

INTERNATIONAL SCIENTIFIC JOURNAL

trans & MOTAUTO WORLD

YEAR IV, ISSUE 1/2019

ISSN PRINT 2367-8399

ISSN WEB 2534-8493



SCIENTIFIC TECHNICAL UNION OF MECHANICAL ENGINEERING "INDUSTRY 4.0"
108 RAKOVSKI STR., SOFIA 1000, BULGARIA

PUBLISHER:

**SCIENTIFIC TECHNICAL UNION OF MECHANICAL ENGINEERING
INDUSTRY-4.0**

108, Rakovski Str., 1000 Sofia, Bulgaria

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tel./fax (+359 2) 986 22 40,

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ISSN PRINT 2367-8399

ISSN WEB 2534-8493

YEAR IV, ISSUE 1 / 2019

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FINITE AREA ALGORITHM FOR THIN FILM CAVITATION IN OPENFOAM

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Abstract: Numerical algorithm for calculating thin film cavitation effects is presented in this paper. Cavitation is a common phenomenon in diverging parts of thin film contacts, such as: journal bearings, ball bearings, seals, etc. Locating and calculating cavitation effects is very important for their applicability, efficiency and safety. The thin film flow solver based on the Reynolds equation, together with cavitation algorithm is implemented using the Finite Area Method inside the OpenFOAM framework. OpenFOAM is an open source C++ toolbox for computational fluid dynamics (CFD). The Finite Area Method is a two-dimensional counterpart of the Finite Volume Method, used for discretising partial differential equations over curved surfaces. Discretisation is performed on user selected patches of computational mesh, with values calculated at face centres and fluxes calculated at edge centres of each finite area face. Reynolds equation is a 2D partial differential pressure equation used for calculating thin film flows between two surfaces in relative motion, with the following assumptions: fluid viscous forces dominate over body, inertia and surface tensions forces; fluid film curvature can be neglected; variation of pressure across the fluid film is negligibly small. The implemented cavitation algorithm is capable of capturing both rupture and reformation boundaries during cavitation, therefore it is considered to be mass conserving. The implemented solver is validated on three test cases: single parabolic slider (1D), twin parabolic slider (1D) and microtexture pocket bearing (2D).

Keywords: OPENFOAM, REYNOLDS EQUATION, FINITE AREA METHOD, CAVITATION

1. Introduction

In this work a numerical algorithm for calculating thin film flow with mass-conserving cavitation procedure is presented. The algorithm is implemented inside foam-extend framework, a community driven fork of OpenFOAM.

When analysing thin film flows the cavitation effects should be taken into account. Cavitation might occur in diverging parts of the contact, between individual asperities and multiple times inside a single contact [1]. Since it can occur multiple times, the method for calculating cavitation must be mass-conserving, which is imperative for accurate prediction of rupture and formation cavitation boundaries. At the rupture boundary the continuous liquid breaks into a mixture of gas bubbles and liquid, while at the formation boundary the mixture turns back into continuous liquid, i.e. full film.

One of the first notable cavitation models for the Reynolds equation was the Swift-Stieber model [2, 3], which forces a null pressure gradient at the rupture boundary and does not consider the reformation of the full film. Jakobsson and Floberg [4] and Olsson [5] derived mass-conserving boundary conditions widely known as Jakobsson-Floberg-Olson (JFO) boundary conditions. The JFO boundary conditions satisfy the mass conservation between the full-film and cavitating regions both at the rupture and at the formation boundary. A significant number of cavitation models in the literature is based on JFO boundary conditions [6]. Elrod and Adams (EA) [7] developed the first efficient algorithm for calculating cavitation when analysing thin film via Reynolds equation. Their algorithm was implemented using the Finite Difference Method and implicitly incorporates JFO boundary conditions by iteratively dividing the computational domain into full-film and cavitating region. They introduced a switch function inside the Reynolds equation which terminates the pressure term (Poiseuille) in the cavitating region, leaving only Couette flow. Vijayaraghavan and Keith [8] made a significant improvement to the EA algorithm, by using an implicit numerical finite difference scheme for orthogonal and non-orthogonal grids. They considerably increased numerical stability of the cavitation algorithm. Sahlin et al. [9] developed a general cavitation algorithm considering an arbitrary density-pressure relation of the fluid. Their model is similar to the model by Elrod and Adams, also using a switch function for terminating the pressure gradient. In the recent years the concept of complementarity was regularly used for resolving cavitation by several authors: Giacomini et al. [10], Bertocchi et al. [11], Almqvist et al. [12], Almqvist and Wall [13], etc.

Presently, the most papers dealing with thin film cavitation are oriented either on algorithm efficiency, e.g. [14], or specific computational methodologies, e.g. hybrid Finite Volume-Finite Element Method [15].

2. Methodology

In the current work the hydrodynamic pressure in thin film lubricant flow is calculated using the Reynolds equation. The Reynolds equation is a two-dimensional partial differential equation governing the pressure distribution in thin film flow between two surfaces in relative motion, with the following assumptions [1]:

- The fluid is considered to be Newtonian or non-Newtonian depending on the formulation of Reynolds equation;
- Variation of pressure across the fluid film is negligibly small;
- Width and length of the fluid film is significantly larger compared to the thickness, thus the film curvature can be neglected;
- Fluid viscous forces dominate over body, inertia and surface tension forces.

The Reynolds equation expressed in its most common form states:

$$\nabla_s \cdot \left(\frac{\rho h^3}{12\eta} \nabla_s p \right) = \nabla_s \cdot \left[\frac{\rho h (\mathbf{U}_1 + \mathbf{U}_2)}{2} \right] + \frac{\partial(\rho h)}{\partial t} \quad (1)$$

where ∇_s and $\nabla_s \cdot$ are the surface gradient and divergence operator, respectively; p , ρ and η are pressure, density and viscosity of the fluid; h is the film thickness; $\mathbf{U}_1$ and $\mathbf{U}_2$ are velocity vectors of surfaces 1 and 2.

In order to take cavitation into account, the algorithm based on the method by Elrod and Adams (EA) [7] was implemented, where the calculation domain is divided into two regions: the full film region and the cavitating region. The switch function α is introduced into Eq. 1, defined as:

$$\begin{aligned} \rho &\geq \rho_{\text{cav}} \rightarrow \text{full-film} \rightarrow \alpha = 1, \\ \rho &< \rho_{\text{cav}} \rightarrow \text{cavitating} \rightarrow \alpha = 0, \end{aligned} \quad (2)$$

where ρ_{cav} is the fluid density at the cavitation pressure p_{cav} . Elrod and Adams derived a new form of the Reynolds equation, where the equation is solved for fractional film content $\theta = \rho/\rho_{\text{cav}}$ instead of pressure p , which enables the calculation of both regions using a

single expression. In this work, a modified form of their Reynolds equation [16] is implemented:

$$\nabla_s \cdot \left(\alpha \frac{\beta h^3}{12\eta} \nabla_s \rho \right) = \nabla_s \cdot \left[\frac{\rho h (\mathbf{U}_1 + \mathbf{U}_2)}{2} \right] + \frac{\partial(\rho h)}{\partial t}, \quad (3)$$

where α is the switch function, $\beta = \rho(\partial p / \partial \rho)$ is the fluid bulk modulus, and ρ represents the density of gas and liquid mixture in the cavitating region and liquid density in the full film region. In the full-film region ($\alpha = 1$) all terms of Eq. 3 are active, however, in the cavitating region the left-hand side term (Poiseuille) is equal to zero, while the first (Couette) and second (squeeze and local expansion) term on the right-hand side remain active. The model implicitly incorporates the mass-conserving Jakobsson-Floberg-Olsson (JFO) [4,5] boundary conditions between the regions. At the rupture boundary, where cavitation initiates, the null pressure gradient condition applies with constant cavitation pressure:

$$\nabla_s p = 0 \quad p = p_{cav}, \quad (4)$$

where p_{cav} is the cavitation pressure. At the formation boundary, where the gas-liquid mixture turns back into the full-film, the non-null pressure gradient represents the supply of the full-film zone by the cavitating region:

$$\frac{\beta h^2}{12\eta} \nabla_s \rho = \frac{\mathbf{U}_1 + \mathbf{U}_2}{2} \left(\rho_{cav} - \frac{\rho_{form}}{\rho_{cav}} \right), \quad (5)$$

where ρ_{form} is the density of gas-liquid mixture next to the formation boundary in the upwind direction.

In order to calculate the lubricant pressure using Eq. 3, a pressure density relation by Dowson and Higginson [17] was implemented:

$$p = \frac{(\rho_0 - \rho)C_1}{\rho - \rho_0 C_2}, \quad (6)$$

where ρ_0 is lubricant density at atmospheric (zero-gauge) pressure, while coefficients C_1 and C_2 are lubricant specific. The bulk modulus β , using the same pressure-density relation, can be expressed as:

$$\beta = \rho \frac{\partial p}{\partial \rho} = \rho \frac{C_1(C_2 - 1)\rho_0}{(\rho - \rho_0 C_2)^2}. \quad (7)$$

Viscosity η can be considered either constant or pressure-dependent and/or shear-dependent. In the current work the well-known Barus viscosity law was implemented for calculating low-shear viscosity μ :

$$\mu = \mu_0 \exp(\alpha p), \quad (8)$$

where μ_0 is the reference viscosity at atmospheric pressure and α is the pressure exponent. In order to account for shear-dependence of lubricant viscosity the Ree-Eyring model [18] was implemented:

$$\eta = \frac{\tau_E}{\dot{\gamma}} \sinh^{-1} \left(\frac{\mu \dot{\gamma}}{\tau_E} \right), \quad (9)$$

where τ_E is the Eyring stress and $\dot{\gamma} = |\mathbf{U}_1 - \mathbf{U}_2|/h$ is the shear rate.

3. Implementation

Models presented in this work are implemented inside `foam-extend` framework, a community driven fork of `OpenFOAM`. `OpenFOAM` is an open source C++ toolbox for computational fluid dynamics (CFD).

The Reynolds equation is discretised using the Finite Area Method (FAM) [19], a two-dimensional counterpart of the Finite Volume Method (FVM). FAM is used for discretising partial differential equations over curved surfaces, i.e. computational surface meshes, with values calculated at face centres and fluxes calculated at edge centres of each finite area face, Fig. 1. The finite area discretisation of the spatial domain, and of the transport

equation is described in [19]. The Poiseuille coefficient in Eq. 3 is a diffusion term in which the values on the edges of a finite area face are calculated using the linear interpolation scheme. In case of the Couette coefficient, being a convection term, the edge values are calculated using the upwind differencing scheme. All numerical examples presented in this work are considered to be steady-state, thus the temporal term in Eq. 3 is equal to zero. The calculation workflow, including the cavitation algorithm and calculation of the lubricant properties, is given in Fig. 2.

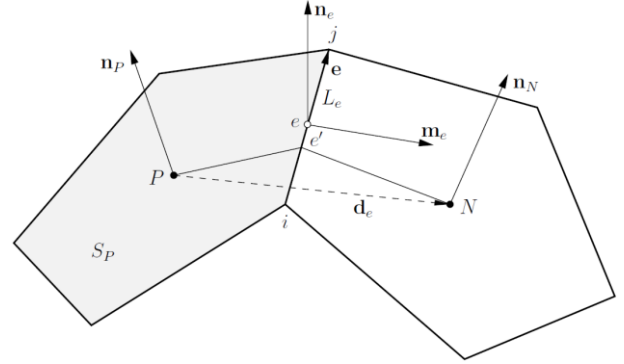


Fig. 1 Finite area faces P and N [19].

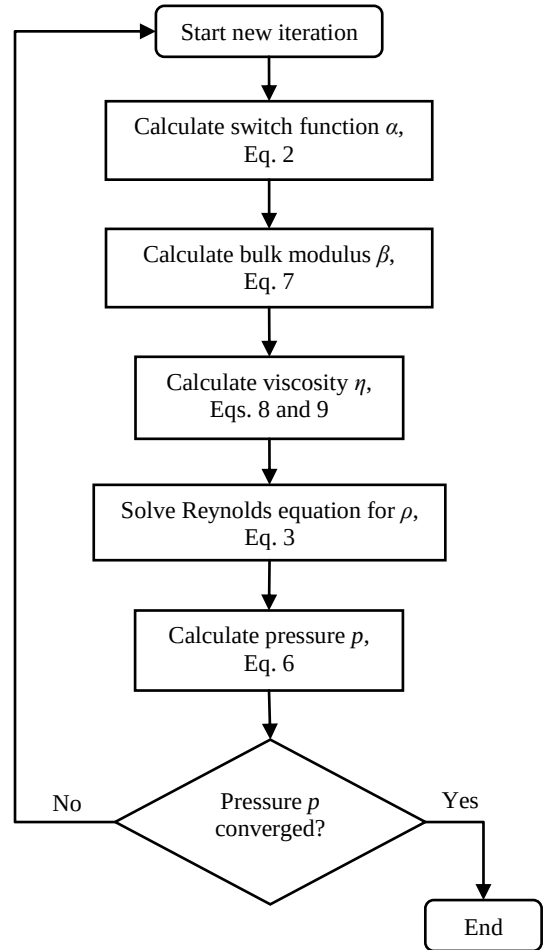


Fig. 2 Thin film pressure calculation workflow.

4. Numerical Examples

In this chapter the previously described thin film pressure model is applied to three numerical examples: single parabolic slider, twin parabolic slider and microtexture pocket bearing.

Single Parabolic Slider

A one-dimensional case of a single parabolic slider is presented here. The case consists of a moving flat plate and a fixed parabolic wall, Fig. 3.

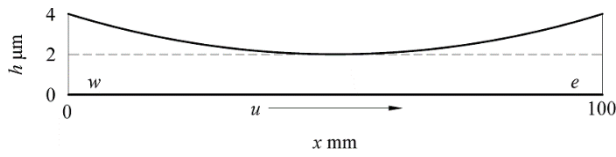


Fig. 3 Single parabolic slider geometry.

The types of the boundary conditions for fluid density ρ are Dirichlet boundary condition at the inlet (w) and zero gradient Neumann boundary condition at the outlet (e). The calculation was performed for two inlet values of density ρ . The fluid viscosity is considered to be constant, while pressure p relates to density ρ following the Dowson-Higginson expression, Eq. 6. The boundary conditions and fluid properties are specified in Table 1. The simulation results are compared to the results by Sahlin et al. [9].

Table 1: Boundary conditions and fluid properties for the single parabolic slider case.

Sliding speed, U	0.25	m/s
Cavitation pressure, p_{cav}	10^5	Pa
Inlet density ratio, ρ_w/ρ_{cav}	1	-
	0.55	
Viscosity, η	0.04	Pa s
Dowson-Higginson coefficients	C_1	$2.22 \cdot 10^9$
	C_2	1.66

In case of the inlet density ratio $\rho_w/\rho_{cav} = 1$ the results are presented in Figs. 4 and 5, while for the inlet ratio $\rho_w/\rho_{cav} = 0.55$ they are given in Figs. 6 and 7. The first case considers the rupture cavitation boundary, while the second one the reformation boundary. For both cases, i.e. inlet conditions, the fluid density ratio and pressure show excellent agreement to the results given by Sahlin et al. [9] in the whole computational domain.

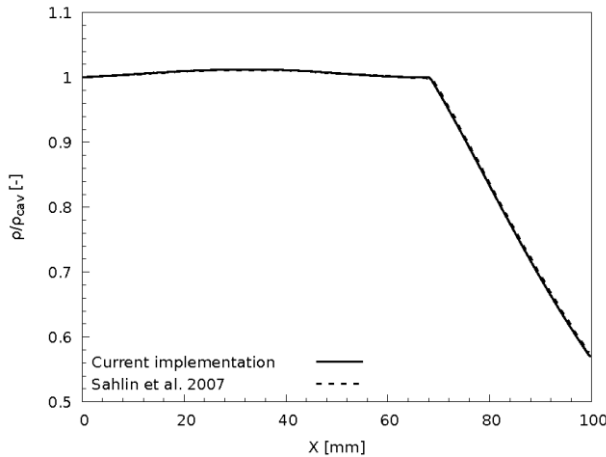


Fig. 4 Fluid density ratio for $\rho_w/\rho_{cav} = 1$ in case of the single parabolic slider.

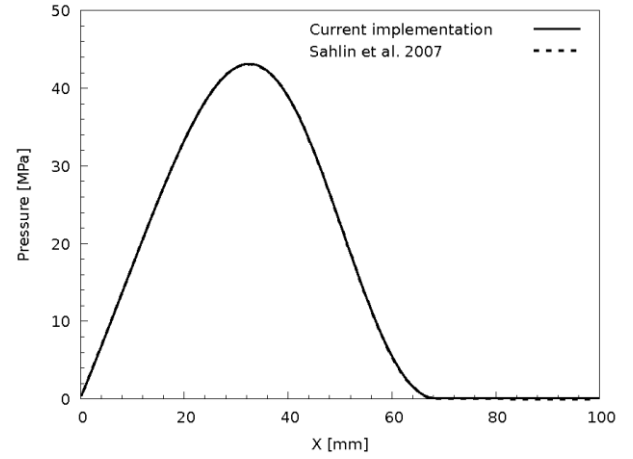


Fig. 5 Fluid pressure for $\rho_w/\rho_{cav} = 1$ in case of the single parabolic slider.

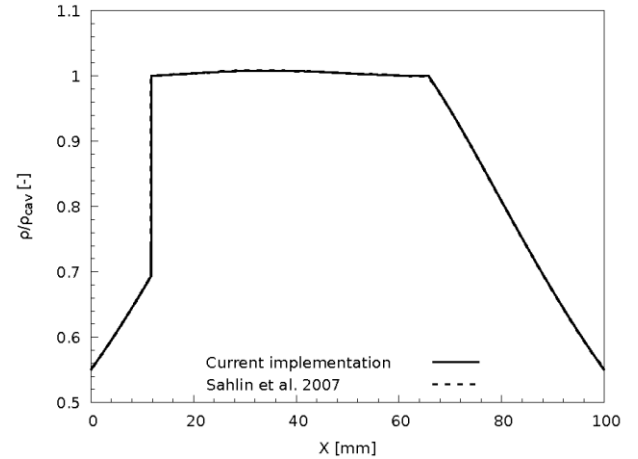


Fig. 6 Fluid density ratio for $\rho_w/\rho_{cav} = 0.55$ in case of the single parabolic slider.

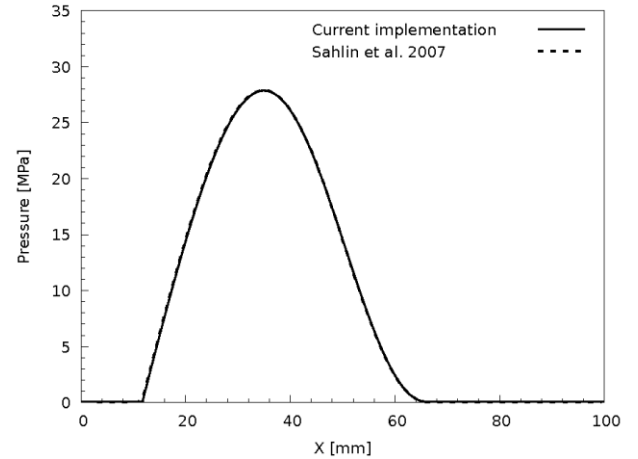


Fig. 7 Fluid pressure for $\rho_w/\rho_{cav} = 0.55$ in case of the single parabolic slider.

Twin Parabolic Slider

A one-dimensional case of a twin parabolic slider is presented here. The case consists of a moving flat plate and a fixed twin parabolic wall, Fig. 8. Compared to the single parabolic slider, this case considers both rupture and reformation cavitation boundaries at the same time.

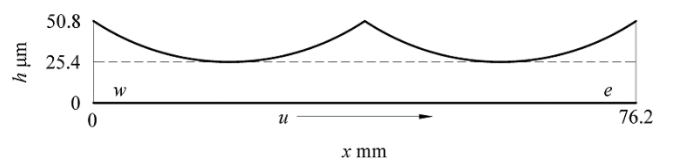


Fig. 8 Twin parabolic slider geometry.

The types of the boundary conditions for fluid density ρ are the same as in the single parabolic slider. The boundary conditions and fluid properties are specified in Table 2. The simulation results are again compared to the results by Sahlin et al. [9].

Table 2: Boundary conditions and fluid properties for the twin parabolic slider case.

Sliding speed, U	4.57	m/s
Cavitation pressure, p_{cav}	0	Pa
Inlet density ratio, ρ_w/ρ_{cav}	1.0001	-
Viscosity, η	0.039	Pa s
Dowson-Higginson coefficients	C_1	$2.22 \cdot 10^9$
	C_2	1.66

The simulation results of fluid pressure are presented in Fig. 9, which show very good agreement to the results given by Sahlin et al. [9], with small underprediction at the second pressure peak.

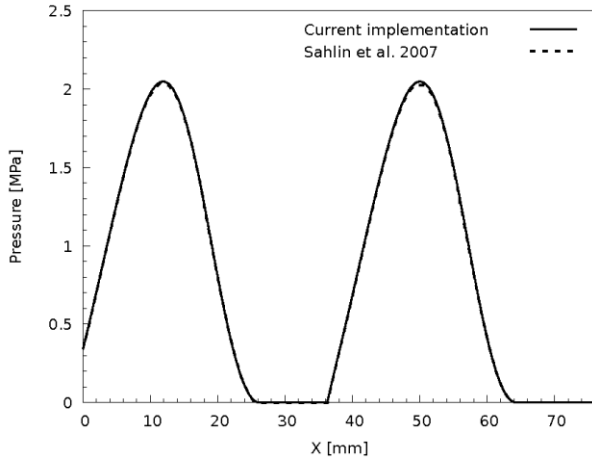


Fig. 9 Fluid pressure in case of the twin parabolic slider.

Microtexture Pocket Bearing

A two-dimensional case of a microtexture pocket bearing [11] is presented here. The case consists of a moving flat plate and a fixed skewed wall with a pocket, Fig. 10.

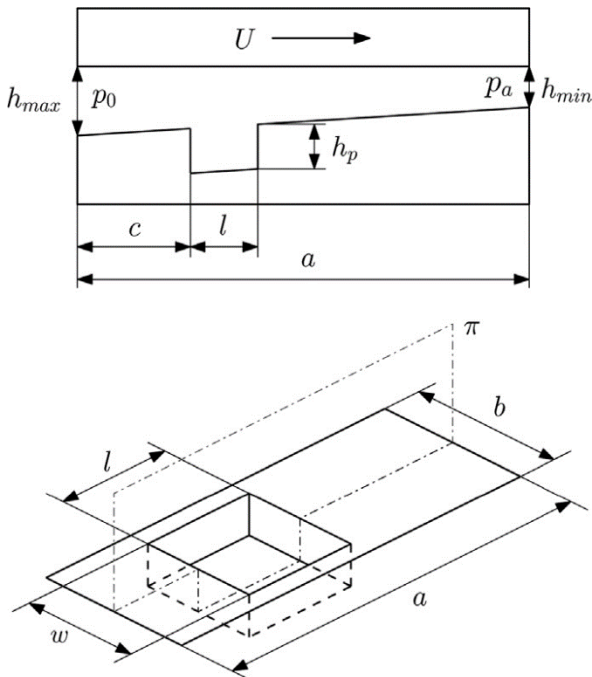


Fig. 10 Microtexture pocket bearing geometry [11].

The geometrical properties are given in Table 3, while the boundary conditions and fluid properties are specified in Table 4. The simulations are conducted for two geometries, narrow and wide pocket bearing. The Dirichlet boundary condition is used for all four boundaries (inlet, outlet, left side, right side) for density ρ , with all values equal to the inlet value.

Table 3: Geometrical properties of the microtexture pocket bearing case.

Length		a	20	mm
		b	6	
		c	4	
Thickness		h_{min}	1	μm
		h_{max}	1.1	
		h_p	0.4	
Width	A	b	10	mm
		w	7	
	B	b	300	
		w	210	

Table 4: Boundary conditions and fluid properties for the microtexture pocket bearing case.

Sliding speed, U		1	m/s
Cavitation pressure, p_{cav}		0	Pa
Inlet density ratio, ρ_w/ρ_{cav}		1.00003	-
Dowson-Higginson coefficients	C_1	$2.22 \cdot 10^9$	-
	C_2	1.66	
Viscosity: Barus law with Ree-Eyring model	μ_0	0.01	Pa s
	α	$1.2 \cdot 10^{-8}$	Pa^{-1}
	τ_E	5	MPa

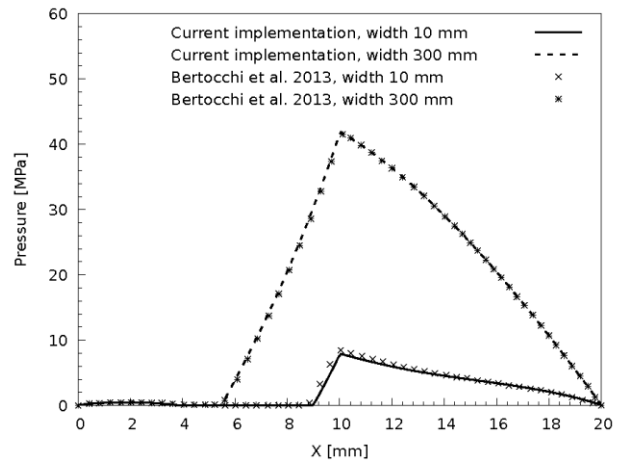


Fig. 11 Fluid pressure at the centerline in case of microtexture pocket bearing.

Fig. 11 shows the results of pressure at the centerline of the bearing for both geometries, with comparison to Bertocchi et al. [11]. The simulation results are in very good agreement with the literature for the narrow bearing, with small underprediction of pressure at the exit of the pocket. In case of the wide bearing, the results are in excellent agreement throughout the centerline.

5. Conclusion

A numerical model for calculating thin film pressure with cavitation effects is presented in this work. The model is implemented in the foam-extend framework, with the Reynolds equation discretised using the Finite Area Method. The model is compared to the literature via three numerical examples, and the results are in very good agreement. For future work, the implemented model will be tested on physically and geometrically more complex numerical examples, e.g. dimple bearings.

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CONCEPT FOR DETERMINING DRIVING FACTOR OF GEARBOX SPEED REDUCER

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Abstract: The driving factor is an important technical feature of dynamical loaded working machines. It is especially important to know its value for the working wheel speed reducer of the rotating excavators. In this paper is presented a concept for determining the driving factor of a concrete gearbox speed reducer. To calculate the value of the driving factor according to the specified concept, knowledge of the load regime and the carrying capacity of the gearbox is necessary. The load regime is known from the load function, which is given in the paper, and is determined on the basis of the experimental measurements carried out for this purpose. The carrying capacity of the speed reducer is calculated by theoretical analysis according to the established methodology in this paper, for the characteristic pairs of gears determined in this analysis. As the most important gear pair, from the aspect of determining the driving factor, the most loaded gear pair is taken, in which during the exploitation of the speed reducer the most interventions are made, compared to the other gear pairs of the gearbox.

Keywords: DRIVING FACTOR, GEARBOX REDUCER, ROTATING EXCAVATOR, COALMINE

1. Introduction

The load regime of the gearbox speed reducer of the working wheel of rotating excavator for normal and specific exploitation conditions is defined by the torque distribution function of the output shaft, depending on the number of load changes in the expected working life of the speed reducer.

The load regime of the speed reducer of the working wheel of the rotating excavator SRs-630 which is used in the coalmine Suvodol-Bitola in Macedonia, is determined on the basis of conducted experimental measurements and extensive theoretical analysis. The load function of this speed reducer is shown in this paper.

The carrying capacity of the gearbox speed reducer of the working wheel of rotating excavator is determined for its most loaded gear pair, i.e. for the gear pair that has certain operational weaknesses.

The carrying capacity for all gear pairs from the working wheel speed reducer of the rotating excavator SRs-630 has been obtained theoretically, with a set methodology for this purpose, presented in the continuation of the paper. The practice of maintaining the speed reducer showed that certain weaknesses during the exploitation were observed in the first, second and last (the fifth) gear pair, but most was intervened at the last gear pair which is the most loaded in the speed reducer, that is, the following gear of this gear pair which is mounted on the output shaft of the speed reducer.

The driving factor is a technical feature of dynamical loaded working machines. The driving factor is an important indicator of the dynamic load of the speed reducers in the rotating excavators, in particular for the working wheel speed reducer of the excavator.

In this paper, the concept for determining the driving factor of the working wheel speed reducer of the rotating excavator SRs-630 is presented and a numerical value for the driving factor based on such a methodology has been obtained.

2. Load regime of the speed reducer

The loading of the speed reducer is a change in the torque T_2 of the following shaft depending on the time t in the total exploitation life of the speed reducer.

The load function of the speed reducer of the excavator SRs-630 (shown in Figure 1) was obtained by processing the deformations of the output shaft of the speed reducer under the characteristic working regimes for the excavator, which deformations were obtained by experimental measurements.

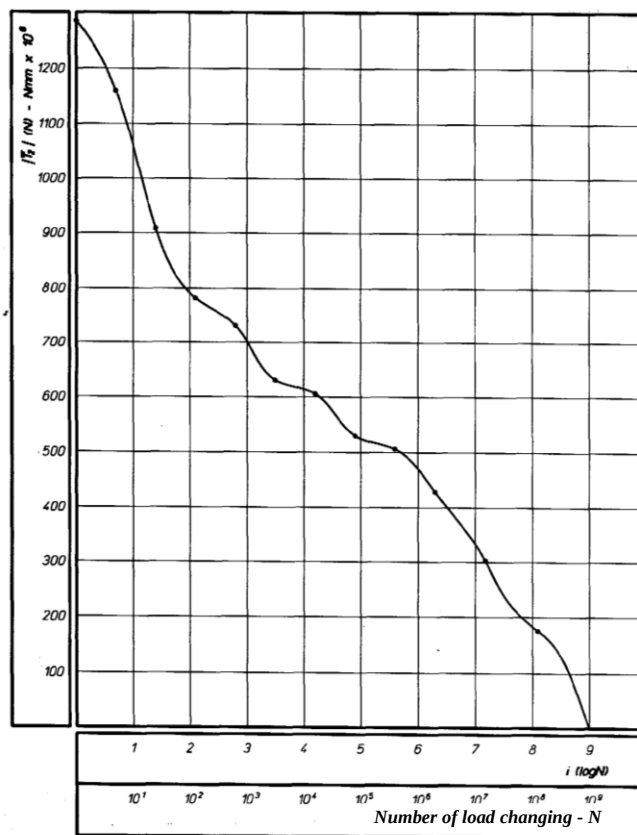


Fig. 1 Load function of the working wheel speed reducer of the rotating excavator SRs-630 under working conditions at the coalmine Suvodol-Bitola

3. Carrying capacity of the speed reducer

3.1. Generally for the carrying capacity and characteristics of the speed reducer

The carrying capacity is the ability of the speed reducer to receive certain types of maximum loads with a certain number of changes, under certain working conditions. The carrying capacity of the speed reducer can be associated with the Veler's curves of fatigue of the material for the gears in the speed reducer, obtained by examining the gears (in particular on the surface pressure of the tooth side, and in particular the flexion in the root of the tooth). This is necessary for theoretically obtaining the curve of the load function, while for practical use it is sufficient to find only two points of the curve and to determine the value of the driving factor for those two points.

The initial basis for this calculation is the knowledge of the materials characteristics from which the gears are made:

- Fatigue strength for Hertz surface pressure σ_{Hlim}
- Static strength for Hertz surface pressure σ_{H0}
- Fatigue strength for strain at the root of the tooth σ_{Flim}
- Static strength for strain at the root of the tooth σ_{F0}

Since this is a speed reducer that is in exploitation, whose scheme is shown in Figure 2, in addition to these data are also known the type of stresses, working conditions, shape and dimensions of the gears, etc., which can determine the value of the static and fatigue capacity.

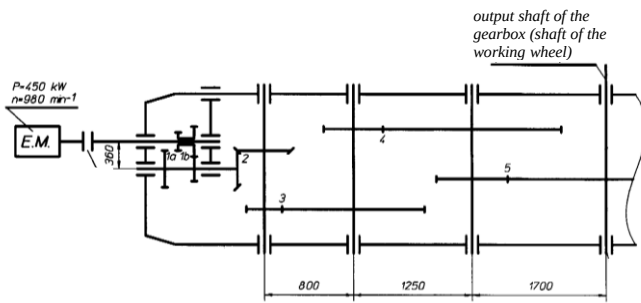


Fig. 2 Scheme of the gearbox speed reducer with the working wheel of the excavator SRs-630

The practice showed that in the work of the speed reducer, the weakness of the first, second and last gear pair was observed (the gear pair 1, the gear pair 2 and the gear pair 5, marked in Figure 2). For the third and fourth gear pair, no problems were encountered in the speed reducer's current exploitation life, so they are practically working as delivered by the manufacturer. Because of the above, only the first, second and last gear pair will be treated.

The first gear pair is double, its driving gear is made of two cylindrical gears with slanted teeth made on a common carrier that fit together with two following gears respectively, whereof the following gears are mounted on the same shaft. Depending on which part of the pair is coupled, two speeds can be achieved by rotating the output shaft for one speed at the input shaft and thus two gear ratio of the speed reducer. In practice, the excavator operates with the higher speed, so for this reason only the right part of the gear pair is analyzed in the paper (gear pair 1b).

The second gear pair is conical with curved teeth (conical helical gear pair), and the last gear pair is a cylindrical with a arrow teeth (herring-bone gear pair).

Data on the characteristics of the materials from which the gear pairs are made are used by the theoretical data from the excavator manufacturer, TAKRAF-Germany.

3.2. Nominal loading of the speed reducer

The nominal loading of the speed reducer defines the average load during a lasting drive.

The nominal load is calculated as the average load for all working regimes of the excavator (for which measurements were made), with the exception of the extreme values achieved during the experimental measurements. The indicator of the load is taken showing of the ammeter during the measurement (the measurement methodology is not subject to analysis in this paper). The average value of the electric current during the experimental measurements was:

$$I_{sr} = \frac{40 + 48 + 30 + 40 + 24 + 32 + 30 + 60}{8} = 38 \text{ A}$$

The power corresponding to this load is rated (nominal) power, so on the output shaft of the speed reducer has a value

$$P_R = (\sqrt{3} \cdot 6 \cdot 38 \cdot 0,86) \cdot 0,97 \cdot 0,92 = 303,08 \text{ kW}$$

and the nominal torque of the output shaft is:

$$T_{20} = 159155 \cdot \frac{P_R}{n} = 159155 \cdot \frac{303,08}{0,11885} = 405858445,3 \text{ Nmm}$$

3.3. Calculation of the carrying capacity of the gear pairs 1b and 5

3.3.1. Carrying capacity of surface pressure

a) Fatigue carrying capacity

- Maximum peripheral force on the tooth side

$$F_{tH\infty} = \frac{b \cdot d}{S_{Hlim}} \cdot \frac{u}{u+1} \cdot \frac{(Z_x \cdot Z_L \cdot Z_R \cdot Z_v \cdot Z_N \cdot Z_w \cdot Z_{Eht})^2}{K_v \cdot K_{H\alpha} \cdot K_{H\beta} \cdot Z_\epsilon^2 \cdot Z_E^2 \cdot Z_{H\beta}^2} \cdot \sigma_{Hlim}^2$$

- Maximum peripheral torque from the peripheral force

$$T_{H\infty} = \frac{F_{tH\infty} \cdot d}{2}$$

b) Static carrying capacity

- Maximum peripheral force on the tooth side

$$F_{tH0} = \frac{b \cdot d}{S_{Hlim}} \cdot \frac{u}{u+1} \cdot \frac{(Z_x \cdot Z_L \cdot Z_R \cdot Z_v \cdot Z_N \cdot Z_w \cdot Z_{Eht})^2}{K_v \cdot K_{H\alpha} \cdot K_{H\beta} \cdot Z_\epsilon^2 \cdot Z_E^2 \cdot Z_{H\beta}^2} \cdot \sigma_{H0}^2$$

- Maximum peripheral torque from the peripheral force

$$T_{H0} = \frac{F_{tH0} \cdot d}{2}$$

c) Carrying capacity at $N = 10^4$ load changes

$$T_{H(10^4)} = \sqrt{T_{H\infty} \cdot T_{H0}}$$

3.3.2. Carrying capacity of flexion in the root of the tooth

a) Fatigue carrying capacity

- Maximum peripheral force at the root of the tooth

$$F_{tF\infty} = \frac{b \cdot d \cdot \cos \beta}{z} \cdot \frac{\sigma_{Flim}}{S_{Flim}} \cdot \frac{Y_x \cdot Y_{ST} \cdot Y_N \cdot Y_\delta \cdot Y_R \cdot Y_{Eht}}{K_v \cdot K_{F\alpha} \cdot K_{F\beta} \cdot Y_{FS} \cdot Y_\epsilon \cdot Y_\beta}$$

- Maximum peripheral torque from the peripheral force

$$T_{F\infty} = \frac{F_{tF\infty} \cdot d}{2}$$

b) Static carrying capacity

- Maximum peripheral force at the root of the tooth

$$F_{tF0} = \frac{b \cdot d \cdot \cos \beta}{z} \cdot \frac{\sigma_{F0}}{S_{F\lim}} \cdot \frac{Y_X \cdot Y_{ST} \cdot Y_N \cdot Y_\delta \cdot Y_R \cdot Y_{Eht}}{K_V \cdot K_{F\alpha} \cdot K_{F\beta} \cdot Y_{FS} \cdot Y_\varepsilon \cdot Y_\beta}$$

- Maximum peripheral torque from the peripheral force

$$T_{F0} = \frac{F_{tF0} \cdot d}{2}$$

c) Carrying capacity at $N = 10^4$ load changes

$$T_{F(10^4)} = \sqrt{T_{F\infty} \cdot T_{F0}}$$

The influential factors in the preceding formulas are determined by an appropriate methodology for this purpose, for gears of the gear pairs 1b and 5.

In order to cover the remaining impacts on the work of gears, which are not yet covered by their own factor, the factor of safety is still being used. Because the maximum potential for loadings on the gears (maximum carrying capacity) should be determined here, its minimum values are taken for the factors of safety: $S_{H\lim} = 1$ and $S_{F\lim} = 1.2$.

According to the known data for the gears and the formulas of items 3.3.1. and 3.3.2. the carrying capacity of the gear pairs 1b and 5 are calculated and given in Table 1 for the gear pair 1b and table 2 for the gear pair 5.

Table 1

carrying capacity	gear pair 1 b	
	$z_1 = 56$	$z_2 = 45$
$F_{tH\sim}$ [N]	127164.2	103166.1
$T_{H\sim}$ [Nmm]	25390875.81	16553000.74
F_{tH0} [N]	5208646.8	3337790.79
T_{H0} [Nmm]	1040010506	535548533.5
$T_{H(10^4)}$ [Nmm]	162501623.3	94153785.21
$F_{tF\sim}$ [N]	127918.8	135094.1
$T_{F\sim}$ [Nmm]	25542546.8	21675848.3
F_{tF0} [N]	447715.7	472829.2
T_{F0} [Nmm]	89395392.8	75865443.1
$T_{F(10^4)}$ [Nmm]	47783853	40551791.9

3.4. Calculation of the carrying capacity of the gear pair 2

3.4.1. Carrying capacity of surface pressure

a) Fatigue carrying capacity

- Maximum peripheral force on the tooth side

$$F_{tH\infty} = \frac{b \cdot d_m \cdot u}{S_{H\lim} \sqrt{u^2 + 1}} \cdot \frac{(Z_X \cdot Z_L \cdot Z_R \cdot Z_V \cdot Z_W \cdot Z_N \cdot Z_{Eht})^2}{K_V \cdot K_{H\alpha} \cdot K_{H\beta} \cdot Z_{H\beta}^2} \cdot \sigma_{H\lim}^2$$

where $d_m = d - b \sin \varphi$ and it is a diameter of the pitch circle

- Maximum peripheral torque from the peripheral force

$$T_{H\infty} = \frac{F_{tH\infty} \cdot d_m}{2}$$

Table 2

carrying capacity	gear pair 5	
	$z_1 = 23$	$z_2 = 98$
$F_{tH\sim}$ [N]	447077.43	2198172.23
$T_{H\sim}$ [Nmm]	143210077.3	3000186363
F_{tH0} [N]	16094788.15	54954306.2
T_{H0} [Nmm]	5155563010	75004659000
$T_{H(10^4)}$ [Nmm]	859260482.6	15000931000
$F_{tF\sim}$ [N]	892999.97	819305.64
$T_{F\sim}$ [Nmm]	286050216.6	1118233401
F_{tF0} [N]	3162708.6	3098378.6
T_{F0} [Nmm]	1013094644	4228837513
$T_{F(10^4)}$ [Nmm]	5383226984.5	21745867708

Note for Table 2: The listed carrying capacity for gear pair 5 in the table refers for half gear. For whole herring-bone gear the given values are multiplied by 2 and the carrying capacity is obtained with respect to surface pressure and flexion.

b) Static carrying capacity

- Maximum peripheral force on the tooth side

$$F_{tH0} = \frac{b \cdot d_m \cdot u}{S_{H\lim} \sqrt{u^2 + 1}} \cdot \frac{(Z_X \cdot Z_L \cdot Z_R \cdot Z_V \cdot Z_W \cdot Z_N \cdot Z_{Eht})^2}{K_V \cdot K_{H\alpha} \cdot K_{H\beta} \cdot Z_{H\beta}^2} \cdot \sigma_{H0}^2$$

- Maximum peripheral torque from the peripheral force

$$T_{H0} = \frac{F_{tH0} \cdot d_m}{2}$$

c) Carrying capacity at $N = 10^4$ load changes

$$T_{H(10^4)} = \sqrt{T_{H\infty} \cdot T_{H0}}$$

3.4.2. Carrying capacity of flexion in the root of the tooth

a) Fatigue carrying capacity

- Maximum peripheral force at the root of the tooth

$$F_{tF\infty} = \frac{b \cdot d_m \cdot \cos \beta_m}{z} \cdot \frac{\sigma_{F\lim}}{S_{F\lim}} \cdot \frac{Y_X \cdot Y_N \cdot Y_R \cdot Y_\delta \cdot Y_{Eht}}{K_V \cdot K_{F\alpha} \cdot K_{F\beta} \cdot Y_{FS} \cdot Y_\varepsilon \cdot Y_K \cdot Y_{LS}}$$

- Maximum peripheral torque from the peripheral force

$$T_{F\infty} = \frac{F_{tF\infty} \cdot d_m}{2}$$

b) Static carrying capacity

- Maximum peripheral force at the root of the tooth

$$F_{tF0} = \frac{b \cdot d_m \cdot \cos \beta_m}{z} \cdot \frac{\sigma_{F0}}{S_{F\lim}} \cdot \frac{Y_X \cdot Y_N \cdot Y_R \cdot Y_\delta \cdot Y_{Eht}}{K_V \cdot K_{F\alpha} \cdot K_{F\beta} \cdot Y_{FS} \cdot Y_\varepsilon \cdot Y_K \cdot Y_{LS}}$$

- Maximum peripheral torque from the peripheral force

$$T_{F0} = \frac{F_{tF0} \cdot d_m}{2}$$

c) Carrying capacity at $N = 10^4$ changes

$$T_{F(10^4)} = \sqrt{T_{F\infty} \cdot T_{F0}}$$

The influential factors listed in the preceding formulas are determined for the gears of the gear pair 2. For the factors of safety, it is taken: $S_{Hlim} = 1.2$ and $S_{Flim} = 1.4$.

According to the specified data for the gears and the formulas of items 3.4.1. and 3.4.2. the carrying capacity of the gear pair 2 is calculated and given in Table 3.

Table 3

carrying capacity	gear pair 2	
	$z_1 = 29$	$z_2 = 38$
$F_{tH\sim}$ [N]	2667868701	3354330573
$T_{H\sim}$ [Nmm]	375996070000	619460990000
F_{tH0} [N]	58100247000	64939839000
T_{H0} [Nmm]	9422117200000	13799715000000
$T_{H(10^4)}$ [Nmm]	1882200500000	2923762000000
$F_{tF\sim}$ [N]	41943.4	42162.1
$T_{F\sim}$ [Nmm]	5911293.1	7786285.8
F_{tF0} [N]	114924.9	115524.1
T_{F0} [Nmm]	18637382.9	24548855.1
$T_{F(10^4)}$ [Nmm]	10496239	13825498.3

4. Driving factor

4.1. Standardized load numbers

$$a_{(10^0)} = \frac{T_{2(10^0)}}{T_{20}} = \frac{1286377308}{405858445,3} = 3,17$$

$$a_{(10^4)} = \frac{T_{2(10^4)}}{T_{20}} = \frac{624750000}{405858445,3} = 1,54$$

$$a_{(10^7)} = \frac{T_{2(10^7)}}{T_{20}} = \frac{333200000}{405858445,3} = 0,82$$

$$A_1 = 4 \quad A_2 = 1.6 \quad A_3 = 1$$

The magnitudes $T_{2(10^0)}$, $T_{2(10^4)}$, $T_{2(10^7)}$ are read from the diagram in Figure 1 and the magnitudes T_{20} is from item 3.2.

4.2. Standardized carrying capacity numbers

$$b_{(10^0)} = \frac{T_{(10^0)}}{T_{F\infty}} = \frac{4228837513}{1118233401} = 3,78$$

$$b_{(10^4)} = \sqrt{b_{(10^0)}} = 1,94$$

$$b_{(10^7)} = 1$$

$$B_1 = 3.15 \quad B_2 = 1.8 \quad B_3 = 1$$

$T_{(10^0)}^0 = T_{F0}$, the magnitudes T_{F0} и $T_{F\infty}$ are from Table 2 for the following gear of the gear pair 5.

Because the carrying capacity of flexion in the root of the tooth is less favorable (smaller), it is considered the most relevant when defining the standardized carrying capacity numbers.

4.3. Value of the driving factor

$$K_A = \max [A_1/B_1, A_2/B_2, A_3/B_3] = \max [4/3.15; 1.6/1.8; 1/1] = \max [1.27; 0.89; 1] = 1.27$$

It is adopted $K_A = 1.25$ for the value of the driving factor of the speed reducer, which is the number from the standard array R20.

5. Conclusion

In this paper is shown the load function of the working wheel speed reducer for a real excavator and known working regimes, which is a function derived from the experimental measurements carried out for that purpose. The load function gives a description of the speed reducer's load regime in its total lifetime.

The paper presents a methodology for theoretical calculation of the carrying capacity of the speed reducers, that is, for its gears, both fatigue and static carrying capacity, for surface pressure on the tooth side and flexion in the root of the tooth. The presented methodology is confirmed by analysis of the specific reducer. Also this paper presents a concept for determining the value of the driving factor of the speed reducer as an important characteristic of its dynamic load, and defines the value of that factor on the basis of this concept for the mentioned speed reducer.

The gain value for the driving factor can serve as a benchmark for the driving factors of the rotating excavators with similar technical characteristics operating in approximate working conditions.

6. References

Hristovska E.: Own papers with this topic and data from experimental measurement

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

Numerous documentation for modifications and repairs of gearboxes from the coalmine Suvodol-Bitola.

INFLUENCE OF THE AMBIENT TEMPERATURE CHANGE ON STEAM PRESSURE REDUCTION VALVE EXERGY DESTRUCTION AND EXERGY EFFICIENCY

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Abstract: The paper presents an exergy analysis of pressure reduction valve mounted in the steam propulsion system on conventional LNG carrier. From exploitation are obtained that the valve pressure and temperature decrease become as higher as steam system load increases. Valve exergy power input and output decreases during the increase in steam system load, mostly because of the steam mass flow decrease. Steam system load increase in exploitation also causes a decrease in valve exergy destruction with a simultaneous decrease in valve exergy efficiency (from 68.42 % to 68.09 %). The ambient temperature variation showed that the valve exergy destruction is the lowest for the lowest observed ambient temperature, in any steam system load. The exergy efficiency of the pressure reduction valve is reverse proportional to valve exergy destruction. An increase in the ambient temperature for 10 °C causes a decrease in analyzed valve exergy efficiency for between 2.5 % and 3 %.

KEYWORDS: AMBIENT TEMPERATURE, STEAM PRESSURE REDUCTION VALVE, EXERGY DESTRUCTION, EXERGY EFFICIENCY

1. Introduction

The main function of pressure reduction valves is reducing pressure of operating medium (in steam plants that operating medium is usually superheated steam). If the operating medium is superheated steam, along with the pressure reduction through the pressure reduction valve was also reduced steam temperature while steam specific entropy increases [1]. In such way the steam system maintained desired operating parameters [2] in all of its parts.

The basic rule for pressure reduction valve operation is that before and after valve specific enthalpy of operating medium remains constant [3]. It is irrelevant to investigate steam pressure reduction valves from the viewpoint of energy, because without any mass flow leakage, steam pressure reduction valves have energy efficiency of 100 % and energy power losses equal to zero, in any observed operating point.

In the land-based steam power plants pressure reduction valves are very rare [4], because in that kind of steam power plants is not necessary to reduce the masses of plant components. In order to remain the walls of every component from the marine steam plant as thick as possible and thus reduce their mass, pressure reduction valves are necessary on ship steam systems [5].

A detailed analysis of any valve type can be rarely found in the scientific literature. If some were found, mostly it is investigations of control valves for steam turbines [6], in some cases along with its actuation systems [7]. Investigations of steam pressure reduction valves are rare, especially for several steam system loads [8].

In this paper was analyzed steam pressure reduction valve, mounted on the main condenser “dump” line, through a several steam system loads. For each load is presented a decrease in steam temperature and pressure on the analyzed valve from the ship exploitation. Based on the measurements of valve operation parameters are presented valve exergy power inputs and outputs, as well as exergy destruction and exergy efficiency in each observed system load. Exergy power inputs and outputs, as well as exergy efficiency and exergy destruction of any steam system component are changeable when the ambient temperature increases or decreases. Engine room temperature variation is performed from 10 °C to 50 °C in steps of 10 °C what is usually expected change of engine room temperature. For observed ambient temperature variation is calculated and presented pressure reduction valve exergy destruction and exergy efficiency in each observed steam system load.

2. Analyzed pressure reduction valve elements and operation characteristics

Analyzed pressure reduction valve is mounted in LNG carrier steam propulsion plant near steam generators. Main characteristics of the LNG carrier which steam propulsion system includes the analyzed pressure reduction valve are presented in Table 1:

Table 1. LNG carrier characteristics

Dead weight tonnage	84.812 DWT
Overall length	288 m
Max breadth	44 m
Design draft	9.3 m
Propulsion turbine	Mitsubishi MS40-2 (29420 kW)
Steam generators	2 x Mitsubishi MB-4E-KS

The position of the analyzed pressure reduction valve in the LNG carrier steam system is near the steam generators. The pressure reduction valve is mounted on steam generators “dump” pipeline. This pressure reduction valve is involved in steam system operation during the system startup. During system startup, steam generators produce much more superheated steam than is necessary for system operation. These facts occur because from the ecological point of view, it is more appropriate to burn LNG surplus in steam generators, than release it into the atmosphere. The superheated steam amount which is not used in steam system was directed to the main condenser. Before entering the main condenser, it is necessary to reduce superheated steam pressure after which follows steam cooling by water spray injection. For pressure reduction of superheated steam before its entrance into the main condenser is responsible analyzed valve. So, the measurements of necessary steam operating parameters before and after pressure reduction valve are and can be performed only during the steam system startup period. General pressure reduction operating range of the analyzed valve is reduction from 6.13 MPa to 0.4 MPa.

Analyzed steam pressure reduction valve intersection, along with all main components can be seen in Fig. 1. This type of pressure reduction valve has two valves (main and auxiliary) for pressure pulsation compensation and for ensuring accurate outlet pressure.

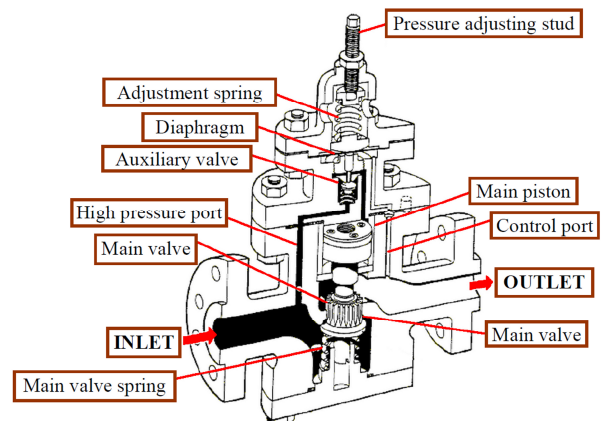


Fig. 1. Analyzed steam pressure reduction valve intersection [9]

3. Exergy analysis equations

3.1. Governing exergy analysis equations

Mass balance equation for a volume in steady state disregarding potential and kinetic energy can be expressed as [10]:

$$\sum \dot{m}_{IN} = \sum \dot{m}_{OUT} \quad (1)$$

Exergy analysis is based on the second law of thermodynamics [11]. The main exergy balance equation for a volume in steady state is [12]:

$$\dot{X}_{heat} - P = \sum \dot{m}_{OUT} \cdot \varepsilon_{OUT} - \sum \dot{m}_{IN} \cdot \varepsilon_{IN} + \dot{E}_{ex,D} \quad (2)$$

where the net exergy transfer by heat ($\dot{X}_{heat}$) at the temperature T is equal to [13]:

$$\dot{X}_{heat} = \sum (1 - \frac{T_0}{T}) \cdot \dot{Q} \quad (3)$$

Specific exergy can be defined according to [14] by an equation:

$$\varepsilon = (h - h_0) - T_0 \cdot (s - s_0) \quad (4)$$

The exergy power of a flow can be calculated according to [15]:

$$\dot{E}_{ex} = \dot{m} \cdot \varepsilon = \dot{m} \cdot [(h - h_0) - T_0 \cdot (s - s_0)] \quad (5)$$

Exergy efficiency [16] is usually defined as:

$$\eta_{ex} = \frac{\text{Exergy output}}{\text{Exergy input}} \quad (6)$$

3.2. Steam pressure reduction valve exergy analysis

For the analyzed pressure reduction valve, all necessary operating points were presented in Fig. 2. The required specific enthalpies and specific entropies were calculated from measured steam pressures and temperatures by using NIST REFPROP software [17].

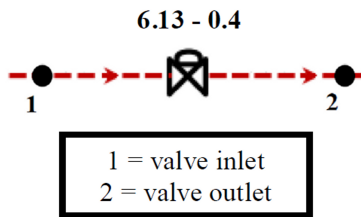


Fig. 2. Steam pressure reduction valve scheme with a general operating range

Mass and exergy balances for the analyzed steam pressure reduction valve are:

Mass balance:

$$\dot{m}_1 = \dot{m}_2 \quad (7)$$

Exergy balance:

- Exergy power input:

$$\dot{E}_{ex,IN} = \dot{m}_1 \cdot \varepsilon_1 \quad (8)$$

- Exergy power output:

$$\dot{E}_{ex,OUT} = \dot{m}_2 \cdot \varepsilon_2 \quad (9)$$

- Exergy destruction:

$$\dot{E}_{ex,D} = \dot{E}_{ex,IN} - \dot{E}_{ex,OUT} = \dot{m}_1 \cdot \varepsilon_1 - \dot{m}_2 \cdot \varepsilon_2 \quad (10)$$

- Exergy efficiency:

$$\eta_{ex} = \frac{\dot{E}_{ex,OUT}}{\dot{E}_{ex,IN}} = \frac{\dot{m}_2 \cdot \varepsilon_2}{\dot{m}_1 \cdot \varepsilon_1} \quad (11)$$

The ambient state in the LNG carrier engine room during the provided measurements was:

- pressure: $p_0 = 0.1 \text{ MPa} = 1 \text{ bar}$,
- temperature: $T_0 = 25 \text{ °C} = 298.15 \text{ K}$.

4. Measuring equipment and measurement results of pressure reduction valve

Measurements were performed in three different LNG carrier steam system operation points during the system startup. After system startup, all produced superheated steam is used in steam system and “dump” line is not in operation from that moment on. Required operating parameters (pressures, temperatures and mass flows) for each steam pressure reduction valve operating point are presented in Table 2 in relation to the main propulsion propeller speed. Main propulsion propeller speed is directly proportional to steam system load.

In Table 2 can be seen that as propulsion propeller speed increases, the steam mass flow, which is sent directly to the main condenser decreases. This fact presents that steam system uses more and more produced superheated steam as propulsion propeller speed (steam system load) increases. After 41.78 rpm, steam system uses all of produced superheated steam and steam “dump” line is closed.

Table 2. Steam pressure reduction valve inlet and outlet - measurement results

Propulsion propeller speed (rpm)	Pressure reduction valve - steam inlet (1*)			Pressure reduction valve - steam outlet (2*)		
	Temperature (°C)	Pressure (MPa)	Mass flow (kg/h)	Temperature (°C)	Pressure (MPa)	Mass flow (kg/h)
25.58	312.5	6.010	15767	232.6	0.4	15767
34.33	309.0	6.080	13175	225.6	0.4	13175
41.78	304.0	6.110	3695	216.5	0.4	3695

* Streams numeration refers to Fig. 2.

Measurement results presented in Table 2 were obtained by using the existing measuring equipment mounted before and after analyzed pressure reduction valve. List of used measuring equipment is presented in Table 3. From Table 3 only the Shaft Power Meter used for propulsion propeller speed measuring is not mounted at the analyzed pressure reduction valve inlet or outlet, it is mounted directly on propulsion propeller shaft.

Table 3. List of used measurement equipment

Steam temperature (valve inlet and outlet)	Greisinger GTF 401-Pt100 - Immersion probe [18]
Steam pressure (valve inlet)	Yamatake JTG960A - Pressure Transmitter [19]
Steam pressure (valve outlet)	Yamatake JTG940A - Pressure Transmitter [19]
Steam mass flow (valve inlet and outlet)	Yamatake JTD960A - Differential Pressure Transmitter [20]
Propulsion propeller speed	Kyma Shaft Power Meter (KPM-PFS) [21]

5. Pressure reduction valve exergy analysis results with the discussion

5.1. Pressure reduction valve exergy analysis - exploitation

Decrease in pressure and temperature on the analyzed pressure reduction valve is presented on Fig. 3. During the increase in steam system load, steam pressure at the analyzed pressure reduction valve inlet increases, Table 2, and this occurrence causes a simultaneous increase in pressure reduction. At the propulsion propeller speed of 25.58 rpm, analyzed valve reduces steam pressure for 5.61 MPa, at 34.33 rpm steam pressure reduction amounts 5.68 MPa, while at the highest observed propulsion propeller speed of 41.78 rpm, right before the closing of “dump” line, pressure reduction amounts 5.71 MPa.

Increase in steam pressure reduction resulted also with the increase in steam temperature reduction. During the increase in steam system load, the steam temperature reduction increases from 79.9 °C (at 25.58 rpm) to 87.5 °C (at 41.78 rpm).

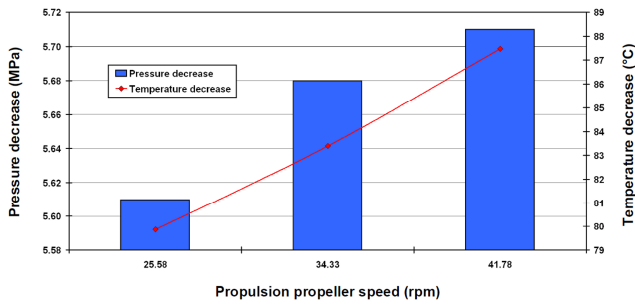


Fig. 3. The steam pressure and temperature decrease on the pressure reduction valve - exploitation

Exergy power input and output of the analyzed pressure reduction valve continuously decreases during the increase in steam system load, Fig. 4. According to the equations (8) and (9) the main reason for such decrease in the exergy power input and output can be found in the decrease in steam mass flow, Table 2. After 41.78 rpm, steam system uses all of the produced steam, so amount of steam, which passes directly from steam generators to main condenser is equal to zero.

Increase in steam system load causes decrease in valve exergy power input from 4823.8 kW (at 25.58 rpm) to 1115.7 kW (at 41.78 rpm) while exergy power output simultaneously decreases from 3300.3 kW to 759.7 kW between the same propulsion propeller speeds.

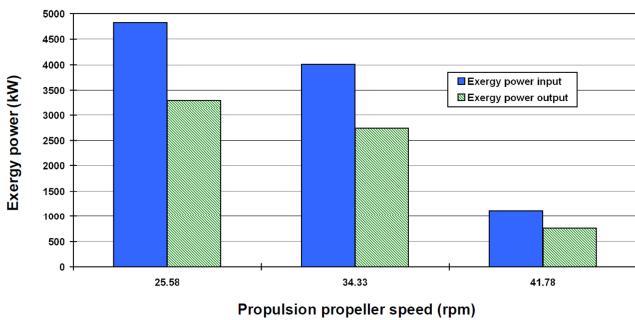


Fig. 4. Change in exergy power input and output of analyzed valve - exploitation

Increase in steam system load causes decrease in pressure reduction valve exergy destruction (exergy power losses), Fig. 5, which is caused mostly because of a decrease in steam mass flow. Pressure reduction valve exergy destruction decreases from 1523.5 kW at 25.58 rpm to 356 kW at 41.78 rpm.

The same decrease trend during the increase in propulsion propeller speed can also be seen in pressure reduction valve exergy efficiency, Fig. 5, which decreases from 68.42 % at 25.58 rpm to 68.09 % at 41.78 rpm.

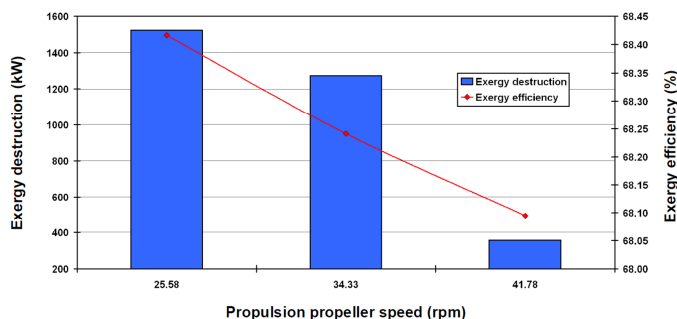


Fig. 5. Exergy destruction and exergy efficiency of the pressure reduction valve - exploitation

5.2. Pressure reduction valve exergy analysis during the ambient temperature variation

Exergy analysis of any steam plant component is dependable on the ambient pressure and temperature. In realistic conditions,

ambient pressure can not be changed for a significant value, so the influence of the ambient pressure on exergy analysis is almost negligible. Change in the ambient temperature is significant for land-based steam plants during the season change (summer/winter). For the marine steam plants change in the ambient temperature significantly influenced exergy destruction and exergy efficiency of any plant component because it depends on the geographical area in which ship operates. It is realistic to expect that the ambient temperature in the analyzed LNG carrier engine room can be changed from 10 °C (when ship operates in Nordic areas) to 50 °C when ship operates in warm areas (for example Persian Gulf).

At any observed pressure reduction valve operating point, exergy destruction (exergy power loss) is the lowest for the lowest observed ambient temperature. An increase in the ambient temperature causes an increase in valve exergy destruction, Fig. 6.

Valve exergy destruction is also related to steam mass flow through the valve; higher steam mass flow resulted in the higher exergy destruction at any steam system load and at any temperature.

Increase in the ambient temperature causes an increase in valve exergy destruction. Between propulsion propeller speeds of 25.58 rpm and 41.78 rpm, valve exergy destruction amounts from 1446.62 kW to 338.05 kW for the ambient temperature of 10 °C, while for the ambient temperature of 50 °C valve exergy destruction amounts from 1651.16 kW to 385.86 kW in the same operation range.

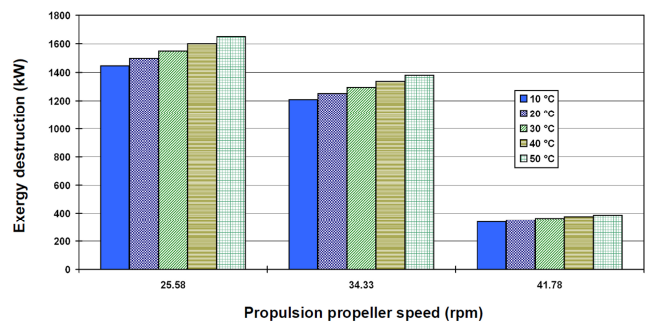


Fig. 6. Change in the exergy destruction of analyzed valve during the ambient temperature change

During the ambient temperature variation, exergy efficiency of the analyzed pressure reduction valve is reverse proportional to valve exergy destruction. The highest valve exergy efficiency is obtained for the lowest ambient temperature of 10 °C (and for the lowest exergy destruction), while the lowest valve exergy efficiency is obtained for the highest ambient temperature of 50 °C (and for the highest exergy destruction), Fig. 7.

At each observed ambient temperature, pressure reduction valve exergy efficiency slowly decreases during the increase in steam system load. Increase in the ambient temperature for a 10 °C causes decrease in valve exergy efficiency for between 2.5 % and 3 %.

When compared analyzed pressure reduction valve exergy efficiency with steam turbines, it can be concluded that valve exergy efficiency are much more affected by the change in the ambient temperature than steam turbines in general, because an increase in the ambient temperature for 10 °C causes decrease in steam turbine exergy efficiency for 1 % or less [5].

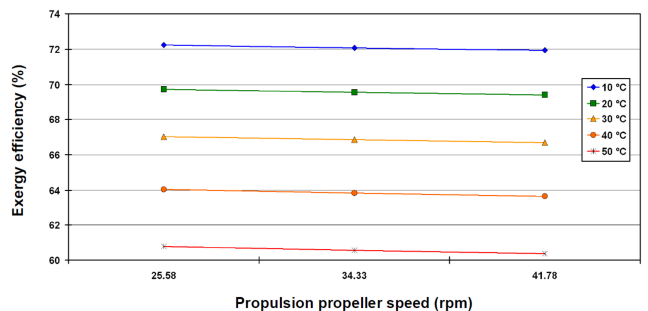


Fig. 7. Change in exergy efficiency of analyzed valve during the ambient temperature change

6. Conclusion

This paper presented an exergy analysis of pressure reduction valve mounted in the steam propulsion system on conventional LNG carrier. The pressure reduction valve is analyzed in two different ways: based on measurement data from exploitation and based on the ambient temperature variation. Analyzed valve is mounted on the steam generators "dump" line, which led superheated steam surplus direct to the main condenser.

From LNG carrier exploitation data are obtained that the valve pressure and temperature decrease become as higher as steam system load increases. Valve exergy power input and output decreases during the increase in steam system load, mostly because of the steam mass flow decrease (superheated steam surplus is as lower as the system load increases). Steam system load increase in exploitation also causes a decrease in valve exergy destruction (from 1523.5 kW at 25.58 rpm to 356 kW at 41.78 rpm) with a simultaneous decrease in valve exergy efficiency (from 68.42 % at 25.58 rpm to 68.09 % at 41.78 rpm).

Pressure reduction valve exergy destruction is the lowest for the lowest observed ambient temperature, in any steam system load. An increase in the ambient temperature causes an increase in valve exergy destruction. Valve exergy destruction is related to steam mass flow through the valve; higher steam mass flow resulted in the higher exergy destruction at any steam system load and at any temperature.

The exergy efficiency of the pressure reduction valve is reverse proportional to valve exergy destruction - the highest exergy efficiency is obtained for the lowest observed ambient temperature (and for the lowest exergy destruction), while the lowest valve exergy efficiency is obtained for the highest observed ambient temperature (and for the highest exergy destruction). An increase in the ambient temperature for 10 °C causes a decrease in analyzed valve exergy efficiency for between 2.5 % and 3 %.

7. Acknowledgment

The authors would like to extend their appreciations to the main ship-owner office for conceding measuring equipment and for all help during the exploitation measurements. This work has been fully supported by the Croatian Science Foundation under the project IP-2018-01-3739.

NOMENCLATURE

Abbreviations:

LNG Liquefied Natural Gas

Greek symbols:

ε specific exergy, kJ/kg
 η efficiency, -

Latin Symbols:

$\dot{E}$ stream flow power, kJ/s
 h specific enthalpy, kJ/kg
 $\dot{m}$ mass flow rate, kg/s or kg/h
 p pressure, MPa
 P work done, kJ/s
 $\dot{Q}$ heat transfer, kJ/s
 s specific entropy, kJ/kg·K
 T temperature, °C or K
 $\dot{X}_{\text{heat}}$ heat exergy transfer, kJ/s

Subscripts:

0 ambient conditions
D destruction
ex exergy
IN inlet (input)
OUT outlet (output)

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FATIGUE ANALYSIS APPROACHES FOR VEHICLE COMPONENTS MADE OF RUBBER

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Abstract: Generally, the most frequently used structural materials are metals which have high strength and stiffness. However, there are many cases, when other important properties come to the fore as well as high deformation by elastic behavior, high viscosity namely good damping effect. Vehicle components made of rubber usually exhibit large deformations. One of the most important properties of rubber is the ability to withstand large strains without permanent fractures. This feature makes it ideal for many engineering applications. On the other hand, the task becomes more complicated because of some features of rubber parts. The temperature of rubber increases significantly after deformations. Material properties of rubber change after these above mentioned temperature changes. Thus it is necessary to understand the mechanics underlying the failure process. This paper summarizes the applied equations and the basic physical laws which are responsible for the theoretical background of the strain and temperature changes and the analysis approaches that are available for predicting fatigue life in rubber, especially in vehicle components made of rubber.

Keywords: RUBBER, HIGH DEFORMATIONS, THERMODYNAMICS, FATIGUE ANALYSIS

1. Introduction

Rubber can be classified as a so-called hyperelastic polymer which has a typical geometrical and material nonlinear behavior. It means that the relationship between displacements and internal forces can be described by functions whose order is higher than linear. The geometrical nonlinearity is easy to handle mathematically, however the material nonlinearity is only described approximately [1], [2]. Independently of the experimental investigations which deal with the material behavior of rubber, a number of theoretical works treated rubber as an ideally nonlinear elastic, in particular hyperelastic material. One of the properties of the constitutive equations of hyper-elastic material is that stresses are derived from stored elastic energy function. Hyper-elasticity can be described by particularly convenient constitutive equation given its simplicity and it constitutes the basis for more complex material models such as elastoplasticity, viscoplasticity, and viscoelasticity [3,4,5,6].

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

It is necessary to summarize the applied equations and the basic physical laws which are responsible for the theoretical background [8,9]. Clarification of these relationships is essential because the material laws of rubber cannot violate those basic physical laws. It is necessary to extend these relationships like balance of linear momentum and balance of angular momentum, the first and second law of thermodynamics to high deformation of rubber and rubberlike polymers. An algorithm will be presented which allows to calculate strain changes and temperature changes of the rubber part of a vehicle component under certain conditions. The present numerical algorithm is the basis of the further fatigue and lifetime-calculations. After this, it will follow the literature survey which is responsible for predicting fatigue life.

2. Governing equations

2.1 Equilibrium of linear momentum

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

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

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

2.2 Equilibrium of angular momentum

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

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

2.3 First law of thermodynamics

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

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

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

2.4 Second law of thermodynamics

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

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

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

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

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

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

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

2.4 Constitutive model

The property of an elastic element is that the total mechanical energy is reversible. The free energy of the body is the function of the strain and temperature. Dissipation comes only from heat conduction.

In order to make the further calculations easier it is necessary to split the Eq. (5) to temperature-dependent and temperature-independent parts. Based on known functions $\tilde{\psi}_0(\underline{\underline{C}})$ and $e_0(\underline{\underline{C}})$ for the free energy and the internal energy at a given reference temperature and the given heat capacity at a reference temperature, one obtains the following general structure for the thermoelastic free energy from the Eq. (5):

$$\psi(\underline{C}, T) = \tilde{\psi}(\underline{C}, \tilde{T}) = \frac{T}{T_0} \psi_0(\underline{C}) + (1 - \frac{T}{T_0}) e_0(J) + \int_{T_0}^T \hat{c}(\underline{C}, \tilde{T}) \left(1 - \frac{T}{\tilde{T}}\right) d\tilde{T} \quad (7)$$

where $\underline{C}$ is the right Cauchy-Green strain tensor [2]. In the following section we are going to investigate the isotrop materials and we are going to apply the Neo-Hookean material law. It means that ψ which is used in free energy depends on the scalar invariant of the right Cauchy-Green strain tensor. The internal energy is zero applying the entropic theory and the heat capacity is constant with good approximation.

2.5 Equation of heat conduction

Starting from the first law of thermodynamics and introducing the internal energy and changing the variable from entropy to temperature, the equation will have the next form:

$$\rho_0 c \dot{T} = \left(\underline{S} - \rho_0 \frac{\partial \psi}{\partial \underline{C}} \right) \cdot \frac{1}{2} \dot{\underline{C}} + \rho_0 \frac{\partial^2 \psi}{\partial T \partial \underline{C}} \cdot \dot{\underline{C}} T - q_0 \nabla_0 + h_0 \quad (8)$$

where $\left(\underline{S} - \rho_0 \frac{\partial \psi}{\partial \underline{C}} \right) \cdot \frac{1}{2} \dot{\underline{C}}$ is the non-recoverable part of the mechanical power, which is zero in the case of a pure elastic element [1], [2]. In this case the rheological model is regarded to be a pure elastic element. So the free energy of the body is characterized by the deformation and temperature. Furthermore, we are assuming that there are not heat sources in the rubber and the temperature field shows homogeneous distribution. Thus, the equation of the heat conduction is the following:

$$c \dot{T} = T \frac{\partial^2 \psi}{\partial T \partial \underline{C}} \cdot \dot{\underline{C}} \quad (9)$$

2.6 Example

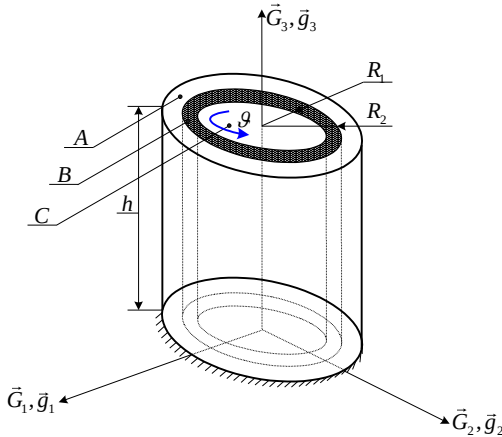


Fig. 1. Mechanical model of a silent block

Let us consider the mechanical model of a silent block, thus the A, B, C axisymmetric bodies (see Fig.2). The A and C bodies are rigid bodies, and B is a deformable one. Regarding the structure of the silent block it consists of two metal elements whose are connected by the rubber which is vulcanized between them. The inside rubber part provides a non-linear elastic connection between the two metal elements in the following way: it transfers loads however filters out the harmful vibrations, i.e. it has damping effect. All three bodies are axysymmetric and their symmetry axes are the same.

The external body (A) is fixed and the internal one is imposed by a given rotation.

Further assumptions:

Planes perpendicular to the symmetric axis will be planes after the deformation. The magnitude of the displacement is linear function of the measured distance from the axes of symmetry. Furthermore, we are assuming that there aren't heat sources in the rubber and the temperature field shows homogeneous distribution, $h_0 = 0, q_0 = 0$. Fig.2 presents the temperature change by the effect of vibration frequency 1 Hertz.

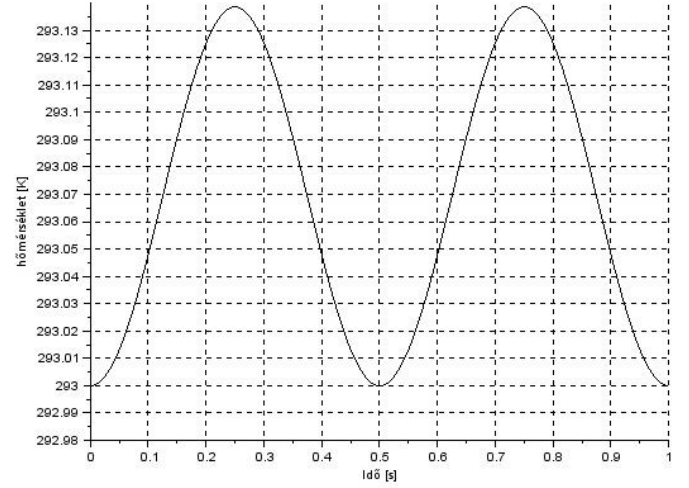


Fig. 2. Temperature change as the function of time

The calculation of temperature changes which is caused by the effect of vibration frequency 1 Hertz is obtained by the using of

Eq.(9), i.e. $c \dot{T} = T \frac{\partial^2 \psi}{\partial T \partial \underline{C}} \cdot \dot{\underline{C}}$. Substitution of the next relation to the

Eq.(9), $\varphi = \varphi \sin(\omega t) \frac{R_2 - R}{R_2 - R_1}$ the temperature changes can be

calculated. Generation of calculations and the representation of the temperature changes as the function of time were determined by the SCILAB program. This calculation will be the basis of the further fatigue and lifetime calculations. In the next a brief literature survey on fatigue analysis approaches will be presented.

3. Literature survey: types of fatigue analysis approaches of rubber

The facility of rubber is the resistance finite strains without permanent deformation makes it ideal for engineering applications e.g. vehicle components made of rubber, as can be seen in the first figure. These loading cases impose large static and time-varying strains over a long period. The fatigue failure analysis have two distinct sections. The first section is a time during which cracks nucleate in regions. The second section is a time during nucleated cracks grow to the point of failure. Two further models for predicting fatigue life in rubber have to be considered. One theory occupies with the predicting crack nucleation life, given the history of quantities like stress and strain in the current configuration. The other theory derives from fracture mechanics, focuses on prediction of the growth of each cracks. Some of the information described in this paper has been reviewed previously [10-15]. Another paper summarizes factors that influence the fatigue life of rubber [16]. These factors are the effects of the mechanical loading history, environmental effects, rubber formulation, and effects due to dissipative aspects of the constitutive response of rubber.

3.1 Models for predicting crack nucleation life

Let us consider fatigue crack nucleation life which is defined as the number of cycles required to cause the appearance of a crack or cracks. The first study of this was Wöhler's work in the 1860's [17]. The maximum principal strain and the strain energy density

are the two widely used fatigue life parameters for crack nucleation approach in rubber. Strain can be directly determined from displacements, which can be measured in rubber. The strain energy density can be estimated from a hyperelastic strain energy density function, which can be defined in terms of strains. The alternating and mean values of maximal principal strain predict nucleation life. The earliest fatigue studies in rubber occupied with the developing of the description of the number of cycles to failure as a function of strain. Cadwell et al. [18] considered unfilled rubber and investigated minimum engineering strains and strain amplitudes in determined range. They stated that, for constant strain amplitude, the fatigue life of rubber improves with extending minimum strain, up to a high strain level, and the additional minimum strain decreased the life. Generally, in the case of strain crystallized rubbers, by increasing the minimum strain of the strain cycle can significantly elongate the fatigue life.

The strain energy density is the second most important parameter which can be used for prediction of fatigue crack initiation. The energy release rate is proportional to the product of strain energy density in certain conditions and the crack size [19, 20]. Several paper can be mentioned in which researchers investigated strain energy density as a fatigue life parameter in rubber. According to the research of Roberts and Benzies [21], and Roach [22], the equibiaxial tension fatigue life of natural rubber is longer than simple tension fatigue life, in the case of using equal strain energy density. Furthermore, Roach gave the best correlation between simple and equibiaxial tension fatigue data, i.e. Roach proposed –in the case simple tension– that all of the strain energy density is available for flaw growth.

3.2 Models by using crack growth approach

The pre-existing cracks or flaws are in the focus of the crack growth approaches. The two main bases of this type of approach are the works which were published by Inglis and later Griffith [23]. Griffith offered a fracture criterion based on energy balance including both the mechanical energy of a cracked body, and the energy associated with the crack surfaces. This approach was further developed by for the case of rubber by Thomas, Lake, Mullins, Lindley and Rivlin. The original application of this approach in the case of rubber was predict to static strength, and later Thomas extended it to the analysis of the growth of the cracks under cyclic loadings in natural rubber. Thomas discovered a square-law context between energy release rate and crack growth rate in the case of unfilled natural rubber. According to Griffith's hypothesis the crack growth is due to the conversion of the stored potential energy to surface energy is in connection with new crack surfaces. He presented that the surface energy associated with the crack faces of a broken glass cane was equal to the elastic energy caused by the fracture. The potential energy (in rubber) released from surrounding material is spent on both reversible and irreversible changes to create the new surfaces. The energy release rate is simply the change in the stored mechanical energy ∂u , per unit change in crack surface area ∂A . This quantity is often called tearing energy T in rubber literature.

$$T = \frac{\partial u}{\partial A}$$

The energy release rate was first applied to the analysis of rubber specimens under static loading, and the above mentioned concept also applied to crack growth under cyclic loading. The experience was that the maximum energy release rate achieved during a cycle determined the crack growth rate, for $R=0$ cycles [24].

4. Summary

An algorithm was presented which allows to calculate strain changes and temperature changes of the rubber part of a vehicle component under certain conditions. In the future I would like to develop a solving computer program in order to apply it as a thermodynamically consistent description. The present numerical algorithm is the basis of the further fatigue and lifetime-calculations. The literature survey will be used to create the connection with the calculations.

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COMPARISON OF NUMERICAL METHODS FOR MODELING THE EFFECT OF EXPLOSION ON PROTECTIVE STRUCTURES

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Abstract: Comparison of numerical methods for modeling the impact of an explosion on a metal plate such as LOAD_BLAST; LOAD_BLAST_ENHANCED; Arbitrary Lagrangian-Eulerian; Particle Blast Method; Smooth Particle Hydrodynamics which are implemented in the program LS-DYNA. The adequacy and accuracy of these methods are evaluated depending on the distance ratio to the explosive charge. The advantages and disadvantages of each method and recommendations for their use based on the results of this modeling and the experience of the authors of this article are presented.

Keywords: EXPLOSION, MINE RESISTANCE, MODELING, LS-DYNA

1. Introduction

The results of the combat losses analysis in armed conflicts over the past decades have shown that a large number of damages to armored combat vehicles (ACV) and their crews are caused by mines and improvised explosive devices [1]. Therefore, a topical solution to the problem of determining the effectiveness of protective structures during explosive loading is to improve of the anti-mine protection of ACV. Nowadays, numerical methods are used for solving physic problems on high-speed processes. This is quite effective tool for modeling processes occurring in conditions of explosive load of the protective structure. Modern software with integrated numerical methods allows us to estimate the influence of design parameters and physical and mechanical characteristics of the material and its elements on the operation of the structure as a whole during impulse influence. Numerical modeling can't replace traditional experimental research methods, but can significantly reduce their number and make the whole process more efficient and improve the anti-mine protection of ACV.

Currently, one of the most popular programs that allows to model the impact of explosion on protective structures is LS-DYNA [2]. Usually researchers choose one of the methods of numerical modeling that is implemented in this software package without substantiation of their choice. Each of these methods has errors for specific modeling conditions, so we must compare its accuracy and adequacy for determining the explosive loads and deflections of armored shell of ACV.

2. Preconditions and means for resolving the problem

2.1. Theoretical Model

The main criterion for assessing the impact of explosion on protective structures is the coefficient of distance Z (Table 1) to the explosive charge, which is determined in such a manner [3]:

$$Z = \frac{R}{\sqrt[3]{W}} \quad (1)$$

where: Z – coefficient of distance to the explosive charge; R – distance from center of charge; W – charge weight in TNT.

Table 1: Conditional classification of the explosion depending on the value of Z

Classification	Z , kg/m ^{1/3}
Far field	$>4 - \approx 40$
Average field	0,4-4
Near field	$\approx 0,053-0,4$
Contact explosion	$\approx 0,053$

The application of the coefficient of distance Z to the explosive charge avoids the attachment to the actual mass of charge,

the distance to it, and conducts an assessment taking into account similarity laws.

The following calculation methods are implemented in the form of commands in LS-DYNA [2]:

LOAD_BLAST;

LOAD_BLAST_ENHANCED;

Arbitrary Lagrangian-Eulerian (ALE);

Particle Blast Method (PBM);

Smooth Particle Hydrodynamics (SPH).

Application of these calculation methods is possible depending on the value of Z (Table 2).

Correspondence of theoretical results with experimental data allows to control the legality of the main references laid down in the calculation scheme, as well as the certainty of the resulting numerical results. As experimental data, these works are selected [4,5].

Table 2: Possibility of application of calculation methods depending on [6]. *Note: in conjunction with the LOAD_BLAST_ENHANCED method

Classification	Z , kg/m ^{1/3}	LOAD_BLAST	LOAD_BLAST_ENHANCED	ALE	PBM	SPH
Far field	$>4 - \approx 40$	Yes	Yes	Yes*	No	No
Average field	0,4-4	Yes	Yes	Yes/Yes*	possibly	No
Near field	0,053-0,4	Yes	Yes	Yes	Yes	Yes?
Contact explosion	$\approx 0,053$	No	No	Yes	Yes	Yes

The general view of the experimental setup for the explosive load of the armor plate is shown in Fig. 1.[4, 5]. When performing research, the armor plate was clamped between two plates (Fig. 1). The explosive charge of trinitrotoluene (TNT) in the form of a sphere was placed at different distances (R) from the target plate. The value of the charge weight also varied. The scheme of conducting an experiment is shown in Fig. 1b.

Cowper-Symonds model (2) for the material of plate was used in the course of numerical modeling [2].

$$\sigma_T = \left[1 + \left(\frac{\dot{\varepsilon}}{C} \right)^{1/p} \right] \left(\sigma_0 + \beta E_p \varepsilon_{eff}^p \right) \quad (2)$$

$$E_p = E_{tg} E / (E - E_{tg})$$

$$\varepsilon_{eff}^p = \int_0^t \left(\frac{2}{3} \dot{\varepsilon}_{ij}^p \right)^{1/2} dt,$$

where: C – parameters of strain rate; $\dot{\varepsilon}$ – equivalent strain rate;

σ_0, σ_T – Static and dynamic yield stress; E – Young's modulus;

E_{tg} – hardening modulus.

The equation of the state of explosive detonation products [2]:

$$p = A(1 - \frac{\omega}{R_1 V}) \exp\{-R_1 V\} + B(1 - \frac{\omega}{R_2 V}) \exp\{-R_2 V\} + \frac{\omega}{V} \bar{E}, \quad (3)$$

where: $V = \rho_0 / \rho = v / v_0$ – relative specific volume; $A, B, C, R_1, R_2, \omega$ – empirical constants; $\bar{E}$ – internal energy.
The equation of state of air [2]:

$$p = e \cdot (\gamma - 1) \rho / \rho_0 \quad (4)$$

where: $\gamma = 1,4$ – indicator of air adiabatic.

The mechanical characteristics of the armor plate are summarized in Table 3. Characteristics for the equation of the state of explosive detonation products are given in Table 4.

Table 3: Characteristics of the plate material [4, 5]

Characteristic	Value
Young's modulus, GPa	210
Poisson's ratio	0,28
Static yield stress, MPa	950
Dynamic yield stress, MPa	1250
Relative elongation, %	9
Density, kg/m ³	7838

Table 4: Characteristics of the equation of state of detonation products [7]

explosive index	Chapman-Jouget Parameters, *MAT_HIGH_EXPLOSIVE_BURN				
	ρ_0 kg/m ³	P GPa	D m/c	E_0 GJ/m ³	G
TNT	1630	21,0	6930	7,00	2,727

The coefficients of the state equation of detonation products in the form Jones-Wilkins-Lee, *EOS_JWL

A	B	C	R_1	R_2	ω
GPa	GPa	m ³ /kg			
371,2	3,231	1,045	4,15	0,95	0,30

2.2. Experimental stand

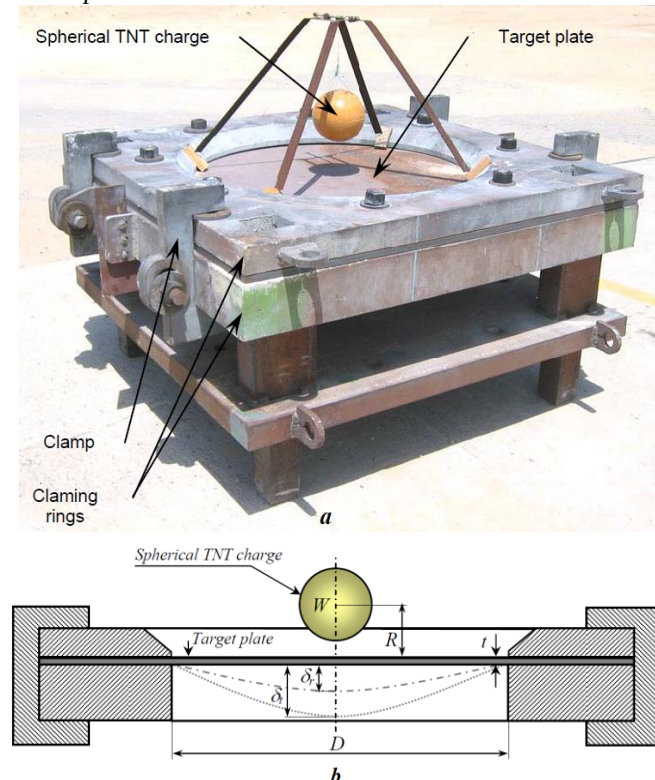


Fig. 1 The general view of the experimental setup (a) and the scheme of conducting an experiment (b) [4,5]: R – distance from center of charge, t – plate thickness; D – plate diameter; δ_r – maximal mid-point transient deflection; δ_r – residual mid-point deflection.

2.3. Numerical modeling

The plate on which the explosion acts, is modeled as a three-dimensional, four-node, shell finite elements (Shell) and three-dimensional, eight-node, solid elements (Solid) in all variants. Accordingly, in each calculation scheme, the weight of the explosive varied and the distance to it varied.

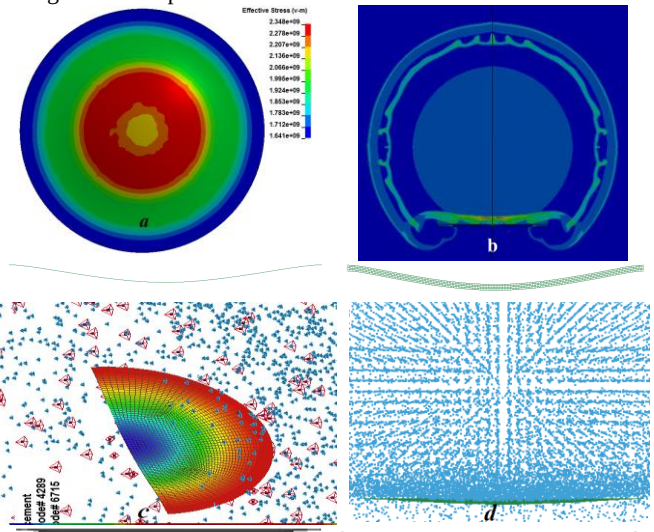


Fig. 2 Different methods of numerical modeling of explosive loading of a plate: a – LOAD_BLAST ma LOAD_BLAST_ENHANCED; b – ALE 2D; c – PBM; d – SPH; under each of the figures - maximum deflections of the plate with the application of the appropriate method

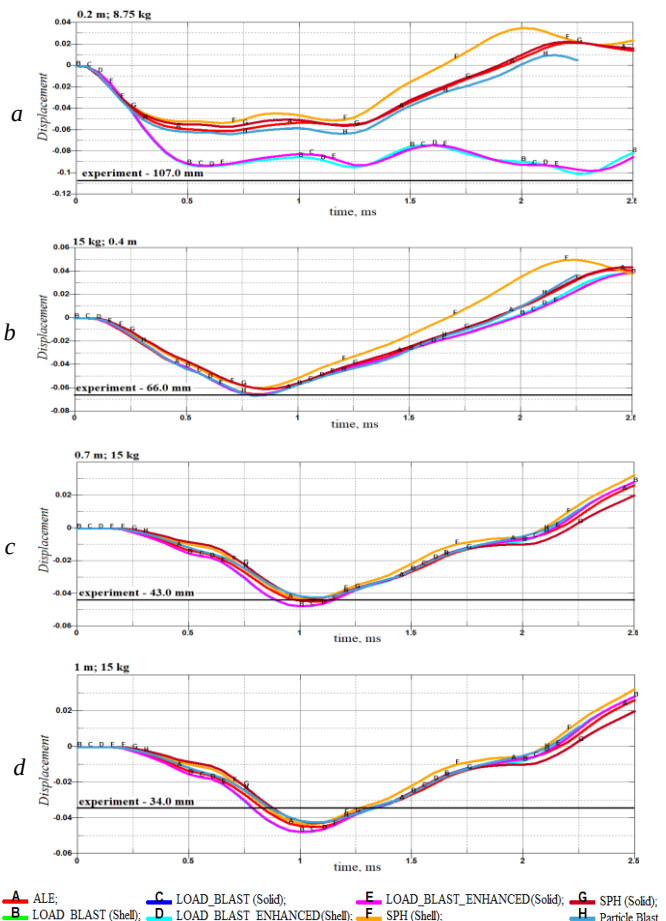


Fig. 3 Graphs of the displacement of the central node of a plate during the simulation of the explosion of charge with the use of various calculation methods: a – $W=8,75$ kg, $R=0,2$ m; b – $W=15$ kg, $R=0,4$ m; c – $W=15$ kg, $R=0,7$ m; d – $W=15$ kg, $R=1$ m.

Table 5: Results of numerical modeling of the impact of explosion on a plate using various calculation methods

Calculation method	W, kg	R, m	δ_r , mm experiment	δ_r , mm calculation	Δ , %	Time of calculation, s	Data volume, MB	Dimensions of a finite element, mm	Number of finite elements, units.	Z, kg/m ^{1/3}
LOAD_BLAST (Shell)	8,75	0,13	165,0	109,7	33,5	81	41,8	12x12	4380	0,063
		0,2	107,0	100,9	5,7	78				0,097
	15	0,4	66,0	66,2	0,3	79				0,162
		0,7	43,0	47,7	10,9	80				0,284
LOAD_BLAST (Solid)		1	34	34,6	1,8	83	128	14x14	18150	0,405
	8,75	0,13	165,0	105,2	36,2	377				0,063
		0,2	107,0	98,1	8,3	374				0,097
	15	0,4	66,0	66	0,0	371				0,162
LOAD_BLAST (ENHANCED) (Shell)		0,7	43,0	47,6	10,7	367	55,8	12x12	4380	0,284
		1	34	34,6	1,8	371				0,405
	8,75	0,13	165,0	109,8	33,5	98				0,063
		0,2	107,0	100,9	5,7	96				0,097
LOAD_BLAST (ENHANCED) (Solid)	15	0,4	66,0	66,3	0,5	88	151	14x14	18150	0,162
		0,7	43,0	47,7	10,9	86				0,284
		1	34	34,6	1,8	85				0,405
	8,75	0,13	165,0	105,2	36,2	376				0,063
SPH (Shell)		0,2	107,0	98,1	8,3	374	498	12x12	65752 (SPH)	0,097
	15	0,4	66,0	66	0,0	366				0,162
		0,7	43,0	47,6	10,7	361				0,284
		1	34	34,5	1,5	367				0,405
SPH (Solid)	8,75	0,13	165,0	84,1	49,0	608	583	14x14	18150(Solid)	0,063
		0,2	107,0	54,5	49,1	596				0,097
	15	0,4	66,0	59,6	9,7	585				0,162
		0,7	43,0	43,2	0,5	505				0,284
ALE 2D (Solid)		1	34	27,2	20,0	498	2100	5x5	196400	0,405
	8,75	0,13	165,0	97,8	40,7	2630				0,063
		0,2	107,0	56,5	47,2	2421				0,097
	15	0,4	66,0	60,8	7,9	2305				0,162
Particle Blast (with air) (Shell)		0,7	43,0	42,6	0,9	2032	2100	5x5	100 000 (PBM)	0,284
		1	34	25,9	23,8	1918				0,405
	8,75	0,13	165,0	70,8	57,1	37850				0,063
		0,2	107,0	61	43,0	37630				0,097
Particle Blast (with air) (Shell)	15	0,4	66,0	65,1	1,4	21030	2100	5x5	100 000 (PBM)	0,162
		0,7	43,0	44,8	4,2	8951				0,284
		1	34	32,1	5,6	9748				0,405
	8,75	0,13	165,0	100,9	38,8	136				0,063
Particle Blast (with air) (Shell)		0,2	107,0	63,8	40,4	133	2100	5x5	100 000 (PBM)	0,097
	15	0,4	66,0	66,4	0,6	130				0,162
		0,7	43,0	42,2	1,9	122				0,284
		1	34	42,1	23,8	65				0,405

3. Result and Discussion.

The obtained simulation results using the considered calculation methods showed high accuracy in comparison with the experimental data, which were selected as reference ones. An additional point is that obtained results correlate with results of other works. [8-10].

The use of Solid or Shell elements in the numerical model has approximately the same level of precision while solving tasks. However, the amount of data needed to store the solution of the same task for Solid elements will be on average three times more than Shell. It is clear that the time of solving the problem with the use of Solid elements will be significantly higher compared with the use of shell elements. Such conclusions aren't new and completely logically proceed from the mathematical content of both elements. For the majority of tasks regarding the assessment of the impact of explosions on protective structures, it is quite sufficient to use Shell elements except for the presence of significant plastic deformations and the destruction of structural elements. In this case, the use of Solid elements will be expedient.

Application of the LOAD_BLAST and LOAD_BLAST_ENHANCED method has significant advantages over other methods because of their ease of use, relative simplicity of the preparation of the finite element model, the smallest volumes of data compared with other methods, the smallest hardware requirements for computers. The disadvantages of such calculation methods are the impossibility of taking into account "shading" by one design of another, the absence of reflection and overlay of shock waves, which leads to an increase of measure of inaccuracy in the evaluation of the protective properties of complex structures. By the way, there is a fairly small choice of charge explosive form. The effect of a spherical charge in the air or on the surface of the soil is integrated in this calculation method. This requires a recalculation of the charge value of the charge, taking into account the soil and coefficients taking into account the shape of the charge. LOAD_BLAST_ENHANCED further allows us to estimate the pressure acting on the protective structure on its surface and the effect of the negative phase of the shock wave (this isn't counted in LOAD_BLAST).

Using the SPH method allows us to take into account processes of reflection and overlaying of shock waves, but requires considerable hardware resources. In addition, this method is very sensitive to grid density, the number of SPH elements, and the correct choice of contact between the elements of the SPH and the elements of the Lagrange.

The ALE method requires a lot of hardware resources, requires the greatest amount of time in order to calculate the task and consumes the highest volume of data to note.

The Particle Blast method has the advantage because of the lack of a mesh, needs a relatively small amount of data to note, has a fairly high accuracy and high speed of problem solution. The application of this method is quite promising at the present time.

4. Conclusion

The obtained results show that for the conditions of charge demolition under the bottom or chassis of the ACV, all the methods of modeling the impact of the explosion described in this article have high adequacy and accuracy. The use of each method is expedient in view of their disadvantages and advantages at the discretion of the researcher. Taking into account the authors' experience, it is expedient to carry out the research of the protective structure in the initial stages using the methods LOAD_BLAST and LOAD_BLAST_ENHANCED. We should use one of the methods such as Arbitrary Lagrangian-Eulerian, Corpuscular Particle Method, Smooth Particle Hydrodynamics in order to receive more detailed information about the protective structure.

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TRANSPORT AND ENVIRONMENT: NEED FOR SUSTAINABLE SOLUTIONS

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Abstract: Transport has irrepressible contribution for the effective development of economy and society. But, these contributions are exceeded by harmful environmental consequences that transport growth causes especially in urban areas. Air pollution, acid rain, noise, climate change are just few of the problems which every urban area is dealing with. Hence, it is very important to provide development of the transport that at the same time could increase its sustainability and minimize environmental degradation. Numerous sustainable transport solutions are available with a potential to decrease these negative influences and promote mobility with higher sustainability. This paper presents a review of sustainable solutions, successfully applied in practice, as guidance for those urban areas that are dealing with the environmental problems from transport, but still are a step behind in undertaking activities.

KEYWORDS: TRANSPORT, ENVIRONMENT, SUSTAINABILITY

1. Introduction

Transport is a cause for many serious environmental problems, especially in urban areas. Urban congestion with the current trends of transport growth is expected to be worse. Although the significant reduction in the vehicle exhaust emissions has been accomplished, the poor air quality remains as a biggest problem [1]. Transport also contributes for other environmental problems, to mention just the formation of acid rain and global warming as more significant. Therefore, to be able to control or minimize these problems, urban transport system needs to be improved towards sustainability.

In order to recommend sustainable solutions, firstly, a categorization of environmental problems for which transport has significant impact, is presented. Then, several sustainable solutions are proposed, selected from the current practice from European towns, where their implementation has given good results. These solutions could be viewed in the term as a starting point for those urban areas, such as macedonian towns, that are dealing with serious environmental problems caused by transport, but still don't have a starting point for effective definition or taking the initiative for concrete solutions.

2. Transport and environment

2.1. Environmental impacts from transport

The high dependence of the vehicles in urban transport leads to the inevitable harmful impacts at the environment. Although the vehicles performances have been continuously improved in term of better energetic efficacy and lower emissions and noise levels, the increasing number of vehicles in urban areas diminishes all of these improvements [2].

Taking aside the number of vehicle and the level of congestion at urban streets, the range of harmful impacts from the vehicles depends of:

- engine type (diesel, petrol)
- vehicle category (passenger cars, high duty vehicles, buses etc.)
- technological – constructive characteristics
- fuel type (fossil fuel or alternative fuel)
- the mode of vehicle usage and
- the level of vehicle maintenance.

Dealing with the serious problems in the environment is feasible only if those problems are precisely defined. Therefore, when selecting the solutions for control and management of environmental problems, categorization of the impacts that transport has on the environment is of vital importance (table 1).

Table 1: Categorization of environmental impacts caused by transport

Category	Impacts
local level	- urban air pollution, with a direct health effect - water pollution - soil pollution - waste - noise - esthetic impact and fragmentation.
regional level	- acid rains
global level	- global warming and climate changes - decrease of the ozone layer - decrease of the natural resources

At local level, the location where emissions are generated is the location of their impact. At regional and global level the location of emission generation is of less of importance. For example, the health impacts of the emissions of particles and hydrocarbons are highest in the urban areas, where their concentrations are highest; carbon dioxide emissions are most significant for the global level, and the location of the sources of carbon dioxide here doesn't have any significance.

The bigger part of the impacts is felt in urban areas, where higher concentrations of vehicles are located. Hence, urban areas have priority for study and research of the problems with exhaust emission pollution [3].

At local level, nitrogen oxides (especially nitrogen dioxide), hydrocarbons, particles and carbon monoxide cause air pollution, which results with numerous health issues (table 2). The regional impacts are presented through acid rains, with nitrogen and sulfur oxides having the biggest contribution for their formation. At global level, vehicles emit carbon dioxide, whose increase of concentrations brings to the climate change and global warming.

At global level, transport contributes for worsening of the problem with global warming. This is connected with a change of many climate aspects, for which higher frequency, duration and intensity is predicted in the future:

- weather events: storms, fires, floods, droughts, heat waves etc.
- changes in the ecosystems
- losses of species in flora and fauna
- changes in agricultural production (decrease of crops) etc.

It is evident that increase in the global temperature will influence at all the aspects of the planetary eco-system and ecological balance of the earth [4].

Table 2: Range of impacts and effect of pollutants from vehicle exhaust emissions

Emissions	Impact	Harmful effects
Carbon dioxide	global	climate change
Carbon monoxide	local	health impacts
Particles	local	health impacts
Hydrocarbons	local and regional	health impacts, ozone formation
Lead	local	health impacts
Nitrogen oxides	local and regional	health impacts, ozone formation, acid rain
Ozone	local	health impacts
Sulfur dioxide	local and regional	health impacts, acid rains
Benzene	local	health impacts

2.2. The range of environmental impacts: data for environmental degradation

According to the report of the European Commission, it has been estimated that in 2014 specific air pollutants have caused the following state of premature mortality in EU-28 [5]:

- PM_{2.5} have caused around 399,000 cases
- NO₂ have caused around 75,000 cases
- O₃ have caused around 13,600 cases.

The consequences from the exposure to specific pollutant are defined by:

- pollutant concentration
- exposure duration in the polluted areas.

Concentrations of local pollutants, especially particles and nitrogen oxides, are several times higher near the roadways compared with the total ambient levels. Air pollution is higher in those urban areas where geographic and climatic conditions trap pollution and produce ozone.

Highly exposed groups are:

- persons living near the roadways with high capacity flows
- persons in the "street canyons" with high traffic intensity
- persons spending long time on roads because of their profession.

The most exposed group are the drivers themselves, who constantly travelling through "tunnel" of polluted air can be exposed more than three times higher local air pollution, compared with the cyclists and pedestrians near the roads [6].

Referring acid rains, it has been projected that until 2020 only 4% from the total eco-systems in EY-28 will exceed the critical levels of acidification, if the current legislation is completely implemented [7].

IPCC (Intergovernmental Panel on Climate Change) considers that the ambitious reduction of 50-80% of the global CO₂ emissions is needed until 2050 (compared with 2000 levels), in order to achieve [8]:

- limitation of the temperature increase from 2-2.4°C
- stabilization of CO₂ concentrations to 450 ppm.

However, the IPCC foresees that until 2100, by different scenarios, the increase of the global temperature will be in the range 0.9 - 4 °C [8].

3. Sustainable transport solution

Sustainability in transport could be achieved through solutions that are relevant and important in longer time period. Sustainability options should be implemented after qualitative and extensive planning and more importantly, through integration of urban and transport planning and designing.

Sustainable solutions applicable in urban areas dealing with the environmental problems could be categorized in several categories (table 3):

Table 3: Categorization of sustainable solutions in transport [9]

Sustainable solution	Activities
Transport demand	- road fees, - parking fees, - city center fees (cordon fees), - public transport fees, - regulation for land use planning.
Traffic management	- parking restrictions, - low-emissions zones, - traffic calming, - speed limitations, - bus priorities, - other physical and regulative measures for traffic flows and modal distribution.
Effectiveness of transport mode	- mobility management, - promotion of walking and cycling - good deliveries from the terminals to the urban centers.
Improvement of transport services	- effective public transport - effective passengers' transfer.
Introduction of clean-fuel vehicles	- alternative fuels.
Public campaigns	- information campaigns - awareness campaigns.

Actually, sustainability in transport is a long-term policy in which sustainable balance between mobility and protection of the environment is defined and should be accomplished. Proposed sustainable (and other) solutions should be combined between each other for bigger effectiveness and should be based at clearly defined and implemented political obligations.

3.1. Transport demand

Introduction of fees

When introducing fees in transport, it is necessary to build a consensus at national and local level. This is very feasible and economic effective approach, although very often has political connotation. Local authorities, urban planners and transport operators have available practical guidance for evaluation of the costs and for financing of urban transport systems. Electronic payment on road is very popular, but also other payment measures, such as parking fees and cordon fees, are more financially effective and more practical [9]. Cordon fee is partially effective when used in congested central areas or during the peak periods. Parking fee is also effective referring the limitation of cars in urban areas where exists a continuous high demand.

Integration of transport and land use planning

Research shows that there isn't a simple strategy which significantly will decrease traffic congestion levels in urban areas in short or long term. Hence, a policy is needed that will change the way of land use in a long term, which will be directed to the reduction of the vehicle usage. Very good lessons are identified from the current practices for combined land use planning and transport planning [9]:

- these policies are successful only if the attractiveness of car travelling is decreased (slower or more expensive travel)
- policies directed to the limitation of the car in city centers aren't harmful for the economic progress of those centers.

3.2. Traffic management

Implemented solely or in combination, suggested activities for traffic management can improve environmental impacts from transport and support attractiveness of green transport policies. This can contribute for measurable difference of the urban life quality, encouraging the local authorities to meet specified policy targets.

3.3. Effectiveness of transport mode*Mobility management*

Mobility management includes information services and mechanisms for coordination for better use of the current transport capacities, which could minimize the number of travel by car. This could result with a change of the transport mode, for example, through green travel plans, car pooling, dedicated lanes for high occupancy vehicles etc [9]. Mobility management also tries to make more effective usage of current public transport developing strategies for improvement of public transport services.

Cycling and walking promotion

Different cities have different success in the usage of this strategy, and hence there is a strong interest for learning from the wider European experiences [9]. Providing the continuous network of walking and cycling lines, segregation of the walking and cycling lines, increasing the number of parking places for cyclists in urban centers, reduction of the vehicle speed to 30 km/h are just some of the measures that could contribute for promotion of cycling and walking.

3.4. Improvement of transport services

An effective framework should be directed to the improvement of the performances of public transport through controlled competition, implementation of effective public transport services and financial support for the operators. Research shows that if improvement of public transport services are solely implemented, they are far ineffective for the stimulation of modal change. In combination with other measures, such as parking and cordon fees, zones with limited access etc. contribute for the highest reductions in number of car travels.

For encouragement of the passenger to use public transport, passenger transfer between different transport modes should be especially efficacious and user-friendly [9]. In the past, the attention was put on the operators' effectiveness. Today, practical guidance for users, planners, public transport operators and managers of the terminals are developed. Also, market research and modeling tools for designing more efficient transfer are developed, available through manuals or Internet [9].

3.5. Introduction of clean-fuel vehicles

In the next several decades, Europe should change its fuels sources in transport, in order to decrease the high dependence of imported oil. Alternative fuels contribute for the mitigation of global warming and air quality improvement. Cleaner vehicles provide improvement of urban area, but they are facing with the barriers

such as high price and lack of fuel infrastructure. Policies for support of alternative fuels are [9]:

- fiscal stimulations for introduction of particular fuel at the market
- definition of low-emission zones with access only for clean vehicles
- contractions between local authorities and operators for promotion of clean vehicles and regulatory framework for their implementation by the national government.

3.6. Public campaigns

Information campaigns give information for all the changes in urban transport and for future transport plans supporting transport modes directed to the sustainable transport solutions. Awareness campaigns provide change in travel behavioral and should be repeated in specific time intervals – in contrary, they will lose their power to impact at the behavioral. If both types of campaigns are directed to specific groups: schools, working places and local communities, they have stronger and long term effects [9].

4. European good practices for sustainable transport solutions

The average modal change achieved by these sustainable solutions for seven European towns where testing was performed, showed a reduction of 4.5% in the usage of vehicles and increase of 12% in the usage of public transport [9]. This corresponds with an energy savings of 7% and similar reductions in the carbon dioxide emissions. Particle emissions were reduced for almost 20% [9]. Usage of the information systems, buses monitoring and driver/passenger information is the most effective measure that decreases the travel times and reduces the emissions for 20% [9].

Modeling results show that fees could reduce traffic flows by 30% in the peak period, mostly by encouraging the drivers to travel in different time or using different routes [9].

The best results for the environment were achieved for the towns who decided to limit the access for cars in order to improve public transport system [9]. This implies that the new vehicle technologies are most effective when combined with the transport management measures.

Some towns achieved reduction of 50% in the emissions using vehicle with alternative fuel, which positively had impacted at local air quality [9]. Electrical energy and gaseous fuels were the most tested option, as well as biofuels and hybrid vehicles.

5. Research development for transport: current and future trends

Research in the transport has shown that the problems which urban transport is facing with could be surpassed only with radical change, introducing proved and new sustainable solutions as a part of one integrated strategy [9]. Research defines a range of advantages that are feasible and provides information base and useful recommendations as a support for the future implementation of these and other solutions.

Directions for the current research are:

- definition, implementation and testing of strategies which integrate more sustainable solutions
- in depth analysis of the results for selected and implemented options
- development of new solutions, for example integrated payment schemes, integration of intelligent systems for management of transport and passenger services etc.

These directions are designed to prove that integrated strategies are capable of achieving significant change in the modal change and reduction in traffic congestion [9].

Research shows that payment measures have the potential to be one of the most used in urban transport management. Still, additional efforts are needed for their practical implementation

before these measures will get the necessary social and political acceptance.

Future research referring sustainable solutions will have a need for a wide range of support tools, such as [9]:

- compilation of good practices
- definition of indicators for environmental and urban life quality
- methods for selection, design and assessment of suitable and applicable sustainable solutions
- experience sharing and training of all the stakeholders involved in the process of transport sustainability.

6. Conclusion

Transport is a cause for very serious environmental problems today, ranging from local to global level. In presented analysis, categorization of sustainable solutions is given using European good practices that have a potential to decrease or mitigate negative environmental impacts from transport. The aim of the analysis is to increase the awareness for the importance of sustainability course in transport and to give support to the policy creators who design and plan implementation of these sustainable solutions.

The European cities today have implemented different sustainable solutions in transport. The main acquisition of their success could be merging of those experiences and as a result creation of information, methods and instructions for more wider and more successful use. This is of special importance for those urban areas who are facing with environmental degradation from transport, but who are still lacking in ideas or activities to deal with the problems. The paper could serve as an initiative intended to start with a change in their transport system towards sustainability.

For better effect, a combination of solutions could be created, if financial or political circumstances allow, using the mutual experiences and practical suggestions from urban planners from different cities. Those solutions with improved communication of

involved stakeholders and better cooperation will distinguished themselves as dominant and more effective solutions.

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THE METHODOLOGICAL FOUNDATIONS FOR CREATING THE SMALL-CAPACITY PORTS ON THE BLACK SEA COAST OF GEORGIA

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Abstract: *As of today, establishment of small port network has an important place in the development of small and medium businesses in maritime countries. They are among the most important tools having a positive impact on sustainable and equal development of small and medium businesses in the country. Small ports appear to us as the centers of regional economic activities. Export technology from a small port is carried out much faster and easier compared to the large deep-water port hubs.*

A small port equipped with appropriate infrastructural facilities may also combine functions of harbor required for small fishing vessels and yachts. A small port can easily develop the yacht and cruise tourism.

KEYWORDS: SMALL AND MEDIUM MARITIME BUSINESS, EFFICIENT SHIPPING, FISHERIES, YACHT-CLUB, CRUISE TOURISM.

1. Introduction

Georgia has been and remains the main land-sea transit hub in the whole South Caucasus. That is why the improvement of transport infrastructure is essential to the country's economic and social development.

The ports and harbors being subject to the border control, are the places for implementing cargo transfer, administrative service and trade procedures, as well as connecting link of land and maritime transport routes and are one of the most important parts of the transport and logistics chain. Within the TRACECA transport corridor, they accord special significance, since this transport corridor crosses the Black and Caspian seas, and its effective functioning as an international transport corridor largely depends on the successful operation of the port and the maritime transport sector.

Navigation in the the Black Sea dates back to the Antique time, thus the Black Sea region countries and peoples have ancient maritime traditions and history, including in terms of international traffic. These historical circumstances still play an important role in the processes of reflection of the current situations and objectives that have arisen in the development of maritime affairs in our country.

The multi-modal transport corridor crossing the Black Sea through Eurasia competes mostly with road and railway routes crossing Russia, and to a lesser extent, with other land routes passing through the territories of Iran and Turkey. However, transit traffic flows, which these corridors are trying to apply, are mostly transported by sea between the Asian and European ports of the Caspian, Black and Mediterranean seas. According to expert estimates, this tendency will be maintained in the long run, and it is therefore important to correctly determine the strong and weak points of the transportation of cargoes through the Black Sea, and how the existing system can be developed to make maritime transport more effective in the near future. This will allow us for determining where and how the land transport corridors will find their functioning area in the common intercontinental transport-logistics networks to increase their share in the transport service market and eventually become an indispensable additive for marine transportation.

The Black Sea ports experience intensive development due to continuous growth of imports and exports, as well as owing to rapid modernization and investments. There are being constructed new powerful terminals in Ukraine and Russia, and the process of

privatization of ports in Turkey and the concession process in Bulgaria are still ongoing. The structure of cargo flows is changing, containerization process is moving fast.

2. Preconditions and means for resolving the problem

Around 80% of international trade is carried out by maritime transport, which in most cases form the so-called "major stakeholder" of shipping. Key elements of the marine-river transport network are harbors and ports, which, at the same time, represent the territories of certain size equipped with the appropriate infrastructure. These territories are used for business activities and often have a special status. Modern trends in the development of international maritime traffic have dictated the need for creating logistics hubs, which play the role of the largest regional distribution centers, effectively carry out consolidation of bulk cargo and shipping operations.

It should be noted that the operation of large-capacity vessels is accompanied by a reduction in the number of their direct entries into ports (since from the technical and technological standpoint, not all ports can serve these types of ships), due to which it is necessary to use the shipment system on the principle of the "hub and spoke", which lowers the quality of process flexibility, and presumably, we obtain the lower effectiveness of the frequency of the voyages.

Cargo carriers believe that all this results in failures in the supply chain, delays, creation of large stock overhangs, and therefore, an increasing need for working capital, which often becomes a main problem for powerful port-hub.

To increase the effectiveness of the frequency of the voyages of large-capacity vessels, carriers sometimes use slow movements, which is regarded by them as the principal response to to excess transportation capacity of ship and the increased fuel prices, which ultimately forces consignors to search for even greater volume of capital investment with objective of storing increased stock. Today, consignors have questioned savings that they would reap as a result of slow movements of large-capacity ships.

In most of the deep-water ports, goods are delivered by railway transport, for which the cargo owner must order wagons at least a month in advance, in besides, the acquisition of a quota in the deep-water terminal is relatively more difficult. Finally, shipping requires more time, documents and efforts. Export of some

types of goods (such as cereal), from a deep-water large port-hub requires up to three months, while the same process from a small port may be completed within a few weeks.

Thus, today, the establishment of small ports network is an important matter in the development of small and medium business in maritime countries. For example, in addition to large port-hubs (Liepaja, Ventspils, Riga, etc.), Latvia has also seven small ports in the Baltic Sea (Pavilosta, Roja, Mersrags, etc.). They represent one of the most important instruments that contribute positively to sustainable and equal development of small and medium businesses in the country. Smaller ports appear to us as the centers of regional economic activity [1].

The establishment of small ports network on the Black Sea coast of Georgia can be considered to be one of the directions of the implementation of the plan of the effective spatial arrangement of business enterprises and transport infrastructure facilities declared by the government of our country.

Export technology from a small port is implemented much faster and easier, compared to large deep-water port-hubs, because in small ports, it is possible to deliver goods to the terminal directly by road transport, and this is very cost-effective, since the preliminary reload and transportation from the cargo yard or from the warehouse to the terminal is excluded. In case of transportation with such technology, in a small port, it is possible to load several ships with capacity of 3-5 thousand tons per day quickly and directly from vehicles.

The main direction of Georgia's small ports activities should be provision of fast and effective processes of transportation of goods produced in the region, as well as small transit cargo (including perishable agricultural products, which require special packaging and transportation conditions), to European countries and in the reverse direction. A small port equipped with appropriate infrastructural facilities may also combine functions of small fishing vessels and yachts [2].

Today, the the volume transported by small ports is relatively small, compared to large port-hubs, but in terms of small and medium business development in the country, small ports are an integral part of the development of the country's coastal regions. Smaller harbors create favorable preconditions in coastal areas for developing such sectors as fishing and fish processing, primary processing of fruits and vegetables, tourism, marine and land transport [3].

State or private investments in the establishment of small ports and their infrastructure will be positively reflected on the reduction of maritime traffic risks and increasing their safety, minimizing environmental pollution risks and facilitating the improvement of the functioning of small ports and harbors.

Small harbors are an important actors in small and medium logistics businesses that perform not only export-import, but also industrial and cargo processing operations. The priority directions of the work of the Black Sea small ports should be as follows:

- transportation of local and transit small cargo between the ports of the Black and Mediterranean seas and their tributaries;
- receiving, dispatching and locating small fishing vessels;
- yacht and cruise tourism development.

An important place in activities the small Black Sea port can be taken by transportation of agricultural goods produced in the South Caucasus region within the effective logistics chain "Field-motor transport - small port-marine transport- destination marine or river port". The high effectiveness determiners of the shipment process from a small port are: the use of motor transport distinguished for

high maneuverability and velocity directly in the process of loading and unloading; small amount of time for transportation and an intensive turnover of marine vehicles between the small ports. For example, forecasts show that in the nearest years, the small seaports' cargo turnover will be increased considerably. A small port should have the petroleum storage reservoirs and warehouses, the border sanitary control point, the distribution and logistics center, and also, it is possible to implement the industrial fleets and wind generators projects. The works should be performed to attract private investments in construction and reconstruction of the petroleum storage reservoirs; with the support of the EU, it is possible to develop the cruise traffic projects.

With a view to increasing brand strength of Georgia, as a seaside resort country and improving the living condition standards of the local population in the country, a significant aspect of activities of ports should be the yacht tourism. With the support of private investments, it is possible to establish a small yacht-building enterprise, as well as to create infrastructure for yacht tourism. At present, there is every reason for creating in our coastal zone small harbors with modern and developed yacht-tourism infrastructure, which will not only promote the development of yacht tourism in the South Caucasus region, but also in the perspective, it will become an important attribute of attracting tourists wanting to "travel around the Black Sea" by yachts. **3. Conclusion**

The following complex objectives should be solved to achieve the set goal:

1. To analyze and activities of small ports and harbors existing in the Black Sea area and their development prospects;
2. Determining the role and place of a small port important, as an important historical and economic factor of ensuring the intensive international business links;
2. To systematize the theoretical approaches to determining the small ports' parameters;
3. To identify the role of a small port's transport complex in the economic development of the region, based on the theoretical analysis of the experience of other maritime countries;
4. To define the principles of mutually beneficial cooperation between large ports and promising small harbors existing in the region;
5. To provide a feasibility study to establish small ports network on the Black Sea coast of Georgia.

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WHY COMMUNICATION THROUGH SOCIAL MEDIA IS IMPORTANT FOR RAILWAY BUSINESS PUBLIC RELATIONS?

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Abstract: Railway transport has been seen as a backbone for sustainable transport system around the world. It is an essential contributor to the economies and the quality of life in many developed countries. In order to increase the volume of railway passenger transport, the railway managers have to influence customers' satisfaction and to improve the quality of rail services. One of the communication tools railway companies uses to reach their passengers is Facebook. In this paper, the comparative analysis among different railway companies in selected European countries is done, in order to track good and less good examples of the usage of sites for social networking, i.e. Facebook. There are defined relevant criteria for making this relative comparison. After the analysis is done, the measures for improving the relationships with railway passengers are proposed.

Keywords: RAILWAY PASSENGERS, MARKETING, COMMUNICATION TOOL, FACEBOOK

1. Introduction

Railway transport has been seen as a backbone for sustainable mobility. It is the most environmentally-friendly, but also economically-cost effective, mode of transport. However, in many countries around the world the train utilization is pretty low, due to low customers' satisfaction. There are many marketing activities that can be conducted, in order to manage a customer satisfaction.

One way, in this regard, is to communicate with customers through social media, especially through networking sites, such as Facebook or Instagram. The research shows that social media today are key tools for maintaining relationships with customers [1,2]. Furthermore, benefits of social media for a company are various, but some of the most relevant are: brand awareness growing, brand reputation increasing, customer relationship building, customer loyalty management, measuring the effectiveness of marketing and other company's activities, and so on.

In this paper, we have analyzed several Facebook pages of railway companies in Europe. The aim was to investigate whether social networking sites are important for railway companies or not? Do railway companies use these sites to communicate with their customers or not, and how much are they effective?

The paper is organized as follows. After the introduction, second section is devoted to defining social media role in public relations. The third section is dedicated to the research description. This section presents and discusses the main results of the conducted research. Finally, the last section is devoted to main conclusions and future research directions.

2. Social media and creative consumers

Comparing to earlier periods when companies were clearly separated from the organisational environment and were able to control how and when they will communicate with their environment, today public relations (PR) works quite differently. Due to ICT development modern companies are interconnected with their environment in numerous ways and they are increasingly losing possibility to maintain control over communicational flows. Especially, development of the internet and WEB 2.0 and 3.0 had significantly changed the practice of PR [3]. Social media plays the key role in this change, especially sites for social networking such as Facebook, Instagram and Twitter. After the rapid development of social media, companies had become aware that traditional forms of PR in the new social and technological environment may look

anachronistic. Today even the use of websites seems as an old-fashioned way for companies to present themselves and to communicate with public. They are now looking for new ways to develop and maintain the long-term relationships with customers rather than one-time coupon and sales transactions that take place through websites. [4] Unlike conventional media and websites whose use is controlled by companies, social networking sites have different logic. They could be a powerful tool for sharing knowledge about particular company among consumers. Through these sites consumer can exchange positive impressions and praises but also, very often, a criticism. As a result of this development, controlling the organizational environment in new technological settings is becoming real challenge for the companies. However, if and when a company manages to adapt to the new circumstances then it gets the opportunity to make consumer its best promoters. [5]

In that regard, Berthon et al. propose five axioms to help international marketing strategists make effective use of social media and constructively engage creative consumers [6]:

1. Social media is a function of the technology, culture, and government of a particular country.
2. In the age of social media, local events seldom remain local.
3. In the age of social media, general issues seldom remain general, that is, macro issues tend to be (re)interpreted locally.
4. The actions and creations of creative consumers tend to be a function of the technology, culture, and government of a particular country.
5. Technology tends to be historically dependent; that is, technologies in different countries evolve along unique trajectories due to inertia rather than they are the optimal solution.

3. Research results and discussion

Aim of our research was to make comparative analysis among different railway companies in selected European countries in order to define good and less good examples of the usage of Facebook as a tool for communication with customers. Our previous research had shown that different strategies of social media usage bring different results to the company and that is possible with relatively low budget to have very successful PR strategy [7].

For the purpose of this research we had analyzed Facebook pages of selected railway companies: ÖBB (Austria), SNCB (Belgium), DSB (Denmark), SNCF (France), DB (Germany), ŽSR (Slovakia),

SBB (Switzerland), SV (Serbia). All named companies are rail passenger transport companies.

In Table 1 we have presented some general data regarding railway, internet and Facebook use characteristics in selected countries.

Table 1 General data for the eight select countries

Country	Railway company	Population	Length of railway lines (km)	Number of passengers (million PKM)	Percent of residence using the internet	Percent of residence using the Facebook
Austria	OBB	8 401 940	5 491	12 077	88	53
Belgium	SNCB	11 000 638	3 578	10 498	90	73
Denmark	DSB	5 560 628	3 181	6 280	98	79
France	SNCF	64 933 400	28 364	96 690	89	42
Germany	DB	80 219 695	38 466	95 529	93	53
Slovakia	ŽSR	5 397 036	3 206	3 754	83	60
Switzerland	SBB	7 954 662	5 196	19 616	95	53
Serbia	Srbija voz	7 186 862	3 809	377	75	52

Data sources:

Population: <https://ec.europa.eu/eurostat/web/population-and-housing-census/census-data/2011-census>, <http://popis2011.stat.rs/>

Total length of railway lines: <https://ec.europa.eu/eurostat/data/database>

Rail transport of passengers: <http://www.stat.gov.rs/sr-latn/publikacije/publication/?p=11525>

Internet and Facebook penetration: <https://ec.europa.eu/eurostat/web/digital-economy-and-society/data/database>

As we can observe from Table 1 all EU countries have high internet penetration, i.e. above 83% of all citizens who regularly use the internet. Serbia is lagging behind but even there 3/4 of all citizens now use the internet. For all countries the Facebook penetration is ranging from 42 to 79 per cent. These differences in the Facebook usage probably depend on some cultural characteristics of specific country.

When it comes to data connected with railway we can see that biggest countries such as Germany and France have the biggest length of railway lines and higher numbers of transported passengers. However, based on other countries we can conclude that extent of

country and number of citizens is not the only indicators of railway infrastructure development and use. The other indicators are: GDP, investments in railway infrastructure, geographical position, history of industrialization of particular country, and so on.

The main results of our Facebook study can be observed in Table 2. As mentioned before, we conducted research on selected European railway companies' Facebook pages. The research was conducted with help of Netvizz v1.6 application. The observed data are related to the Facebook pages content that was produced during the October 2018.

Table 2 Data related to the Facebook pages of selected railway companies

Railway company	Facebook page launched	Page followers	The number of posts	The percent of users' posts	Reactions Total/per day	Comments Total/per day	Shares Total/per day	Engagement Total/per day
OBB (AT)	2010	173 000	238	74	3 443	12 344	1 543	17 330
					51.87	14.47	6.48	72.82
SNCB (BE)	2013	46 000	50	88	514	1 090	607	2 211
					21.80	10.28	12.14	44.22
DSB (DK)	2010	105 000	166	92	2 463	10 886	571	13 920
					65.58	14.84	3.44	83.86
SNCF (FR)	2012	261 000	145	89	1 690	3 127	610	5 427
					21.57	11.66	4.21	37.43
DB (DE)	2010	580 000	30	30	6 340	6457	975	13 772
					215.23	211.33	32.50	459.07
ZSR (SK)	2017	4 700	33	64	205	757	252	1 214
					22.94	6.21	7.64	36.79
SBB (CH)	2012	17 100	120	100	339	240	8	587
					2.00	2.83	0.07	4.89
SV (RS)	2015	1 600	14	0	46	292	29	367
					20.86	3.29	2.07	26.21

As we can see from Table 2 first companies' Facebook pages had been launched in 2010 (OBB, DSB and DB) following by pages launched in 2012 (SNCF, SBB) and 2013 (SNCF). The last two pages are launched in 2015 (Srbija Voz) and 2017 (ZSR).

When it comes to the followers the Facebook page of the German railway company (DB) has the largest followers number comparing to other railway companies. France Company (SNCF) is on the second place, followed by Austrian Railway Company (OBB). On the other hand, Serbian and Slovakian railways' pages have a smallest number of followers.

When it comes to page posts it is interesting to note that majority of posts are generated by followers instead of companies itself (from 30 to 100 per cent, excluding non-EU country Serbia). This is the best example of communicational role of sites for social networking. As we can see, the companies' Facebook pages are not exclusively reserved for dissemination messages from companies to the costumers but the quite contrary. This is the space for two way communication between companies and consumers which is often held or dictated from the consumers' point of view.

The correlations between number of followers and independent variables (Pearson and Spearman correlation tests, Table 3) have shown that there is strong interdependence between number of followers on one hand and the population size, length of railway lines, number of passengers and the Facebook page launch year, on the other hand. In another words, those railways companies who are from the biggest countries with the most developed railway transport can count on the largest number of follower, under the condition that they had launched their Facebook pages 8-10 years ago.

Table 3 Main Correlations

Pearson				
	Facebook launch year	Page followers	Total posts	Users' posts
Engagement	-0.805**	0.604	0.611	0.499
Reactions	-0.730*	0.924**	0.197	0.093
Comments	-0.758*	0.394	0.724*	0.627*
Shares	-0.650*	0.566	0.567	0.416
Spearman Sig. (1-tailed)				
Engagement	0.822**	0.786*	0.690*	0.690*
Reactions	-0.908**	0.929**	0.476	0.476
Comments	0.761**	0.738*	0.595	0.595
Shares	-0.602	0.833**	0.381	0.381

** . Correlation is significant at the 0.01 level

* . Correlation is significant at the 0.05 level

The most interesting results of our study are ones which reviled the rate of posts engagement i.e. the number of reaction (predominantly likes), comments and shares to company's page posts. We have used two measures to present this figures, total numbers and average numbers (per post) of reactions, comments and shares, i.e. engagement rate. Total numbers reviled that posts from OBB page had generated the highest engagement rate (17 330), followed by DSB page posts engagement rate (13 920) and DB page posts engagement rate (13 772). The lowest total engagement rate was generated by SV (367) and SBB (587). On the other hand, in terms of average numbers, post from DB generated highest engagement rate per post (459.07), followed by DSB (83.86) and OBB (72.82). The posts from SBB, with average of only 4.89, had been the least successful in attracting reactions from page the Facebook users. The correlation from the table 3 (only for total engagement rate) shows that engagement rate is dependent on the Facebook launching year, number of followers and posts number (total and average). That is reason why the Facebook

pages of OBB, DSB and DB are the most successful ones in the terms of engagement rate (total and average). Besides this correlation one can also note that posts from these companies, especially OBB and DSB, had generated the highest share of comments in the overall engagement rate. This is also important indicator because it shows that it is crucial to "provoke" the Facebook users to actively engage with post i.e. to comment instead of just clicking likes and shares.

4. Conclusion

The aim of the conducted research was to analyze whether the railway passengers are active on social media, i.e. Facebook, or not, and to highlight the advantages of Facebook that can be a good communication tool between a railway company and passengers. The results of this research can be useful for the management of railway companies around the world.

Our main conclusion is that active and toward consumers open communication through the Facebook company page is the most effective way of interacting with organisation's environment in the age of the internet. To be active means to publish as many posts as possible and to inspire the customers to do the same. The post should be of various sorts (statuses, photos, videos, links, etc...).

Our research has shown that for bigger countries are easier to generate the higher number of followers. This is important because the number of followers is one of the leading predictors of post high engagement rate. But OBB and DSB are good examples that even companies from smaller countries can be very effective and visible in cyber space if they use social media in proper way.

On the other hand, by "open" we have in mind regular and responsive communication which is led by desire to answer to costumers' questions and remarks, and not to present some imagined picture of company as in the old day of PR. To be open also means that company's Facebook page is equally space for the company and consumer which means that consumer are allowed to make posts and comments however displeased they can be. In this way, consumers have impression that company really care for their suggestions and complains. We believe that this is the best way to build mutual understanding and trust.

Future research is dedicated to the model development that can be a guideline for the railway company's management. Actually, the model will give the suggestions to railway company's marketers how they can improve the attractiveness and effectiveness of the company's Facebook page.

Acknowledgment

This paper is part of the "Critical infrastructure management for sustainable development in postal, communication and railway sector of Republic of Serbia" project, funded by the Ministry of Education and Science of the Republic of Serbia, Project number: TR36022.

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RECYCLED TIRE RUBBER MODIFIED BITUMENS FOR IMPROVING THE QUALITY OF THE ROAD CONSTRUCTIONS IN ALBANIA

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Abstract: Nowadays, only a small percentage of waste tires are being landfilled in Albania. The large environmental impact has made the government to support some companies for building up the recycling industry of the waste tire in Albania. The Recycled Tire Rubber have been used in many fields such as agricultural uses, sport applications, civil engineering, rubber modified asphalt applications etc. Therefore, different parts of the world have used rubber modified asphalts where the benefits were being more widely experienced and recognized. Based on it, our paper will be focused on the asphalt mixtures produced with Recycled Tire Rubber Modified Bitumen's (RTR-MBs). Our proposal can reduce environmental impact from the waste tires and improving the quality of the road constructions in Albania.

Keywords: RTR-MBs, ROAD CONSTRUCTIONS, ASPHALT APPLICATION, WASTE TIRES, ENVIRONMENTAL IMPACT.

1. Introduction

In recent decades, the worldwide growth of the automobile industry and the increasing use of vehicles as the main means of transport have tremendously boosted tire production [1]. This has generated massive stockpiles of used tires [1, 2]. Extensive research works were carried out on how to use used tires in different applications [2]. Furthermore, some of the fields are linked with agricultural uses, sport applications, civil engineering, rubber modified asphalt applications which have used Recycled Tire Rubber.

The increasing use of vehicles and traffic intensity on the roads has pushed Albanian government to speed up the quality policies on the road constructions and to reduce the environmental impact caused from the waste tires. The quality of the road constructions and the large environmental impact has made also the government to support some companies for improving the paving industry and building up the recycling industry through the waste tires. The use of rubber is of interest to the paving industry because of the additional elasticity imparted to the binder and enhanced safety related to improved roadway skid resistance [3].

Tire recycling is one of the most important processes in the industry [4-6]. The Fig. 1 depicts the technologies of the possible waste tire treatment routes.

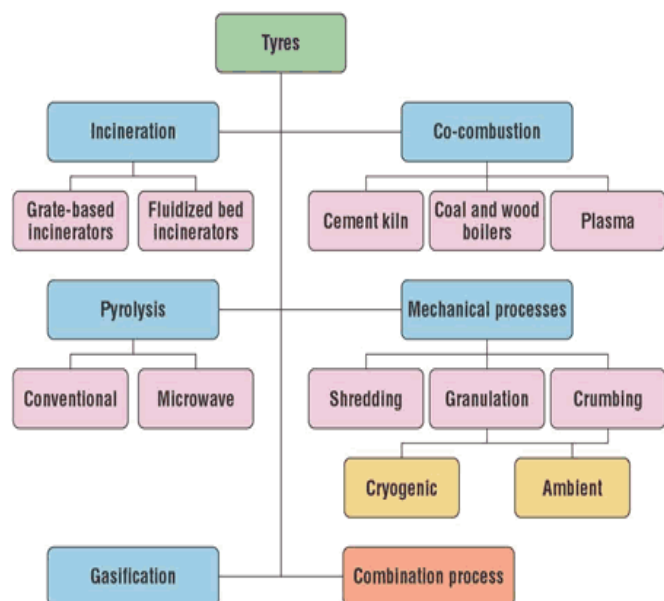


Fig. 1 Technologies for managing scrap tires [7]

Due to economic reason the recycling technology in ambient temperature has been used in this research work. This is also the widespread method used in most of the countries for recycling

technology. The importance of the usage of Recycled Tire Rubber Modified Bitumen's for improving the quality of properties of this material will briefly described in this paper.

2. Materials and Methods

Tire recycling is one of the most important processes in the industry [1-11]. Table 1 depicts the materials characterization of the waste tire that is used in recycling process.

Table 1. Materials characterization of the waste tire.

Ingredients	Percentage (%)
Elastomers	47
Carbons	21.5
Metals	16.5
Textile fibers	5.5
Zn-oxide	1
Sulfur	1
Additive	7.5

Through tire recycling process we can eliminate metal materials and fibrous with textile nature from inside the rubbers. Afterward, particle size of the rubbers would influence in the performance of the rubber modified asphalt. Non-regular sizes of the rubbers having a specific area of greater than regular size were more likely to react with bitumen at high temperature through wet process asphalt forming. In our research work, the wet process asphalt forming has been used to produce rubber modified asphalts, see Fig. 2.

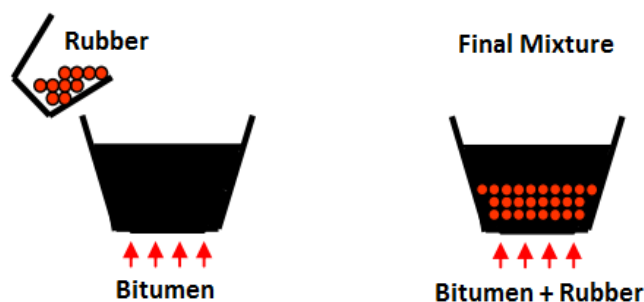


Fig. 2 Rubber modified asphalts.

The quantity of the Recycled Tire Rubber (RTR) in the mixture of the asphalt was 10%. Particles size of the rubbers varied from 0.85 mm to 2.36 mm and 1 mm to 4 mm. The modification of this material is carried out to reinforce the bitumen properties through the physical-chemical coupling between bitumen and rubber at high temperatures.

3. Summary Results

Addition of elastomeric rubber to the asphaltic bond, regardless of the method used for obtaining the final product, makes it possible to reinforce the properties of the asphalts which are as follows. The Fig. 3 depicts the comparisons of the crack levels during the years.

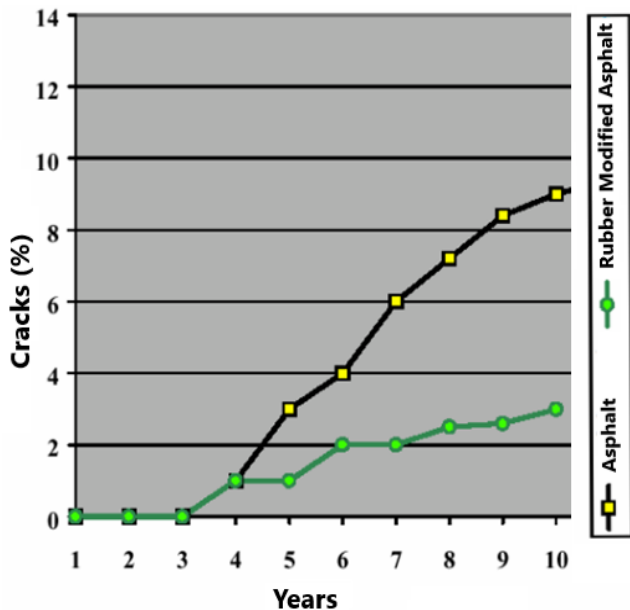


Fig. 3 Comparisons of the cracks level during the years.

The results have shown that the cracks level during the years were more lower in rubber modified asphalt than conventional asphalt. Another important analysis is focused on the comparisons of the penetration values between rubber modified asphalt and conventional asphalt, see Fig. 4.

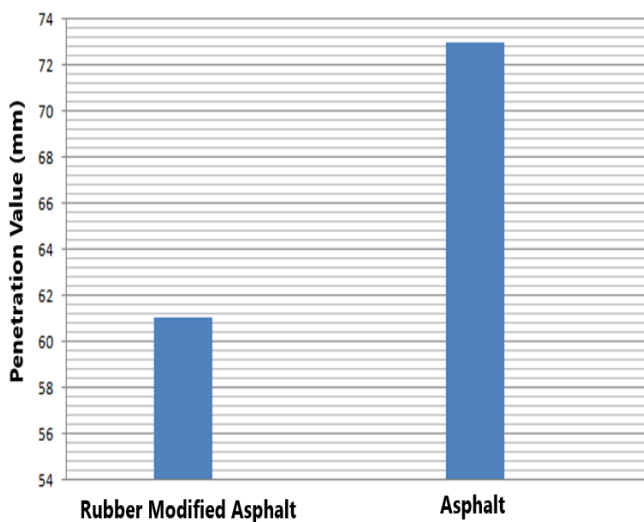


Fig. 4 Penetration value for rubber modified asphalt and conventional asphalt

The results have shown that penetration value is lower for rubber modified asphalt in comparisons of the conventional asphalt.

4. Conclusions

The importance of the usage of Recycled Tire Rubber Modified Bitumens for improving the quality of properties of this material has been briefly described. This proposal have shown that rubber modified asphalts will reduce environmental impact by using recycling process and improve the quality of the road constructions in Albania. Further studies should be done to investigate long term performance of field test sections under various traffic conditions.

5. Acknowledgment

The authors gratefully acknowledge ELBA ltd. Laboratory and the Polytechnic University of Tirana for partial funding of this research work.

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ROADS FOR IMPROVING THE QUALITY OF AVIATION EDUCATION

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Abstract: *The most important element for achieving a sustainable development of aviation is human resources and the provision of opportunities for acquiring the necessary knowledge, skills and competencies. In this sense, aeronautical training schools are becoming more and more crucial for increasing productivity and competitiveness, as well as helping people realize their full potential and achieve their personal goals.*

Keywords: QUALITY, ASSESSMENT, AVIATION EDUCATION, SELECTION AND TRAINING

1. Introduction

The analysis of global aviation and aviation market trends shows that excellent opportunities for their realization both in the country and on the international labor market are being created. There is an opportunity to turn Bulgarian aviation education into an export industry for highly qualified aviation specialists.

This requires the provision of quality education and training of aviation specialists, which will effectively respond to the needs of key government stakeholders, students, higher education institutions and employers directly related to aviation development.

But also the education and training of aviation specialists depends on their motivation. Students expect a significant benefit from this education and how it will help them perform their work better. Here is the role of higher education institutions and trainers to create favorable conditions for this motivation.

2. Concept on the quality of higher education

The quality of higher education is a criterion in the rating and accreditation of higher education institutions. The concept of quality, however, often remains blurry and leaking or "by default". Quality ideas can be grouped into several categories, the most important of which are [10]:

- quality as consistency of designation;
- quality as an outstanding achievement;
- quality as a faultless one;
- quality as a transformation;
- quality as a threshold or accented medium level;
- quality as a continuous improvement.

According to J. Juran, the author of the Annual Quality Improvement (AQI) concept, which develops the famous "spiral of quality," "quality is an outgrowth of results that has already been achieved, and is related to man's desire to achieve new records." [12]

According to the specialists, quality is a complex multidimensional set of properties and characteristics of higher education (university, specialty, training program) that give it the ability to satisfy determined or supposedly dynamically changing requirements for it. [5]

Quality is as good as the difference between dynamically changing requirements and the results obtained from the educational site is smaller and the faster it is overcome.

This understanding of quality is also taken into account by the National Evaluation and Accreditation Agency, which measures it as a compliance of the state of the site with the requirements set by it.

One of the main issues is "What requirements should a given educational establishment or aviation specialist meet?"

The answer is not simple and should start from the strategic goal (at least for the EU) "to achieve a high quality of life by increasing the potential of human resources in the context of a dynamic knowledge economy and sustainable socio-economic development" [1]; through the mission of higher education "to satisfy the personal and social educational, scientific, professional, qualification and cultural needs of every person throughout his life, in order to achieve his effective professional and personal realization in the society of knowledge" [2].

It is not possible to give a comprehensive answer to quality measurability if one of the most distinctive features of modern higher education is not taken into account - the diversification of higher education from which the different missions of higher education institutions and their professional fields derive.

Diversification - structural, programmatic, in form and purpose - is not an end in itself. It is a consequence of the diversified economic, social, scientific, technical and cultural needs - public, market and personal. In order to stimulate economic growth, it is necessary not only to increase the number of students but also to prepare them adequately for their needs. The "educational pyramid" of society is rising, and its degrees become more and more diverse.

When the European Commission's Program for Advanced Higher Education [10] states that by 2020 more than 35% of all jobs in the EU will need a high level of qualification, it is intended to include in this heading all types of higher education. In this context, it is pointless to question whether higher aviation education should provide a qualitative fundamental preparation that is orientated towards the next 10 years, or a qualitative preparation for today's public practice and the labor market. The question is solved by negating the alternative. **The answer is both one and the other.**

3. Assessment of the quality of aviation education

There is no unambiguous definition of quality education. Every world organization has its own definition, trying to unequivocally fix the most important characteristics of an education that is capable of guaranteeing the highest results. Irrespective of the differences, in the definitions two elements are present in them:

- The ability of the individual to understand and orient in the environment, which determines the cognitive development of learners as the main goal of all educational systems. Orientation is a major goal in all education systems; the success the systems achieve is an indicator of their quality.

- The role of education to promote and promote the values and attitudes of civil consciousness and to promote the creative and emotional development of the learner in a spirit of peaceful coexistence, security and civil responsibility; equality and continuity of cultural values throughout the generations.

Quality in aviation education can also be seen as the extent to which a set of properties and inherent characteristics (status and

results) of the object being evaluated (unit, structure, process, activity) corresponds to a dynamically changing goal, need or expectation mission of the higher school, academic standard, specialty or discipline goal, etc.). Quality is at the heart of aviation education. It influences what learners are learning, how well they learn, and what benefits they have from this education. Seeking a guarantee that trainers will achieve good learning outcomes and acquire knowledge and skills to help them play a positive role in society is a matter of political agenda in almost every country. Bulgarian military aviation education has already seen the correctness of this in the distant past. In all classes in which the training of the jankers in the Military School of His Majesty takes place, a consistent, purposeful and strictly sophisticated system of exams is set up and operates continuously throughout the duration of school education. [6]

The assessment may be considered as an established market value of the benefits or classification of something or someone in consideration of its value. It is the result of an act of judging or assessing a person, situation or event. Assessment is a process used to measure the performance or performance of a system or its elements. It is used as a all-inclusive term to denote any of the following activities: audit / review, performance appraisal, review of management systems, review or peer review (review by professionals, peer review), revision or monitoring.

Assessment is the systematic acquisition of information and feedback on the use, qualities and impact of an object, program or process in relation to expected outcomes [9]. It is seen as a process measuring the extent to which the expected learning outcomes have been achieved. In the context of education and training, assessment is a systematic method of gathering information on the impact and effectiveness of learning. Does the result show that the objectives set have been achieved and can give a sense of quality and help improve the learning process. Assessment is a process of characterizing and evaluating different aspects of educational activity, and these are:

- quality assurance - continuous attention to quality, keeping in mind and improving it. One of the tools in the area of quality care is its assessment.

- maintaining quality - such as managing it, keeping it in constant consistency with the goals and academic standards, and constantly updating them due to changing market and consumer demands.

- improving quality - a part of its management aimed at enhancing the ability to meet quality requirements.

Quality assessment defines any structured activity that determines the quality of the education and training process and the acquisition of knowledge and / or research, while self-assessment and / or evaluation by an external expert. In education, quality assessment involves determining (usually quantitatively) the achieved level of quality by adopted methods, mechanisms, criteria and indicators, as well as a review of the status of a unit or activity in order to establish the level of quality achieved. Every educator who provides an educational service is interested in doing so in the best possible way, to attract the most users, to provide the highest quality service. Of course, this can be understood after getting an evaluation. This assessment can be: self-assessment; assessment by users, students; assessment by colleagues in the team in which he works; assessment by the employer; assessment by the employer of his / her graduates; an external evaluation by an accreditation commission.

The criteria used to assess the activities of the higher aviation education institutions today represent a rich variety of forms in different countries and universities. Diversity depends on the different organizational and structural models of higher education systems as well as on the different funding models.

Here, it can be concluded that the quality of university activity is determined on the basis of long-term observation to take into

account the subsequent career of graduates or the impact of research on the development of society. Typically, indicators are developed for a more short-term perspective. When a complex activity is evaluated by several statistical metrics, a certain amount of information is lost and this can lead to a distorted picture of the actual state. There is a tendency for people to succumb to the influence of numbers and the importance they bear. Experience shows that, following the publication of activity assessment metrics, they are "inherently important". Using evaluation metrics is a tool for a proper judgment, not a substitute for the evaluation itself. Figures can not speak for themselves. Only data reporting is not enough; it is always necessary to interpret them.

4. Approaches for determining the quality of aviation education

In world university practice, there are different approaches to measuring and assessing the quality of higher education [9].

- *Output Control Approach* - This approach assesses the final outcome - the quality of the training of the trainees. This approach can be applied to both internal and external evaluation, for example in accreditation [5].

The Expert Committee, consisting of external to university and independent scientists and training professionals, conducts exams according to a pre-established regulation. On the scientific methods of qualification and electoral statistical control a sample of all graduating students in a given specialty (Training Program) is taken. The students' knowledge, skills, values, competencies and attitudes are examined, and their achievements (publications, developments, creative events) are introduced. In the case of a specialty, it receives accreditation when a certain proportion of the students under examination (eg 80%) meet the established standards for general, fundamental, general vocational, special and specialized training.

The disadvantage of the method is the strong dependence of the results not only (and often not so much) on the skills, creativity and input to the students' preparation but on their pre-training and personal qualities. Therefore, output control can not be indicative of the contribution of the university and academic staff to the quality of education.

Two examples:

- A large number of trainees have a good in-service training for computers. By controlling the outcome of their training in the IT / IT course, they receive high marks that are not a testimony to the new knowledge and the quality of the training course;

- Higher school of local significance, which has a weak competition at the entrance, will hardly give a good result at the exit. On the contrary, universities with attractive majors and old glory who have many and strong candidates will receive an undeservedly high score even at the time when they did not add much in their training during their training.

- *Value Added Approach* - A more equitable approach to evaluation is the so-called added value [8]. Methods of control at the level of the preliminary (incoming) preparation of the students are established. Input control is performed. After finishing the training, the result is controlled at the exit. The difference is the so-called. value added. The higher the score for the team responsible for the quality of the discipline or program is higher. Because it is not so important what the learner's entry level was, how much new knowledge and new applicable skills and competences are accumulated during the training. The disadvantage of this approach is the need for a reliable system of credible input and output preparation and determination of the difference. To this end, science-based methodologies for verifying the knowledge and skills of input and output should be developed. With a one-time exam or test, reliability can not be guaranteed, as is the case with outbound control.

- *Step by step* - Different aspects and processes of learning (academic composition, learning content, material basis, organization and course of the learning process, examination procedures, etc.) are evaluated, the overall value and importance of which determines the quality [8]. It monitors whether all the steps that pass and control the training are done properly. If they are done in the best possible way, the best results at the outputs, which do not necessarily have to be controlled, can be expected. This approach is conceptually close to total quality control. It is also applied in ISO quality control standards. His application at the higher school means that there is no need for special attention to what the entrance is and what the outcome of the given subject is, since everything "inside" is done in the best way, "creating" quality. The disadvantage of the approach is not always the clear connection of the quality being sought with the criteria on which the judgment is judged for its level and hence sometimes the erroneous assessment of the object. Many of the activities, processes, and phenomena in the university appear to be more positive (in terms of "paper") than they really are.

5. Factors on the basis of the quality of aviation education and training.

Obviously, the quality of education and training depends on many interrelated factors, some of which are priorities for the market competitive environment:

First. Quality policy

In Bulgaria, there is no established state policy on the quality of higher aviation education. It is true that the Higher Education Act requires each higher education institution to have an "internal system for assessing and maintaining the quality of teaching and the quality of academic staff" [11], and the control is entrusted to the National Evaluation and Accreditation Agency. However, this does not negate the need for state policy and related resource quality assurance.

Higher aviation education is one of the few areas of public life left behind by the 28 years ago, where the natural law on supply and demand has limited effect. The existing regulatory base keeps the development of the market environment, which presumably "gives birth" to quality. There is still a resource-oriented model of institutional funding for state higher education institutions, which provides a state subsidy for the maintenance of normative training for a learner. For the budget subsidy, it is not important what kind of product the university offers and whether it is sought and the number of students. As a consequence, requirements for admission, training, examinations, etc. are reduced.

In order to protect their state and budget, Higher Education Schools also abusively detain unprepared students. Competition is conducted in another field - "number of students at the entrance". Stimulating the "cylinder principle" (as they came to the university - so out of date) and not the "funnel principle" (no matter how many come in, the important thing is that only the well-trained ones get out of the way). Higher schools compete not on the basis of the quality they give at the exit but on the number of students they accept. It is the responsibility of the university to create conditions and prerequisites for good preparation, to motivate the student. But it is not his problem if he does not meet the high requirements. The mechanism of the funnel must work without the economic pressure to expand it.

The beginnings of state policy in this regard are already the rating of professional trends and differentiated subsidies, which are given to the best. For all, however, the question remains whether the assessment tool and methods of measurement are true whether it is done in scientific methods.

It is obvious that the state has something to give to the "unleashing" of market forces in the conditions of balanced regulation.

Second: Financing Higher Education.

Higher education across the world faces serious funding difficulties. They are the result of a combination of rising costs and static or even declining revenue in the sector. The reasons for the scarcity of financial resources can be sought in several directions: [4]

- increased participation of the population in higher education, expressed through the net enrollment rate of students in HEI.

- the tendency for average costs in higher education to grow faster than the average cost of the economy;

- limited revenue in the higher education sector. In most countries, the volume of higher education funding is even rising, this is usually at a rate lower than those with increasing costs. The slow increase in higher education funding is due to competition and the prioritization of other public sectors;

- aviation education and qualification of aviation specialists require the mobilization of a significant amount of financial resources. Responsibility for the study, development, evaluation and implementation of the various financial instruments and schemes to secure this resource from domestic and foreign financial institutions or from the private sector through alternative funding paths is the responsibility of the state.

Third: Role of the market

New theories of growth have a major impact on the development of public sector management. More and more methods and approaches for private sector management are beginning to find application in various public systems. These processes also affect the higher education sector. The reason is the nature of higher education as a good. Carefully examined, higher education has a number of characteristics typical of private, not public, goods. In the higher education, the characteristics of competitiveness (limited supply) and divisibility (supplied against a certain price) of goods are manifested.

In addition, the users of the higher education service are well informed, in many cases better than the suppliers themselves, which is a prerequisite for the effective operation of market forces. The perception of higher education as a service of receipt, which should be paid by the consumer, irreversibly alters the relations between the main actors in the system - state, educational institutions, learners, business - turning the latter two into a major driving force for long-term development of the sector.

Fourth: The motivation of the learner

A very important factor for the quality of higher education is the learner's motivation. It makes him carefully select his specialty because he thinks economically motivated for his future realization (which is not a fact now). And as soon as it is realized in its specialty, it strives to improve its preparation. It also puts pressure on its teachers for up-to-date knowledge and applicable skills. It is about the mass trainee, not the small percentage of truly internally motivated learners who learn and achieve high results and without external (financial) motivation. An even more immediate impact on the learner's motivation may be the way of self-financing of his / her education. Students pay graduate fees depending on their success during the previous school year. Prize-winners are taught free of charge, and the average trainees pay the full amount. This requires a strong and direct link between the quality of learning and the learner's success, i.e. objective assessments that adequately reflect not only the knowledge but also the applicable skills, values, competencies and relationships that are needed by the prospective specialist to be fit for employment in the knowledge based economy. In this way, quality management systems will work more efficiently. The model will be welcomed by the business, as its future human resources will be better motivated and quality prepared according to its needs. An effective link is created between

the results of higher education and the requirements of the labor market.

This will reduce the share of learners studying at the expense of the state "for diploma" and unwanted majors without the intention to apply them. Weak and unattractive higher schools and professional fields will remain with some trainees. In order to survive, they will unite and group together naturally, which will solve the problem of the ineffective higher education network (unaccountably administratively).

It should be noted that the implementation of incentives in our military education system, and in particular in the field of aeronautical specialists, has gained experience in the 40s of the last century. It uncovers a well-thought-out and flexible system for the preparation of technical staff for the needs of the Air Force. It is based on a consistent and precise selection among candidates with the necessary education. Placed in a suitable environment to prove their professionalism, at a later stage, they are given another opportunity to refine their knowledge and training and to grow up in the service hierarchy. In such a combination of professionalism and further stimulation and development of the acquired knowledge, habits and skills, a well-prepared theoretical and practical personnel base is obtained [7].

6. Conclusion

In a certain way, the quality of higher aviation education can be defined as the level and degree of usefulness of complex learning and research. Managing this utility implies bringing quality to a higher level and keeping it in line with labor market needs (above all national / high quality aviation professionals). All this calls for the establishment and maintenance of a comprehensive quality management system for activities in aviation education institutions [3].

The aim is systematic and planned increase of the quality of the training of the aviation specialists. In concrete terms, this means:

- studying the current and future trends in demand requirements, respectively the quality of supply of aviation specialists on the labor market in the country and the world;
- quality management of the elements of the educational cycle in the two main and parallel courses: first - admission, training and realization of the graduates and second - learning process, research process and additional complex service process / choice of scientific directions and specialties, organization of contemporary teaching, marketing the created educational product, etc .;
- maintaining two coordinates in quality management: systemic design, implementation and improvement of the mechanisms of functioning of the aviation education institutions and complexity in covering the factors of reproduction of the educational product - cadres and research;
- diversification of the educational product in the aviation education institutions in view of market requirements and elimination of non-conforming educational services and scientific products;
- precise selection of training contingents, teaching staff, service staff and building and maintenance of a modern educational and material base, which meets the modern requirements;
- scientific service of the processes for improving the quality of the activities related to the training of aviation specialists;
- a clearly defined vision of market behavior of aeronautical education institutions, incl. first, moving from a product-oriented to market-oriented management approach; secondly, business relations with the aviation business, public administration, users of educational products, presentation of the created goods, etc .;
- thirdly, strengthening the capacity of the educational institution to

implement strategic management; fourth, maintaining a quality enhancement and quality control system, etc .;

- harmonization of activities with those of the European Educational Area, which is expressed in: mobility / of students and teachers /, harmonization of educational content, mutual recognition of diplomas, etc .;
- urgently organizing continuing further training, in line with dynamic labor market developments;
- effective use of the available resources of the higher education institution.

Quality control is becoming a priority of aviation education as a factor of social and cultural engagement and economic potential. In modern Bulgaria, the guarantee of quality education of aviation specialists is a key factor for the development and efficient functioning of the aviation sector as a whole. For this reason, European quality education criteria must be applied to ensure the process of adequate assessment with the subsequent provision, maintenance and improvement of quality in higher aviation education.

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THERMODYNAMICAL ANALYSIS OF HEAT EXCHANGE AND FUEL CONSUMPTION IN MARINE RE-HEAT STEAM GENERATOR

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Abstract: The paper presents analysis of heat exchange and fuel consumption in the entire Marine Steam Generator (MSG) with steam re-heating and in all of its components. Analysis is performed by using operating parameters from the steam generator exploitation. The highest heat amount transferred from combustion gases is used in the evaporator (48.17 % of the cumulative heat amount transferred in MSG). Proportionally, evaporator uses the highest fuel mass flow of 0.5172 kg/s when compared to other MSG components. In the high-pressure pipeline heat losses amounts 82.64 kW. Cumulative heat transferred from combustion gases to water/steam in all MSG components amounts 42048.47 kW. Cumulative water/steam specific entropy and temperature increase in the entire MSG is 4.5677 kJ/kg·K and 454.18 K, while the fuel mass flow in the entire MSG is equal to 1.0736 kg/s.

KEYWORDS: MARINE STEAM GENERATOR, STEAM RE-HEATING, HEAT EXCHANGE, FUEL CONSUMPTION

1. Introduction

Marine propulsion systems are today mainly based on diesel engines, regardless of its type [1], [2]. Marine steam propulsion systems [3] still have a dominant role in the propulsion of LNG carriers. New propulsion systems for LNG carriers, which are at least partially based on steam turbines, are under development [4].

Marine steam propulsion systems have a high level of complexity due to a number of components from which they are assembled [5]. An essential element of such marine steam propulsion systems is steam generator. Older versions of marine steam generators produced superheated steam for all marine steam turbines [6] and for proper operation of other system components, but such steam generators do not possess ability of steam additional re-heating [7], [8]. Newer versions of marine steam generators possess that additional ability of steam re-heating.

In this paper is performed analysis of Marine Steam Generator (MSG) with additional steam re-heating. Analysis is based on the first law of thermodynamics. For each component of MSG with steam re-heating is calculated amount of heat transferred from combustion gases during the superheated steam production. Also, for each component of analyzed MSG is calculated fuel consumption. The analysis takes into account steam specific entropy and temperature increase on each analyzed MSG component along with heat and pressure losses of high-pressure steam pipeline.

2. Description of MSG with steam re-heating and its thermodynamic process

Main scheme of Marine Steam Generator (MSG) with steam re-heating is presented in Fig. 1. Water which enters in MSG (point 1, Fig. 1) is heated in all steam generator components and from that water is produced superheated steam (point 2, Fig. 1). Heating in all MSG components is ensured with fuel burning (with the inevitable presence of air) which is delivered in steam generator furnace.

After production, main superheated steam is led through a high-pressure pipeline into the high-pressure turbine. Due to steam high pressure and temperature, heat and pressure losses in high-pressure pipeline cannot be neglected. Therefore, at the high-pressure turbine inlet (point 3, Fig. 1), superheated steam pressure and temperature are lower in comparison with outlet from the MSG (point 2, Fig. 1).

After expansion in the high-pressure turbine, steam was led back to the MSG for the re-heating process. Due to high-pressure steam turbine extractions, steam mass flow in re-heater is lower in comparison with the main steam mass flow. Re-heating process increases steam temperature (between points 4 and 5, Fig. 1) and after re-heating steam was lead to medium-pressure steam turbine.

Each steam generator consists of three main components in the same housing. Those components are water heater, evaporator and superheater. MSG from this analysis in the housing also has fourth component - steam re-heater. To be able to present heat transfer and fuel consumption of each MSG component, operating points from

Fig. 1 are not sufficient, so between operating points 1 and 2 are added two additional operating points which divided MSG on its constituent components. That division is presented in Fig. 2.

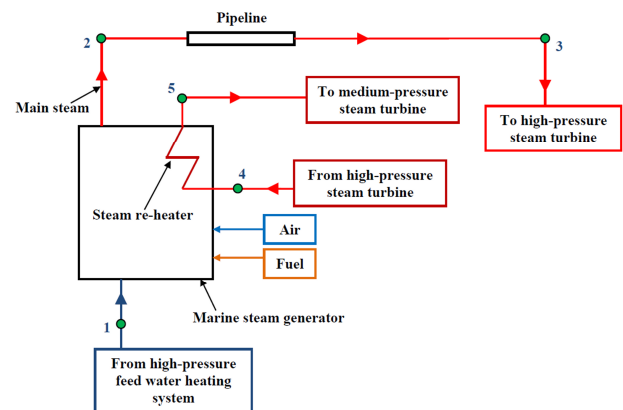


Fig. 1. Scheme of the MSG with marked operating points for the analysis

Division of MSG into the components required some assumptions. The first assumption is that the process of water heating, evaporation and superheating is happening at the same pressure (pressure losses at each mentioned component is neglected). The second assumption is that water heating, evaporation and superheating processes begins or ends at the saturation line (for the evaporator process begins and ends at the saturation line), which varies from the real process. Only pressure loss which has taken into account is in re-heater due to high steam volume and speed through re-heater.

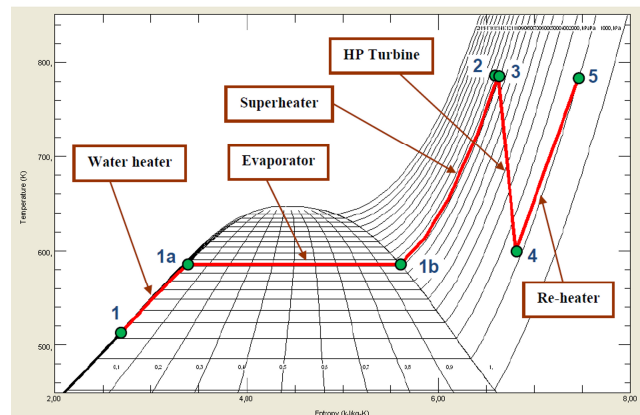


Fig. 2. T-s diagram of the MSG process with marked operating points necessary for the analysis of each component

Besides analyzed process at each MSG component, in Fig. 2 can also be seen temperature (and pressure) loss in high-pressure steam

pipeline (between points 2 and 3) as well as the real (polytropic) steam expansion process in a high-pressure steam turbine. The analysis in this paper ends with point 5, after steam exits the steam re-heater and was lead to medium-pressure steam turbine.

3. Equations for the thermodynamic analysis of MSG with steam re-heating

Analysis of MSG is performed according to the first law of thermodynamics [9] which is related to the conservation of energy [10]. Equations for the analysis are defined according to Fig. 1 and Fig. 2:

- Heat transferred from combustion gases to water in the MSG water heater:

$$\dot{Q}_{WH} = \dot{m}_1 \cdot (h_{1a} - h_1) \quad (1)$$

- MSG water heater fuel consumption:

$$\dot{m}_{f,WH} = \frac{\dot{Q}_{WH}}{H_{low} \cdot \eta_{WH}} = \frac{\dot{m}_1 \cdot (h_{1a} - h_1)}{H_{low} \cdot \eta_{WH}} \quad (2)$$

- Heat transferred from combustion gases to water/steam in MSG evaporator:

$$\dot{Q}_{EVAP} = \dot{m}_{1a} \cdot (h_{1b} - h_{1a}) \quad (3)$$

- MSG evaporator fuel consumption:

$$\dot{m}_{f,EVAP} = \frac{\dot{Q}_{EVAP}}{H_{low} \cdot \eta_{EVAP}} = \frac{\dot{m}_{1a} \cdot (h_{1b} - h_{1a})}{H_{low} \cdot \eta_{EVAP}} \quad (4)$$

- Heat transferred from combustion gases to steam in MSG superheater:

$$\dot{Q}_{SH} = \dot{m}_{1b} \cdot (h_2 - h_{1b}) \quad (5)$$

- MSG superheater fuel consumption:

$$\dot{m}_{f,SH} = \frac{\dot{Q}_{SH}}{H_{low} \cdot \eta_{SH}} = \frac{\dot{m}_{1b} \cdot (h_2 - h_{1b})}{H_{low} \cdot \eta_{SH}} \quad (6)$$

- Heat transferred from combustion gases to steam in MSG re-heater:

$$\dot{Q}_{RH} = \dot{m}_4 \cdot (h_5 - h_4) \quad (7)$$

- MSG re-heater fuel consumption:

$$\dot{m}_{f,RH} = \frac{\dot{Q}_{RH}}{H_{low} \cdot \eta_{RH}} = \frac{\dot{m}_4 \cdot (h_5 - h_4)}{H_{low} \cdot \eta_{RH}} \quad (8)$$

- Heat loss in MSG high-pressure pipeline:

$$\dot{Q}_{PL} = \dot{m}_2 \cdot (h_2 - h_3) \quad (9)$$

- Heat transferred from combustion gases to water/steam in the entire MSG:

$$\begin{aligned} \dot{Q}_{MSG} &= \dot{m}_1 \cdot (h_2 - h_1) + \dot{m}_4 \cdot (h_5 - h_4) = \\ &= \dot{Q}_{WH} + \dot{Q}_{EVAP} + \dot{Q}_{SH} + \dot{Q}_{RH} \end{aligned} \quad (10)$$

- Entire MSG fuel consumption:

$$\begin{aligned} \dot{m}_{f,MSG} &= \frac{\dot{Q}_{MSG}}{H_{low} \cdot \eta_{MSG}} = \frac{\dot{m}_1 \cdot (h_2 - h_1) + \dot{m}_4 \cdot (h_5 - h_4)}{H_{low} \cdot \eta_{MSG}} = \\ &= \frac{\dot{Q}_{WH} + \dot{Q}_{EVAP} + \dot{Q}_{SH} + \dot{Q}_{RH}}{H_{low} \cdot \eta_{MSG}} \end{aligned} \quad (11)$$

Data for the MSG thermodynamic analysis were found in [5]. In this document the authors specified that used fuel have a lower heating value (H_{low}) equal to 43038 kJ/kg and steam generator

efficiency (η_{MSG}) is 91 %. In this analysis were taken the same fuel lower heating value, while the efficiency of 91 % is assumed for the entire MSG and each of its components. Water/steam specific enthalpies (h) and specific entropies (s) were calculated with NIST REFPROP 9.0 software [11] from known temperature and pressure at each operating point of Fig. 1 and Fig. 2.

4. Operating parameters at each MSG point necessary for the analysis

Water/steam temperature, pressure and mass flow at each operating point of MSG (Fig. 1) were found in [5] and presented in Table 1. In Table 1 are also presented water/steam specific enthalpies and specific entropies calculated with NIST REFPROP 9.0 software [11] at each observed MSG operating point.

Table 1. Data for the MSG analysis at each operating point [5]

Op. point*	Temperature (K)	Pressure (kPa)	Mass flow (kg/s)	Specific enthalpy (kJ/kg)	Specific entropy (kJ/kg·K)
1	514.87	10300	15.593	1046.5	2.7028
2	786.00	10300	15.593	3404.6	6.6247
3	783.00	10100	15.593	3399.3	6.6263
4	599.95	2260	12.859	3079.2	6.8087
5	783.00	2030	12.859	3489.7	7.4545

* Operating point numeration refers to Fig. 1 and Fig. 2

Analysis of heat transfer and fuel consumption at each MSG component requires widening of data between operating points 1 and 2 (Fig. 1). Operating points 1 and 2 has the same pressure (Table 1), so at each main component of the analyzed MSG (re-heater not included) is assumed the same pressure of 10300 kPa. Through each MSG component the water/steam mass flow is equal (15.593 kg/s). Widening of operating data is performed according to Fig. 2 (new added operating points are 1a and 1b) to be able to analyze each MSG component individually. Operation parameters for all added operating points are presented in Table 2.

Table 2. Expanded data for the MSG analysis between operating points 1 and 2

Op. point*	Temperature (K)	Pressure (kPa)	Mass flow (kg/s)	Specific enthalpy (kJ/kg)	Specific entropy (kJ/kg·K)
1	514.87	10300	15.593	1046.5	2.7028
1a	586.33	10300	15.593	1420.9	3.3819
1b	586.33	10300	15.593	2719.9	5.5974
2	786.00	10300	15.593	3404.6	6.6247

* Operating point numeration refers to Fig. 2

5. The results of the MSG with steam re-heating thermodynamic analysis

Heat transferred from combustion gasses to operating medium (operating medium is water/steam) at each analyzed MSG component, along with heat losses in the high-pressure pipeline is presented in Fig. 3. In high-pressure pipeline occurs decrease of steam pressure (Table 1) and simultaneous decrease in steam temperature which resulted with heat losses equal to 82.64 kW. Heat losses also occur in all the other steam pipelines, but in high-pressure steam pipeline heat losses are the highest (due to the highest steam pressure and temperature), so it should not be neglected as in other pipelines.

Heat transfer at MSG components shows that the highest heat amount of 20255.31 kW is transferred from combustion gases to operating medium in evaporator during the change in water aggregate state (from water to saturated steam). Superheater is the second heat consumer which increases saturated steam temperature and for that temperature increase uses heat amount of 10676.53 kW. In the analyzed MSG, water heater uses a higher heat amount of

combustion gases than re-heater (5838.02 kW for water heater and 5278.62 kW for re-heater). The main reason for such occurrence is that water mass flow, which passes through water heater, is notably higher when compared with superheated steam mass flow, which passes through the re-heater, Table 1. Heat transferred in the entire analyzed MSG (water heater, evaporator, superheater and re-heater) amounts 42048.47 kW, Fig. 3.

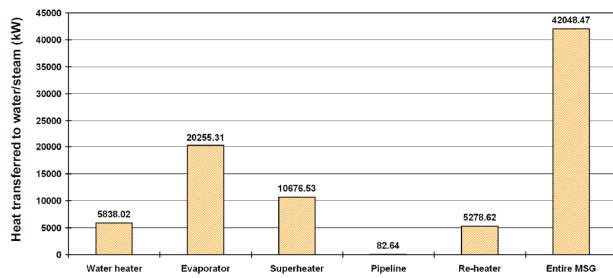


Fig. 3. Distribution of heat transferred from combustion gases in all analyzed MSG components along with heat loss in the high-pressure pipeline

From the entire heat amount which is transferred from combustion gases to water/steam in the analyzed MSG, 48.17 % is used by the evaporator, Fig. 4. After evaporator, 25.39 % of the cumulative transferred heat amount uses superheater, water heater uses 13.88 % and the lowest percentage of the cumulative transferred heat amount uses re-heater (12.55 %).

Analyzed MSG acts as many other marine steam generators and steam generators from the land-based steam power plants because the largest heat amount is used for changing the water aggregate state. Change of water aggregate state, according to early adopted assumptions did not result in water/steam temperature increase in evaporator (water/steam temperature in the evaporator can only decrease due to losses).

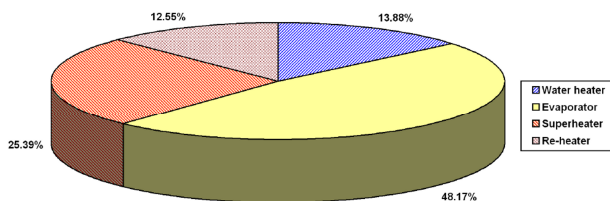


Fig. 4. Percentage distribution of transferred heat in all MSG components

Increase of water/steam specific entropy in the analyzed MSG is the highest in the evaporator after which follows the superheater, two components which uses the highest delivered heat amount, Fig. 4 and Fig. 5. Increase in specific entropy of operating medium is higher for water heater than for re-heater, Fig. 5. From this observation can be concluded that heat transferring process in the components of analyzed MSG is directly proportional to operating medium specific entropy increment - increase in delivered heat amount from combustion gases is followed by a proportional increase in specific entropy.

The increase in operating medium temperature of analyzed MSG is not proportional to heat transfer process, Fig. 4 and Fig. 5. Evaporator which gets the highest heat amount of burned fuel does not increase water/steam temperature because of the aggregate state change. The highest operating medium temperature increase can be seen in superheater and re-heater, which both increase temperature of superheated steam (superheater for 199.67 K and re-heater for 183.05 K). MSG water heater increases water temperature for 71.46 K, Fig. 5.

The sum of operating medium specific entropy and temperature increment at each analyzed MSG component resulted with a conclusion that analyzed MSG increases the specific entropy of operating medium for a cumulative value of 4.5677 kJ/kg-K, while

the cumulative increase in operating medium temperature in this steam generator is equal to 454.18 K.

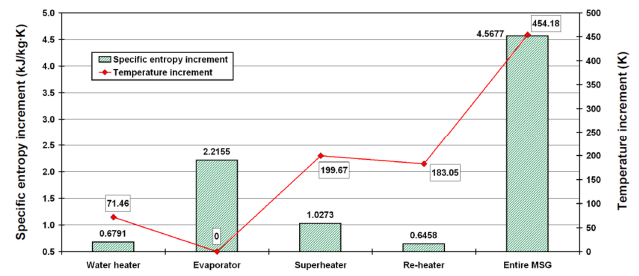


Fig. 5. Increase in water/steam specific entropy and temperature in each component and in the entire MSG

Fuel consumption for each MSG component is also directly proportional to heat transferred on each component, Fig. 6. The highest fuel consumer (and proportionally the highest heat consumer) is evaporator which uses fuel mass flow of 0.5172 kg/s. After the evaporator follows superheater and water heater which uses fuel mass flow of 0.2726 kg/s and 0.1491 kg/s. The lowest fuel mass flow (and proportionally the lowest heat amount) of all analyzed MSG components can be seen at re-heater which uses fuel mass flow of 0.1348 kg/s. All the MSG components use the same fuel, which lower heating value amounts 43038 kJ/kg.

Cumulative fuel mass flow used in the entire analyzed MSG is the sum of fuel mass flows used in each component. Fuel mass flow in the entire MSG is 1.0736 kg/s, Fig. 6.

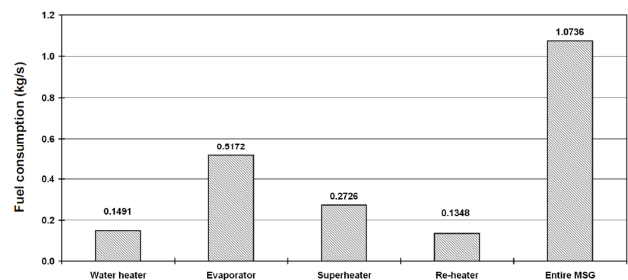


Fig. 6. Fuel consumption for each component and in the entire MSG

NIST REFPROP 9.0 software [11] allows calculation of many water/steam operating parameters (besides specific enthalpies and specific entropies) at each observed operating point of the analyzed MSG. Some of important water/steam operating parameters necessary for detail calculation of MSG operation is presented in Fig. 7. Those operating parameters are presented in each operating point of MSG according to Fig. 2.

Water/steam specific volume, which is directly proportional to operating medium volume, is one of the indicators for losses at each MSG component (especially pressure losses). In MSG water heater (points from 1 to 1a) increase in water specific volume is low, so pressure losses in that component can be neglected. In evaporator (points from 1a to 1b) and in superheater (points from 1b to 2) increase in operating medium specific volume is significant and in that MSG components can be expected a notable operating medium pressure decrease during MSG operation, which are neglected in this analysis (according to used operating parameters from [5]). So, in the real exploitation of analyzed MSG will be important precise measurements of steam pressure and temperature at the evaporator and superheater inlets and outlets, if possible. Temperature and pressure decrease in high-pressure pipeline (points from 2 to 3) resulted with an increase in superheated steam specific volume from 0.03250 m³/kg to 0.03302 m³/kg. The most significant increase in superheated steam specific volume can be seen during the steam expansion in the high-pressure turbine (points from 3 to 4) and in steam re-heater (points from 4 to 5). Such significant increase in superheated steam specific volume through the MSG re-heater resulted in steam pressure decrease between re-heater inlet

(operating point 4) and outlet (operating point 5), which can be seen in Table 1.

Compressibility factor denotes the difference between the real and ideal gas. Compressibility factor value equal to 1 denotes an ideal gas, while its value lower than or greater than 1 represents the real gas. Compressibility factor is usually used to observe how much the current operating medium characteristics deviate from the ideal gas. For the analyzed MSG in the water heater (points from 1 to 1a) operating medium characteristics significantly deviates from ideal gas because in water heater operating medium is water, so compressibility factor is 0.05299 at the water heater inlet and 0.05569 at the water heater outlet. Saturated steam at the evaporator outlet (operating point 1b) has compressibility factor 0.66152 which means that saturated steam characteristics getting closer to an ideal gas (in comparison with water in water heater). At the superheater outlet (operating point 2) superheated steam has a compressibility factor equal to 0.92266, what indicates that superheated steam has a characteristics very close to ideal gas. After the expansion in high-pressure steam turbine (at the re-heater inlet-operating point 4) superheated steam has a compressibility factor of 0.95199, while at the re-heater outlet (operating point 5) superheated steam has a compressibility factor of 0.98525.

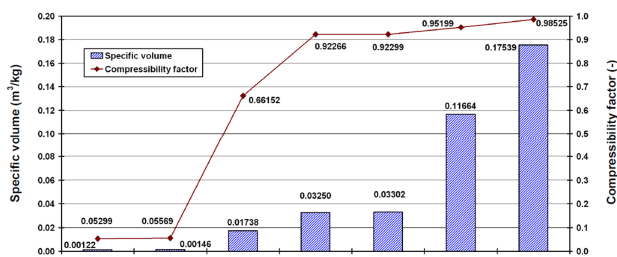


Fig. 7. Change in water/steam specific volume and compressibility factor at each analyzed MSG operating point

6. Conclusions

This paper presents analysis of heat exchange and fuel consumption in Marine Steam Generator (MSG) with steam re-heating and in all of its constituent components. The presented analysis provides insight into the operation of each MSG component and presented the most important temperature and pressure losses during the superheated steam production. The most important conclusions of this analysis are:

- The highest transferred heat amount of the entire analyzed MSG is used in the evaporator and amounts 20255.31 kW (48.17 % of the cumulative heat amount transferred in MSG).
- The lowest transferred heat amount of the entire analyzed MSG is used in steam re-heater and amounts 5278.62 kW (12.55 % of the cumulative heat amount transferred in MSG) due to lower operating medium mass flow through re-heater when compared to other MSG components.
- Due to the highest transferred heat amount, evaporator uses the highest fuel mass flow, which amounts 0.5172 kg/s and causes the highest specific entropy increase of operating medium (2.2155 kJ/kg·K), when compared to other MSG components.
- The highest operating medium temperature increase occurs in superheater and re-heater (superheater for 199.67 K and re-heater for 183.05 K), while the evaporator uses the highest heat amount but did not increase operating medium temperature (in evaporator operating medium change its aggregate state).
- Superheated steam, after it leaves MSG superheater, has characteristics very similar to ideal gases.
- Cumulative heat transferred from combustion gases to water/steam in all MSG components (in the entire MSG) amounts 42048.47 kW. Cumulative water/steam specific entropy and temperature increase in the entire analyzed MSG is 4.5677 kJ/kg·K and 454.18 K, while the fuel mass flow in the entire MSG is 1.0736 kg/s.

7. Acknowledgments

This work has been fully supported by the Croatian Science Foundation under the project IP-2018-01-3739.

8. Nomenclature

Abbreviations:		Greek symbols:	
MSG	Marine Steam Generator	η	efficiency, %
Latin symbols:		Subscripts:	
h	specific enthalpy, kJ/kg	EVAP	Evaporator
H_{low}	fuel lower heating value, kJ/kg	f	fuel
$\dot{m}$	mass flow, kg/s	PL	Pipeline Loss
p	pressure, kPa	RH	Re-Heater
$\dot{Q}$	heat transfer, kW	SH	Super Heater
s	specific entropy, kJ/kg·K	WH	Water Heater
T	temperature, K		

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