The speed and maximum torque at which it is achieved, is usually given also.

2. Data Needed for Building an Engine Model

AVL Boost is a software tool that consists in a pre-processing program, used for initial data entry and technical characteristics of the engine to be designed as model. After forming the engine assembly with annexes systems, mathematical equations and algorithms of the model with the graphical user interface (GUI) will analyze and calculate the processes that are required during simulation [6, 7]. The model for the engine designed in AVL BOOST application is shown in figure 1.

![Fig. 1 Model of Single Cylinder Diesel Engine: SB-system boundaries, MP-measuring points, C-cylinder.](image)

A list of information that is needed to create a model in AVL BOOST is included in library. The main features of the diesel engine that have been used as initial data to define the cylinder parameters are presented in table 1. Cylinder (C1) of the model in AVL Boost is connected with element Engine (E1), and it defines the type of engine used, operating speeds on it, moments of inertia and break mean effective pressure (BMEP). Combustion method is Mixing Controlled Combustion model that predicts the rate of heat released (ROHR) and NOx emissions on the quantity of fuel in the cylinder and the turbulent kinetic energy introduced by the injection of fuel [8].
Table 1: Specification of the engine

<table>
<thead>
<tr>
<th>Engine Parameters</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bore</td>
<td>76</td>
<td>[mm]</td>
</tr>
<tr>
<td>Stroke</td>
<td>65</td>
<td>[mm]</td>
</tr>
<tr>
<td>Displacement</td>
<td>295</td>
<td>[cc]</td>
</tr>
<tr>
<td>Power</td>
<td>4</td>
<td>[kw]</td>
</tr>
<tr>
<td>Speed</td>
<td>3000</td>
<td>[rpm]</td>
</tr>
<tr>
<td>Compression ratio</td>
<td>17:1</td>
<td>-</td>
</tr>
<tr>
<td>Valve lift</td>
<td>8.5</td>
<td>[mm]</td>
</tr>
<tr>
<td>Piston pin offset</td>
<td>10</td>
<td>[mm]</td>
</tr>
</tbody>
</table>

The Diesel engines by small power have a wide range of application for the mechanization of the most activities in industry and agriculture. This type of engines, with power up to 10kW, is using, in most of the cases, the air-cooling.

3. Result and Discussion

After definition of the engine parameters has been run a series of simulations, and then plot the results in Impress Chart were was analyzed.

![Fig. 2 Evaluation of the cylinder pressure and temperature](image)

The running simulation result is all of the engine performance data with the different engine speed (rpm). This model was running at any speed from 2000 to 3000 rpm.

![Fig. 3 Mean effective pressure of engine model](image)

In figure 1 show the evolution of the cylinder pressure and temperature for the 3000 rpm engine speed. Maximum cycle pressure is 5.9 MPa and temperature of 2065.97 K. The maximum duration of the ignition delay is 5.5°, at this stage are visible the processes resulting with heat absorption (latent heat vaporization of diesel fuel, the first reaction of oxidation) and causes a reduction in air pressure and temperature increase. The Diesel engines by small power have a wide range of application for the mechanization of the most activities in industry and agriculture. This type of engines, with power up to 10kW, is using, in most of the cases, the air-cooling.

Mean effective pressure variation is illustrated in figure 3. The range of it is between 0.54 to 0.58 MPa, the values are close to the real case of engines in this class.

![Fig. 4 Residual gas coefficient](image)

Figure 4 presents the influence of speed on residual gas coefficient. Specific range of diesel engines is between 0.03 and 0.06. The values obtained indicate a proper discharge of the cylinder.

![Fig. 5 Brake specific consumption of engine model](image)

The brake specific fuel consumption is shown in Figure 5. The model simulation result shown that the minimum brake specific fuel consumption is 263 g/kWh at 2300 rpm.

The brake power of the engine model is shown in Figure 6. Brake power is usually measured by attaching a power absorption device to the drive-shaft of the engine (any type of brake). If the engine speed is increased the brake power is increased too until engine speed 2900 rpm. The maximum brake power of the engine model is 3.95 kW at engine speed 2900 rpm and after that the brake power decreases.
3. Conclusion

This paper presents the results of the engine cycle simulation of a single cylinder direct injection diesel engine using AVL Boost software. The engine cycle simulation processes show the evolution of the main specific parameters. Analyzing the obtained results, it finds that they correspond to the real range of variation for this type of engines. The residual gas coefficient can be improved by optimizing the intake and exhaust systems of the engine. The results obtained by simulation show that engine cycle simulation offers an accurate picture of the progress of real processes from a diesel engine.

Acknowledgment

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Reference


Fig. 6 Brake power of engine model
STUDYING OF VIBRATIONS, ACTING OF THE DRIVERS OF THE ROAD-BUILDING MACHINERY AND AUTOMOBILES

ИЗСЛЕДВАНЕ НА ВИБРАЦИИТЕ, ДЕЙСТВАЩИ НА ВОДАЧИТЕ НА ПЪТНО-СТРОИТЕЛНИ МАШИНИ И АВТОМОБИЛИ

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Abstract: The vibrations caused by road-building machinery affect drivers equally harmful to the three axes of the coordinate system. Assessment of exposure to vibration hand-arm system is based on the calculation of daily exposure value for the 8-hour A (8).

Keywords: VIBRATIONS, ROAD-BUILDING MACHINERY, AFFECT DRIVERS, HAND-ARM SYSTEM.

1. Introduction

The impact of vibration on humans is associated with the oscillation of certain internal variable force effects on the machine, or on its system. At the beginning of this kind of oscillations may be associated not only with power but with kinematic excitation, typically for vehicles in their movement on rough roads.

The minimum requirements to protect workers from existing or potential risks to health and safety associated with exposure to vibration at work are set out in Ordinance № 3 of 5.05.2005. According to this Ordinance, the vibrations are divided into vibration system "hand-arm" and vibrations transmitted to the whole body. [2]

Vibrations of the whole body disturbs the human body. It is normalized by taking into account the source of the impact that a sign is divided into:
- transport - result from movement of machinery in areas and roads;
- transportation technology - are formed during operation of machines performing technological operation in stationary and/or moving of a specially prepared portion of the production area or industrial site;
- technology - arise when working on stationary machines or transmitted jobs, have not got source of vibration.

The purpose of this study is to identify and demonstrate the values of vibration experienced by drivers of different types of road-building machinery (trucks, excavators, tractors and other vehicles).

The transport machineries and vehicles are randomly selected, the only condition for their research is more common types operating in the region of Smolyan.

In accordance with the ordinance, the values of vibrations of hand-arm system should not exceed the daily exposure limit value set for the 8-hour (5 m/s²) and the daily exposure value action specified period 8 h (2,5 m/s²).

The values of the vibrations of the whole body must not exceed the daily exposure limit value set for the 8-hour (1,15 m/s²) and the daily exposure value action set for the 8-hour (0, 5 m/s²).

2. Theoretical formulation and methodology of the study

Used in the article terms and definitions of Ordinance № 3 of 5.05.2005 and BDS EN ISO 5349-1 Vibrations: exposure, daily value of exposure action, daily value of exposure set for the 8 hours (A (8) or \(a_{hv(eq,8h)}\) (m/s²), the total value of vibration frequency weighted rms acceleration \(a_{hv}\) (m/s²), RMS acceleration frequency-weighted vibration in hands, one axis \(a_{hv}\), (m/s²).[3]

Assessment of exposure to vibration hand-arm system is based on the calculation of daily exposure value for the 8-hour A (8).

The daily vibration effect is obtained from the magnitude of vibration (vibration of a total amount) and the length of day effects.

Daily exposure value for the 8-hour is calculated by the formula:

\[
A(8) = a_{hv} \sqrt{\frac{T}{T_0}},
\]

where:
- \(A(8)\) is the daily amount of exposure to vibration in m/s²;
- \(a_{hv}\) - total vibration values in m/s²;
- \(T\) - total daily duration of exposure in h (s);
- \(T_0\) - duration 8 h (28 800 s).

The total value of the vibration is determined by the formula:

\[
a_{hv} = \sqrt{a_{hv_x}^2 + a_{hv_y}^2 + a_{hv_z}^2},
\]

where:
- \(a_{hv}\) is the total value of vibration in m/s²;
- \(a_{hv_x}, a_{hv_y}, a_{hv_z}\) are frequency-weighted RMS acceleration in m/s², measured in three axes - x, y and z the vibrating surface in contact with the hand.

From the cited methods, it is clear that the longer the time of impact, the driver will be exposed to a higher exposure and at eight-hour working day, the exposure will be equal to the total value of the vibrations - \(a_{hv}\), and when the length of the impact is less than 8 hours, it is necessary recalculated leftmost value of the daily exposure to vibrations A (8).

Measurement of vibrations in the system "hand-arm" and the whole body were made with a meter conforming to ISO 8041. Device is calibrated by an accredited laboratory valid until 2015. Before each measurement, the device is controlled with vibrokalibrator, which is also calibrated by an accredited laboratory valid till October, 2014.

The apparatus allows measurements to be made in three axes simultaneously.
Vibration arm were recorded for the three strands of rectangular coordinate system, as shown in Figure 1. Orientation of the coordinate system in measurements correspond to BDS EN ISO 5349-1.

Methodology in Ordinance № 3 of 5.05.2005, and BDS EN ISO 5349-1 requires the sensor to be placed in areas where there is contact of the body with a vibrating surface. When measuring vibration system "hand-arm", the sensor is located between the arm and the vibrating surface, which in this case the steering wheel.

Drivers of road-building machinery - trucks, excavators, tractors and other vehicles are tested parameters of vibration system "hand-arm." Observed parameters apply to both hands in contact with the steering wheel.

According to standard BDS EN ISO 5349-1, the vibrations in each of the three directions defined by the axes of the rectangular coordinate system shown in Figure 1, are equally harmful and that the same frequency weighting can be used for each axis. Therefore, the risk of damage caused by vibrations, transmitted by hand, are evaluated by the total value of the vibrations, which is equivalent to the energy for a period of 8 hours, the appropriately reflects the relationship between the different sizes of the vibration and the duration of the daily action.

3. Analysis of the results of studies of production of vibration

The purpose of the statistical survey is to identify and demonstrate the level of vibration experienced by drivers of different types of road-building machinery (trucks, excavators, tractors and other vehicles), without focusing on the duration of exposure. They focused reported and registered by the device totals vibration \( a_{th} \) and RMS acceleration \( a_{rms} \), frequency weighted RMS acceleration \( a_{w} \), measured of the three axes - x, y and z the vibrating surface.

Conditions under which the measurements were performed are the same for the groups of machines: type of road surface (asphalt, stone, rough road); movement of the car (loaded or unloaded, in no time flat, horizontal gradients); state of the road surface (wet, dry, snow, smooth, rough, flat, downhill, uphill) and instantaneous technical condition of machines.

The researches has been aimed to determining exposure, which requires measurement of the vibration level for the time of impact, i.e. for the entire period of operation of the machines, the data presented in this article apply only to the level of vibration measurement time of 30 min.

Of each machine have been studied a number, and reported by the device parameters are averaged for each species.

Object of study in this article the vibration during the work in the operating conditions of the following types of road construction machines and vehicles:

- excavator: excavator "JCB" - 4 pcs.; wheel excavator "ATLAS 1304" - 2 pcs.; front loader "ATLAS 52 D" - 2 pc.; mini excavator "Bobcat" - 2 pcs.;
- Tractors wheel - "Universal" 651 M - 2 pcs.; UMZ 61 - 2 pcs.; "TK-80" - 2 pcs.;
- Tractors chain - T 170-1 pc.; DT 75 - 1 pc.
- Trucks: KAMAZ 5511-12 pcs.; DAF cf 85-5 pcs.; MAN TGS - 10 pcs.; STEYR 91; IFA L60; IFA W 50; Mercedes 914; Mercedes 409.

In carrying out a statistical measurements are made as one of the two factors of the said standard, which affects the impact of the vibration arm, namely the magnitude of the vibrations. The parameters considered were: the total value of the vibration of hand-arm system, frequency-weighted RMS acceleration in m/s², measured along three axes - x, y and z of the hand-arm system.

<table>
<thead>
<tr>
<th>Types of machines</th>
<th>Source of vibrations</th>
<th>( a_{th} ) - RMS acceleration of the frequency weighted vibration axis &quot;a&quot; (m/s²)</th>
<th>( a_{rms} ) - Total vibration frequency weighted rms acceleration (m/s²)</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Excavators</strong></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Backhoe &quot;JCB&quot;</td>
<td></td>
<td>0,31</td>
<td>0,33</td>
</tr>
<tr>
<td>Wheel excavator &quot;ATLAS 1304&quot;</td>
<td></td>
<td>3,15</td>
<td>4,01</td>
</tr>
<tr>
<td>Front loader &quot;ATLAS 52 D&quot;</td>
<td></td>
<td>2,14</td>
<td>3,23</td>
</tr>
<tr>
<td>Mini excavator &quot;Bobcat&quot;</td>
<td></td>
<td>1,87</td>
<td>0,93</td>
</tr>
<tr>
<td><strong>Wheeled tractors</strong></td>
<td>Universal 651 M</td>
<td>3,27</td>
<td>3,71</td>
</tr>
<tr>
<td></td>
<td>UMZ 61</td>
<td>2,75</td>
<td>3,38</td>
</tr>
<tr>
<td></td>
<td>TK-80</td>
<td>4,72</td>
<td>6,83</td>
</tr>
<tr>
<td><strong>Crawler tractors</strong></td>
<td>T 170</td>
<td>4,88</td>
<td>5,21</td>
</tr>
<tr>
<td></td>
<td>DT 75</td>
<td>4,25</td>
<td>4,83</td>
</tr>
<tr>
<td><strong>Trucks</strong></td>
<td>KAMAZ 5511</td>
<td>2,51</td>
<td>1,48</td>
</tr>
</tbody>
</table>
Results obtained from the survey were processed using methods of mathematical statistics and probability theory, and are summarized in tabular and graphical dependencies.

6. References


[2] Ordinance № 3 of 5.05.2005 on the minimum requirements for the provision of health-veto and safety of workers on the risks related to exposure to vibration.


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4. Conclusion

1. From the measurements, calculations and built graphical relationships shows that in the tractors levels of vibration of the "hand-arm" are higher than other tested machines.

2. Investigations show that the vibration measured on vehicles and road construction equipment are higher in tractors and trucks from older models.

3. Vibration system "hand-arm" are higher in axis "y" in the majority of the machines.
THE NEW METHOD OF DETERMINATION OF VESSEL WEIGHT AND APPLICATE OF ITS CENTRE OF GRAVITY BY USING OF ELASTIC INFLATABLE TANKS

Abstract: The new and modified method of experimental determination of the vessel weight and the coordinates of its center of gravity without launching using inflatable elastic containers are shown in this article.

KEYWORDS: VESSEL, INCLINING, TEST, ELASTIC, TANKS

1. Introduction

The methods of inclining and weighing tests which are proposed in article “Determination of vessel weight and its centre of gravity by using of elastic inflatable tanks”, published in “trans&MOTAUTO’13 digest” had some difficult and limits. The main difficulty in case of transverse arrange of elastic inflatable tanks (EICT) is to determine the contact area of EICT with the bottom of the vessel. This is due to the fact that when the ship heeled standing on the elastic vessels perpendicular the vessel center line, the contact area takes the curved shape. (fig. 1-3).

To facilitate and accelerate inclining and weighing test the longitudinal placement of inflatable tanks scheme (parallel center line) had proposed. The heeling of the vessel in use of this scheme is due to change the EICT pressure in the EICT located from one side of center line. The advantages of this scheme is that the use of cargo for inclining is not required, hence not require the deck strengths calculations.

To simplify the determination of the contact area are encouraged to use the flat “platform” between EICT and ship’s hull. When using this “platform”, contact area with EICT and bottom of vessel is known and does not change when the vessel heeled.

These solutions help to simplify and expedite the inclining and weighing tests by using of EICT.

2. Description of new method

The proposed method is based on two solutions to simplify the calculation process. The main difference - longitudinal position of inflatable elastic tanks symmetrically relative to the center plane. (Fig. 4).

To facilitate and accelerate inclining and weighing test the longitudinal placement of inflatable tanks scheme (parallel center line) had proposed. The heeling of the vessel in use of this scheme is due to change the EICT pressure in the EICT located from one side of center line. The advantages of this scheme is that the use of cargo for inclining is not required, hence not require the deck strengths calculations.

To simplify the determination of the contact area are encouraged to use the flat “platform” between EICT and ship’s hull. When using this “platform”, contact area with EICT and bottom of vessel is known and does not change when the vessel heeled.

These solutions help to simplify and expedite the inclining and weighing tests by using of EICT.

EICT group of each side connected to compressed air systems and fitted with pressure gauge. Thus, by changing the pressure in tanks on one side, can change the position of the hull. Consequently, applying bank housing can be produced without the use of pressure roll changing ballast tanks on one side. (Fig.5).

The second solution is to use a rigid rectangular pad is positioned between the EICT and the vessel’s hull. (Fig. 6).

Use of this site makes known and constant contact area with the hull capacity and greatly simplifies the calculations.

Using this method can greatly simplify the calculations. In contrast to the long and complex calculations proposed in the first article, the new method of calculation reduces to determining the pressure in the cylinders, the calculation of the total contact area and the calculation of the angle of the hull.

Drafted by a mathematical model allow to calculate the weight of the vessel and the applicate of vessel center of gravity.

Vessel weight \( G \), equal to the sum of responses \( P_i \):

\[
G = \sum P_i
\]
Applicate the center of gravity is calculated by equating the moments of Equation 2.

\[-z_g \cdot \sin \theta + P_1 \cdot Y_{p1} - P_2 \cdot Y_{p2} = 0\]

\[-z_g = \frac{P_1 \cdot Y_{p1} - P_2 \cdot Y_{p2}}{G \cdot \sin \theta} \quad (1)\]

\[z_g = \frac{P_1 Y_{p1} - P_2 Y_{p2}}{(P_1 + P_2) \cdot \sin \theta \cdot \cos \theta} \quad \frac{2(P_1 Y_{p1} - P_2 Y_{p2})}{(P_1 + P_2) \cdot \sin \theta} \quad (2)\]

3. Conclusion

As seen from the calculations, the mathematical apparatus developed a new technique allows to save time and simplify the calculations. At the moment, to test the theory developed a model experiment.

When heeled vessel weight of the vessel is determined by the formula 1.

\[G = \Sigma P_1 \cdot \cos \theta \quad (1)\]

When heeled vessel heeling and righting moment are, respectively:

\[M_{kp} = -G \cdot z_g \cdot \sin \theta;\]
\[M_a = P_1 \cdot Y_{p1} - P_2 \cdot Y_{p2};\]
\[\Sigma M_i = 0;\]
\[M_{kp} + M_a = 0;\]