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МАШИНСКИ ФАКУЛТЕТ, СКОПЈЕ, РЕПУБЛИКА МАКЕДОНИЈА

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С О Д Р Ж И Н А

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Original scientific paper

FUNCTION STRUCTURE ANALYSIS OF THE CURRENT PASSANGER VEHICLE BUMPERS

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A b s t r a c t: The basic purpose of the vehicle bumper is absorption of impact energy in the collision with another vehicle ***at smaller velocities.*** On the other side, numerous statistical data show that over 80% head-on collisions with serious or fatal injuries occur with impact speed range 40–60 km/h, which is considerably more than velocities issued by existing standards [3].

There is no doubt, that just now, the conditions for further high-quality development and improvement of the current and future passenger vehicle bumpers are “ripened” (independently of the drive system).

In the paper a modern designing approach for creating technical product with top performances is presented, by which the current passenger vehicle bumper is analyzed as a system.

The existence of the problem is noticed, and on the basis of those data the new task of the bumper is defined. The requirements list is assembled, with the help of abstraction the General Technical Function (GTF) of the new bumper is defined and finally (by division of GTF to partial and elementary functions) the functional structure of the bumper is elaborated.

Key words: requirements list (RL); abstraction; General Technical Function (GTF); functional structure

1. INTRODUCTION – DEFINING OF THE PROBLEM

The current passenger vehicle is a complex technical system, realized as a result of a hundred years evolution and team work of highly qualified experts, whose mutual goal is satisfaction of a set of most different requirements.

First vehicles did not had a bumper. The increase in number of vehicles, and the increase of the possibility of unpleasant collision of two vehicles, actualizes the need of existence of an element sufficiently strong to reduce the damage of ***less heavy collisions,*** and which is at the same time cheap with simple construction and appropriate appearance.

Up to 70-ies, the vehicle bumpers were produced as separate elements of the outside vehicle equipment, from hard-chrome plated sheet of steel. These steel bumpers in the 70-ies were replaced by nonmetal bumpers. The first nonmetal bumpers were made of polyester (Renault 5), and then polyester was replaced of thermoplastics [6].

The last few years, the current passenger vehicle bumper, became an integral part of the body, and together with it, it makes functional and aesthetic entirety. Concurrently with the evolution of the shape, the evolution of applied materials happens as well [4, 5, 6].

The technical function of the bumper is becoming more and more complex, and just now it is not a simple detail, but ***system*** with numerous functions. Parallel with the evolution of the bumper, numerous limitations as standards, laws, regulations and recommendations were developed, too.

International regulations define the term ***bumper*** as: an outside protective device, which is set at the front and rear end of the vehicle and which is so constructed to allow touches and little shocks, without causing bigger damage.

The basic purpose of a vehicle bumper is absorption of impact energy in the collision with another vehicle, at ***smaller velocities.*** On the other side, numerous statistical data show that over 80% frontal collisions with serious or fatal injuries occur with impact speed range 40–60 km/h, which is considerably more from velocities issued by existing standards.

There is no doubt however, that just now, the conditions for further high-quality development and improvement of the current and future passenger vehicle bumpers are "ripened" (independently of the drive system).

In the paper, on the basis of the existing problem, the passenger vehicle bumper is analyzed as a system, and the new task of the bumper is defined. The new requirements list is assembled with abstraction, the new General Technical Function (GTF) of the new bumper is defined, and finally, the new functional structure of the bumper is elaborated.

2. DEFINING OF THE NEW GENERAL TECHNICAL FUNCTION

During the evolution of the passenger vehicles, the bumper also evolves, from simple shape to more and more important subsystem with many functions: ***partial prevention*** from front and rear collision, reducing of ***the pedestrian injuries*** in an accidental contact with vehicle, reducing of the ***aerodynamic resistances*** and of the ***consumption of fuel*** , reducing of the ***noise*** of air flow, correct ***directing of air flow*** under and over the body, ***carrying*** of various signal and other elements, important ***aesthetic*** function, etc.

In accordance with the usual procedure [1, 7, 8], all the requirements resulted from a clarified task, as well as the existing limitations are classified according to the requirement category (yes/no demands, tolerated minimal demands and wishes) (see Table 1).

The requirements list is a base for many designer's activities. On the occasion of new product creation, the abstraction of the task to reduce the problem to its most essence form has a first-rate importance. (By abstraction of the requirements, the problem reduces to general, solution-neutral terms. Abstracting to identify the essential problem by reducing it to its most essential form).

According to the usual methodical treatment, the process of abstraction is done in five steps [8]:

1) omitting of all wishes,

2) omitting unimportant demands,

3) transforming demands from quantitative to qualitative,

4) broadening the designer's knowledge and the level of abstraction,

5) stating the problem in solution-neutral terms (to generalize the task, i.e. formulation of GTF).

Many inexperienced designers perceive the fourth step of abstraction as formality. This position may bring many difficulties in the defining of GTF. For that reason, in this concrete design task, the fourth step of abstraction was specially attended. Three very important knowledges resulted from this analysis:

a) The basic principle of passive safety is to have the passengers retained, i.e. well fixated at their seats during the collision. In a sitting position the human body, in the direction back-belly (or vice versa), can endure acceleration of 15 g for several seconds, up to 30 g for 0.1 second. However, in a collision with a bigger velocity, the level of deceleration is even to –40 g, so that normal passengers could not endure it, and their detainment in the seats ***does not have any effect.*** In that case, plastique deformation could play "leading role" in decreasing the acceleration at a bearable level.

b) It is noticeable that all cited effects and requirements can be attained by forming the ends of the body in such way, that these ends have the bumper features, i.e. the bumper is unnecessary. However, it is as clear as day that any small damages will be impractical for frequent repairs. It is much better, if a changeable part exists, which can be changed and frequently, if it is necessary, without any influence at the body entirety.

c) At this step, it can be recognized that the bumper performs two kinds of functions:

– ***Permanent***, i.e. providing of aesthetic, attractive and aerodynamic form of the body, reducing of the aerodynamic resistance, reducing of the noise of wind, reducing of the fuel consumption, carrying of various signal and light devices, directing of the air flow around, over, under and in the vehicle. This function is performed by carefully modelling and shaping, and by selection and processing suitable materials as well.

T a b l e 1

Requirements list for the current passenger vehicle bumpers

	REQUIREMENT	y/n	tol.	wis.
1.	**Geometry:** The bumper dimensions are dependent on vehicle dimensions and harmoniously and esthetically pleasant, merge with the car body in aerodynamical, functional and esthetic ensemble:	*		
	– the bumper is changeable exterior part of the body (front or rear),	*		
	– the upper line of the bumper coincides with the bottom line of the headlights,			*
	– the car passing (front approach angle min. 16 deg., and rear exit angle min.10 deg.) in accordance with SAE J 689,		*	
	– the size of the aperture for the passing air in the refrigerated device is min 0,08…0,1 m^2.		*	
2.	**Kinematics:** In activated state, the bumper can endure deformation or carry out limited rectilinear motion in the impact direction:			
	– the size of motion is limited by the condition, that in the collision there must not be any damages of the vital vehicle systems (fuel, cooling, braking, steering, and light and signal devices),	*		
	– the size of the motion is in accordance with ECE 42 (min. 70 mm),		*	
	– the security velocity range, for the bumper to be functional is 35…65 km/h.		*	
3.	**Forces, loading:** The basic bumper loading is an impact force from the collision with enother vehicle or an obstacle. The impact direction may be from the left side, from the right or head-on collision:			
	– the forces are a result of impact with another solid at 35…65 km/h velocity,		*	
	– reduction of the level of deceleration during collision up to –7 g,		*	
	– the bumper mass ought to be as small as possible, max. 1 % of the vehicle mass,		*	
	– aerodynamical form, small aerodynamical resistance (as smaller as possible *Cx* factor),		*	
	– the bumper loadings are transferred to the body by support system which accepts the collision forces and is specially designed for that purpose.			*
4.	**Energy:** The bumper functions upon the principle converting kinetic impact energy of vehicles in motion to deformation labor for deforming of the bumper, or some other kind of energy (friction-warmth, hydropneumatical damping, shearing of the adjusted screws):	*		
	– a favorable aerodynamical shape directly influences the engine fuel consumption.			*
5.	**Material:** It is possible to use all customary technical materials (alloy steels, stainless steels, aluminium or magnesium alloys, thermoplastics, polypropylene – PP+EPDM, PC+PBT), which have to require mechanical features, high dimensional stability at high and low temperatures (–40…+80 ^{o}C), high abrasion resistance, impact-resistant, resistance to chemicals and weather), possibility of shaping with productive and rational methods and treatments (welding, gluing, squirting) and economical recycling.	*		
6.	**Ergonomy:** Easy to handle, i.e., simple montage and changing (a damaged bumper with new one), or simply regeneration of an activated bumper:			*
	– rounded edges and surfaces to minimize danger of injuring during "contact" with pedestrians. Outside edges have to be rounded with r_{min} = 5 mm to avoid injures at handling,		*	
	– serviceability, recycling, varnishing with usual varnishes,			*
	– nonflammable, incombustible, nonpoisonous (when burning, it doesn't produce toxic agents),	*		
	– aerodynamical form for decreasing the air flow noise and optimal air flow directing under, around and over the body, as well as, along holes for air to cool (engine, brakes, air conditioner) and so on…		*	
7.	**Fabrication and control:** Simple design solution, possibility of shaping with productive and rational methods and treatments (welding, gluing, squirting):			*
	– satisfaction law obligation from rules ECE 42, ECE 26, ECE 48, FMVSS581, SAE J 689,	*		
	– surface for license plates of the vehicle (CEE 520*120 mm, USA 12"*6"),	*		
	– a hole for a hook for pulling of the vehicle,			*
	– a hole for the exhaust pipe from the engine (min. clearance 30 mm because of vibrations),			*
	– placing a lighting in accordance with the rule ECE 48 (a positional light, a blinking light of the turn signal, a fog light, a light for rear license plate),	*		
	– placing a pump for head-light washer.	*		
8.	**Assembly and transport:** Simple montage and/or regeneration of the working ability of the activated bumper:	*		
	– transport in wood container and plastic bag, for eliminating the possibility of mechanical injuries, the influence of the humidity and of the corrosive agents.			*
9.	**Exploitation:** The bumper must be made from material resistant at temperature fluctuation (–40…+80 ^{o}C), and capable of retention of dimensions stability:	*		
	– resistance at usual chemical agents (salt, oils, acids, humidity, antifreeze, different industrial gases),	*		
	– cleaning with water and usual body care shampoo,			*
	– repair of damaged surfaces with usual materials for body repair,			*
	– possibility for varnishing the bumper in two colors.			*
10.	**Costs:** A price of a bumper ready for montage max. 0,2 % from the vehicle worth.		*	

– ***Incident***, i.e. reducing of the level of deceleration during the collision, kinetic energy absorption, passenger protection from fatal injuries, protection of vital vehicle systems (fuel, cooling, braking, steering), and reducing pedestrian injuries on the occasion of possible contact with them. This function is accomplished by creating the kind of forms capable to perform safety requirements without jeopardizing the permanent function and the costs of the vehicle.

In the fifth step, the GTF of the bumper is defined in the following form: *To make a concept of a changeable exterior part of the body with variable dimensions and aerodynamical form from material resistant at temperature fluctuation and different chemical agents, which assured dimension stability, which enables converting of kinetic impact energy to deformation labor or some other kind of energy, with the task of passenger protection from injuries in a collision, as well as protection of the vital vehicle systems (fuel, cooling, braking, steering) and reducing of pedestrian injuries on the occasion of possible contact with them.*

3. CREATING OF THE NEW FUNCTIONAL STRUCTURE OF THE BUMPER

The GTF formulation gives us the possibility of creating the bumper functional structure. A rough structure can be formulated by just two elements, according with the two kinds of established functions performed by the bumper [9]:

– ***Permanent*** (providing of aesthetic, attractive and aerodynamic form of the body, reducing of the aerodynamic resistance, reducing of the noise of wind, reducing of the fuel consumption, carrying different signal and light equipment, directing of the air flow around, over, under and in the vehicle).

– ***Incident*** (reducing the level of deceleration during the collision, kinetic energy absorption, passenger protection from fatal injuries, protection vital vehicle systems for fuel, cooling, braking, steering and reducing pedestrian injuries on the occasion of possible contact with them, and transfer the bumper loadings to the body by the supporting structure.

It is easy to see that incident function can be divided in two parts, dependently on the level of the realized forces, on the occasion of vehicle impact with another solid. Consequently, we have a function for small velocities and a function that bumper realizes at greater velocities. An analysis of the rough structure gives more detailed structure of partial functions (Fig. 1) [9].

By permanent function, we obtain: decreasing of the negative aerodynamically effects (resistance motion decreasing, decreasing of the noise of wind and fuel consumption), realization of likeable appearance of the body for conquering the customer's heart, regular directing of the air flow for cooling the engine, the brakes, the condeser of air conditioner and so on.

By the incident function we obtain: amortization of smaller impacts, decreasing the impact consequences in a possible contact with pedestrians, reduction of deceleration, impact absorption, vehicle vital systems protection, transfer of the loadings to the supporting vehicle structure. Such formed rough functional structure, makes it possible for us to define the executors of the partial functions, i.e. to discover more suitable executors for each partial function (Table 2) [9].

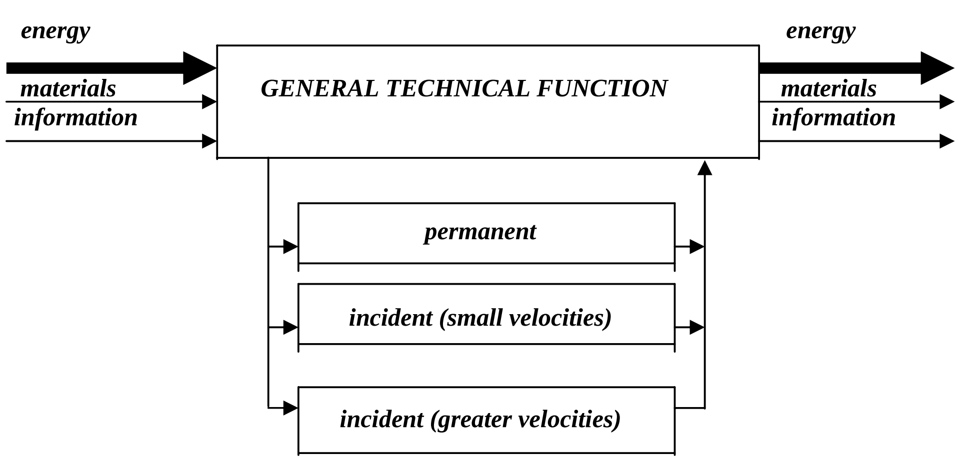

Fig. 1. Rough structure of bumper partial functions

T a b l e 2

Possible executors of the separate bumper partial function

Function	Executor
Reducing of harmful aerodynamical effects (resistances, noise, turbulence and so on).	Aerodynamical form obtained with CAD program module modelling, FEM analysis and with testing and checking in an aerotunnel.
Creating attractive looks to "conquer customer's heart".	Attractive form, obtained with CAD program module modelling (NURBS, B-spline…).
Supplying a suitable form for regular directing of the air flow (for cooling the engine, the brakes, the condenser of airconditioner and so on.	System of spoilers and suitably placed openings and slits in the bumper structure.
Decreasing of the impact consequences in a possible contact with pedestrians.	Application of rounded forms, optimal locating and application of soft materials.
Amortization of smaller impacts.	Elastic deformation of the bumper structure.
Reducing of the level of deceleration, during the collision to a bearable level.	Elastic and/or plastic deformation of the bumper structure.
Absorption and amortization of the kinetic energy of bigger impacts (damping).	Plastic (permanent) deformation of the bumper structure.
Protection of the vehicle vital systems from collision consequences.	Elastic deformation of the bumper structure, and limitation of the bumper operation (passage).
Transfer of loadings from the bumper to the supporting vehicle structure.	Specialy designed system of supports and amortisseurs.

4. DEFINING THE EXECUTORS OF THE SEPARATE BUMPER PARTIAL FUNCTIONS

It is customary, the functional structure with executors of partial functions all together to be presented graphically (Fig. 2). The separate partial function and their principled executors are presented in Table 2. Further analysis of the content of the table shows the possibilities for detailed defining of the separate executors.

The possible executors are given in Table 2 and the finite form of the expanded functional structure is given in Figure 2.

5. CONCLUSION

On the occasion of new product creation or an essential improving of an existing, it is necessary to perform a complete methodical concepting, i.e., to analyze the functional structure of the technical product as it is done in the paper.

The further process of concepting has a generally scheme-like character: from the created functional structure, with the help of a morphological matrix, more reliable and real feasible solution variants are obtained, and finally, by using an evaluation system, the optimal concepting variant is selected. This variant will be developed (worked out) by the embodiment designing and by constructive elaboration. This part goes out of the intentions of this paper and can be found in the cited references [1, 8, 10].

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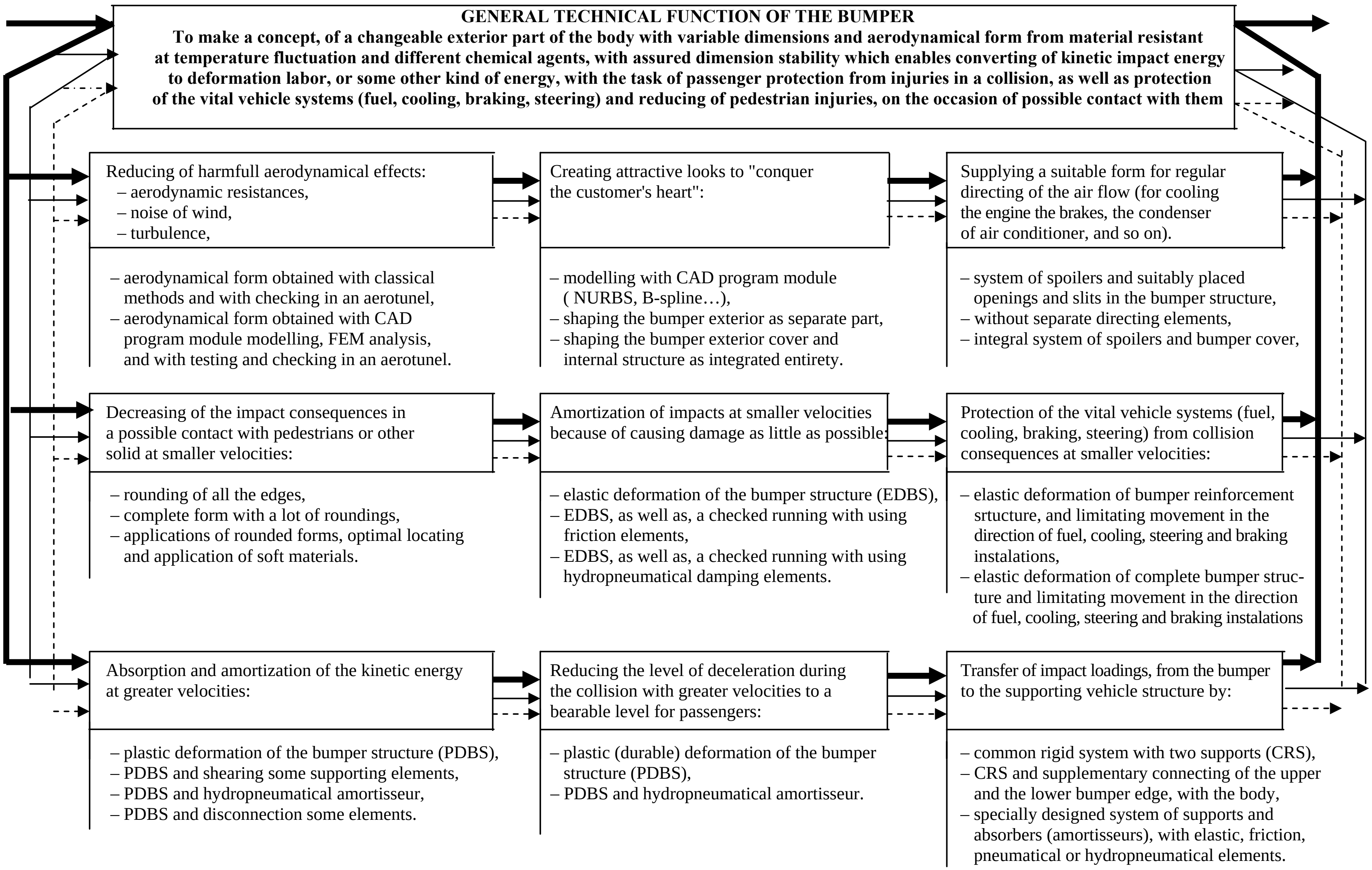

Fig. 2. The functional structure of the bumper

Р е з и м е

АНАЛИЗА НА ФУНКЦИОНАЛНАТА СТРУКТУРА НА БРАНИЦИ НА СОВРЕМЕНИ ПАТНИЧКИ АВТОМОБИЛИ

Владимир Андоновиќ, Александар Андоновиќ

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Клучни зборови: браник; апстрахирање; техничка функција; функционална структура

Основната намена на браникот е апсорпција на ударната енергија при судир со друго возило при помали брзини. Од друга страна, статистичките податоци покажуваат дека повеќе од 80% челни судири со сериозни или фатални последици се случуваат при брзини од 40 до 60 km/h, што е значително повеќе од брзините на кои постојните стандарди се повикуваат. Нема сомневање дека созреале услови за понатамошно квалитетно усовршување и подобрување на браниците на современите и идните патнички автомобили (независно од видот на погонот). Во трудот е прикажан модерен конструктивен приод кон создавање технички производ со врвни перформанси, при што браникот на патничко моторно возило се анализира како систем. Уочено е постоење на проблемот и врз основа на тоа е дефинирана нова задача за браникот. Составена е листа на побарувања, со апстрахирање е дефинирана општа техничка функција на новиот браник (ОТФ), а со нејзиното разложување на парцијални и елементарни функции е разработена функционална структура на браникот.

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Original scientific paper

CALCULATION OF COMPLEX PROFILES IN TECHNICAL MECHANICS USING "TMI" PROGRAM

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A b s t r a c t: This article presents specific method of calculations in technical mechanics, especially in mechanical engineering. Worldwide in technical faculties and schools curriculum of technical mechanics exist. This is the basis for other further disciplines in the knowledge of any mechanical engineer. The statics, kinematics, dynamics and the strength theory, vibrations are parts of the technical mechanics. Some parameters like a position of center of gravity, areas, axial and polar moments of inertia, resistance moments of inertia are the most frequently used parameters in mechanics. There are many exact equations for their calculations. Sometimes, for complex profiles special methods of calculation and determination exist.

This paper shows the abilities of computer program "TMI.EXE" based on the "SCREEN CONTACT" methodology [1], for parameter calculation of complex profiles in the machines.

Key words: center of gravity; moment of inertia; profile; mechanics; machines

1. INTRODUCTION

The center of gravity and moment of inertia of the regular geometrical plane profiles are easy to calculate. There are equations for exact calculation. However, the machines and other technical systems are often assembled of elements with very complex shapes (irregular profiles) (Fig.1, Fig. 2).

These elements are produced by casting, forging, welding, etc. It is important to calculate such profile of axial or tangential stress. The parameters like the position of center of gravity, and moments of inertia are necessary for stress calculation, and also for calculation of real security factor, and at least to give a compact design.

Now-a-days various programs are offered based on finite elements methodology. However, all of these programs are expensive, hulk and very complicated for use.

There are many parameters needed in mechanics:

A (cm^2) – cross-section area of the profile,

m_1 (kg/m) – mass of 1 m long beam of this profile,

x_T, y_T (cm) – position of center of gravity of the profile,

x_{max}, y_{max} (cm) – maximal distances of the contour of the profile,

r_{max} (cm) – maximal absolute distance between center of gravity and contour,

S_x, S_y (cm^3) – statical moments of the profile,

I_x, I_y, I_p (cm^4) – axial and polar moments of inertia of the profile,

I_{xy} (cm^4) – centrifugal moment of inertia,

I_{max}, I_{min} (cm^4) – capital (main) moments of inertia of the profile,

i_{min}, i_{max} (cm) – minimal and maximal radius of inertia of the profile,

φ (grad) – capital (main) course of inertia of the profile,

W_x, W_y, W_p (cm^3) – resistance moment of inertia of the profile.

Geometrical and other values of the parameters can be presented in millimeters (mm), (mm^2), (mm^3), (mm^4) or meters (m), (m^2), etc.

Along with the aforementioned parameters there is another information presented: deformaion of a 1 meter long beam of this profile under the force of 1 kN. This information is included in

TMI.EXE program, but information of the polar resistance moment of inertia W_p (cm^3) is not included, because of several reasons.

2. COMPLEX SHAPES OF ELEMENTS IN MECHANICAL ENGINEERING

In many cases machines are consisted of complex elements. There are economic reasons: saving material and decreasing of weight. Instead of simplicity – the present modern designs are very complicated shapes.

Older designs are also very complex [3], (Fig. 1).

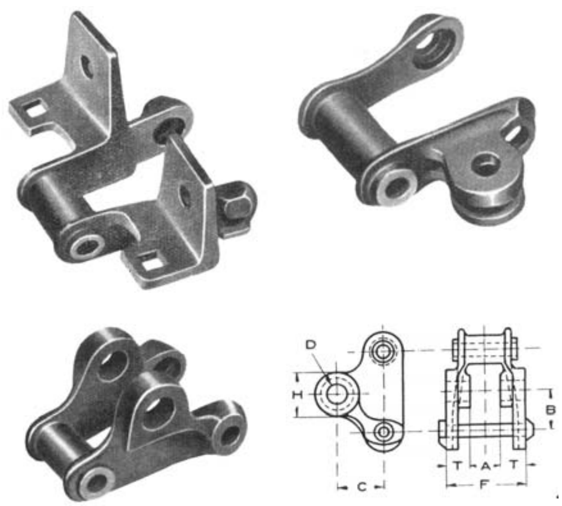

Fig. 1. Some complex shaped elements used in chain conveyors and chain bucket excavators

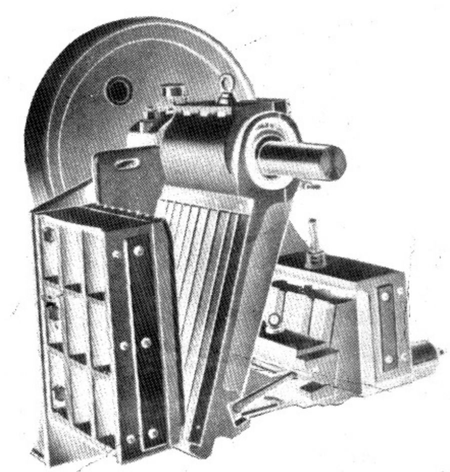

Fig. 2. Jaw crushers consisted of many complex shaped elements given by casting

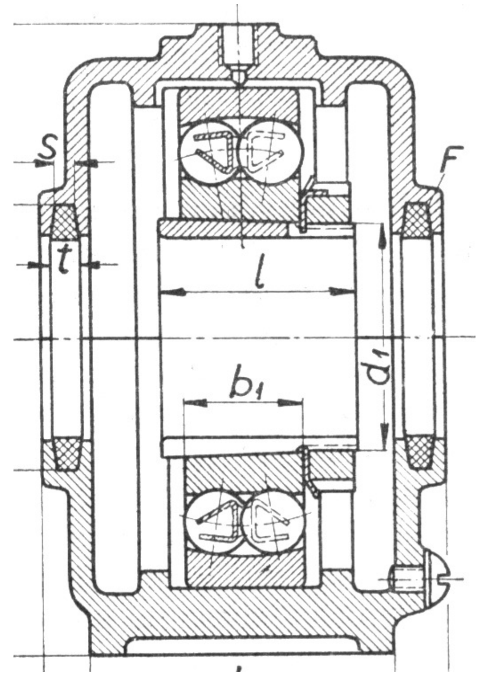

Fig. 3. Journal bearing with removable cap [4]

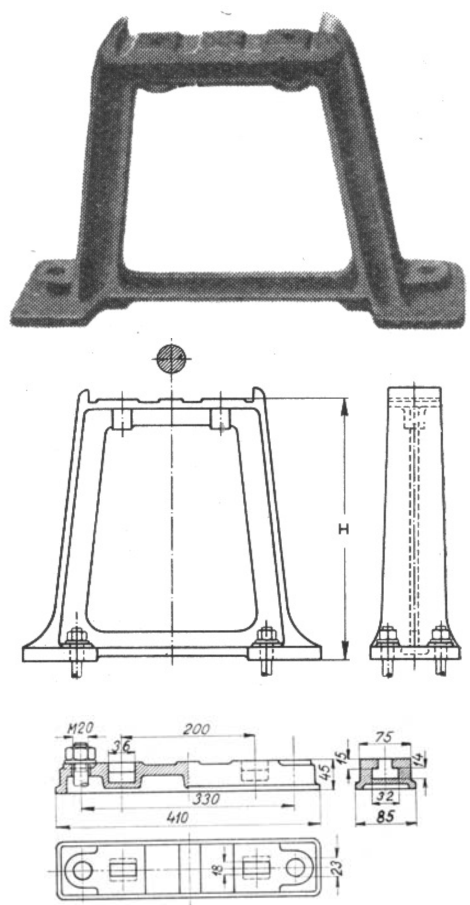

Fig. 4. Floor frames for journal bearing [4]

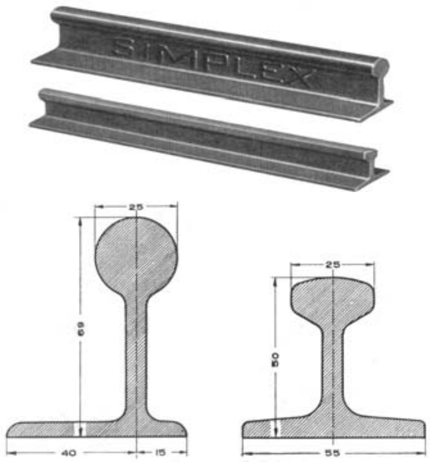

Fig. 5. Special rails for conveyors and trains

Many of the designed parts have very complex profile. For stress analysis or determination of factor of safety it is necessary to calculate some of parameters mentioned above (Fig. 3, 4, 5).

For example, fixed or movable jaw of crushers shown on the Fig. 2, are under strong bending stress [5, 6]. Thus, it is required to calculate the resistance moment of inertia W_x (cm^3). It is the same with the rails, frames, journal bearings, splined shaft (Appendix D), etc.

3. FEATURES OF THE TMI PROGRAM

The TMI program is a C/C++ computer program based on the "SCREEN CONTACT" methodology introduced in dissertation [1]. The main principles of this methodology are contacts between pixels of two or more different colored objects. The motion of the objects on the computer screen produces contacts or/and overlapping. The first state of a contact is defined as an overlapping of only one pixel of the one object with one pixel of the other object. Actually, this is the most important status between objects. The positions of the objects in this instant are very important information.

This methodology is used in kinematic analysis of complex mechanisms containing kinematic groups of 3-rd class (trada), and especially in the mechanisms with higher kinematic pairs (gears, chains, geneve-mechanisms, cams, etc.).

However, some variations of this methodology can be used in other different tasks. Calculations of parameters of complex profiles enumerated above can be implemented by program based on this methodology.

The main principle of TMI is the "**pixel counting**" then, the pixels of the same color are made a conversion in real coordinate (ratio), and other computation of different arithmetic operations (adding, multiplication, dividing, etc.), for any pixel or group of pixels.

However, at first, the objects must be shown on the screen. The TMI program enables to draw or scan the drawings in BMP graphic format. Then, the image can be colored with some of the 16 VGA colors (the old version is under DOS).

Then program allows import (load) the BMP image, and after input the RATIO, the program gives the results (Appendices A, B, C).

The inside region of the image must be of one color, and other information (dimension, arrows, lines, accessories, etc.) must be differently colored. The precision of the TMI program is tested and analyzed in [2]. This program gives very precise results when the image on the screen is larger. The complexity of the profile is not important. The figures are showing very complex profiles of machine elements used in the industries.

4. SOME PRACTICAL EXAMPLES

The examples presented here are very often used in the industries (Fig. 6).

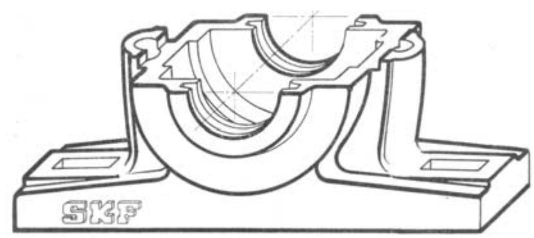

Fig. 6. Basis of a journal bearing with removable cap

The load of the working forces on the journal bearing with removable cap, creates a bending stress of the profiles denoted in the Fig. 7.

The resistance moment of inertia

$$W_x = (b\,h^2)/6\ (\mathrm{cm}^3).$$

However, it is clear that the profile is not a rectangle, but it is a very complex image. One way to calculate is to assume the approximate height h (cm) or h_2 (cm). But there is no accuracy (precision).

Instead of such arbitrary approximation, the TMI program offers great accuracy.

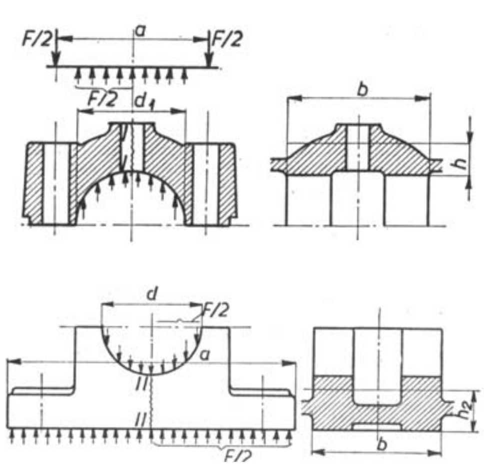

Fig. 7. Approximate determination of resistance moment of inertia of journal bearing

Figure 8 shows a profile of basis of the journal bearing, scanned and prepared for calculation. The results of this calculation by TMI are presented in the Appendix A.

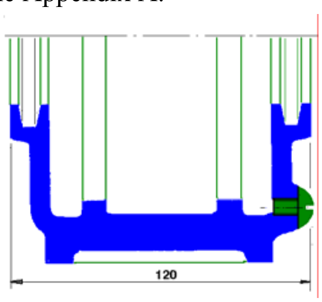

Fig. 8. The profile of the journal bearing basis prepared for program TMI (from Fig. 3)

The standard rails and also the special rails are intensively loaded on the bending stress. Figure 5 shows the special rails used in chain overhead conveyors. Appendix B presents the results of these special rails.

Appendix C presents the TMI results of jaw crushers profile (movable jaw), and Appendix D shows the results of a splined shaft with six splines.

5. CONCLUSION

The TMI program is a very compact program, written in C/C++ program language in 1995 [1]. Although it works under DOS operating system, similar to other DOS applications, it can be easily adjusted for Windows operating system. Because the procedure is very simple, the manual (direction) is very short – it takes only one page text. It needs only one hour for training. This paper presents only a part of machine elements of complex profiles, but it can be used not only in mechanical engineering, but also in construction, mining, etc. The "Screen Contact" is computer based methodology which also gives many opportunities for variety calculations in mechanics.

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Р е з и м е

ПРЕСМЕТКИ НА СЛОЖЕНИ ПРОФИЛИ ВО ТЕХНИЧКАТА МЕХАНИКА СО КОРИСТЕЊЕ НА ПРОГРАМАТА "TMI"

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Клучни зборови: тежиште; момент на инерција; пресек; механика; машина

Овој труд презентира еден специфичен метод за пресметки во техничката механика, особено во областа на машинството. На сите технички факултети и училишта во светот постои дисциплина техничка ме-

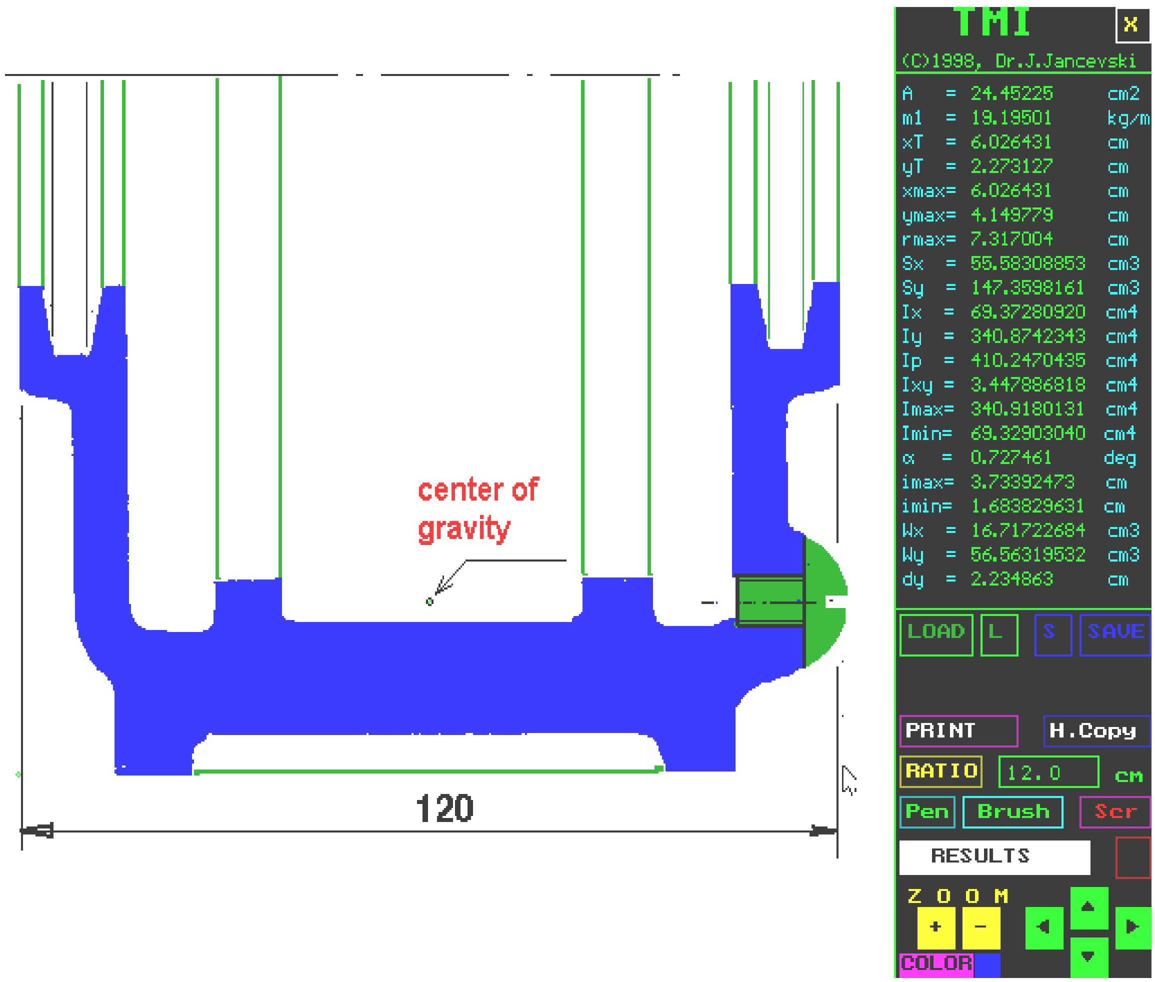
ханика. Таа е основа за сите други наредни предмети потребни на еден машински инженер. Статиката, кинематиката, динамиката, јакоста на материјалите, како и теоријата на осцилациите, всушност се делови на техничката механика.

Некои параметри и величини како што се тежиштата, површините на напречните пресеци, аксијалните и поларните моменти на инерција, отпорните моменти, се многу често користени во механиката. Постојат точни равенки за нивно пресметување. Сепак, понекогаш за сложени профили (пресеци) постојат специјални методи за нивно утврдување и пресметка.

Овој труд ги прикажува можностите на компјутерската програма TMI.exe која е заснована на методот „екрански контакти“ [1], според кој се пресметуваат величини на сложени профили кај машините.

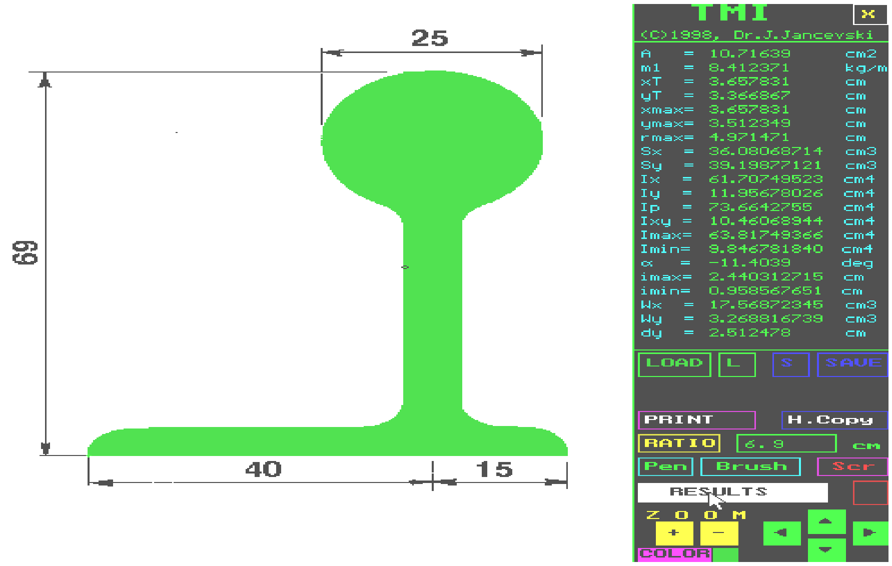
APPENDIX A

TMI Results – Basis of journal bearing



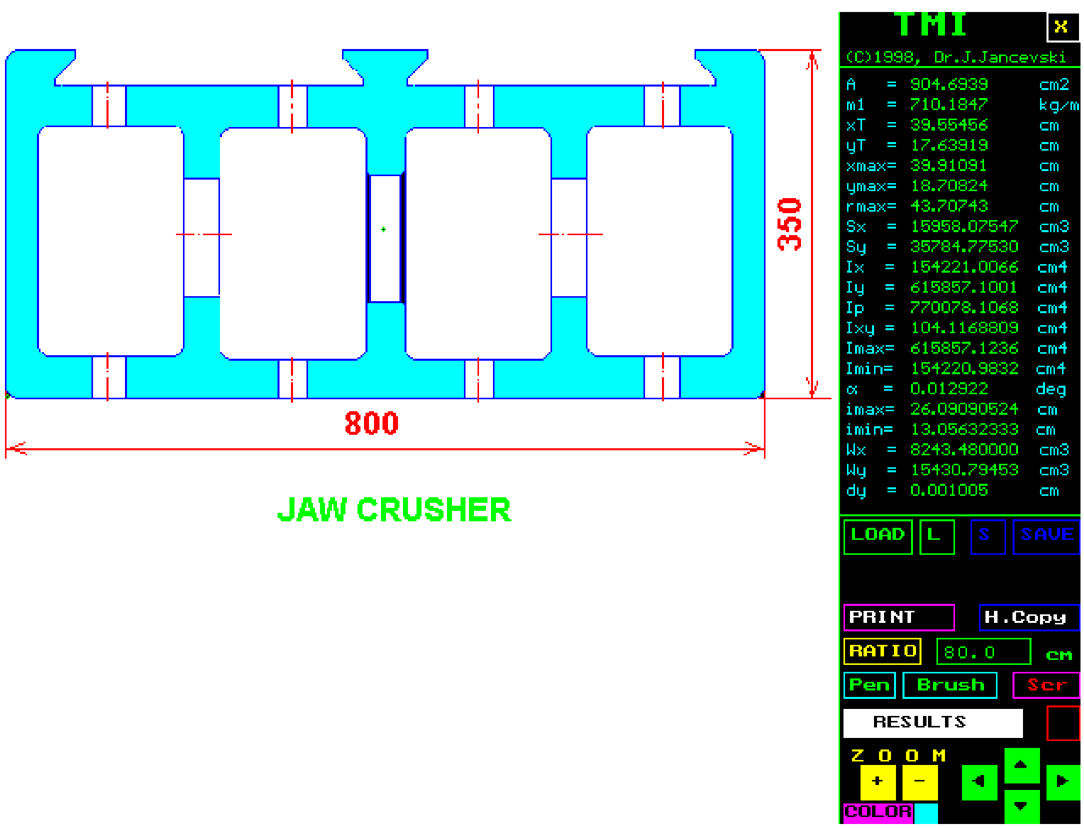
APPENDIX B

TMI Results – Special rail for conveyors



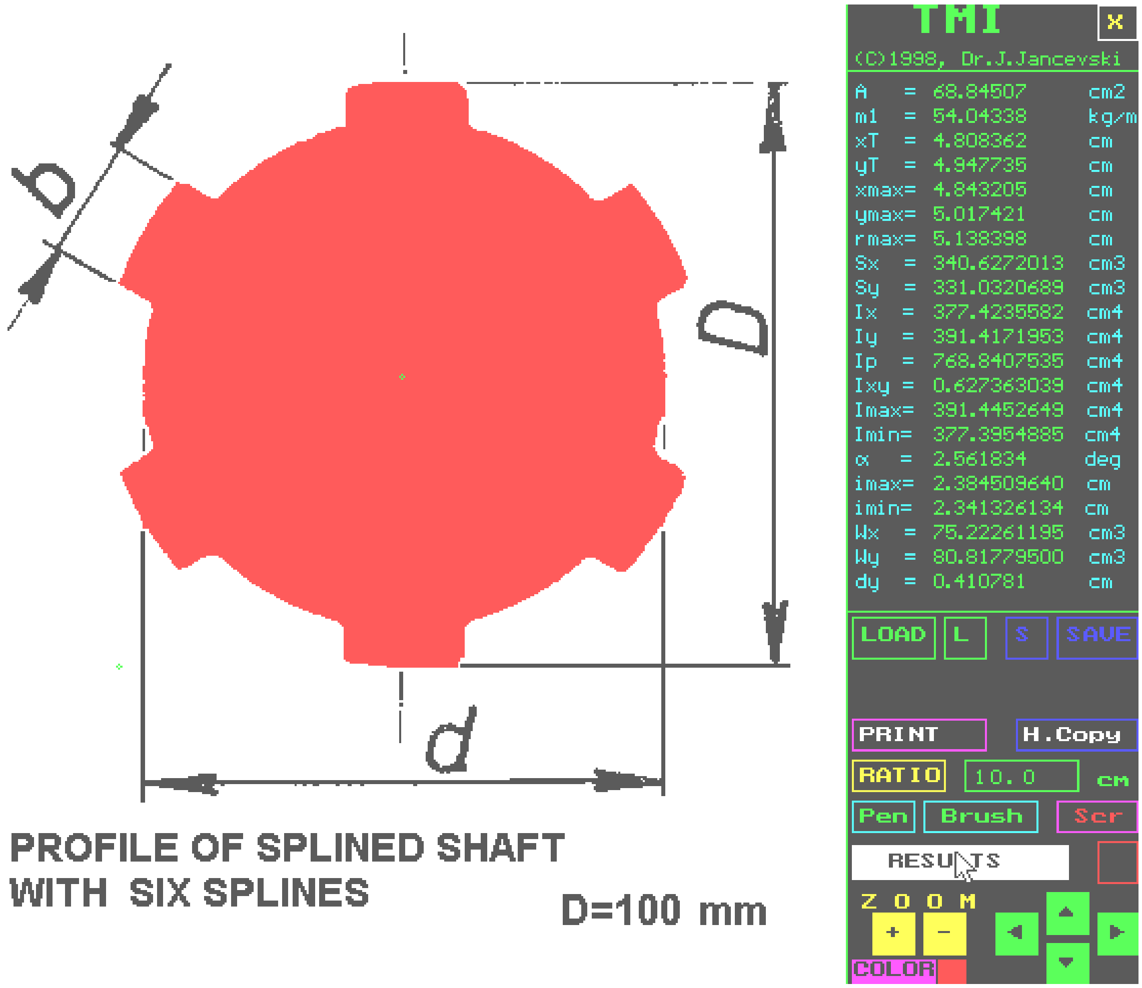
AHPPENDIX C

TMI Results – Movable part of a jaw crusher



APPENDIX D

TMI Results – Profile of a splined shaft with six splines



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Original scientific paper

RESISTANCE TO ORGANIZATIONAL CHANGE VERSUS MANAGEMENT STYLES ACCORDING TO ADIZES

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A b s t r a c t: Success of organizational change process is in direct connection with the resistance that appears to that process and the issues related to successfully overcoming that resistance. For successfully overcoming resistance to organizational change the managers or the initiators of those changes must predict the reaction of the people in the organizational system to the proposed changes. Therefore, in this paper the authors represent the results of analysis of the reaction of managers to the organizational change against management styles according to Adizes. The research is obtained with a questionnaire in 50 enterprises in the Republic of Macedonia.

Key words: organizational change; resistance to change; reaction to change; management styles

1. INTRODUCTION AND THEORY REVIEW

For this part of paper literature that is cited in reference section [1], [2], [3] is used. Instead of nonmanagerial staff in one organizational system resistance to organizational change may come from the organizational members from management levels. The initiators of organizational changes may predict resistant behavior to change of organizational members in many ways. One of those ways that are usually in use is the management style. Ichak Adizes derives four different roles that separate every management style. With their combination, we may predict the behavior of managers about accepting or resisting the change process. The basic management styles according to Adizes are:

(P) – Producer

(A) – Administrator

(E) – Entrepreneur

(I) – Integrator

According to Adizes these styles present the four roles that are need in management. The rule of the *producer* in management is rule that should produce results through products and services. The role of the *administrator* is rule that provides implementation of politics and procedures. The administrator is the one who does not produce the results, but produces effective and efficient functioning of the whole system. The role of the *entrepreneur* provides new ideas and initiatives. This role is the one that initiates organizational changes and that shows the direction in which the organizational system must go in order to adapt to the changed environment. The role of the *integrator* is the last role in management according to Adizes. Integrators provide teamwork, integrate people and provide perfect functioning of the system from people's viewpoint.

The ideal manager is the manager who can incorporate all these roles in his/her management style (PAEI). This type of manager is impossible to find in practice. The useless manager is manager who has neither of these roles in his/her management style (_ _ _ _). Emphasizing one role in regard of other role and their combination provide different management styles that have different names according to Adizes. For example, Paei style presents producer, while P_ _ _ style presents recluse rider. In Table 1 some of the characteristic management styles according to Adizes with basic characteristic of that style and way of respond to organizational change are given.

Table 1

Management styles according to Adizes and (reaction) resistance to organizational change

Code	Management styles	Respond to change
----	Useless man	He/She does **not resist** the change because he/she is disinterested in it.
P---	Recluse rider	He/She **resists the change** because he/she has not got enough time. He/She **accepts the change** as he/she provides fast results.
-A--	Bureaucrat	He/She **resists the change** because he/she fears from loosing control.
--E-	Incendiary	He/She **accepts the changes** if those changes are his/her initiative. He/She **resists** if he/she did not initiatie it.
---I	Super follower	He/She **accepts** if they increase their role in solving of conflicts and not threat the cohesion. He/She **resists** if with the change is not increasing conflict solving and cohesion.
Paei	Producer	Because of the expression of the role of producer, he/she **resists** to change that did not give fast results.
pAei	Administrator	Expressing role of administration provides **resistance** to change that made destroying stability of the organization.
paEi	Entrepreneur	He/She **accepts** the change when it comes from them. He/She **resists** when change provides increasing of the rules, politics, and procedures in working.
paeI	Integrator	He/She **resists the change** that destroys social relationship because of expressed role of integration.

Looking this table we may conclude that only managers with entrepreneurial style of management are those who give active support to organizational change.

2. ANALYSIS OF MANAGEMENT STYLES VERSUS RESISTANCE TO ORGANIZATIONAL CHANGE

The research obtained here was conducted with one questionnaire that was given to 62 managers in 50 enterprises in the Republic of Macedonia. The questionnaires include questions which help us identify management style of the responders of the questionnaire and the way of reaction (potential resistance) of responders to organizational change. The statistical sample for this research obtains managers from top level, middle managers and first-line managers. The research was realized in the period of December 2005 to July 2006. The questionnaire for this segment of research obtains two types of question:

1. Questions about analysis of reaction of managers to organizational change [4].

2. Questions about analysis of management styles of responders of questionnaire [5].

For the first type of questions related to the reaction of the managers to the organizational change are used questionnaire RTCI – Reaction to Change Inventory. The reaction that was obtained in questions about analysis of people's reaction to change is:

1. Great change support.
2. Mean change support.
3. Inferior behavior to change.
4. Mean change resistance.
5. Great change resistance.

The second type of questions from this analysis related to management styles conducted questions which help us identify what style of management the responder of questionnaire has. A management style that is analyzed in this research was management styles according to Adizes, that was obtained in the introduction text of this paper.

These two types of questions and their combination give us the answer to how the managers from certain management style react to organizational change.

In Table 2 management levels at whom belonging analyzed managers are given. From this table we may note that managers from top level are managers that are mostly included in research (38 managers), while first level managers are only 9. This is due to the fact that the top managers have better picture for entire organization and because they have more experience with resistance to change since they passed all the levels of the hierarchical structure in the organization.

Table 2

Management level of managers obtained in research

Management level	Number of analyzed managers
Top level	38
Middle level	15
First-line level	9

The purpose of this analysis is to get interdependence (correlation) of way in which managers react to organizational change with management style of managers. In Table 3 results from research for all management styles according to Adizes with certain reaction to organizational change of analyzed managers are given.

Table 3

Management styles versus reaction to change

Management styles according to Adizes	Reaction to change	Number of appearance
Producer	Great change support	1
	Mean change support	10
	Inferior behavior to change	14
	Mean change resistance	1
	Great change resistance	0
Administrator	Great change support	3
	Mean change support	1
	Inferior behavior to change	7
	Mean change resistance	0
	Great change resistance	0
Entrepreneur	Great change support	12
	Mean change support	5
	Inferior behavior to change	0
	Mean change resistance	0
	Great change resistance	0
Integrator	Great change support	0
	Mean change support	5
	Inferior behavior to change	2
	Mean change resistance	1
	Great change resistance	0

From Table 3 we can note that neither the manager who is a part of this analysis is not reacting with great resistance to change. This may be due to the fact greater part of the analyzed managers belong to the top level of management, and initiative for change coming from that level. Great supporting of change has entrepreneurs that is normally and that is expected. To find what kind of management styles have usually reaction to change, this reactions is ranged from 5 for great supporting of change to 1 for great resistance to change and from that is derived basic statistics (mean, medians, mode, standard deviation and variance) for all management styles (Tab. 4).

Table 4

Statistics for management styles

	Producer	Administrator	Integrator	Entrepreneur
N	26	11	8	17
Mean	3.42	3.6364	3.5000	4.1176
Median	3.00	3.0000	4.0000	5.0000
Mode	3	3.00	4.00	5.00
Standard deviation	0.643	0.92442	0.75593	1.40900
Variance	0.414	0.855	0.571	1.985

From this table where the basic statistics from analyses of management styles against reaction to organizational change is presented, we may to note that only entrepreneurial style giving great support of organizational change. The interpretations of the results of this analysis are given in the next few lines:

1. The producers are usually inferior to change which approve theoretical hypothesis from Table 1. This means those managers that have this kind of style will accept organizational changes only when those changes give fast results. Producer present potential enemy for proposed organizational changes, unless they do not conform to those changes.

2. The administrators according mean value (3.6364) sometimes may give mean supporting to change, but according values of median and mode these managers are inferior to organizational change. If with organizational change administrators do not lose control they will give mean supporting or will be inferior to that change, but if they lose some part of controlling role, they will resist to that change. Here the theoretical hypothesis from Table 1 is approved.

3. The entrepreneurial style is the style that gives to much support for organizational change, and in the larger number of situations these managers are initiators of organizational change. This is approved from values of median and mode, but the mean value (4.1176) give us the right to conclude that sometimes these managers may have mean supporting to organizational change. Sometimes skepticism may exist about changes from these managers if changes increase rules and procedures in working.

4. The integrator gives mean supporting of organizational change, but often is inferior to it (mean value of 3.5 and median and mode value of 4). From this we may conclude that it is true theoretical hypothesis (Table 1) that managers who have this type of style will support change only if those changes do not destroy social relationships in the organizational system.

This analysis approve theoretical hypothesis from first part of this paper where we conclude that from management style that some managers have, we may predict the reaction to change of the managers. However, to check whether correlation exists between the way of reacting to organizational change of the managers and management styles of that managers, we accomplish correlation analysis with Pearson's coefficient of correlation. The results from this analysis is given in Table 5.

Table 5

Correlation analysis of management styles and reaction to change

		Management style	Reaction to change
Management style	Pearson correlation	1	$-0.350^{(**)}$
	Sig. (2-tailed)		0,005
Reaction to change	Pearson correlation	$-0.350^{(**)}$	1
	Sig. (2-tailed)	0.005	

** Correlation is significant at 0.01 level (2-tailed).

From this table we may to note that between the management styles and reaction to organizational change from managers have poor negative (inverse) correlation with value of Pearson's coefficient of –0.350 with level of significantly of 0,01. This negative correlation means that when the value of one variable increases, then the value of other variable decrease. However, this correlation is too small to give us some powerful relation. This led us to conclusion that all the reactions to the organizational changes against management styles will depend on certain situation, and will differ from situation to situation of organizational changes.

3. CONCLUSION

The analyses of the management styles according to the reaction of managers to organizational changes approve the theoretical hypothesis that the entrepreneurial style is one who give more powerful support to organizational change (or that style initiates the change process), while the other styles of management have mean support that may outgrow to resistance to that process. This is a very significant for the initiator of the change processes because they may predict the behavior of the managers.

Correlation analyses do not give us some strength relationship between the management styles and the reaction of the managers to the organizational change processes. These analyses give weak inverse correlation that lead us to conclude that this type of predicting potentially resistance to organizational change must be estimated from situation to situation of processes of organizational changes.

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Р е з и м е

ОТПОРНОСТА СПРЕМА ОРГАНИЗАЦИСКИТЕ ПРОМЕНИ ВО ОДНОС НА СТИЛОТ НА МЕНАЏМЕНТОТ СПОРЕД АДИЖЕС

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Клучни зборови: организациски промени; отпорност спрема промени; реакција на промени; стилови на менаџментот

Успешноста на процесот на организациските промени е во директна врска со отпорноста спрема промените и е поврзана со успешно справување со таа отпорност. За успешно справување со отпорноста спрема организациските промени менаџерите или иницијаторите на тие промени мораат да ја предвидат реакцијата на луѓето во организацискиот систем. Затоа во овој труд авторите ги претставуваат резултатите од анализа на реакциите на менаџерите на организациските промени во однос на стилот на менаџментот според Адижес. Истражувањето е спроведено со прашалник во 50 претпријатија во Република Македонија.

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THE EFFECT OF OVERMATCHING ON A BEHAVIOR OF TENSILE SPECIMENS WITH A CRACK IN HEAT-AFFECTED-ZONE

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A b s t r a c t: An experimental and numerical investigation of the behavior of the overmatched welded joint with a crack in the heat-affected-zone (HAZ) is performed. Two cases are considered: crack tip located into the fine-grain HAZ (FG HAZ) and into the coarse-grain HAZ (CG HAZ). In both cases, after the initial growth phase of the crack in FG HAZ or CG HAZ, the overmatching acted as a barrier to further propagation of a crack, directed its propagation towardsweaker and more ductile base metal instead of structures with lower toughness. A detailed three-dimensional finite element analysis was performed in order to explain such a behavior.

Key words: overmatched welded joint; fine-grain heat-affected-zone; coarse-grain heat-affected-zone; crack propagation; finite element method

1. INTRODUCTION

The complexity of the welded joint regarding its understanding as material system with significant heterogeneity in the microstructure and mechanical properties is more than evident. Furthermore, its fracture behavior depends on different factors, especially on the properties of the regions surrounding the crack tip, too. The shielding effect of overmatching is well known [1, 2], but for a determined welded joint there is still need for detailed analysis and investigation.

In the scope of assessment of the integrity of cracked welded pressure vessel, described elsewhere [3], the experimental and numerical investigation of the overmatched welded joints with a crack in Fine-Grain Heat-Affected-Zone (FG HAZ), and Coarse-Grain Heat-Affected-Zone (CG HAZ) is performed. The experimental investigation encompassed chemical and metallographical analysis of the base metal and welded joint, hardness and toughness testing, fracture mechanics testing on standard SENB specimens for fracture toughness evaluation and direct measurement of J integral on tensile specimens. Numerical investigation included three-dimensional finite element analysis of the investigated welded joints, too. An extract of this investigation regarding the metallographical and numerical analysis is shown in this paper, together with an analysis of the welded joint behavior. The experimental and the numerical investigation encompassed two cases: case 1 – welded tensile specimen with artificial crack located in the fine-grained (FG) HAZ, and case 2 – welded tensile specimen with artificial crack located in the coarse-grained (CG) HAZ.

2. EXPERIMENTAL INVESTIGATION AND RESULTS

The base material is micro-alloyed steel St. E 420 (according to DIN), 30 mm thick plate. The chemical analysis and measured tensile properties are given in Tables 1 and 2, respectively. The X welded joint (Fig. 1) is made by multipass manual metal arc welding, using base coated electrode AWS E 7018-1 (EN 499: E 46 5 B42 H5), of chemical composition given in Table 1, too.

T a b l e 1

Chemical analysis

Chemical composition of base metal, %												
C	Si	Mn	P	S	Ti	Cr	Al	Cu	Ni	V	Mo	Nb
0.2	0.44	1.35	0.012	0.01	0.12	0.15	0.06	0.05	0.1	0.008	0.015	0.001
Chemical composition of base coated electrode, %												
C	Si	Mn	P	S	Ti	Cr	Al	Cu	Ni	V	Mo	Nb
0.05	0.51	1.56	0.011	0.008	–	0.04	–	0.1	0.02	0.03	0.1	0.01

Table 2

Tensile properties of base metal

Yield strength, $R_{p0.2}$ (MPa)	Tensile strength, R_m (MPa)	Elongation, A_5 (%)
420	604	25

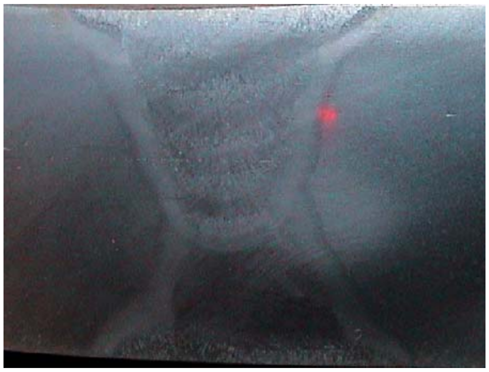


Fig. 1. Weldment cross section

Micro-hardness measurement

The micro-hardness was measured at subsurface layers (2 mm below the surface, lines H1 and H3), and at depth corresponding to the location of the crack tip (lines H2), but also in vertical direction through the weld metal (line V1), Fig. 2.

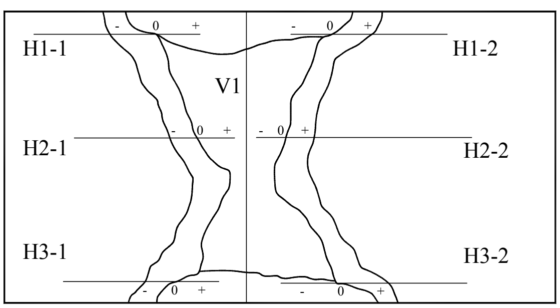


Fig. 2. Micro-hardness measuring positions

The micro-hardness measurement exhibited small differences for different weld metal values, measured along V1 line (Fig. 3 – adopted average values: $HV1_{root}$ = 212, $HV1_{fill}$ = 205, $HV1_{surface\ pass}$ = 215) and the base metal ($HV1_{BM}$ = 185), indicating moderate overmatch of welded joint. The width of HAZ is approximately 3 mm and typical hardness level near the surface for "true" coarse-grained structure is about HV1 = 350, whereas at depth near the crack tip, which was relevant for the σ–ε curves determination, the hardness varied between HV1 = 281 for the "quasi" CG HAZ and HV1 = 221 for the FG HAZ (Fig. 4). These results are used for σ–ε curve determination, and also for the FEM analysis of the welded joint.

The stress-strain curves

The σ–ε curve of the base metal was obtained by the standard tensile test, and the σ–ε curves for weld metal (WM) and CG HAZ and FG HAZ of the heterogeneous welded joint, are assessed through a combination of micro-hardness measurement and Ramberg-Osgood law (Fig. 5). Input parameters were the micro-hardness, measured by HV1 (enabling evaluation of the yield strength, $R_{p0.2}$, of the tensile strength, R_m, and indirectly of the elongation, A), and cooling time, $\Delta t_{8/5}$, from 800 °C and 500 °C (from which strain hardening exponent n^* could be evaluated), as shown in [4]. The following equations were applied in order to evaluate the basic tensile properties for the HAZ [4]:

$$R_{p0.2} = 3.1 \cdot \mathrm{HV} \cdot (0.1)^{n^*} - 80 \ \text{(Mpa)}$$

$$R_m = 3.5 \cdot \mathrm{HV} \cdot (1 - n^*)(12.5 \frac{n^*}{1 - n^*}) - 92$$

$$A = 5.75 \times 10^4 (R_m)^{-1.25} \ (\%)$$

$$n^* = 0.065 \cdot (\Delta t_{8/5})^{0.17}.$$

For the estimated $\Delta t_{8/5}$ = 15 s, obtained values are:

n^* = 0.103,

$R_{p0.2}$ = 461 MPa (for FG HAZ), and

$R_{p0.2}$ = 605 MPa (for "quasi" CG HAZ) [4].

For the weld metal [4] following equations were used:

$$R_{p0.2} = 3.15 \cdot \mathrm{HV} - 168 \ \text{(MPa)}$$

$$R_m = 3.3 \cdot \mathrm{HV} - 8 \ \text{(MPa)}$$

for 100 < HV < 250

$$R_m = 3.15 \cdot \mathrm{HV} + 93 \ \text{(MPa)}$$

for 250 < HV < 400.

Thus, the average yield strength for weld metal (in fill passes) is $R_{p0.2}$ = 478 MPa, obtained by the average HV1 measurement values.

Finally, by applying the Ramberg-Osgood relation:

$$\frac{\varepsilon}{\varepsilon_0} = \frac{\sigma}{\sigma_0} + \alpha \left(\frac{\sigma}{\sigma_0} \right)^{\frac{1}{n}},$$

where σ_0 is the reference yield stress, ε_0 equivalent strain and α non-dimensional constant, the true σ–ε curves of the welded joint regions are determined and shown in Fig. 5.

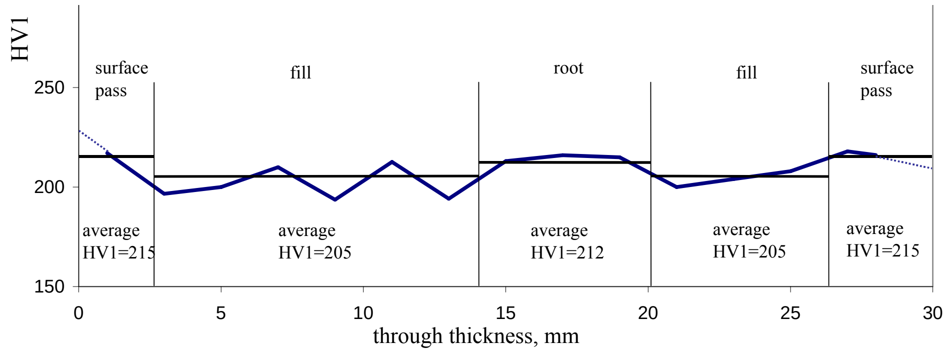
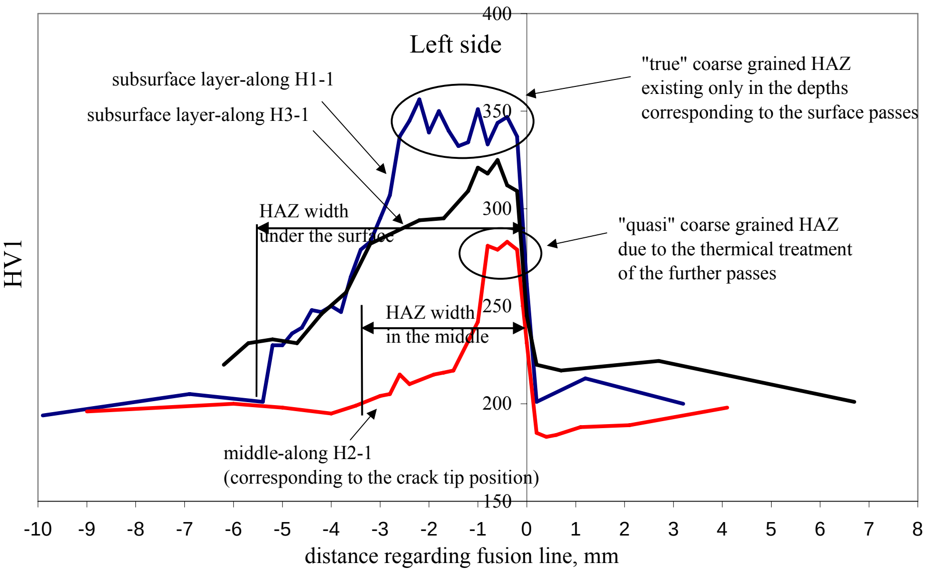


Fig. 3. Distribution of the measured micro-hardness HV1 along V1 (see Fig. 2)



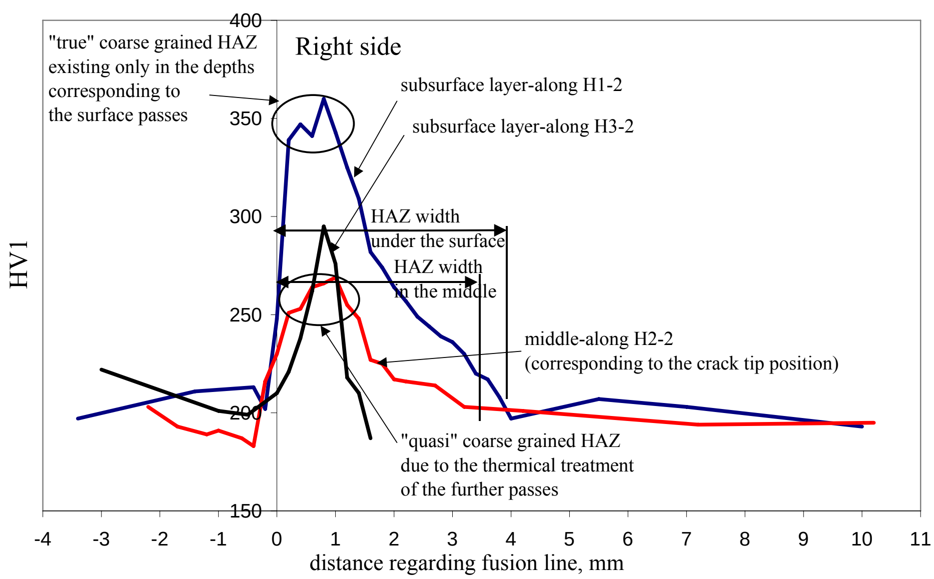


Fig. 4. Distribution of the measured micro-hardness around the fusion line (“the zero points" in Fig. 2)

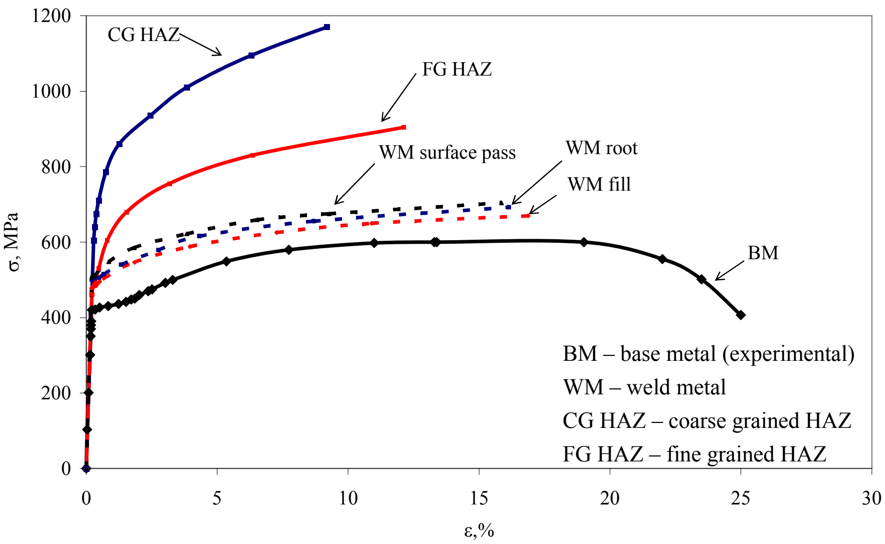


Fig. 5. Determined stress-strain curves of the welded joint's regions

Direct measurement of J integral

The main reason to apply direct measurement of J integral in addition to the standard fracture toughness testing was to get, in the same time, the strain distribution along the weldment. Toward this end, tensile specimens (width W = 24 mm, thickness B = 20 mm, length L = 300 mm, Fig. 6) were machined out of the spare welded plate (400 × 400 mm). Cracks were produced by electroerosion, since by standard fatigue procedure it was practically impossible to locate the crack tip in very narrow HAZ subregions (fine grain – FG HAZ, and coarse grain – CG HAZ). Special procedure was applied, including very small amperage (cca. 1 A) in order to get crack tip as sharp as possible (below 0.05 mm, as shown schematically in Fig. 7). The nominal length of crack was 12 mm.

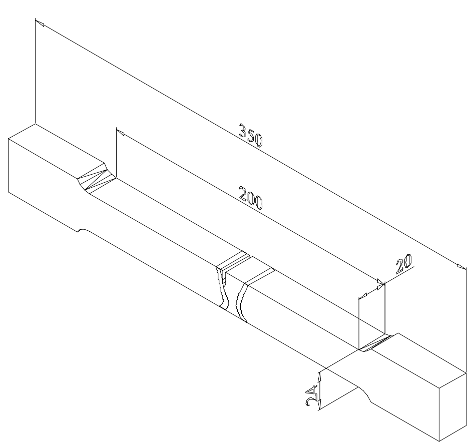


Fig. 6. Tensile test specimen precracked in heat-affected-zone

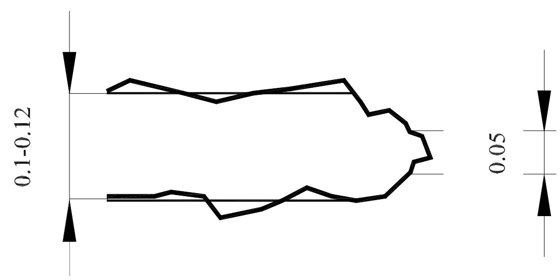
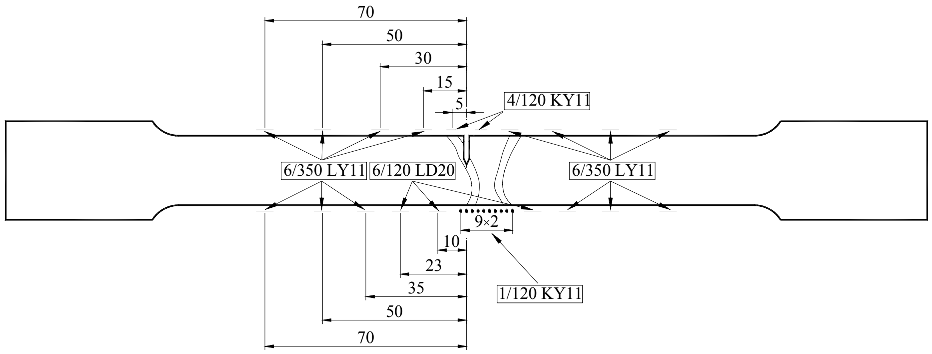


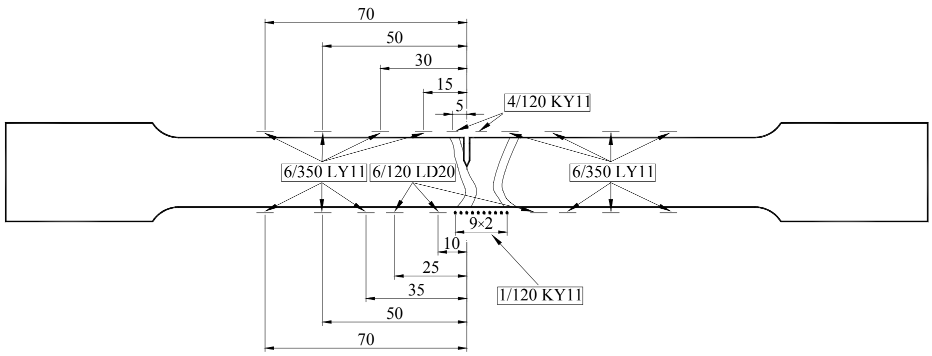
Fig. 7. The shape of the crack tip obtained by electroerosion

Strain gauges were positioned following properly selected J integral path and CMOD clip gauge was applied, as shown in Fig. 8. The procedure for evaluation of J integral is described elsewhere [6]. Strains were recorded at successive loading levels during tension, enabling detailed analysis of strain distribution along the weldment, both from cracked and uncracked side. At the same time, as the final result, this method provides relationship force, F, vs. crack mouth opening displacement, CMOD, and resistance curves in the form of J integral vs. CMOD.

Two specimens with different crack tip position (case 1 – specimen E5-2 with crack tip located in FG HAZ, and case 2 – specimen E4-1 with crack tip located in CG HAZ) were tested. Another pair of specimens was tested in the same way, except for the instrumentation, which consisted only of CMOD clip gauges, in order to verify results. The results, in a form of strain distribution are shown in Fig. 9 for case 1 and in Fig. 10 for case 2.



crack tip located in FGHAZ (specimen E5-2)



crack tip located in CGHAZ (specimen E4-1)

Fig. 8. Instrumentation for direct J integral measurement

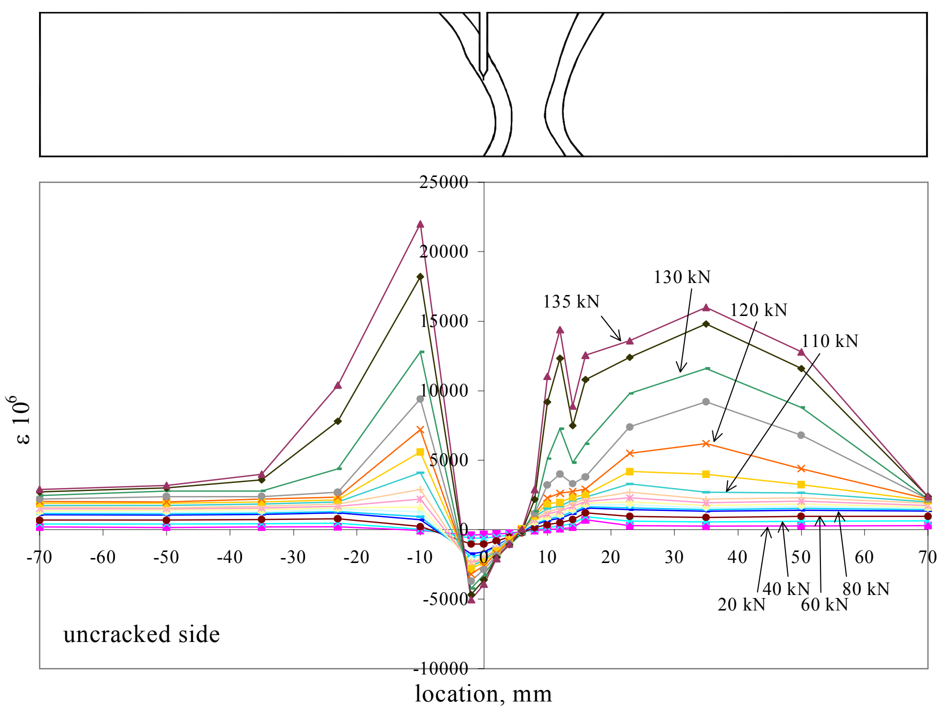


Fig. 9. Strain distribution – specimen E5-2 (case 1 – crack tip located in FG HAZ)

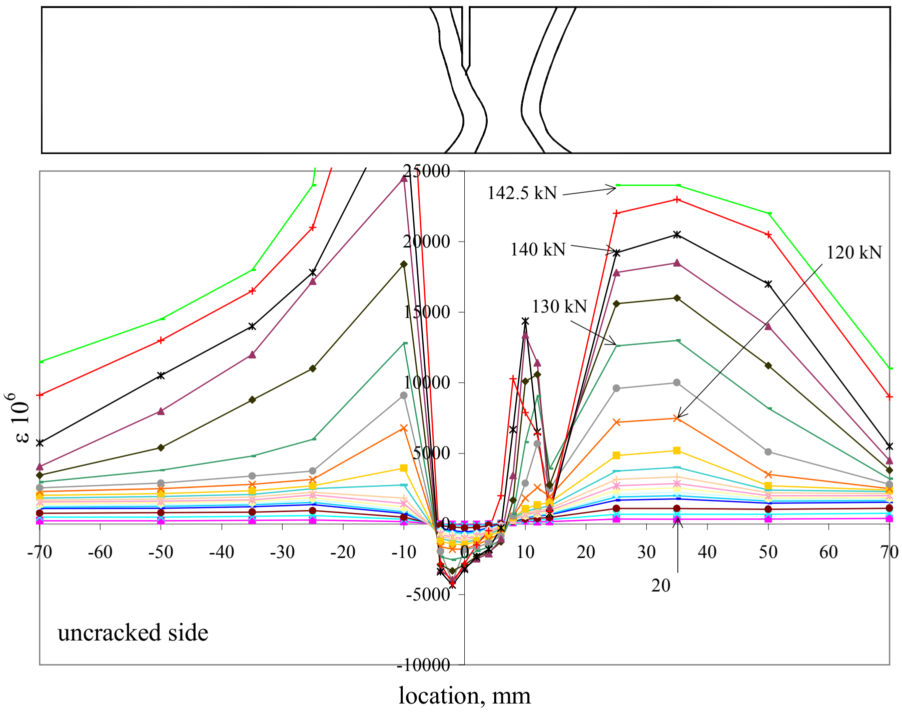


Fig. 10. Strain distribution – specimen E4-1 (case 2 – crack tip located in CG HAZ)

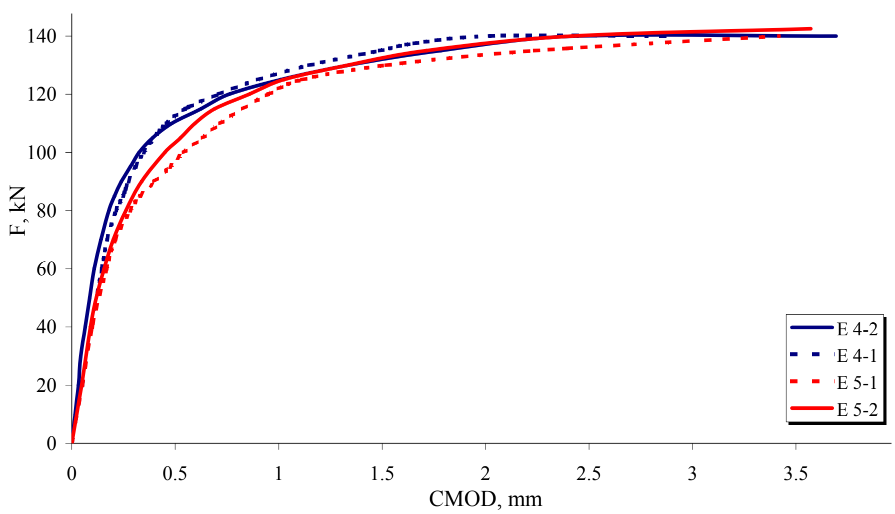


Fig. 11. F vs. CMOD curves for specimens with crack tip located in CG HAZ (E4-1) and FG HAZ (E5-2)

The results in form of force *F* vs. CMOD are shown in Fig. 11 for two specimens in each considered case. In Figure 12 the resistance curves J integral vs. J-Δa are presented for one specimen in each case, due to the fact that only one pair of specimens was instrumented with strain gauges.

J-Δa revealed stabile crack growth. The J-Δa resistance curve for the homogeneus materials consists of two ranges of different slope, one with steeper slope indicating the blunting of the crack tip, and the second one of lower angle indicating the stabile crack growth. Instead of it, these specimens revealed J-Δa resistance curves consisting additional, third, section of steeper slope, indicating the entrance of the crack tip into the material of higher fracture resistance, i.e. the base metal. The resistance curves for both of the cases revealed similar trend except the propagation of higher rate of crack propagation in case of crack in CG HAZ, due to the fact that the coarse grained HAZ has lower toughness and the crack initiation started sooner.

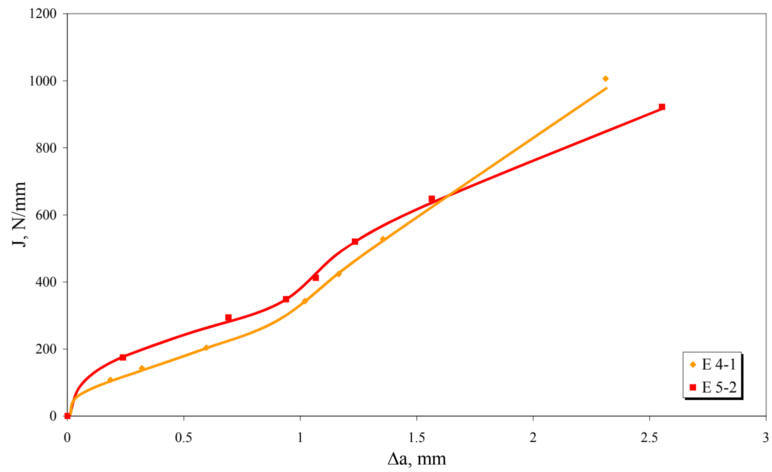


Fig. 12. J vs. Δa curves for specimens with crack tip located in CG HAZ (E4-1) and FG HAZ (E5-2)

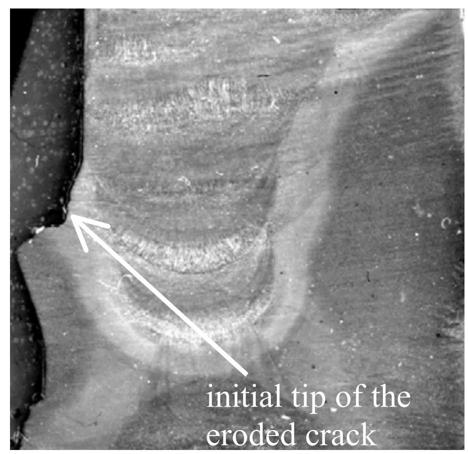
3. METALLOGRAPHIC INVESTIGATION

The metallographic investigation was performed on both types of specimens. The results are shown in Fig. 13 and in more details in Fig. 14 and Fig. 15. One should notice that for both crack tip locations (CG HAZ and FG HAZ) crack propagated into the base metal.

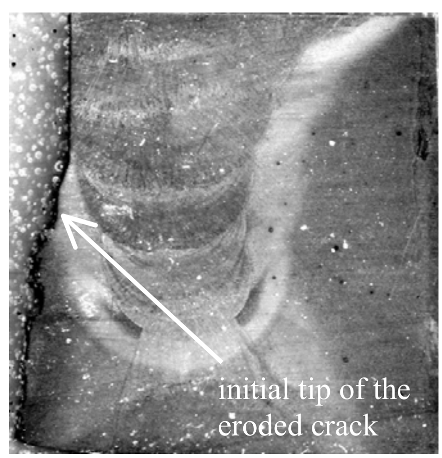
a)

b)

Fig. 13. Propagation of a crack from HAZ toward the parent metal: a) specimen E5-2, crack tip located in FG HAZ, b) specimen E4-1, crack tip located in CG HAZ

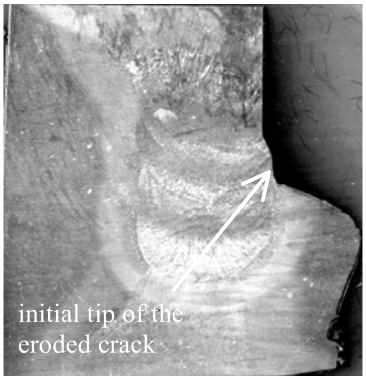


a)

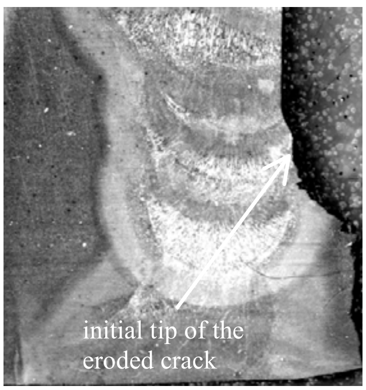


b)

Fig. 14. Crack propagation – specimen E5-2, crack tip located in FG HAZ: a) free surface, b) weld centre



a)



b)

Fig. 15. Crack propagation – specimen E4-1, crack tip located in CG HAZ: a) free surface, b) weld centre

Both of the cases, the case 1 and the case 2, exhibited crack propagation towards the base metal.

In the case 1 the crack was located in the finegrained region of the HAZ. On the free surfaces the crack has started its propagation straight towards the base metal (Fig. 14-a). After that, it keeps the same direction of propagation as a result of free contraction, until the start of unstable crack growth. In the middle, small deviation of the crack path towards the base metal could be noticed, and than the propagation has been directed in broken line directly through-thickness (Fig. 14-b). Such behavior is resulting from the high tri-axial state in the middle of the specimen due to constraining of the surrounding material leading to decreasing of the fracture resistance of the material of crack propagation [8]. Such stress state could result with brittle fracture if the crack is surrounded by material of higher strength. Anyhow, in this case the base material has absorbed certain amount of the deformation and has prevented the brittle fracture.

Likewise, in the case 2 the crack has propagated towards the base metal, with exception of the longer way of crack propagation till the entrance in the base metal structure, since the initial tip was located in the CG HAZ. In this case, the crack was located in CG HAZ (Fig. 15) and, in certain points along the welded joint, directly by the fusion line. At the free surfaces (Fig. 15-a), the crack propagation moves directly through the CG HAZ, and then shifts towards the base metal. Furthermore, Figure 15-b shows multiple directioning of the crack propagation, lightly shifting towards the base metal, then sudden sharp shunt towards the weld metal, and again sudden sharp shunt towards the base metal, each time in angle of app. 180^o, as explained in details in [4, 7]. Such sharp shuntings of the propagation path are resulting from the nature of the CG HAZ and the fusion line, as well the ingredients concentrated around the grain boundaries. The crack propagation in the middle has exhibited certain brittle behaviour due to the plane strain state explained above.

4. NUMERICAL ANALYSIS

Tensile behavior of specimens was analyzed numerically by the finite element method (FEM). The two-dimensional (2D) and three-dimensional (3D) models were generated and run by using ANSYS with incremental loading in large number of steps. Special care was taken to model HAZ and crack region, as shown in Fig. 16, and with magnified crack tip region, given in Fig. 17. This region requested large number of 3D 20-node elements (total amount is 4560), as presented in Fig.18.

The tip with radius of 0.05 mm, i.e. the front of the crack in the threedimensional analysis is modelled with big refinement of the mesh, thus requiring large number of 2D 8-node, i.e. 3D 20-node elements. This provides singularity of the stress and the strain around the small zone in front of the crack tip. Each element at the tip has 3 independent nodes which is a postulate for elastic-plastic behavior of the material in the zone of increased stresses and straines, according to the ESIS recommendations.

Material behavior in the ANSYS input file was modeled by multi-linear stress-strain curves. Numerical results for remote force F vs. CMOD are shown in Fig. 19 for the specimen with crack tip in FG HAZ and in Fig. 20 for the specimen with crack tip in CG HAZ, together with results of 2D FEM calculation and the experimental results. Three models for each case are taken into the numerical analysis: 2D – plane stress and plane strain, and 3D. Numerical results for J vs. CMOD curve are shown in Figs. 21–22. The equivalent strain distribution, obtained by the 3D FEM is shown in Fig. 23 for both specimens and for three different levels of loading (remote stress $\sigma = 250$ and 292 MPa, corresponding to the remote force $F = 120$ and 140 kN, respectively). Significant influence of overmatching is obvious, acting as the barrier for strain development, pushing the strain to the more ductile base metal of lower strength and protecting the welded joint from fast fracture, in both of the cases.

Case 1 – crack in FG HAZ   Welded joint sight   Case 2 – crack in CG HAZ

Fig. 16. The simplified FEM model of the welded joint

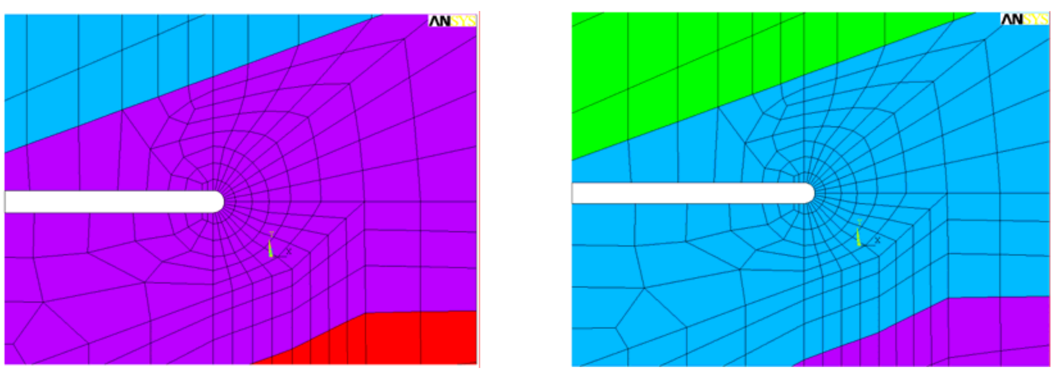


Fig. 17. The simplified FEM model of the welded joint – enlarged crack tip region

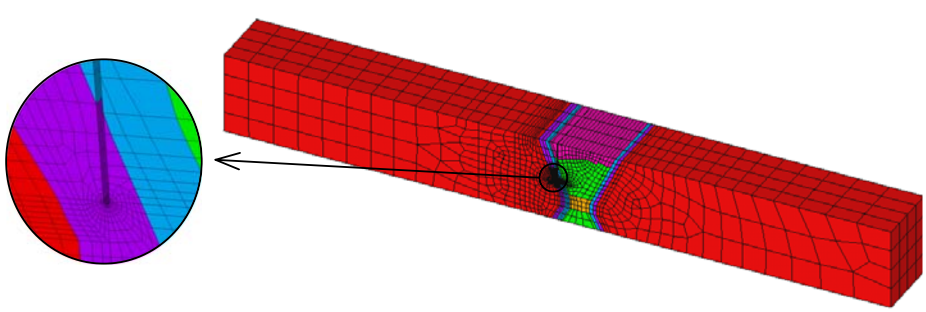


Fig. 18. The 3D model with enlarged crack tip region

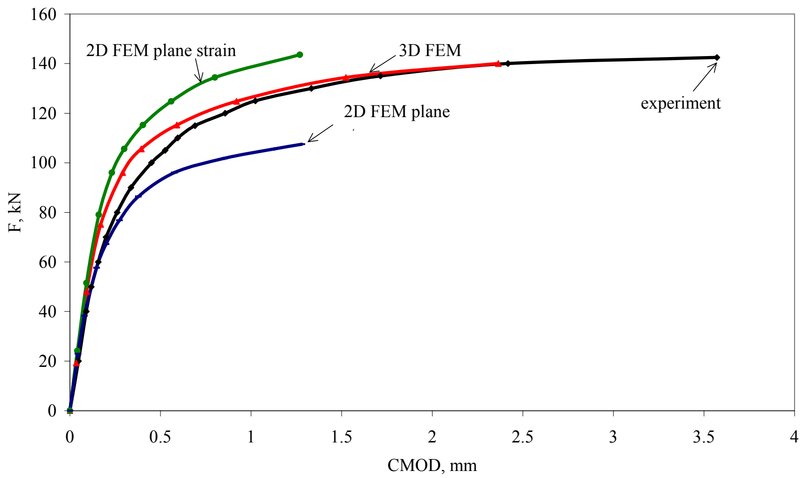


Fig. 19. F-CMOD curves for the specimen with crack tip in FG HAZ

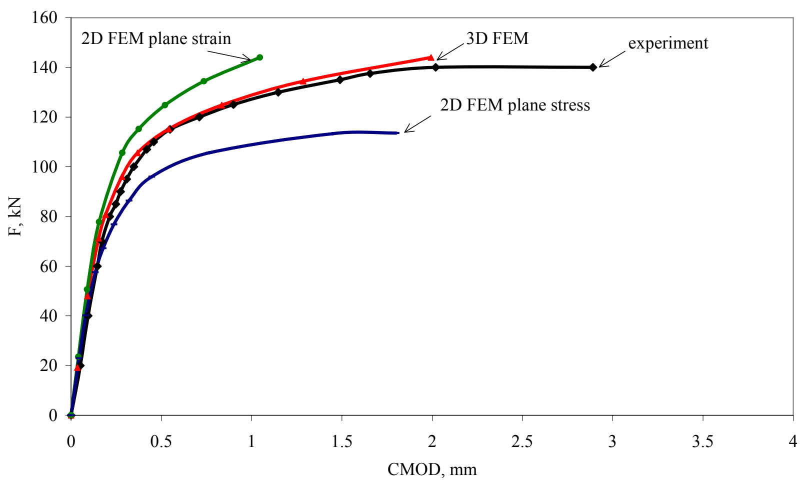


Fig. 20. F-CMOD curves for the specimen with crack tip in CG HAZ

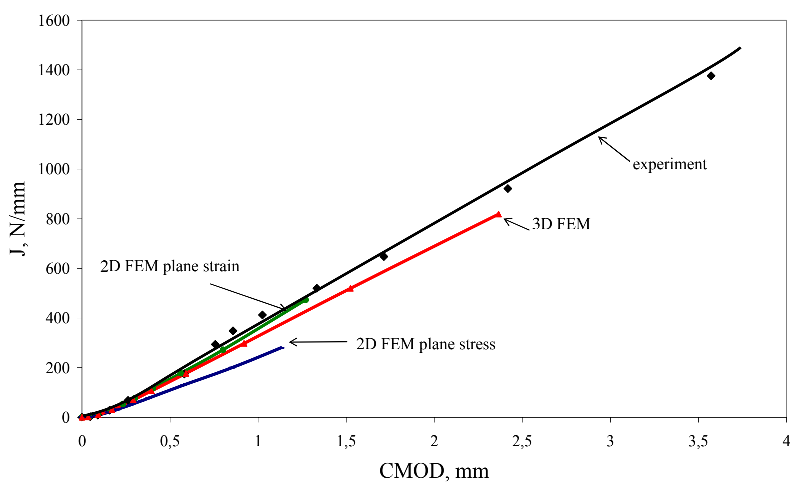


Fig. 21. J-CMOD curves for the specimens with crack tip in FG HAZ

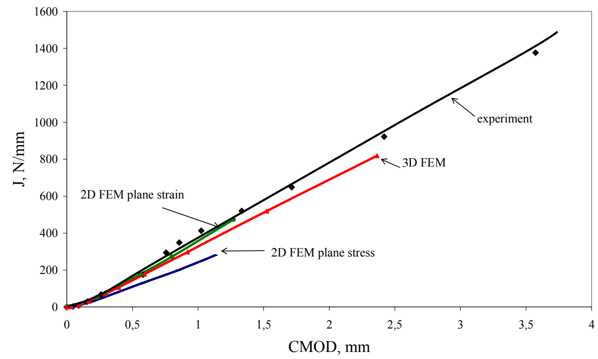
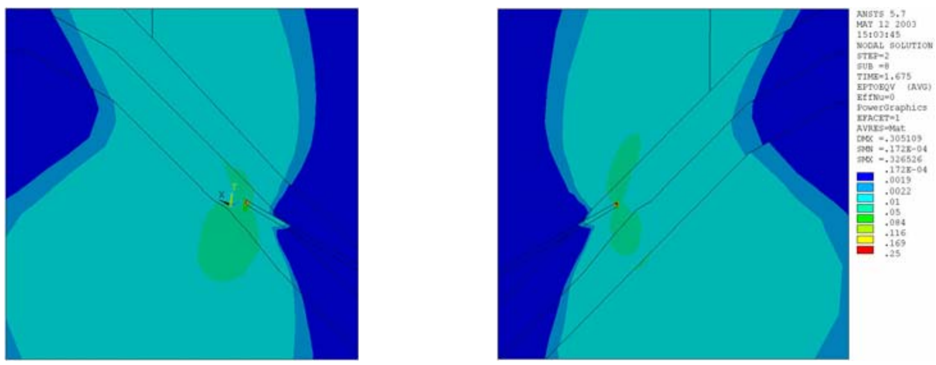
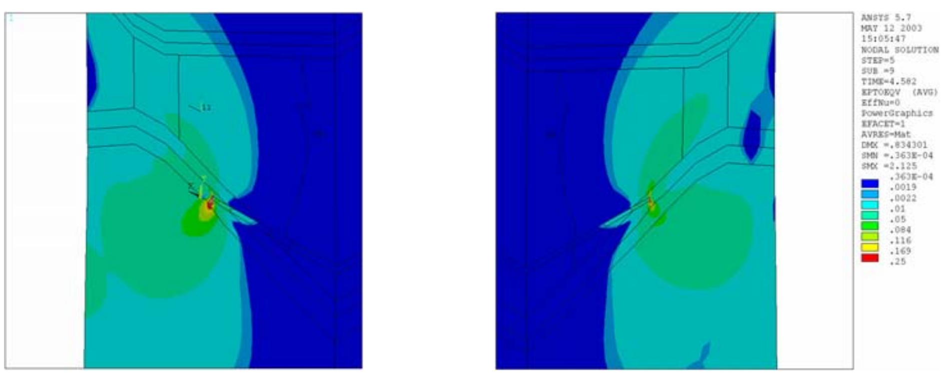


Fig. 22. J-CMOD curves for the specimens with crack tip in CG HAZ



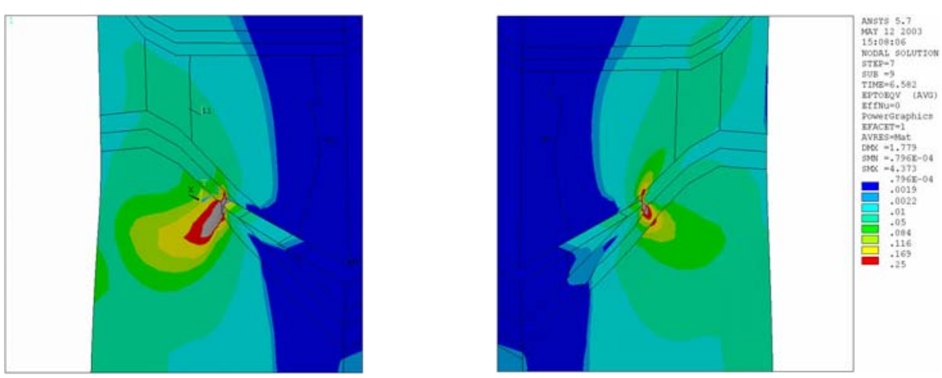
a) crack tip in FG HAZ b) crack tip in CG HAZ

at remote stress σ = 167 MPa corresponding to load *F* = 80 kN



a) crack tip in FG HAZ b) crack tip in CG HAZ

at remote stress σ = 250 MPa corresponding to load *F* = 120 kN



a) crack tip in FG HAZ b) crack tip in CG HAZ

at remote stress σ = 292 MPa corresponding to load *F* = 140 kN

Fig. 23. Equivalent strain distribution

5. DISCUSSION

Experimental investigation

Although the direct measurement of J integral is primarily used to get the J-R curves, in this investigation the strain distribution was of primary interest instead. As one can see from Figs. 9 and 10, the strain distribution is not significantly different in two considered cases, except at very high levels. The specimen E4-1 exhibited higher level of the strain, in general, since the crack tip was located in region with higher strength, thus shifting the greater amount of the deformation to the surrounding material, primarily into the base metal. The strain reaches its maximum at approximately 10–12 mm left from the crack tip for both cases. This is in accordance with the ligament length (approximately 12 mm) and propagation of the deformation along the planes of maximum shear stresses (45^o to the crack tip plane). On the right side of the crack tip, the deformation tends to behave in the same way, except for the lower level due to the higher strength of weld metal. Sharp decrease of strain at approximately 10–15 mm, being more expressed for the case 2 than for the case 1, is due to the effect of the HAZ. The absence of such an effect on the right side is due to a dominant influence of crack.

For the specimen E5-2, the level of the strain, in general, is lower comparing with the first specimen, since the crack tip was located in more ductile structure (FG HAZ), thus not shifting the complete deformation into the surrounding base metal, but resuming certain part of it. The crack propagation at the free surface started in the HAZ (Fig. 14-a), but immediately after that it turned to the base metal, due to the weld metal, acting as a barrier, and also supported by the free contraction, i.e. plane stress condition. Nevertheless, in the specimen center, after an initial "intention" to turn toward the base metal, crack propagated straight downwards (Fig. 14-b) due to the constraint deformation (plane strain condition) resulting with high triaxial stress condition, which decreased the fracture resistance of the welded joint in general.

The specimen E4-1 exhibited similar behavior, except that the crack had to pass greater length to reach the base metal. At the free surfaces the crack initially propagated straight through the CG HAZ, but afterwards it turned toward the base metal (Fig. 15-a). In the specimen center, the crack propagated through the base metal and than turned to the uncracked side (Fig. 15-b).

It should be emphasized here that the triaxial stress condition in the specimen center can cause brittle fracture if regions with higher strength, acting like barriers, surround the crack. Nevertheless, in both cases analyzed here, the stronger weld metal surrounded the crack tip only from one side. On the other side was the weaker metal, either the base metal or the FG HAZ, absorbing the deformation and increasing the fracture resistance of the welded joint in general [5].

Numerical analysis

The numerical analysis consisted of 3D and 2D modelling. In general, the behavior of a specimen is 3D, but 2D analysis is also beneficial since the plane stress and plane strain conditions are the two thresholds limiting the structure behavior. The first one, plane stress state, is dominant on the free surface i.e. it is typical for structures of small width, and the second one, the plane strain state is dominant in the middle of the specimen, i.e. it is typical for structures of greater width. The three-dimensional condition is between these two extreme conditions, since the width of the tested specimens is between the definition of small and great width.

Both pairs of curves, J vs. CMOD and F vs. CMOD exhibited good agreement between the experimental and the 3D numerical results (Figs. 19–22). Comparing with the 2D analysis, the experimental curves were between the two extreme conditions, as expected. The partial mis-agreement between the numerical and experimental results can be explained by following reasoning:

– geometrical simplification of the welded joint, including its shape, dimensions and its composition which is approximated as being built of root, fill, and surface passes, although it contains much more different microstructural regions with different mechanical properties;

– material behavior simplification, having in mind the analytical determination – assessment of the σ-ε curves for the welded joint regions using the Ramberg-Osgood law;

– the finite element analysis used here did not include crack extension.

The strain distribution for the case 1 (crack tip located in fine-grained HAZ, E5-2), as presented in Fig. 23-a, shows strong influence of the surrounding regions with different properties and size, pushing the deformation in the direction of the base metal. This effect is a consequence of several associated influences. Namely, the crack is located in the fine-grained HAZ region, which is placed between coarse-grained region with higher yield strength ($R_{p0.2}$ = 605 MPa) and the base metal with lower yield strength ($R_{p0.2}$ = 420 MPa). The deformation is blocked in the direction of the weld metal and is directed towards the base metal for lower but also for higher loads. This could be explained by the following reasons: first, the high strength CG HAZ acts like barrier, preventing the propagation of deformation in the direction of less ductile structures, secondary, due to this constraint, and high stress concentration around the crack tip, deformation was directed towards the base metal, and third, the shifting of deformation towards the base metal is supported by the weld metal as well, although having only slightly higher strength than the base metal (Fig. 5), but much larger size compared to HAZ.

In the case 2 (E4-1) (Fig. 23-b) the crack is located in the material with high yield strength, surrounded by the more ductile fine-grained HAZ region and the weld metal, having similar yield strengths, $R_{p0.2}$ = 461 and 478 MPa, respectively. At lower loads, having better ductility, the surrounding fine-grained HAZ, as well the weld metal absorbed the deformation, thus relaxing the crack tip region placed in coarse-grained HAZ. At higher loads, not only because of slightly lower yield strength, but also because of the shifted rotation center, deformation shifted toward the base metal through the fine-grained HAZ (Fig. 23-b). Due to this absorption of the deformation, the plastified zone around the crack tip is smaller comparing to the one in case 1.

6. CONCLUSION

The behaviour of welded joint with crack placed in fine-grained HAZ is under significant influence of the adjacent coarse-grained HAZ supported by the weld metal as well. By blocking the free deformation, this effect causes high three axial stress state and, although on the other side the more ductile base metal accepts the deformation, this leads to deterioration of the fracture behavior of the fine-grained HAZ, compared with its behaviour in a state without the effect of mismatch [4]. In the other case, there is coarse-grained HAZ, having smaller fracture toughness, but, since it is surrounded by regions with smaller strength and bigger plasticity, this effect leads to absorption of the deformation by these more ductile regions, causing more favourable stress conditions for the coarse-grained HAZ, compared with its behavior in a state without the effect of mismatch [4].

Anyhow, both of the cases exhibited high fracture resistance due to the barrier effect of the mismatching. After the initial propagation of the crack in FG HAZ or CG HAZ, relatively small overmaching has changed crack propagating direction toward weaker and more ductile base metal, thus decreasing the speed of the crack propagation.

In general, more favourable stress-strain state around the crack tip is being achieved in case when the surrounding adjacent material has lower strength and higher plasticity (overmatching), so this material absorbs the deformation under loading, thus reducing the stresses in the crack tip zone, thus improving the fracture resistance. This mostly relates to the weld metal, as most dangerous zone for crack occurence.

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Р е з и м е

ВЛИЈАНИЕ НА ХЕТЕРОГЕНОСТА НА ЗАВАРЕНА ВРСКА СО ЗГОЛЕМЕНИ МЕХАНИЧКИ ОСОБИНИ НА ОДНЕСУВАЊЕТО ПРИ ЗАТЕГНУВАЊЕ НА ЕПРУВЕТИ СО ПУКНАТИНА ВО ЗВТ

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Клучни зборови: заварена врска со зголемени механички особини; финозрнеста ЗВТ; грубозрнеста ЗВТ; пропагација на пукнатина; метод на конечни елементи

Спроведено е експериментално и нумеричко истражување на заварена врска со зголемени механички особини (overmatching) со пукнатина во ЗВТ. Разгледани се два случаја: пукнатина лоцирана во финозрнеста ЗВТ и во грубозрнеста ЗВТ. И во двата случаја на зголемени механички особини на металот, заварот во однос на основниот метал се однесувал како бариера за понатамошна пропагација на пукнатината во структури со намалена жилавост, а насочувајќи ја кон послабиот и подуктилен основен метал. Деталната нумеричка 3-Д анализа со конечни елементи беше спроведена заради објаснување на ваквото однесување.

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Professional paper

OPTIMUM MANIPULATOR MOBILITY, SIMULATED BY USING MATLAB/SIMULINK AND VIRTUAL REALITY TOOLBOX

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A b s t r a c t: In this paper the original developed method for measurement of needed precision regarding realization of assembly between cylindrical elements using of industrial robot with help of MATLAB/Simulink program language is presented. Original MATLAB/Simulink schemes for measurement of kinematics and dynamic parameters of modular pneumatic robot is presented. Algorithm for optimization of specified link movement is developed. Increasing of air pressure in pneumatic cylinders (in our experiment we do such a phase by increasing of force into kinematical pairs), which means increasing of acceleration. Kinematical schemes of PMIR by using MATLAB module Virtual Reality toolbox with a simulation possibility of robot movement, and also changing kinematical and dynamical parameters are developed.

Key words: manipulator; optimization; error; simulation

1. INTRODUCTION

The main task of this paper is to define the suggestions about modernization of modular manipulators. Regarding this, in this paper we used inverse kinematics algorithms for theoretical research and calculation implemented by MATLAB/Simulink. Special accent is put on optimization of end-effector trajectory as a mathematical interpretation of productivity, which can be rapidly increased up to 20 times.

Problem definition

Define optimal values of the coordinates *x*, *y* and *z* shown in Fig. 1. After the optimization routine calculate position and velocity errors of the end-effector. The following are given:

– trajectory: $z^2 = x^2 + y^2$,

– initial postures: $x = 0.3\,\text{m}$, $y = 0.4\,\text{m}$ and $z = 0.5\,\text{m}$,

– forces: $F_1 = 1\,\text{kN}$ and $F_2 = 10\,\text{kN}$,

– masses: $m_1 = 16800\,\text{g}$, $m_2 = 2500\,\text{g}$ and $m_3 = 3500\,\text{g}$.

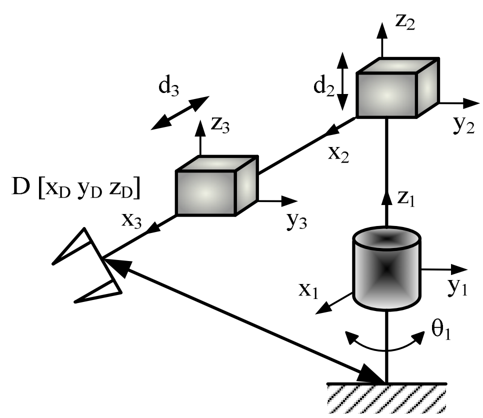

Fig. 1. Manipulator kinematics scheme

2. MANIPULATOR DIRECT KINEMATICS

The homogeneous transformation matrices for manipulator are:

$$A_1^0(\theta_1) = \begin{bmatrix} \cos(\theta_1) & -\sin(\theta_1) & 0 & 0 \\ \sin(\theta_1) & \cos(\theta_1) & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix} \tag{1}$$

$$A_2^1(d_2)=\begin{bmatrix}1&0&0&0\\0&1&0&0\\0&0&1&d_2\\0&0&0&1\end{bmatrix} \tag{2}$$

$$A_3^2(d_3)=\begin{bmatrix}1&0&0&d_3\\0&1&0&0\\0&0&1&0\\0&0&0&1\end{bmatrix} \tag{3}$$

direct kinematics function is:

$$T_3^0=\begin{bmatrix}\cos(\theta_1)&-\sin(\theta_1)&0&d_3\cos(\theta_1)\\\sin(\theta_1)&\cos(\theta_1)&0&d_3\sin(\theta_1)\\0&0&1&d_2\\0&0&0&1\end{bmatrix} \tag{4}$$

according to equations 1, 2, 3 and 4:

$$\begin{bmatrix}x_D^{(0)}\\y_D^{(0)}\\z_D^{(0)}\end{bmatrix}=\begin{bmatrix}d_3\cos(\theta_1)\\d_3\sin(\theta_1)\\d_2\end{bmatrix} \tag{5}$$

3. MANIPULATOR DIFFERENTIAL KINEMATICS

Jacobian of the manipulator can be calculated with equation:

$$J(\theta_1,d_2,d_3)=\begin{bmatrix}\frac{\partial x}{\partial\theta_1}&\frac{\partial x}{\partial d_2}&\frac{\partial x}{\partial d_3}\\\frac{\partial y}{\partial\theta_1}&\frac{\partial y}{\partial d_2}&\frac{\partial y}{\partial d_3}\\\frac{\partial z}{\partial\theta_1}&\frac{\partial z}{\partial d_2}&\frac{\partial z}{\partial d_3}\end{bmatrix} \tag{6}$$

The Jacobian in differential kinematics equation of a manipulator defines a linear mapping $v=J(q)\dot{q}$ between the vector $\dot{q}$ of joint velocities and the vector $v=\begin{bmatrix}\dot{p}&\dot{\omega}\end{bmatrix}^T$ of end-effector velocity. By considering the previous mentioned, joint velocities can be obtained via simple inversion of the Jacobian matrix:

$$\dot{q}=J^{-1}(q)v\,. \tag{7}$$

If the initial manipulator posture $q^{(0)}$ is known, joint positions can be computed by integrating velocities over time:

$$q(t)=\int_0^t\dot{q}(t)dt+q^{(0)}. \tag{8}$$

The integration can be performed in discrete time by using Euler integration method; given an integration interval Δt; if the joint positions and velocities at time t_k are known, the joint positions at time $t_{k+1}=t_k+\Delta t$ can be computed:

$$q(t_k+1)=q(t_k)+\dot{q}(t_k)\Delta t \tag{9}$$

or:

$$q(t_k+1)=q(t_k)+J^{-1}(q(t_k))v(t_k)\Delta t\,. \tag{10}$$

It follows that the computed joint velocities do not coincide with those satisfying (7) in continuous time. This inconvenience can calculate error between desired and actual end-effector position (11) and velocity (12):

$$e_{pos}=x_d-x \tag{11}$$

$$e_{vel}=\dot{x}_d-\dot{x} \tag{12}$$

4. SIMULINK MODEL AND SIMULATION RESULTS

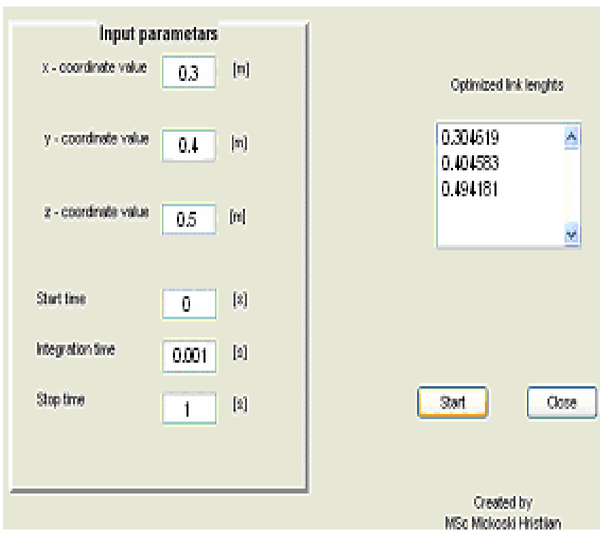

Fig. 2. Graphic User Interface (GUI)

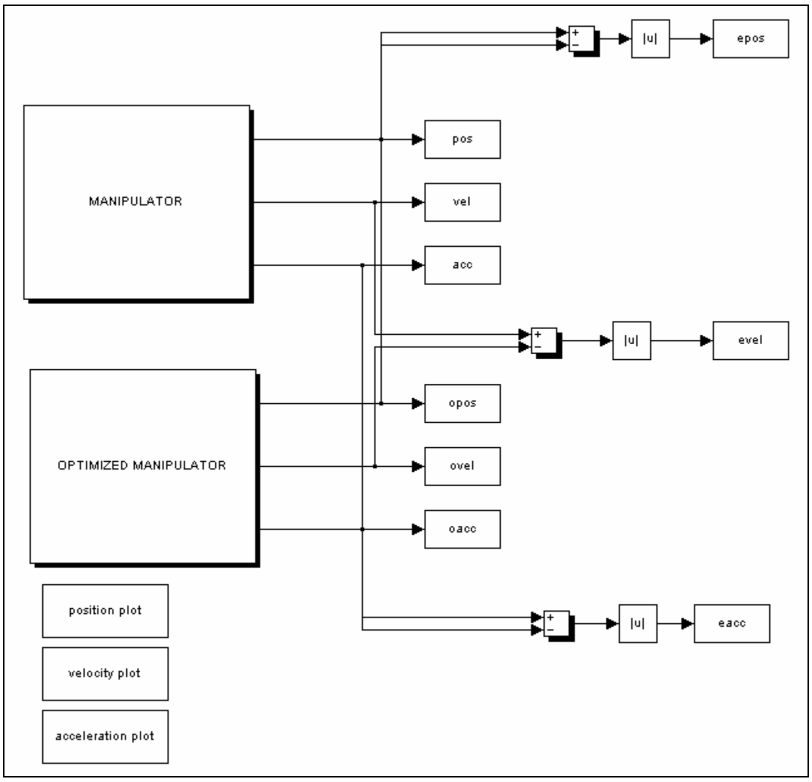

Fig. 3. Simulink model

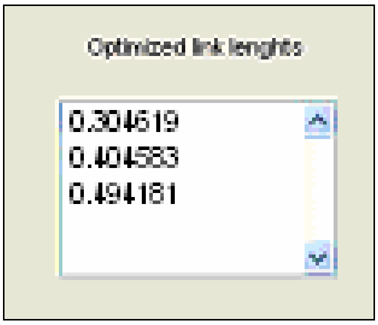

Fig. 4. Optimization results

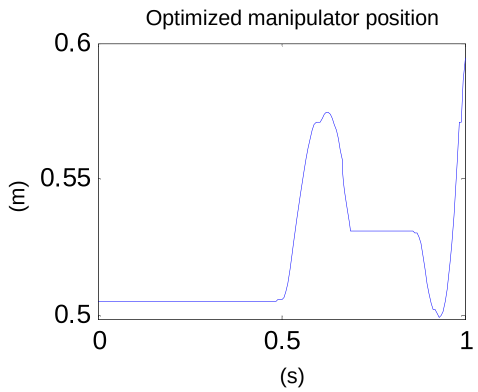

Fig. 5-b

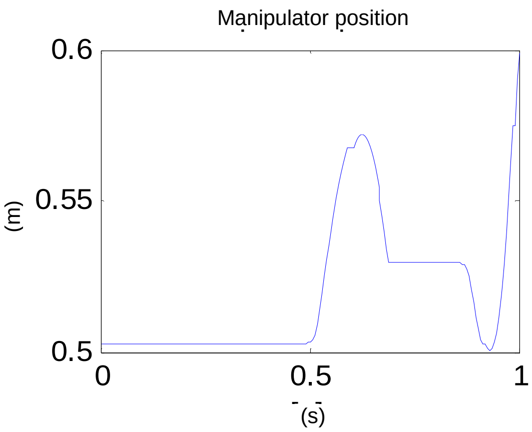

Fig. 5-a

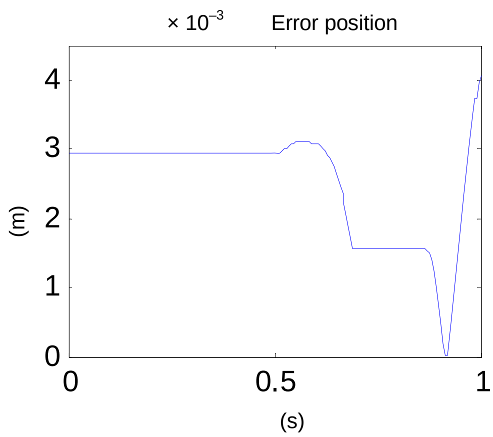

Fig. 5-c

Fig. 5. Position error results

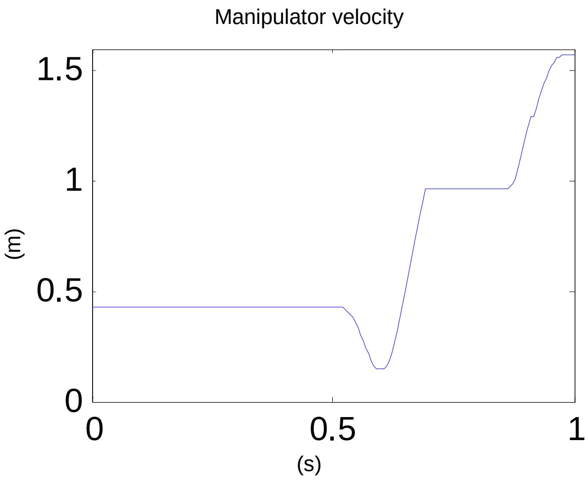

Fig. 6-a

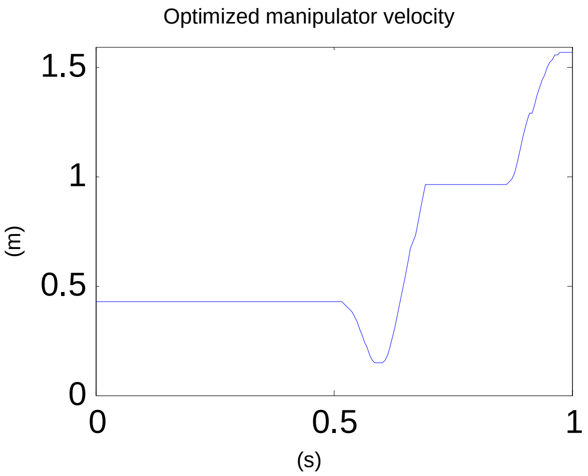

Fig. 6-b

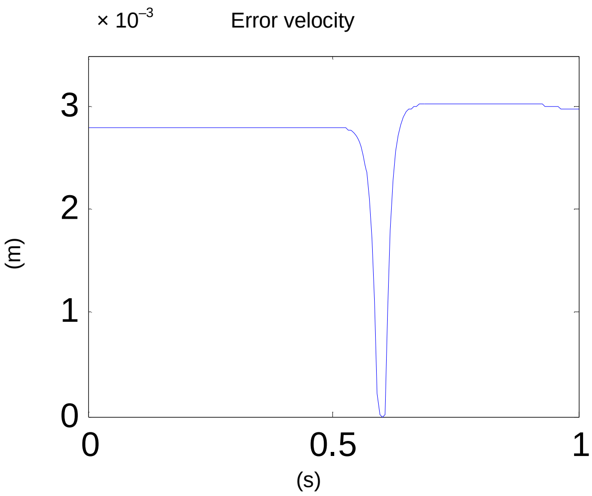

Fig. 6-c

Fig. 6. Velocity error results

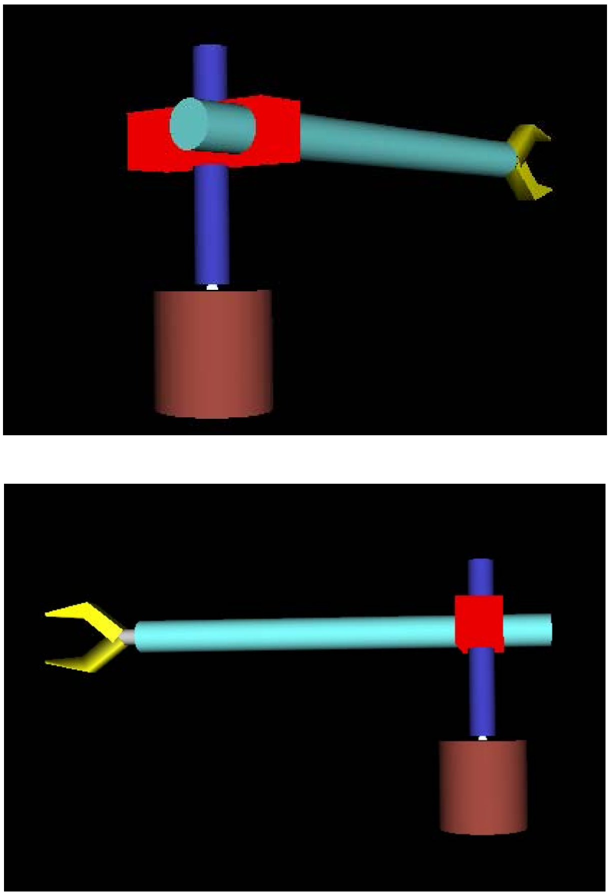

Fig. 7. VR animation

5. CONCLUSION

In this paper we developed original MATLAB/Simulink program that calculates end-effector position and velocity error. Increasing pressure and forces into kinematics pairs result such end-effector errors. Visualization of results is shown in this paper. Performing optimization routines and calculation of optimal structure can cause increasing of production. Theoretical results were confirmed by practical results, regarding difference of approximately up to 10%.

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Р е з и м е

ОПТИМАЛНА ПОДВИЖНОСТ НА МАНИПУЛАТОР, СИМУЛИРАНА СО MATLAB/SIMULINK И VIRTUAL REALITY TOOLBOX

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Клучни зборови: манипулатор; оптимизација; грешка; симулација

Главна цел на овој труд е дефинирање на сугестии за модернизација на модуларни манипулатори. За теоретските истражувања и пресметки е користен инверзниот кинематички алгоритам, имплементиран во програмскиот пакет MATLAB/Simulink. Посебен акцент е даден на оптимизационата процедура како математичка интерпретација на продуктивноста. Оптимирана е траекторијата која ја реализира фаќачот. Врз база на теоретските и практичните резултати е докажано зголемување на продуктивноста над 20 пати