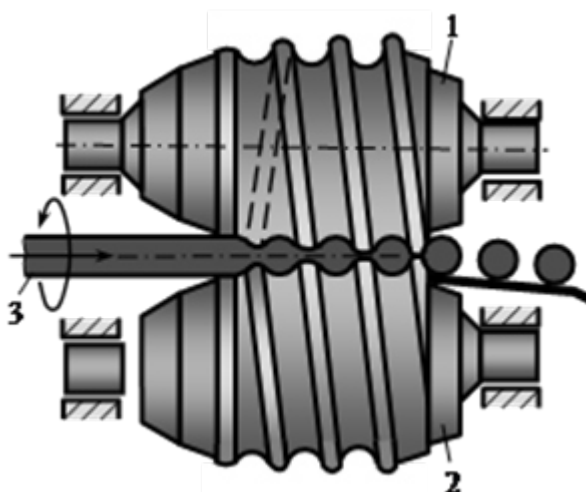


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NEW STRUCTURE OF THE GANTRY CRANES LEVEL LUFFING JIB SYSTEM

НОВА КОНСТРУКЦИЯ НА СИСТЕМАТА ЗА ИЗМЕНЕНИЕ НА ОБСЕГА НА ПОРТАЛНИТЕ КРАНОВЕ

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Abstract: A new construction of the gantry cranes level luffing jib system has been proposed. Its important operating characteristics has been studied. An optimization mathematical model of these structure is built. The parameters, optimization criteria and their limitations are defined. The optimization of construction is done using the Pareto optimization procedure in MATLAB. All parts and assembled units of the gantry cranes level luffing jib system have undergone optimization in practice. The results of the optimization study of the new design are compared with those of the traditional structure.

KEYWORDS: NEW STRUCTURE, GANTRY CRANES, LEVEL-LUFFING JIB SYSTEM, OPTIMIZATION MATHEMATICAL MODEL, PARETO OPTIMIZATION PROCEDURE, MATLAB.

1. Introduction

Gantry cranes provide a substantial part of the cargo flow of the economy in each country. The moving assemblies belonging to the level-luffing jib system, which determine to a large extent the performance of these cranes, have the largest mobile mass in crane structure. This is sufficient reason to have a effort for improvement and therefore a new design of this system has been created.

The aim of the report is to present the operation parameters re-search of the new structure of gantry cranes level-luffing jib system. It is made using a new complex of Pareto multi-criteria optimization programs.

2. Preconditions and means for resolving the problem

We have applied a system approach to conducting the research. The principles of the system approach are limited to the following:

1. Purposefulness - Have to define the purpose of the work and to ensure efficiency in the operation of the machine;
2. Relativity - Each element should be considered **as part of the whole** in the studies and calculations;
3. Modeling - The possibility to build a model of the machine under study.

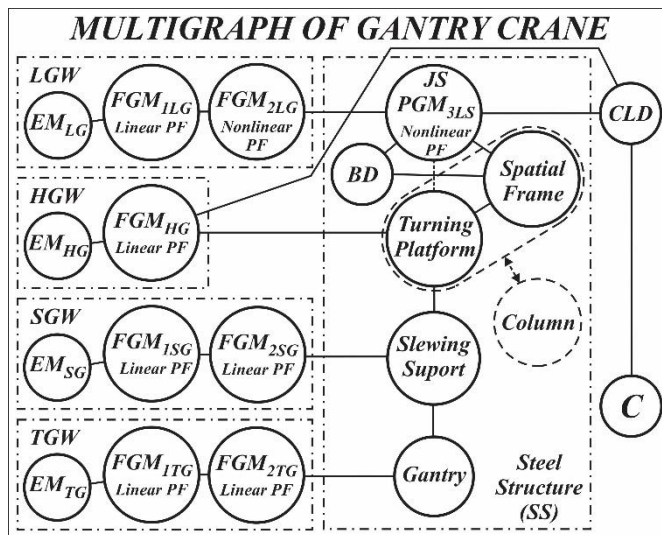


Fig. 1. Multigraph of gantry crane

Structural interaction of all systems and their elements of the gantry crane is indicated in Fig. 1. The structure of the gantry crane is shown in Fig. 2. It is composed of four main systems with their drive winches. The systems and devices favoring the motions at the

respective degrees of mobility of the gantry crane are also presented. The gantry crane has four degrees of mobility. They are determine by the four drive winches operation. The gantry crane luffing gear winches in two variants are shown in Fig. 3. The first is of crane ZUBR in Fig. 3 a) and the second is of crane KIROVEC in Fig. 3 b). The symbols and denominations of the crane systems and elements in Fig. 1, Fig. 2 and Fig. 3. are written and clarified in Table 1.

Table 1. Symbols and denominations in Fig. 1, Fig. 2 and Fig. 3

№	SYMBOL	DENOMINATION
1.	$EM_{LG}, EM_{HG}, EM_{SG}, EM_{TG}$	Power aggregates of crane gear winches (electric motor, clutch, brake)
2.	$FGM_{1LG}, FGM_{HG}, FGM_{1SG}, FGM_{2SG}, FGM_{1TG}, FGM_{2TG}$	Function gear mechanisms with linear position function (reduction units, open spur gearing) in the respective crane gear winches
3.	$FGM_{2LG}, JS (FPM_{3LG}), BD$	Function gear mechanisms or function position mechanisms with nonlinear position function (rack and pinion, two planar four-bar linkages) in the respective crane gear winches or systems
4.	LGW, HGW, SGW, TGW	Crane gear winches accordingly for operating radius changing, hoisting of cargo, turning platform, crane traveling and steel structure (foundation) above which, they are mounted
5.	LJS, HS, CSS, TS	Crane systems accordingly for operating radius changing, hoisting of cargo, turning platform, crane traveling and gantry crane steel structure adjacent part, which is driven or manipulated by crane gear winches
6.	SS, CLD, C	Steel structure of the gantry crane, Cargo-lifting device, Cargo

The level-luffing jib system shown in Fig. 4 and Fig. 5 consists of the following subsystems and their elements:

- A) Jib system (JS) comprising: jib - 4, jib arm - 3 and guy - 5;
- B) Balancing device (BD) consisting of: tie-bar - 7, rocker arm - 9 and counterweight - 10;
- C) Driving mechanism of system (LGM) with: winch - 11 and rack - 8.

The hoisting system is shown in Fig. 4 and Fig. 5 by cargo - 1, wire ropes - 2 and hoisting gear winch - 6.

The lower axle support point O_2 of the guy is stationary assembled to the level-luffing jib system base, which can be a spatial farm, frame or column, traditionally. Such a traditional construction is thoroughly studied and researched in [1] and is shown therein in Fig. 6 and in Fig. 2 of present report. The new changing in the proposed and studied construction of the level-luffing jib system is that point O_2 on Fig. 1 now is moved and attached to the rocker arm - 9 of the counterweight - 10.

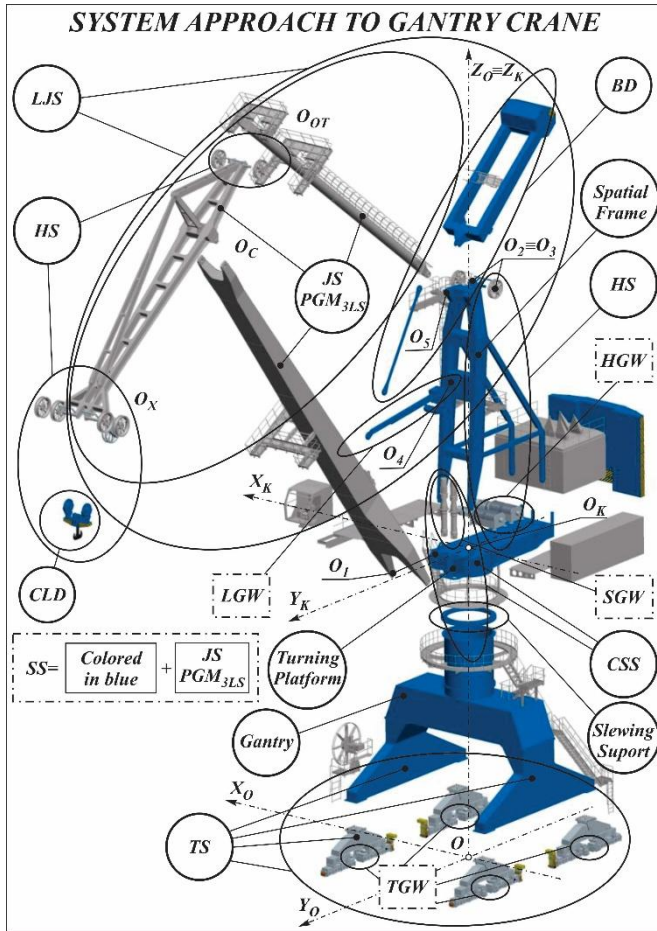


Fig. 2. System approach to gantry crane
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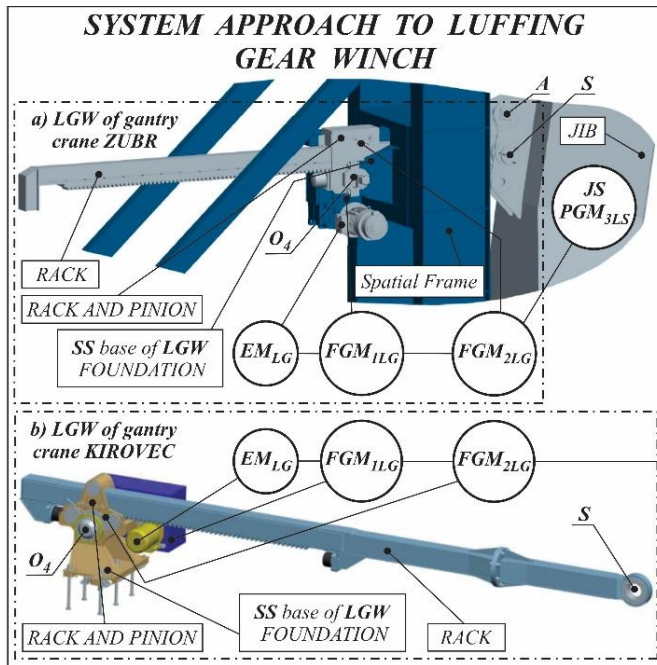


Fig. 3. Luffing gear winches in two variants:
a) of crane ZUBR and b) of crane KIROVEC.
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Let us recall the structure of a level-luffing jib system with a profiled jib arm. This old construction have purpose to achieve an

unloading of the jib from bending practically, and reach straight cargo trajectory, by changing the lengths of the rear arm of the jib arm – l_{2X} and the guy – l_{OT} .

The change in the new construction is achieved by moving of point O_2 onto the rocker arm - 8 of the counterweight - 10. This time are changed the length of the guy - l_{OT} again, and also the polar radius from point O_1 to point O_2 - d_{OT} and its corresponding polar angle - η_{OT} . The two four-bar linkages JS and BD from the traditional construction of the level-luffing jib system, shown on Fig. 6 in [1] and researched there, which is now widespread, are transformed into the six-bar linkage structure of the new design of level-luffing jib system, shown in Fig. 4, Fig. 5 and Fig. 6 in present report in this way.

We aim to achieve a positive effect in performance characteristics by this change of structure.

3. Solution to the problem

An new optimization model of the gantry cranes level-luffing jib systems based on the universal methodology [3, 8, 11] is built. It corresponds to the Pareto optimization procedure explained in Fig. 4, Fig. 5 and Fig. 6 in [1], but the new things of structure are shown in Fig. 4, Fig. 5 and Fig. 6 in present report. These new optimization model of the gantry cranes level-luffing jib systems was created in MATLAB [4, 6]. Experimental studies of the new gantry cranes level-luffing jib systems were done using this model.

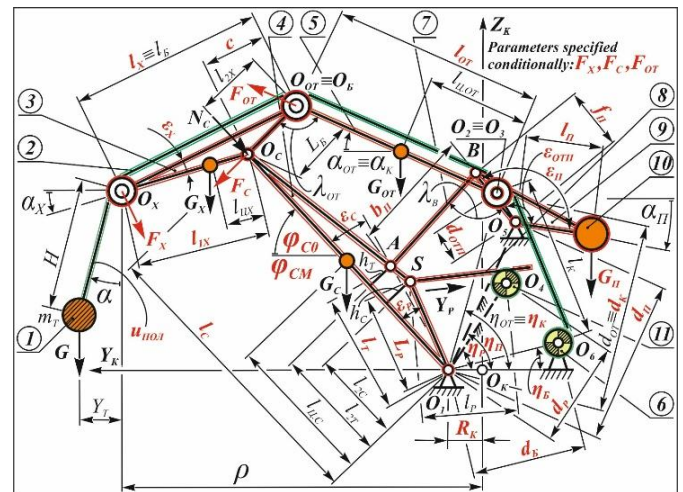


**Fig. 4. Optimization mathematical model schema when
the hoisting wire ropes pass through the guy**

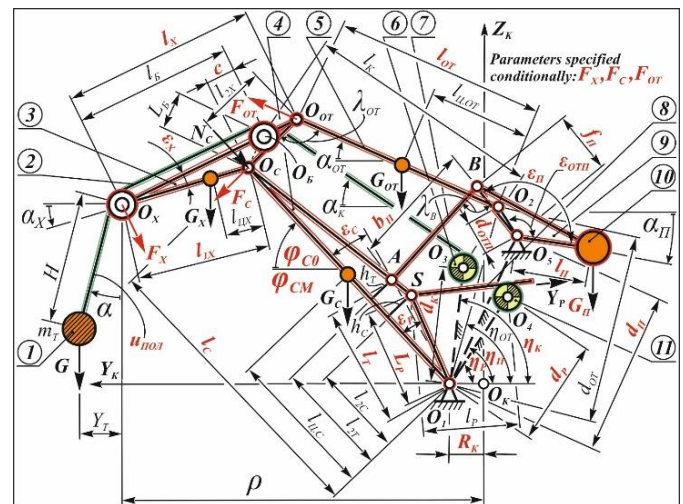


**Fig. 5. Optimization mathematical model schema when the
hoisting ropes pass arbitrary between the jib and the guy**

The Pareto optimization procedure uses the interaction of the 30 parameters characterizing units and parts of the gantry cranes level-luffing jib system shown in red in Fig. 4 and Fig. 5, which are involved in the formation of 20 criteria defining the quality of its work [5, 7, 10]. Parameters are graphically depicted in Fig. 4 and Fig. 5, and their names are shown below in the next paragraph text. The adequacy of the optimization model has been verified by setting a minimum range of parameters variations adjusted to a specific construction, where the result is similar to previous studies of this type of level-luffing jib system [2].

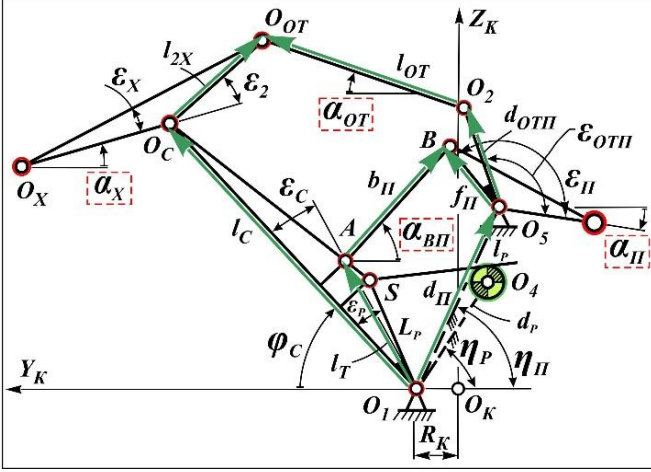


Fig. 6. Schema of the closed vector loop method

The kinematic study in the optimization procedure is carried out by the method of closed vector loop [9] and its adequacy is checked by a graphical method. The first step of kinematic research is to solve the problem of the position of linkages of mechanism. So that, we compile the output system of the equations for analysis, which connects the unchangeable and the variable parameters of the mechanism. This system is easiest and most convenient to compose by the method of projection the closed vector contours on coordinate axes.

The symbols in next four formulas correspond with Fig. 4, Fig. 5 and Fig. 6 and their denomination are shown in next paragraph.

We introduce vectors associated with the linkages of mechanism, organized into two closed vector loops as shown in Fig. 6. We compose the following two vector equations for them:

$$(1) \begin{cases} \vec{O_1O_C} + \vec{O_CO_{OT}} = \vec{O_1O_5} + \vec{O_5O_2} + \vec{O_2O_{OT}}; \\ \vec{O_1A} + \vec{AB} = \vec{O_1O_5} + \vec{O_5B}. \end{cases}$$

We project the vector equations of this system on the coordinate axes and we get the following system (2) of four non-linear equations for the unknown four angles: α_X , α_{OT} , α_{II} and α_{BII} . The angles are shown in Fig.6.

$$(2) \begin{cases} l_C \cos \varphi_C - l_{2X} \cos (\alpha_X + \varepsilon_2) = -d_{II} \cos \eta_{II} + \\ + d_{OTII} \cos (\pi - \varepsilon_{OTII} + \alpha_{II}) + l_{OT} \cos \alpha_{OT}; \\ l_C \sin \varphi_C + l_{2X} \sin (\alpha_X + \varepsilon_2) = d_{II} \sin \eta_{II} + \\ + d_{OTII} \sin (\pi - \varepsilon_{OTII} + \alpha_{II}) + l_{OT} \sin \alpha_{OT}; \\ l_T \cos (\varphi_C + \varepsilon_C) - b_{II} \cos \alpha_{BII} = \\ = -d_{II} \cos \eta_{II} + f_{II} \cos (\pi - \varepsilon_{II} + \alpha_{II}); \\ l_T \sin (\varphi_C + \varepsilon_C) + b_{II} \sin \alpha_{BII} = \\ = d_{II} \sin \eta_{II} + f_{II} \sin (\pi - \varepsilon_{II} + \alpha_{II}). \end{cases}$$

The angle φ_C we treat as known and it is the generalized coordinate of this mechanism. It determines a degree of mobility of a six-linkages mechanism.

As a result of solving this system (2) we get the values of the four angles: α_X , α_{OT} , α_{II} and α_{BII} , so that the position of the linkages is already known.

The second stage of the mechanism analysis is the determination of first order kinematic transmission functions. To do this, we differentiate the system (2) on the time. We obtain the following system (3) of four linear equations:

$$(3) \begin{cases} -l_C \sin \varphi_C + [l_{2X} \sin (\alpha_X + \varepsilon_2)] U_X = \\ = -[d_{OTII} \sin (\pi - \varepsilon_{OTII} + \alpha_{II})] U_{II} - [l_{OT} \sin \alpha_{OT}] U_{OT}; \\ l_C \cos \varphi_C + [l_{2X} \cos (\alpha_X + \varepsilon_2)] U_X = \\ = [d_{OTII} \cos (\pi - \varepsilon_{OTII} + \alpha_{II})] U_{II} + [l_{OT} \cos \alpha_{OT}] U_{OT}; \\ -l_T \sin (\varphi_C + \varepsilon_C) + [b_{II} \sin \alpha_{BII}] U_{BII} = \\ = -[f_{II} \sin (\pi - \varepsilon_{II} + \alpha_{II})] U_{II}; \\ l_T \cos (\varphi_C + \varepsilon_C) + [b_{II} \cos \alpha_{BII}] U_{BII} = \\ = [f_{II} \cos (\pi - \varepsilon_{II} + \alpha_{II})] U_{II}. \end{cases}$$

In this system, kinematic transmission functions U_X , U_{OT} , U_{II} and U_{BII} of first order are unknown. After solving the system we get the values of these kinematic transmission functions.

The third stage of the mechanism's analysis is the determination of the second-order kinematic transmission functions. To do this, we differentiate the system (3) by the time. We get the following system again from four linear equations:

$$(4) \begin{cases} -l_C \cos \varphi_C + [l_{2X} \cos (\alpha_X + \varepsilon_2)] U_X^2 + \\ + [l_{2X} \sin (\alpha_X + \varepsilon_2)] dU_X/d\varphi_C = \\ = -[d_{OTII} \cos (\pi - \varepsilon_{OTII} + \alpha_{II})] U_{II}^2 - [l_{OT} \cos \alpha_{OT}] U_{OT}^2 + \\ - [d_{OTII} \sin (\pi - \varepsilon_{OTII} + \alpha_{II})] dU_{II}/d\varphi_C + \\ - [l_{OT} \sin \alpha_{OT}] dU_{OT}/d\varphi_C; \\ -l_C \sin \varphi_C - [l_{2X} \sin (\alpha_X + \varepsilon_2)] U_X^2 + \\ + [l_{2X} \cos (\alpha_X + \varepsilon_2)] dU_X/d\varphi_C = \\ = -[d_{OTII} \sin (\pi - \varepsilon_{OTII} + \alpha_{II})] U_{II}^2 - [l_{OT} \sin \alpha_{OT}] U_{OT}^2 + \\ + [d_{OTII} \cos (\pi - \varepsilon_{OTII} + \alpha_{II})] dU_{II}/d\varphi_C + \\ + [l_{OT} \cos \alpha_{OT}] dU_{OT}/d\varphi_C; \\ -l_T \cos (\varphi_C + \varepsilon_C) + [b_{II} \cos \alpha_{BII}] U_{BII}^2 + \\ + [b_{II} \sin \alpha_{BII}] dU_{BII}/d\varphi_C = \\ = -[f_{II} \cos (\pi - \varepsilon_{II} + \alpha_{II})] U_{II}^2 + \\ - [f_{II} \sin (\pi - \varepsilon_{II} + \alpha_{II})] dU_{II}/d\varphi_C; \\ -l_T \sin (\varphi_C + \varepsilon_C) - [b_{II} \sin \alpha_{BII}] U_{BII}^2 + \\ + [b_{II} \cos \alpha_{BII}] dU_{BII}/d\varphi_C = \\ = -[f_{II} \sin (\pi - \varepsilon_{II} + \alpha_{II})] U_{II}^2 + \\ + [f_{II} \cos (\pi - \varepsilon_{II} + \alpha_{II})] dU_{II}/d\varphi_C. \end{cases}$$

In this system, second-order kinematic transmission functions $dU_X/d\varphi_C$, $dU_{OT}/d\varphi_C$, $dU_{II}/d\varphi_C$ and $dU_{BII}/d\varphi_C$ are unknown. As a result of solving system (4) we get their values.

Now, after determining the first and second order kinematic transmission functions, we can easily determine the angular velocities and angular accelerations of the all linkages of the mechanism, as well as the speeds and accelerations of points from these linkages.

4. Results and discussion

Two program complexes have been compiled for the purposes of the study. They correspond to the two design schemas studied, which are shown in Fig. 4 and Fig. 5. Results are obtained for both shown structures.

A third type of level luffing jib system structure exists when hoisting wire ropes pass through the boom, but it is constructively inconvenient and not addressed in this report. The results shown below refer only to Fig. 4.

In the optimization procedures, is taking into account that the force in the guy acts on the counterweight rocker arm, and when the hoisting ropes pass along the guy, hoisting wire ropes act on the counterweight rocker arm too in point O_2 .

The experimental optimization study was carried out with a cargo of 160 kN, and results are pictured on the graphs in Fig. 7, Fig. 8 and Fig. 9. Specified conditional forces are needed to close the circle of "force - inter sectional dimensions - mass" needed for the study. The forces are shown in Fig. 4 and Fig. 5.

The parameters shown in Fig. 4 in red, of selected optimal solution are described by the names and the values obtained after the optimization as follows:

1. The coordinate of jib base point O_1 along coordinate axis $O_K Y_K - R_K = 1.920$ m;
2. The polar radius of guy base point O_2 in regard to point $O_5 - d_{OTII} = 0.5650$ m;
3. The polar angle of distance d_{OTII} to guy base point O_2 in regard to the rear arm of the counterweight $l_{II} - \varepsilon_{OTII} = 2.6472$ rad;
4. The polar radius of axis in point O_3 wire rouse deviating pulleys on the guy base point $O_2 \equiv O_3$ these points are coincident - d_K = it changes by moving of point O_2 ;
5. The polar angle of distance d_K to point $O_2 \equiv O_3$ wire rouse deviating pulleys on the guy base point $O_2 - \eta_K$ = it changes by moving of point O_2 ;
6. The length of jib - $l_C = 27.693$ m;
7. The length of jib arm - $l_X = 16.048$ m;
8. The length of jib arm front part - $l_{IX} = 11.511$ m;
9. The angle in jib arm front part between axes $O_X O_C$ and $O_X O_{OT} - \varepsilon_X = 0.0592$ rad;
10. The length of guy - $l_{OT} = 23.548$ m;
11. The coordinate of wire rouse deviating pulleys along the length of jib arm - c = according to l_X , l_{IX} and ε_X ;
12. The distance from point O_1 to point $A - l_T = 6.751$ m;
13. The angle between the longitudinal axis of jib $O_1 C$ and $O_1 A - \varepsilon_C = 0.1553$ rad;
14. The polar radius of the swing counterweight axis point O_5 in regard to point $O_1 - d_{II} = 11.792$ m;
15. The polar angle of distance d_{II} to point O_5 in regard to the negative direction of the axis $O_K Y_K - \eta_{II} = 1.0604$ rad;
16. The length of counterweight arm - $l_{II} = 3.518$ m;
17. The angle on the swing counterweight axis in point $O_5 - \varepsilon_{II} = 2.5998$ rad;
18. The length of rocker arm front arm - $f_{II} = 2.185$ m;
19. The length of tie-bar - $b_{II} = 8.652$ m;
20. The distance from point O_1 to $S - L_P = 6.011$ m;
21. The angle between the longitudinal axis of jib $O_1 C$ and $O_1 S - \varepsilon_P = 0.1503$ rad;

22. The polar radius of axis in point O_4 of LGM in regard to point $O_1 - d_p = 6.552$ m;

23. The polar angle of distance d_p to point O_4 in regard to the negative direction of the axis $O_K Y_K - \eta_p = 0.9089$ rad;

24. The maximum bending force of jib arm, specified conditionally - $F_{XMax} = 189$ kN;

25. The maximum bending force of jib, specified conditionally - $F_{CMMax} = 53$ kN;

26. The maximum tensing force of guy, specified conditionally - $F_{OTMax} = 608$ kN;

27. Mass of counterweight - $m_{II} = 20.993$ t;

28. The jib inclination angle, which corresponding to maximum operating radius - $\varphi_{C0} = 0.8127$ rad;

29. The jib inclination angle, which corresponding to minimum operating radius - $\varphi_{CM} = 1.4184$ rad;

30. The cargo block and tackle reduces factor - $u_{IIOL} = 1$;

The dependencies of the forces acting on the level luffing jib system elements for a selected prototype of gantry crane and the results for the forces of the optimized new structure according to Fig. 4, are shown in Fig. 7. It can be seen that the values of the forces in the new optimized structure are significantly smaller than the existing gantry crane level luffing jib system. This is a very good indicator. In this way, the total mass of the level luffing jib system

structure can be reduced. This is followed by other positive effects. When reducing the total mass of the level luffing jib system, the mass of the gantry

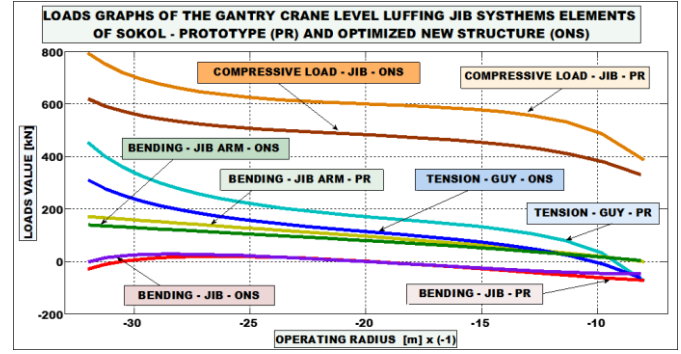


Fig. 7. Dependencies of the forces acting on level luffing jib system elements

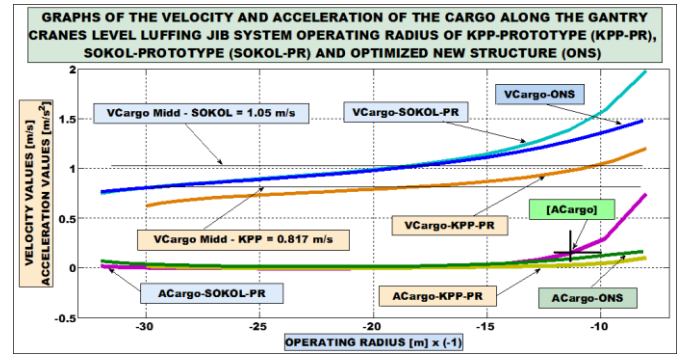


Fig. 8. Graphs of the cargo velocity and acceleration

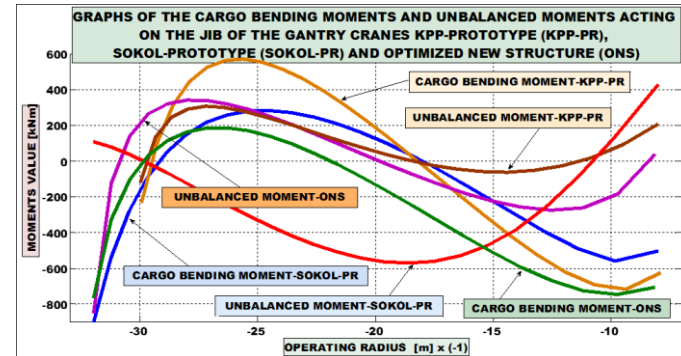


Fig. 9. Graphs of the moments acting on the jib

crane rotating part is reduced. It is possible to reduce the power of the electric motor in the crane slewing system also. When the mass of moving elements decreases, the forces of inertia acting on them and other elements are also reduced.

The cargo velocity and acceleration graphs for two models of gantry cranes selected for prototypes and the graphs of cargo velocity and acceleration for the new structure are shown in Fig. 8. The graphs show that for the new structure on minimum operating radius, significantly lower velocity and acceleration values have been achieved. Decreasing speed is beneficial as the diapason of speed variation along the operating radius is reduced. This makes the movement of the cargo evenly. The acceleration reduction at minimum operating radius is also beneficial because it limits the possibility of cargo pendulum. This makes crane and cranedriver work calm and precise, which is essential for quality and productive work.

The graphs of the cargo unbalanced moments and jib unbalanced moments for the two model of gantry cranes selected for prototypes and the same graphics for the optimized new structure of level luffing jib systems are shown in Fig. 9. It is seen that the

values of these criteria are commensurate between the prototypes and the new construction.

In the optimization studies to date the power of the electric motor has not been reduced.

The deviation of the cargo from the horizontal trajectory is also maintained within half a meter. When working with both complexes of programs feels the influence of equidistant and non-equidistant trajectories of the top of the jib arm point O_X and the cargo. These trajectories are reached by the passing of the lifting ropes through the level luffing jib system.

5. Conclusion

5.1. The systematic approach to research, analysis and synthesis of gantry cranes level luffing jib system has significant positive aspects. It allows that the level luffing jib system to be viewed in conjunction with other systems of the crane, load system of the crane and human systems. The system approach creates a clear and universal point of view for the denominations and interactions between the separate systems of the crane and between the elements in the systems themselves.

5.2. No additional elements have been added to the gantry crane level luffing jib system when constructing the new construction. Only the unit in point O_2 of the guy mast has been changed, this node being mounted on the counterweight jib arm. This way we have done, point O_2 movable. This implies that the designs of the counterweight and of the guy moves so that all the gantry crane level luffing jib system elements can be moved reciprocally. The construction must ensure that the hoisting ropes when pass through the guy and after that lead them to a machine room, point O_6 in Fig. 4 where the hoisting winches are located.

5.3. The study of the new construction has shown that can achieve better, lower values of the acceleration of the cargo along the operating radius, especially at minimum operating radius, while preserving the other basic parameters.

5.4. Trajectories at the top of the jib arm point O_X and the cargo can be controversial criteria.

5.5. Improved criteria values are obtained when the location of point O_2 is in the second and third quadrants of a coordinate system centered in point O_5 and a positive direction of the abscissa is in the l_H direction.

5.6. The forces values in the new optimized structure are significantly smaller round about 15...25 %, than the existing gantry crane level luffing jib system. Then come the positive effects caused by the lower loading forces of the gantry crane level luffing jib system elements - the smaller masses of the elements and the smaller inertial forces in the work of both systems, luffing sysytem and crane slewing system.

As a future work to improve and develop of this new structure of gantry cranes level luffing jib systems, it may be envisaged to develop the optimization procedure in order to improve other criteria defining the quality of the level luffing jib system work. It is a good idea to make a design improvement of the unit in point O_2 , which now is mounted on the counterbalance jib arm, taking into account the specificity of the new structure and the passage of the hoisting wire ropes through the level luffing jib system. It is possible after improving to check with the help of optimization procedure the

performance of the level luffing jib system in different operating modes and typical technological cycles.

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OPTIMAL SELECTION OF COMPONENTS FOR A HYDRAULIC EXCAVATOR SWINGING MECHANISM

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Abstract: In the present paper by the use of the Monte Carlo method, an optimal selection of components for a hydraulic excavator swinging mechanism has been made. Three objective functions are defined – the total mass of the components “hydraulic motor”, “gearbox”, “slewing bearing” and “small sprocket”, as well as the product of the total price and the total mass and the price of the same components. By computation of the objective function for multiple random combinations of the components and subsequent ranking of the objective function values, an optimal combination of the components is selected. A major advantage of the method is that it is suitable for computer realization and the number of required calculations is relatively small. Another advantage is that by the objective function value rankings close to the optimal additional combinations are obtained, which significantly extends the design possibilities.

Keywords: HYDRAULIC EXCAVATOR, SWINGING MECHANISM, OPTIMAL SELECTION

1. Introduction

The swinging mechanism of the hydraulic excavators is intended to rotate the excavator platform together with the front digging manipulator about the vertical axis. It does not participate directly in the earthmoving operations and has auxiliary functions, mainly by performing the transport operations of the excavated soil, as well as the positioning of the digging manipulator in the required position [1,2]. The hydraulic motor and the gearbox are essential components of the swinging mechanism. The most common kinematic scheme of the swinging mechanism consists of an axial-piston hydraulic motor (Fig.1,pos.1) coupled to a vertical two-stage planetary gearbox (Fig.1, pos.2), whose output shaft with the coupled small sprocket is engaged with the slewing bearing toothed ring (Fig.1,pos.3).

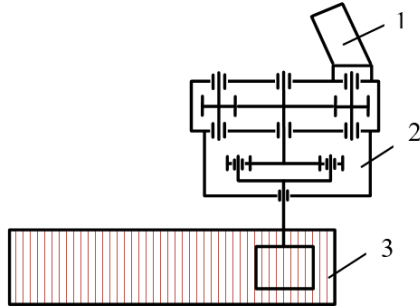


Fig. 1 Kinematic scheme of an excavator swinging mechanism

The hydraulic motor, the two-stage planetary gearbox and the slewing bearing are standardized units with known technical specifications that's why there are possible multiple combinations between them for achieving the needed kinematic and torque specifications of the swinging mechanism - it is possible to connect a hydraulic motor with a smaller torque to the gearbox with a larger gear ratio and vice versa. Obviously, an optimal combination satisfying the predefined requirements must be selected.

2. Definition of the optimization task

A possible objective function is the minimum of the total mass m of the hydraulic motor, the gearbox, the slewing bearing and the small sprocket:

$$m = m_{hm} + m_r + m_v + m_m \rightarrow \min \quad (1)$$

where the following notations are used: m_{hm} – the mass of the hydraulic motor; m_r – the mass of the planetary gearbox; m_v – the mass of the slewing bearing; m_m – the mass of the small sprocket.

If by H denote the set of the available different hydraulic motors, then they can be represented by a matrix consisting of the technical characteristics of every motor (in columns):

$$H = \begin{bmatrix} 1 & 2 & 3 & \dots & n \\ M_{hm}^1 & M_{hm}^2 & M_{hm}^3 & \dots & M_{hm}^n \\ \omega_{hm}^1 & \omega_{hm}^2 & \omega_{hm}^3 & \dots & \omega_{hm}^n \\ m_{hm}^1 & m_{hm}^2 & m_{hm}^3 & \dots & m_{hm}^n \\ p_{hm}^1 & p_{hm}^2 & p_{hm}^3 & \dots & p_{hm}^n \end{bmatrix} \quad (2)$$

where the following notations are used: $i=1, 2, 3, \dots, n$ – the sequential element numbers of the available hydraulic motors; n – the dimensionality of the set H ; $M_{hm}^1, M_{hm}^2, M_{hm}^3, \dots, M_{hm}^n$ – the corresponding nominal torques of the hydraulic motors; $\omega_{hm}^1, \omega_{hm}^2, \omega_{hm}^3, \dots, \omega_{hm}^n$ – the corresponding nominal angular velocities of the hydraulic motors; $m_{hm}^1, m_{hm}^2, m_{hm}^3, \dots, m_{hm}^n$ – the masses of the hydraulic motors; $p_{hm}^1, p_{hm}^2, p_{hm}^3, \dots, p_{hm}^n$ – the prices of the hydraulic motors.

In a similar manner, by R is denoted the set of the available gearboxes:

$$R = \begin{bmatrix} 1 & 2 & 3 & \dots & k \\ i_r^1 & i_r^2 & i_r^3 & \dots & i_r^k \\ \eta_r^1 & \eta_r^2 & \eta_r^3 & \dots & \eta_r^k \\ m_r^1 & m_r^2 & m_r^3 & \dots & m_r^k \\ p_r^1 & p_r^2 & p_r^3 & \dots & p_r^k \end{bmatrix} \quad (3)$$

where: $j=1, 2, 3, \dots, k$ – the sequential element numbers of the available gearboxes; k – the dimensionality of the set R ; $i_r^1, i_r^2, i_r^3, \dots, i_r^k$ – the corresponding gear ratios of the gearboxes; $m_r^1, m_r^2, m_r^3, \dots, m_r^k$ – the corresponding masses; $\eta_r^1, \eta_r^2, \eta_r^3, \dots, \eta_r^k$ – the efficiencies of the gearboxes; $p_r^1, p_r^2, p_r^3, \dots, p_r^k$ – the prices of the gearboxes.

By O is represented the set of available slewing bearings:

$$O = \begin{bmatrix} 1 & 2 & 3 & \dots & l \\ z_v^1 & z_v^2 & z_v^3 & \dots & z_v^l \\ m_v^1 & m_v^2 & m_v^3 & \dots & m_v^k \\ p_v^1 & p_v^2 & p_v^3 & \dots & p_v^k \end{bmatrix} \quad (4)$$

where: $r=1, 2, 3, \dots, l$ - the sequential element numbers of the available slewing bearings; l - the dimensionality of the set O ; $z_v^1, z_v^2, z_v^3, \dots, z_v^l$ - the number of the toothed ring teeth; $m_v^1, m_v^2, m_v^3, \dots, m_v^k$ - the corresponding masses of the slewing bearings; $p_v^1, p_v^2, p_v^3, \dots, p_v^k$ - the prices of the slewing bearings.

By K is denoted the set of the available small sprockets:

$$K = \begin{bmatrix} 1 & 2 & 3 & \dots & q \\ z_m^1 & z_m^2 & z_m^3 & \dots & z_m^q \\ m_m^1 & m_m^2 & m_m^3 & \dots & m_m^q \end{bmatrix} \quad (5)$$

where: $s=1, 2, 3, \dots, q$ - the sequential element numbers of the available small sprockets; q - the dimensionality of the set K ; $z_m^1, z_m^2, z_m^3, \dots, z_m^q$ - the number of the small sprocket teeth; $m_m^1, m_m^2, m_m^3, \dots, m_m^q$ - the masses of the small sprockets..

The price of the small sprocket can be expressed in terms of the number of the teeth by the following continuous approximate equation:

$$p_m = 0.252(z_m)^3 - 10.464(z_m)^2 + 149.29z_m - 696.86 \quad (6)$$

which is defined for the interval $z_m = 12 \div 18$.

By the superscripts are denoted the sequential element numbers of the elements in the corresponding sets. Obviously, the combinations of the elements with a certain number of different sets will be optimal with respect to the defined objective function and constraints. The number of possible combinations between the elements of the sets is $nklq$.

The constraints of the defined optimization task are defined by the kinematical and the torque requirements to the swinging mechanism.

The torque, created by the hydraulic motor and applied to the rotational platform, must be larger or equal to the needed one $M_{pl}^{\max}$, determined for the platform rotation with the predefined angular acceleration, so the first constraint is:

$$M_{hm}\eta_r\eta_z i_z \geq M_{pl}^{\max} \quad (7)$$

where by i_z is denoted the gear ratio of the internal gearing (between the small sprocket and the slewing bearing toothed ring); η_z - the efficiency of the internal gearing, which is considered as known and constant; η_r - the efficiency of the gearbox, considered as a function of the gearbox ratio i_r .

For the known teeth number of the small sprocket z_m toothed ring of the slewing bearing z_v , the gear ratio i_z is determined as:

$$i_z = \frac{z_v - z_m}{z_m} \quad (8)$$

The second constraint is formulated from the requirement the nominal angular velocity of the hydraulic motor not to be exceeded when the predefined maximal angular velocity of the platform $\omega_{pl}^{\max}$ is reached:

$$\omega_{pl}^{\max} \leq \frac{\omega_{hm}}{i_r i_z} \quad (9)$$

The defined optimization task belongs to the class of the discrete optimization tasks, which are solved by suitable mathematical methods [4]. The objective function (1) has a discrete character due to the discrete nature of the elements and their specifications in each set. Since the optimization task solution set consists of a finite number of objective function values, the optimal value can be determined by calculating all possible combinations between the elements and determining the combination with minimal objective function value that satisfy the defined constraints [6]. In the case of a big dimensionality of the problem, this method is ineffective.

3. Solution of the optimization task

In the present paper solution of the optimization task is performed by the Monte-Carlo method [3]. The defined optimization task is deterministic in nature. That is why to use the Monte-Carlo method, there is a need to impart a stochastic character to the optimization task. This is done as in each simulation step in a random fashion with the same probability a sequential number of the elements from every set is selected. In Excel, this is done by defining the discrete cumulative distribution function of consecutive numbers in sets. Figure 2 shows the cumulative distribution function the set H . The selection of a specific sequence number is performed by generating a random number using the RAND() and VLOOKUP functions. To ensure the independence of the generated random sequential element numbers from the four sets, the selection of the numbers from each of the sets is performed by four independent random number generators.

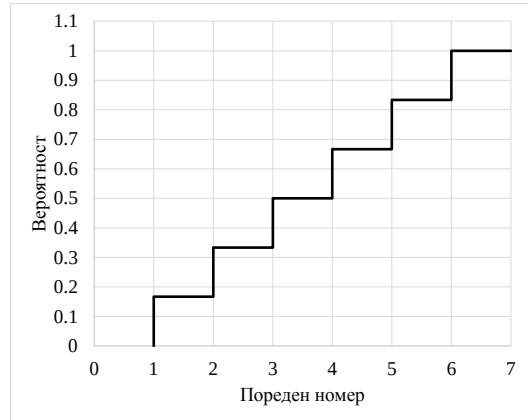


Fig.2 Discrete cumulative distribution function of the sequential element numbers for the hydraulic motors set

The determined in random fashion sequential numbers of elements in every set and their corresponding characteristics are used to calculate the left-hand sides of the constraints (10) and (11):

$$O_1: i_r^j i_z^{(r,s)} - \frac{M_{pl}^{\max}}{M_{hm}^i \eta_r^j \eta_z} \geq 0 \quad (10)$$

$$O_2: i_r^j i_z^{(r,s)} - \frac{\omega_{hm}^i}{\omega_{pl}^{\max}} \leq 0 \quad (11)$$

For every randomly generated combination (i,j,r,s) is calculated the indicator variable $I_{(i,j,r,s)}$, which takes the value 1 if both constraints are satisfied and 0 - else:

$$I_{(i,j,r,s)} = \begin{cases} 1, & \text{if } O_1 \cap O_2 = \text{true} \\ 0, & \text{else} \end{cases} \quad (12)$$

In some cases, the combination of two adjacent elements (hydraulic motor-gearbox, gearbox-small sprocket, small sprocket-slewing bearing) is not possible, and an another indicator variable

$K_{(i,j,r,s)}$ is used. It takes the value of 1 if the generated random combination is possible and 0 – else:

$$K_{(i,j,r,s)} = \begin{cases} 1, & \text{if } (i,j,r,s) \rightarrow \text{possible} \\ 0, & \text{else} \end{cases} \quad (13)$$

The value of the objective function is computed for every of the randomly generated combinations:

$$m(i,j,r,s) = K_{(i,j,r,s)} I_{(i,j,r,s)} (m_{hm}^i + m_r^j + m_v^r + m_m^s) \quad (14)$$

The steps described are repeated many times, and with the exception of the zero values, the smallest value of the objective function is selected and the corresponding combination of elements (i,j,r,s) is identified.

A practical approach to reducing the number of conducted experiments is to use the method in two or more stages. At the first stage, the determination of the minimum value is carried out by conducting a relatively small number of experiments [5]. At the second stage in the vicinity of the extremum, additional experiments are carried out to determine more accurately the optimal combination.

The described optimization algorithm is illustrated by a numerical example. For Bosch-Rexroth A6VM series 65 hydraulic motor, the H matrix has the following form:

$$H = \begin{bmatrix} 1 & 2 & 3 & 4 & 5 & 6 \\ 349 & 509 & 681 & 891 & 1019 & 1273 \\ 708 & 621 & 565 & 518 & 494 & 462 \\ 28 & 36 & 46 & 61 & 62 & 78 \\ 520 & 630 & 720 & 954 & 1225 & 1600 \end{bmatrix} \begin{matrix} [Nm] \\ [rad/s] \\ [kg] \\ [€] \end{matrix}$$

For the gearboxes set the matrix R is:

$$R = \begin{bmatrix} 1 & 2 & 3 & 4 & 5 & 6 & 7 \\ 10 & 13 & 18 & 24 & 28 & 42 & 50 \\ 0.98 & 0.98 & 0.94 & 0.92 & 0.9 & 0.9 & 0.88 \\ 140 & 142 & 148 & 153 & 166 & 172 & 180 \\ 250 & 320 & 380 & 380 & 440 & 500 & 490 \end{bmatrix} \begin{matrix} [kg] \\ [€] \end{matrix}$$

For the slewing bearings set the matrix O is:

$$O = \begin{bmatrix} 1 & 2 & 3 \\ 113 & 127 & 151 \\ 855 & 1120 & 1493 \\ 3300 & 3890 & 4450 \end{bmatrix} \begin{matrix} [kg] \\ [€] \end{matrix}$$

For the small sprockets set the matrix K is:

$$K = \begin{bmatrix} 1 & 2 & 3 & 4 & 5 & 6 & 7 \\ 12 & 13 & 14 & 15 & 16 & 17 & 18 \\ 8 & 12 & 14 & 17 & 20 & 22 & 24 \end{bmatrix} [kg]$$

At every step, the price of the small sprocket is calculated according to (6). The required values are $M_{pl}^{\max} = 65.5 kNm$ and $\omega_{pl}^{\max} = 0.8 rad/s$. Also, the efficiency is $\eta_z = 0.97$.

The total number of the possible combinations is $6.7.3.7 = 882$. When applying the algorithm with 300 experiments, the optimal combination found is $(i,j,r,s) = (2,3,1,1)$, and the objective function value is found to be 1047 kg. Figure 3 shows the objective function values for all conducted experiments for which the indicator variables are equal to 1. Since the mass of the slewing bearing significantly exceeds the masses of the other elements, the objective function values are grouped in three intervals.

One advantage of the used approach is that it allows in addition to the identified optimal combination to distinguish other combinations for which the values of the objective function are close enough to the optimal value. For example, in Fig. 4 are shown all combinations (26) for which the objective function value lies in the range $1040 \div 1080$ kg. Such a multivariance of the solution considerably widens the number of possible design decisions without having a significant deviation from the optimum value.

Table 1 shows the first 6 combinations that slightly deviate from the optimal value.

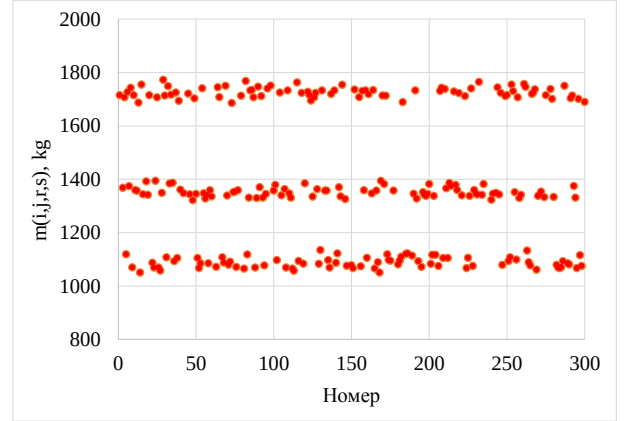


Fig.3 The objective function values

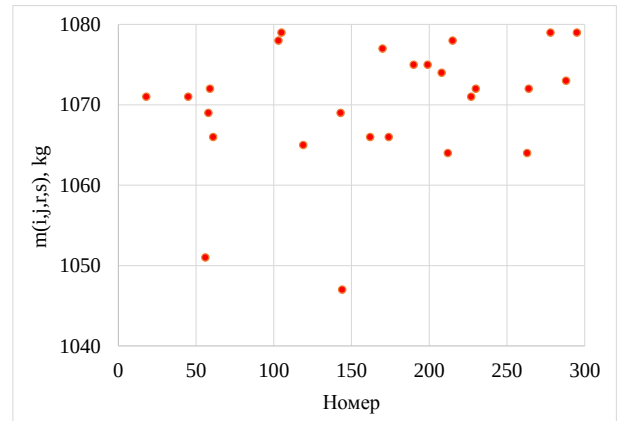


Fig.4 The objective function values, situated in the interval $1040 \div 1080$ kg

Table 1 Combinations that are close to the optimal

Combination (i,j,r,s)	Objective function values $m(i,j,r,s)$, kg	ε_M , %	ε_{ω} , %
(2,3,1,1)	1047	7.4	412.4
(3,2,1,1)	1051	8.1	545.5
(2,4,1,1)	1052	40.1	284.3
(2,4,1,2)	1056	28.0	320.5
(1,5,1,1)	1057	9.6	275.5

The table also shows the percentage deviations ε_M and ε_{ω} of the generated torques and angular velocities with respect to the required values. It can be seen that the deviations from the required torque values are considerably smaller than the deviations from the required angular velocity values. This points out that the constraint O_1 is "stronger" than the constraint O_2 , i.e. the torque constraint is

decisive for the choice of the hydraulic motor and the motor speed reserve is significant. This allows the O_2 condition to be removed from the optimization task, and after selecting a combination satisfying the O_1 constraint to perform verification that the nominal angular velocity of the hydraulic motor is not exceeded for this combination.

In addition, the objective function can be modified to take into account the price of the elements:

$$\begin{aligned} Z(i, j, r, s) &= K_{(i, j, r, s)} I_{(i, j, r, s)} m(i, j, r, s) p(i, j, r, s) = \\ &= K_{(i, j, r, s)} I_{(i, j, r, s)} (m_{hm}^i + m_r^j + m_v^r + m_m^s) (p_{hm}^i + p_r^j + p_v^r + p_m^s) \rightarrow \min \end{aligned} \quad (14)$$

where by $p(i, j, r, s)$ is denoted the price of the combination (i, j, r, s) . In Fig. 5 are shown the objective function values divided by 1000.

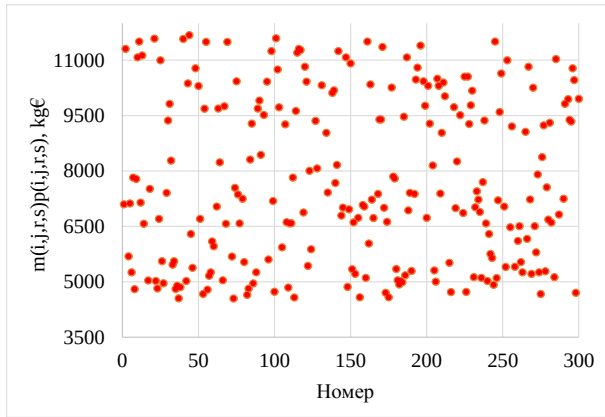


Fig.5. The objective function values

The optimal value of the objective function (14) is 4523.8×10^3 kg€ for the optimal combination (1,5,1,1). This combination is also present in the Table 1. If the objective function is based only on the price then the optimal value is 4279.8 € and again the optimal combination is (1,5,1,1).

By means of the already found optimal combination of the swinging mechanism elements, the size of the hydraulic pump can also be determined. The maximum flow rate Q_{hm} through the hydraulic motor is represented as [7]:

$$Q_{hm} = \frac{D_m^i \omega_{hm}^{\max}}{\eta_{hm}^v} \quad (15)$$

where D_m^i is the nominal volume of the selected hydraulic motor with number i from the set; $\omega_{hm}^{\max}$ is the maximum angular velocity of the hydraulic motor:

$$\omega_{hm}^{\max} = \omega_{pl}^{\max} i_r^j i_z^{(r, s)} \quad (16)$$

and η_{hm}^v is the volumetric efficiency of the hydraulic motor.

The flow rate of the hydraulic pump is:

$$Q_p = D_p \omega_p \eta_p^v \quad (17)$$

where ω_p is the constant angular velocity of the pump; η_p^v is the volumetric efficiency of the pump; D_p is the adjustable volume of the pump.

Taking into account that the flows of the motor and pump are equal, the maximum volume of the adjustable pump is determined by the following relationship:

$$D_p^{\max} = \frac{D_m^i \omega_{pl}^{\max} i_r^j i_z^{(r, s)}}{\omega_p \eta_{hm}^v \eta_p^v} \quad (18)$$

One need to choose the pump with standard volume $(D_p^{\max})_{st}$ larger then computed by (18).

4. Conclusions

The presented study has shown that, despite the stochastic nature of the Monte Carlo method, it is suitable for solving the task of optimizing the selection of the excavator swinging mechanism components. Three objective functions are defined – the total mass of the hydraulic motor, the gearbox, the small sprocket and the slewing bearing, as well as the product of the total price and the total mass and the total price of the same components. By the use of the multiple selections in a random fashion of combinations and subsequent ranking of the calculated objective function values, an optimal combination of components is chosen. A major advantage of the method is that it is suitable for computer realization and the number of required calculations is relatively small. Another advantage is that by the objective function value rankings close to the optimal additional combinations are obtained, which significantly extends the design possibilities.

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ENERGY EFFICIENCY AND ENERGY POWER LOSSES OF THE TURBO-GENERATOR STEAM TURBINE FROM LNG CARRIER PROPULSION SYSTEM

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Abstract: Turbo-generator (TG) steam turbine energy efficiency and energy power losses in a wide range of turbine loads were presented in this analysis. For TG steam turbine was investigated influence of steam specific entropy increment from the real (polytropic) steam expansion on energy power losses and energy efficiency. TG turbine energy power losses, during the all observed loads, were in the range from 646.1 kW to 685.5 kW. The most influenced parameter which defines change in TG turbine energy power losses is steam mass flow change, while for small steam mass flow changes, influence of steam specific entropy increment on steam turbine energy power losses is the most influential. Steam specific entropy incremental change can be used to estimate the change of TG steam turbine energy efficiency. Increase in steam specific entropy increment resulted with a decrease in TG turbine energy efficiency and vice versa. Analyzed steam turbine energy efficiency ranges from 53.84 % to 60.12 %, what is an expected range for low power steam turbines.

Keywords: TURBO-GENERATOR, STEAM TURBINE, ENERGY EFFICIENCY, ENERGY POWER LOSSES

1. Introduction

Marine steam turbine propulsion plants nowadays can be found in a number of LNG carriers [1]. Such steam propulsion plant consists of many constituent components [2] and one of them is turbo-generator (TG) which steam turbine is analyzed in this paper from the aspect of energy.

The analyzed LNG carrier has at disposal two identical turbo-generators which are designed to cover all ship requirements for electrical power. Each TG turbine has identical operating parameters (inlet and outlet temperatures, pressures and mass flows) and for the analysis is selected one of them. Steam turbine for each electric generator comprises of nine Rateau stages. Steam turbines with Rateau stages and their complete analysis can be found in [3]. Many details of the classic and special designs of marine steam turbines and their auxiliary systems are presented in [4] and [5].

The goal of the TG steam turbine analysis was to determine the specific entropy increment increase during steam expansion from the real exploitation for different steam turbine loads. Increase in steam specific entropy increment, usually indicate an increase in system energy power losses (in this analysis system is a TG steam turbine). It was examined the influence of steam specific entropy increment change on TG turbine energy power losses and energy efficiency change, at each observed operating point.

Main characteristics of the LNG carrier in which steam propulsion system is mounted analyzed turbo-generator are presented in Table 1.

Table 1. Main specifications of the LNG carrier

Dead weight tonnage	84,812 DWT
Overall length	288 m
Max breadth	44 m
Design draft	9.3 m
Propulsion turbine	Mitsubishi MS40-2 (max. power 29420 kW)
Turbo-generators	2 x Shinko RGA 92-2 (max. power 3850 kW each)

2. Equations for steam turbine energy analysis

2.1. General equations for the energy analysis

Energy analysis is defined by the first law of thermodynamics, which is related to the conservation of energy [6]. Mass and energy balance equations for a standard volume in steady state disregarding potential and kinetic energy can be expressed according to [7] and [8] as:

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

$$\dot{Q} - P = \sum \dot{m}_{OUT} \cdot h_{OUT} - \sum \dot{m}_{IN} \cdot h_{IN} \quad (2)$$

Energy power of a flow for any fluid stream can be calculated according to the equation [9]:

$$\dot{E}_{en} = \dot{m} \cdot h \quad (3)$$

Energy efficiency may take different forms depending on the type of the system. Usually, energy efficiency can be written as [10]:

$$\eta_{en} = \frac{\text{Energy output}}{\text{Energy input}} \quad (4)$$

2.2. Turbo-generator turbine energy efficiency and energy power losses

Steam turbine for each turbo-generator drive is condensing type and consists of nine Rateau stages [11]. Schematic view of steam turbine directly connected to an electric generator (the whole set of steam turbine and electric generator is called turbo-generator) is presented in Fig. 1. In Fig. 1 is also presented steam mass flow along with specific enthalpy and specific entropy at the steam turbine inlet and outlet.

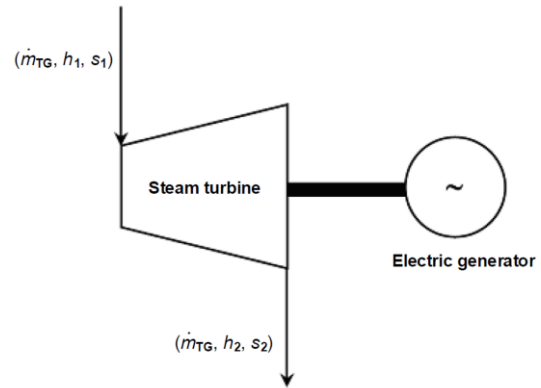


Fig. 1. Specific enthalpy, specific entropy and steam mass flow through the turbo-generator turbine

Steam mass flow in relation to the real developed turbine power for TG turbine, regarding the producer specifications, is presented in Fig. 2.

Real TG turbine power calculation at different loads was necessary for the TG turbine correct energy analysis. The turbine real developed power curve was approximated by the third degree polynomial using data from Fig. 2:

$$P_{TG,RE} = -4.354 \cdot 10^{-10} \cdot \dot{m}_{TG}^3 + 6.7683 \cdot 10^{-6} \cdot \dot{m}_{TG}^2 + 0.251318 \cdot \dot{m}_{TG} - 256.863 \quad (5)$$

where $P_{TG,RE}$ was obtained in (kW) when $\dot{m}_{TG}$ in (kg/h) was placed in the equation (5). Steam mass flow through the TG turbine ($\dot{m}_{TG}$) was measured component, while the developed real TG turbine power was calculated according to equation (5).

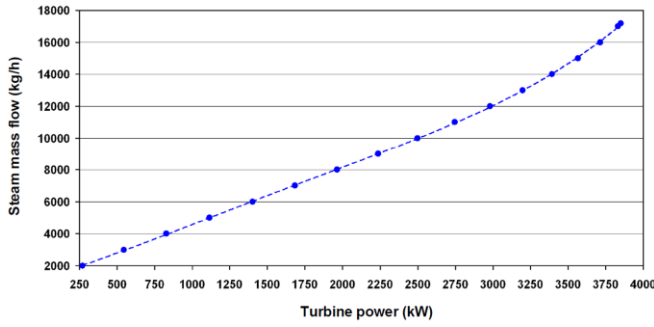


Fig. 2. Real TG turbine power in relation to the steam mass flow [11]

During measurements, no steam leakage on the analyzed TG turbine was observed, so the mass balance for the TG steam turbine inlet and outlet is valid as:

$$\dot{m}_{TG,1} = \dot{m}_{TG,2} = \dot{m}_{TG} \quad (6)$$

According to Fig. 1 and Fig. 3, h_1 is steam specific enthalpy at the turbine inlet, and h_2 is steam specific enthalpy at the turbine outlet after real (polytropic) expansion. Steam specific enthalpy at the turbine inlet was calculated from the measured pressure and temperature. Steam specific entropy at the turbine inlet s_1 was also calculated from measured steam pressure and temperature at the turbine inlet. Steam real specific enthalpy at the turbine outlet was calculated from the turbine power $P_{TG,RE}$ in (kW) and measured steam mass flow $\dot{m}_{TG}$ in (kg/s) according to [12] by using an equation:

$$h_2 = h_1 - \frac{P_{TG,RE}}{\dot{m}_{TG}} \quad (7)$$

The steam real specific entropy at the turbine outlet s_2 was calculated from steam real specific enthalpy at the turbine outlet h_2 , calculated by using equation (7), and measured pressure at the turbine outlet.

Steam specific enthalpy after isentropic expansion h_{2S} was calculated from the measured steam pressure at the turbine outlet p_2 and from known specific entropy at the turbine inlet s_1 . Ideal isentropic expansion assumes no change in steam specific entropy ($s_1 = s_{2S}$), Fig. 3.

Steam specific enthalpy at the turbine inlet, steam specific enthalpy at the end of isentropic expansion and both steam specific entropies (at the turbine inlet and outlet) were calculated by using NIST REFPROP 8.0 software [13].

To proper described TG turbine energy power losses, in any steam turbine operating range, it must be known the real turbine developed power and isentropic power, which can be developed in the ideal situation (when the change in steam specific entropy does not occur). Isentropic steam turbine power, according to Fig. 3, should be calculated as:

$$P_{TG,IS} = \dot{m}_{TG} \cdot (h_1 - h_{2S}) \quad (8)$$

Isentropic steam turbine power will always be higher than the real developed power, because of higher specific enthalpy difference (increment) during the isentropic expansion in comparison to the real polytropic expansion.

Steam turbine (TG turbine) energy power losses can be calculated as:

$$\dot{E}_{TG,en,PL} = P_{TG,IS} - P_{TG,RE} = \dot{m}_{TG} \cdot (h_2 - h_{2S}) \quad (9)$$

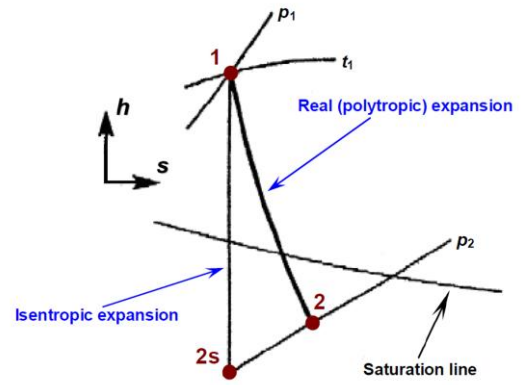


Fig. 3. Turbo-generator steam turbine real (polytropic) and ideal (isentropic) expansion

Energy efficiency of TG steam turbine can be calculated according to [14] and [15] by using the following equation:

$$\eta_{TG,en} = \frac{(h_1 - h_2)}{(h_1 - h_{2S})} = \frac{P_{TG,RE}}{P_{TG,IS}} \quad (10)$$

3. Measurement results and measuring equipment of the analyzed TG steam turbine

Measurement results of required operating parameters for TG turbine are presented in relation to the propulsion propeller speed, Table 2. Propulsion propeller speed is directly proportional to steam system load, higher propulsion propeller speeds denote higher steam system loads and vice versa.

Table 2. Measurement results for TG turbine

Propulsion propeller speed (rpm)	Steam pressure at the TG turbine inlet (MPa)	Steam temperature at the TG turbine inlet (°C)	Steam pressure at the TG turbine outlet (MPa)	Steam mass flow through TG turbine (kg/h)
25.00	6.21	491.0	0.00541	4648.83
41.78	6.22	491.0	0.00489	4556.16
56.65	5.97	490.5	0.00425	4000.58
65.10	6.07	491.0	0.00392	3838.78
70.37	6.07	502.5	0.00397	3778.91
76.56	6.01	504.5	0.00420	4070.84
80.44	5.89	501.5	0.00554	4689.03
83.00	5.90	493.5	0.00561	4487.93

All the measurement results were obtained from the existing measuring equipment mounted on the TG turbine inlet and outlet. List of all used measuring equipment was presented in the Table 3.

Table 3. Used measuring equipment for the TG turbine analysis

Steam temperature (TG inlet)	Greisinger GTF 601-Pt100, Immersion probe [16]
Steam pressure (TG inlet)	Yamatake JTG980A, Pressure Transmitter [17]
Steam pressure (TG outlet)	Yamatake JTD910A, Pressure Transmitter [18]
Steam mass flow (TG inlet)	Yamatake JTD960A, Pressure Transmitter [18]
Propulsion propeller speed	Kyma Shaft Power Meter, (KPM-PFS) [19]

4. TG steam turbine energy analysis results with the discussion

Steam specific entropy difference (increment) between the inlet and outlet of the TG steam turbine is presented in Fig. 4 for all the observed steam system loads. As the steam specific entropy at a TG turbine inlet is almost constant during the all propulsion propeller speeds, specific entropy increment is the most influenced by steam specific entropy at the TG turbine outlet.

From the lowest to the highest observed propulsion propeller speeds, TG turbine steam specific entropy increment (difference) firstly increases from 1.69 kJ/kg·K at 25.00 rpm up to 2.02 kJ/kg·K at propulsion propeller speed of 70.37 rpm, after which decreases to the lowest value of 1.69 kJ/kg·K at 83.00 rpm.

Increase in steam specific entropy increment, usually indicates an increase in system energy power losses, for a large number of different systems [14]. It will be interesting to analyze does the same conclusion is valid for the TG steam turbine.

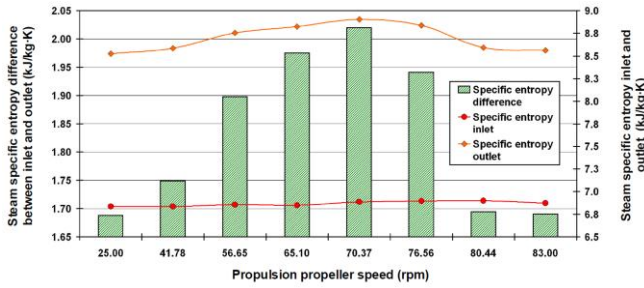


Fig. 4. Steam specific entropy change at the TG turbine inlet and outlet along with specific entropy difference (increment) between inlet and outlet

TG steam turbine isentropic and real power is presented in Fig. 5. The change in both TG turbine power for all observed propulsion propeller speeds must have the same trend. Isentropic TG turbine power can theoretically be developed by using the real operating parameters, but without any losses (without change in steam specific entropy). TG real power is the power developed according to real measured operating parameters in the LNG carrier propulsion system during navigation.

In the whole range of observed steam system loads, isentropic TG turbine power varies from 1423 kW up to 1711 kW, while in the same load range real TG turbine power varies from 766 kW up to 1025.5 kW. The real TG turbine power depends on the current need for electricity and it changes depending on the inclusion or exclusion of the individual electrical consumers.

The difference in isentropic and real TG turbine power represents energy power losses of the real TG steam turbine process in comparison with ideal one.

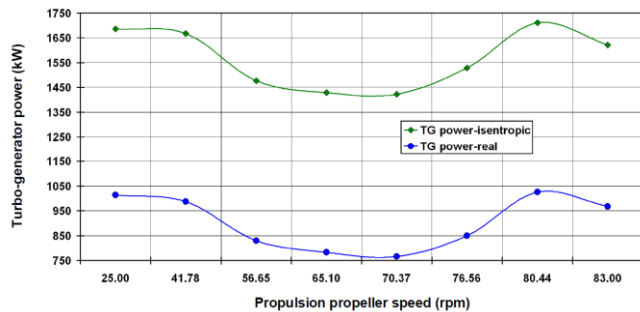


Fig. 5. Change in TG turbine power (real and isentropic) for all observed propulsion propeller speeds

Increase in steam specific entropy increment reduces available steam specific enthalpy difference which will be used in steam turbine. As a result, increase in steam specific entropy increment will cause a decrease in real developed steam turbine power.

From Fig. 4 and Fig. 6 can be seen that the change in TG turbine specific entropy increment does not have the most significant influence on TG steam turbine energy power losses. For example, between propulsion propeller speed of 41.78 rpm and 56.65 rpm, steam specific entropy increment increases, Fig. 4, while for the same propulsion propeller speeds TG turbine energy power loss decreases, Fig. 6. The same occurrence is visible between propulsion propeller speeds of 76.56 rpm and 80.44 rpm when steam specific entropy increment decreases while for the same propulsion propeller speeds TG turbine energy power loss increases.

The most significant influence on TG steam turbine energy power losses has steam mass flow through the turbine. In general, increase in steam mass flow will increase TG turbine energy power losses, while a decrease in steam mass flow will decrease TG turbine energy power losses. TG steam turbine energy power losses, between some observed operating points, during a small change in steam mass flow are also influenced by steam specific entropy increment.

Between propulsion propeller speed of 41.78 rpm and 56.65 rpm, TG turbine energy power losses decrease, Fig. 6, because of a noticeable decrease in steam mass flow from 4556.16 kg/h to 4000.58 kg/h, Table 2. Increase in steam specific entropy increment between these two propulsion propeller speeds does not have significant influence on energy power losses change, Fig. 4.

Also, between propulsion propeller speeds of 76.56 rpm and 80.44 rpm TG turbine energy power losses increases, Fig. 6, because of noticeable increase in steam mass flow from 4070.84 kg/h to 4689.03 kg/h, Table 2, regardless of steam specific entropy increment noticeable decrease, Fig. 4.

On the other side, for a small change in steam mass flow, steam specific entropy increment can have an important influence on TG turbine energy power losses change. When compared propulsion propeller speeds of 65.10 rpm and 70.37 rpm, steam mass flow decreases from 3838.78 kg/h to 3778.91 kg/h, Table 2, but this steam mass flow decrease does not cause a decrease in TG turbine energy power losses, Fig. 6. Between these two propulsion propeller speeds, increase in TG turbine energy power losses occurs because of a notable increase in steam specific entropy increment, Fig. 4.

Final conclusion which can be derived is that the most influenced parameter on TG turbine energy power losses is steam mass flow. For a small change in steam mass flow, steam specific entropy increment takes a leading role in affecting the change of TG turbine energy power losses.

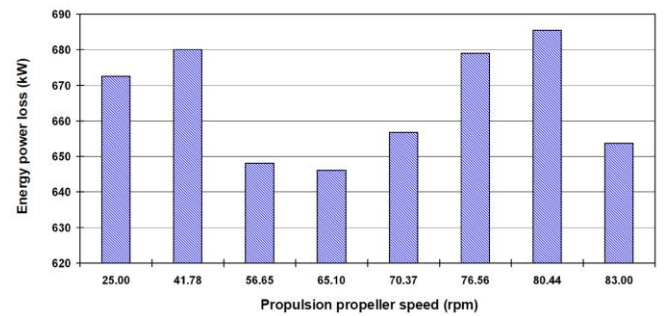


Fig. 6. TG turbine energy power loss change in all observed propulsion propeller speeds

Steam specific entropy increment can be used as an essential parameter for evaluation of TG steam turbine energy efficiency change. During the increase in steam specific entropy increment, TG turbine energy efficiency decreases and during the decrease in steam specific entropy increment, TG turbine energy efficiency increases. This conclusion is valid for every two observed propulsion propeller speeds, during the whole investigated TG turbine load range, Fig. 4 and Fig. 7.

The highest TG turbine energy efficiency of 60.12 % was obtained for the lowest steam specific entropy increment of 1.69

kJ/kg·K at the propulsion propeller speed of 25.00 rpm (the lowest observed TG turbine load), Fig. 7. The lowest TG turbine energy efficiency of 53.84 % was obtained for the highest steam specific entropy increment of 2.02 kJ/kg·K at the propulsion propeller speed of 70.37 rpm.

The analyzed TG steam turbine is a low power steam turbine. Its energy efficiency, for the observed loads, ranges from 53.84 % to 60.12 %, what is an expected range of energy efficiency for low power steam turbine in general [9].

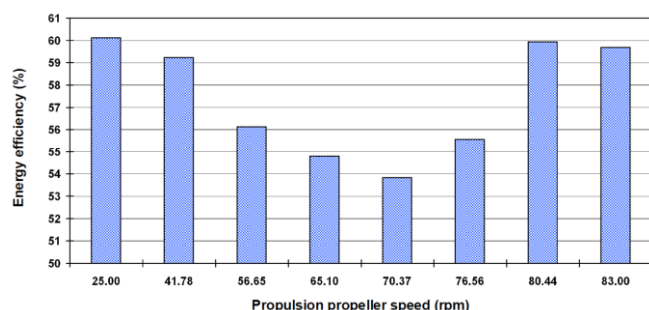


Fig. 7. TG turbine energy efficiency change in all observed propulsion propeller speeds

5. Conclusions

This paper presents an analysis of energy efficiency and energy power losses for low power steam turbine, in a wide range of turbine loads. For the analyzed TG steam turbine was investigated influence of steam specific entropy increment from the real (polytropic) expansion process on energy power losses and energy efficiency.

TG steam turbine energy power losses were calculated as a difference between steam turbine real developed power (polytropic steam expansion) and power which can be developed in an ideal situation without any specific entropy increment (isentropic steam expansion). It was found that TG turbine energy power losses, during the all observed loads, were in the range from 646.1 kW to 685.5 kW. Steam specific entropy increment does not have a major influence on TG turbine energy power losses change in general, but for small steam mass flow change, influence of steam specific entropy increment on steam turbine energy power losses is dominant. The most influenced parameter which defines change in TG turbine energy power losses is steam mass flow - increase in steam mass flow caused an increase in TG turbine energy power losses and vice versa.

Steam specific entropy increment change can be used to estimate the change of TG steam turbine energy efficiency. Increase in steam specific entropy increment resulted with a decrease in TG turbine energy efficiency and decrease in steam specific entropy increment resulted with an increase in TG turbine energy efficiency.

NOMENCLATURE

Abbreviations:

LNG Liquefied Natural Gas
TG Turbo generator

Latin Symbols:

$\dot{E}$ stream flow power, kJ/s
 h specific enthalpy, kJ/kg
 $\dot{m}$ mass flow, kg/s or kg/h
 p pressure, MPa
 P power, kJ/s
 $\dot{Q}$ heat transfer, kJ/s
 s specific entropy, kJ/kg·K

Greek symbols:

η efficiency, -

Subscripts:

en energy
IN inlet
IS isentropic (ideal)
OUT outlet
PL power loss
RE real

6. Acknowledgment

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THE DEVELOPMENT OF CALIBRATION FOR THE ROLLING BALLS OF DIAMETER 40 MM IN CONDITIONS OF JSC "SSGPO"

РАЗРАБОТКА КАЛИБРОВКИ ДЛЯ ПРОКАТКИ ШАРОВ ДИАМЕТРОМ 40 ММ В УСЛОВИЯХ АО «ССГПО»

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Abstract: Paper presents a preliminary calculation of the calibration rolls for rolling balls with a diameter of 40 mm in a rolling mill JSC "SSGPO", which will be rolled from round workpieces with a diameter of 40 mm. The main elements of ball caliber are considered, their design dimensions are determined. It is established that the height of the flange varies linearly and the width of the spherical area increases during rolling.

Keywords: ball, rolling, roll, calibration, calculation.

1. Introduction

For shape rolling mill 320 of JSC "SSGPO" two calibration schemes of round steel rolling were developed [1]:

1) from the billets 150×150 mm are produced round billets with a diameter of 100 mm, which are used as the grinding rods into rod mills factory complex of JSC "SSGPO".

2) from the initial billets 120×120 mm are produced round billets with a diameter of 40 mm, which are used on ball mills in the production of thermally hardened grinding balls diameter 40 mm.

In the analysis of techno-economic indicators was revealed that during the production of round bars with a diameter of 40 mm with the initial billet is 120×120mm, it has the untapped potential of production capacity of the rolling mill. The solution to this problem is the transformation of mill in the production of a round billet with a diameter of 40 mm from the initial billet 150×150 mm (figure 1). Such technological scheme of obtaining of a round billet with a diameter of 40 mm will allow to transfer the workpieces, unsuitable for rolling circle diameter of 100 mm, to the production of a circle with a diameter of 40 mm.

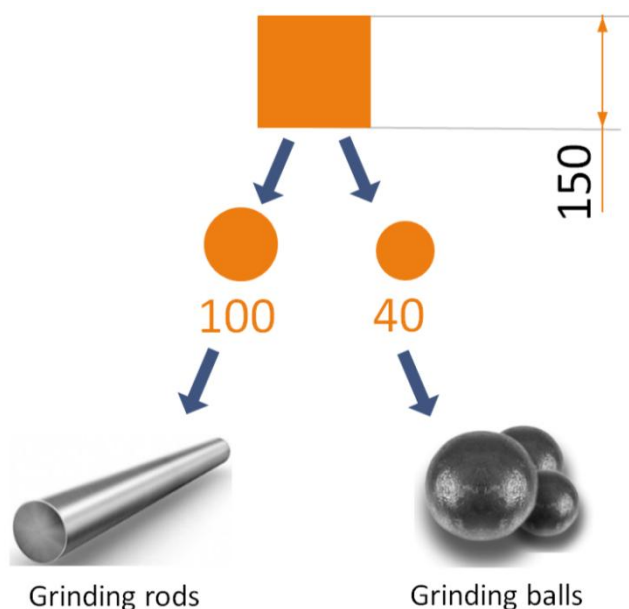


Figure 1 –New scheme of production

In work [2] a calibration of rolls for rolling round steel № 40 from the billet of cross section 150 x 150 mm in a mill 320 was developed. For this goal the total number of passes was increased from 9 to 11, the evaluation of compliance with emerging values of the power parameters of permissible values was considered (for the exception of equipment failure), and also conformity assessment of the shape and size of the final profile plant requirements was performed (to prevent defects of the geometry). In the result, it was

found that new developed calibration for obtaining the round steel with a diameter of 40 mm fully conforms to the technical parameters of shape mill 320. In this work the calculation of the calibration rolls for rolling balls with a diameter of 40 mm under conditions of JSC "SSGPO" from the original round billet with a diameter of 40 mm is presented.

2. Statement of the problem and initial data

Rolling of ball workpieces is performed between the two rolls 1 and 2 rotating in one direction, on their barrels the screw gauges are cut (figure 2). The axis of rolls usually tilted at a small angle to the axis of the rolled workpiece 3, thereby providing axial flow of the metal into the rolls. Rotating workpiece is crimped by flanges of the gauge and takes the form of a ball connected by a bridge with the rest of the workpiece. During the further promotion the ball is calibrated and completely separated from the rod.

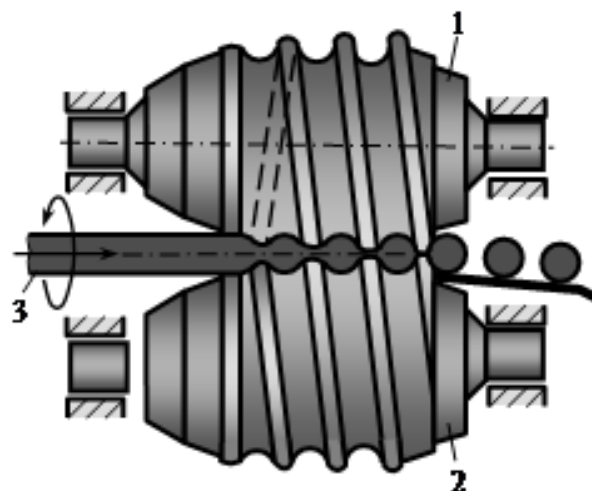


Figure 2 – Scheme of the ball rolling

During balls rolling, the deformation zone has two main sections [3]: forming section, where the workpiece is crimped with a change in the shape and size of the screw flange; and finishing section, where the shape and size of the flange remain constant, but the compression is due to the ovalization of the workpiece. The forming of the ball is performed by the flanges of the rolls, the height of which gradually increases. For ease of calculation, calibration and manufacturing of the rolls it is accepted that the height of the flange of the caliber changes according to the law of a straight line. For normal rolling process the profile and dimensions of the forming section of caliber are calculated so that during the compression of the workpiece there are two basic principles [4]:

1) volume of metal, crimped in the caliber, should remain constant during the forming process;

2) change the profile and size of the caliber flange should match

the stretching of crimping workpiece.

According to the first position, it is necessary that the volume of a certain part of the workpiece captured by the rolls, was unchanged as it passes through the remaining sections of the caliber. In this case, at any time of rolling will not be an excess of metal. When the excess metal the geometric form of sphere is distorted and the formation of voids in the axial zone of the workpiece is possible. The presence of a small excess of metal is permitted only at the beginning of caliber, when the flange is still relatively low and does not interfere with the displacement of the metal from the caliber.

The second condition is that for the normal formation of the ball shape, the stretching of compressed area should match to the changing shape and sizes of flange. The length of compressed bridge must be equal to the width of the straight portion of the flange. If the change of width of the flange is smaller of caliber stretching, the metal will move away from the flange, and the surface of the workpiece to form a reel, which upon further compression of the workpiece is rolled out in defect. If the change of width of the flange is more than the stretching of the workpiece, then in crimped bridge arise axial tensile stresses, which can lead to rupture of the bridge. Thus, to perform both the first and second conditions, the flange on different parts of the caliber must have a certain thickness. In this regard, the forming section of the caliber has a variable cutting step.

When developing the rolls calibration, it was decided to use the calculation algorithm described in detail in works [5-6]. In this work the results of preliminary calculation are presented. The initial data for the calibration are the technical characteristics of the rolling mill, size of the rolls, diameter of the resulting ball.

The calculation of the calibration ball is performed for the left roll, calibration of right roll is similar. The only difference is that after cutting the flange of the right roll it is cut on height by a certain value. The source data for calibration: diameter of the ball $d_B = 40$ mm; diameter of rolls $d = 300$ mm. In the calculation and construction of the sizing rolls it should be guided by the recommendations are given in table 1.

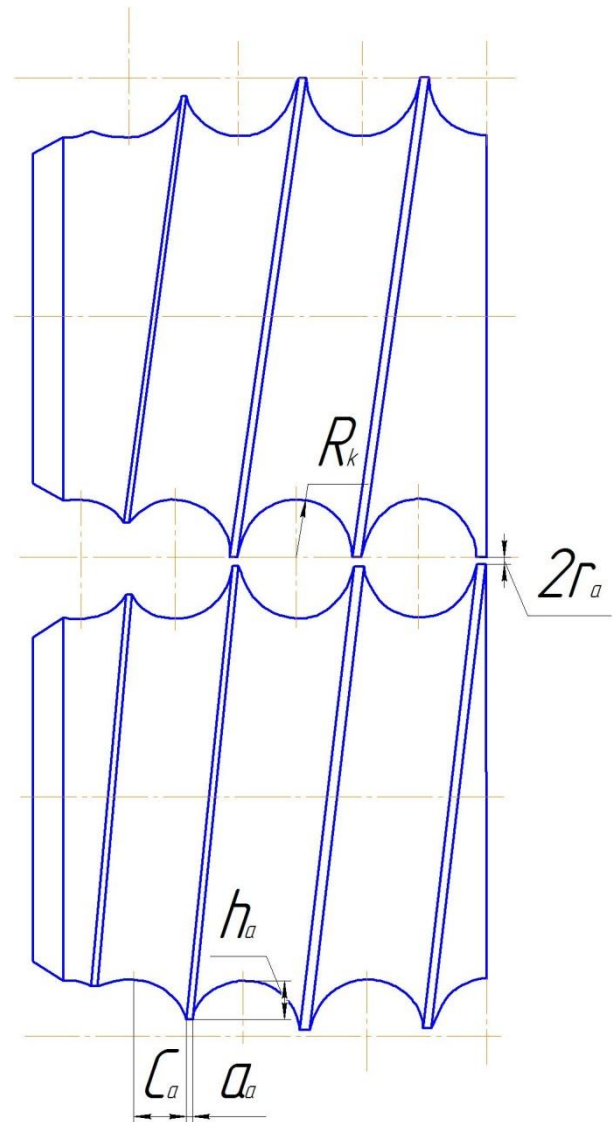


Figure 3 – Basic elements of a calibration for balls rolling:
 h_a and a_a – height and width of flange;
 r_a and R_K – radii of the bridge and caliber;
 C_a and S_a – width of spherical and cylindrical sections of caliber

Table 1 – Initial data for calculation of calibration

d_B , mm	d , mm	Length of the caliber, grad		$2r$, mm	The size of flange, mm		
		α_{FULL}	α_{FIN}		height on the capture	width at the finishing area	width on a area of 270° from the capture
40 - 50	180 - 300	900 - 1080	540 - 720	3,0 – 3,2	2,5 - 3,8	3,7 - 5,4	2,0 - 2,4
60 - 80	280 - 460	1080 - 1350	630 - 810	3,6 - 5,0	4,4 - 4,5	5,8 - 8,4	3,0
100 - 125	550 - 690	1260 - 1350	630 - 810	6,0	5,4	9,0 - 11,0	3,5 - 4,0

According to GOST 7524 - 2015 "Hot-rolled steel balls for ball mills" choose the nominal diameter of the ball for subsequent calibration of the rolls subject to the permissible deviations. Accept that $d_B = 40$ mm, allowable deviation of diameter ± 2 mm. Then the diameter of the workpiece is $d_w = d_B / \eta_0 = 40 / 1,03 = 38,8$ mm, where η_0 - coefficient taking into account the radial growth of the ball during rolling (accept that $\eta_0 = 1,03$).

According to the regulations, the balls rolled from a billet of diameter 40 mm. Then the diameter of the ball is equal to:

$$d_B = d_w / \eta_0 = 40 \cdot 1,03 = 41,2 \text{ mm} \quad (1)$$

Basic sizes of rolls for ball-rolling mill are shown in figure 3.

3. Results and discussion

Diameter of caliber

$$d_K = d_B \cdot \eta_t = 41,2 \cdot 1,013 = 41,7 \text{ mm} \quad (2)$$

where $\eta_t = 1,013$ - coefficient taking into account thermal expansion of the metal.

The minimum width of the flange

$$a = 0,04R_K + 1,3 = 0,04 \cdot 20,85 + 1,3 = 2,1 \text{ mm}, \quad (3)$$

where $R_K = d_K / 2$ - radius of the caliber, mm;
The initial height of the flange

$$h_0 = 1,5 + 0,07(R_K - 10) = 1,5 + 0,07(20,85 - 10) = 2,3 \text{ mm}; \quad (4)$$

The minimum radius of the bridge

$$r = 1 + 0,04(R_K - 10) = 1 + 0,04(20,85 - 10) = 1,4 \text{ mm}; \quad (5)$$

The width of the flange before cutting

$$a_{\text{CUT}} = 4 + 0,14(R_K - 10) = 4 + 0,14(20,85 - 10) = 5,5 \text{ mm}. \quad (6)$$

The length of the caliber (number of turns) is conventionally measured in degrees of the spiral line. For the beginning of the caliber (0°) the center of the radius of the caliber is adopted, located on a straight line extending along the end face of the roll on the output side of the finished ball. During rolling balls are guided by the recommendations of table 1 and set the total length of caliber as $\alpha_{\text{FULL}} = 900^\circ$, the length of the forming area $\alpha_{\text{FORM}} = 270^\circ$ and the length of the finishing area $\alpha_{\text{FIN}} = \alpha_{\text{FULL}} - \alpha_{\text{FORM}} = 900 - 270 = 630^\circ$. The length of a caliber is divided in equal portions; for pre-calculation it is sufficient $\Delta\alpha = 90^\circ$.

The radius of the jumper at the end of the forming at $\alpha = \alpha_{\text{FORM}}$ - $\Delta\alpha$ is determined from the ratio

$$r_{\text{CUT}} = r + \frac{R_K - h_0 - r}{\alpha_{\text{FORM}}} \Delta\alpha = 1,4 + \frac{20,85 - 2,3 - 1,4}{270} 90 = 7,1 \text{ mm} \quad (7)$$

Change the height of a flange during turn the roll on 90° :

$$\Delta h = r_{\text{CUT}} - r = 7,1 - 1,4 = 5,7 \text{ mm} \quad (8)$$

For each angle of the rolls rotation from α_{FULL} till α_{FORM} after a period of 90° , the height of the flange will be:

$$h_{\alpha-90} = h_\alpha + \Delta h = h_\alpha + 5,7 \quad (9)$$

where h_α - current height of flange, mm.

Then the radius of the bridge is calculated, its size in the finishing area will be $r = 1.4 \text{ mm}$, and the radius of the bridge on the forming section is defined by the formula

$$r_{\alpha+90} = r_\alpha + \Delta h = r_\alpha + 5,7 \quad (10)$$

The width of the spherical section of the caliber is defined by the expression

$$C_\alpha = \sqrt{R_K^2 - r_\alpha^2} = \sqrt{20,85^2 - r_\alpha^2} \quad (11)$$

So the cut of bridge begins during forming of the ball, then take the width of the spherical area on the angle of rotation of the roll $\alpha_{\text{FIN}+90}$, the previous finishing area of the caliber, i.e. $C_{\text{CUT}} = C_{\alpha_{\text{FIN}+90}}$. Basic design data of pre-calibration stage are given in table 2.

Table 2 - Basic design data of pre-calibration stage of calibration rolls for rolling ball with a diameter of 40 mm

$\alpha, ^\circ$	$h_{\alpha-90}, \text{ mm}$	$r_{\alpha+90}, \text{ mm}$	$C_\alpha, \text{ mm}$
900	2,3	18,5	9,6
810	8,0	12,8	16,4
720	13,7	7,1	19,6
630	19,4	1,4	20,8
540	19,4	1,4	-
450	19,4	1,4	-
360	19,4	1,4	-
270	19,4	1,4	-

4. Conclusion

Paper presents a preliminary calculation of the calibration rolls for rolling balls with a diameter of 40 mm in a rolling mill JSC "SSGPO", which will be rolled from round workpieces with a diameter of 40 mm. The main elements of ball caliber are considered, their design dimensions are determined. It is established that the height of the flange varies linearly and the width of the spherical area increases during rolling.

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EXTRACTION OF PRECIOUS METALS FROM A PYRITIC CONCENTRATE PRETREATED BY MICROBIAL OXIDATION

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Abstract: A sulphide flotation concentrate containing 15.2 g/t gold and 893 g/t silver finely disseminated in pyrite (4.1 % sulphidic sulphur in the concentrate) was treated by a two-stage process to recover these precious metals. Initially the concentrate was subjected to microbial oxidation by means of different acidophilic chemolithotrophic microorganisms (bacteria at 37 °C and archaea at 59 and 86 °C) to expose the precious metals encapsulated in the pyrite. The precious metals liberated in this way were then subjected to leaching by means of solutions containing different reagents (protein hydrolysate, thiosulphate, cyanide and some chemical oxidizers). The leaching was carried out in agitated reactors and up to 93.6 % of the gold and 80.8 % of the silver were solubilised in this way for 48 hours from a pulp density of 20 % at 57 °C.

Keywords: PRECIOUS METALS, MICROBIAL OXIDATION, EXTRACTION

1. Introduction

Gold-bearing sulphide concentrates, ores and wastes in which the gold is finely disseminated as submicron size particles in the sulphide matrix are refractory to hydrometallurgical treatment for gold extraction since the leachants cannot penetrate into the interior of the minerals to reach the enclosed gold. However, since a relatively long period of time it is known that several acidophilic chemolithotrophic microorganisms (bacteria and archaea) are able to oxidize efficiently the sulphides containing these precious metals and to make them exposed for leaching by means of suitable reagents. Furthermore, it has been found that in most cases such microorganisms are able to attack preferentially just the defect sites of the sulphide minerals in which the precious metals are located (Livesey-Goldblatt et al., 1983; Groudev, 1989; Rawlings and Johnson, 2007; Karavaiko et al., 1985; Van Aswegen et al., 2007). The gold and silver can be extracted from the pretreated sulphidic concentrates by means of cyanides or by other less toxic reagents. The present paper contains some data about laboratory experiments for microbial pretreatment and subsequent leaching of a sulphide concentrate containing gold and silver present mainly as metals finely dispersed in sulphide minerals.

2. Materials and Methods

The concentrate used in this study was produced by flotation of low-grade sulphide ore taken from a dump located near the deposit Zlata in the north-west part of Bulgaria, in a short distance from the small town Trun. The concentrate contained 4.8 % of sulphur (from which 4.1 % was sulphidic, present in pyrite), 8.2 % of iron, 1.49 % copper, 15.2 g/t gold and 893 g/t silver as the valuable components. Data about the phase composition of the precious metals in the concentrate are shown in Table 1.

Table 1: Phase composition of the precious metals in the flotation concentrate.

Phase composition	Distribution, %	
	Au	Ag
Free exposed metals	11.3	-
Metals capsulated in iron oxides	35.2	37.0
Metals finely dispersed in sulphide minerals	50.3	59.4
Metals finely dispersed in silicates	3.2	3.6
Total content, %	100.0	100.0

The preliminary microbial oxidation of the concentrate to expose the precious metals encapsulated in the sulphides was performed by means of different acidophilic chemolithotrophic microorganisms (bacteria at 37 °C, and archaea at 57 and 86 °C). The microbial pretreatment was carried out in the 9K nutrient medium (Silverman and Lundgren, 1959).

The pretreatment was connected with an efficient extraction of the sulphur, copper and iron from the concentrate. The progress of the microbial oxidation was followed by analysis of the leach solutions for ferrous, ferric and total iron species, copper, sulphates, pH, Eh and number of the chemolithotrophic microorganisms. Solvent extraction plus electrowinning were used to recover copper from the pregnant solution after bioleaching. The solvent extraction was carried out by means of the reagent LIX 984N (10 volume percents in a kerosene diluents).

The precious metals exposed in the solid residue as a result of the acidic pretreatment of the concentrate were subjected to leaching by means of different solutions for the comparative testing of their leach efficiency:

Solution type No 1: microbial protein hydrolysate consisting of mixture of protein hydrolysates produced from the biomass of three different microbial species with different amino acids composition and mixed in different correlations with pH from 9 – 11 (by NaOH);

Solution type No 2: microbial protein hydrolysate from the type mentioned above but containing also a chemical oxidizer of the precious metals (KMnO₄, NaNO₂ or H₂O₂) used in concentrations from 5 to 10 g/l, at pH from 9 – 11 (by NaOH);

Solution type No 3: thiosulphate in concentrations from 5 to 20 g/l, at pH from 9 – 11;

Solution type No 4: combinations of the microbial protein hydrolysate and thiosulphate at different ratios and in concentrations from 10 to 20 g/l, at pH from 9 – 11;

Solution type No 5: solutions of NaCN in concentrations from 5 to 10 g/l, at pH from 9 – 11 (by NaOH).

The microbial pretreatment and the subsequent chemical leaching of the pretreated concentrate were formed in agitated reactors with a working volume of 500 ml each and a pulp density from 10 to 20 g/l. The duration of these two operations was within 72 to 168 hours for each of them.

Elemental analysis of the liquid samples was carried out by means of atomic adsorption spectrometry and induced coupled plasma spectrometry. Elemental analysis of the solid samples before and after the leaching were carried out by the above-mentioned methods. Control analyses of the gold and silver in the solid samples before and after the leaching were carried out by cupellation (fire assay).

The isolation, identification and enumeration of microorganisms were carried out by methods described elsewhere (Karavaiko et al., 1988; Hallberg and Johnson, 2001; Rawlings and Johnson, 2007).

3. Results and Discussion

The extraction of precious metals from the initial concentrate not subjected to preliminary oxidation was not efficient due to their significant dispersion and encapsulation in the sulphide minerals, mainly in the pyrite. The addition of chemical oxidizers (KMnO₄, NaNO₂ or H₂O₂) to the protein hydrolysate acting as a complexing agent for the precious metals increased to some extent the level of their extraction but even in these cases the effectiveness of leaching was relatively low (Table 2).

Table 2: Leaching of gold and silver from the flotation concentrate by means of different leach solutions.

Leach solutions	Initial concentrate		Concentrate subjected to preliminary oxidation	
	Metals extraction, %			
	Au	Ag	Au	Ag
Thiosulphate	48.2	37.0	92.3	77.0
NaCN	49.5	39.2	93.6	78.5
Protein hydrolysate	14.0	8.6	20.3	14.5
Protein hydrolysate+ chemical oxidizer:				
- KMnO ₄	43.1	32.0	90.5	74.3
- NaNO ₂	36.1	28.2	85.1	69.1
- H ₂ O ₂	32.0	24.2	84.2	64.0

Portions of the precious metals solubilised during the leaching, especially of silver, precipitated in the cases in which the pH of the leach solution was lower than 9. The maintenance of the pH at levels higher than 9 decreased considerably the precipitation of the dissolved precious metals. The most efficient chemical leaching of these metals was performed within pH of about 9.5 – 10.5.

The optimum concentrations of the chemical oxidizers during the leaching were within the limits of about 5 – 10 g/l. These concentrations were sufficient to maintain the Eh of the leach solutions of values higher than 400 mV for a period of about 55 – 60 hours. The addition of the oxidizers in portions during the leaching was more efficient and decreased to some extent the consumption of these reagents. Regardless of this, the consumption of the reagents were high (within 0.5 – 0.8 g/g concentrate). These high consumptions were due to the fact that the oxidizers reacted not only with the precious metals but also with the sulphides in the concentrate and with the amino acids contained in the protein hydrolysate.

The leaching of the concentrate by means of thiosulphate was much more efficient than this by means of protein hydrolysate alone and even from these achieved by means of the protein hydrolysate in combinations with the chemical oxidizers (Table 2). However, the combination of the protein hydrolysate with the thiosulphate was the most efficient and practically equal to the extraction achieved by means of cyanide.

The microbial oxidative pretreatment of the sulphidic concentrate by means of acidophilic moderately thermophilic chemolithotrophic bacteria at 55 – 60 °C or by means of the extremely thermophilic archaea at 86 – 90 °C gave similar results during the subsequent leaching of the precious metals by means of the leach solutions mentioned above. The extraction of these metals was clearly connected with the level of oxidation of the sulphidic minerals containing the precious metals (Table 3). It must be noted, however, that the subsequent leaching of the pretreated concentrate by means of thiosulphate at temperatures highest than 50 °C was connected with a considerable increase of the consumption of this reagent. The further decrease of the content of sulphidic sulphur in the concentrate (to about 0.7 – 0.8 %) by means of the microbial pretreatment had partially no additional effect on the extraction of silver.

Table 3: Effect of microbial oxidative pretreatment of the sulphidic concentrate by means of moderately thermophilic chemolithotrophic bacteria on the subsequent extraction of the precious metals

Content of sulphide sulphur in the concentrate	Levels of the sulphidic sulphur oxidation, %	Extraction of the precious metals, %	
		Au	Ag
4.1	0	30.7	14.5
3.2	24.4	55.4	41.0
2.6	36.6	73.8	59.0
2.1	51.2	92.3	78.6
1.5	63.4	93.0	80.2
1.0	75.6	93.2	80.8

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ADVANCED HYDROGEN STORAGE TECHNOLOGIES

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Abstract: The development of ecological and efficient vehicles powered by hydrogen or electricity is one of the priorities of the automotive industry at the moment. These new propulsion systems should come as a replacement for the internal combustion engines, which use gasoline or diesel fuel and therefore produce toxic emissions. One of the problems, that hydrogen powered vehicles face, is hydrogen storage and that is the main topic of our paper. Conventional technologies for hydrogen storage, like high pressure and cryogenic reservoirs, have many drawbacks in terms of compactness, mass (weight), efficiency and safety. In order to get over these problems, there have been many researches throughout the years in effort to develop new, commercially available technologies. The most advanced storage technologies today, are tanks filled with metal hydrides which absorb hydrogen by forming chemical bonds with it, and tanks filled with large contact surface materials that are able to adsorb hydrogen. In this paper we will present the requirements that these solutions have to fulfill, their current level of development and the expectations for future improvement and usage.

Keywords: HYDROGEN STORAGE, TANKS, METAL HYDRIDES, COMPLEX HYDRIDES, CARBON NANOTUBES

1. Introduction

The development of ecological and efficient vehicles powered by hydrogen or electricity is one of the priorities of the automotive industry at the moment. The idea of using hydrogen as a fuel originates from the 20-th century. It was based on the fact that there are no toxic components in the exhaust gases as a result of hydrogen combustion and it doesn't rely on the usage of fossil fuels. If we take into account that hydrogen must be produced, because there is no free hydrogen in nature, we can conclude that hydrogen is not a power source, but a medium for power transfer.

The development of a safe, reliable, compact and energy efficient technology for hydrogen storage is one of the main obstacles for using hydrogen as a propulsion medium in vehicles. In order to be competitive with the other vehicles available on the market, hydrogen storage reservoirs must provide a range of at least 500 km, as small as possible mass and dimensions, high energy efficiency and the most important of all, safety for the passengers.

2. Conventional storage technologies

Nowadays, there are two prevalent and commercially available technologies for hydrogen storage: high pressure reservoirs and cryogenic reservoirs.

2.1. High pressure reservoirs

In high pressure reservoirs (Fig. 1), hydrogen is stored under extremely high pressure of approximately 700 bar. These tanks are widely available on the market today, but have a drawback in terms of their large mass and dimensions. They are also very inefficient, because a lot of energy is wasted on hydrogen compression.

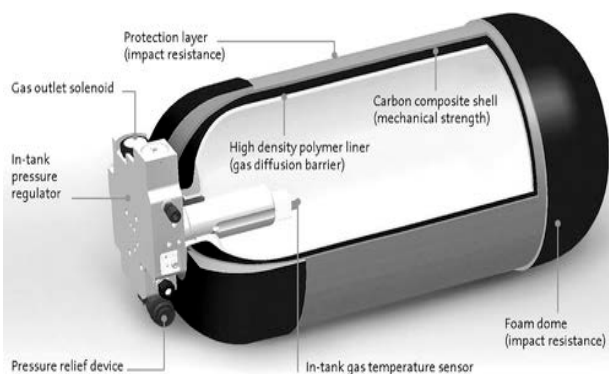


Fig. 1 Section view of a high pressure reservoir

Because of the high pressure they are classified as unsafe. With the increased usage of composite materials, the modern reservoirs have smaller mass and dimensions, but the other problems mentioned remain unsolved.

2.2. Cryogenic reservoirs

Cryogenic reservoirs (Fig. 2) are filled with liquid state hydrogen and are able to store a greater quantity of hydrogen in a smaller volume. However, it requires a lot of energy to bring the hydrogen in a liquified state. Liquid hydrogen has a much greater specific energy than the hydrogen in high pressure reservoirs. The hydrogen must maintain a very low temperature, which means that these reservoirs must have great thermal isolation.

In the last couple of years, hybrid hydrogen reservoirs were made. They allow hydrogen storage under high pressure and at very low temperatures. There reservoirs are more compact, but they also inherit some of the disadvantages of both previously described reservoirs.

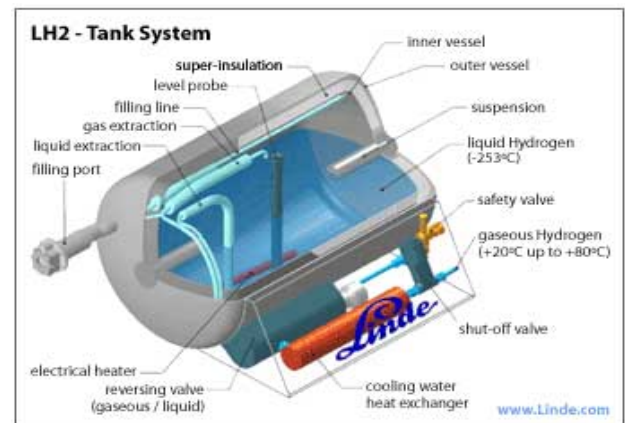


Fig. 2 Section view of a cryogenic reservoir

3. Advanced storage technologies

Because of the major disadvantages of the conventional storage technologies, in the last two decades, there were exhaustive researches, whose goal was development of new, energy efficient and commercially available technologies for hydrogen storage. The most advanced storage technologies today include: storage through absorption in materials (forming chemical bonds between the hydrogen and the material inside the reservoir) and storage through adsorption on the surface of materials (Fig. 3).

3.1. Absorption in materials

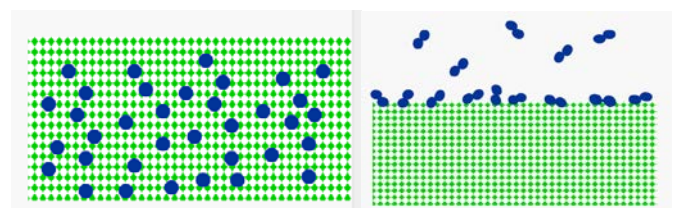


Fig. 3 Mechanism of absorption and adsorption

3.1.1. Metal hydrides

Metal hydrides are compounds formed when a metal element is connected with an atom or molecule of hydrogen. Usually the bond between the elements is covalent, but some hydrides are formed with ionic bonds. Metal hydrides appear in solid state at atmospheric pressure and room temperature. The different types of hydrides, those of biggest interest for hydrogen storage and their structure are mentioned in Table 1.

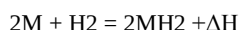
Table 1: Types of metal hydrides and their structure

No.	Typ	Base	Hydrid	Structure
1	AB	TiFe	TiFeH ₂	Cubic
2	AB ₃	CeNi ₃	CeNi ₃ H ₄	Hexagonal
3	AB ₅	LaNi ₅	LaNi ₅ H ₆	Hexagonal
4	AB ₅	TiMn ₂	TiMn ₂ H ₅	Hexagonal or cubic
5	A ₂ B	MgNi	Mg ₂ NiH ₄	Cubic
6	A ₂ B ₇	Y ₂ Ni ₇	Y ₂ Ni ₇ H ₃	Hexagonal

Metal hydrides consist of two different metals. The first one is usually some rare element or alkaline metal, which has the property to bond with the hydrogen and form a stable compound. The other is a so called transitional metal which has the property to form unstable compounds with hydrogen and acts as a catalyst during the phases of hydrogen storage and release.

Metal hydrides which are able to absorb and release hydrogen under conditions such as room temperature and atmospheric pressure, and are able to achieve high values for volumetric and gravimetric density of absorbed hydrogen, can be used as materials in hydrogen storage reservoirs. However, hydrides which are able to absorb hydrogen under atmospheric conditions, are usually composed of transitional metals and have a low gravimetric density. Such an example is LiNi₅H₆, which has a gravimetric density of only 1,15 %wt. An opposite example is TiCr₂H₆, which has a gravimetric density of 2.43 %wt, but needs high temperature to release the stored hydrogen. Recent studies deal with simpler hydrides which are formed from light metals and are easier to produce, such as MgH₂.

The formation of metal hydrides is an exothermal reaction, which means that during the storage phase a certain amount of heat is released. The same amount of heat must be brought back in the system in order for hydrogen to be released. However, if we find a way to absorb the energy released during the formation phase, the same energy can be used later in the release phase. Another way is to use the waste heat from the fuel cell or the ICE.



Where the designations mean the following: M-metal or alloy; MH₂ – metal hydride; ΔH - released heat during the absorption phase. The reaction is reversible and can be used for both hydrogen absorption and release, depending on pressure and temperature.

3.1.2. Complex hydrides

Complex hydrides are inorganic materials that are best described as salts, which are built from complex anions containing hydrogen as terminal ligand, such as the BH₄⁻ (tetrahydroborate or borohydride) or AlH₄⁻ (alanate) anions and counteractions from many different groups in the periodic table. Most of the attention drawn by complex hydrides in the recent past is owed to their large gravimetric hydrogen capacity, i.e. nominal hydrogen-content per weight unit, which makes them competitive candidates for solid state on-board hydrogen storage. This was first suggested in 2001 for LiBH₄. In such materials the hydrogen molecule can be eliminated either by hydrolysis or electrolysis at elevated temperatures.

Although their production was more difficult because of the complicated structure, researchers thought that complex hydrides would be the next phase in the development of hydrogen storage

technologies. However, a problem arose with the slow dynamics of the process of hydrogen release. The process was consisted of a couple of phases for which different conditions were needed. There was a big difference between the theoretical capacity of the reservoirs and the practically achieved capacity. Another problem was the low reliability and the short life span of these reservoirs.

3.2. Adsorption on the surface

Adsorption of hydrogen is based on Van der-Waals interactions between gases and solids (Fig. 4). These interactions are attractive, intermolecular forces which allow hydrogen to bond with solids. Van der-Waals forces originate from charge distribution in atoms and molecules when they get close to each other. Atoms and molecules are attracted by electrostatic forces.

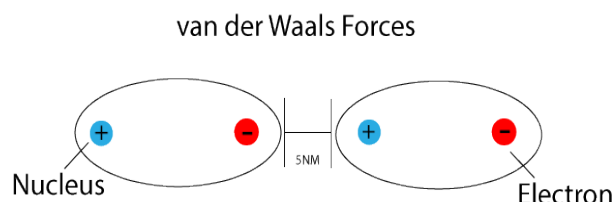


Fig. 4 Van der Waals forces scheme

The balance between attractive and repulsive forces between two molecules causes an energetic minimum of 1-10 kJ/mol and the average energy of adsorbed hydrogen on a carbon surface is 4-5 kJ/mol. This low energy level means a weak bond. That is why hydrogen is released as temperature increases and in high temperature conditions the level of adsorbed hydrogen is very low.

Usually only a single layer of hydrogen is adsorbed on one surface. The maximum amount of adsorbed hydrogen on the surface of a material can be represented as the maximum amount of hydrogen that can be placed in a single layer. In ideal conditions, the minimal surface area needed to adsorb one mol of hydrogen is 85.917 (m²/mol). A single layer of graphite has surface area of 1315 (m²/g) so the maximum amount of hydrogen it can adsorb is 3% wt. Larger quantity of hydrogen can only be adsorbed in low temperature conditions.

3.2.1. Carbon nanotubes as hydrogen storage materials

Porous carbon is considered to be good adsorbent, due to its large contact surface. Adsorption capabilities of normal porous carbon are proportional to its contact surface and pore volume, but it can achieve good gravimetric density (4-6 %wt) only at extremely low temperatures.

Unlike porous carbon, carbon nanotubes have a smaller contact surface and pore volume, but they show remarkably good hydrogen adsorption. At high pressure (10MPa) hydrogen atoms form a dense, compact structure. Using mathematical models, it is shown that at high pressure, a single layer of graphite can adsorb up to 4.1% wt of hydrogen. Single-walled nanotubes (SWNT) can adsorb up to 4% wt of hydrogen (Fig. 5).

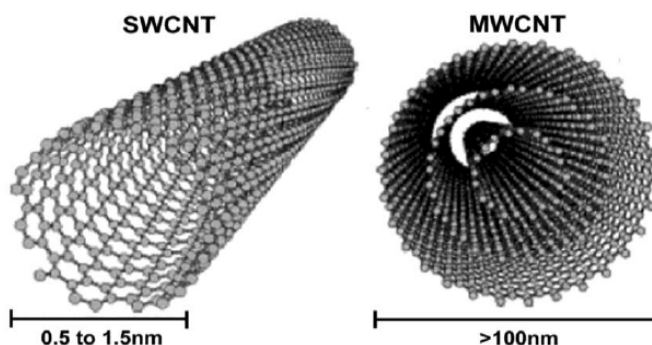


Fig. 5 Single and multi walled carbon nanotubes

Multi-walled nanotubes (MWNT) are nested, concentric, cylindrical layers of graphite with hollow centre and they show great capability for hydrogen adsorption. Catalysts are used to increase the amount of adsorbed hydrogen. Researches show that MWNT activated with KOH (potassium hydroxide) can adsorb up to 4,7% wt hydrogen, whereas ordinary MWNT can adsorb only 0,7% wt.

4. Energy efficiency comparison

In order to obtain a reasonable estimation of the energy required for hydrogen storage with the three different technologies, we must calculate the energy required for compression, cooling and heating. We must also take into account the efficiency of the generator and compressor, and the efficiency of the cooling cycle.

To calculate the energy required to store hydrogen in a high-pressure reservoir one may use the adiabatic compression equation Compressing hydrogen to 680 bar requires 29 MJ/kg. If we add the efficiency of the compressor and generator, assuming that their combined efficiency is 60%, we obtain that compressing hydrogen to 680 bar requires 47 MJ/kg. These correspond to 30 and 40% of the low heat value (LHV) of hydrogen. Storing hydrogen as a compressed gas is quite energy intensive.

To store hydrogen as a liquid, the energy required to cool hydrogen to the liquid state is critical. Theoretical heat to cool hydrogen from 25 °C to 20 K and condense it to liquid is about 3.4 MJ/kg (2.94 sensible and 0.45 condensation). But the actual required energy is much higher due to the inefficiency of refrigeration at extremely low temperature. The minimum required energy may be calculated using an ideal refrigeration cycle, the reversed Carnot cycle. The efficiency of this cycle depends on the temperatures at which heat is added and rejected, and is equal to $T_1/(T_2-T_1)$, where T_1 =evaporator temperature=20 K, T_2 =condensing temperature=298 K. Therefore the efficiency in our case is $20/(298-20)=7.2\%$. To generate 3.4 MJ to liquefy 1 kg of hydrogen will require 47 MJ ($3.4/0.072=47$). This is 39.2% of the LHV. The actual required energy can be significantly more because the refrigeration cycle will be less than ideal. Heat will also be required to evaporate the liquid and warm up the gas before feeding it to the energy conversion device such as a fuel cell. This may require an additional of 3.4 MJ/kg. This makes the total required energy to store hydrogen as a liquid to be 50.4 MJ/kg. This is 42% of the low heat value of hydrogen. Liquid hydrogen storage is as energy intensive as the compressed hydrogen storage.

Using metal hydride as hydrogen storage requires an absorption step and a desorption step. A typical absorption step requires a supply of 20 bar hydrogen and the removal of about 14.6 MJ/kg heat of absorption. The energy needed to compress hydrogen to 20 bar is about 12 MJ/kg (10% of LHV). The heat of absorption is removed by coolant at temperatures of about 10 °C. The energy required for cooling will be obtained from the adiabatic compression equation and is $14.6/5=3$ MJ/kg. That is $3/120=2.5\%$ of the LHV. Similar amount of heat would be needed to desorb the hydrogen, but this can be provided by the waste heat of the energy conversion device (fuel cell or internal combustion engine) and is practically free. The total energy required to operate a metal hydride storage system is therefore about 15 MJ/kg, or about 12.5% of LHV. This is the lowest operating energy compared to those of compressed hydrogen and liquid hydrogen. Therefore, metal hydride for hydrogen storage has the advantage of low operating energy, moderate pressure and temperature, and high volumetric density.

The analysis performed shows that storing hydrogen in metal-hydride reservoirs improves the energy efficiency 1.45 times in relation to high-pressure reservoirs and 1.5 times in relation to cryogenic reservoirs.

5. Current situation

The properties of modern metal hydrides such as efficiency, volumetric and gravimetric density and the pressure and

temperature at which hydrogen is stored, can be obtained from table 6.1. The newest materials, which are currently tested, are MgH_2 , $LiNi_5H_6$ and $LiAlH_4$. Metal hydride tanks are commercially available but they are not used in vehicles. The most advanced tanks that are used in commercial vehicles are composite tanks made of carbon fibre, where hydrogen is stored under high pressure.

Researchers in Germany in 2015 have created a 3D model which shows the optimal structure that nanotubes should form. This is a mathematical model and theoretically, a tank with this structure should adsorb 5.5%wt hydrogen at room temperature. Nanotubes are placed parallel to each other and they don't intersect, but they have a large contact surface. This structure allows great storage capabilities. An ideal structure like this is nearly impossible to achieve with the current technology and that is the reason why no carbon nanotube tanks have been made yet.

Table 6.1. Properties of different metal-hydrides

Typ	Base	Temp. (K)	Pressure (bar)	Wt (%)	Density (g/cm ³)	Price (\$/kg)
AB	TiFe	265	4.1	1.86	6.5	4.68
AB2	TiCr2	182	18.2	2.43	6	8.64
AB2	TiMn2	252	8.4	1.86	6.4	5
AB5	MmNi5	217	23	1.46	8.6	7.94
AB5	LaNi5	285	1.8	1.49	8.3	9.87

Table 6.2. Properties of different carbon nanotubes

Material	Wt (%)	Temperature (K)	Pressure (bar)
SWNT (low purity)	3.5-4.5	298	0.4
SWNT (high purity)	5-10	273	0.4
Li-MWNT	20	473	0.1
K-MWNT	14	300	0.1
GNFs	5	300	10.1
Nano-graphite	7.4	300	1

6. Conclusion

The research carried out in this paper presents a series of findings that are connected with the possibilities for more intensive application of hydrogen powered vehicles. Hydrogen storage is one of the basic problems for its commercialisation as a fuel for vehicles. Conventional storage technologies have limited application due to their safety and performance deficiencies. Metal hydrides and carbon nanotubes are imposed as alternative technologies with the ability to overcome these problems.

During the last two decades, a number of research and experiments have been carried out, which have enabled the improvement of the properties of metal hydrides. With further improvement in terms of reliability and cost of production, it is possible to use them commercially in the near future. Carbon nanotubes show even greater potential for hydrogen storage. The research that is ongoing in this direction is still theoretical due to the technological limitations associated with the possibilities of achieving precise nanostructure.

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TECHNOLOGICAL LINE CONTROL VERIFIED ON HIL PLATFORM

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Abstract:

The paper deals with the design and experimental verification of a new control structure with reference model for the tension control in the technological processing line the stability of which is derived on basis of the second Lyapunov method. The properties of the proposed controller were verified by experimental measurement on a new concept of hardware-in-the-loop (HIL) simulation platform based on programmable logic controllers (PLC). The experimental measurements confirmed that the proposed control structure is robust over a wide range of controlled system significant parameters changes together with invariance and desired dynamics prescribed by the reference model.

KEYWORDS: CONTINUOUS LINE, TENSION CONTROL, ROBUST CONTROL, HIL SIMULATION

1 Introduction

Technological processing line represents a multi-motor drive system in which the individual drives are mechanically coupled through the web of material. The tension control quality in this technological line significantly affects the quality of the line final product in many web processing industries (textile, paper, metal, polymer). Web tension control systems can be regarded as a nonlinear multivariable system. Due to the strong coupling between web velocity and web tension during normal operation cycle there exist many sources of disturbances e.g. web sliding, vibrations, change of web-elasticity. These disturbances are transmitted to the web tension that can result in bad quality or even in the destruction of the material being processed.. For this reason the control strategy for continuous technological lines should be robust with respect to the uncertainties and disturbances [1].

Web tension regulation is a rather challenging industrial control problem. A review of the web tension control problems can be found in [2]. Proportional-integral-derivative (PID) [3, 4], fuzzy logic [5, 6], neural network [7, 8], optimal control [9, 10], and robust control approaches [11, 12] are used. Recently, robust Lyapunov-based feedback control [13, 14] and multivariable H_{∞} controller with one or two degrees of freedom control strategies have been proposed for industrial web transport systems [15].

Regarding control structure, centralized control structure has many drawbacks and researchers have proposed decentralized control [16-18] and overlapping decentralized control [19] to improve the performance over centralized control structure.

The increasing requirement on tension control performance in technological lines requires the use of more better solutions and sophisticated control approaches. Due to the fact that multi-motor drives occur in practice as parts of larger technological assemblies, it is necessary to look for such methods of their control that would be simply and easily physically interpretable. Otherwise their wide application in industrial practice cannot be expected.

Therefore our aim was to design a simple controller that would ensure the required value of tension in the web independent from its speed. The basic idea of the control method lies in extending the system by a new suitable state variable which represents additional information into the controlled system easily obtained from the system's output variable. The proposed controller then secures asymptotic stability of the extended system and also zero control deviation based on the application of the Lyapunov synthesis.

The properties of the proposed controller were verified by experimental measurement on a new concept of hardware-in-the-loop (HIL) simulation platform based on generally available hardware (PLC) and software means (MATLAB).

2 Preconditions and means for resolving the problem

2.1 Technological line structure

The middle part of the continuous line is a typical representative of technological line for processing continuous flows of material (e.g. sheet metal strips, tubes, processing lines in paper mills and printing

works, etc.) in the field of elastic or plastic deformation, which influences the material's mechanical properties. Figure 1 shows the structure of the middle part of a continuous line (CL).

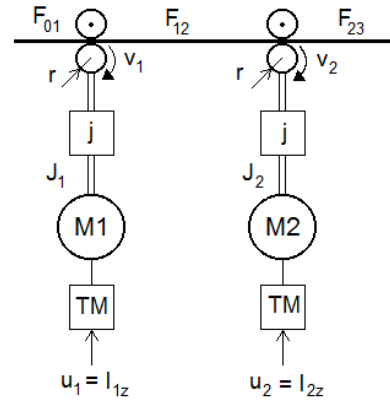


Fig. 1 Structure diagram of middle section of continuous line

The legend for the diagram in Fig. 1 is as follows:

F_{01} , F_{23} - tension before 1st machine, tension before 2nd machine,
 v_1 , v_2 - 1st machine rolls velocity, 2nd machine rolls velocity,
 F_{12} - tension between 1st and 2nd machine,
 r - roll radius, j - gearbox, M_1, M_2 - DC motors,
 J_1 , J_2 - moments of inertia of rolls, TM - static converter,
 I_{2z} - desired current value of motor M2,
 I_{1z} - desired current value of motor M1.

Assume that the converter has proportional transfer and an integrated subordinated current loop the time constant of which is substantially shorter compared to the time constants of the remaining subsystems of the CL middle section, so it can be ignored.

The block diagram of CL middle section under this (practically real) assumption is shown in Fig. 2. The parameters of the CL model are specified in the Appendix.

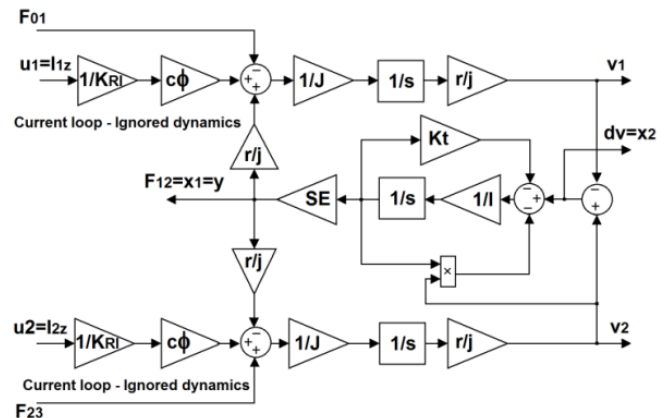


Fig. 2 Continuous line block diagram

In terms of control this is a 3rd order MIMO system with two inputs [$u_1=I_{1z}$, $u_2=I_{2z}$] and two outputs [F_{12} , v_2], which can be influenced by two additive disturbances [F_{01} , F_{23}].

2.2 Technological line tension control

The precise tension control is the main goal of control in such systems. The tension should have a constant and non-oscillating course in all operation stages (start-up, deceleration) under consideration the influence of disturbances before and after the section of the line as well as of changes of parameters (e.g. the moment of inertia).

In terms of tension control the controlled system will be a 2nd order system with two inputs [$u_1=I_{1z}$, $u_2=I_{2z}$] and one output $y=F_{12}$. The first state variable will be the tension ($F_{12}=x_1$) and the second state variable is the difference of the circumferential velocities of the rolls ($dv=x_2=v_2-v_1$), where velocity v_2 is considered as an additional disturbance.

The state description of the middle section of the CL in terms of tension control according to Fig. 2 has the form

$$\begin{bmatrix} \frac{dx_1}{dt} \\ \frac{dx_2}{dt} \end{bmatrix} = \begin{bmatrix} -\left(\frac{K_t + v_2}{l}\right) \frac{SE}{l} \\ -\frac{2r^2}{j^2 J} \end{bmatrix} \begin{bmatrix} x_1 \\ x_2 \end{bmatrix} + \begin{bmatrix} 0 \\ -\frac{rc\phi}{jjK_{RI}} \end{bmatrix} u_1 + \begin{bmatrix} 0 \\ \frac{F_{01}r}{jj} + \frac{F_{23}r}{jj} + \frac{rc\phi}{jjK_{RI}} u_2 \end{bmatrix} \quad (1)$$

The dynamic properties of tension in the middle section of the continuous line will be prescribed by a reference model. As for the 2nd order controlled system we need to obtain additional information on the unknown parametrical and also additive disturbances, we will extend its reference model by one state variable $x_{3M} = x_{eM}$, which will secure that in steady state the tension control disturbance is equal to zero. The reference model will be a 3rd order linear system the dynamics of which can be set according to [20] by a single optional positive parameter α . This reference model can secure optimal dynamical properties of the controlled system in terms of the minimal control disturbance and minimal input power criterion [20]. If we need to settle the tension in the web of material within approximately 1s, then based on the Shannon-Kotelnikov theorem the value of the parameter will be $\alpha=5$. Then the extended state description of the reference model for the 2nd order system is

$$\begin{bmatrix} \frac{dx_{eM}}{dt} \\ \frac{dx_{1M}}{dt} \\ \frac{dx_{2M}}{dt} \end{bmatrix} = \begin{bmatrix} 0 & 1 & 0 \\ 0 & 0 & 1 \\ -\frac{\alpha^3}{2} & -\frac{3\alpha^2}{2} & -\frac{3\alpha}{2} \end{bmatrix} \begin{bmatrix} x_{eM} \\ x_{1M} \\ x_{2M} \end{bmatrix} + \begin{bmatrix} -1 \\ 0 \\ 0 \end{bmatrix} w \quad (2)$$

Note: The tension is generated by the first drive in opposite direction to the line speed, which is represented by the work rolls circumferential velocity v_2 of the second drive.

The block diagram of the considered reference model is shown in Fig.3.

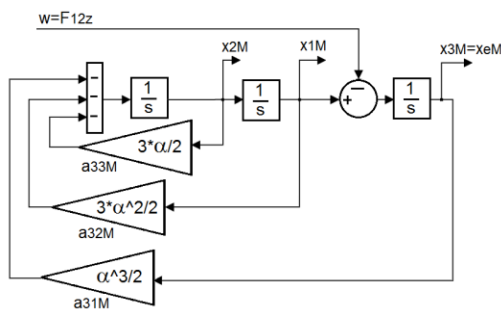


Fig. 3 Reference model extended by an additional state variable

It is clear in Fig. 3 that in steady state the input of integrator x_{3M} will equal zero and tension will equal the desired value w .

In terms of CL middle section tension control this is a 2nd order controlled system (eq.1) which similarly to the reference model will be extended by an additional state variable x_e . Its state description is as follows

$$\begin{bmatrix} \frac{dx_e}{dt} \\ \frac{dx_1}{dt} \\ \frac{dx_2}{dt} \end{bmatrix} = \begin{bmatrix} 0 & \frac{1}{l} & 0 \\ -\left(\frac{K_t + v_2}{l}\right) & \frac{SE}{l} & 0 \\ -\frac{2r^2}{j^2 J} & 0 & 0 \end{bmatrix} \begin{bmatrix} x_e \\ x_1 \\ x_2 \end{bmatrix} + \begin{bmatrix} 0 \\ 0 \\ -\frac{rc\phi}{jjK_{RI}} \end{bmatrix} u_1 + \begin{bmatrix} 0 \\ 0 \\ \frac{F_{01}r}{jj} + \frac{F_{23}r}{jj} + \frac{rc\phi}{jjK_{RI}} u_2 \end{bmatrix} \quad (3)$$

If we establish the reference model and system deviation as

$$\mathbf{e} = \mathbf{x}_M - \mathbf{x} \quad (4)$$

then through simple modifications we will obtain a system whose states are represented by state deviation $\mathbf{e}$

$$\frac{d\mathbf{e}}{dt} = \mathbf{A}_M \mathbf{e} + \mathbf{f} - \mathbf{B} \mathbf{u} \quad (5)$$

where we have denoted

$$\mathbf{f} = (\mathbf{A}_M - \mathbf{A})\mathbf{x} + \mathbf{B}_M w - \mathbf{d} \quad (6)$$

then $\mathbf{f}$ is the generalized disturbance vector which comprises all parametrical and additive disturbances affecting the system with regard to its reference model.

If we choose the Lyapunov function for system (5) as

$$V = \mathbf{e}^T \mathbf{P} \mathbf{e} = \mathbf{e}^T \mathbf{z} \quad (7)$$

where $\mathbf{z}$ is the weighted state deviation vector

$$\mathbf{z} = \mathbf{P} \mathbf{e} \quad (8)$$

and $\mathbf{P}$ is a positive definite matrix which elements according to [20] have form

$$\mathbf{P} = \begin{bmatrix} \frac{\alpha^5}{2} & \frac{\alpha^4}{2} & \frac{\alpha^3}{2} \\ \frac{\alpha^4}{2} & \frac{5\alpha^2}{2} & \frac{3\alpha^2}{2} \\ \frac{3\alpha^2}{2} & \frac{3\alpha^2}{2} & \frac{3\alpha}{2} \end{bmatrix} \quad (9)$$

The derivation of Lyapunov function (7) with regard to system (5) while it applies that $\mathbf{Q} = -\alpha \mathbf{P}$ is

$$\frac{dV}{dt} = -\mathbf{e}^T \mathbf{Q} \mathbf{e} + 2(\mathbf{f}^T \mathbf{z} + b_2 \mathbf{u}) = -\alpha \mathbf{e}^T \mathbf{P} \mathbf{e} + 2(\mathbf{f}^T \mathbf{z} + b_2 \mathbf{u}) = -\alpha \mathbf{e}^T \mathbf{z} + 2(\mathbf{f}^T \mathbf{z} + b_2 \mathbf{u}) \quad (10)$$

The first element of equation (10) is always negative, as the expression $\mathbf{e}^T \mathbf{z} = \mathbf{e}^T \mathbf{P} \mathbf{e}$, where $\mathbf{z} = \mathbf{P} \mathbf{e}$ is always positive. System (5) will then be asymptotically stable, i.e. its derivation will be negative, if for input u the following will apply

$$\mathbf{u} = -\mathbf{K} \mathbf{e}^T \mathbf{z} \quad (11)$$

Note: The second element of equation (10) will be negative, if $b_2 \mathbf{u} = -b_2 \mathbf{K} \mathbf{e}^T \mathbf{z}$ will be larger than $\mathbf{f}^T \mathbf{z}$, which can always be assured by a sufficiently large value of optional positive parameter K .

For generating state quantities deviations of the reference model and the system according to equation (4) the following modification can be applied

$$e_3 = x_{eM} - x_e = \int x_{1M} dt - \int x_1 dt = \int (x_{1M} - x_1) dt = \int e_1 dt \quad (12)$$

The resulting block diagram of CL middle section tension control is shown in Fig. 4.

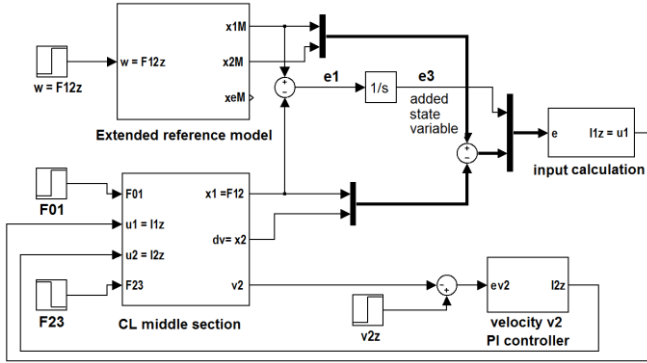


Fig. 4 Block diagram of CL tension control

2.3 Experimental verification on HIL platform

The properties of the proposed controller were verified by experimental measurement on a new concept of hardware-in-the-loop (HIL) simulation platform (Fig.5).

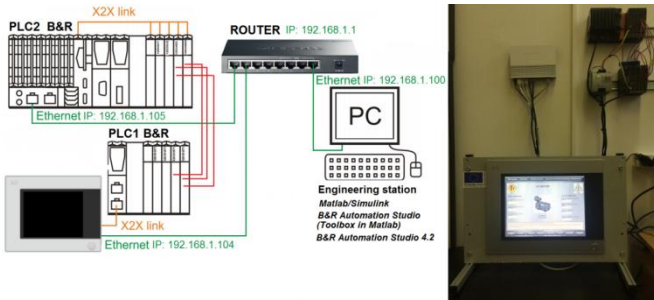


Fig. 5 HIL simulation workplace

The workplace consists of two standard PLCs and one standard PC. Communication between these devices is via standardized Ethernet interface. The first PLC, equipped with an operator display, is used for modelling the controlled technological process. The operator display enables independent animation of the process model and also external modelling of various process situations (change of drive load torque). The second PLC is used as the controller member of the system and enables testing of various controller options. Interface of control and information signals between the controller and the process model is implemented by standardized electric signals.

A standard PC with Windows operating system enables programming of both the control PLC and the PLC for technological process simulation. For this purpose it has to be provided with a programme package from the manufacturer of the PLCs being used, and possibly also software for dynamic systems modelling. As in this particular case PLCs from B@R were used, for their programming the programme package *B@R Automation Studio 4.2* was installed on the PC. The *Matlab* package with the relevant tools was installed for modelling purposes.

The whole process of implementation of the specified task in the HIL simulation workplace is executed in four basic steps.

In the *first step*, the CL control (Fig. 4) was computer modelled in *Matlab* environment using *Simulink*.

In the *second step* it is necessary to build a hardware configuration

for the process simulation PLC1 (PP500) and for the control PLC2 (X20 CP 1484-1).

In the *third step* it is necessary to transform from *Matlab* the CL model (Fig. 2) and its reference model into the source code for PLC1 for process simulation and the controller structures (Fig. 4) into the source code for the control PLC2. It was implemented by using the *B&R Automation Studio toolbox* in *Matlab* for automatic generation of the code for PLCs according to the *Simulink* block diagram created in the first step.

In the *fourth step* we verify the correctness of the tension controller design.

3 Results and discussion

The dynamical performance of the output controlled variable of tension F_{12} for the operation cycle which includes three stages – start-up, line running at constant operational speed, line run finish. for the value of parameter $K=2$ is illustrated in Fig. 6. It is evident that tension in the middle section of the continuous line practically follows the tension prescribed by the reference model during the whole operation cycle, and it does so even when influenced by step disturbances at its input and output at time $t=3s$ and $t=6s$ and also under the influence of all changes of speed (Fig. 7), which fact verifies the invariance of the proposed control against additive disturbances.

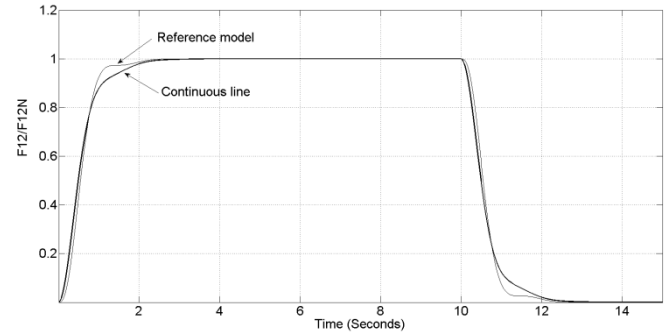


Fig. 6 Time course of tension control of CL

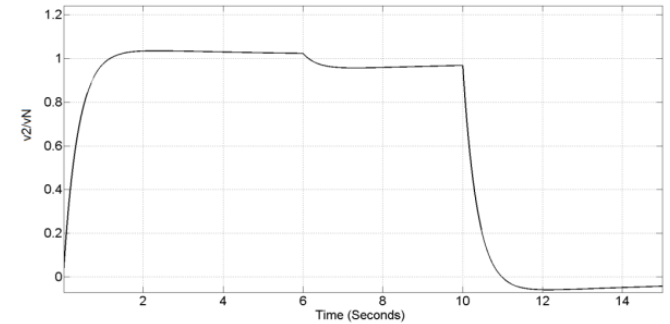


Fig. 7 Time course of velocity v_2 control of CL

The robustness of the proposed control structure was verified for changes in the two most important controlled system parameters that significantly influence the properties of flexible coupling, namely damping of the material being processed (material elasticity), and the moment of inertia of the drives (pulling thicker sheet metal, material weld). Figures 8 and 9 show the time plots of tension F_{12} for as much as fivefold reduced material damping (i.e. five times more elastic material) and for a twofold increase of the drives' moment of inertia (parameter $K=2$).

It is clear from the quoted figures that the dynamics, autonomy and invariance of tension control practically did not change with the significant change of the parameters considered, which fact points at the strong robustness of the proposed controller.

Proposed control structure is very simple, and there is no need for a mathematical model of the system for the calculation of the controller parameters. The dynamical properties of tension setting

are in general prescribed by a linear reference model in state space (Eq. 2), and stability is secured by the elements of positive definite matrix $\mathbf{P}$ (eq.9) and by choosing a positive parameter K .

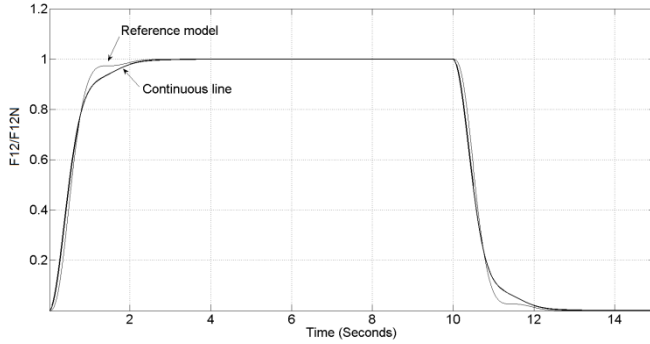


Fig. 8 Time plot of CL tension control for $K_t=0.2 \cdot K_{IN}$ and $J=2 \cdot J_N$

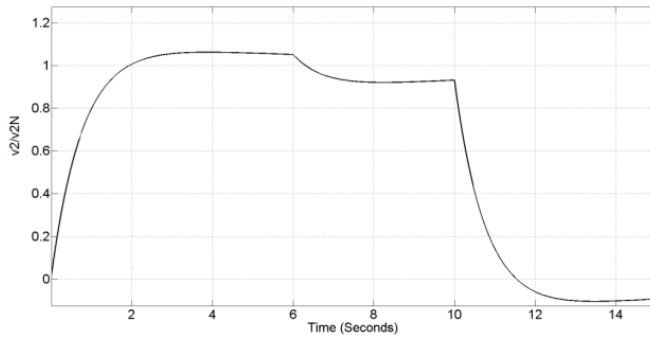


Fig. 9 Time plot of CL velocity v_2 control for $K_t=0.2 \cdot K_{IN}$ and $J=2 \cdot J_N$

Controller synthesis requires data about all state quantities of the controlled system, which is the main disadvantage of the proposed stable control structure. However, the state quantities can be obtained either by measurement or from various types of observers.

4 Conclusion

The paper deals with the design and verification of a new stable control structure with reference model for the control of tension in the middle section of a continuous processing line.

Because the speed and tension subsystems of the line were considered as independent systems and the coupling between them was regarded as a disturbance the control of the continuous line output quantities (tension and speed) is decentralized. The controllers for each subsystems were then designed independently.

The properties of the proposed control structure were verified by experimental measurement on HIL platform based on PLC's. The main advantage of the proposed control structure is its strong robustness over a wide range of changes of significant parameters of the controlled system, together with other important objectives of CL control, i.e. invariance and desired dynamics prescribed by the reference model both in transient and in steady states. What is more, the Lyapunov synthesis tools guarantee the stability of the whole controlled system.

The proposed control strategy is very simple and therefore it is possible to assume its wide application in industrial practice.

APPENDIX

Parameters of CL used for simulation:

DC motors:

$U_N = 24 \text{ V}$	$n_N = 3650 \text{ rpm/s}$	$R_A = 0.7 \Omega$	$I_N = 8.5 \text{ A}$
$P_N = 140 \text{ W}$	$J = 0.002 \text{ kgm}^2$	$M_N = 0.37 \text{ Nm}$	$L_A = 0.1 \text{ mH}$
$j = 24$	$c\phi = 0.043 \text{ Vs}$	$v_{\text{MAX}} = 0.6 \text{ ms}^{-1}$	$I_{\text{MAX}} = 20 \text{ A}$
$F_{12N} = 25 \text{ N}$	$K_{RI} = 1 \text{ V/A (current sensor)}$		

Processed material:

$$b=0.03 \text{ m}, h=0.1 \cdot 10^{-3} \text{ m}, S=bh=3 \cdot 10^{-6}, E=1.8 \cdot 10^9 \text{ Nm}^{-2}, SE=5400 \text{ N}, T_{12}=2.25 \text{ s}, l_{12}=1.35 \text{ m}, K_{IN} = \frac{Jj^2}{T_{12}2r^2} = 160 \text{ kgs}^{-1}$$

Work rolls: $r=0.04 \text{ m}$ Reference model: $\alpha=5$

Parameters of PI controller of velocity v_2 : $K_P=20, K_I=2$

ACKNOWLEDGMENT

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FUNCTIONAL AREAS AND ARCHITECTURAL FRAMEWORK FOR THE MANAGEMENT OF THE ELECTRON BEAM WELDING INSTALLATION

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Abstract: Management is the process of processing the information aiming decision making to resolve problem or to achieve the target. High advanced automated processes, including the vacuum system with pump and pressure control, management of the cooling system, management of the manipulator, high voltage control and emission current, the control of the electron beam movement and its characteristics, as well as computer-based automatic control of the beam power distribution, must be integrated on the basis of the use of systems for operational management of production. The present work presents the hierarchical structure and architectural framework for the management of an installation for electron beam welding, evaporation and surface modification.

Keywords: AUTOMATIC CONTROL, INFORMATION, HIERARCHICAL STRUCTURE, ARCHITECTURAL FRAMEWORK, CONCEPTUAL MODEL

1. Introduction

Management is a process of processing the information to make a decision in order to solve a problem or to achieve the target. Information systems are intellectual means of reducing the risk of decision making in the management of a selected real process.

For the modeling of objects and processes in the software industry, a standardized approach should be used to ensure the integrity and expediency of the models used. Standards provide a formal model for data exchange between business systems and manufacturing systems. The models define the management of production operations, the activities of an enterprise that take production schedules and perform the actual work required to produce products and ensure production visibility [1].

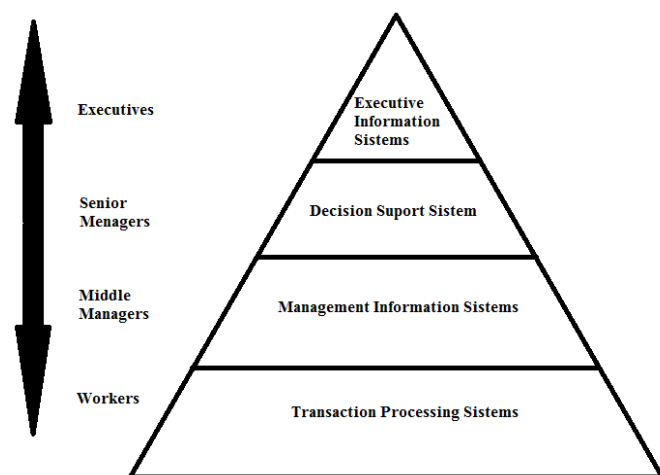


Fig.1. Basic groups of information systems

According to the hierarchical level of management and the used information technologies, the following main groups of information systems are differentiated (Fig.1):

- systems that automate office operations and collaborative work - serve all levels of government and ensure the integration of newly acquired knowledge in business;
- systems handling transactions (TPS – Transaction Processing Systems) – serve the operational management level;
- management information systems (Management Information Systems)– serve the tactical level of management;
- systems, decision support (Decision Support Systems) – serve the tactical level of management;

- systems supporting the Executive Director (Executive Support Systems)- serve the strategic level of management.

2. Models, hierarchical structure and conceptual ontology meta model of EBW

The hierarchical structure according to IEC62264-3 standard, which divides production into different stages of its management. The standard defines the activities that occur in the management of production operations [1].

The choice of a model for designing and developing the specific system or functionality is subjective, each company has to find or build its own approach (method) in developing their projects, while trying to exploit the strengths and abilities of each member of the company [2].

Storage modules can be dedicated to a particular material, group of materials or storage.

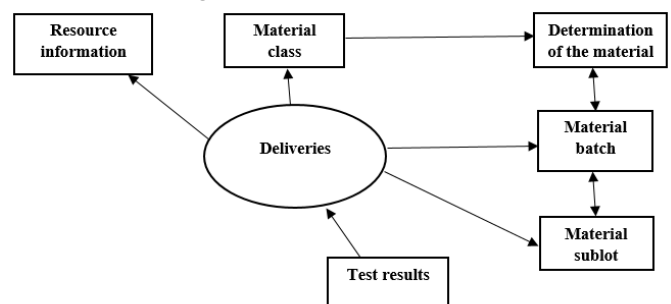


Fig. 2. Model Deliveries

For the Model Deliveries (Fig. 2) the methods for inserting a new record, editing an existing record, adding a record as a line to another object, for changing an existing row in another object. The methods for adding and editing lines in another object can be used for objects in the classes "Resource Information" and "Material Definition". The same methods have been implemented for class objects "Batch of Materials". The methods implemented in the "Batch Materials" and "Subtotal Materials" classes respectively are to create a new record, edit an existing record, and delete a record.

The conceptual model (Fig.3) is a recommended model representing a set of things (artifacts) through a set of concepts, the relationships between them and constraints on the assumption of a "closed environment". Highly sophisticated automated processes, including the vacuum system with pump and pressure control, management of the cooling system, management of the manipulator, high voltage control and emission current, the control of the electron beam movement and its characteristics, as well as computer-based automatic control of the beam power distribution, must be integrated

on the basis of the use of systems for operational management of production.

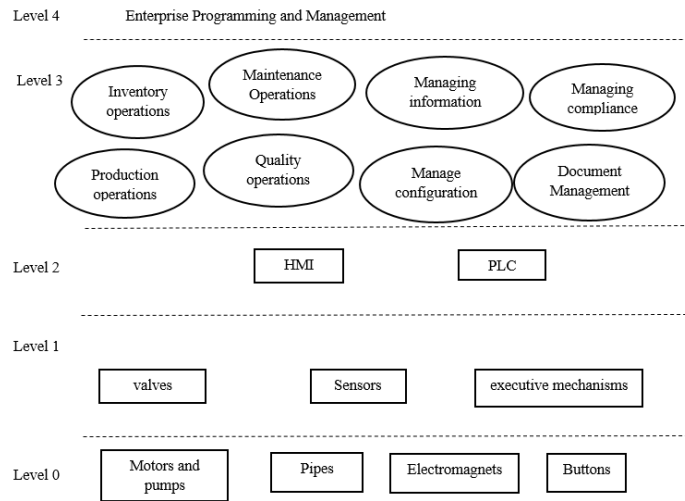


Fig. 3. Conceptual model

In Fig. 3 conceptual model for electron beam welding plant is presented:

Level 0 - Production - at this level are included all field execution devices that serve to directly collect information and manage the processes of starting and stopping available pumps, cranes and valves.

Levels 1-2 - Production Control and Automation - at these levels control and visibility of the process takes place;

- Level 1 receives sensor data, which also serves as a feedback for the vacuum pressure obtained in the vacuum chamber.

- To Level 2 is connected PLC and HMI, which serve as a connection between management and executive devices and for direct management and process change.

Level 3 - Management of production - the maintenance of production information is centralized to provide greater control and

availability of records. The conceptual model is a recommended model representing a set of things (artifacts) through a set of concepts, links between them, and constraints on the assumption of a "closed environment". The highly sophisticated automated processes, including the vacuum system with pump and pressure management, cooling system control, handler control, high voltage and emission control, electronic beam movement control and its features, as well as computer based automatic distribution control of the power of the beam must be integrated based on the use of systems for operational management of production. Level 3 is being implemented:

- Inventory operations.
- manufacturing operations;
- maintenance operations;
- quality operations;
- information management;
- configuration management;
- compliance management;
- document management.

Level 4 - Enterprise Programming and Management - includes designing, deploying, controlling, analyzing, and optimizing, combining management methods and IT tools techniques.

The conceptual ontological meta-model is a description of a number of objects and concepts, knowledge about them and links between them, and ontology is called the explicit specification of conceptualization and consists of terms organized in taxonomy, their definitions and attributes, as well as the related their axioms and rules of inference. Concept means categorization. Object-independent ontologies containing top-level categories (concepts) such as space, time, object, event, action, quantity, measure [3,4]. It can be represented by an architectural frame (Fig. 4).

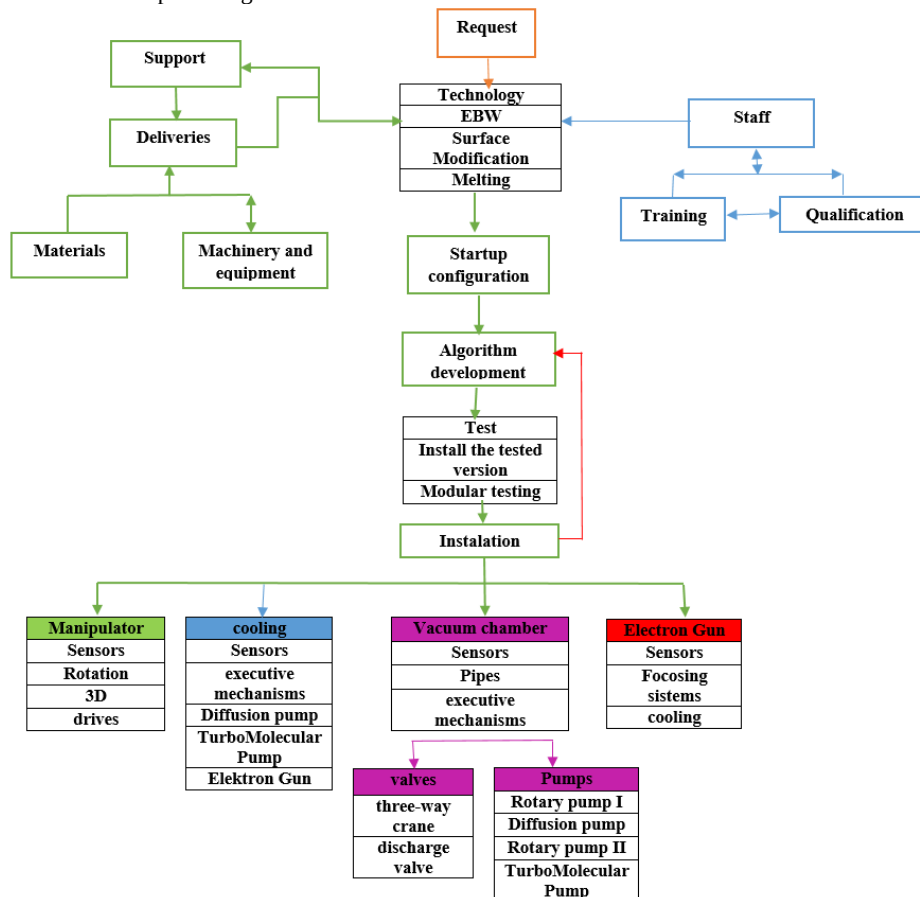


Fig. 4. Architectural framework Installation for Electron Beam Welding

Fig. 4 presents the electron beam welding installation's algorithm with the interconnections between the different models. The technology may be included depending on the request (EBW, Surface Modification, Melting). From the Deliveries model the required materials are obtained. The Personnel model is implemented to identify and select the required qualified staff. A record of the staff working hours can be made and the algorithm is followed to work on request.

3. Conclusion

In the present work is presented an integrated system for management of the electron beam welding process based on a conceptual system, including other additional processes: surface modification, electron beam evaporation, melting and electron beam diagnostics. The main objective is to integrate and organize the knowledge of the process and to present the possibilities for improvement, modeling and control of the technological processes in the installations for the electron beam processes.

Acknowledgements

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THE RESEARCH PROCESS OF ACTIVE DIELECTRICS: SYSTEM INTERCONNECTIONS

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Abstract: Ferroelectric thin films are widely used in devices of micro- and nanoelectronics. In particular, their unique properties (high values of relative permittivity, hysteresis loop, close to rectangular) are connected with the presence of a phase transition leading to a change of the functional dependencies form between the material parameters at temperatures below and above the phase transition temperature. Depending on the purposes of materials research consisting in the analysis of their properties or the materials synthesis with specified characteristics, two tasks can be solved: materials analysis ; controlled synthesis of a material with specified properties. The authors considered the generalized structure of the study process of ferroelectric materials; the interconnection of the parameters of active dielectrics was analyzed. It made it possible to optimize the process of studying the materials used in elements of functional electronics.

Keywords: ACTIVE DIELECTRIC, PROPERTIES RESEARCH, FUNCTIONAL ELECTRONICS

Ferroelectric thin films find wide application in devices of micro- and nanoelectronics. In particular, their unique properties (high values of relative permittivity, hysteresis loop, close to rectangular) are connected with the presence of a phase transition, leading to changes in functional dependencies between the parameters of the material at temperature below and above the phase transition temperature. Depending on the purposes of materials research consisting in the analysis of their properties or the materials synthesis with specified characteristics, two tasks can be solved:

a) the task of analysis of materials with specific properties, subject to experimental determination. A special case of the problem is the study of a variety of materials to select those properties which have desirable quantitative characteristics;

b) the task of synthesis of materials with the specific properties. The peculiarity of this task is the need to not only measure material properties, but also study ways of their correction under the influence of influencing factors. In turn, the allocation of these tasks leads to the distinction of morphological structures, that is, the relationship of the individual components within the subsystems, at the same time some relations are common in the solution of both objectives, which can be shown using the so-called "tree of problems," used in research organizations structuring [1].

The generalized structure of study process of ferroelectric materials in the solution of the problem of synthesis of materials with specified properties is shown in detail in this article (Fig. 1).

Fig. 1 shows a structure applicable to the study process of the ferroelectrics properties, in general case, the task is the establishment of multiple functions $\{y_1, y_2, \dots, y_j, \dots, y_k\} \in Y$ of independent variables $\{x_1, x_2, \dots, x_i, \dots, x_n\} \in X$ such that:

$$y_j = f_j(x_1, x_2, \dots, x_i, \dots, x_n).$$

A random process $F(X)$ is given by the set (system) of functions $f_j(x_1, x_2, \dots, x_i, \dots, x_n)$, where $j = 1, 2, \dots, k$, each of which hypothetically describes the j -process. Accordingly, the experimental study of ferroelectrics is directed by the multi-factor experiment. Then for each function $f_j(x_1, x_2, \dots, x_i, \dots, x_n)$ in the presence of q levels of measurement-required number of experiments O_j will be determined by the ratio

$$O_j = q^{i_j} \quad (1)$$

Where $i_j \in [1, n]$ is the number of arguments – factors, on which the j -function F_j depends, where $j = 1, 2, \dots, j, \dots, k$.

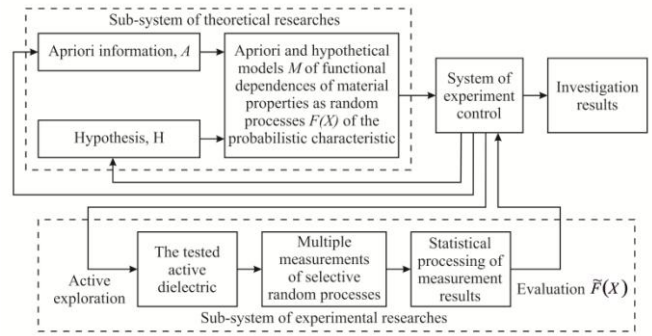


Fig. 1 Generalized structural scheme of the study process of active dielectrics for solution of tasks of materials synthesis

Due to the fact that a comprehensive study of material is not limited by the study of the reaction of one parameter of a set $\{y_1, y_2, \dots, y_j, \dots, y_k\} \in Y$, in general, the number of experiments is determined by the sum of O_{Σ} :

$$O_{\Sigma} = \sum_{j=1}^k O_j \quad (2)$$

Here it is assumed that on the basis of the analysis of a priori information from set A is known, from which specific factors X_{ij} from the set X (i.e. $X_{ij} \subset X$) depends each function y_j . In the case when a priori information is not enough to highlight the set X_{ij} , to obtain the reliable information about the properties of the test material, you need an experimental study of the functions y_j depending on all possible factors n . In this case, the expression (1) due to the fact that $i_j = n$ it always takes the form:

$$O_1 = O_2 = \dots = O_j = \dots = O_k = q^n.$$

Accordingly, the expression (2) is also transformed:

$$O_{\Sigma} = kq^n.$$

In general, when you want the behavior of multiple measurements, the value q takes into account the number of measurements at each level.

It should be noted that (2) gives an inflated number of experiments in case of intensive parameters – factors of the set X serve as arguments of several functions. Look at the example of ferroelectric materials having pronounced piezoelectric properties. Analysis of a number of scientific works and results of modern researches shows that practical interest for materials of this class is the study of the dependencies the following extensive parameters: the polarization P and the mechanical deformation x from strain X , electric field E , temperature T , frequency f . In addition, the presence of the inverse piezoelectric effect causes the necessity of measurement of the dependency of E and X (in this case, they

characterize intrinsic properties of a ferroelectric, they are extensive parameters) on intensive parameters – effects. These dependencies can be represented as a family of univariate functions under fixed parameters denoted by the subscript indices

$$\begin{aligned} &P(X)_{E,T,f}, P(E)_{X,T,f}, P(T)_{X,E,f}, \\ &P(f)_{E,X,T}, P(x)_{X,E,T,f}; \\ &x(X)_{E,P,T,f}, x(E)_{X,P,T,f}, x(P)_{X,E,T,f}, \\ &x(T)_{X,E,P,f}, x(f)_{X,E,P,T}; E(x)_{X,T,f}; X(P)_{E,T,f}. \end{aligned}$$

The totality of the single-factor dependencies on a particular extensive parameter carries information equivalent to the corresponding multivariate dependencies:

$$\begin{aligned} &P(X, E, T, f, x), x(X, E, T, P, f) \\ &E(x, X, T, f), X(P, E, T, f) \end{aligned} \quad (3)$$

If the factors are x and P , it has the reverse effect. Identifying each of the four types of experiments depending on ordinate indices $j = 1 \dots 4$ and that $i_1 = i_2 = 5$, $i_3 = i_4 = 4$, according to (2) we determine that in this case the number of experiments:

$$O_s = q_1^5 + q_2^5 + q_3^4 + q_4^4.$$

The analysis of a set of dependencies (3) can detect the redundancy of its member functions. Since constants of the inverse piezoelectric effect are equal to the corresponding constant direct effect, the dependency $E(x, X, T, f)$ is duplicated by dependency $P(X, E, T, f, x)$ and the dependency $X(P, E, T, f)$ by $x(X, E, T, P, f)$. Given the above, we imagine (3) to be an equivalent set from the point of view of the amount of information about the material properties:

$$P(X, E, T, f, x), x(X, E, T, P, f).$$

In this case, the number of measuring operations will be reduced to

$$O'_s = q_1^5 + q_2^5.$$

Before measuring, the drawing up of the apriori model M of the process $Y = F(X)$, which is a set of functional dependencies $\{y_j\}$. Then the initial model M represents a mapping of apriori and hypothetical ideas about the properties of the test material in the form of some blurred region, such that with some probability the actual process $Y = F(X)$ belongs to the blurred area. This probability, in its turn, is an assessment of the adequacy of the original model.

Methods of experimental study of field dependencies of ferroelectric materials are showed in the work [2], in [3] – the modeling of temperature dependences of the parameters of ferroelectrics with phase transition of the second kind, the explanation of the discrepancies of the results of experiment and theoretical calculations derived from the theory of Landau-Ginzburg-Devonshire is given. In this regard, the methodical error caused by the difference of the dielectric constant of the ferroelectric, measured in adiabatic conditions and calculated in isothermal conditions is appreciated.

The following are some aspects of modeling of the field dependences of the ferroelectrics parameters on the basis of thermodynamic theory. Use the expression for the dependence of polarization P on electric field E :

$$P = \frac{E\epsilon_0}{\chi^x} \left(1 - \frac{(\epsilon_0 E)^2 \xi_{11}}{(\chi^x)^3} - \frac{(\epsilon_0 E)^4 \zeta_{111}}{(\chi^x)^5} \right), \quad (4)$$

where the coefficients χ^x , ξ_{11} , ζ_{111} , can be determined experimentally, based on their physical meanings. For example, according to [4] the coefficients ξ_{11} and ζ_{111} are calculated by the experimentally obtained dependence $P(E)$ by solving the system of equations:

$$\begin{cases} P_0^2 = \frac{3}{4} \left(-\frac{\xi_{11}}{\zeta_{111}} \right) \\ \chi_0^x = \frac{3}{16} \frac{(\xi_{11})^2}{\zeta_{111}} \end{cases}, \quad (5)$$

where P_0 , χ_0^x – values of the spontaneous polarization and inverse dielectric susceptibility at the Curie temperature.

The expression (4) is used to model the main loop of the dependence polarization $P(E)$ (Fig. 2).

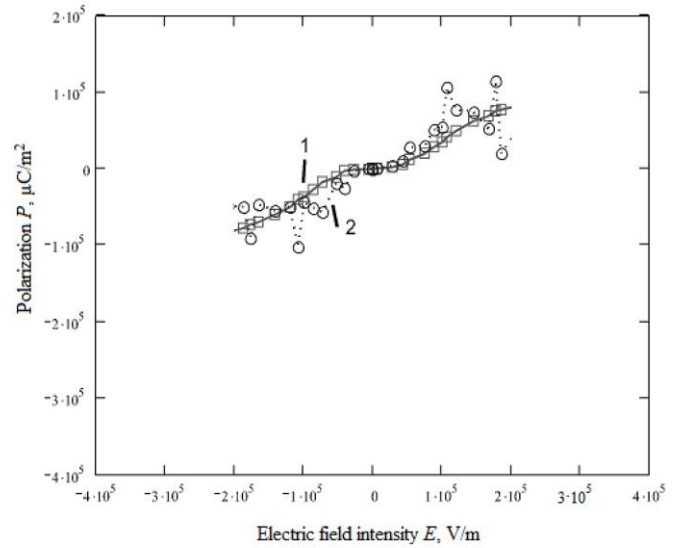


Fig. 2 Basic curve of polarization: 1 – experimental; 2 – simulated based on the thermodynamic theory

The results of the experiment coincide with theoretical values only in a narrow range of small applied fields (in the range from 100 kV/m to 100 kV/m). With increasing field on module the experimental curve deviates significantly from the theoretical one. A similar result was observed, for example, in [4] modeling dependencies $\epsilon(E)$.

The reason of this discrepancy is that in the result of applying the DC field in the crystal charged layers located near the electrodes spatially appear. This in turn reduces the dielectric constant measured in the equilibrium state.

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EFFECT OF BORONIZING PARAMETERS AND MATRIX STRUCTURES ON THE WEAR PROPERTY OF DUCTILE IRON

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Abstract: In this study, the effects of matrix structure (pearlitic, tempered martensitic, lower ausferritic, and upper ausferritic), boronizing temperature (800, 825, and 850°C) and time (3, 4.5 and 6 hours) on the wear behaviour of Cu-Ni-Mo alloyed ductile iron were investigated. Wear tests were performed on ball-on-disc type wear tester under the load of 6.8 N, at sliding speed of 6.5 mm/s, at room temperature and dry sliding conditions. The mass losses were measured after wear tests and the friction coefficients were obtained during wear tests. The hardnesses and thicknesses of boride layers, microstructures and worn surface examinations (SEM) of the matrix structures and borided layers were performed. The surface hardnesses of borided samples were obtained three or four times more than that of the matrix structures. The best wear performance was observed for the sample borided at 850°C for 6 h. The mass loss of this boronizing condition is 0,2 mg and this value is nine times less compared with that of the as-cast pearlitic structure.

KEYWORDS:DUCTILE IRON, MATRIX STRUCTURE, BORONIZING, MASS LOSS

1. Introduction

Bronizing, which is conventionally carried out by holding the materials at 700-1100°C in a boron-rich environment for diffusion of boron atoms into the material in order to form a boride layer, is very attractive thermochemical surface treating technique for ferrous alloys[1]. Boronizing medium can be in the form of a solid powder, paste, liquid, or gas. Pack boronizing is the most widely used boriding process because of its relative ease of handling, safety, and the possibility of changing the composition of the powder mix, the need for limited equipment, and the resultant economic savings[2,3].

The boronizing treatment mostly forms two boride phases; FeB (more externally) and Fe₂B. The significant features of these borides are their high melting points (1540°C for FeB and 1390°C for Fe₂B), metallic resistivity, high hardness, and excellent wear, friction, and corrosion resistance[4]. The coefficient of expansion of Fe₂B ($2,9 \times 10^{-8} \text{ K}^{-1}$) is less than that of iron ($5,7 \times 10^{-8} \text{ K}^{-1}$) and hence this phase remains in compression after cooling, while that of FeB ($8,7 \times 10^{-8} \text{ K}^{-1}$) is greater than iron or Fe₂B and therefore remains in tension[2]. This disparity in residual stress can result in the formation of cracks in the region of the FeB-Fe₂B interface, especially when a component is subjected to thermal and/or mechanical shock[5].

Generally, the formation of a monophase (Fe₂B) with saw-tooth morphology is more desirable than a double-phase layer with FeB and Fe₂B for industrial applications[6]. The degree of teething depends on the quantity of alloying elements, carbon concentration, temperature and time[7]. Industrial boriding can be carried out on most ferrous materials such as structural steels, cast steels, Armco iron, grey iron and ductile iron[8]. The most significant parameters that determine the characteristics of boride layer are processing temperature and time. Alloying elements mainly retard the boride layer thickness (or growth) caused by restricted diffusion of boron into the substrate because of the formation of a diffusion barrier[9].

Ductile iron is a member of cast iron family which draws great interest due to its unique mechanical properties. Most of this superiority is achieved by heat treating and surface hardening processes. Ductile iron can gain variety of matrix structures via heat treating, an example of which is ausferrite. Almost all surface hardening processes, applicable for steels, can be successively

performed on ductile irons. The aim of the present study is to investigate the effect of boronizing parameters (temperature and time) and matrix structure on the wear properties of Cu, Ni, and Mo alloyed ductile iron.

2. Experimental Procedures

The chemical composition of the ductile iron is given in Table 1. The nodule diameter and nodule count range between 29–37 μm and 100–155 mm^{-2} , respectively. The nodularity is above 90%.

Table 1. Chemical composition of the ductile iron (wt.%)

C:	3.730	Mg:	0.044
Si:	2.550	Cu:	1.030
Mn:	0.300	Ni:	1.250
P:	0.045	Mo:	0.180
S:	0.023	Cr:	0.032

The heat treatments, given in Table 2, were carried out in order to obtain various matrix structures, namely tempered martensitic (TM), low ausferritic (LA), and upper ausferritic (UA). Test materials were machined and ground to a cylindrical shape with dimensions of 15 mm diameter and 20 mm length. Pack boronizing technique was performed in a solid commercial Ekabor 2 (B₄C - SiC - KBF₄) powder with grain size range of 75-106 μm . All samples were packed in the powders mix and sealed in a stainless steel container. Boronizing was performed in an electrical resistance furnace under atmospheric pressure at various temperatures (800, 825, and 850°C) for 3, 4.5, and 6 h, respectively. The steel box was followed by cooling in air to room temperature.

Table 2. Matrix structures and heat treatment parameters

Matrix	Austenitizing temperature/time, T _γ / t _γ	Tempering conditions and/or cooling
TM	900 °C /1 h	Oil quenching, tempering at 400 °C for 1 h, air cooling
LA	900 °C /1 h	Austempering at 300 °C for 1 h, air cooling
UA	900 °C /1 h	Austempering at 365 °C for 1 h, air cooling

The hardness of borided surfaces and matrix structures were measured on the cross-sections using Metkon Mh-3 Vickers indenter with a load of 1 kg.

An Eclipse MA100 optical microscope was used to examine the microstructures and depth of boride layers of polished and etched (with 5% nital solution) specimens, and a Zeiss Evo40 scanning electron microscope for the worn surface of specimens.

The ball-on-disc abrasive wear tests were performed on metallographically polished samples at room temperature with the humidity of $45 \pm 5\%$. A constant load of 6,8 N, a sliding velocity of 6,5 mm/s, and a sliding distance of 140 m were used. For the counter body, 5 mm diameter tungsten carbide ball with 68 HRC hardness was selected. The samples were cleaned by ethyl alcohol and dried before and after the wear tests. Then, the mass losses were measured by a balance with sensitivity of 10^{-4} g. The friction force was monitored continuously by means of a force transducer. The friction coefficients were recorded during the tests. The wear tests were repeated three times for each boriding parameters and matrix structures.

3. Results and Discussion

The microstructures of as-cast (pearlitic), TM, LA, and UA matrix structures of the ductile iron are illustrated in Fig.1. The as-cast microstructure consisted of bull's eye structure; ferrite surrounding graphite nodules in a pearlitic matrix (Fig 1a). While LA structure shows fine acicular ferrite (dark needles) in Fig 1c, UA structure shows coarse feathery ferrite (Fig 1d) in the ausferritic structure.

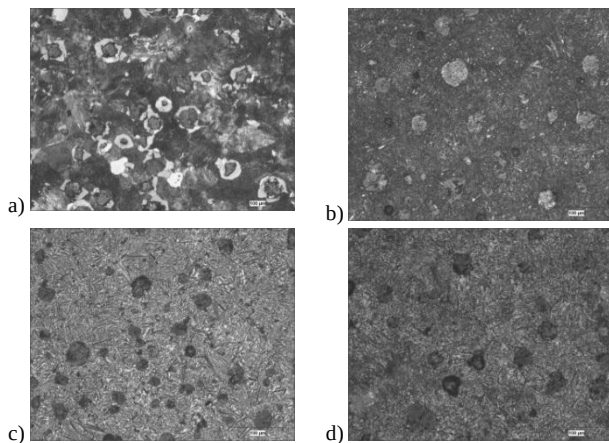


Figure 1. Microstructures of the matrix structures of
a) as-cast, b) TM, c) LA, and d) UA (200X)

The hardness values of matrix structures are given in Table 3. The TM structure has the highest hardness due to the brittleness effect of martensite. The UA structure has higher hardness than the LA one. This result may be due to the second stage reaction. This reaction occurs at high austempering temperatures, and/or long durations. High carbon austenite decomposes into carbides and ferrite by the second reaction. These carbides rise the hardness and lower the strength and toughness of ductile iron. [10]

Table 3. Hardnesses and mass losses of the matrix structures

Matrix	Hardness [HV ₁]	Mass loss [mg]
As-cast(pearlitic)	413	1,8
TM	614	1,2
LA	464	1,4
UA	494	1,3

The surface hardnesses of borided samples are given in Figure 2. The values varied between 1280-1685 HV₁. The hardness of the boride layer was found to be three or four times higher than that of the as-cast structure. This result is well agree with the previous study[11]. The hardnesses increased with both the boronizing temperature and time. This increase is more clear at 850°C. The durations at 800 and 825°C made no significant variations in surface hardness values.

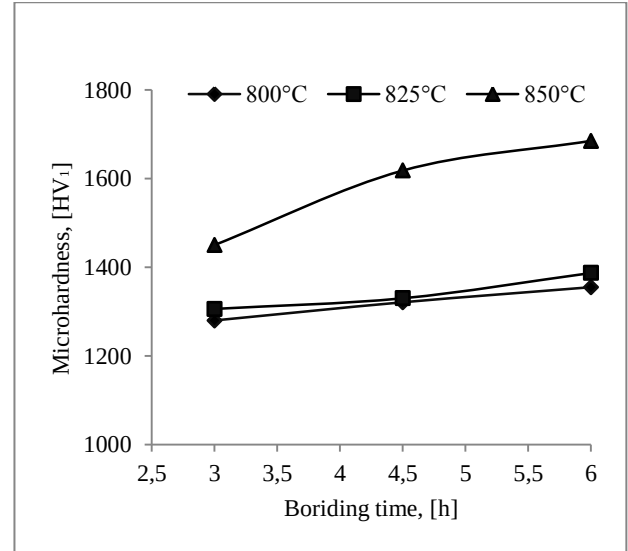


Figure 2. Microhardnesses of the borided surfaces

The microstructures of the borided layers are given in Figure 3 for 6h boriding time at 800, 825, and 850°C temperatures, respectively. The layer thickness is increased with the boronizing temperature. The boride layer has a tooth-shaped structure. It is observed that nonuniform light lines present at interface between the matrix and the borided layer. These lines are thought to be silicon-rich zones. Silicon has no solubility in iron boride. Therefore, during the boronizing process, silicon atoms diffuse inwards and produce silicon-rich zone[11].

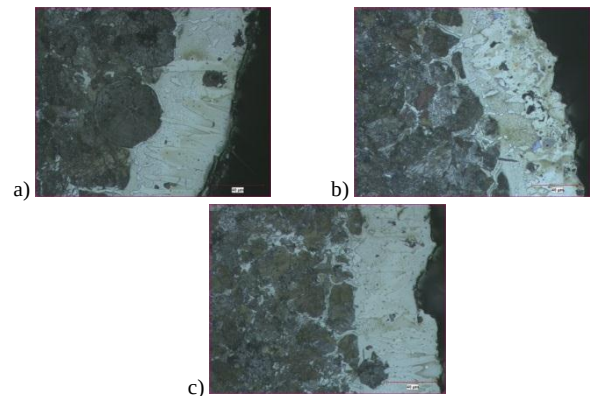


Figure 3. Microstructures of the boride layers boronized for 6 h at
a) 800°C, b) 825°C, and c) 850°C (500X)

The effect of boronizing parameters on the boride layer thickness is given in Figure 4. As the boronizing temperature and time increased, the boride layer also increased as consistent with the previous studies[12,13]. The thickness and hardness of the boride layer depends on the substrate material being processed, boron potential of the boronizing compound, boronizing temperature-time, boronizing mediums and their compositions[3]. The increase rates

of boride layer by the boriding time have close values of approximately 64, 59, and 60% at 800, 825, and 850°C, respectively.

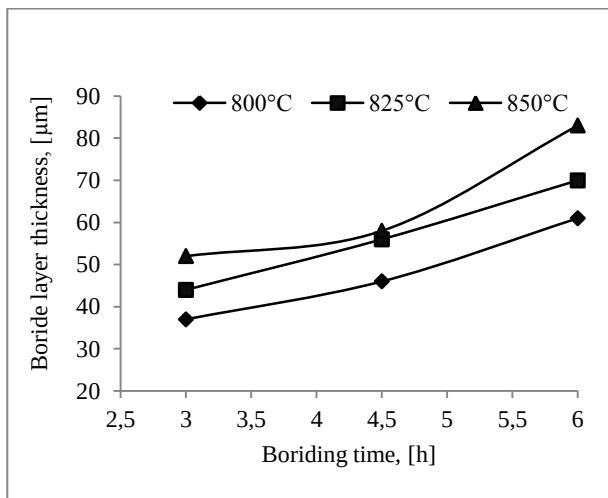


Figure 4. Thicknesses of the boride layers

The mass losses of matrix structures are given in Table 3. The values are varied inversely proportion with their hardness values. Namely, the highest hardness structure showed the least mass loss such as in TM structure. This result confirms that hard surfaces are more resistant to wear.

The mass losses by the boronizing temperature and time are given in Fig 5. At 825 and 850°C boronizing treatments, mass losses are decreased by the process duration. An oppose and unexpected result is observed at 800°C. For all durations at 800°C, due to having low microhardness values (Fig 2.), the mass losses are unexpectedly lower than that of 825°C. It can be concluded that 800°C boronizing temperature is not enough for the studied alloyed ductile iron. As the alloying elements retard the diffusion rate of boron into the substrate, the higher boronizing temperatures are required for the expected performance. The durations at 850°C is more effective in mass loss as in boride layer. Boronizing at 850°C for 6h showed the least mass loss among all parameters as 0,2 mg. C. Li et al. [14] reported that higher temperature and longer process duration result in more excellent wear resistance. The pearlitic as-cast structure showed nine times more mass loss comparing to that of specimen boronized at 850°C for 6h (Table 4 and Fig. 5).

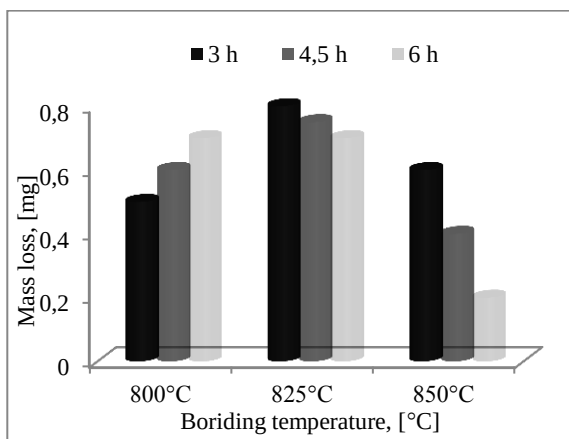


Figure 5. Mass losses by the boronizing temperature and time

The friction coefficients of the pearlitic, TM, LA, and UA structures are given in Fig 6. At the beginning of wear tests, the friction of coefficients increased to high values and then lowered and caught the steady state for all structures. The as-cast pearlitic structure has the lowest friction coefficient with more steady distribution.

Having many boriding variables, it was not possible to illustrate the friction coefficient distributions of borided surfaces in a graph. Wider distributions in friction coefficients of borided surfaces were observed compared with that of the matrix structures. The values are ranged between 0.15-0.65 at 800 and 825°C boriding temperatures for all durations. However, a narrow range of values between 0.15-0.3 was observed at 850°C. Moreover, the distributions of friction coefficients were more steady at 850°C.

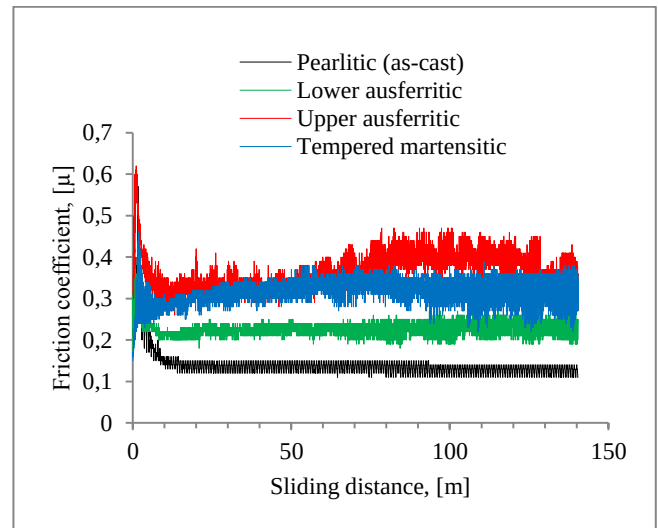


Figure 6. The friction coefficients of the matrix structures

Figure 7 illustrates the SEM graphs of worn surfaces of the matrix structures (TM and LA) and surfaces borided for 6 h at 800°C and 850°C temperatures. The micro scratches in the matrix and the micro cracks, formed at graphite-matrix interface, are dominant in the TM structure (Fig 7a). An adhesive wear mechanism is more effective with significant plastic deformation in the LA structure (Fig 7b). The worn surface of sample borided at 800°C for 6 h showed remarkable amount of cracks (Fig 7c)

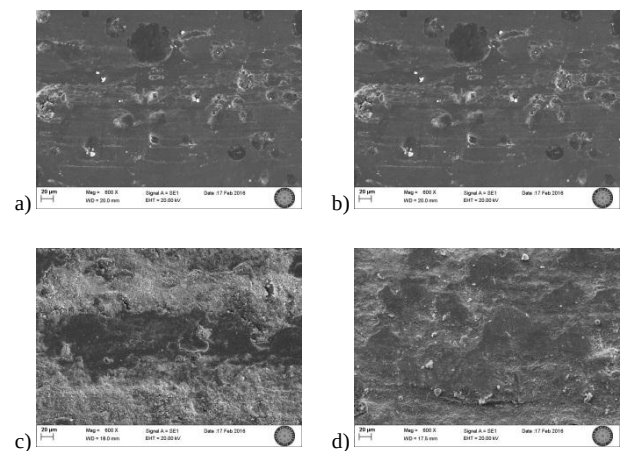


Figure 7. SEM graphs of the worn surfaces of structures a) TM and b) LA and surfaces borided for 6 h at c) 800°C and 850°C (600X)

4. Conclusion

The following conclusions can be drawn from the present study:

- The hardness values of the borided surfaces increased three and/or four times higher than that of the as-cast structure.
- The surface hardnesses increased with both the boronizing temperature and time. This increase is more clear 850°C temperature.
- The boride layer thickness increased with both the boronizing temperature and time. This increase rates by the boriding time have approximately close values as 64, 59, and 60% at 800, 825, and 850°C, respectively.
- 800°C boronizing temperature is found to be low to generate expected boride layer on the studied alloyed ductile iron. Many micro cracks were detached on the worn surface of 800°C/6 h borided sample.
- Boronizing at 850°C for 6 h showed the least mass loss among all parameters as 0,2 mg. The pearlitic structure showed nine times more mass loss according to that of this boriding parameter.
- The friction coefficients were more steady for the matrix structures than the borided ones. The as-cast pearlitic structure has the most steady and lowest friction coefficient value.

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EFFECTS OF MECHANOCHEMICAL TREATMENT OF ThO_2 WITH UO_3 AND CeO_2

ЕФЕКТИ ОТ МЕХАНОХИМИЧНО ТРЕТИРАНЕ НА ThO_2 С UO_3 И CeO_2

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Abstract: The paper presents the effects of the mechanochemical activation of ThO_2 - UO_3 and ThO_2 - CeO_2 mixtures in air and (for ThO_2 - UO_3) in suspension in H_2O or CHCl_3 . Planetary ball mill (Pulverisette 5, Fritch) with stainless steel triboreactors and milling balls from the same materials are used. Milling for 5 h of the ThO_2 - UO_3 mixture does not affect significantly the crystal structure of the ThO_2 but leads to amorphization of UO_3 , sharp decrease of its crystallites size and increase of the ionic character of the U-O bond. Storage of the activated product (2 months at room temperature) or heating (3 h, 165 °C) leads to partial relaxation of the UO_3 crystal structure. The X-ray diffraction data does not give proves for formation of ThO_2 - UO_3 solid solution.

Formation of solid solution $\text{Ce}_{0.6}\text{Th}_{0.4}\text{O}_2$ accompanied with some amount of amorphized ThO_2 , is established as a result of co-milling at the same conditions of the mixture ThO_2 - CeO_2 with a mole ratio 1. XRD-determined lattice parameter (5.4804(9) Å) and crystallites size (14 nm) of the obtained solid solution are in satisfactory agreement with literature data for the same product obtained via wet chemistry route.

Keywords: THORIUM DIOXIDE, URANIUM TRIOXIDE, CERIUM DIOXIDE, MECHANOCHEMISTRY, SOLID OXIDE SOLUTIONS, XRD, IR-SPECTROSCOPY

1. Introduction

At least two factors determine the research interest for mixed actinide dioxides of the type ThO_2 - UO_2 and ThO_2 - PuO_2 : (i) they are considered as a promising fuel for some types of nuclear reactors. The application of solid solutions is preferred because they insure uniform mixing of the individual oxides; (ii) thorium-based dioxides are expected to have good performances in long-term storage of radioactive waste because of the low solubility and very low dissolution rates. In the same time it is known that cerium/ CeO_2 is suitable surrogate for Pu/ PuO_2 due to the closeness of the cation radii and the identity of the oxides crystal structure. So solid solutions of the type ThO_2 - CeO_2 are of significant interest as model systems [1-3], strengthened by the fact that Ce/ CeO_2 is one of the major fission products produced in the nuclear fuel.

The mentioned solid solutions are usually prepared through dry chemistry routes but wet chemistry methods were also proposed recently. A short review of the methods proposed for U-Th mixed oxides is presented in [4]. Practically, all of the used methods require transformation of U(VI) to U(IV) oxidation state, realized electrochemically or by heating in reducing atmosphere or in presence of reducing agent. Later on, the photochemically-induced preparation of nano-powders of crystalline UO_2 - ThO_2 solid solution has been investigated [5-7]. Preparation of $\text{Ce}_x\text{Th}_{1-x}\text{O}_2$ solid solutions through thermally induced transformation of oxalate precursors [1, 2] or by citric sol-gel combustion technique followed by a thermal annealing (1000 °C for 24 h) [3] is proposed.

The literature data on the effects of the mechanoactivation on the 5f-elements compounds are rather limited. The mechanochemical behavior of uranium oxides is reported in [8-11]. It was shown that they are comparatively stable upon mechanical treatment. The main effect of the mechanoactivation on the U_3O_8 and UO_3 in air or in suspension with organic solvents is the reduction of U(VI) with the formation of UO_2 , U_3O_7 and U_2O_5 ; the degree of reduction and the leaching of ^{238}U and the decay product ^{234}Th (the later - as a result of mechanoactivation in suspension with organic solvents) is found to depend on the mechanical treatment conditions and of the complexing agent nature. Mechanochemically assisted preparation of U-Th [4], U-La and U-Ce [12] oxide solid solutions was proposed: the polyphase product of external gelation process with the participation of U- and Th nitrates (analogous to the one described by Kumar et al. [13] for synthesis of U-Ce mixed oxides) is undergone to mechanical treatment, the later leading to

phase homogeneous product. Solid (U,Pu) O_2 solution is obtained by co-milling of UO_2 and PuO_2 [14]. Nanocrystalline UO_{2+x} powders have been prepared by high-energy ball milling and subsequently consolidated into dense fuel pellets under high pressure [15].

In the present work the potential is studied of the solely mechanochemical treatment for synthesis of Th-U and Ce-Th oxide solid solutions via co-milling of the individual oxides ThO_2 - UO_3 and ThO_2 - CeO_2 , respectively.

2. Materials and Methods

Pure ThO_2 (5 g, Carlo Erba Reagents, Italy) and its mixture with $\text{UO}_3 \cdot 0.8\text{H}_2\text{O}$ (Chemapol) (3 g ThO_2 and 2g UO_3) were mechanically activated in air or as suspension in 8 ml CHCl_3 or in 6 ml distilled H_2O as shown on Fig. 1. After activation in suspension the solid phase was separated by filtering through fine-porous filter, rinsed few times with a total volume of 50 ml CHCl_3 or H_2O (when suspensions are used) and dried at room temperature. Part of the mixture milled in CHCl_3 was heated for 3 h at 165 °C.

Another set of experiments was performed with a mixture of ThO_2 and CeO_2 (total mass about 5.2 g mole ratio Th:Ce = 1), mechanically activated in air.

Planetary ball mill (Pulverisette 5, Fritch) with stainless steel vessels of 80 cm³ volume and balls from the same material (7-15 mm in diameter) were used. The activation was performed for 5 h at mechanical load of 12g.

The X-ray diffractograms of the samples before and after the mechanochemical processing were taken by a powder diffractometer Siemens D500 using CuK_α radiation filtered by a secondary monochromator (40 kV, 30 mA, 0.05° 2 θ /2 s). The relative content and mean crystallite size of the oxides were determined from XRD pattern using the program POWDER CELL [16]. The crystal cell parameter, size of crystallites and density (for the ThO_2 - CeO_2 system) were determined by the BRASS program [17].

The IR spectra in the range 4000-400 cm⁻¹ were recorded by a Bruker spectrometer in capillary layer.

Optical microscope B-130 (Optika, Italy) equipped with Celestron digital microscope Imager with 2 mega-pixel sensor and IR cut-off filter was used for observation of the mixture of ThO_2 and CeO_2 before and after the mechanical processing.

3. Results and Discussion

3.1. Mechanochemical effects on ThO₂ in air and in suspension

The mechanical treatment of ThO₂ neither in air [4], nor in CHCl₃-suspension (at above described experimental conditions) leads to any changes in its X-ray diffractogram (Fig. 1) which completely correspond to the one of the referent ThO₂ (JCPDS 78-0685, 2001).

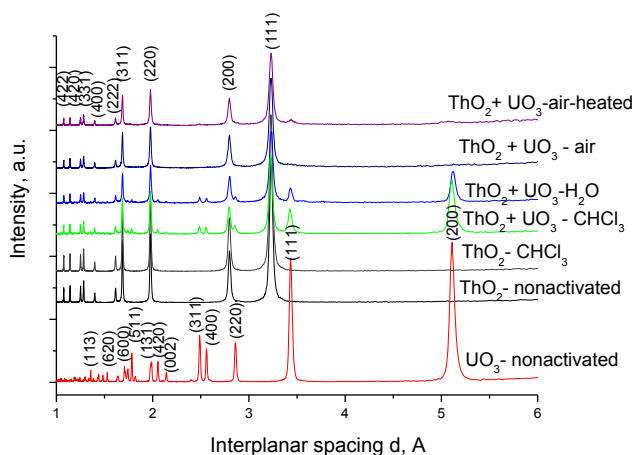


Fig. 1 X-ray diffractograms of the initial UO₃ and ThO₂; mechanoactivated products; ThO₂-UO₃ mixture post activation heated. Miller indices for UO₃ and ThO₂ are shown.

3.2. Mechanochemical effects on mixture of ThO₂ and UO₃

Milling in air of the ThO₂-UO₃ mixture practically does not change the X-ray diffraction behavior of the ThO₂ (compared with the initial oxide) but drastically decreases the crystallites size of UO₃ (Table 1). Amorphization of the UO₃ as a result of mechanoactivation is reported in [10], also. The intensity of the UO₃-peaks vigorously decrease (the diffractogram of the milled mixture becomes rather close to that of the initial thorium, Fig. 1). Due to that the relative content of UO₃, calculated from the diffraction data is much lower (Table 1) than that in the initial mixture (60 %). The activation of the mixture in CHCl₃- or H₂O-suspension leads to similar, but less (especially in CHCl₃ medium) pronounced effect of UO₃ amorphization (Fig. 1). The X-ray diffraction data does not give proves for formation of ThO₂-UO₃ solid solution.

Table 1: Relative content (RC, %) and crystallites size (CS, nm) of ThO₂ and UO₃ in studied samples before and after mechanical treatment in air

Sample	Mechano-activation	Post-activation heating	ThO ₂		UO ₃	
			RC	CS	RC	CS
ThO ₂	No	-	100	32	-	-
UO ₃			-	-	100	56
ThO ₂ -UO ₃ mixture	Yes	No	95	29	5	5
		160 °C, 3 h	85	28	15	17
		25 °C, 2 months	64	25	36	47

An interesting effect of relaxation of the UO₃ crystal structure is observed with time at ambient temperature which is promoted by heating for 3 h at 165 °C (Fig. 1, Table 1). Relaxation of the crystal structure of mechanoactivated UO₃ promoted by heating (at 300 °C) is also reported in [9].

The mechanochemically-induced amorphization is the probable reason for the changes in the general pattern of the samples IR-spectra (Fig. 2). Besides that, some changes in the region 950-850 cm⁻¹ (connected with uranyl-type U-O-bond vibration) are observed. While the band around 937 cm⁻¹ (asymmetric stretching

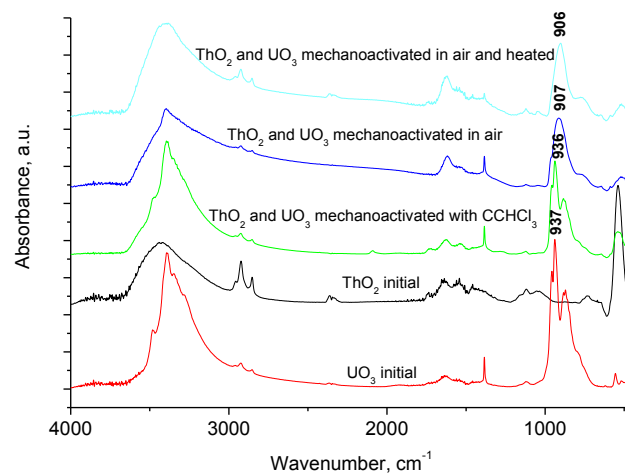


Fig. 2 IR-spectra (4000 – 400 cm⁻¹) of the initial UO₃ and ThO₂; mechanoactivated products and of activated ThO₂-UO₃ mixture heated for 3 h at 165 °C.

vibration $\nu_{as}(\text{U-O})$ [9]), existing in the initial UO₃, is slightly affected by the activation in CHCl₃- suspension, it is significantly shifted to lower wave-lengths (to 907 cm⁻¹) when activation in air is performed. The shift reveals an increase of the ionic character of the U-O bond and, respectively, its weakening. The subsequent heating of the activated sample (3 h, 165 °C) does not lead to restoration of the bond character, i.e. it seems that the heating leads to relaxation of the crystal, but not the molecular structure.

3.3. Mechanochemical effects on mixture of ThO₂ and CeO₂

The X-ray diffractograms of the initial ThO₂ and the product obtained after mechanoactivation of a mixture ThO₂-CeO₂ are shown on Fig. 3. Table 2 contains the data for the interplanar distances and the diffraction peaks relative intensity of ThO₂, CeO₂ and the product of activation of their mixture.

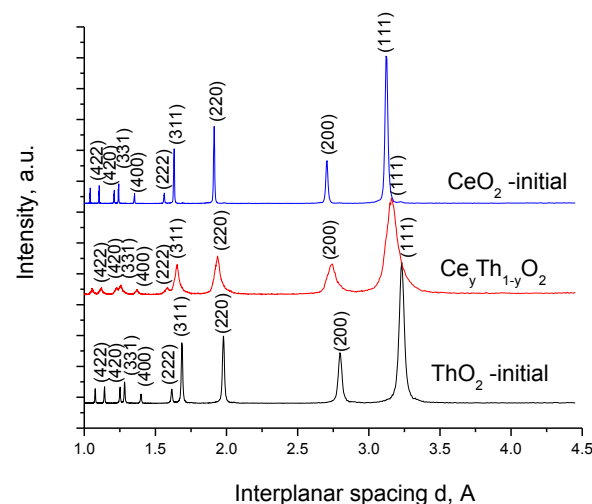


Fig. 3 X-ray diffractograms of the initial ThO₂ and CeO₂ and the product obtained after mechanoactivation of the ThO₂-CeO₂ mixture.

Data for the lattice constant, the density and the size of crystallites of the components of the studied system are shown in Table 3. The sharp decrease of the size of the crystals as a result of milling, seen on the optical microscopy images (Fig. 4), is accompanied with more than two times decrease of crystallites size of the obtained mixed oxide compared with that of the initial ThO₂ and more than four times, compared to the initial CeO₂.

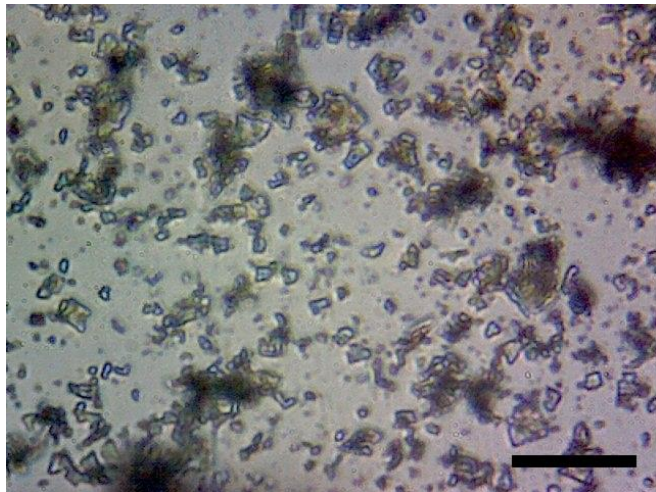
The density and the lattice parameter of the activated product (Table 3) takes an intermediate values between the constants of the

Table 2: Interplanar distances and relative intensity of ThO_2 , CeO_2 and the product of their mixture activation

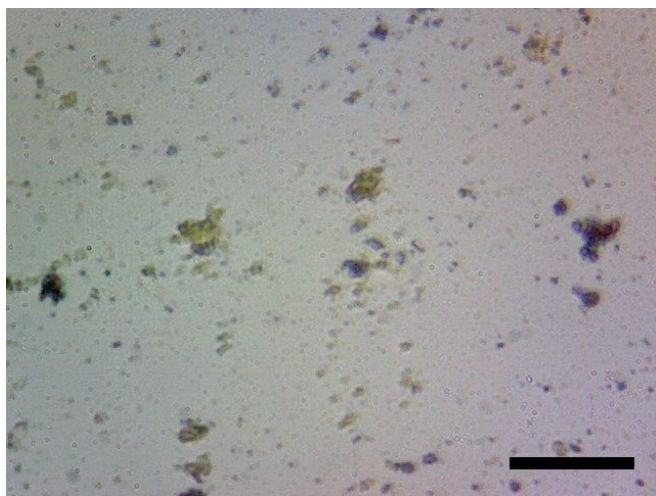
ThO_2 JCPDS 42-1462		CeO_2 JCPDS 43-1002		$\text{Ce}_y\text{Th}_{1-y}\text{O}_2$ Present work	
d, Å	I, %	d, Å	I, %	d, Å	I, %
3.232	100	3.1240	100	3.166	100
2.799	41	2.7060	27	2.739	33
1.9787	45	1.9132	46	1.937	39
1.6877	46	1.6316	34	1.654	32
1.6159	11	1.5621	6	1.588	8
1.3993	7	1.3528	6	1.366	6
1.2841	18			1.264	10
1.2516	14	1.2414	12	1.223	8
1.1426	13	1.2100	7	1.119	7

Table 3: Some physical parameters of the initial oxides and the mechanoactivated product

Parameter	ThO_2	CeO_2	$\text{Ce}_y\text{Th}_{1-y}\text{O}_2$
Lattice constant, Å	5.5915	5.4037	5.4804(9)
Density, g/cm ³	10.0223	7.2261	8.7986
Crystallites size, nm	32	65	14



a



b

Fig. 4 Microscopic images of the mechanical mixture of ThO_2 and CeO_2 before (a) and after (b) mechanochemical treatment; magnification 40x, size bar 20 μm .

initial oxides (without any distortion in unit cell) suggesting formation of a solid solution. The same FCC fluorite-type structures

of both initial oxides and closeness of the ionic radii (96 pm for Ce^{4+} and 105 pm for Th^{4+} [2]) are good grounds for a solid solution formation.

The stoichiometry of the product of mechanoactivation was determined using the Vegard's law (linear dependence exists between the lattice parameter and the molar part of the elements in the solid solution). The dependence of the lattice parameter a and Th-content c_{Th} in the activated product is shown on Fig. 5 using literature values for the lattice constants of the initial oxides (Table 3). From the derived straight line equation

$$a [\text{\AA}] = 5.4037 + 0.00188c_{\text{Th}} [\text{mol \%}],$$

accounting for the found (from the X-ray data) value of the solid solution constant (Table 3) follows that $c_{\text{Th}} = 40 \text{ mol\%}$, i.e. the solid oxide solution has to be written as $\text{Ce}_{0.6}\text{Th}_{0.4}\text{O}_2$. The so found composition is different from the composition of the initial oxide mixture in which the mole ratio $\text{Ce/Th} = 1$. This fact leads to two important deductions: (i) it has to be supposed that in this system ThO_2 is much more inclined to amorphization than in the ThO_2 - UO_3 system and some amount of strongly amorphized ThO_2 is present in the mechanically treated system; (ii) despite the availability of ThO_2 in the milled system, solid solution, richer to ThO_2 , is not produced. Other synthetic methods permit preparation of such mixed oxides, up to $\text{Ce}_{0.1}\text{Th}_{0.9}\text{O}_2$ [3].

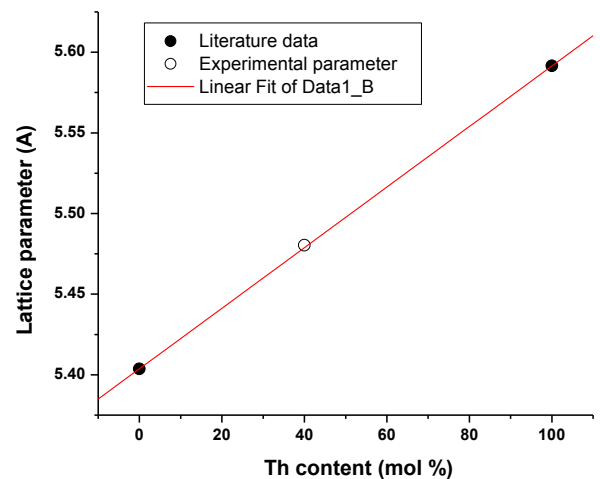


Fig. 5 Dependence of the lattice parameter on the relative content of ThO_2 in the mechanochemically formed solid solution

Table 4: Lattice parameter (a) in dependence of the mole part (y) of Ce in the $\text{Ce}_y\text{Th}_{1-y}\text{O}_2$

y	a, Å	Reference
0.4	5.43	3, JC-PDF 78-0704
0.5	5.5044(1)	2
0.6	5.4942(1)	2
0.6	5.4804(9)	Present work

A satisfactory agreement is found between literature data for the lattice constants of the mixed oxides of the type $\text{Ce}_y\text{Th}_{1-y}\text{O}_2$ and the value determined in the present paper (Table 4).

The size of the particles of the newly synthesized product is two times smaller than that of the initial ThO_2 suggesting that radical changes take part in the course of the mechanochemical treatment.

An interesting comparison can be made between the crystallites size of the mechanochemically obtained $\text{Ce}_{0.6}\text{Th}_{0.4}\text{O}_2$ and that of the analogous mixed oxide prepared through thermally induced transformation of oxalate precursors [2]. The size of crystallites of the later product increases with the increase of the heating temperature and is found to be 5 nm after heating at 400 °C, 15 nm at 900 °C and 50-60 nm at 1200 °C [2]. As it is shown in Table 3, 14 nm were determined in the present work. So, one can speculate that the mechanical treatment (at the conditions described in the

Materials and Methods part of the paper) is equal to heating of roughly 800-900 °C. Such an approximation is supported by the general pattern of the X-ray diffractograms of the compared products. Shapely, but relatively broad XRD lines are seen on Fig. 3, similar to the ones obtained in the diffractogram of the solid solution produced via wet chemistry route after heating at about 800 °C [2].

4. Conclusions

The present study shows that the ThO₂ is rather stable upon mechanical treatment in air or as CHCl₃-suspension. The mechanoactivation of a mixture of ThO₂ and UO₃ in air and in suspension in H₂O or CHCl₃ affects the UO₃ only, leading to its amorphization and destroying of its crystal structure; this effect is strongly revealed at air-milling and is lowest in CHCl₃-suspension. No X-ray proves for mechanochemical formation of solid solution ThO₂-UO₃ is found.

The response of the ThO₂ to the mechanochemical treatment in mixture with CeO₂ is rather different. The main effect is formation of solid solution Ce_yTh_{1-y}O₂ where, it seems that at the conditions applied in this work, $y \leq 0.6$. Most probably, the excess of Th in the activated system remains in amorphous state. The lattice constant and crystallites size of the obtained solid solution are in satisfactory agreement with literature data for the same product obtained via wet chemistry route.

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PECULIARITIES OF Fe POWDER CONSOLIDATION IN CONDITIONS OF SPARK-PLASMA SINTERING

ОСОБЕННОСТИ КОНСОЛИДАЦИИ ПОРОШКА ЖЕЛЕЗА В УСЛОВИЯХ ИСКРО-ПЛАЗМЕННОГО СПЕКАНИЯ

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Abstract: Theoretical and experimental data on the impact of heating rate during spark-plasma sintering on densification kinetics, grain size and capillary pressure in powder compacts, based on Fe, are given. It is found out, that an increase of heating rate in range from 10 °C/s to 20 °C/s leads to acceleration of process of obtainment of non-porous compacts and decrease of structure grain size.

KEYWORDS: HEATING RATE, SPARK-PLASMA SINTERING, DENSIFICATION, POROSITY, GRAIN SIZE

1. Introduction

It is known that physical-mechanical properties of powder compacts are largely dependent on their porosity and grain structure [1, 2], and method of spark-plasma sintering (SPS) provides accelerated consolidation of powder materials, which influences the process of compacts densification [3, 4]. Fe-based composites are the most common among metal-matrix composite materials, used in industry [5]. Thus, studies of impact of heating rate on densification kinetics and structure of pure Fe compacts are of high scientific urgency.

The goal of present work is to study the impact of heating rate during SPS on densification kinetics and structure of Fe compacts.

2. Preconditions and means for resolving the problem

Studies were performed on PZhR-3 (GOST 9849-86) (ПЖР-3 (ГОСТ 9849-86)) Fe powder with mean particle size of 60 μm.

Consolidation of Fe powder was performed by SPS method on “GEFEST” (“ГЕФЕСТ”) experimental complex [6], which allows powders consolidation at mechanical loading in vacuum by passage of superposition of direct and pulsing currents of 10 kHz frequency and total amplitude of 1.1 kA.

Consolidation mode, considered in [7], was taken as base: heating rate was 10 °C/s, isothermal holding temperature was 1100 °C, isothermal holding time was $\tau = 180$ s, cooling rate was 10 °C/s. Change of heating rate was due to increase of current amplitude rise rate from 14 A/s to 30 A/s.

Porosity of consolidated specimens was studied according to GOST 9391-80 (ГОСТ 9391-80). Theoretic evaluation of grain size change was performed using MS Excel software.

Experimental studies of compacts grain structure were performed using methods of computer metallography according to DSTU ISO 643:2009 (ДСТУ ISO 643:2009) and [8] using “Biolam-I” (“Біолам-І”) optic microscope and ImageJ software.

3. Results and discussion

In order to define the possibility of impacting structure formation process in Fe compact during heating, an SPS process model, based on views of continuum sintering theory [3], which describes macroscopic behavior of porous body during consolidation and connects external pressure and deformation rate tensor components, was considered. On its base, authors [3, 9] created a system of differential equations, which describes changes of pores shape, porosity and grain size depending on heating temperature at constant pressure. Yet, in order to use equations, given in [3] for case of Fe powder SPS, coefficients A_0 , m , which characterize system viscosity and presence of liquid phase, must be known, as well as Fe diffusion characteristics at SPS conditions, which, according to papers [10 – 16] lie in wide range. Therefore, it was decided to find values of A_0 , m coefficients for powder

consolidation in presence of liquid phase [4] experimentally in order to be able to prognose Fe powder behavior during SPS, and evaluations from paper [3] were written in such form:

$$G' = \begin{cases} 6.67 \times 10^{-4} \omega \cdot \ln \frac{235}{\omega} \cdot G_0 \cdot \left(\frac{G_0}{G}\right)^2 \cdot \theta^{-1.24}, & \text{if } T > 733 \text{ K} \\ 0, & \text{if } T \leq 733 \text{ K} \end{cases} \quad (1)$$

$$\theta' = (1 - \theta) \cdot \left\{ \left(\frac{3 \cdot \theta}{2} \right)^{\frac{m+1}{2}} \cdot \frac{\frac{3 \cdot \theta}{2} \cdot (1 - \theta) - \sigma_x}{A_0 \cdot \exp \left(\frac{Q_{cr}}{R \cdot T} \right) (1 - \theta)^{\frac{m+1}{2}}} \right\}^{\frac{2}{m+1}}$$

where G' – change of grain size, G_0 – initial grain size, ω – heating rate, θ' – change of porosity, α – surface tension, σ_x – external stresses, A_0 , m – creep law constants, Q_{cr} – dislocation crawl activation energy, R – universal gas constant, T – temperature.

In order to check the possibility of these dependences usage as well as to find A_0 , m coefficients, series of experiments, aimed at determination of porosity θ and grain size G in consolidated specimens of initial clean Fe powder, were performed.

Variation of heating rate from 10 °C/s to 20 °C/s was due to change of current amplitude rise rate in range from 14 A/s to 30 A/s. Changes of current value during Fe powder SPS are shown on Fig.1

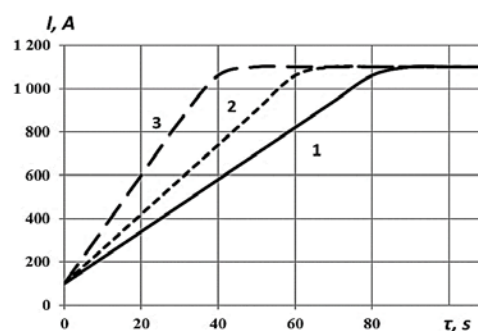


Fig. 1 Experimental curves of changes of current amplitude during SPS with different heating rate
1 – 10 °C/s; 2 – 15 °C/s; 3 – 20 °C/s

Curves of heating and porosity change are shown at Fig. 2 and Fig. 3 respectively. Temperature sensor was installed on matrix surface, which leads to offset of temperature curves by time axis. During the studies, it was found out that specimens, consolidated without further isothermal holding at heating rate higher, than 20 °C/s, are destroyed after they are taken out of matrix.

Analysis of obtained data shows that an increase of heating rate from 10 °C/s to 20 °C/s leads to decrease of densification time from 60 s to 40 s (see Fig. 3). Optical microphotographs confirm significant porosity decrease after 30 s of sintering with heating rate of 20 °C/s (see Fig. 4).

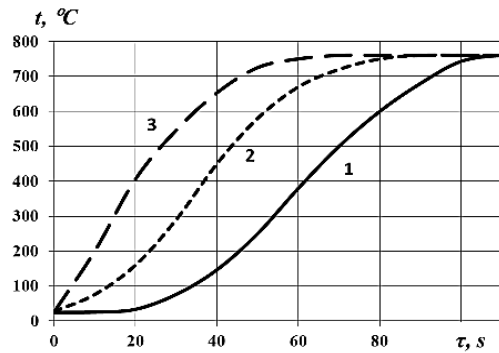


Fig. 2 Experimental curves of changes of temperature during SPS with different heating rate
1 – 10 °C/s; 2 – 15 °C/s; 3 – 20 °C/s

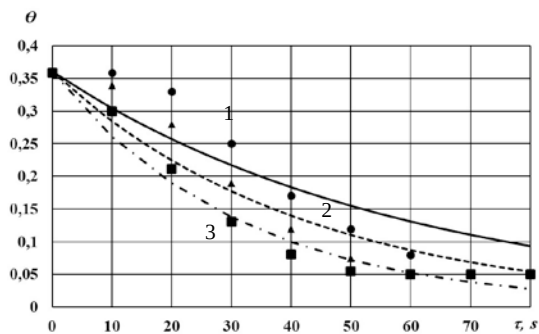
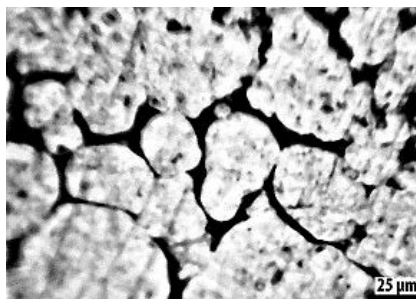
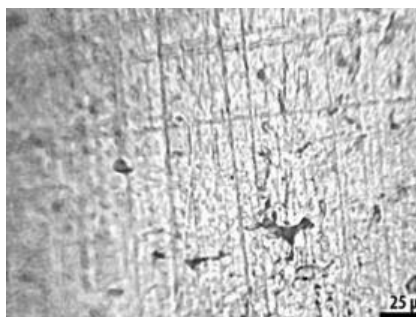


Fig. 3 Experimental curves of changes of porosity of consolidated Fe powder specimens after SPS with different heating rate
1 – 10 °C/s; 2 – 15 °C/s; 3 – 20 °C/s



a



b

Fig. 4 Optical microphotographs of consolidated Fe powders, holding time 30 s, magnification $\times 250$, pores are dark
a – heating rate of 10 °C/s; b – heating rate of 20 °C/s

Obtained experimental data was approximated with 90 % accuracy by exponential curves:

$$\theta_{10} = 0,36 \cdot e^{-0,017\tau}, \quad (2)$$

$$\theta_{15} = 0,36 \cdot e^{-0,024\tau}, \quad (3)$$

$$\theta_{20} = 0,36 \cdot e^{-0,032\tau}, \quad (4)$$

Analysis of grain size changes (see Fig. 5) during SPS at heating up to 1100 °C shows, that at heating rate of 10 °C/s mean grain size of specimens increases from 2.4 μm to 16 μm, and use of heating rate of 20 °C/s leads to almost two times decrease of grain growth (mean grain size is 10 μm), which is confirmed by optical microscopy (see Fig. 6) and leads to increase of specimens hardness from 60 HRB to 85 HRB.

In order to avoid negative values and characterize absence of grain growth in Fe before recrystallization begins, experimental data was approximated with 80 % accuracy by exponential curves:

$$G_{10} = 1,8663 \cdot e^{0,0292\tau}, \quad (5)$$

$$G_{15} = 1,91128 \cdot e^{0,0379\tau}, \quad (6)$$

$$G_{20} = 3,1594 \cdot e^{0,0452\tau}, \quad (7)$$

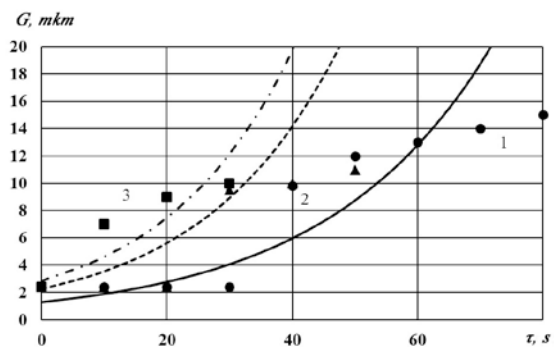
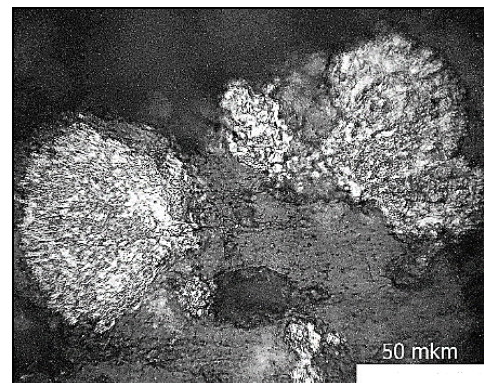


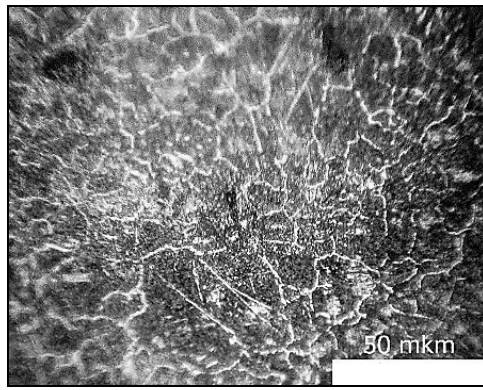
Fig. 5 Experimental values of grain size in consolidated Fe powder specimens after SPS with different heating rate and curves of their approximation

1 – heating rate of 10 °C/s; 2 – heating rate of 15 °C/s;
3 – heating rate of 20 °C/s

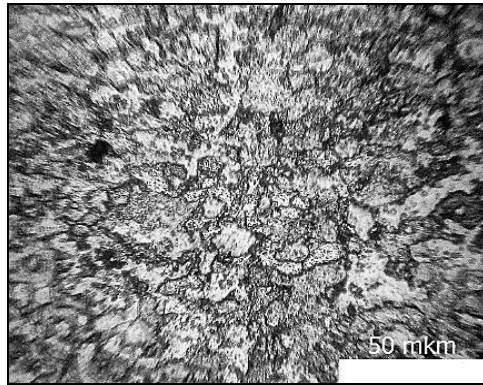
Obtained approximation evaluations allowed determination of A_0 and m coefficients at value of surface tension of $\alpha = 1.2 \text{ J/m}^2$ and activation energy of $Q_{cr} = 200 \text{ kJ/mol}$ for case of liquid phase presence ($m \rightarrow 1$) [4, 13,16]. Found values are $A_0 = 240 \text{ MPa}\cdot\text{s}$ and $m = 0,9$. This leads to a solution of differential equations system (1), which allows description of Fe powder behavior during SPS with different heating rates with 80 % accuracy (see Fig. 7 and Fig. 8).



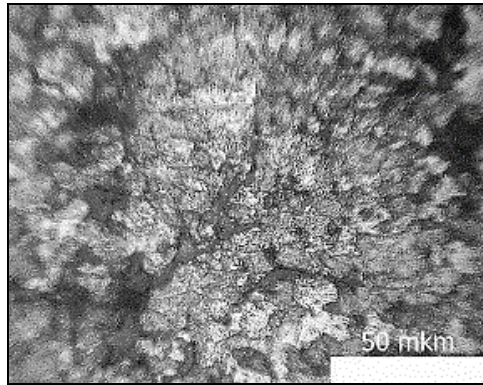
a



b



c



d

Fig. 4 Microstructures of initial Fe powder and consolidated Fe powder specimens after SPS with different heating rates, magnification $\times 450$
a – initial Fe powder; b – consolidated specimen, heating rate of 10 °C/s; c – consolidated specimen, heating rate of 15 °C/s; d – consolidated specimen, heating rate of 20 °C/s;

$G, \mu\text{m}$

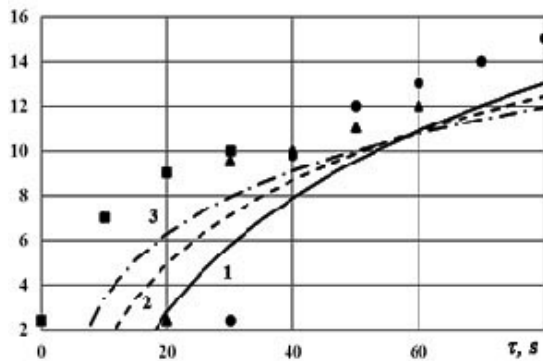


Fig. 7 Experimental values and theoretical curves of grain size in consolidated Fe powder specimens after SPS with different heating rate
1 – heating rate of 10 °C/s; 2 – heating rate of 15 °C/s; 3 – heating rate of 20 °C/s

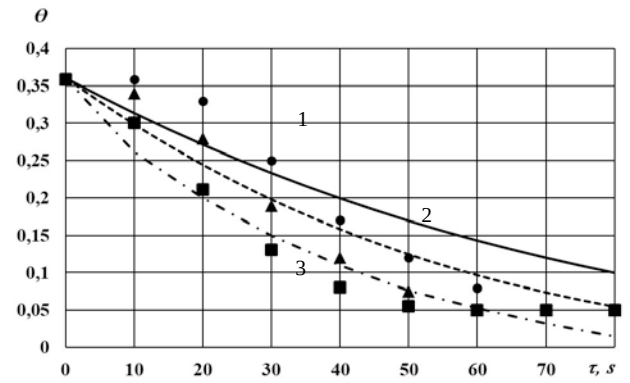


Fig. 8 Experimental values and theoretical curves of changes of porosity of consolidated Fe powder specimens after SPS with different heating rate
1 – 10 °C/s; 2 – 15 °C/s; 3 – 20 °C/s

Dependences of porosity changes during SPS with different heating rate, shown on Fig. 8, and A_0 coefficient allowed theoretical determination of changes of capillary pressure between Fe powder particles dependence on specimens density, based on continuum sintering theory. A model of densification of cylindric specimen in hard matrix under external pressure of 30 MPa was considered (see Fig. 9). Material was considered as having pores and linearly viscous non-porous phase.

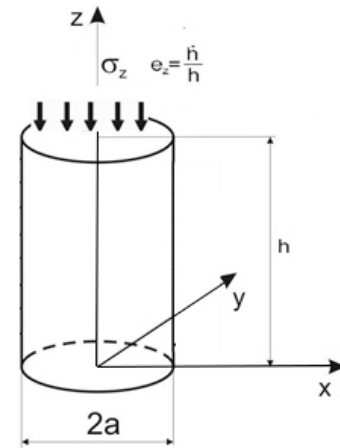


Fig. 9 Model of densification of cylindric specimen in hard matrix under external axial pressure

This model was described using evaluations (8–10) [3, 4, 17]:

$$\sigma_r = 2\eta \left(e_r - \frac{1}{3}e \right) + \zeta e + P_l, \quad (8)$$

$$\sigma_z = 2\eta \left(e_z - \frac{1}{3}e \right) + \zeta e + P_l, \quad (9)$$

$$e = \frac{1}{3}(e_z + 2e_r). \quad (10)$$

where η and ζ – coefficients of shear and volumetric viscosity respectively;

P_l – capillary pressure;

e – rate of volume change.

Behavior of consolidated material can be described by the following evaluations:

$$\eta = \eta_m (1 - \theta)^2, \quad (11)$$

$$\zeta = \frac{4}{3} \eta_m \frac{(1 - \theta)^3}{\theta}. \quad (12)$$

where η and ζ – shear and volumetric viscosities respectively;
 η_m – coefficient of shear viscosity, which, according to [4],
 can be represented as:

$$\eta_m = \frac{A_0}{2}. \quad (13)$$

Theoretical dependences of change of capillary pressure on porosity change during SPS with heating rates of 10 °C/s, 15 °C/s and 20 °C/s were obtained as a result (see Fig. 10). It is found out, that during Fe powders consolidation capillary pressure impacts their densification process only when total specimens density is higher than 80 %. Values of capillary pressure depend on heating rate and are 0.5 MPa for heating rate of 10 °C/s and 2 MPa for heating rate of 20 °C/s, which is due to formation of liquid phase and decrease of pores size, rate of excretion of which depends on heating rate.

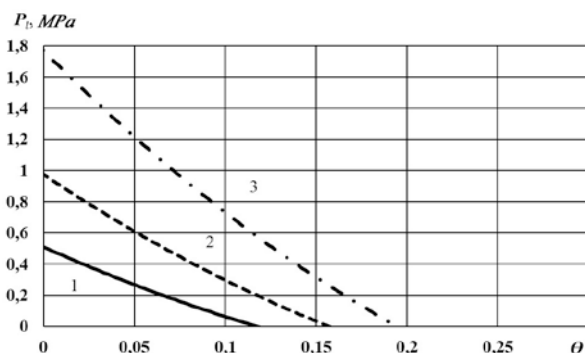


Fig. 10 Theoretical curves of changes of capillary pressure depending on porosity of consolidated Fe powder specimens after SPS with different heating rate
 1 – 10 °C/s; 2 – 15 °C/s; 3 – 20 °C/s

4. Conclusions

Regularities of impact of heating rate during SPS in range from 10 °C/s to 20 °C/s on Fe compact densification kinetics are found.

Basing on evaluations of continuum sintering theory, it is shown and experimentally confirmed that an increase of heating rate during Fe powders SPS from 10 °C/s to 20 °C/s leads to a decrease of densification time from 60 s to 40 s, increase of capillary pressure from 0.5 MPa to 2 MPa and increase of consolidated specimens hardness from 60 HRB to 85 HRB due to obtaining of more fine-grain structure.

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