CONTENTS

TRANSPORT TECHNICS, INVESTIGATION OF ELEMENTS, RELIABILITY

DEVELOPMENT OF MODEL OF WORK BY MODULAR WHEELS FOR SELF-PROPELLED MACHINES

ON THE NEW DESIGN OF HELICOPTER’S MAIN REDUCER
Prof. Dr Eng. Abdullayev A.H., Prof. Dr. Eng. Najafov A.M., Conf Dr., Eng. Talibov A.R. ......................................................................................... 48

REDUCING THE DYNAMIC LOAD OF MECHANICAL WATER-JET DRIVE OF AMPHIBIOUS MACHINES
Prof. Dr. Eng. Taratorkin I., Prof. Dr. Eng. Derzhanskii V., PhD Taratorkin A., postgraduate Volkov A. ................................................................. 51

INFLUENCE OF ASYMMETRY OF SOLAR PANELS ON THE MICROACCELERATION LEVEL IN A SPACE LABORATORY ENVIRONMENT
Prof. Phd Sedelnikov A.V., Potienko K.I. ......................................................................................................................................................... 54

EXPERIMENTAL ANALYSIS OF BRAKE DISC COOLING CAPACITY
Bombek G., Lešnik L., Biluš I. ........................................................................................................................................................................... 58

TRANSPORT. SAFETY AND ECOLOGY. LOGISTICS AND MANAGEMENT

DEVELOPMENT OF INTERMODAL TRANSPORT AND LOGISTICS SCHEMES FOR PASSENGER TRANSPORTATION BY RAILWAY TRANSPORT WITHIN UKRAINE-EUROPE DIRECTION
Candidate of technical sciences, senior lecturer Primachenko A. A. ................................................................................................................. 61

TECHNICAL ASPECTS OF AUTOMATION OF TRANSPORT SECURITY
Prof. Dr. Yelisov L., Prof. Dr. Ovchenkov N. .................................................................................................................................................... 64

ASSESSMENT OF RISKS CAUSED BY RAILWAY CAR UNCOUPLING
Prof. PhD Garbaruk V.V., Prof. PhD Fomenko V.N. ........................................................................................................................................ 68

OPERATIONAL SAFETY OF WORK PLATFORMS
doc. Ing. Marianna Tomašková, PhD. ................................................................................................................................................................ 72

THE ANALITICAL RESEARCH OF THE PROCESS OF FORMING THE MOTOR-GRADER MOTION PATH AT IMPLEMENTATION OF WORKING OPERATIONS
Cand. Eng. Sc., Associate Professor Shevchenko V., Post-graduate student Chaplygina A., Junior Researcher, Beztsennaya Zh. ....................... 81

VEHICLE ENGINES. APPLICATION OF FUELS TYPES, EFFICIENCY

ECOLOGICAL CONTRIVANCE FOR ANALISIS AND MANAGING THE PRODUCTION OF BIOETHANOL AS A FUEL FOR INTERNAL COMBUSTION ENGINES
PhD Student Dzhelil Y. R., Ass. Nikolova D.St., Prof. Ivanov B. B. .................................................................................................................. 85

ENERGY FLOW STREAMS IN THE MARINE STEAM PLANT DURING THE MAIN PROPULSION PROPELLER SPEED VARIATION
PhD Student Eng. Igor Poljak, PhD. Vedran Mrzljak, PhD. Josip Orovic ........................................................................................................ 89
DEVELOPMENT OF MODEL OF WORK BY MODULAR WHEELS FOR SELF-PROPELLED MACHINES

Bekeno T.N., doctor of technical sciences (L.N.Gumilyov Eurasian National University, Astana); Nussupbek Z.T., candidate of technical sciences (Saken Seifullin Kazakh Agrotechnical University, Astana); Tassybekov Z.T., candidate of technical sciences (Nazarbayev University, Astana)

From the literary sources known the equation of the traction balance of vehicles, including cars, which is derived from the condition of the interaction of the entire machine with the road (base). The inertia force entering the traction balance equation is calculated through the coefficient of accounting for the rotating masses of the entire machine. In addition, in the literature describes the calculation of the interaction of single wheels by the road, depending on the nature and direction of the forces and moments acting on these wheels, that is, different modes of power loading of the wheels. At the same time, the connections of the front and rear wheels in the general regime of their force loading are not given and are not considered, respectively, the equations of the power balance of the wheels separately in cases of their uneven motion are absent. Therefore, the consideration of the individual processes of interaction of modular wheels with the road is necessary to establish the relationships between them, as well as with the machine as a whole. In particular, this concerns the redistribution of tractive forces on the wheels of machines in assessing their patency and efficiency.

To assess the traction and coupling properties of self-propelled wheeled all-wheel drive vehicles and justify the reasons for the redistribution of tractive forces on their wheels, it is necessary to choose the simplest design scheme, that is, their four wheel variant. In this case, the parameters of the wheels of machines, depending on the properties of the wheels themselves and the type of power transmission, will be considered, assuming that the support is solid. At the same time, the effect of transient processes is excluded from the analysis, since we will only consider the initial steps in the formation of parameters in conditions of acceleration of machines.

With the motion of full-drive vehicles with a blocked transmission even in a linear trajectory, at low speed there is a circulation of power in the transmissions due to the redistribution of traction forces on the wheels. The reason for such phenomena is applied to cars with a wheel formula 4x4 is that their front wheels are relative to the rear lagging (with the equality of weight conditions of bridges and air pressures in tires) when starting during acceleration /1, 2/. The torques on the wheels of the machine in the process of its starting (at constant air pressures in the tires of all wheels) will obey the condition:

$$M_{k2} > M_{k1},$$

$$M_{k1}, M_{k2}$$ – Torque respectively on the front and rear wheels, which is confirmed by experiments performed on the machine 4BC-10 /3/. At the same time, the pressure in the tires of the front wheels above the air pressure in the tires of the rear wheels ensures an increase in the rolling radius of the front wheels over the radius of the rear wheels /1, 2/, and the condition is fulfilled accordingly:

$$M_{k1} > M_{k2},$$

which is also confirmed in the work /3/.

The cars Ural 377 (6x4) and Ural 375 (6x6) also have the following reasons for the redistribution of traction forces regularities in the differences in the radius of movement of the wheels of various drive train /1, 2/. All of this also agrees with the experiments /4/.

Proceeding from the analysis it can be seen that in order to more fully determine the reasons for the redistribution of tractive forces between the wheels, it is necessary to choose the design scheme of a vehicle consisting of 4 driving wheels as the simplest in this respect and for this calculation scheme to develop a model of redistribution of traction forces on modular wheels its various axes.

Consider the interaction of the front drive wheel of the machine with the road (picture 1).

![Picture 1 – Interaction of the front drive wheel with the road](image)

According to the calculation scheme, we obtain equations /5, 6/:

$$X_1 = X_{u1} + P_{u1} = X_{u1} + jm_1,$$

$$z_i q + X_1 r + M_{al} = M_{r1},$$

$$X_1 = P_{r2} - P_{f1} - \frac{j}{r^2} (J_g i_{tp}^2 + J_1),$$

$$P_{u1} = jm_1, \quad \varepsilon = \frac{j}{r}, \quad M_{u1} = J_1 \varepsilon ,$$
\( X_1 \) – the tangential reaction of the road to the front wheels, H; \( X_{u1} \) – reaction of the rear of the car to the front wheels, H; \( z_1 \) – reaction of the road to the front wheels, H; \( M_{u1} \) – moment of inertia of the wheel, kNm; \( M_{T1} \) – traction moment of the front wheel, Hm; \( P_{u1} \) – inertial force of the forward moving wheel, H; \( J_M \) – moment of inertia of the motor rotor, kgm\(^2\); \( J_1 \) – moment of inertia of the front wheel, kgm\(^2\); \( P_{T1} \) – tractive power of the front wheel, Нм; \( \pi \) – inertial force of the forward moving wheel, Н; 

Solving equations (1) and (2) we obtain (5, 6, 7):

\[
X_{u1} = P_{T1} - z_1 f - \frac{j}{r^2} (J_M i_{TP}^2 \eta_{TP} + 2J_1) - jm_1 = P_{T1} - z_1 f - jm_1 \left(1 + \frac{(J_M i_{TP}^2 \eta_{TP} + 2J_1)}{m_1 r^2}\right),
\]

\[
X_1 = P_{T1} - P_{f1} \frac{j}{r^2} (J_M i_{TP}^2 \eta_{TP} + 2J_1),
\]

\[
\delta_{ap1} = 1 + \frac{(J_M i_{TP}^2 \eta_{TP} + 2J_1)}{m_1 r^2} \quad \text{attached to the front wheels coefficient accounting rotating masses; } P_{f1} \text{ - rolling resistance of front wheels, H; } G_3 \text{ - the weight falls on the front wheels, H.}
\]

Consider the interaction of the rear driving wheel of the machine with the road (picture 2).

\[ X_2 = X_u + P_{u2} = X_u + jm_2, \]

\[ z_2 a + X_2 r + M_{u2} = M_{T2}, \]

\[ X_2 = P_{T2} - z_2 f - \frac{j}{r^2} (J_M i_{TP}^2 \eta_{TP} + 2J_2), \]

\[ P_{u2} = jm_2 , \quad \epsilon = \frac{j}{r}, \quad M_{u2} = J_2 \epsilon , \]

\[ X_2 \] – the tangential reaction of the road to the rear wheels, H; \( X_{u2} \) – reaction of the front of the car to the rear wheels, H; \( z_2 \) – normal reaction of the road to the rear wheel, H; \( M_{u2} \) – inertia of the rear wheels, Hm; \( M_{T2} \) – traction moment on the rear wheels, Hm; \( P_{u2} \) – inertia force of the forward-moving rear wheels, H; \( P_{T2} \) – tractive power of the rear wheels, H; \( J_M \) – moment of inertia of the motor rotor, kgm\(^2\); \( J_2 \) – moment of inertia of the rear wheel, kgm\(^2\); \( m_2 \) – weight to the rear wheels, kg; \( j \) – acceleration, \( \text{m/s}^2 \); \( f \) – rolling resistance coefficient; \( \epsilon \) – angular acceleration; \( r \) – wheel radius, m.

Solving equations (6) and (7), we obtain (5, 6, 7):

\[ X_{u2} = P_{T2} - z_2 f - \frac{j}{r^2} (J_M i_{TP}^2 \eta_{TP} + 2J_2) - jm_2 = P_{T2} - z_2 f - jm_2 \left(1 + \frac{(J_M i_{TP}^2 \eta_{TP} + 2J_2)}{m_2 r^2}\right), \]

Picture 2 – The interaction of the rear driving wheel with the road.

According to the calculation scheme, we obtain equations:
\[ X_2 = P_{f2} - P_{f2} - \frac{j}{r^2}(J_i^2\eta + 2A) \]  

(10)

\[ \delta_{m2} = 1 + \frac{(J_i^2\eta + 2A)}{m_2r^2} \] - attached to the rear wheels coefficient accounting rotating masses; \( P_{f2} \) - The rolling resistance of the rear wheels, Н.

**LIST OF USED SOURCES**


7. Бекенов Т.Н., Нусупбек Ж.Т. Разработка модели расчета тяговой проходимости передних и задних приводных колес самоходных машин / В кн.: Теория и расчет тяговой и опорно-сцепной проходимости самоходных машин и их приложений (Бекенов Т.Н.).- М.:ООО «Буки Веди», 2012.- 112с.
ON THE NEW DESIGN OF HELICOPTER’S MAIN REDUCER
О НОВОМ КОНСТРУКТИВНОМ РЕШЕНИИ ГЛАВНОГО РЕДУКТОРА ВЕРТОЛЁТА

Abstract: A new design of Mi-8/Mi-17 helicopter’s main reducer with a higher technical level and built on the basis of the kinematic scheme of the three–stage double-flow reducer AN is being considered. At the same time the requirements for the reducer to be multi-flow and to have the required kinematic parameters at all of its output shafts for driving the main and tail rotors and all the auxiliary units are being observed.

KEYWORDS: MAIN REDUCER, KINEMATIC SCHEME, TECHNICAL LEVEL, NEW DESIGN

1. Introduction

Modern helicopters are mainly characterised by power and weight-dimensional parameters of their internal combustion engines and main reducer respectively. The weight of the transmission is about 10% of the total helicopter weight and the weight of the main reducer is – 75% of the total transmission weight [1]. Therefore any improvement in the kinematic scheme of the main reducer, i.e. its design, with increased efficiency and reliability level and decreased weight-dimensional parameters presents a very topical problem.

2. Objective and research methodology

In order to decrease the main reducer’s weight they are currently being designed on the basis of multi-flow schemes, i.e. the input torque is divided into several equal parts, which are transmitted in parallel and then summed up on the main rotor’s shaft. One of the main problems to be solved while designing a main reducer is to provide the most possible equal division of the torque into parallel flows [2].

Taking into consideration that the main reducer’s scantlings are restricted along the main rotor’s axis all the widely used earlier planetary kinematic schemes of the main reducer are lately used less and less often although being quite effective (compact set up in horizontal direction; all loads from gear wheels are applied to the internal wheel of internal engagement and by this taking the load from the reducer’s body) as they can not contribute any further towards decreasing of the reducer’s scantlings and subsequently its weight. The main drawback of planetary gears is the difficulty to provide an equal division of the load between satellites as the relative angular orientation of each satellite depends on a big number of randomly combined errors of the members of the planetary gear. Low-frequency vibrations from the main rotor especially at the last stage of reduction influencing inequality of load distribution between the satellites. As a result the coefficient of inequality of load distribution between the satellites at the last stage of the main reducer reaches the value of 1,35-1,4. This leads to gearings becoming heavier at the last stage of reduction as well as the main reducer itself. Apart from the above said planetary mechanisms quite labour intensive in production and assembly because of the certain technological difficulties. Therefore it is recommended to use multi-stage kinematic schemes with simple gearing and with division of the torque between the flows at the first stages and their subsequent merging at the last stage on a gear wheel with a big number of teeth. An example of such combined three-stage multi-flow kinematic scheme (simple and planetary gearing) of the reducer BP-8(A) / BP-14 is shown on the Fig. 1. and is currently being used in almost all modifications of the helicopter series Mi-8 / Mi-17.

The torque is transmitted to the main rotor through three stages of reduction. The power coming from both engines merging and then being transmitted by means of free-wheeling clutches at the first stage from the driving cylindrical single-helical cogs to the driven cylindrical single-helical cog. The second stage consists of two spiral bevel cogs and is used for transmitting the horizontal axis of rotation into the vertical one. The third stage represents a differential gear with three straight cylindrical cogs (all three cogs rotating) and the other three make power return differential gear. Thus the torque at this stage is transmitted to the main rotor in two ways: through differential gear and its power return gear [3, 4].

Classification and the main technical parameters of the helicopters series Mi-8 / Mi-17 are presented in the Table 1.
3. Problem solution

A new design of helicopter’s main reducer with a high technical level is being considered. At that three-stage double-flow reducer AN [5] is used as a basis for the kinematic scheme of the helicopter’s driving gear, which is presented later as a four-flow mechanical system consisting of the two embedded single-stage bevel gearing with a set of cylindrical and bevel wheels. The input shaft of the three-stage double-flow cylindrical reducer AN (Abdulhayev – Najafov) is connected on both sides (the first and the second flows) to the internal combustion engines located in the horizontal plane by means of bevel gearing.

Free-wheeling clutches are used for transmitting the rotation from the engines to the bevel gearing, i.e. helicopter’s main reducer. The transmission is transmitted with the required frequency (the third flow) to the intermediate and tail reducers by means of the sets of cog blocks consisting of cylindrical and bevel wheels, which are rigidly connected to the output axis.

Three-row cog blocks consisting of cylindrical and bevel wheels and non-ridge fixed to the counter shaft are used for transmitting rotation with the required frequency in the vertical plane to helicopter’s main rotor (the fourth flow). Bevel gearing embedded into the main part of the three-stage double-flow reducer AN can also be used for transmitting rotation to fans, hydro pumps, oil pumps, tachometer, generator and air compressor.

4. Results and problem discussion

An optimal distribution of the overall gear ratio between the stages and the approximate dimensions of the diameters of the bevel reducer’s and reducer AN’s input shafts as well as reducer AN’s counter and output shafts and the main rotor’s shaft with consideration of their torsion are presented in the Table 2.

<table>
<thead>
<tr>
<th>No.</th>
<th>Parameter Description</th>
<th>Reducer Type</th>
<th>BP-8(A)</th>
<th>BP-14</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Main reducer’s overall gear ratio</td>
<td></td>
<td>62,5</td>
<td>78,125</td>
</tr>
<tr>
<td>2</td>
<td>Bevel reducer’s gear ratio</td>
<td></td>
<td>3,97</td>
<td>3,9</td>
</tr>
<tr>
<td>3</td>
<td>First stage’s gear ratio of reducer AN</td>
<td></td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>4</td>
<td>Second stage’s gear ratio of reducer AN</td>
<td></td>
<td>4,5</td>
<td>5</td>
</tr>
<tr>
<td>5</td>
<td>Third stage’s gear ratio of reducer AN</td>
<td></td>
<td>3,5</td>
<td>4</td>
</tr>
<tr>
<td>6</td>
<td>Gear ratio of bevel gearing to main rotor</td>
<td></td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>7</td>
<td>Gear ratio of bevel gearing to tail rotor</td>
<td></td>
<td>0,297</td>
<td>0,333</td>
</tr>
<tr>
<td>8</td>
<td>Gas turbine turbo shaft engine power, kW</td>
<td></td>
<td>1102,94</td>
<td>1617,65</td>
</tr>
<tr>
<td>9</td>
<td>Bevel reducer’s input shaft torque, Nm</td>
<td></td>
<td>877,76</td>
<td>1030</td>
</tr>
<tr>
<td>10</td>
<td>Diameter of bevel reducer’s input shaft, Mm</td>
<td></td>
<td>44,44</td>
<td>46,88</td>
</tr>
<tr>
<td>11</td>
<td>Reducer AN’s input shaft torque, Nm</td>
<td></td>
<td>3355</td>
<td>3775</td>
</tr>
<tr>
<td>12</td>
<td>Diameter of reducer AN’s input shaft, Mm</td>
<td></td>
<td>69,48</td>
<td>73,03</td>
</tr>
<tr>
<td>13</td>
<td>Reducer AN’s counter shaft torque</td>
<td></td>
<td>3287,9</td>
<td>3699,5</td>
</tr>
<tr>
<td>14</td>
<td>Diameter of reducer AN’s counter shaft, Mm</td>
<td></td>
<td>69,02</td>
<td>71,79</td>
</tr>
<tr>
<td>15</td>
<td>Main rotor’s shaft torque, Nm</td>
<td></td>
<td>50601</td>
<td>74222</td>
</tr>
<tr>
<td>16</td>
<td>Calculated minimal diameter of main rotor’s shaft, mm</td>
<td></td>
<td>171,18</td>
<td>193,38</td>
</tr>
</tbody>
</table>

At that the following mandatory conditions and requirements for designing a new scheme of helicopter Mi-8 / Mi-17’s main reducer BP-8 (A) or BP-14 are observed:
- provision of low vibration activity for maximum reduction of vibration, which is transmitted from the main rotor to the last stage of the main reducer;
- equal distribution of the load between the parallel members of multi-flow gears;
- elimination of load concentration along the length and height of gear teeth;
- direction of rotation of gas turbine turbo shaft engine (TSE) (anticlockwise when looking from the main reducer towards the engine), which is connected to the main reducer’s input shaft by means of free-wheeling clutch (FWC);
- direction of rotation of the main rotor (clockwise when looking at the rotor from top);
- direction of rotation of the tail rotor (clockwise when looking at the rotor);
- availability of enough number of the main reducer’s output shafts (flows) for transmitting rotation with the required frequency to the auxiliary units (fans, generator, air pump, tachometer, oil pump and air compressor);
- roller bearings’ reliability level ≥0,98 [6, 7].

Taking into account all the above said the algorithm of designing of a new scheme of helicopter Mi-8 / Mi-17’s main reducer can be presented as:
1. Calculation of the axle base distance value of reducer AN is done based on the solution of the contact problem for cylindrical wheels of the most heavy loaded stage of the reducer.

2. Calculation of the pitch circle diameter of the cogs and wheels for each stage based on the determined value of the axle base distance for the most heavy loaded stage and distribution of the overall gear ratio between the stages of reducer AN.

3. Determination of the number of teeth and their modulus based on the calculations for cylindrical gear wheels’ bending resistance with consideration of the wheel material’s mechanical characteristics.

4. After a generalised parameter of the helicopter’s main reducer selected all the main geometrical parameters are calculated based on the generalised coordinate and a block-scheme of automated design of the suggested mechanical system is developed.

5. Conclusion

A new design of the main reducer’s kinematic scheme can provide the required gear ratios for all four flows according to the required technical parameters of helicopter’s transmission. At this mechanical system’s reliability level and efficiency increasing and weight-dimensional parameters decreasing, i.e. raising the system’s technical level and this has an important practical value at design stage of new schemes of helicopter’s main reducer.

The developed algorithm of designing of new schemes of helicopter Mi-8 / Mi-17’s main reducer with consideration of the durability and rigidity characteristics has a very important theoretical value.

As an application for patent has been submitted and due to the requirement to observe the main principle of non-disclosure the kinematic scheme and a more detailed description of the suggested helicopter’s main reducer will be presented during reporting at the conference.

6. Literature


2. Unit drives, reducers, clutches of Gas Turbine Engines. Reducers – Chapter 10 (10.2.2.1 – Kinematic schemes of helicopter’s main reducers), pp. 17 – 19.


REDUCING THE DYNAMIC LOAD OF MECHANICAL WATER-JET DRIVE OF AMPHIBIOUS MACHINES

СНИЖЕНИЕ ДИНАМИЧЕСКОЙ НАГРУЖЕННОСТИ ПРИВОДА ВОДОМЕТНОГО ДВИЖИТЕЛЯ АМФИБИЙНОЙ МАШИНЫ

Prof. Dr. Eng. Taratorkin I.1, Prof. Dr. Eng. Derzhanskii V.1, PhD Taratorkin A.1, postgraduate Volkov A.1 – Institute of Engineering Science of the Ural Branch of the Russian Academy of Sciences (IES UB RAS), Russia
Corresponding author - Igor Taratorkin
E-mail: ig_tar@mail.ru

Abstract: The article provides an analysis of the excitation conditions and the results of the detuning of parametric resonance in mechanical drive amphibious water-jet machines. The paper substantiates the possibility of solving the problem of reducing the dynamic load on the basis of the synthesis filter low-frequency vibrations.

Index words: dynamic loading, oscillations, analysis, transmission, water-jet, frequency.

Introduction

Transport vehicle power plant durability is limited largely to fatigue failure of the components due to the resonant mode excitation at a deficient level of vibration protection.

In machines, the method for resonance elimination is widely applied by reducing transmission natural frequency due to reduction in spring and friction damper rigidity [1,2]. However, the possibilities of rigidity reduction are restricted because angular compliance does not exceed 2 … 3 degrees in case of the required 8 … 12. When using springs of the Swedish company "Oteva", the compliance reaches 5 … 8 degrees [3]. The value of the required compliance can be reached when using, as a damper, a Centa elastic coupling (with angular compliance up to 15 degrees) [4]. However, the possibility of wide application of such couplings is defined by durability limited by biological aging of elastic details, by stability of their properties in the wide range of temperatures.

In mechanical engineering, while solving engineering problems, the method of eliminating resonant modes stated in A.N. Grishkevich's reference book [5] is widely known and applied. This method deals with performing the following operations:

1) Determination of transmission natural frequencies according to rigid body drawings of the components.
2) Computation of the function for the polyharmonic excitation of the engine torque according to the indicator diagram of a single cylinder, considering an operating procedure and construction features, based on spectrum analysis of the acquired function for determination of major harmonics.
3) Creation of the superimposed frequency response characteristic of the engine and transmission. Forecast of the resonant mode according to the crosspoints of lines of the system natural frequencies and engine motor harmonicas, as well as determination of a corresponding range of engine shaft speeds.
4) Definition of the acceptable speed range of engine shaft speeds, beyond which the resonance should be removed.
5) When forecasting the resonant mode, it is necessary to calculate required parameters of elastic and dissipative characteristics (rigidity, friction torque, preloading forces) eliminating the resonant mode; and to build an elastic and dissipative characteristic. Depending on required angular rigidity, it is necessary to choose a damper type (spring and friction, torsion, in the form of an elastic coupling) which is to be established on the flywheel of the heat engine. However, the efficiency of a damper synthesized on the known method can be sufficient only to eliminate some local resonance, for example, in pretorque converter zone, but at the same time, the conditions of exciting resonances at other frequencies remain.

The results of research

The object of the experimental research is dynamic loading of the transport vehicle transmission ramified dynamic system. According to the modal analysis of the mechanical system, which turns on the YaMZ-780 engine, the GTK-430 hydraulic torque converter, inertial mass of a divider of the transmission gear, inertial masses and compliances of drive components of support equipment, for example, of the track water propulsion unit of the amphibious vehicle, the resonant modes according to the ramified frequency characteristic (Fig. 1, Table 1) caused by the main motor harmonics of the engine (3,6,9,12) can arise at all natural frequencies of the system.

Table 1. Natural frequencies of the dynamic system
According to the kinematic scheme, the dynamic model of the system (Fig. 2) is developed, the elastic and inertial parameters of which are determined according to the drawing technical documentation.

While creating this model, the possibility of regulating the frequency of engine shaft rotation within the working range is provided. The results of the harmonic analysis of the engine torque, considering gas and inertial forces, are given in Fig. 3, from which it follows that the main harmonics forming excitation in the dynamic system are the third and sixth.

The results of modeling the dynamics of the system, executed in LMS AMESim software package [6] are shown in Fig. 4. Analysis of the results shows that all the inertial mass of the system are loaded with essential dynamic torque. As an example, analyzes the change of the dynamic torque is given to the mass Ja during acceleration from minimum to maximum sustainable revolution. The graph shows the 3D-spectrum of amplitudes and frequencies of the engine torque, with dependence on its rotation. The maximum values of torque amplitudes up to 3 kNm are fixed at frequencies of 25, 40 and 76 Hz and are excited by the third motor harmonica. The entire population of high value of amplitudes and recurrence of loading the water-jet drive components restricts their durability.

To reduce dynamic loading of the drive, the option of the low frequency oscillation filter design and optimization of its parameters are worked over. Filter parameters are selected from a condition of reducing a variable component of an engine torque, providing a dynamic response factor at the transmission inlet to be no more than $K_d=0.2$. At the same time, the given dynamic response factor is provided with a necessary value of a detuning factor [7]. The results of modelling the system dynamics with the optimized low frequency filter are given in Fig. 5. It follows from the calculated data that the filter provides a considerable (up to 3 times) reduction of torque amplitudes in the water-jet drive.

### Table

<table>
<thead>
<tr>
<th>Motion mode</th>
<th>Natural frequencies, Hz</th>
</tr>
</thead>
<tbody>
<tr>
<td>D</td>
<td>25 34 40 76 109 173 263 337 475</td>
</tr>
<tr>
<td>R</td>
<td>27 33 39 81 100 173 263 337 475</td>
</tr>
</tbody>
</table>

Fig. 2 – Dynamic model of the system "Engine – HTC – Divider - Water-jet Drive"

Fig. 3 Results of the harmonic analysis of the engine torque.
Limitations

The results of the research are received for the linear system though existence of shaft drives in the driving gear and gear backlashes leads to emergence of nonlinear dynamic effects, such as parametrical oscillations, sub- and super-harmonic oscillations. Research of nonlinear system dynamics can be an object of follow-up studies.

Conclusion

As a result of the research, the following has been established.

1) The developed computational scheme and mathematical model of the system "Engine – HTC – Divider - Water-jet Drive", the results of modeling the polyharmonic excitation of the engine and defining modal characteristics of the system show that dynamic loading in many respects is defined by a condition of a resonance excitation.

2) Novelty of the research findings is in using a modern software package [*] for detailed research of harmonic components of the internal combustion engine torque, which are the source of oscillation system excitation.

3) Introduction of a low-frequency filter to the system provides three-fold decrease of the magnitude of the dynamic torque that creates opportunities for ensuring the required durability of drive components.

References

6) LMS Imagine.Lab AMESim Training – HYD1, 2013. – 50 p
INFLUENCE OF ASYMMETRY OF SOLAR PANELS ON THE MICROACCELERATION LEVEL IN A SPACE LABORATORY ENVIRONMENT

ВЛИЯНИЕ АСИММЕТРИИ ПАНЕЛЕЙ СОЛНЕЧНЫХ БАТАРЕЙ НА УРОВЕНЬ МИКРОУСКОРЕНИЙ ВНУТРЕННЕЙ СРЕДЫ КОСМИЧЕСКОЙ ЛАБОРАТОРИИ

Prof. Phd Sedelnikov A.V.1, Potienko K.I.2
Department of Space Engineering, Samara University – Samara, Russian Federation, axe_backdraft@inbox.ru 1
Faculty of Electronics and Instrument Engineering, Institute of IT, Mathematics and Electronics, Samara University – Samara, Russian Federation, potienko97@mail.ru 2

Abstract: There are viewed the following types of asymmetry: mass, linear and asymmetry of fixing points of big elastic elements to the spacecraft’s body. There is shown increase of the microacceleration module while asymmetry of the mentioned types appears. The conclusions about importance of geometrical symmetry of space laboratory for successful realization of gravity-sensitive technological processes on its board were made.

Keywords: MICROACCELERATION FIELD, SOLAR PANELS, SPACE LABORATORY, GRAVITY-SENSITIVE PROCESSES, ASYMMETRY

1. Introduction

Modern requirements for design of new space technic, which is intended for realization of gravity-sensitive processes in space, such as growing of monocrystals, foamed metal, creating of specific medicines etc., presuppose serious limits of microacceleration level in area with technological equipment. It is very important to provide near zero gravity conditions into the inner environment of space laboratories for successful gravity-sensitive experiments and technological processes. The features of some of them are described in [1-4]. The maximum permissible microacceleration level in the facilities hosting technological equipment is a requirement provided in the preliminary technical design for the development of the space laboratory. This level is one of the most important operational and technical features of the future laboratory [5]. For instance, during the design study program of the “NIKA-T” spacecraft a microacceleration level less than 20 µm/s² was prescribed [6], while for the perspective spacecraft “OKA-T” the requirement was less than 10 µm/s² [7]. These requirements will become increasingly stringent with the evolution of space technologies [8].

One of the important factors to meet microgravity acceleration requirements is to avoid asymmetries of solar panels of the space laboratory during the developmental stage. This article is devoted to the analysis of consequences of possible asymmetries of solar panels in three forms:

- mass asymmetry, caused by differences in mass of solar panels;
- linear asymmetry, due to different lengths of solar panels;
- asymmetry of connection points of solar panels to spacecraft’s body (connection points are not aligned with the mass center of the spacecraft).

It is known that orbiting space laboratories have been launched into space by Russia and China. Russian spacecraft from “BION” family provide a platform for conducting fundamental space biology or bio-medical experiments [9, 10]. The “FOTON” family program is dedicated to physical science and technological experiments [8, 11]. Chinese spacecraft from SJ family are used as a platform for conducting gravity-sensitive processes in space [3]. Spacecraft «FOTON – М» № 4 launched on 19th of July 2014 [11] was applied as a prototype of space laboratory for the research.

2. Source data for modelling

The spacecraft model is represented by a perfectly rigid central body with two elastic elements rigidly fixed to it (figure 1).

Elastic elements are represented by Euler–Bernoulli beam. Studies performed according to such simplified model show that a rigid connecting lug of the elastic element to the spacecraft’s body gives some overestimation because some energy of elastic elements’ oscillations would be dispersed by elastic attaching lug [12]. It is proved that a beam-model of elastic elements also gives rather inflated estimate in comparison with the model of homogeneous plate [13]. Therefore, application of the above mentioned model of the spacecraft provides some overestimation resulting in some safety margin for microacceleration disturbances that are very undesirable for certain gravity-sensitive processes on board the space laboratory. Data corresponding to the reference spacecraft, chosen for modelling, are shown in table 1.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Notation</th>
<th>Value</th>
<th>Dimension</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mass</td>
<td>m</td>
<td>6535</td>
<td>kg</td>
</tr>
<tr>
<td>Centroidal moment of inertia of spacecraft</td>
<td>l₁, l₂, l₃</td>
<td>10000; 11000; 10000</td>
<td>kg·m²</td>
</tr>
<tr>
<td>Length of elastic elements</td>
<td>l₁, l₂</td>
<td>4; 4</td>
<td>m</td>
</tr>
<tr>
<td>Thickness of elastic elements</td>
<td>h₁, h₂</td>
<td>0,003; 0,003</td>
<td>m</td>
</tr>
</tbody>
</table>

Fig. 1 Model of spacecraft
3. Equations for modelling and calculation results

The following coordinate systems have been used for derivation of the equations of the mathematical model (1): OXYZ – the main coordinate system; A XA γA ZA – local coordinate system, centered at the fixing point of elastic element to spacecraft’s body; Ci xi yi zi – local coordinate system, centered at the mass center of i-section of elastic element.

The equations of the mathematical model of the spacecraft dynamics around its mass center are shown below:

\[
\begin{align*}
\dot{\omega}_i &= \sum_{m=1}^{N} \left[ \frac{\sum_{i=1}^{N} \left( \omega_{ixi} + \omega_{iyi} \right) \left( x_i x_i - x_i y_i \right) \left( y_i x_i - y_i y_i \right) \left( x_i - y_i \right)}{M_{z_i}} - \omega_{iz} \right] - \mathbf{I}_z \dot{r}_c; \\
\dot{\omega}_r &= \sum_{m=1}^{N} \left[ \frac{\sum_{i=1}^{N} \left( \omega_{ixi} + \omega_{iyi} \right) \left( x_i x_i - x_i y_i \right) \left( y_i x_i - y_i y_i \right) \left( x_i - y_i \right) \left( y_i - x_i \right) \left( x_i - y_i \right)}{M_{z_i}} - \omega_{iz} \right] - \mathbf{I}_z \dot{r}_c; \\
\dot{\omega}_t &= \sum_{m=1}^{N} \left[ \frac{\sum_{i=1}^{N} \left( \omega_{ixi} + \omega_{iyi} \right) \left( x_i x_i - x_i y_i \right) \left( y_i x_i - y_i y_i \right) \left( x_i - y_i \right) \left( y_i - x_i \right) \left( y_i - x_i \right) \left( x_i - y_i \right)}{M_{z_i}} - \omega_{iz} \right] - \mathbf{I}_z \dot{r}_c.
\end{align*}
\]

where \( \mathbf{I}_z \) is the inertia tensor of i-section of elastic element in the coordinate system located at the mass center of the section with axes parallel to the axes of the main coordinate system (figure 1); \( \mathbf{O}_i (\mathbf{O}_{ixi}, \mathbf{O}_{iyi}, \mathbf{O}_{iz}) \) is the angular velocity vector (relative to the central body) in the main coordinate system (figure 1); \( \mathbf{O}_i \) is absolute angular velocity vector of the central body in the inertial coordinate system; \( m_j \) is the mass of i-section of elastic element; \( \rho_i \left( x_i, y_i, z_i \right) \) is the vector radius of mass center of i-section of elastic element in the main coordinate system (figure 1); \( \rho_i \left( x_0, y_0, z_0 \right) \) the vector radius of the mass center of the spacecraft in the main coordinate system (figure 1); \( N \) is the number of the elastic elements of the spacecraft; \( \mathbf{M}(M_x, M_y, M_z) \) is the moment vector provided by the attitude control thrusters used to keep the spacecraft at the desired orientation.

In this model, as opposed to the article [6], elastic elements are represented by a finite number of separated rigid elements. Such approach allows observing of the system’s motion as a whole and not to divide it into rigid and elastic parts.

Components of the angular velocity vector of the spacecraft are derived from (1) in an explicit form:

\[
\begin{align*}
\omega_x &= \frac{-\sum_{m=1}^{N} \left( \sum_{i=1}^{N} \left( \omega_{ixi} + \omega_{iyi} \right) \left( x_i x_i - x_i y_i \right) \left( y_i x_i - y_i y_i \right) \left( x_i - y_i \right) \left( x_i - y_i \right) \left( y_i - x_i \right) \left( y_i - x_i \right) \right)}{M_{z_i}} \dot{r}_c; \\
\omega_y &= \frac{-\sum_{m=1}^{N} \left( \sum_{i=1}^{N} \left( \omega_{ixi} + \omega_{iyi} \right) \left( x_i x_i - x_i y_i \right) \left( y_i x_i - y_i y_i \right) \left( x_i - y_i \right) \left( y_i - x_i \right) \left( x_i - y_i \right) \left( y_i - x_i \right) \right)}{M_{z_i}} \dot{r}_c; \\
\omega_z &= \frac{-\sum_{m=1}^{N} \left( \sum_{i=1}^{N} \left( \omega_{ixi} + \omega_{iyi} \right) \left( x_i x_i - x_i y_i \right) \left( y_i x_i - y_i y_i \right) \left( x_i - y_i \right) \left( y_i - x_i \right) \left( y_i - x_i \right) \left( x_i - y_i \right) \right)}{M_{z_i}} \dot{r}_c.
\end{align*}
\]
elastic element attached in point B from the value mentioned in table 1.

As shown in figure 2, significant mass asymmetry (Δm = 4 kg) can increase the amplitude of the microacceleration disturbances more than three times; the waiting time for favorable conditions corresponding to relatively small microaccelerations compatible with gravity-sensitive processes is increased to almost 20%.

The linear asymmetry is discussed below. In what follows the masses of both panels are the same and equal to the nominal value (reported in table 1), and points A and B are symmetric relative to the center of mass of the spacecraft. The length of the elastic element attached to the point A is also equal to the nominal value in table 1. The length of the elastic element attached to the point B is reduced from the nominal value in Δl. Such asymmetry will also lead to asynchrony of vibrations of elastic element resulting in additional microaccelerations related to the displacement of the spacecraft center of mass. The quantitative impact of the linear asymmetry on the microacceleration level has been identified. Time dependences of microaccelerations, analogous to those shown in figure 2, demonstrate that disturbances, in the presence of linear asymmetry of elastic elements, are higher than in the case of symmetric spacecraft. The same above mentioned parameters considered to investigate the influence of mass asymmetry have been evaluated.

Figure 3 shows the behavior of the relative times and relative microacceleration amplitudes versus the value of linear asymmetry.

4. Results and discussion

In summary, all the discussed results show that almost each asymmetry is a source of acceleration disturbances in the microgravity environment. This can be explained by the constant position of the mass center of the spacecraft when elastic elements and their oscillations are symmetric, because displacements of separated parts along axes of the main coordinate system have equal magnitudes and opposite directions. Microaccelerations are caused by inertia moments excited by displacements of separated parts of elastic elements synchronously clockwise or anticlockwise. Appearance of mass asymmetry caused by reduced mass of one elastic element, in the present case, excites the displacement of the center of mass of the spacecraft while solar panels oscillate. The resulting microaccelerations are higher than the corresponding reduction caused by rotational motion. This reduction is conditioned by decrease of inertia moment of elastic elements while mass of one of them is reduced. An increase of mass of elastic element causes even larger microaccelerations caused by the combination of the rotational contribution (increase of inertia moment of elastic elements) with the displacement of the mass center of spacecraft.

A similar situation occurs when linear asymmetry appears. The length of an elastic element was reduced providing reduction of microaccelerations caused by rotational movement of the spacecraft. However, the overall microaccelerational level increases because of the displacement of the center of mass. Increase of the length of elastic element makes this growth even more significant.

5. Conclusion

The study provides the following conclusions.

1. The level of microaccelerations is underestimated when they are evaluated based only on the platform characteristics assuming symmetrical oscillations of symmetric elastic elements, because only evolution of the spacecraft around the mass center is taken into account. Displacements of the center of mass, caused by fluctuations of asymmetric elastic elements, give rise to additional constructive microacceleration. Despite the fact that the model (rigid attachment of elastic elements to the spacecraft body the beam-models of elastic elements), as shown in [13], overestimates the microaccelerational level, it is shown that neglecting significant asymmetries the microgravity disturbances can be underestimated leading to incorrect decisions about the feasibility of gravity-sensitive processes aboard the space laboratory.

2. Mass and linear asymmetry negatively influence the microgravity environment, especially in cases when the length or weight of the elastic element is greater than its designed value. Therefore, in the development stage of a space laboratory it is necessary to minimize these types of asymmetries.

3. The mathematical model used in this research allows a qualitative estimation of the impact of the considered types of asymmetries on the microaccelerational level. For reliable quantitative estimations a significantly more complicate model of the spacecraft must be developed. However, even a qualitative analysis is very useful to design a new generation of space laboratories.

4. The effects of the asymmetry of solar panels and, consequently, the microgravity disturbances can be reduced with ad hoc spacecraft asymmetries, for instance the center of mass of the spacecraft should be removed closely to heavier or longer panel. The effect of such asymmetry is opposite to the influence of mass or linear asymmetries of solar panels.
References


7. A.I. Belousov, A.V. Sedelnikov, Problems in formation and control of a required microacceleration level at spacecraft design, tests, and operation, Russian Aeronautics, 57 (2), 2014, 111–117.


EXPERIMENTAL ANALYSIS OF BRAKE DISC COOLING CAPACITY
Bombek G.1, Lešnik L.1, Biluš I.1
Faculty of Mechanical Engineering – University of Maribor, Slovenia1
ignacijo.bilus@um.si

Abstract: The car brake system performance has to be reliable at wide range of operating conditions. The friction based braking system reliability strongly depends on cooling capability of disc. According to this, different testing is needed to assure the performance of discs during their design and prototyping phase. The contribution presents the study of cooling capability of different vented brake disc geometries.

Brake disc temperatures during thermal capacity test can exceed 600°C. The temperatures are highly dependent on disc cooling capability. Vented disc have higher cooling capability. There is an open issue which factor has the most significant impact. Is it air flow, air flow distribution, turbulence or ...? Brake discs with the same external geometry but different internal geometry were tested on brake dynamometer and subsequently air flow parameters were analysed in cold condition to attempt to explain different maximum temperatures.

Keywords: brake disc, cooling, flow measurements

1. Introduction
Braking performance is crucial for the safety of the vehicle and its occupants. Braking performance can be influenced by wear, maintenance and braking history. Most vehicles use cast iron discs or drums. Brake drums are used in smaller cars and mostly for rear brakes since the majority of brake force is distributed on front axle (60-70%). The most important task of the brake system is to convert kinetic energy into heat. Braking performance is brake disc temperature dependent. Disc temperature rises with every braking and the braking performance decreases with the temperature. Brake disc is mostly cooled by air, although some heat is transmitted via radiation too. The cooling efficiency was studied by prof. Limpert (Limpert, 1975) and vented disc were proposed and a whole new area of study was opened dedicated to the question which remains actual: Which internal geometry of the disc is the best? Is a straight vanes design, curved vanes design or fins design? Beside the braking efficiency and temperature there are still other requirements to be fulfilled. Minimizing the disc mass, moment of inertia and increasing of resonant frequencies are also important but the production costs have very high priority when design is considered.

2. Problem
Two prototypes with aerofoil shaped vanes (B and C) were produced to find possible replacement for existing disc design with straight vanes (A). The discs were used on a B class car and are presented in Figure 1.

The results on dynamometer test were below expectations and the temperature after thermal capacity test (SAE J2522) was higher than in the case of the original design in case C and the same in case B. A more detailed analysis was performed and some of the results will be presented and discussed.

External disc geometry was the same for all three cases. The distance between the friction plates was 7 mm in type A, 8 mm in type B and 6 mm in type C. The number of vanes was 41 in case of type B and C and 53 in case of type B. The inner diameter of the friction surface was 145 mm for discs A and B and 154 mm for disc C. Vane length in case of type C disc was 4-5 mm shorter.

3. Air flow measurements
The first idea that comes to mind is to measure the air flow through the disc. The disc was placed on a rotating device driven by frequency controlled electro-motor. Most rotating device parts were the same as in appropriate car, although some modifications to the axle were made and the brake pads and the jaws were removed. It is quite complicated to measure the air flow at the disc exit. Air exit velocity in case of radial fan with straight vanes can be approximated by circumferential velocity of the vane tip which was approx. 10 m/s. Low velocities are difficult to measure with traditional methods like Pitot tube due to very low pressure differences. Additionally, exit velocity profile is not constant due to finite number of vanes and their thickness. Most of the studies were performed at 15.7 Hz rotational speed (corresponding to 100 km/h vehicle speed). When multiplying rotational frequency with the number of vanes frequency above 600 Hz can be expected. This frequency makes Pitot based measurements practically impossible. There are some reports of HWA measurements but our first idea was to simplify the mass flow measurements and the disc orientation was changed (friction surface faced outwards instead inwards). This enabled free access to the intake and enabled measurements of the intake velocity profile. Since the disc had no cross drilled holes the conservation of mass was assumed. Anemometer Testo 435-4 was placed on a traverse system and velocity profile in horizontal and vertical plane measured. Program controlling anemometer and traverse system was written in LabVIEW. The measurement results for disc type A are presented in Figure 2.

![Fig. 1 Different shapes of disc vanes.](image)

![Fig. 2 Intake velocity profile type A disc at 100 km/h.](image)
The procedure was repeated for several rotational speeds and the results for original disc (type A) are presented in Figure 5.

Velocity profile was used to calculate volumetric flow. Results for volumetric flow are presented and compared in Figure 6.

At least 240 images were averaged and radial velocity component at desired radius extracted. The procedure was repeated in several layers beginning at outward friction surface and ending at inward friction surface. The layers between friction plates were placed in 1mm interval and two layers were located in the middle of the friction plate. Radial velocity profiles are presented in Figure 9.

By comparing results presented in Figure 9 it is possible to observe different radial velocity distribution. The distribution in type A disc is along the vane, while in case of type B and type C it is concentrated in the middle of the channel. This fact can offer possible explanation for lower cooling capacity of type B and C discs. It is desirable to have cooling flow near hot surfaces. The hottest surfaces are friction planes and vanes.

4. Validation and comparison

Radial velocity field was used to calculate volumetric flow. It was assumed that there is no significant difference between the vanes and that 3 or 4 vanes covered by PIV analysis are representative. All internal surfaces are un-machined and sand-cast.
The results for volumetric flow at 100 km/h are presented in Table 1.

**Table 1: Comparison of the results.**

<table>
<thead>
<tr>
<th></th>
<th>vol. flow A (m³/s)</th>
<th>vol. flow B (m³/s)</th>
<th>vol. flow C (m³/s)</th>
</tr>
</thead>
<tbody>
<tr>
<td>PIV</td>
<td>0.025268</td>
<td>0.026803</td>
<td>0.019168</td>
</tr>
<tr>
<td>anemometer</td>
<td>0.024063</td>
<td>0.026267</td>
<td>0.020346</td>
</tr>
</tbody>
</table>

As presented in Table 2 quite good agreement between the measurement methods was achieved for internal air flow which contributes the major part in convective cooling of the brake disc.

5. Influence of external disc diameter

We were fortunate to have at our disposal an enlarged version of the original disc A. The number and shape of the vanes is the same. The main difference is increase of external diameter of the disc by 22 mm. The comparison of mass flow is presented in Figure 10.

![Fig. 10 Comparison of volumetric flow type A and A+](image1)

As presented in Figure 10 there is a less than 3% increase in air flow although the diameter increase is more than 8%. The vane length was increased by approx. 11 mm. The distance between the friction plates was 7 mm (the same as in case A) although casting tolerances must be considered. Simple expansion of the disc does not increase its cooling capacity much.

6. Influence of the internal vane diameter

A very similar disc (according to A+) was studied too. This disc had the same external diameter but only 40 vanes and the internal diameter of the friction plate 154 mm (like disc C). The air flow measurements are presented in Figure 11.

![Fig. 11 Comparison of volumetric flow type A+ and A++](image2)

As presented in Figure 11 internal vane diameter has significant influence on air flow. The difference between the numbers of vanes (41 and 40) was neglected. Vane length was reduced by approx. 5 mm (according to A+). External vane diameter was the same. The distance between the friction surfaces was again 7 mm.

7. Discussion

Air volumetric flow and radial exit velocity distribution were acquired during an attempt to explain unexpected results during thermal capacity test. Volumetric air flow is an important factor, but velocity distribution must not be neglected. It is convenient to have high velocity near hot surfaces even if it may cause lower cumulative flow. The radius between the vanes and the friction plates should be minimal since larger radius causes air flow concentration in the middle of the channel which is not desirable from the cooling point of view.

External vane diameter has lower influence on mass flow than internal diameter. The cooling air channel is defined by friction plates and two consecutive vanes. Since all vanes are oriented to the axis of rotation the channel has the bottleneck at vane internal diameter. Air flow is limited by the smallest distance between the vanes. Longer vanes (at limited external diameter) may increase pressure difference but they reduce intake area which at least in our case showed to be more influential.

Linear dependence between air flow and vehicle speed was observed in all presented cases. This fact is a bit surprising since quadratic dependence is expected in closed channels. Possible explanation could be existence of the recirculation area on the suction side of the vane. PIV visualization of recirculation area is presented in Figure 12.

![Fig. 12 PIV image of recirculation area in the inter vane channel](image3)

White (fog) area in figure 12 represents recirculation area. Disc was rotated clockwise. Recirculation area reduces the mass flow and heat transfer on the suction side of the vanes due to the lower air velocity and temperature difference.

It might be beneficial to modify the disc design to reduce recirculation area and maybe achieve better cooling capacity with lower air flow and consequently reduce acoustic emissions and deposition of moisture (corrosion) in recirculation areas.

5. References

DEVELOPMENT OF INTERMODAL TRANSPORT AND LOGISTICS SCHEMES FOR PASSENGER TRANSPORTATION BY RAILWAY TRANSPORT WITHIN UKRAINE-EUROPE DIRECTION

РАЗРАБОТКА ИНТЕРМОДАЛЬНЫХ ТРАНСПОРТНО-ЛОГИСТИЧЕСКИХ СХЕМ ДОСТАВКИ ПАССАЖИРОВ ЖЕЛЕЗНОДОРОЖНЫМ ТРАНСПОРТОМ В НАПРАВЛЕНИИ УКРАИНА – ЕВРОПА

Candidate of technical sciences, senior lecturer Primachenko A. A.¹
Faculty of Management processes traffic ¹ – Ukrainian State University of Railway Transport, Kharkov, Ukraine, e-mail: AnnaPrimachenko@i.ua

Abstract: The main issues are considered in this topic: the new logistic schemes for passenger transportation between Europe and Ukraine; the developing an integrated short-term planning system for the purpose of interaction between passengers and carriers; the intermodal transportation of passengers.

KEYWORDS: INTERMODAL PASSENGER TRANSPORTATION, RAILWAY TRANSPORT, PASSENGER LOGISTICS, PASSENGER TRAFFIC.

1. Introduction

The logistics of passenger railway transportations is a complex and interconnected solution of problems in organization of passenger transportation. The objective of logistics management of passenger rail transportation system is a transfer of a passenger from the station of departure to the station of destination ensuring the optimization of the following criteria: effective use of rolling stock and rail infrastructure, minimal costs, optimal route, traffic schedule and the level of transportation quality. A modern passenger became more demanding – the requirements to the quality of transportation, comfort and convenience, the interaction of different transport routes and directions, the quality and range of associated services, the way of the fare payment, the direct and high-speed transit to the places of destination increased have been increased.

Management, based on the principles of logistics in passenger railways sector, is directed at the optimization of costs for transport services while increasing their quality and competitiveness. Currently, the logistics measures of railway passenger sector management include [1]:

- transit from information to information-analytical systems of transportation process management;
- the establishment of high-speed passenger corridors;
- the introduction of additional routes or additional cars in areas where there is a steady demand for transportation;
- optimal organization suburban traffic;
- the introduction of a single travel document for all kinds of passenger transportation;
- the elaboration of options of logistic chains to transfer passengers by railroad;
- technical and economic evaluation of options to transfer passengers on each section of logistics system;
- the assessment of transport service quality according to transportation options;
- the creation of virtual information and logistics centers;
- the completion of existing information systems for tracking the terms of travel documents realization and obtaining timely information as to the profitability of each train;
- the search for new market niches related to the production of goods or the provision of specialized services in order to attract workers and manufacturing capacities of Railways released during the reformation process in the sector.

Currently the management of passenger rail transportation based on logistics has already taken place in a Public joint-stock company «Ukrzaliznytsia». Thus, the audit of passenger traffic on each train on the basis of the introduced information system has enabled the Railways of Ukraine to undertake a number of activities to reduce of unprofitableness of passenger traffic, such as the replacement of suburban trains by rail buses on not intensive sections, the replacement of passenger trains by high-speed passenger trains, accelerated regional trains and high-comfort trains, and the abolishment of unprofitable sparsely populated trains etc.

The improvement of logistics approaches to the management process in passenger railway transport sector will put them to a new level and will ensure the growth of competitiveness of passenger transportation.

2. Problem discussion

Recently the passenger turnover between Europe and Ukraine is increasing in leaps and bounds, and this fact is a stimulus for transport companies of Ukraine to create new logistic schemes for passenger and tourists transportation (fig. 1) [1].

Fig. 1. Ukraine tourism potential model for calculating the prospective demand for intermodal passenger tourists by train

The most attractive for residents of Ukraine is tourism in European Union (EU) countries in northern and central Europe, in particular, the Baltic countries, Czech Republic, Slovakia, Hungary, Romania. Particular urgency tourist visits aforementioned countries takes in connection with the citizens in 2017 year Ukraine visa-free travel to the EU, and therefore the expected sharp increase in demand for international tourism from Ukraine to neighboring states.
The undeniable advantage of railway tourism (versus bus) is reliability, comfort and the possibility to navigate in all weather conditions travel by rail, which allows any season easily to transport tourists between cities visits mostly at night, providing quality recreation and meals on the go.

For tourists from Ukraine can become the most attractive areas: Ukraine – Poland – Baltic countries – Ukraine, Ukraine – Poland – Central Europe – Ukraine (fig. 2).

Besides, it is very important to develop an integrated short-term planning system for the purpose of interaction between passengers and carriers. The main point under creation of an intermodal system for passenger transportation is good organization, which influences in direct proportion to the quality of services. That is why it demands constant implementation of new technologies. The main problem solved by the intermodal transportation of passengers is the creation of an optimized travel due to transfer from one mode of transport to another one. Passengers who do not have a direct route with no transfer may need it, or it can be used by travelling tourists.

Public joint-stock company «Ukrzaliznytsia» is interested in the cooperation with other modes of transport and countries, and also in the development of the railway tourism, which is proved by the approved strategies of development. The creation of intermodal (transportation of passengers by different modes of transport with liability imposed on one operator) schemes for passenger transportation acts as an essential innovation of the transport area. The main goal of modern intermodal passenger transportation is minimization of private cars usage by the population and attraction of potential passengers to the public transport. The participants of intermodal logistic scheme within Ukraine-Europe direction are railway, air, road modes of transport and ferries as well (fig. 3).

Thus, the intermodal transport system represents integration of some modes of transport where, in most cases, the basic one, nevertheless, is the railway transport. The railway transport acts as the master link of the intermodal system because it is closely situated to other transport junctions of every city.

Passenger intermodality is a policy and planning principle that aims to provide a passenger using different modes of transport in a combined trip chain with a seamless journey [2].

Intermodality can be seen as a characteristic of a transport system that allows at least two different modes to be used in an integrated manner in a door-to-door transport chain. The adjective intermodal can be used for a service, facility, consignment of journey, involving transference between different modes of transport. Moreover, intermodal travel necessarily involves transferring from one mode to another. This usually takes place at modal interchanges.

The focus of this project is on long-distance passenger intermodality (journeys >100 km), including also the «first/last urban mile» (the connection with the regional and urban transport system) [1]. The strategic objectives of this research were:

- to support a more favourable environment for intermodal passenger travel across Europe;

- to foster the integration of intermodality policies for passenger travel;

- to facilitate cooperation to implement intermodal solutions;

- to overcome the fragmentation of the current transport market.

To achieve these objectives three main tasks had to be studied [2]:

- exchange to build a European network for intermodal passenger transport to exchange experience and work on better (transnational) solutions;

- transfer to set up a knowledge centre for intermodal passenger transport which structures research, defines research questions, formulates policy recommendations and disseminates information;

- promotion to promote passenger intermodality across Europe, mobilize political support, activate stakeholders and eventually develop a long term perspective for the project as an active organization to make sure every objective and task was treated to its full complexity. Three main work areas were defined. One of the Examples National Projects is Effective terminals for intermodal transports in Sweden [3]. The objectives of the project are to develop intermodal transhipment centres with respect to sustainability and efficiency in cargo handling in the terminal and towards connecting transport systems by sea, rail and road.

This being clear, there are a few related concepts and issues which may blur newly built picture about intermodality. In some texts, policy documents or other sources we might come across some concept that is closely related or even worse, that is wrongly used to point at intermodality. So listed them below and gave a definition for each of them.

Interoperability – capability to operate on any stretch of the transport network (especially crossborder) without any difference (regulatory, technical and operational systems need to be compatible) [2].

Intermodal transport – transport using different elements of a modal subsystem (requiring their cooperation) [2].

Multimodal – use of different modes of transport at different opportunities (trips/trip chains); policy principle not to stick to one single mode. The development of a seamless web of integrated transport chains, linking road, rail and waterways. Such integration would lead to improved flexibility, quality and cost effectiveness and would stimulate competition between transporters instead of between transport modes.
So, one of the most popular routes by railway transport within Ukraine-Europe direction is Kiev – Berlin – Hamburg – Paris (fig. 4) under questioning of 983 respondents that was conducted in 2016 year. On this route we can unite train and bus in intermodal transportation. The main idea was in using single travel document and implemented by a single company – Public joint-stock company «Ukrzaliznytsia». On fig. 4 red color was using for train direction and green color – for bus direction.

Fig. 4. The example of intermodal transport and logistics schemes for passenger transportation by railway transport within Ukraine-Europe direction

Also was analyzed route Kiev – Krakow – Amsterdam – Paris (fig. 5). There is combine also train and bus directions. On fig. 5 crimson and green colors was using for bus direction and blue and gray colors – for bus direction.

Fig. 5. The route of intermodal transport and logistics schemes for passenger transportation by railway transport within Ukraine-Europe direction

3. Conclusion

Due to the analysis of passenger railway traffic condition it has been grounded that the introduction of modern logistics management gives the opportunity to invest funds in new rolling stock at the expense of obtained revenues, to improve the quality of services and to refuse cross-subsidization of passenger transportation at the expense of freight transportation [4]. The methods presented in this work allow improving not only the efficiency of the technology of passenger train operation organization [5] but getting the highest profit possible from this kind of activity at the expense of reducing cost price.

While creating the intermodal system it is important to have online software products for the transfer of information and its correction in online regime. Besides, the essence of intermodal transportation lies in the fact that it is put into practice with a single travel document and implemented by a single company, which is more profitable both for passengers and carriers. Thus, we see implementation of the business-strategy «win-win».

4. Literature

1. Бараш, Ю. Перспективи розвитку залізничного туризму в Україні. – Журнал «Українська залізниця», №1-2 (43-44), 2017, стр. 18-23.
5. Aleshinskiy, E. Using the Petri nets for forming the technological lines of the passenger trains processing in Ukraine. – Archives of Transport, Warsaw, Poland, Volume 38, Issue 2, 2016, Pages 7-15.
TECHNICAL ASPECTS OF AUTOMATION OF TRANSPORT SECURITY

ТЕХНИЧЕСКИЕ АСПЕКТЫ АВТОМАТИЗАЦИИ ПРОЦЕДУР ОБЕСПЕЧЕНИЯ ТРАНСПОРТНОЙ БЕЗОПАСНОСТИ

Prof. Dr. Yelisov L.1, Prof. Dr. Ovchenkov N.2
Moscow State Technical University of Civil Aviation, Moscow, Russia
lev.el@list.ru1, ovchenkov@electronika.ru2

Abstract: This research relates to air transport security and includes studies aimed at improving systems to ensure an acceptable level of transport security by solving technical challenges in the field of adaptive incident management.

KEYWORDS: TRANSPORT SECURITY, SECURITY SYSTEMS, ACCEPTABLE LEVEL, TECHNICAL CHALLENGES, ADAPTIVE INCIDENT MANAGEMENT

1. Introduction

In recent years, the issue of air transport security has become of extreme importance given the unfavorable external conditions for activities related to transport services, such as the sharp rise in the number of illegal actions, including terrorism.

Professionals from different countries have been making significant efforts to solve this challenge and are increasingly creating specialized automated transport security systems. The most advanced achievements in various branches of science and technology are used as the technical basis for such systems.

Electronika Security Manager (ESM) is one such system. It was developed with the direct participation of the authors. ESM is a software and hardware platform that includes rather unique technical and organizational methodologies providing effective solutions for the entire complex of challenges related to securing the transport infrastructure. ESM was used as the foundation for the security system at Sochi Airport and demonstrated its effectiveness during the 2014 Winter Olympics.

2. Research Subject and Methods

ESM solves the entire range of tasks for managing facility security (see Fig. 1):
– Managing and integrating the facility's security subsystems – Collection and processing of data provided by different equipment and devices
– Alarm detection
– Assessment of the reliability and severity of alarms
– Identification of interrelated events and scenarios for the materialization of a particular threat
– Preliminary classification of incidents and deciding whether to assign incident status to the event – Initialization of the procedure for prompt response to the incident
– Collection of information on incidents; incident monitoring and management
– Ensuring the coordination of the Security Service and other departments
– Generation of alerts for senior managers to inform them of critical deviations
– Managing the enterprise security levels
– Real-time reporting
– Data distribution between users and convenient data presentation
– Automated submission of data on emergencies to law enforcement agencies

ESM includes the following subsystems: security alarm; alarm and warning system; process alarm; perimeter alarm; fire-safety systems; access control; video surveillance and video analysis; navigation satellite systems.

Alerters include the system's graphical interface, e-mail, texting, VoIP gateways connected to UTN/PBX, GSM, radio communications.

Figure 1. Generalized structure of the ESM
The multicamera video analytics system can point a dome camera at an object and display people, vehicles, and other objects on the map in real time. Instant search for events in the archive using visual tools and the map; report generator with key video frames and descriptions. The monitoring system allows mobile objects to be monitored and tracked using GLONASS/GPS trackers. The integrated radar-optical complex provides indiscriminate monitoring of the territory of the facility and restricted areas and also creates several virtual echeloned security lines. ONVIF is fully supported.

The video-surveillance system for the airport and landside is built on the basis of Milestone XProtect Corporate. This is one of the most powerful software products in the world that supports an unlimited number of servers, cameras, users, and virtually all vendors. This solution provides centralized management for all devices, servers, and users and also supports flexible rules with time-based and event-based triggers. This system includes the following components: system management server combined with SQL Express; Milestone recording servers; storage servers; AgentVI analytical servers; Milestone workstations (see Figure 2).

**Figure 2. Structure of the Milestone XProtect Corporate solution**

The video-analytics system uses Agent VI software in conjunction with Milestone XProtect. Agent VI is a platform for managing video analytics. The Data Collection and Processing System (DCPS) consists of a server based on ESM-server software and automated workstations based on ESM-client software.

The Transport Security Command Console (TSCC) is the central guard post for centralized output of information to digital monitors. The TSCC has operational control capabilities covering all the subsystems of the airport's Technical Security Equipment Set (TSES) (see Figure 3). The TSCC consists of several Automated Workstations (AWS). The centralized information environment and a single database enable a unified approach to information collection and processing. The interfaces were designed taking into account the goals and tasks of each user role.

Electronika Security Manager is a software platform deploying a single complex based on different equipment, devices, and software.
The integrated engineering safety systems and security alarm provide the following options: automatic response to an intruder attempting to enter a protected area and/or perimeter section, prompt notification of security personnel, and ensuring coordination of their actions.

Integration with the Intellect video-surveillance system provides the following capabilities: monitoring and control of video cameras, servers, video monitors; importing photos and events from Intellect; displaying video signals taking into account actuated alarm-initiating devices; analysis of archived alarms in the ESM-client interface; monitoring of events triggered by video-analytics sensors.

Integration with the Milestone XProtect video-surveillance system provides the following capabilities: receive and save XProtect alarms in the ESM database; put XProtect alarms on the list of alarm events; obtain the time of the beginning of the alarm event from the video archive and save it in the nESM database to be able to view video recordings associated with this event; put alarms of the AgentVI video analytics on the list of ESM alarm events; display video signals from all cameras associated with the alarm on the alarm monitor in real time; obtain the time of the beginning of the alarm event from the video archive and save it in the database; execute commands bound to the alarm.

Integration of existing security systems that are based on BIS (BOSCH) software provides the following features: centralized creation and management of users in BIS; centralized creation and management of access cards in BIS; centralized import of the list of access rights from BIS to ESM; centralized assignment of access rights and access cards to users in BIS; centralized management of access cards (activation and deactivation) and BOSCH equipment from BIS; centralized removal of access rights and access cards in BIS; centralized export of events related to users entering protected facilities in BIS; centralized export of events related to denial of user access to protected facilities in BIS; export of the list of access zones from BIS to correctly decipher events related to user access.

ESM provides step-by-step instructions for operators on responding to alarms and adjusts them taking into account the dynamics of negative impacts (see Figure 4). This approach increases response speed and accuracy, reduces requirements regarding the level of operator training, makes the situation more controllable, and increases the effectiveness of the measures taken. The response algorithm consists of the following three stages: receiving an alarm, verifying it, and executing the instructions.
The operating instructions include: malfunction of equipment or devices, security-system alarm, alarm-system alarm, people running in a controlled area, vehicles in the controlled area for too long, a large number of people in the controlled area, vehicles going too fast in the controlled area, activated Antipanic mode at the checkpoint, manual access control at the checkpoint.

Automated management of responses to incidents that are treated as groups of events obtained from different subsystems and correlated with one incident. In this case, comparison criteria (time and place of the incident) are taken into account. The system contains the following types of incidents: incidents related to alarms and incidents related to user access. Each incident can have three states: active, inactive, closed (see Figure 5).

Incident management is reduced to the operator processing alarm events. These events are triggered by security systems in accordance with the incident model implemented in ESM. Alarm events may be associated with activated security alarms, unclosed doors, pressed emergency buttons, suspicious objects in the surveillance area of the video camera, or activated detectors of the video camera. After processing the incidents, the necessary information for an Act of Unlawful Interference (AUI) is gathered, and AUI reports are generated for the competent authorities.

2. Conclusion

The proposed technical solutions implementing the methods for dynamic integration of transport-security systems at the airport are based on principles that imply that these technical solutions are a single dispersed system with a reconfigurable structure. Technical solutions related to integration management are based on the concept of object vulnerability, which is considered a qualitative characteristic. The set of security equipment and systems must be adequate for the level of vulnerability expressed as the quality of the systems for ensuring transport security. The control parameter expressed in quantitative terms is determined using qualimetry. The proposed technical solutions implement all necessary data connections to ensure proper management within the transport security system and solve tasks related to the dynamic integration of security systems at the airport.

Bibliography


1. Introduction

Modern approaches to traffic safety ensuring imply introduction of basic security norms. As for passenger transportation, such norms are probabilities of death or injury whereas damage or loss of cargo are relating to freight traffic. Areas for current repair with car uncoupling are intended to remove malfunctions which appear in the car operation in between scheduled repairs or after the car production and before its first normal repair. All defects of cars are to be removed disregarding their cause that may be normal aging of components and units, mistakes in performing loading-unloading and switching operations, violation of rules laid down in corresponding regulations in repairing cars and their components.

In work [1], the probability of a car accident is computed basing on a test sequence. However, the problem to determine the probability of failure of several cars running on a specific route rather than that for a single one has not been posed in that and other works. Similar investigations were performed in paper [2]. The author suggested a computational scheme to estimate the probability of accident proneness of a car. To perform the analysis a tree-like event graph was constructed which decomposed the final state (e.g., derailment of a car) into elementary events such as failure of specific units. In paper [3] the concept of the trouble-free life of car was used, the train safety being treated in such a way that failure of a single car might result in a complete train wreck under fatal circumstances. However, train wreck is a rare event. Other cases are more frequent when a defect is revealed in the course of inspection at the station. In this situation either a slight repair without uncoupling is performed or cars are uncoupled for a more serious repair. In such case risk of delivery delay or cargo damage occurs. The risks of just this kind are of main interest for consignors and insurance agencies: what is the safety level of a car group coursing in transit car detachments caused by technical malfunctions with the probability of uncoupling of N cars out of the train. In paper [4] the technique is outlined for estimation of car safety basing on the Markov process theory. In that work the transitions induced by random factors (failures) between states of the car-and-technical-service system have been considered. The state and transition probabilities are computed for a statistically average car, the process being assumed to be ordinary, i.e., excluding that several events occur within short time interval. Thus, like in the abovementioned works, the transition rate of a car to failure state is determined, rather than the probability of uncoupling for the car’s extraordinary technical maintenance and repair. The probabilistic estimate of uncoupling of several cars from the same train on a specific route is not discussed by the author either. Some forecast mathematical models have been suggested using correction factors in a number of works [5, 6] investigating the mutual influence of car reliability and performance of repair with uncoupling. The factors take into account the quality of operations with wagons at railway stations, on the one hand, and the influence of trends and seasonality, on the other. However, so far the models mentioned above are isolated from each other and they do not take account the full set of factors causing car uncoupling. For instance, in paper [6] the forecast model predicting the number of car detachments on the supply route is only developed for axle equipment heating.

Freight insurance is a sort of insurance of property which defends cargo against various risks such as incidents in the course of up-, down- and reloading operations, threats arising in cargo transportation etc. The insurer can properly estimate the risks related to freight transport if one possesses appropriate information not only on cargo and delivery distance, but also the transport operator. An increase of delivery distance over the nominally fixed average enhances the chance of insurable event. For this reason, insurers allow for specific multiplying factors depending on transport distance. To successfully solve problems arising in insuring cargo it is important that information is available on the number of car detachments occurring during transportation, each detachment being treated as random event with a definite probability.

In the present paper, the risk of failure to make delivery is estimated if there emerges a need to detach one or more wagons for extra repair because of technical malfunction on a specific route.

2. Computation of probability of a single-car uncoupling

Non-failure operating time is often considered in reliability theory to be an exponentially distributed random variable \( F(t) = 1 - e^{-\lambda t} \) (see, for example, GOST 51901.12-2007, Risk management. Method of analysis of failure types and effects). It results from the fact that the probability of non-failure operating of a device within a time interval \( t \) does not depend on duration of the preceding failure-free run from its start to the beginning of the time interval under consideration, but it only depends on the duration \( t \) [7]. One can adopt exponential distribution also in analyzing in-transit car detachments caused by technical malfunctions with the covered distance \( L \) as its argument instead of elapsed time. Distribution function \( P(L) = 1 - e^{-\lambda L} \) allows computing the probability \( P \) of a car uncoupling on the route of length \( L \). Prediction of the uncoupling rate \( \lambda \) can be done on the basis of statistical data. A large size of data makes it possible to reliably predict the car-uncoupling rate per 1 wagon-km.

Assuming that a malfunction implying car detachment can appear on any route segment of fixed length with equal probability, calculating the uncoupling probability of a loaded car can be performed by the formula:

\[
P = 1 - e^{-\lambda L \eta_{\text{fp}} (1 - d) (1 \pm K)},
\]

where \( L \) is transportation distance;
\( \Psi_p \) – car-uncoupling rate per 1 wagon-km;

\( K \) – seasonal transportation factor;

\( d \) – portion of empty cars in current repair.

One can use the following approximate formula for small values of \( L \Psi_p \) instead of formula (1):

\[
P_{pr} \approx L \Psi_p (1 - d) (1 \pm K).
\]

The car-uncoupling rate per 1 wagon-km \( \Psi_p \) is determined by the relation:

\[
\Psi_p = \frac{n_p}{\sum_{i=1}^{k} L_i},
\]

where \( k \) is the number of cars;

\( \sum_{i=1}^{k} L_i \) – the total run of all \( k \) wagons (km);

\( n_p \) – the number of car detachments for current repair.

Known the portion of cars of each type that are in current repair, the value of \( n_p \) can be calculated from the data of branch statistical.

**Fig.1 Number of car detachments by quarters**

The number of car detachments noticeably increases in autumn and winter period. In this work it is suggested to take this fact into account by seasonal transportation factor \( K \). It enters into formulae (1) and (2) with plus sign for the autumn and winter period and with minus sign for the spring and summer one.

In this paper we deal with detachments of loaded wagons only, since the transport operator may suffer a loss in that case. For this reason the factor \( d \) is introduced in the formulae to exclude detachments of empty cars from consideration.

### 3. Risk of a car uncoupling from the train

Let us assume that several identical wagons are combined to one train for cargo transportation. An in-transit uncoupling of any wagon due to its technical malfunction is a random event with the probability which does not depend on the fact if some other wagon went wrong. In other words considerations are dealt with to be independent events. Let uncoupling probabilities of all wagons have the same value. Suppose \( N \) is the number of cars in a wagon batch. Then one has to do with a sequence of independent tests, in each of which a car may be uncoupled because of malfunction occurred in the course of transportation with equal probability. Let us denote by \( n \) the number of cars that may be uncoupled. The probability \( P(n; N) \) that \( n \) cars are uncoupled from a batch of \( N \) wagons can be calculated by the Bernoulli formula [7]:

\[
P(n; N) = C^N_n \cdot P^n \cdot (1 - P)^{N-n},
\]

where \( C^N_n = \frac{N!}{n!(N-n)!} \) is the number of combinations consisting of \( n \) elements chosen from a set of \( N \) elements \((N! = 1 \cdot 2 \cdot \ldots \cdot N)\).

The most probable number of cars \( m(AB) \) that may be uncoupled due to in-transit failure detection on the route between points A and B is determined by the following formula

\[
m(AB) = N \cdot P = L N \Psi_p (1 - d) \cdot (1 \pm K),
\]

suppose that \( N \cdot P \) is an integer. Otherwise the most probable number of uncoupled cars \( m(AB) \) is obtained from the following two-sided inequality

\[
N \cdot P + P - 1 \leq m(AB) \leq N \cdot P + P.
\]

Let us consider a specific example of probability computation for an in-transit car detachment caused by technical failure. Here are source data to make calculations: \( L = 9882 \text{ km}; N = 50 \text{ cars}; \Psi_p = 0,00003741; d = 0,78; K = 0,162 \). The transportation is assumed to be carried out in spring and summer period. The probability \( P \) of a car detachment due to in-transit detection of a technical malfunction is approximately found by formula (2)

\[
P_{pr} = 9882 \cdot 0,00003741 \cdot (1 - 0,78) (1 - 0,162) \approx 0,068.
\]

A more precise value of \( P \) is obtained via formula (1)

\[
P = 1 - e^{-0.068} = 0,666.
\]

The most probable value of the number of cars uncoupled on the route between points A and B \( m(AB) \) is got from the following two-sided inequality

\[
3,4 + 0,066 - 1 \leq m(AB) \leq 3,4 + 0,066.
\]

The integer \( m(AB) = 3 \) obeys this inequality. The probability of a detachment of three cars from a fifty car batch is computed by formula (3) with \( n = 3 \):

\[
P(3; 50) = C^50_3 \cdot P^3 \cdot (1 - P)^{50-3} = 50 \cdot 0,066^3 \cdot (1 - 0,066)^{47} \approx 0,228.
\]

The risk that one, two or no car at all will be uncoupled can be found by formula (3) too:

\[
\begin{align*}
P(1; 50) &= C^50_1 \cdot P^1 \cdot (1 - P)^{50-1} \approx 0,117, \\
P(2; 50) &= C^50_2 \cdot P^2 \cdot (1 - P)^{50-2} \approx 0,202, \\
P(0; 50) &= (1 - P)^{50} \approx 0,033.
\end{align*}
\]

The probability of detachment of an arbitrary number of wagons can be calculated similarly. **Fig.2** illustrates the behavior of the car detachment probabilities.

**Fig.2 Probability of uncoupling of various numbers of cars from a car batch.**

The risk that one, two or no car at all will be uncoupled can be found by formula (3) too:

\[
\begin{align*}
P(1; 50) &= C^50_1 \cdot P^1 \cdot (1 - P)^{50-1} \approx 0,117, \\
P(2; 50) &= C^50_2 \cdot P^2 \cdot (1 - P)^{50-2} \approx 0,202, \\
P(0; 50) &= (1 - P)^{50} \approx 0,033.
\end{align*}
\]
The probability of detachment of an arbitrary number of wagons can be calculated similarly. Fig.2 illustrates the behavior of the car detachment probabilities. Another important way to describe the risks is the interval estimation of the number of cars uncoupled for a specified confidence level \( P_{\text{conf}} \). It is possible to state with a degree of reliability \( P_{\text{conf}} \) that the number of cars uncoupled cannot exceed the value \( n_{\max} \). The maximum value of car detachments \( n_{\max} \) is determined by the system of inequalities

\[
\begin{align*}
P(n > n_{\max}) &\leq 1 - P_{\text{conf}}, \\
(1 - P(n > n_{\max} - 1)) &> 1 - P_{\text{conf}}.
\end{align*}
\]

For instance, we get \( n_{\max} = 8 \) as the maximum number of uncouplings at confidence level \( P_{\text{conf}} = 0.99 \) in the example given above. Fig.3 demonstrates results of calculations of the maximum number of cars detached are shown at various confidence levels.

**Fig.3** Interval estimation of car detachment risk.

### 4. Car detachments at variable length of the route

The probability \( P_L \) of a car uncoupling in case of a technical malfunction detection for variable route length is as follows

\[
P_L = 1 - e^{-L \cdot 0.00003741 \cdot (1 - 0.78) \cdot (1 - 0.162)} =
\]

\[
1 - e^{-L \cdot 0.0000069} \approx L \cdot 0.0000069.
\]

The probability that a car will not be uncoupled is equal

\[
Q_L = 1 - P_L = e^{-L \cdot 0.0000069}.
\]

If there are \( N \) cars in the train, then the probability of no detachments is computed by multiplication theorem [7] for independent events

\[
P_{(0; N)} = Q_L^N = e^{-N \cdot L \cdot 0.0000069}.
\]

Figure 4 demonstrates how the probability of uncoupling of at least one car depends on transportation distance for various numbers of cars in the train.

**Fig.4** Risk of uncoupling of at least one car for various transportation route lengths and numbers of cars in the wagon batch.

The probability of at least one uncoupling, i.e., that of an opposite event, depends on the transportation distance. The expression giving the probability of uncoupling of at least one car which depends on the number of cars in the batch and the transportation distance is as follows:

\[
P_L(n > 0) = 1 - P_L(0; N) = 1 - e^{-N \cdot L \cdot 0.0000069}.
\]

If we assume that the train consists of \( N \) cars the calculation of the probability of uncoupling of just one car can be carried out by means of the Bernoulli formula [7]:

\[
P_L(1; N) = C_N^1 q_L = C_N^1 \cdot P_L \cdot Q_L^{N-1} =
\]

\[
= N \cdot \left(1 - e^{-L \cdot 0.0000069}\right) \cdot e^{-N(1-1) \cdot L \cdot 0.0000069}.
\]

Then the formula for computing the probability that two or more cars will be uncoupled on the route due to technical malfunction has the form:

\[
P(n > 1) = 1 - P_L(0; N) - P_L(1; N).
\]

Fig.5 demonstrates the results are presented for the probability of detachment of one or two cars from a 50 wagon batch, as well as that of no uncoupling at all.

**Fig.5.** Detachment risk on various routes for a batch of 50 wagons

If cars of different types with unlike detachment rates are present in the train one should first calculate the probability of detachment of \( n \) cars for each group of similar cars. Then one computes the probability of uncoupling of \( n \) cars from the train using the total probability formula [7].

Let us discuss a specific example. Suppose the train consists of 30 cars of kind I with detachment probability \( P_{(I)} = 0.066 \) and 20 cars of kind II with detachment probability \( P_{(II)} = 0.052 \). The probability to choose at random a car of the first kind equals \( P(I) = 30/50 = 0.6 \). For the second type cars it is \( P(II) = 20/50 = 0.4 \). The conditional probabilities of uncoupling of the cars of both kinds are

\[
P_L(1; 30) = C_{30}^1 \cdot P_{(I)} \cdot (1 - P_{(I)})^{30-1} \approx 0.273;
\]

\[
P_L(1; 20) = C_{20}^1 \cdot P_{(II)} \cdot (1 - P_{(II)})^{20-1} \approx 0.323.
\]

One can compute the probability of uncoupling of one car of each kind using the total probability formula

\[
P(1; 50) = P(1; 30) \cdot P(II) + P(II) \cdot P(I; 20) = 0.293.
\]

The probabilities of uncoupling of any number of cars are determined similarly.

As our calculations have shown, the probability of an in-transit car detachment is significant considering the actual rather low reliability of the car fleet. The probability of uncoupling of a car is close to 0.37 for transportation distance from 2500 km to 3000 km. Whereas the probability of uncoupling of two wagons becomes 0.27 for a route length from 5500 km to 6000 km. Taking into consideration an increasing cargo turn-over between the countries
of the European Union and Asia-Pacific region [8] it is necessary to estimate risks relating to delay of freight delivery.

5. Uncouplings for a variable number of cars in the train

The results obtained above can be generalized to the case when the number of cars in the train is unknown in advance being a random variable with a definite probability distribution law. In this work we assume that the number of cars in the train is equal to \( Z + 1 \), \( Z \) being a random Poisson distributed variable with the parameter \( \lambda \). We have introduced a shift by unity in order to exclude the case of trains with zero number of cars. Thus the probability that there are \( N \) cars in a train is equal to

\[
q(N) = e^{-\lambda} \frac{\lambda^{N-1}}{(N-1)!}
\]

Here \( \lambda + 1 \) has the meaning of the average number of cars in the train. The probability that the number of cars in a train is within the range from 46 to 55 equals 0.53 at \( \lambda = 49 \). This probability amounts to 0.85 for the range from 41 to 60 cars. Fig. 6 plots this probability versus the number of cars in the train.

![Fig.6. The probability of various numbers of cars in a train.](image)

The detachment probability for \( n \) cars is calculated by the formula of total probability [7]:

\[
P(n) = \sum_{N} P(n; N) \cdot q(N).
\]

The probability of detachment of \( n \) cars from a train containing \( N \) cars is determined using formula (3). Note that \( P(n; N) = 0 \), if \( N < n \). The curve in Fig. 7 plots the probability of detachment of various numbers of cars evaluated by formula (6). As is seen from the figure the most probable number of uncoupled cars is equal to 3.

![Fig.7. Probabilities of the number of cars uncoupled.](image)

Let us notice that fig. 2 and 7 differ only slightly. Therefore variability of the number of cars does not influence the results of calculations of the uncoupling risk probability essentially if the average number of cars is fixed.

The expectation value, i.e., the mean number of wagons uncoupled in the train with a variable number of cars, is computed in the following way

\[
n_{av} = \sum_{n} nP(n) = \sum_{n} \sum_{N} P(n; N) \cdot q(N).
\]

\( n_{av} = 3.4 \) for the parameter set chosen (\( L = 9882 \) km; \( N = 50 \) wagons; \( \Psi_{p} = 0.00003741; d = 0.78; K = 0.162 \)).

6. Conclusion

The methodology of a numerical analysis of operational reliability of the car fleet is given in this paper. When calculating the single-car in-transit uncoupling probability due to a technical malfunction an exponential distribution was used with the distance covered as its argument. Prediction of the uncoupling rate was done on the basis of statistical data of freight cars. A formula was developed to obtain the probability of at least one car uncoupling depending on the number of cars in the wagon batch and the length of transportation route. The probability of detachment of \( n \) cars from \( N \) ones in the batch was calculated by the Bernoulli formula for independent trial sequence. A method was demonstrated for determining the maximum number of car detachments given the confidence level. The results obtained were generalized to the case when the actual number of cars in the train was not known beforehand.

The obtained dependences of the uncoupling probability on transportation distance allow assessing safety level in international transit corridors where each local railway administration shares liability for damage to cargo and railway equipment in the course of transportation through their area of responsibility. The technique of assessment of the car uncoupling risk outlined in this paper can be utilized by insurance companies to justify their financial assets in insuring freight delivery.

7. References


OPERATIONAL SAFETY OF WORK PLATFORMS

doc. Ing. Marianna Tomašková, PhD.
Technical University in Košice, Faculty of Mechanical Engineering, Department of Safety and Quality of Production, Letná 9, 042 00 Košice
marianna.tomaskova@tuke.sk

Abstract: Elevating work platforms are used for work in places that are not easily accessible, and with their design and subsequently appropriate design craftsmanship they ensure safe work in construction and assembly operations, inspections of street lighting and many other activities. The mobile elevating work platforms that are manufactured today, with their level of construction, quality of workmanship and ever-improving built-in safety features, create safer working conditions at heights and thus a quieter and safer working environment not only for the workers themselves but also for their employers. The paper analyzes the dangers and threats when working with platforms and the requirements for operators working with this equipment. Subsequently, risks are identified using the risk-assessment methods in the standards TNI ISO / TR 14121-2 Safety of Machinery. Risk assessment. Part 2: Practical guidance and examples of methods. In the conclusion is a proposal of corrective measures that can improve safety when working with elevating work platforms.

Key words: MOBILE ELEVATING WORK PLATFORMS, OPERATIONAL SAFETY, FUNCTIONAL STRUCTURE, RISK

1. Introduction

Work at heights ranks in terms of work safety among the most dangerous types of work and potential work injuries among the most serious. Thus, claims for occupational safety and health naturally grow as a result. A basic technological safety requirement when performing construction assembly works is to protect workers in places where there is the danger of falling from a height. At present mobile elevating work platforms are becoming ever more popular and are commonly used with work at heights.

Selected basic concepts:

Mobile elevating work platform: [10]
✓ a mobile machine intended for the transport of persons to a workplace, where they will perform work activities from the work platform under the conditions that persons get on and off the work platform at a determined access point from the level of the terrain or from the undercarriage and which is made up minimally of work platform with controls, telescopic construction and an undercarriage.

Work platform:
✓ an enclosed platform or basket, which can be moved while loaded into the required work position and from which it is possible to erect, repair, inspect or perform similar work.

Elevating platform with transport of the operator:
✓ an elevating platform, on the surface of which the operator enters for loading and unloading or on which the operator may be transported, for which the platform is equipped with controls.

Operator:
✓ the person trained for safe operation of an elevating platform according to the manufacturer’s instructions.

Remote controlling:
✓ a control connected by a cable which is not located on the elevating platform itself [1].

Prohibited area:
✓ a space which is reserved only for a person authorised to be in it and which is not accessible to the public.

Nominal loading capacity:
✓ the loading capacity which the equipment lifts when used according to the operating instructions, as guaranteed by the manufacturer [1].

Protective cover:
✓ a part of the machine used especially for protection by means of physical blockage.

Safety position:
✓ state when the elevating platform or part of the elevating platform is sufficiently secured against entry, thus preventing any threat to persons or cargo.

Emergency stop control:
✓ a part of the emergency stopping equipment, which after activation of the incorporated manual control (trigger switch) sends an emergency stop signal.

Description of a work platform:
With respect to the range and various types of performed work the offer of elevating platforms on the market is broad and diverse. Construction of the platforms is intended for specific work environments and use in practice, and thus they are engineered in different forms, versions and sizes. With regard to the mentioned facts it is evident that for good selection of elevating platforms their size or type with regard to their planned use (activities performed and their range) also has a key influence on their effective use.
2. Categorizing of mobile elevating work platforms by functional structure

Lifting equipment as an element of a person–machine–environment system can be evaluated as an integrated machine system whose individual functional structures cannot be assessed separately. Most important is the fact that after launch of the system into operation, risks associated with its operation are evident. In relation
to this, it is necessary to start from the fact that it is necessary to ensure an identical technical level not only in the lifting equipment system but also from the fact that the platform must, as a part of a material flow system, correspond to the technical level of the whole technological process [4].

Lifting equipment as a machine system can be divided into six functional structures:
- the guiding function of the structure,
- the suspension function of the structure,
- the moving function of the structure,
- the strength-load-bearing function of the structure,
- the safety function of the structure,
- the controlling-regulation of the structure.

During operation of mobile elevating work platforms these functional structures can be distinguished:

1. **Moving function of the structure** – the role of this structure is to ensure the mobility, lift and rotation of the platform, a change of position of the boom and additional movements, so that the most effective space is created for its use during work. The moving function of the structure is made up of mechanisms which also ensure the transfer of the power flow to the strength function of the structure [4].

![Fig. 7 Moving function of the structure [5]](image)

2. **Strength-load-bearing function of the structure** – its purpose is to ensure transfer of the external loads influencing the elevating platform. This function at the same time conditions the selection of the material and the form of the load-bearing construction of the elevating platform. At present the construction of this equipment is made from steel – a steel load-bearing construction. The given structure also has a significant impact on stability, which is one of the main safety factors of elevating platforms [4].

![Fig. 8 Strength-load-bearing function of the structure [5]](image)

3. **Safety function of the structure** – its primary task is to ensure safe operation of the elevating platform. This structure must be paramount over other functions of the structure. A component of it is equipment that records the technical status of the machine during operation.

![Fig. 9 Safety function of the structure [5]](image)

Standard safety features:
- double control panel,
- flow cabling for hydraulic rollers,
- safety device with destruction of hoses on all rollers,
- exact horizontal position of the basket in all positions,
- manual emergency starting,
- hydraulic and electrical protection against overloading,
- hydraulic support legs,
- control signal for all functions,
- electronic switch when tilting, with a signalling alarm.

4. **Controlling-regulation of the structure** – the task of this structure is to ensure controlling of the working movements of the elevating platform, such that the loading on its other functional structures is not increased. At present the operator of an elevating platform uses it most often for this purpose. The behaviour of the operator has been the object of several studies, and it acquires, for example, ever greater importance in connection with the expanding use of continuously regulated drives. Two types of controlling of the elevating platform are distinguished – controlling in the basket and remote controlling. This structure in the majority of cases is unable to prevent incorrect usage [8].
3. Dangers and risks when operating mobile elevating work platforms

For the exclusion of risks at work with elevating work platforms the operator must follow the relevant instructions and recommendations.

If this is not done, the following risks and threats can arise:

- fitful controlling of the control stick,
- overloading of the platform,
- uncertain relations on the ground,
- gusts of wind,
- contact with obstacles on the floor or at height, creating the danger that the machine will begin to move uncontrollably forward, backward, to the side or will tilt over.

According to valid provisions on high-tension lines, persons and equipment for elevating persons may not operate closer to electrical outdoor power lines than safety zones permit. If doing work requires a smaller distance from power lines, it is necessary to agree with the operator or owner of the distribution network on a method of execution. For example, the safe distance for power lines in the range of 1000 volts of one-way voltage within a town is for mobile elevating work platforms in non-insulated configurations minimally 1 meter from the lines of the outermost line and from parts under voltage. The safe distance from the power lines for railways of both voltage systems is 2 meters from the lines. In the case of need of a work platform at a distance smaller than 2 meters, it is necessary to fulfill the requirements given in technical standards [6].

During storms the use of mobile elevating work platforms is prohibited. In environments threatened by explosion or fire, such as, for example, the charging of accumulators in a closed space with simultaneous activity of the platform, filling the fuel tank in the vicinity of an open flame, contact with heated parts of a motor, using equipment with oil leaking from the hydraulic system, the danger of explosion or burning arises. Persons may not move in the work space of a machine, and in the case that solid barriers are located there, the operator must prevent dangerous collisions with a moving part of the machine. Therefore, before each use the operator must unconditionally inspect whether persons are moving in a dangerous work space and ensure that no dangerous collision will occur. In order to limit the threat of a fall or injury, dangerous manoeuvres cannot be made, safety and signalization elements cannot be taken out of operation, and it is prohibited to sit and climb on the railings during movement, and the like [7].

- Danger of shock from electric current
  A work platform is not electrically insulated, and therefore it does not ensure protection in the case of contact with electric current or being in its vicinity [13].

- Danger of tilting over
  The weight of persons, equipment and material on a platform cannot exceed the load-capacity of that platform. The weight of supplements and accessories, such as, e.g. pipe holders, sheet metal and welding aggregates, which decrease the nominal load-capacity of the platform, must be included in the total loading of the platform. The boom cannot be lifted nor disengaged if the machine is not standing on a solid and even surface [6].
4. Identification of threats

When assessing the risks of any machine a fundamental step is the systematic identification of threats which are adequately foreseeable, that is determining situations in which during the performance of planned or unplanned operations, a person may be exposed to a dangerous situation which could also lead to endangerment of his life or health. The design engineer must take into consideration all types of threats which can occur at the individual stages of the equipment lifespan, in particular with:
- assembling, installation and transport of the machine,
- putting it into operation,
- use and maintenance,
- taking out of operation, disassembling and disposing of the machine [8].

The types of threats, their sources and potential consequences are listed in Table 1.

**Tab. 1 Examples of types of threats, their sources and potential consequences [9]**

<table>
<thead>
<tr>
<th>Type, or group</th>
<th>Examples of threats</th>
<th>Potential consequences</th>
</tr>
</thead>
<tbody>
<tr>
<td>1. Mechanical threats</td>
<td>Accelerating, slowing</td>
<td>Collision, fall from a height</td>
</tr>
<tr>
<td></td>
<td>Angular parts</td>
<td>Cutting oneself</td>
</tr>
<tr>
<td></td>
<td>Collision of the platform with a solid part</td>
<td>Striking, contusion, fall</td>
</tr>
<tr>
<td></td>
<td>Falling objects</td>
<td>Collision</td>
</tr>
<tr>
<td></td>
<td>Starting of the platform</td>
<td>Smashing of the operator’s hand</td>
</tr>
<tr>
<td></td>
<td>Unstablleness of the machine</td>
<td>Collision, being thrown off, fall,</td>
</tr>
</tbody>
</table>

| 2. Electrical threats | Live parts | Shock by electrical current |
| | Electromagnetic factors | Biochemical changes, breakdown of brain activity |
| 3. Thermal threats | Hot or cold objects or material | Dehydration, burning, freezing |
| | Explosion | Burning, contusions |
| | Radiance from a heat source | Injuries caused by radiance from heat source |
| 4. Vibration threats | Moving equipment | Stress, headache |
| | Wearing down of parts | Inattention |
| 5. Ergonomic threats | Position | Dizziness, pain in the legs |
| | Limited space | Discomfort, mental fatigue |
| | Work strain | Hand pain |
| | snow | Slipping, fall |
| | temperature | Dehydration, seizure, catching cold |
| | rain | Illness, discomfort |
| | wind | Turning over, fall of the platform |

Sources of threats when working with a work platform and the potential consequences following from them are depicted in Table 2.
### Tab. 2 Threats with images

<table>
<thead>
<tr>
<th>Threat</th>
<th>Source</th>
<th>Potential consequences</th>
</tr>
</thead>
<tbody>
<tr>
<td>Work at height</td>
<td>Live parts under voltage</td>
<td>Collisions, contusion</td>
</tr>
<tr>
<td>Moving parts</td>
<td>Charging of the accumulator</td>
<td>Collision, contusion</td>
</tr>
<tr>
<td>Unstable platform</td>
<td>Inexpert operator</td>
<td>Collision, tipping over</td>
</tr>
<tr>
<td>Dangerous behaviour</td>
<td>Unevenness of surface</td>
<td>Fall from a height, collision</td>
</tr>
</tbody>
</table>

### Tab. 3 The most serious risks when working with a work platform [9]

<table>
<thead>
<tr>
<th>Dangerou situation/threat</th>
<th>Possible damage to OSH/ Negative consequence</th>
<th>Seriousness S1/S2</th>
<th>Frequency F1/F2</th>
<th>Possible prevention P1/P2</th>
<th>Risk a+</th>
</tr>
</thead>
<tbody>
<tr>
<td>Collisions, fall from height due to sudden acceleration / slowing of the machine</td>
<td>Broken bones, tearing wounds, internal bleeding, death of a worker</td>
<td>S</td>
<td>F</td>
<td>P</td>
<td>C</td>
</tr>
<tr>
<td>Tipping over, fall of the platform due to instability of terrain</td>
<td>Broken bones, tearing wounds, death of a worker</td>
<td>S</td>
<td>F</td>
<td>P</td>
<td>C</td>
</tr>
</tbody>
</table>

Proposed measures:
- to inform operators about the dangers and safety system of the work, which should be observed
- to organize regular training for the lift platform operator
- selection of a suitable type of platform with sufficiently high railings
- Use of safety belt / rope
- using a platform only on a solid and flat foundation
- extension and failure of supports on the floor against movement of the elevating platform
- design of signalization equipment
### 5. Conclusion

At present elevating work platforms are irreplaceable in construction and handling technologies. When it comes to work on buildings, assembly works or various actions on above-ground cabling for electrical networks,
accidents that do occur with work at heights are among the most common reasons for serious injury and death. In the following short instructions it is possible to see what can be done in order to minimize risks and successfully manage them.

1. Threat of collision, fall from a platform by sudden accelerating/slowing of the machine [9]

A sudden acceleration or slowing of the platform may lead to the collision of the operator with the platform construction, or to a fall of the worker from the working height to the ground, which has in the great many cases very negative consequences in the form of a serious injury, such as broken bones, tearing wounds and in the worst cases ends in death. Application of the following measures could possibly contribute to improving the current state:
- informing of operators about the dangers and the safety system of work which should be observed,
- organizing of regular training sessions for the operator of elevating work platforms,
- selection of a suitable type of platform with sufficiently high railings so that falls are prevented,
- use of a safety belt/rope which would also prevent a fall from the platform.

2. Threat of tipping over the platform due to instability of the terrain

Work and use of a work platform in unsuitable terrain with an unstable base could lead to tilting, and upon subsequent loss of stability, even the tipping over of the platform. This could lead to serious undesired consequences. In the case of such a situation a devastating injury of even death can easily occur. For prevention of a similar event it is possible to use these proposals:
- use the platform only on a solid and even base,
- spread the telescoping supports and set them onto the floor before starting to lift the platform,
- do not overload the platform with a number of operators or different heavy materials which could lead to tipping over,
- design signalization equipment which would announce a critical tilt of the terrain, or tilting of the platform with a sound or light signal.

3. Threat of the tipping over the platform due to strong wind

Unfavourable wind conditions can also threaten the stability of a platform. A strong wind in connection with other factors has the potential to lead to an unfortunate course of events with subsequent tragic consequences. Measures for this case are:
- observing the set maximum safe speed of the wind for operation of the platform,
- not overloading the platform by the operator or material used,
- with strong wind use telescoping supports for increased equipment stability,
- design of signalization equipment for monitoring the wind.

4. Threat of lightning strike with work on a platform

For safety during operation of a work platform natural factors are also considered, since a great amount of work on platforms is performed outdoors. Among those natural factors are lightning, which is very dangerous during operation of platform, as a possible lightning strike could have fatal consequences. The following measures are proposed for avoiding such situations:
- observing the prohibition on using a platform during storms,
- finishing work and leaving the platform with lightning and thunderstorms nearing.

5. Threat of shock by electric current

Anywhere there is work with electrical equipment or electric current, there is a potential threat. With defective insulation, incorrect maintenance, or in cases when the electric current reaches the external construction of the platform, an operator could be shocked, and this could lead to serious health damage. Shock may also occur in cases not directly related to work with the platform, but when the operator of the platform works on above-ground electrical power lines or with maintenance of public lighting. These proposals are in place in order to successfully avoid this:
- observing the prohibition on using work platforms in the near vicinity of unprotected power lines,
- disconnection of electricity when working on electric power lines,
- observing principles of safety maintenance.

6. Threat from heat, cold, rain or snow

The influences of the external environment are significant factors with operation of work platforms, since their effects cannot be avoided and weather cannot be adapted. The influence of different unfavourable situations, which obviously do not help with comfort and safety of work, may follow from different types of weather. Measures with negative effects of weather are:
- ensuring a sufficient amount of fluids,
- increasing the number of breaks with extreme heat or cold weather,
- selection of suitable clothing according to the weather.

7. Threat of explosion when charging an accumulator

Charging an accumulator also belongs among those activities which have the potential to lead to undesired events, for example, an explosion. This as a consequence could cause serious injuries or even death. Proposals for preventing this happening are:
- regular control and maintenance of the work platform,
- ensuring the observance of safety processes with maintenance and charging of the accumulator
- strict observing of the prohibition on smoking or handling an open flame when charging an accumulator,
- charging of accumulators in well ventilated spaces.
8. Threat of crushing and striking of a platform operator when handling the platform

Operation of a platform can lead to crushing and injuring of workers, when moving or folding up of the equipment, which in the majority of cases leads to devastating injuries. These measures are used for avoiding such cases:

- observing the principles of safe behaviour,
- devoting increased attention to solid barriers,
- improving the method of communication between workers.

The contribution was prepared in the scope of APVV-15-0351 Development and application of models for managing risks in the conditions of technological systems in line with the strategy Industry (Industrie) 4.0.

“VEGA 1/0150/15 Development of methods of implementation and verification of integrated systems for safe machines, machine systems and industrial technologies”.

Paper is result of the Project implementation: University Science Park Technicom for Innovation Application Supported by Knowledge Technology, ITMS: 313011D232.

8. Literature


[6] GENIE INDUSTRIES: Návod k obsluze s informacemi o údržbě, USA, tretí výtisk. 2005. 47 s. Part No. 110015CZ.


THE ANALITICAL RESEARCH OF THE PROCESS OF FORMING THE MOTOR-GRADER MOTION PATH AT IMPLEMENTATION OF WORKING OPERATIONS

Cand. Eng. Sc., Associate Professor Shevchenko V.1, Post-graduate student Chaplygina A.1, Junior Researcher, Beztsennaya Zh.2
Faculty of Mechanical – Kharkiv National Automobile and Highway University, Ukraine 1
Department of Industrial Policy and Energy Security – Research Centre for Industrial Problems of Development of NAS of Ukraine, Ukraine 2
olexandrachaplygina@gmail.com

Abstract: Based on the analysis of the mathematical model of the motor grader movement in the process of implementation of working operations, the factors that have the greatest influence on formation of the machine’s motion path are identified. It is proved that some of the factors characterizing the parameters of the machine and technological process have a destabilizing impact on the formation of the grader motion path, while the rest of them have a stabilizing impact. The analytical research enables developing practical recommendations to ensure the movement of the machine along a predetermined path without deviations.

Keywords: ROAD-HOLDING ABILITY, PROCESS OF MOVEMENT, MOTOR-GRADER, MATHEMATICAL MODEL

1. Introduction

As a rule, motor-graders perform working operations in the process of which the main blade is positioned at the working angle different from 90 and with he tilt in the vertical plane. In this case at cutting the developed medium the formation of a cut layer of a triangular shape with the simultaneous dumping of the material to the side takes place. The main peculiarity of such operations is the displacement of the coordinate of application of the resultant resistance vector on the blade, which leads to an asymmetric loading of the machine [1]. The force action of this type can lead to the deviation of the machine form the predetermined motion path, which results in the deterioration of indicators of quality of the implemented working operations, a drop in the productivity of the machine, etc.

2. Analysis of publications

The asymmetric application of external loads to the working body, the action of lateral forces on the working attachment and undercarriage can lead to the deviation of the machine from the planned path of motion, interpreted as the loss of road-holding ability.

The lateral displacement is at a large degree is characteristic for machines with unevenly distributed load affecting the working attachment of agricultural machinery, earth-moving machines (EMM), etc.

Most of developments in this field are dedicated to the problem of the vehicle movement across the slope with its differential unlocked. The resulting redistribution of the vertical reactions onto its wheels causes the emergence of destabilizing moments along with other moments that cause the deviation of the tractor from the predetermined direction. By means of the technology [2, 3] that implies movement of an additional load in the lateral direction in the form of a flat plate to cause a lateral displacement of the center of gravity, the emergence of stabilizing moments proportional to the slip moments acting on the respective axles of the tractor is provoked. This results in the increase of the vehicle’s road-holding ability and it keeps moving within the predetermined motion corridor.

There are many designs of the front axle with the control of the tilt of its propelling devices in the vertical plane in order to maintain a straight-line motion on the slope [4].

Some of the inventions concerning the problem of the road-holding ability solve this task by means of an additional wheel, mounting it along the longitudinal axis of the vehicle either in front of or behind it. This can be a disk, wheel or ripper able to keep the machine on the chosen trajectory of motion [5].

3. The solving of the set purpose

The carried out field researches have allowed to identify features of the process of loosing the road-holding ability by a motor-grader [1]. In particular, at the initial stage of work, despite the asymmetric application of external resistance forces \( \sum W \) (Fig.1), the machine moves along the straight-line trajectory. Then as the material accumulates in front of the blade, there comes a moment when the motor-grader spontaneously turns around with respect to the point corresponding to the coordinate of application of the resultant vector of external resistance forces (point O), after which the machine again continues its straight-line motion. The actual trajectory of the machine motion consists of linear segments and zones of unintentional turning in the points of their joining.

At approaching the zone of unintentional turning, the actual linear speed of the motor grader decreases almost to zero, but due to the considerable engine power there takes place an obvious slippage of the driving wheels. The experiments show that in this case the theoretical speed of the machine determined by the angular velocity of rotation of the crankshaft is reduced by no more than 4 %. This effect is possible when the total resistance forces at movement of the machine \( \sum W + W_f \) are equal to the maximum traction force under conditions of adhesion of the driving propelling devices with the support surface:

\[
T_p = \sum W + W_f, \tag{1}
\]

where \( W_f \) — resistance to the machine rolling.

It should also be noted that at the stage of straight-line motion the dynamic model of this process is described by the following basic equation of dynamics

\[
m \ddot{x} = \sum T - (\sum W + W_f); \]

where \( x \) - coordinate of the horizontal displacement of the machine; \( \sum T \) - total tractive effort developed by the machine; \( m \) - mass of the motor grader.
Fig. 1 The process of forming the actual motion path of the motor-grader at asymmetric application of external resistance forces to the blade

However, at the moment of the unintentional turning the dynamic model changes and can be presented as follows (Fig. 2)

\[ I \ddot{\alpha} = M(T) - M(\sum W_i, f_i, P_i), \]

where \( \ddot{\alpha} \) - coordinate of the angular displacement of the machine; \( I \) - moment of inertia of the machine with respect to the center of turning; \( M(T) \) - torque developed by tractive efforts; \( M(\sum W_i, f_i, P_i) \) - torque developed by resistance forces.

The analysis of the application of external forces (Fig. 2) made it possible to determine the conditions for the transition from the straight-line motion of the machine to its unintentional turning:

\[
\begin{align*}
T_2 + T_3 &= \sum W + W_f \\
M(T) &= M(\sum W_i, f_i, P_i).
\end{align*}
\]

(2)

Figure 2 shows: \( P_{i_1} \) - forces of resistance to lateral displacement of the wheels; \( W_{up} \) - resistance to moving the prism of soil in front of the blade; \( W_p \) - resistance of soil to cutting; \( W_c \) - resistance to moving the cut layer of soil up the blade. The resultant vector of resistances acting on the blade is equal to

\[ \sum W = W_p + W_c + W_{up} \]

The center of turning of the machine is in the zone of the blade jamming in the soil. The torque turning the machine clockwise is calculated by the equation:

\[ M(T) = T_2 \cdot l_1 + T_3 \cdot (l + l_4). \]

The moment of resistance forces preventing the machine from turning,

\[ M(\sum W_i, f_i, P_i) = W_{f_1} \cdot \left( \frac{L^2}{2} + l_1 \right) + W_{f_2} \cdot l_1 + W_{f_3} \cdot (l + l_4) + P_{i_1} \cdot l_2 + (P_{i_2} + P_{i_3}) \cdot l_3 + W_{up} \cdot l_3 + (W_p + W_c) \cdot l_4. \]

Tractive efforts \( (T_i) \), forces of resistance to lateral displacement \( (P_{i_1}) \) and forces of resistance to rolling \( (W_p) \) are determined by certain common equations and depend on the value of the support reaction on the undercarriage [6].

Fig. 2 The scheme of the application of forces to the motor-grader in the horizontal plane at the unintentional turning of the motor-grader

The components of the resultant vector of resistance also can be determined by means of widely-known equations [7, 8].
\[ W_p = k \cdot F; \]
\[ W_{ap} = V_{ap} \cdot \frac{\delta_p}{k_p} \cdot g \cdot \mu_1; \]
\[ W_c = V_{ap} \cdot \frac{\delta_p}{k_p} \cdot g \cdot \mu_2 \cdot \cos^2 \delta. \]

In the presented dependences \( k \) - specific resistance of the developed material to cutting; \( F \) - area of the cut layer; \( V_{ap} \) - volume of the prism of the moved material; \( \delta_p \) - density of the material in the natural state; \( k_p \) - coefficient of loosening of the developed material; \( g \) - acceleration of gravity; \( \mu_1 \) - coefficient of internal friction of the material; \( \mu_2 \) - coefficient of external friction of the material against steel; \( \delta \) - cutting angle.

The tuning of the machine under condition (2) will last until the moment when the dependence (1) changes as follows:

\[ T_2 + T_3 \sum W + W_f, \]

after which the movement of the motor-grader along the straight-line trajectory continues.

The considered mathematical model allows to explain the character of the motor-grader motion depending on the application of forces at its loading. Along with this, the field researches conducted with the real motor-grader ДЗк – 251 on the Testing ground of Kharkiv National Automobile and Highway University allowed to identify a number of additional factors that also influence the parameters of the motor-grader road-holding ability at performing working operations \([1, 9]\). To such parameters, first of all, there can be attributed the turning angle of the driven wheels in the horizontal plane \((\gamma)\), the tilt angle of the front wheels in the vertical plane \((\rho)\), the angle of the transverse gradient of the support surface \((\varphi)\) and the coefficient of adhesion of the driving propelling devices with the support surface \((\varphi_s)\).

On the basis of the factorial experiment, a regression equation was obtained, which allows to determine the lateral displacement of the motor-grader \( H \) depending on the above mentioned factors for the same conditions of external loading:

\[ H(\varphi, \varphi_s, \gamma, \rho) = 2.6589 - 0.1538\varphi + 0.7015\varphi_s - 0.80095\gamma + 0.1441\rho + 0.4219\varphi + 0.0208\gamma + 0.0343\rho - 0.0355\gamma - 0.4967\rho + 0.0343\gamma + 0.0355\rho + 0.0948\varphi + 0.0064\gamma - 0.0033\varphi - 0.0124\varphi_s\rho. \]

In the course of the experiments it was revealed that some of factors have a destabilizing effect on the parameters of road-holding ability, and some, on the contrary, allow the machine to be kept on the planned trajectory. In particular, the turning of the driven wheels in the direction opposite to that of the unintentional turning of the motor-grader and the tilt of the same wheels in the same direction provide for eliminating the displacement of the motor-grader in the lateral direction even with significant torque acting on the machine in the horizontal plane. Unfortunately, currently there are no recommendations for choosing the angles \( \gamma \) and \( \rho \).

However the equation (3) made it possible to determine the dependence between the angles enabling to keep the machine on the straight-line course under the given conditions of asymmetric loading of the motor-grader.

Figures 3 and 4 present graphs of dependence of the lateral displacement of the motor-grader on the angles \( \gamma \) and \( \rho \) of the position of the driven wheels. The tables give rational dependence of the angles allowing to keep the machine on a straight-line trajectory of movement for the loading conditions realized in the experiment.

### Conclusions

The conducted analysis of the power asymmetric loading of the motor-grader in the process of performing working operations made it possible to determine the boundary conditions that change the nature of the vehicle motion and determine the form of its actual trajectory.

The field research made it possible to reveal the influence of additional factors on the formation of parameters of the road-holding ability. The proposed method allows to calculate the values of such parameters as the angle of rotation of the front wheels in the horizontal plane and the angle of their tilt in the vertical plane, allowing to keep the motor-grader on a straight trajectory of motion even with an asymmetric application of external loads. The obtained data can become a basis for the development of automatic systems ensuring the stability of the trajectory of the motor-grader.

### References


2. Пат. RU 2100528 Россия, МПК Е 02 F3/76. Автогрейдер с переменной массой / Воронович В.П., Задеев Е.П., Коробка Б.А., Плотин И.И., Приходько В.И.; владелец Открытое акционерное общество "Крюковский вагоностроительный завод". – №95116814/03; заявл. 29.09.1995; опубл. 27.12.1995, Бюл. №36.

3. Пат. RU 2399538 Россия, МПК В 62 D 37/04. Способ стабилизации положения колесного транспортного средства / Реймер В.В., Стеновский В.С., Черкасов А.А., Сорокин А.А., Асманкин Е.М.; владелец Федеральное государственное
4. Пат. SU 1614941, МПК В 60 K 17/30. Управляемый мост автогрейдера / Воронин А.Н., Антипов Л.А., Епифанов В.С., Беликов В.Ф., Островский С.М.; владелец Московское научно-производственное объединение по строительному и дорожному машиностроению ВНИИстройдормаш. – № 4630828/25-1; заявл. 03.01.1989; публ. 23.12.1990, Бюл. №47.


ECOLOGICAL CONTRIVANCE FOR ANALYSIS AND MANAGING THE PRODUCTION OF BIOETHANOL AS A FUEL FOR INTERNAL COMBUSTION ENGINES

PhD Student Dzhelil Y. R.1, Ass. Nikolova D. St.2, Prof. Ivanov B. B.1
Bulgarian Academy of Science, Institute of Chemical Engineering, Sofia, Bulgaria
Prof. d-r. Asen Zlatarov University, Burgas, Bulgaria
unzile_20@abv.bg

Abstract: The energy crisis and environmental contamination are two of the main problems of the 21st century. Atmospheric contamination from transport is well-studied case of environmental impact and is associated with significant health consequences. The biofuels use is able to reduce both pollution emitted by transport (CO2 and other damaging substances into the atmosphere, contributing to global warming) and the reduction of energy dependence on fossil fuels (gasoline, diesel, etc.). European Union requires their use to reach -10% by 2020. Object of our study is bioethanol, which has great potential to replace gasoline. It was discussed the environmental technique for analyzing and managing the production of bioethanol, aiming to explore all stages of the production chain. It includes a selection of feedstock, site of its cultivation and transport, area for the construction of biorefinery, technology for production and distribution.

Keywords: BIOETHANOL, ENVIROMENTAL ACTION, PRODUCTION TECHNOLOGIES, GHG EMISSION, COORDINATION

1. Introduction

Bioethanol has been used in Germany and France in 1894. Availability of cheap fuel does not allow its production on an industrial scale until it occurs energy crisis and increasing emissions of greenhouse gases that renewed interest in bioethanol and its use [1]. The European Union adopted Directive in 2008, which obliges all member states to use ethanol in a mixture of gasoline as its use gradually increased to reach 10% to 2020. The use of bioethanol would lead both to reduce emissions leading to greenhouse effect - CO2, NOx, CO2 (the combustion of bioethanol is released so amount of CO2 as it was absorbed by the plant in its photosynthesis) and to reduce dependence on gasoline.

Bioethanol consumption in the EU transportation sector is given in Fig. 1 [2]. In Bulgaria the quantity of bioethanol is equal to 5.3% in 2016.

Fig. 1 Bioethanol consumption in the transportation sector in recent years.

2. Problem Statement

The problem described in this paper aims to present ecological technique for analyzing and managing the production of bioethanol, building optimal Supply Chain (SC) for bioethanol which minimizes chain costs, the impact on the environment by making appropriate compromises.

3. Bioethanol as a Transport Fuel

The Bioethanol represents ethyl alcohol which can be obtained from cultures containing sucrose (sugar beet, sugar cane, sweet sorghum and sweet potato), starch (wheat, corn, barley, rice and potatoes) and lignocellulose raw material (corn cobs, straw, and wood / forest waste). It has a higher octane rating, a wider range of high burning rate. Octane number is a standard measure of resistance to detonation motor gasoline and it is conditional unit showing the tendency to the detonation of the test sample, compared with a penchant for detonation of accepted standards. The fuel can withstand higher compression prior to detonate at the higher the octane number.

Bioethanol can be blended in combination with gasoline in different proportions depending on the desired effect. Common proportions: E5, E10, E20, E25, E70, E85, which is a high-level ethanol-gasoline blend containing 85% ethanol and 15% gasoline. E85 and mixtures even with higher concentrations of bioethanol, such as E95, have been considered as alternative fuels. Vehicles fueled called E85 vehicles adaptability to fuel - fuel flexible vehicles (FFVs) and they are available from some major vehicle manufacturers.

Table 1: Fuels properties.

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Gasoline</td>
<td>0.76</td>
<td>0.6</td>
<td>&lt;21</td>
<td>42.7</td>
<td>32.45</td>
<td>92</td>
<td>1.00</td>
</tr>
<tr>
<td>Bioethanol</td>
<td>0.79</td>
<td>1.5</td>
<td>&lt;21</td>
<td>26.8</td>
<td>21.17</td>
<td>&gt;100</td>
<td>0.65</td>
</tr>
</tbody>
</table>

4. Feedstocks for Bioethanol Production

In compliance with the EU Directive it is necessary to provide as available arable land earmarked for the cultivation of biomass and new areas to increase the amount of received and used bioethanol. It should be borne in mind also the geographical
location of biorefinery, the local soil conditions and biomass yield per hectare. Different types of biomass with their production capacity are given in the Table. 2 [4].

<table>
<thead>
<tr>
<th>Biomass</th>
<th>Production potential of bioethanol (l/t)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Sugar cane</td>
<td>70</td>
</tr>
<tr>
<td>Sugar beet</td>
<td>110</td>
</tr>
<tr>
<td>Potatoes</td>
<td>110</td>
</tr>
<tr>
<td>Corn</td>
<td>360</td>
</tr>
<tr>
<td>Rice</td>
<td>430</td>
</tr>
<tr>
<td>Barley</td>
<td>250</td>
</tr>
<tr>
<td>Wheat</td>
<td>340</td>
</tr>
<tr>
<td>Sweet Sorghum</td>
<td>60</td>
</tr>
</tbody>
</table>

Feedstocks containing sugar can easily be converted into bioethanol by fermentation, and those that contain starch are made up of many glucose molecule chains which are cleaved to produce the sugars and fermenting. Bioethanol production from lignocellulosic biomass is more difficult technology, because they consist primarily of lignin, cellulose, proteins, fats, waxes, and a small amount of starch, sugars, proteins and lignin.

Global Bioethanol production in recent years is illustrates in fig. 2. The figure shows that its production gradually increases [5].

5. Bioethanol Production

Bioethanol must be produced in compliance with the following requirements to be clean:
- In the cultivation of biomass to avoid land rich in carbon, as the plant will take it and release it during the combustion.
- By-products must be recovered efficiently and reused.
- To reduce those emissions throughout the entire life cycle of production and use of this fuel. Emissions take place as from cultivation, collection, and transportation and in their conversion to bioethanol and logistics of finished fuel.
- To build a SC for bioethanol by which to choose the optimal technology for its receipt, and the selection of feedstock as well as the locations where they are installed biorefineries with their production capacities to make the necessary compromises.

An environmental SC should consider all stages of production of the fuel: the choice of raw material to its incineration and disposal of waste products. It also considers the emissions released in each stage of production and lead to their reduction by choosing the optimal of all possible feedstocks and technologies.

SC analyzes the different stages of the production of bioethanol. It provides the ability to manage manufacturing processes, including the selection of the most favorable environmental and economic criteria. Development of appropriate SC represents a powerful tool for ensuring sustainable development of modern energy. This includes the joint implementation of the criteria of economy and ecology. It is also possible to include social decisions.

Ecological technique for analysis and management of production of bioethanol (as fuel for internal combustion engines), including design of appropriate SC represents the development of a complex hierarchical structure of decision-making. The main stages of the production of bioethanol are:

- **Choice of biomass** - it is selected depending on the geographical location and climatic conditions. At this stage are chosen and tips and techniques for harvesting, as well as space for its storage and transport (selected type of transport and routing).

- **Choice of areas for building biorefineries** - depending on areas of growing biomass.

- **Choice of technology for conversion, here we have to choose the following stages:**
  - **Pre-treatment** - it depends on the type of the biomass. The sugar feedstocks in this step are cleaned, crushed and is carried out extracting sugars and juices processing. First the starch materials are pulverized and subjected to hydrolysis by a suitable enzyme for the cleavage of the glucose molecules:
    \[ (C_{6}H_{12}O_{6})_{n} + nH_{2}O \rightarrow nC_{6}H_{12}O_{6} \]

  - **D-glucose**

  The step pre-treatment of the lignocellulosic biomass required to separate holocellulose of lignin and further its conversion to glucose. The aim of pretreatment is to break down the structure of lignin and cellulose, which will improve access of enzymes to the pulp during the hydrolysis stage.

  Suitable treatments of lignocellulose are the following: a physical pre-treatment (steam physical treatment, milling, irradiation, temperature and pressure), which would increase the available surface area, pore size and would result in a reduction of crystallinity and degree of polymerization of cellulose, physical-chemical (including automatic hydrolysis), chemical (ozonolysis, alkaline and acid) and organic. After this treatment is carried out the hydrolysis, which can be chemical or enzymatic;

  - **Diluting the sugars in water and then adding yeast;**
  - **Fermentation** - at this stage must be selected type yeast / enzymes;
  - **Distillation** – for remove water;
  - **Dehydration** - drying of bioethanol as it can be used as fuel for internal combustion engines, here choose the appropriate methods for dehydration;

  **Choice of transport and route for transporting of a ready fuel to places for blending with gasoline**

  **Selection of areas for construction of installations for mixing of bioethanol with gasoline**

  **Logistics** - choice of transport route for transportation of bioethanol-gasoline blend to the gas stations.

  The stages of the production of bioethanol vary depending on the type of feedstock.
### Table: Ethanol production steps by feedstock and conversion technique

<table>
<thead>
<tr>
<th>Feedstock</th>
<th>Feedstock conversion to sugar</th>
<th>Process heat</th>
<th>Sugar conversion to alcohol</th>
<th>Co-product</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cane</td>
<td>Sugars extracted through bagasse crushing, soaking, chemical treatment</td>
<td>Typically from fossil fuel</td>
<td>Fermentation and distillation of alcohol</td>
<td>Heat, electricity, molasses</td>
</tr>
<tr>
<td>Sugar beet</td>
<td>Sugar extraction</td>
<td>Typically from fossil fuel</td>
<td>Fermentation and distillation of alcohol</td>
<td>Animal feed, fertilizer</td>
</tr>
<tr>
<td>Wheat</td>
<td>Starch separation, milling, conversion to sugars via enzyme application</td>
<td>Typically from fossil fuel</td>
<td>Fermentation and distillation of alcohol</td>
<td>Animal feed (e.g. distillers dried grains)</td>
</tr>
<tr>
<td>Corn</td>
<td>Starch separation, milling, conversion to sugars via enzyme application</td>
<td>Typically from fossil fuel</td>
<td>Fermentation and distillation of alcohol</td>
<td>Animal feed, industrial use</td>
</tr>
<tr>
<td>Potatoes</td>
<td>Washing, mashing, cooking, starch separation, conversion to sugars via enzyme application</td>
<td>Typically from fossil fuel</td>
<td>Fermentation and distillation of alcohol</td>
<td>Animal feed, industrial use</td>
</tr>
<tr>
<td>Trees</td>
<td>Cellulose conversion to sugar via saccharification (enzymatic hydrolysis)</td>
<td>Lignin and excess cellulose</td>
<td>Fermentation and distillation of alcohol</td>
<td>Heat, electricity, animal feed, bioplastics, etc.</td>
</tr>
<tr>
<td>Forest waste</td>
<td>Cellulose conversion to sugar via saccharification (enzymatic hydrolysis)</td>
<td>Lignin and excess cellulose</td>
<td>Fermentation and distillation of alcohol</td>
<td>Heat, electricity, animal feed, bioplastics, etc.</td>
</tr>
</tbody>
</table>

We must keep the following criteria for Environmental and economical production of bioethanol:

**Economic criteria** include: the costs of handling, transportation and storage of biomass, transport costs of biomass from store to store to biorefinery, costs for its transformation, transportation costs of biofuels by blending facility and the transport cost to transporting of finished product to warehouses for storage.

\[ (1) \quad IK = IR + IBH + IBT + IBI + IBD + IP + IPI + IPT + IPC \]

where,

- \( IR \) - annual capital costs;
- \( IBH \) - total cost of harvest, (€/kg);
- \( IBT \) - total transport cost of biomass, (€/kg);
- \( IBI \) - storage costs of the biomass, (€/kg);
- \( IBD \) - handling costs of the biomass, (€/kg);
- \( IP \) - production costs, (€/kg);
- \( IPI \) - cost of fuel storage, (€/l);
- \( IPT \) - general transportation costs of bioethanol, (€/l);
- \( IPC \) - Government incentives for biofuel production, (€/l);

**Environmental criteria**, its aims to minimize the total annual amount of greenhouse gases (GHG) emitted during the lifecycle of SC.

The three main greenhouse gases emitted by a SC are methane (CH₄), nitrous oxide (N₂O) and carbon dioxide (CO₂).

Total GHG emissions are measured from the point of view of CO₂ - equivalent emissions (CO₂-eq). Greenhouse gases are 1g CH₄ = 25g CO₂-eq, and 1g N₂O = 298g CO₂-eq according to the latest assessment report of the Intergovernmental Panel on Climate Change [6].

Emissions separated from bioethanol production are illustrated in Fig. 3.

\[ (2) \quad EK = Kbh + Kbd + Kbi + Kbt + Kp + Kpt + Kpm + KBcar + KGcar \]

where,

- \( Kbh \) – The emission of cultivating and obtaining biomass, (kg CO₂-eq/ kg biomass);
- \( Kbd \) – The emission of drying unit amount of biomass, (kg CO₂-eq/ kg biomass);
- \( Kbi \) – The emission of producing unit amount biofuel from biomass, (kg CO₂-eq/ kg biomass);
- \( Kbt \) – The emission of transporting unit amount biomass, (kg CO₂-eq/ kg biomass);
- \( Kp \) – The emission of producing unit amount biofuel from biomass, (kg CO₂-eq/ kg biomass);
- \( Kpt \) – The emission of transporting unit amount biofuel, (kg CO₂-eq/ l);
- \( Kpm \) – The emission of blending and distributing unit amount of biofuel, (kg CO₂-eq/ gallon);
- \( KBcar \) – The emission emitted using biofuels in transport, (kg CO₂-eq/ l);
- \( KGcar \) – The emission emitted using conventional fuels in transport, (kg CO₂-eq/ l).

### 6. Conclusion

Biofuel production is expected to quickly develop during the next decades. Biofuel production is expected to develop rapidly in the coming decades because of growing environmental pollution, the inefficient use of energy and energy crisis. Energy future of the world is inextricably linked with the development of techniques for the controlled production of biofuels.

This paper presents an approach for Bioethanol Supply Chain on economic and environmental criteria, taking into account the main characteristics of biofuels such as seasonality for supply of raw materials, geographic diversity and availability of biomass, other conversion technologies, recycling of by-products, distribution demand, and regional economic situation. Presented model allows minimizing the economic costs and the reduction of...
harmful emissions released in the chain by making the necessary compromises. The design of optimal SC for biofuel can solve a wide range of issues related to biofuels, because this area changes very fast (not only economic but also includes strategic decisions relating to domestic consumption or export the produced biofuel producing region biomass, etc.).

The analysis and production management presented the used area is flexibility and the ability to solve many problems simultaneously by the methods of mathematical modeling.

7. Acknowledgements

The authors would like to thank Bulgarian National Science Fund for the obtained financial support under contract DN 07-14/15.12.2016.

8. Literature

ENERGY FLOW STREAMS IN THE MARINE STEAM PLANT DURING THE MAIN PROPULSION PROPELLER SPEED VARIATION

PhD Student Eng. Igor Poljak1, PhD. Vedran Mrzljak2, PhD. Josip Orovic3, Rožići 4/3, 51221 Kostrena1, Faculty of Engineering, University of Rijeka, Vukovarska 58, 51000 Rijeka2, University of Zadar, Maritime Department, Mihovila Pavlinovića 1, 23000 Zadar3, Croatia
E-mail: igor.poljak2@gmail.com, vedran.mrzljak@riteh.hr, jorovic@unizd.hr

Abstract: For the analyzed LNG carrier with Rankine regenerative feed water cycle, energy analysis is presented. The intention was to determine auxiliary flow streams from the useful ones. Auxiliary energy flow supports plant operation on the one side, but reduces overall efficiency on the other. Steam generators energy streams are divided into two major groups, for the main and auxiliary stream consumers. The steam system test was performed by varying main propulsion shaft revolutions. The required thermodynamic data are collected at various steam system locations. For considered plant energy flow streams components are explained and analyzed. In this paper the recommendations for possible energy savings for the mentioned propulsion plant are proposed.

Keywords: MARINE STEAM PLANT, MACHINERY, ENERGY FLOW, ENERGY SAVING

1. Introduction

Due to interpretation variability of power and mass flow of the superheated Rankine cycle, it has risen necessity for clarifying energy motion inside the marine steam plant at LNG carrier.

Although stationary steam power plants were well being elaborated by many authors, marine steam plants have not been researched in that way yet.

In the context of electricity production, auxiliary load is the load or device which consumes electricity while contributing to the process of electricity generation or plant operation. Hence, this does not add to the total plant yield, but reduces the gross yield by a considerable amount. Thus, auxiliary load is the in-facility electrical load which needs to be minimized [1], [2].

In marine power plant, the main auxiliary loads include: motors used to run various service pumps which support plant operation, fans for the boilers, turbine turning gear in port, electric pre-heaters, etc. In addition to that, instrumentation, controls, computers, valve actuators, air compressors and lighting within the power plant also add on to the auxiliary load.

Electrical power which is generated on board the vessel for supporting plant operation is gained from turbo generator units. Beside to auxiliary load, LNG carrier consumes steam, which is used for lube oil heaters, fuel oil heaters, waste oil tank heaters, accommodation services heaters, etc. This amount of steam which is taken from the boiler contributes to efficiency degradation of the total plant cycle.

Considered marine steam plant overview is given in Fig.1. Explanation of all symbols from Fig.1 is presented in Table 1.

Table 1. Marine steam plant chain indexes

<p>| | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Main boiler</td>
</tr>
<tr>
<td>2</td>
<td>High pressure turbine</td>
</tr>
<tr>
<td>3</td>
<td>Low pressure turbine</td>
</tr>
<tr>
<td>4</td>
<td>Turbo generator No1</td>
</tr>
<tr>
<td>5</td>
<td>Turbo generator No2</td>
</tr>
<tr>
<td>6</td>
<td>Main condenser</td>
</tr>
<tr>
<td>7</td>
<td>Main condenser cooling sea water pump</td>
</tr>
<tr>
<td>8</td>
<td>Condensate pump</td>
</tr>
<tr>
<td>9</td>
<td>Fresh water generator</td>
</tr>
<tr>
<td>10</td>
<td>Gland steam cooler</td>
</tr>
<tr>
<td>11</td>
<td>Low pressure feed water heater</td>
</tr>
<tr>
<td>12</td>
<td>Auxiliary condensate pump</td>
</tr>
<tr>
<td>13</td>
<td>Deaerator</td>
</tr>
<tr>
<td>14</td>
<td>Feed water pump</td>
</tr>
<tr>
<td>15</td>
<td>Feed water pump turbine</td>
</tr>
<tr>
<td>16</td>
<td>High pressure feed water heater</td>
</tr>
<tr>
<td>17</td>
<td>De super heater</td>
</tr>
<tr>
<td>18</td>
<td>Water heater</td>
</tr>
<tr>
<td>19</td>
<td>LNG heater</td>
</tr>
<tr>
<td>20</td>
<td>Fuel oil heater</td>
</tr>
<tr>
<td>21</td>
<td>Contaminated service heaters</td>
</tr>
<tr>
<td>22</td>
<td>Contaminated service cooler</td>
</tr>
<tr>
<td>23</td>
<td>Condensate cooler</td>
</tr>
</tbody>
</table>

2. Numerical model

Elaborated power plant uses regenerative feed water cycle in order to increase cycle efficiency. The feed water group will not be considered in this study. Plant flows are divided into two main stream groups from the main boilers (plant consists of two identical steam boilers): superheated and the de superheated streams. Superheated stream may be divided into four sub streams: steam flow to the main turbine, steam flow to turbo generators No1 and No2 and steam flow to feed pump steam turbine. There is one additional superheated sub stream which relates to the losses and is noted as an additional superheated sub stream. The stream flows from the main boilers carry energy and mass notations. De superheated steam stream flow to the service consumers is noted in the opposite direction, as presented in Fig.2.
3. Thermodynamic analysis

Pressure, temperature, and power are measured with standard plant measuring equipment, which is displayed in Table 2.

Table 2. Marine steam plant measuring equipment

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>- Turbo generator and feed pump steam turbine inlet steam pressure</td>
<td>- Main propulsion turbine inlet steam pressure</td>
<td>- Main boiler de superheating outlet steam pressure</td>
<td>- Main propulsion turbine inlet steam temperature</td>
<td>- Main propulsion turbine shaft power</td>
<td>- Turbo generators power</td>
</tr>
</tbody>
</table>

Thermodynamic results were obtained from the measured data at various locations in the engine room. Individual stream flows to dedicated direction is calculated according to [8], [9].

In the steady state process the mass balance of control volume is calculated according to:

$$\sum m_i = \sum m_o$$

(1)

The energy balance of the control volume is presented as:

$$\sum E_i + \dot{Q} = \sum E_o + \dot{W}$$

(2)

Table 3. specifies energy and mass stream flow calculation procedure for the steam plant components observed in this analysis.

Table 3. Mass and energy stream flow calculation routines

<table>
<thead>
<tr>
<th>Mass flow</th>
<th>Energy flow</th>
</tr>
</thead>
<tbody>
<tr>
<td>$m_{TG1} = \dot{m}<em>{TG1} \cdot h</em>{TG1}$</td>
<td>$E_{TG1} = (m_{TG1} \cdot h_{TG1})$</td>
</tr>
<tr>
<td>$m_{TG2} = \dot{m}<em>{TG2} \cdot h</em>{TG2}$</td>
<td>$E_{TG2} = (m_{TG2} \cdot h_{TG2})$</td>
</tr>
<tr>
<td>$m_{FP} = \dot{m}<em>{FP} \cdot h</em>{FP}$</td>
<td>$E_{FP} = (m_{FP} \cdot h_{FP})$</td>
</tr>
<tr>
<td>$m_{MT} = \dot{m}<em>{MT} \cdot h</em>{MT}$</td>
<td>$E_{MT} = (m_{MT} \cdot h_{MT})$</td>
</tr>
<tr>
<td>$m_{SE} = \dot{m}<em>{SE} \cdot h</em>{SE}$</td>
<td>$E_{SE} = (m_{SE} \cdot h_{SE})$</td>
</tr>
<tr>
<td>$m_{LO} = \dot{m}<em>{LO} \cdot h</em>{LO}$</td>
<td>$E_{LO} = (m_{LO} \cdot h_{LO})$</td>
</tr>
</tbody>
</table>

Specific enthalpy of every steam flow was calculated by using measured pressures and temperatures.

Cumulative energy flow from main boilers to all observed steam plant components can be defined with:

$$\sum ALL = \dot{m}_{MT} \cdot h_{MT} + 2 \cdot \dot{m}_{TG} \cdot h_{TG} + \dot{m}_{FP} \cdot h_{FP} + \dot{m}_{SE} \cdot h_{SE} + \dot{m}_{LO} \cdot h_{LO}$$

(3)

It is important to emphasize that both turbo generators have identical mass flows and identical inlet pressures and temperatures (consequently identical inlet specific enthalpies).

Ratio of cumulative energy flow stream distributed to the observed components is defined by the equations:

- Main turbine:

$$E_{MT} = \frac{\dot{m}_{MT} \cdot h_{MT}}{\sum ALL} \cdot 100\%$$

(4)

- Turbo generator No1 and No2:

$$E_{TG1} = \frac{\dot{m}_{TG1} \cdot h_{TG1}}{\sum ALL} \cdot 100\%$$

(5)

$$E_{TG2} = \frac{\dot{m}_{TG2} \cdot h_{TG2}}{\sum ALL} \cdot 100\%$$

(6)

- Feed pump steam turbine:

$$E_{FP} = \frac{\dot{m}_{FP} \cdot h_{FP}}{\sum ALL} \cdot 100\%$$

(7)

- Service:

$$E_{SE} = \frac{\dot{m}_{SE} \cdot h_{SE}}{\sum ALL} \cdot 100\%$$

(8)

- Losses:

$$E_{LO} = \frac{\dot{m}_{LO} \cdot h_{LO}}{\sum ALL} \cdot 100\%$$

(9)

4. Analysis and discussion

The engine run test was made after cargo loading operation. Steam energy stream flows during variation of revolutions at the main propulsion shaft are presented in the Table 4.

Table 4. Energy flow streams under speed variation of the main propulsion shaft

<table>
<thead>
<tr>
<th>Main shaft RPM</th>
<th>Main turbine (kW)</th>
<th>Turbo generator No1 (kW)</th>
<th>Turbo generator No2 (kW)</th>
<th>Feed pump (kW)</th>
<th>Service (kW)</th>
<th>Losses (kW)</th>
</tr>
</thead>
<tbody>
<tr>
<td>25.58</td>
<td>3606</td>
<td>4389</td>
<td>4389</td>
<td>3035</td>
<td>24305</td>
<td>223</td>
</tr>
<tr>
<td>34.33</td>
<td>9060</td>
<td>4424</td>
<td>4424</td>
<td>3079</td>
<td>22431</td>
<td>240</td>
</tr>
<tr>
<td>41.78</td>
<td>15645</td>
<td>4302</td>
<td>4302</td>
<td>3039</td>
<td>14242</td>
<td>225</td>
</tr>
<tr>
<td>53.50</td>
<td>33082</td>
<td>4470</td>
<td>4470</td>
<td>3205</td>
<td>9707</td>
<td>289</td>
</tr>
<tr>
<td>56.65</td>
<td>7214</td>
<td>3779</td>
<td>3779</td>
<td>3143</td>
<td>12230</td>
<td>286</td>
</tr>
<tr>
<td>61.45</td>
<td>35697</td>
<td>3877</td>
<td>3877</td>
<td>2994</td>
<td>10782</td>
<td>300</td>
</tr>
<tr>
<td>62.52</td>
<td>35474</td>
<td>3782</td>
<td>3782</td>
<td>2989</td>
<td>10881</td>
<td>298</td>
</tr>
<tr>
<td>63.55</td>
<td>38090</td>
<td>3928</td>
<td>3928</td>
<td>3032</td>
<td>11218</td>
<td>316</td>
</tr>
<tr>
<td>65.10</td>
<td>41022</td>
<td>3626</td>
<td>3626</td>
<td>3063</td>
<td>11537</td>
<td>320</td>
</tr>
<tr>
<td>66.68</td>
<td>42496</td>
<td>3663</td>
<td>3663</td>
<td>3092</td>
<td>11909</td>
<td>342</td>
</tr>
<tr>
<td>67.68</td>
<td>45808</td>
<td>3562</td>
<td>3562</td>
<td>3146</td>
<td>12440</td>
<td>360</td>
</tr>
<tr>
<td>68.66</td>
<td>47362</td>
<td>3607</td>
<td>3607</td>
<td>3149</td>
<td>10914</td>
<td>359</td>
</tr>
<tr>
<td>69.49</td>
<td>48755</td>
<td>3595</td>
<td>3595</td>
<td>3177</td>
<td>11258</td>
<td>368</td>
</tr>
<tr>
<td>70.37</td>
<td>50270</td>
<td>3599</td>
<td>3599</td>
<td>3198</td>
<td>11278</td>
<td>377</td>
</tr>
<tr>
<td>71.03</td>
<td>50832</td>
<td>3619</td>
<td>3619</td>
<td>3206</td>
<td>11187</td>
<td>379</td>
</tr>
<tr>
<td>73.09</td>
<td>56002</td>
<td>3669</td>
<td>3669</td>
<td>3283</td>
<td>11760</td>
<td>408</td>
</tr>
<tr>
<td>74.39</td>
<td>62370</td>
<td>3767</td>
<td>3767</td>
<td>3314</td>
<td>8495</td>
<td>422</td>
</tr>
<tr>
<td>76.56</td>
<td>67454</td>
<td>3883</td>
<td>3883</td>
<td>3388</td>
<td>8660</td>
<td>449</td>
</tr>
<tr>
<td>78.41</td>
<td>73766</td>
<td>3926</td>
<td>3926</td>
<td>3472</td>
<td>8580</td>
<td>482</td>
</tr>
<tr>
<td>79.46</td>
<td>79287</td>
<td>4199</td>
<td>4199</td>
<td>3457</td>
<td>2620</td>
<td>476</td>
</tr>
<tr>
<td>80.44</td>
<td>82766</td>
<td>4465</td>
<td>4465</td>
<td>3509</td>
<td>2704</td>
<td>495</td>
</tr>
<tr>
<td>81.49</td>
<td>85339</td>
<td>4156</td>
<td>4156</td>
<td>3488</td>
<td>382</td>
<td>491</td>
</tr>
<tr>
<td>82.88</td>
<td>90640</td>
<td>4193</td>
<td>4193</td>
<td>3557</td>
<td>369</td>
<td>518</td>
</tr>
<tr>
<td>83.00</td>
<td>91766</td>
<td>4250</td>
<td>4250</td>
<td>3574</td>
<td>371</td>
<td>525</td>
</tr>
</tbody>
</table>

The energy stream flow ratio of the specified stream flows, to observed machineries or systems, according to the previously explained calculation procedure is presented in the Table 5.
Presentation of main and auxiliary steam energy flows with variations of main propulsion shaft speed is given in Fig.3. Only flow which is related to the ship propulsion is main (useful) flow. For the observed ship steam plant, main energy flow is only one - energy flow to the main turbine. Other observed energy flows are considered as auxiliary flows.

<table>
<thead>
<tr>
<th>Main shaft RPM (min⁻¹)</th>
<th>Main turbine (%)</th>
<th>Turbo generator No1 (%)</th>
<th>Turbo generator No2 (%)</th>
<th>Feed pump (%)</th>
<th>Service (%)</th>
<th>Losses (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>25.58</td>
<td>9.02</td>
<td>10.99</td>
<td>10.99</td>
<td>7.60</td>
<td>60.84</td>
<td>0.56</td>
</tr>
<tr>
<td>34.33</td>
<td>20.75</td>
<td>10.13</td>
<td>10.13</td>
<td>7.05</td>
<td>51.38</td>
<td>0.55</td>
</tr>
<tr>
<td>41.78</td>
<td>37.47</td>
<td>10.30</td>
<td>10.30</td>
<td>7.28</td>
<td>34.11</td>
<td>0.54</td>
</tr>
<tr>
<td>53.50</td>
<td>59.91</td>
<td>8.09</td>
<td>8.09</td>
<td>5.80</td>
<td>17.58</td>
<td>0.52</td>
</tr>
<tr>
<td>56.65</td>
<td>33.90</td>
<td>7.51</td>
<td>7.51</td>
<td>6.25</td>
<td>24.30</td>
<td>0.53</td>
</tr>
<tr>
<td>61.45</td>
<td>62.05</td>
<td>6.74</td>
<td>6.74</td>
<td>5.20</td>
<td>18.74</td>
<td>0.52</td>
</tr>
<tr>
<td>62.52</td>
<td>62.01</td>
<td>6.61</td>
<td>6.61</td>
<td>5.22</td>
<td>19.02</td>
<td>0.52</td>
</tr>
<tr>
<td>63.55</td>
<td>62.95</td>
<td>6.49</td>
<td>6.49</td>
<td>5.01</td>
<td>18.54</td>
<td>0.52</td>
</tr>
<tr>
<td>65.10</td>
<td>64.92</td>
<td>5.74</td>
<td>5.74</td>
<td>4.85</td>
<td>18.24</td>
<td>0.52</td>
</tr>
<tr>
<td>66.08</td>
<td>65.21</td>
<td>5.62</td>
<td>5.62</td>
<td>4.74</td>
<td>18.28</td>
<td>0.52</td>
</tr>
<tr>
<td>67.68</td>
<td>66.51</td>
<td>5.17</td>
<td>5.17</td>
<td>4.57</td>
<td>18.06</td>
<td>0.52</td>
</tr>
<tr>
<td>68.66</td>
<td>68.64</td>
<td>5.23</td>
<td>5.23</td>
<td>4.56</td>
<td>15.82</td>
<td>0.52</td>
</tr>
<tr>
<td>69.49</td>
<td>68.91</td>
<td>5.08</td>
<td>5.08</td>
<td>4.49</td>
<td>15.91</td>
<td>0.52</td>
</tr>
<tr>
<td>70.37</td>
<td>69.51</td>
<td>4.98</td>
<td>4.98</td>
<td>4.42</td>
<td>15.59</td>
<td>0.52</td>
</tr>
<tr>
<td>71.03</td>
<td>69.78</td>
<td>4.97</td>
<td>4.97</td>
<td>4.40</td>
<td>15.36</td>
<td>0.52</td>
</tr>
<tr>
<td>73.09</td>
<td>71.08</td>
<td>4.66</td>
<td>4.66</td>
<td>4.17</td>
<td>14.92</td>
<td>0.52</td>
</tr>
<tr>
<td>74.59</td>
<td>75.94</td>
<td>4.59</td>
<td>4.59</td>
<td>4.03</td>
<td>10.34</td>
<td>0.51</td>
</tr>
<tr>
<td>76.56</td>
<td>76.90</td>
<td>4.43</td>
<td>4.43</td>
<td>3.86</td>
<td>9.87</td>
<td>0.51</td>
</tr>
<tr>
<td>78.41</td>
<td>78.35</td>
<td>4.17</td>
<td>4.17</td>
<td>3.69</td>
<td>9.11</td>
<td>0.51</td>
</tr>
<tr>
<td>79.46</td>
<td>84.13</td>
<td>4.46</td>
<td>4.46</td>
<td>3.67</td>
<td>2.78</td>
<td>0.50</td>
</tr>
<tr>
<td>80.44</td>
<td>84.11</td>
<td>4.54</td>
<td>4.54</td>
<td>3.57</td>
<td>2.75</td>
<td>0.50</td>
</tr>
<tr>
<td>81.49</td>
<td>87.07</td>
<td>4.24</td>
<td>4.24</td>
<td>3.56</td>
<td>0.39</td>
<td>0.50</td>
</tr>
<tr>
<td>82.88</td>
<td>87.60</td>
<td>4.05</td>
<td>4.05</td>
<td>3.44</td>
<td>0.36</td>
<td>0.50</td>
</tr>
<tr>
<td>83.00</td>
<td>87.62</td>
<td>4.06</td>
<td>4.06</td>
<td>3.41</td>
<td>0.35</td>
<td>0.50</td>
</tr>
</tbody>
</table>

Higher relative energy stream flow consumption of the feed pump steam turbine, Fig.4, occurs due to fact that feed pump is recycling certain amount of the water at low plant loads in order to protect itself from the cavitation which will be caused by water overheating through the pump stages at reduced flow. Turbo generator higher energy consumption in the lower propulsion shaft operating zone is caused by the bow thruster run and related electrical consumption.

Conducted analysis shows that 12.38% of the cumulative energy ratio goes for auxiliary requirements at main propulsion shaft speed of 83 min⁻¹, Fig.5. The steam propulsion system load is directly proportional to main shaft speed, so at the highest observed steam system load the majority of energy flow from steam generators goes to the main turbine.

It may be seen that feed pump steam turbine power consumption at 83 min⁻¹ is 3.41%, what is lower in comparison with the similar auxiliary steam turbines in stationary steam plants for the rated loads. The main difference is lower pumping pressure as marine boiler MB-4-KS is designed to operate at steam pressure of 6 MPa, in that respect pumping power is lower than, for example, of a boiler which is operating at 16 MPa.

Considering the overall developed power, steam turbine for a feed pump drive in this mode (83 min⁻¹) is fed with a greater energy flow than required. The reason for that fact can be found in deaerator operation. From the observed steam system, steam after the feed pump turbine is drained to the deaerator. At the highest steam system operating load, calculated energy flow ratio to the feed pump steam turbine is not in function to produce high power of the turbine than that to ensure smooth deaerator operation.

When compared energy ratios at the main propulsion shaft speed of 25 min⁻¹ and 83 min⁻¹ it is obvious from Fig.4 and Fig.5 that energy flow ratio, which is leading to both turbo generators and feed pump steam turbine decreases as steam system load increases. On the other side, the energy flow ratio of the main turbine has the opposite pattern of behavior in comparison with the turbo generators and feed pump steam turbine. Energy flow ratio of the main turbine increases during increase in steam system load.

Finally, it is interesting to mention that steam system losses energy ratio decreases from the lowest to the highest observed main propulsion shaft speeds.
When comparing auxiliary with the main turbine energy flows for the observed steam system, it is interesting to present ratio according to Table 4. Values from Table 4 for all auxiliary energy flows are summarized and divided by the energy flow to the main turbine, at each observed main propulsion shaft speed. Fig. 6 presents that this ratio constantly decreases from 100% at the main shaft speed of 25 min^{-1} to the 14% at the main shaft speed of 83 min^{-1}.

**Fig. 5. Energy ratio in (%) at the main propulsion shaft speed of 83 min^{-1}**

**Fig. 6. Auxiliary requirement vs main turbine ratio for energy stream flows**

**5. Conclusion**

Described marine steam plant generates more auxiliary energy flows at lower main propulsion shaft speeds than on the higher, so the efficiency of the plant at lower operation speeds will be lower.

As for recommendation, reducing of auxiliary flow may be achieved with higher speed of the LNG carrier and one turbo generator operation after manoeuvring zone. Moreover, the steam driven feed pump turbine has to be changed with electrically driven pump, what will act beneficially to turbo generator specific steam consumption.

Additionally, the high rate feed pump should be changed to a lower rate feed pump in port and during manoeuvring zone as to avoid recycling of feed water, what will cause decreased auxiliary load consumption either steam or electrical power.

Further studies should include the profitability of installation of variable speed driven pump electro motors, which could further decrease auxiliary power consumption on given example.