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Politehnica University Timisoara, Faculty of Engineering, Hunedoara, Romania
University of Szeged, Faculty of Engineering, Szeged, Hungary**



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Dear Ladies and Gentlemen, respectable Colleagues and Friends of KOD,

It is a real pleasure and great honor for us to greet You on behalf of the Organizing Committee of the Ninth International Symposium about machine and industrial design in mechanical engineering – KOD 2016. This year, symposium KOD, for the third time, takes place in Hotel Marina in Balatonfüred, Hungary on 9th and 12th June 2016, and I would like to thank You for participating in it.

As we all know, the basic goal of this event is to assemble experienced researchers and practitioners from universities, scientific institutes and different enterprises and organizations from this region. Also, it should initiate more intensive cooperation and exchanging of practical professional experiences in the field of shaping, forming and design in mechanical and graphical engineering, industrial design and shaping, product development and management. Having always present need for making more effective, simpler, smaller, easier, noiseless, cheaper and more beautiful and esthetic products that can easy be recycled and are not harmful for environment, the cooperation between specialists of these fields should certainly be intensive.

Fifty articles are published in the Proceedings. It is the same number of papers as in last symposium. This means our colleagues and friends of KOD are always active. Of course, we believe that time for organizing symposium has not passed and we want to prove it. However, published papers are very interesting, contribute to the understanding of design building relationships across multidisciplinary design domains including engineering and product development, innovation, manufacturing, management, complexity, human behaviour and system design, so that means these topics have potentials and have to be further researched.

Thank You for coming in Balatonfüred to take part in symposium KOD 2016 and for Your interesting articles. I wish You success in Your further researching and great fortune and happiness in personal life.

Prof. D.Sc. Siniša Kuzmanović, Eng.

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Invited Paper in Plenary Session:

DESIGN OF AUTOMOTIVE GEARBOX WITH TOP PROPERTIES BASED AT HYBRID AND CVT APPROACH

Milosav OGNJANOVIĆ
Dragan DŽODAN

Abstract: *The actual trend in engineering design is increase the level of technical systems properties and behaviour. The main approach is hybridisation together with widening the range of function, structure and behaviour. The article contains presentation of this trend and continues with presentation of the main actions in this sense. Automotive gearbox developed by the authors is result of presented approach. This gearbox is hybrid (mechanical, electronic and software) system which provides continual variation of transmission ratio and alternating power transmission, consumption of electricity and recharge the batteries*

Key words: *Engineering design, Hybrid technical systems, Automotive gearbox, CVT*

1. INTRODUCTION

Design methodology in 70-90 years was oriented to Function-based design with the main task to develop the new conceptual design by transformation of function – F into structure – S (Fig. 1), using linear design model. This approach didn't provide expected results. The later period was characterised with dominant development of Property-based design approach with the aim to improve technical system (TS) behaviour – B, using developed V-design model. Nowadays is actual approach to increase the quality of action and behaviour of TS with existing principle of action. This Effect-based design approach is oriented to expand the domain of TS functions – F, domain of structures – S and domain of behaviour – B, using integration of mechanical structure with electronic, software and mental action [1]. The lack of the new principles of action have to be complemented by increase of the level of existing TS. This situation was created due to the fact that in the last hundred years was intensive development of technology with the absence of significant scientific discoveries and inventions. High level of technology provides possibility for increase of the

level of existing TS and absence of significant discoveries and the new sources of power is the reason for absence of the new principles of TS action. [1]

2. DESIGN ACTION TOWARDS INCREASE OF PROPERTIES AND TS EFFECTS

Perfection of TS implies the new secondary functions, the new structure and increase of TS behavior. It can be achieved by application of hybrid structure which except of mechanical, contains electronic, software and similar components, in a word by development of intelligent TS. This kind of reverse engineering with the objective to increase existing TS behavior, performs with the pressure (force) to increase certain indicator of behavior such as efficiency, convenient control, comfort etc. The limitations which define design parameters, got design constraints. These are forced directions and objectives which have to be fulfilled in order to increase the level of TS. The design limitations are the passive limits which have to be satisfied, but design constraints are active and forced directions of design action and for activities which lead to objectives which are not precisely defined in advance, uncertain and will be fulfilled in the measure which allow level of knowledge, available technology and market conditions.

The basic design model established by J. Gero [2] is FBS-model (Fig.1) which shows that in design process it is necessary to bring in close relation function F, behavior B and structure S in order to provide product P i.e. technical system (TS). The suitable example for illustration of widening of F, B and S ranges is automotive gearbox. The main constraints in this sense is presented in Fig.1[3]. The actual constraints of functional F and structural S are derived from references [3-10] and constraints in behavior B, from references [11-17]. Results to date of this development are specified at the bottom of the Fig.1. These are manual automotive transmissions, automated manual transmissions, continuously variable transmissions, dual clutch transmissions, hybrid drive and electric drive. From generation to generation of these automotive power transmissions, by presented design constraints action, the level of design quality is increased, especially in the area of design behavior in exploitation. Majority of constraints are in the strong contradictions, especially structural and behavior constraints. The typical example is contradiction between lightweight design i.e. mass reduction and reliability and durability increase and vibration and noise reduction.

Functional design constraints are arise in order to increase vehicle efficiency by reduction of mechanical power dissipation and to transform surplus of this power in electricity and use it when it is necessary. It was beginning of hybridization of vehicle drive systems by combination of engine drive and electric motor drive. Hybridization continued with hybridization of the power transmission control using sensors, actuators and software components. The actual situation, according to [4] is that exist a few levels of automotive hybrid drive trains. Micro hybrid drive trains contain start-stop use and transformation of electric power. Mild hybrid implies electric boosting and recuperation of the drive and full hybrid electric slow drive in stop and go conditions.

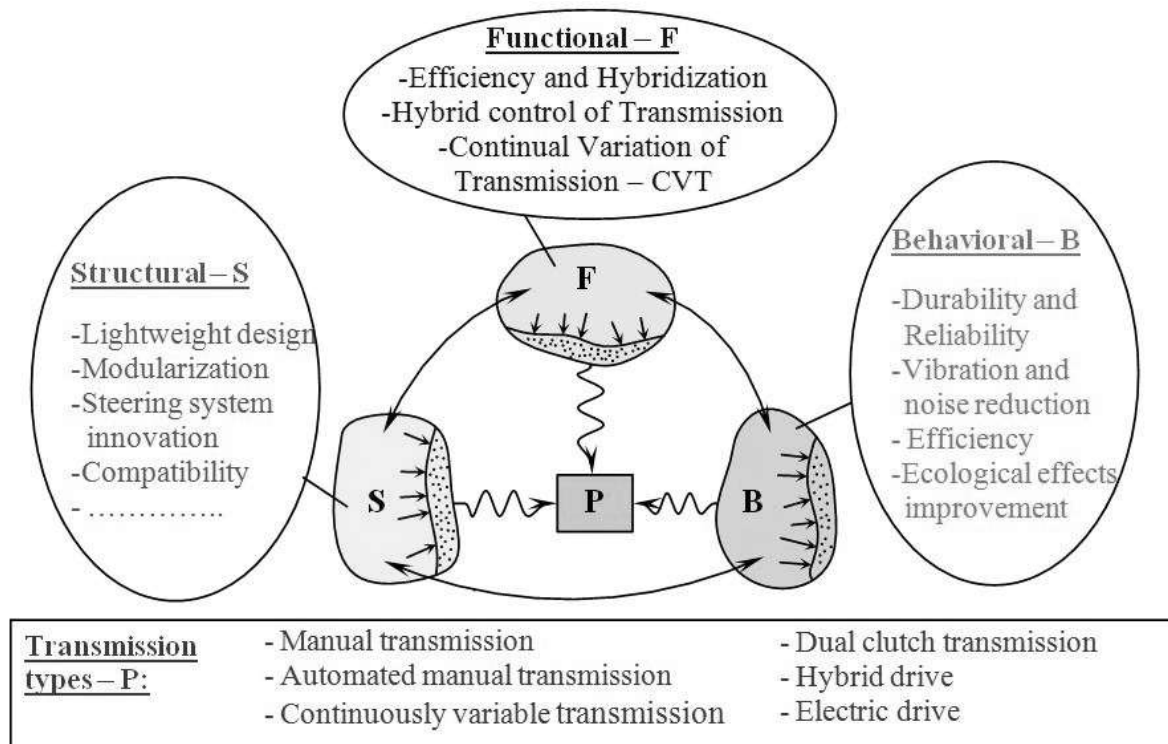


Fig.1. FBS design model with corresponding areas of design constraints

Higher levels of hybridization imply possibility of electric drive till to electric drive in all operating conditions.

Structural design constraints such as lightweight design and high level of compatibility, modularization, etc. got very important especially in the automotive area. In relation with the other design constraints such as functional, very often they can be in strong contradiction which has to be solved with the new design solutions based on the new principle (Fig. 2) or improvement of existing conceptual solution. This contradiction can be salved by the new design solutions which provided compromise of contradiction. In *Lightweight design* the weight reduction is the complex activity which involves phases of design development, starting from conceptual design, material selection, dimensions and shape choice, application of various technological methods such as thermal and plastic treatment etc. Application of these approaches can significantly reduce the mass but also can significantly increase product cost. This contradiction is very often a limit for mass reduction. *Modularization* is structural design constraint directed to the cost reduction [10]. Modularization contains transformation of complex technical system into the set of modular components which can be installed in various design structures. Automotive gearbox or engine in the form of design module can be installed in various cars or one type of car can be with in installed various engines or gearboxes. *Steering system innovation* in automotive area is constraint which can provide significant increase of product quality in functionality, comfort, traffic safety, etc. One sub-area of vehicle steering is power transmission management [11-12]. Reverse energy flow, is not possible to be manual. Control of these processes is complex and has to be carried out by electronic and software systems. This necessity involves in the system function carriers based on various physical principles. Linking of this function carriers presents serious design

problem and constraint. Discussion and procedure of linking is presented in [13]. Realization of this constraint is in contradiction with lightweight or compatibility constraint. Solving of this contradiction produces various specific design solutions and innovations (solution in Fig. 2 and 3).

Behavioral design constraints are oriented to increase quality of action, market quality, environmental and ecological quality etc. Behavior of the system is result of developed structure and required field of functions. In reverse direction, desired behavior imposed necessary functions, design parameters and structure, interrelations inside the system, etc. *Durability and reliability* are the main indicators for market competition in automotive area, including relation between this indicators and cost. Furthermore increase of durability and reliability are in strong contradiction with other constraints such as cost reduction, lightweight design, compatibility, etc. Compromises by them and the new design solutions in details and in structure integration, achieve satisfaction of contradictory design constraints. *Vibration and noise* level reduction is also the design constraint which provides increase of TS quality and better position in the market competition. This action constrains development of technical solution which satisfies legal limitations and also in reverse provides effect at the legal limitations to be stronger in the future.

Design actions oriented to satisfy mentioned and similar design constraints present very complex procedure which not defined in advance and is in strong relation with the task and objective. The first step is observation and systematic analysis of possibility for design action and of possible design contradictions which can arise including possible contradiction compromises. The second step is definition of the set of design activities which can satisfy objectives. Design actions can be in stage of conceptual design, embodiment and detail design.

3. CONCEPTUAL DESIGN

Taking in consideration all design constrains and design solutions of automotive gearboxes, presented in Fig. 1, the new conceptual solution is developed and presented in Fig. 2. The concept and principle of action involves the two planetary gear sets connected by electric motor-generator. The first stage of gears 1-3 is usual planetary gear transmission with stationary outside gear ring 3. The second stage is also planetary gear set 4-6, but with moveable outside gear ring 6. Motion control of gear ring 6 performs electric motor-generator. It is specific electric design which provides controlled relative motion of ring 6 in relation to motion of the first stage satellite carrier. Transmission ratio of planetary gear sets are

$$i_I = 1 + \frac{z_3}{z_1} = 1 + \frac{90}{60} = 2,5; \quad i_{II} = 1 + \frac{z_6}{z_4} = 1 + \frac{90}{60} = 2,5$$

For conditions when outside ring 6 standing, total transmission ratio is $i_{I,II} = 6,25$. By rotation of gear ring 6 in one direction, transmission ratio decreases to the total value of 1 and less. By rotation in another direction, transmission ratio increases to the necessary level, for example to the 10. For the engine speed of 6000-8000 rpm the output speed can vary in the range 600-8000 rpm. Motor-generator is specific design which provides lightweight and compatibility and by specific electronic control, alternative flow of electric power from the batteries and reverse. In order to provide transmission ratio 6.25 it is necessary to provide standing position of the ring 6. Electromagnetic forces have to be reaction to the active forces at the ring 6. This electromagnetic forces provides system of controlled electromagnets settled in the rings 7 and 8. Ring 7 rotates together with the satellite carrier of the first stage with the speed of 2400 rpm for the engine speed of 6000 rpm ($6000/2,5=2400$). In conditions of relative motion with 2400 rpm, standing position of the ring 6 and ring 8, is possible with electricity production and storage into the batteries. By

reduction of transmission ratio from 6.25 until to 2.5 reduces production of electricity to the zero level. For farther reduction of transmission ratio to the 1 or less, it is necessary to consume electricity. Increase of transmission ratio from 6.25 also implies consumption of electricity. For the purpose of reverse consumption and production of electricity, motor-generator can be composed of ring 7 as permanent magnet and ring 8 of electronically controlled electromagnet. However motor-generator has one function in the form of relative speed control between rings 7 and 8. This the reason that both rings (7 and 8) have to be developed as systems of electromagnets with control of electricity flow.

Electronic and software control of the electric system which in Fig. 2 marked as motor-generator is the top important component for presented principle of action. Functions of motor-generator together with control system can be classified into the following three groups.

- 1) Identification and control of relative speed of rotation between rings 7 and 8 (Fig.2). In relation of this speed vary transmission ratio of gearbox. Control of relative speed between disc 7 and 8 is control of gearbox transmission ratio.
- 2) Transfer and control of electricity at electromagnets in the rings 7 and 8 in order to keep necessary relative speed of rotation. Reverse transfer of electricity can provide equilibrium and close to full batteries. In the absence of electricity in batteries, ring 7 can consummate engine mechanical power. In the case of a surplus of electricity, via the ring 7 can be reduced power of engine. Also surplus of vehicle kinetic energy has to be transform into electricity (breaking by transmission).
- 3) Vehicle speed control in interactive action with the driver.

F-B-S relation at conceptual level is established. This conceptual solution is significantly expanded the area of function F, structure S and behavior B of automotive gearboxes. Also the new structure provides very good lightweight and compatibility in relation with existing automotive gearboxes.

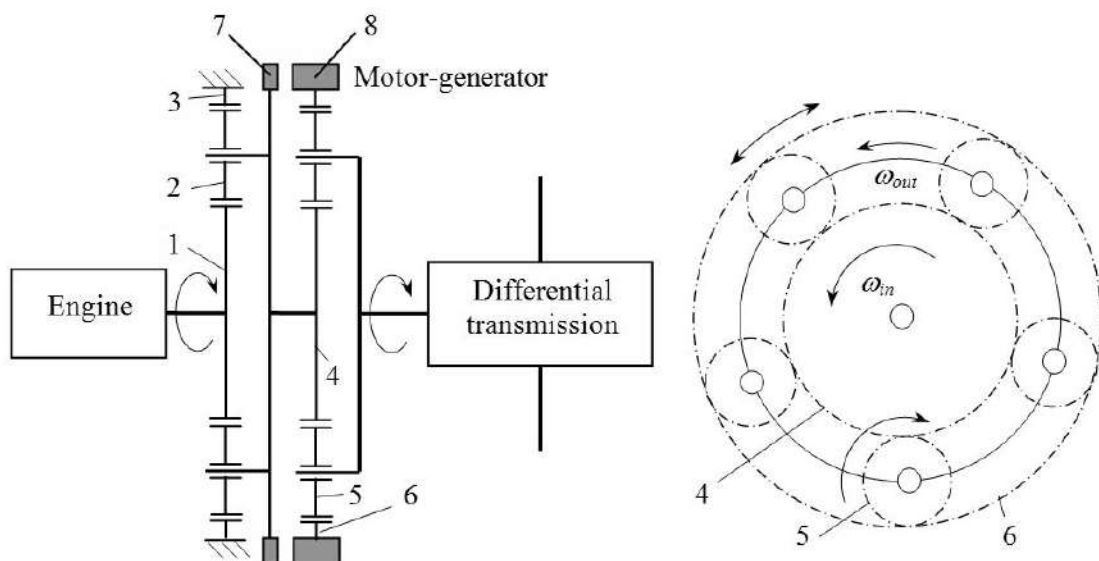


Fig.2. The new conceptual solution of automotive gearbox with continual variation of transmission ratio and with alternating flow of electric power - CV

4. EMBODIMENT DESIGN

4.1 Mechanical structure

According to constraints of structure S presented in Fig.1, **Lightweight design** implies maximal reduction of the weight in relation to caring (load) capacity. In conceptual design this constraint is applied by planetary principle of transmission and by using five (5) gear satellites which divide total torque into five gear teeth connections. At embodiment stage, the stress in the teeth contacts has to provide minimization of overall dimensions, but do not increase the risk of failure (reliability of Behavioral-B constraints). In Fig. 3 is presented preliminary layout of automotive gearbox developed based on concept presented in Fig.2. Maximal impute torque is 500 Nm with input speed of rotation 6000 rpm i.e. power of 315 kW. The gears have to be produced of steel for carbonization of the teeth. Considering necessary reliability (Reliability for design) related to the load spectrum of the vehicle exploitation and service life, allowed (design) contact stress in the teeth contact is limited to be at the maximal level of 1200 N/mm². For this conditions the teeth module is $m_n=2.5$ mm. This module is applied for the both planetary gear sets. Difference in the loads at the first and second transmission stage is solved by difference in the gear width. At the first planetary set gear width is 25 mm and at the second stage 35 mm. Overall outside diameter of gearbox together with the housing is about 300 mm. Equal outside dimensions of outside gear rings (marked in Fig.2 with 3 and 6) provides possibility for motor-generator installation which is of specific design.

Constraints of **compatibility**, as a rule is in relation with lightweight, except in specific conditions. In design presented in Fig.3 compatibility is provided with forms of gears and the forms of satellite carriers which fit together in compact assembly. The problem for farther reduction of dimensions of gears are dimensions of bearings. Less dimensions of gears produce higher bearing loads and higher bearing dimensions. Additionally, less gearbox overall dimensions reduce quantity of heat which outside gearbox surface can emit into surroundings. This contradiction is the limit for increase compatibility and the mass reduction. Developed structure in Fig.3 is result of the high application of structural design constraints together with compromise of mentioned contradiction.

Constraint of **modularization** in the automotive field is with the strong action. It is necessary to create as much as possible modules in order to create various design structures with the same modules. Gearbox is one of the main modules. It is necessary to create possibility for installation the same design of gearbox in the various vehicles and the same vehicle with various engines and gearboxes. Developed gearbox is with high caring capacity (500 Nm/315 kW) and with compact dimensions (small volume), 300 mm outside diameter. It can corresponding to the powerful cars or to the small buses. The design parameters of gearbox are **robust** and corresponds to relative high variation of service conditions. Various service life for various vehicles can spread out the range of robustness of this gearbox.

Constraints of **gearbox steering** is also significantly improved i.e. transformed in interactive software process.

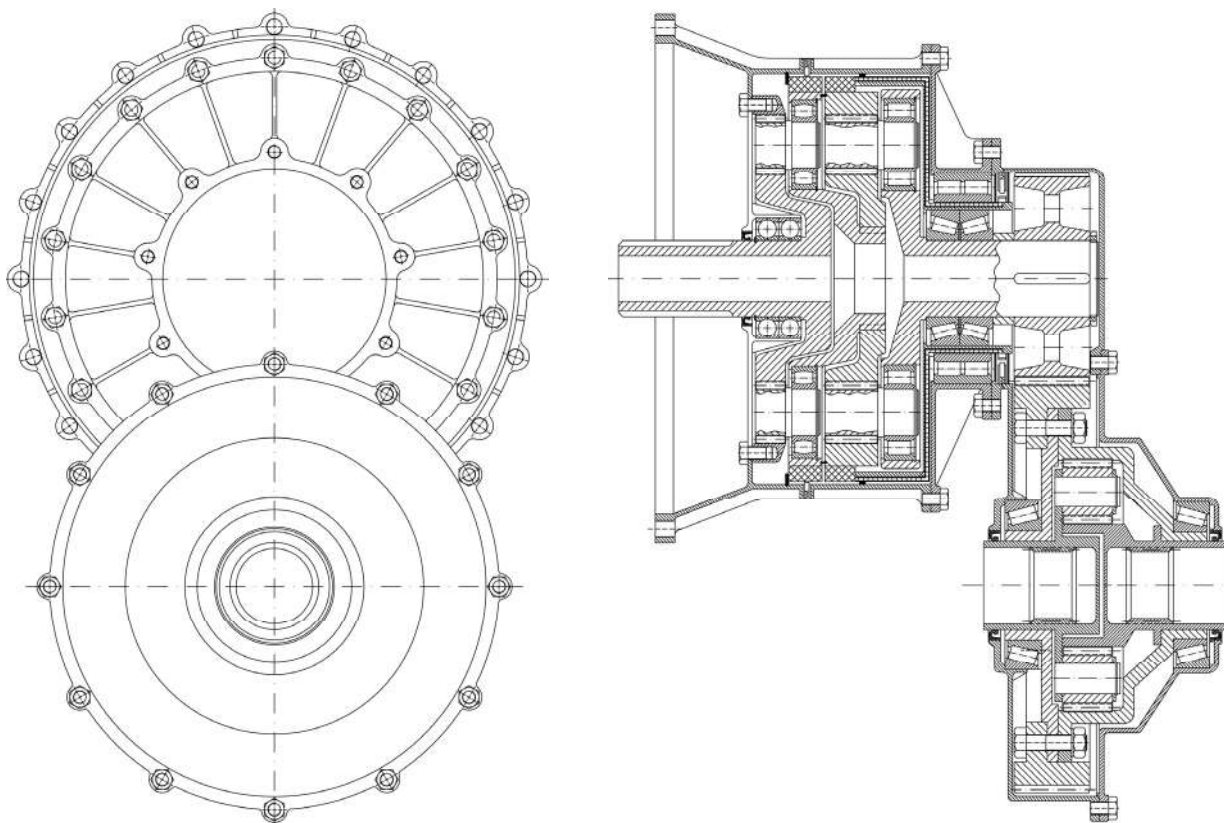


Fig.3. Layout of automotive gearbox with continual variation of transmission ratio, according to conceptual design presented in Fig.

4.2 Electrical, electronic and software structure

Motor-generator (Fig.2) consists of the two rings, 7 and 8 (Fig.2) with the system of electromagnets distributed in each ring similarly to presentation in Fig. 4. The system of electromagnets in the ring 7 supplies direct current. Rotation of this magnetic field corresponds to the frequency of rotation of the ring 7 in relation of engine speed of rotation. The ring 8 has vary relative speed of rotation in relation to the ring 7. Variation of this speed provides difference in frequency of magnetic fields i.e. variation of the frequency of rotating field in ring 8. For this purpose, the system of electromagnets in this ring supply the three phase alternating electric current. Ba variation of current frequency vary the difference in frequencies of this two magnetic fields. In relation with frequency difference varies relative speed of rotation between rings 7 and 8.

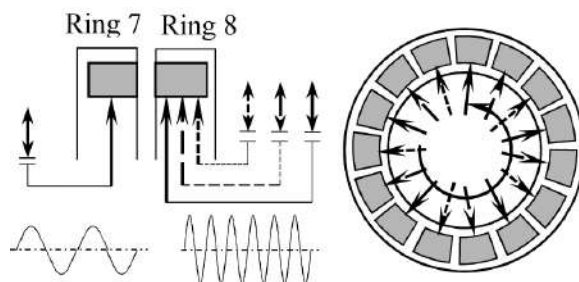


Fig.4. Electromagnetic system supply and produced electricity taking out

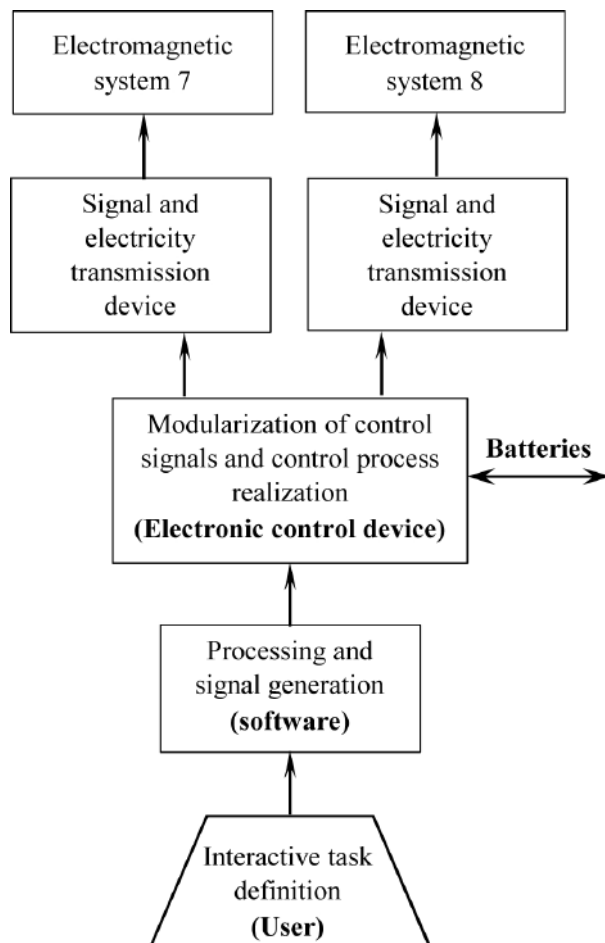


Fig.5. Structure of electric, electronic and software components of gearbox steering

The structure of electric, electronic and software components for gearbox steering is presented in Fig. 5. All of them it is necessary to be developed. Similar design solution of electromagnetic system 7 and 8 existing and is applied in hybrid automotive drives (Fig. 6). This systems already operates as motor-generators, but the design parameters have to be significantly adapted to the new specific function.

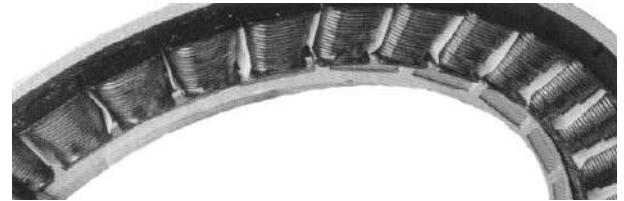


Fig.6. System of electromagnets in motor-generators of automotive hybrid drives

Signal and electricity **transmission devices** (Fig.5) are important components which have to provide this kind of transmission by mechanical contacts. Peripheral speed of the ring 7 is the less in comparison with peripheral speed of the ring 8. This is the reason that transmission device for the ring 7 is settled in the gearbox housing in the area of the ring (Fig.3). Transmission device for the ring 8 is settled in the region with much less diameter in order to reduce sliding speed in the contacts.

Electronic control device (Fig. 5) is electronic component which carry out the set of control functions. The main is modularisation of the signals which provides necessary relative speed between rings 7 and 8. By variation of this speed vary the transmission ratio of gearbox and speed of vehicle. The speed is in relation of difference in frequencies in electromagnetic fields of electromagnetic systems in rings 7 and 8, including electromagnetic sliding in relation to the gearbox load. In order to reduce this sliding it is necessary to increase the intensity of electromagnetic field by variation of electric current which supply electromagnets. This variation is the second function of electronic control device. The third one is communication with the batteries, filling and consumption of electricity. Provision of some ranges of transmission ratio consume electricity and some ranges produced electricity by engine power. Vehicle inertia also produces electricity (vehicle braking). The function is to keep the level of occupancy of the batteries in order to keep reserve of electricity for normal operation of gearbox and to have the free space for battery charge when existing power surplus. Engine power can supplement electricity and when existing electricity surplus usage of engine power is reduced. In this way vehicle efficiency is significantly increased.

Software is component which provides flexibility of the steering process of gearbox and of power transmission of vehicle (Fig. 5). Electronic control device is design solution with fixed carriers of functions. Variation of electronic device function execution can be managed by the software. The software make processing of parameters which identify operating conditions and create signals (electronic values) which manage operation of electronic control device.

5. CONCLUSION

Design solution of automotive gearbox presented in the article presents technical innovation of high level with the new principle of action. This are hybrid technical system with mechanical components (planetary gear system), electric motor-generator with electronic and software control in mental interaction with the user. Gearbox provides continual variation of transmission ratio, balance of energy and significantly increase of efficiency. Dimensions are significantly reduced and specific load capacity is increased. This results are achieved by using the actual approach in engineering design, by constrained action in order to increase the level of existing design solution.

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Invited Paper in Plenary Session:

FORMATION OF A VIRTUAL DESIGN DEPARTMENT FOR DEVELOPMENT OF HIGH-TECH PRODUCTS IN AN SME

Gorazd HLEBANJA
Marjan JENKO

Abstract: *This paper describes a new structured approach to designing a high-tech product in an SME. The design process is centred on a Virtual Design Department (VDD), which provides research, development and design services to the SME. The VDD and SME collaborate on the project base. The objective of the collaboration is to design a high-tech mechatronic product that provides a long-term revenue stream to the SME. The product design takes place concurrently in the SME and the VDD, with progress being shared using contemporary ICTs.*

Key words: *Concurrent engineering, Distributed design, Product development*

1. INTRODUCTION

Sixty-six percent of the Slovenian industrial base is comprised by SMEs – they generate sixty percent of revenue. In Germany, which is the powerhouse of the European economy and is essential for European stability and growth, sixty percent of the industry is made up by SMEs. These account for fifty-three percent of revenue. Despite their apparent importance, SMEs are incredibly sensitive to necessary research and development costs. This is an inherent problem due to the small size of SMEs and the ballooning cost of R&D.

The potential to develop high-tech mechatronic products in SMEs is thus significantly lower than in global enterprises. Production volumes in the latter are high since products are designed for global markets. Large production volumes can sustain the cost of multidisciplinary R&D departments that are needed to develop high-tech products.

Most of new environmentally sustainable products need to include connectivity to the information network, they are to work without failures throughout their life time and must have low energy consumption. They have to be produced in a sustainable manner - most materials have to

be reused and the overall harmful influence to the environment must be kept at lowest possible extent through the product life time.

The majority of modern products get their functionality from different technical domains: namely from mechanical engineering, electronics, ICT, control engineering and others. This is the case for both the products the market accepts as complex (cars, smart passive buildings, personal communication devices) and for supposedly simple products (smart tooth brush, gadgets for telemetry in fitness, running shoes with adaptive sole stiffness, other leisure time products).

2. THE VIRTUAL DESIGN DEPARTMENT AND THE SME

Creativity, innovation and many different fields of expertise are needed in modern product design. A critical mass of different specialists is required for work to be done. Most SMEs do not produce enough operating income to afford the inclusion of design teams with different fields of expertise to their payroll. In the past, when less interdisciplinary development was possible and needed, design departments could be smaller and less expensive.

Many products need to be refreshed every three to six years, i.e. a new version needs to be brought to the market. If a design cycle is shorter, then cost control requires periodical dismissal and rehiring of internal design teams. This is, however, problematic as experts tend to not want to return to an employer after having been dismissed.

The costs and complexity of interdisciplinary development are pushing SMEs out of the field of modern interdisciplinary product design. The result is that only global enterprises can fund the development and production of state of the art industrial products. This is already apparent in the computer industry with smart phones and other consumer electronics.

The majority of SMEs want to grow. However, monitoring shows that the SMEs expand at a slower than predicted rate, since their expertise can only cover a small niche – despite the small size a number of competitors will often be targeting the same niche. Hence, the number of employees and their remuneration needs to be carefully controlled to avoid unnecessary expenses. As a result the SMEs can only focus on improving existing products.

From a design standpoint, a problem arises when incremental improvements are no longer enough or when a new market niche is selected that requires solutions based on expertise that is not present within the SME. The immediate hiring of new experts is not an option since associated expenses can easily endanger financial flows. Microenterprises are even more vulnerable to these issues. The proposed option is to restructure the SME product design process. The formation of a VDD for SMEs addresses the issue of complex work costs: design and development expenses are kept at manageable levels since the VDD provides specialists for interdisciplinary work that is needed in modern product development.

2.1. The specifics of a mechatronic design

Structuring the design work, i.e., design systematics have been developed and put into practice especially in Germany in the second half of the last century. This is founded in the work of Pahl and Beitz from 1980, which has been gradually improved to this day [1].

In the 90s Suh developed [2] more abstract approach, aimed at mathematical formulation of activities of the design process. Customer needs are mapped by axiomatic rules into a functional structure of the specific design process.

The theory of Inventive Problem Solving (TIPS, TRIZ - Teoriya Resheniya Izobretatelskikh Zadatch), developed by the Russian inventor Altshuller, is based on the concept that an invention is a result of technical contradictions being solved. The development of a technical system is managed by the laws of technical system evolution [3]. TRIZ aims to describe problem solving in computer aided innovation [4], [5].

The specifics of a mechatronic design involve the use of many different technologies to deliver the functionality of the mechatronic product, which is defined as a composition of sensors and actuators in a mechanical core, while its informational structure makes its processes possible [6]. Consequentially, the development of modern high tech mechatronic products requires interdisciplinary collaboration – at the very least specialists for mechanical design, electronics, ITs and process control are involved. The first guideline on mechatronic design was VDI-2422 [7]. Mechanical structure, circuit layout and software are to be developed separately, with no interdisciplinary linkage in the design process.

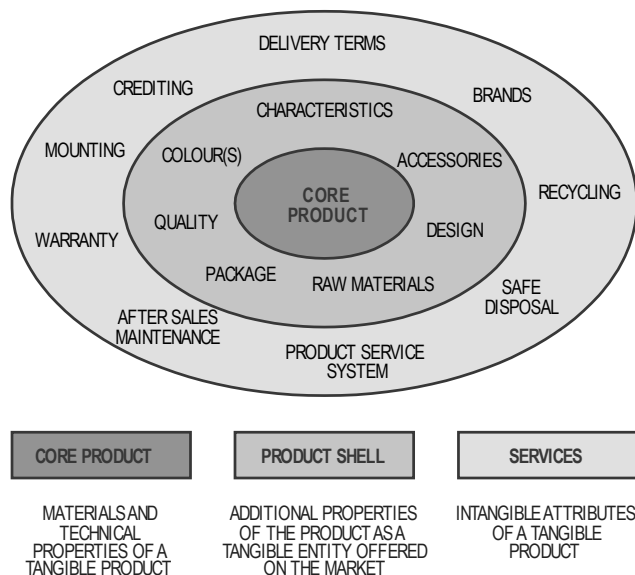


Fig.1. Modern product design from a product life cycle perspective [13]

VDI-2206 [8] defines a Domain integration approach to system design. Introduced is the V-model [9], which is a graphical representation of the proposed structure of system development. It summarizes the main design steps in conjunction with the corresponding deliverables within a computerized system validation framework. The left side of the "V" represents the decomposition of

requirements and the creation of system specifications. The right side of the "V" represents the integration of parts and their verification [10]. The "V" design is structured into sequential phases.

Modern environmentally sustainable design accounts for all the stages of a product's life cycle [11], [12]. The product is designed with the aim of yielding the most functionality, while having the least impact on environment. Many aspects of product design are responsible for delivering a high ratio of functionality to environmental impact.

Figure 1, adapted from [13], illustrates the complexity of modern product design from the perspective of a product life cycle. The model is aimed at encapsulation of most design activities.

Modern product design, including mechatronic design requires expertise, methods and tools from several technical disciplines. The core mechatronic expertise consists of control, mechanical, electrical and software engineering. The implementation of specific processes can involve many other fields of expertise. The need for collaborative interfield development is evident. Collective competence, which contains all the individuals' knowledge and experience, as well as weaknesses in some technical fields, need to be recognized at the beginning of the project. The variety of possible methods and potential uses of different tools make recognition of good design choices more difficult. "Over-informed" engineering task force can easily decrease effectiveness of the design process. The competence of lead engineers is of vital importance in order for design efforts to culminate in a working product.

2.2. The modern ICT and the internet support for the modern product design schemes

Modern products are designed, produced, utilized and recycled in a global environment. Transport and information exchange are essential throughout the phases of a product life cycle.

Contemporary Information and Communication Technologies (ICT) provide different means and methods for e-collaboration in a modern product design. The key requirements for the implementation of the design and development collaboration framework are: the integration of geographically spread resources; synchronous and asynchronous communication; data, file and document management; project management; and a common cooperation space in a distributed environment. The security and reliability of modern ICT platforms are at a level, which adequately complements the informational needs of modern engineering, production and business processes.

2.3. Background on development of the VDD

The formation of a VDD for gaining the mastery over development of a high tech mechatronic product in an SME is derived from [14]. The long term efficiency of an SME within the constraints of global markets is addressed in the cited paper. An innovative Design and Development (D&D) structure has been proposed by Peklenik [14], [15] in order to address discrepancies between the product development capabilities of an SME and the technological needs of modern product design.

functionality could not be found using internet search engines.

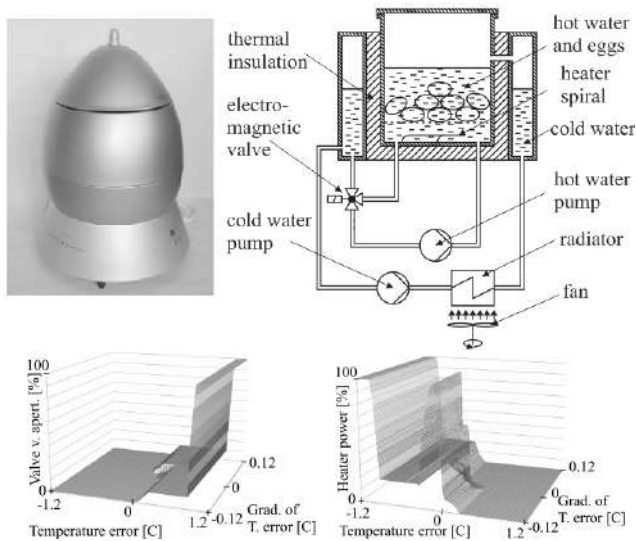


Fig.4. The Golden egg apparatus with block diagram and temperature control surfaces [19]



Fig.5. Functional and ergonomical modernization of an industrial kitchen product line.

A new generation of thermal appliances for the industrial kitchens, Figure 5, has been developed; production costs were reduced by 20 %, while energy management and styling were improved.

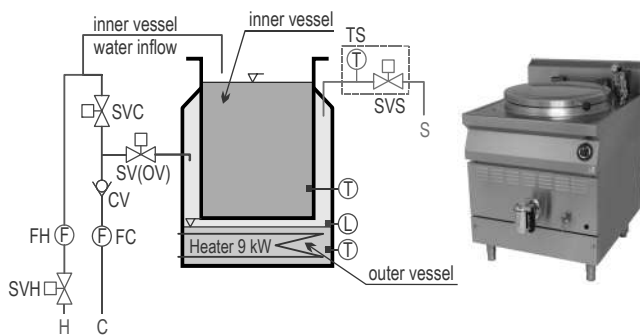


Fig.6. The block diagram and the form of the kettle for cooking under pressure, cooking and boiling foods

Design and development of a multipurpose kettle in Figure 6, which can cook under pressure, boil, cook and warm up various foods, is under way. Two to three separate kettles and an additional generator of steam are usually needed for the purpose. Device specific methods for energy regulation and observability of thermal processes via the information network are essential components of such apparatuses.

4. DISCUSSION

It is authors' understanding that modern ICT have more potential for information transfer, storage and for design support than some established organization of design work can use. The work paradigm in many companies is that members of the design teams must work at the same location; many times even in the same office space, oftentimes even without cubicles, not having even cubicles but literally with each other in plain sight. Many managers, who have not gone through the process of gradual promotion, but have been able to skip ranks lack the experience necessary to judge the progression on a project, which does not take place physically in front of their eyes. In principle they are supportive of modern design, but want to have simple means of control.

To work better, work contributors have to be highly motivated and dedicated to their work. The source of motivation can be their willingness to contribute and to discover things or simply the fear of losing their job and economic existence in the recession.

The proposed organization of development builds on principles of concurrent engineering, remote collaboration, high motivation and use of competent engineering workforce, Figure 7.

Information Technology (IT) security in SMEs is not jeopardized, since collaborators use only a small set of tools for remote cooperation. Modern internet routers and switches in SMEs can be easily configured for controlled information exchange between some internal and external workstations, while simultaneously keeping overall security at the highest level.

The virtual development environment for smaller sized companies became of utmost interest during this short time period and IT technology is enabling it more each day through developed and emerging technologies. So, one could expose cloud concept.

Yet, another example for improving development and innovation rate can be state supported and through big corporations, SMEs and research organisations driven backbone system like Strategic Development Innovation Partnerships – “Factories of the Future” [20]. The organization structure incorporates active members (e.g. value chains in toolmaking or pharmacy) who meet several core technology clusters (e.g. nano-technologies, robotics, new material production technologies, management technology for systems, etc.). Such a system includes development of common solutions (e.g. human resources, CA services and technologies, intellectual properties, etc.). Partners can meet in several floors and in appropriate circumstances projects derive with a common goal in development of a product, service or process.

5. CONCLUSION

The functioning of a VDD linked to an SME for the purposes of product development is presented in the paper. Work differs from classical outsourcing of design activities in the following ways:

a) The seamless integration of design activities between the SME and the VDD is enforced by policies defined by the design process,

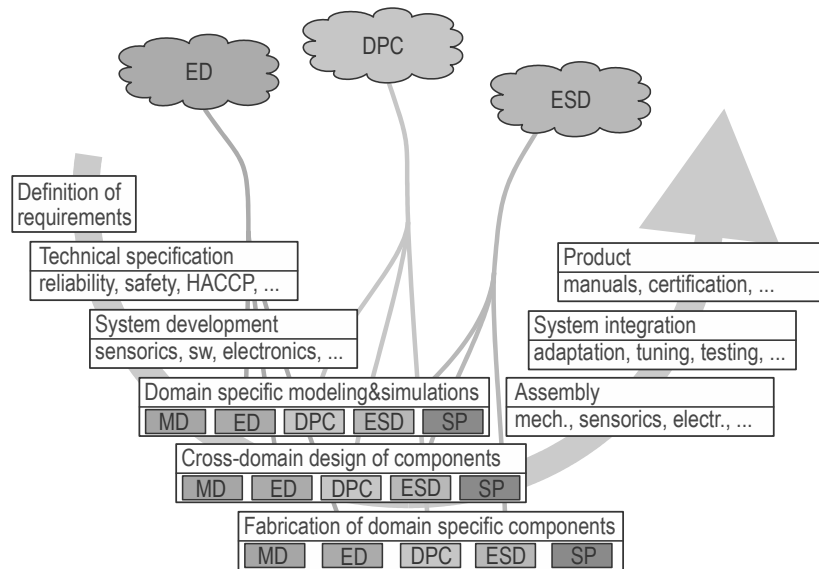


Fig.7. Concurrent engineering in the VDD

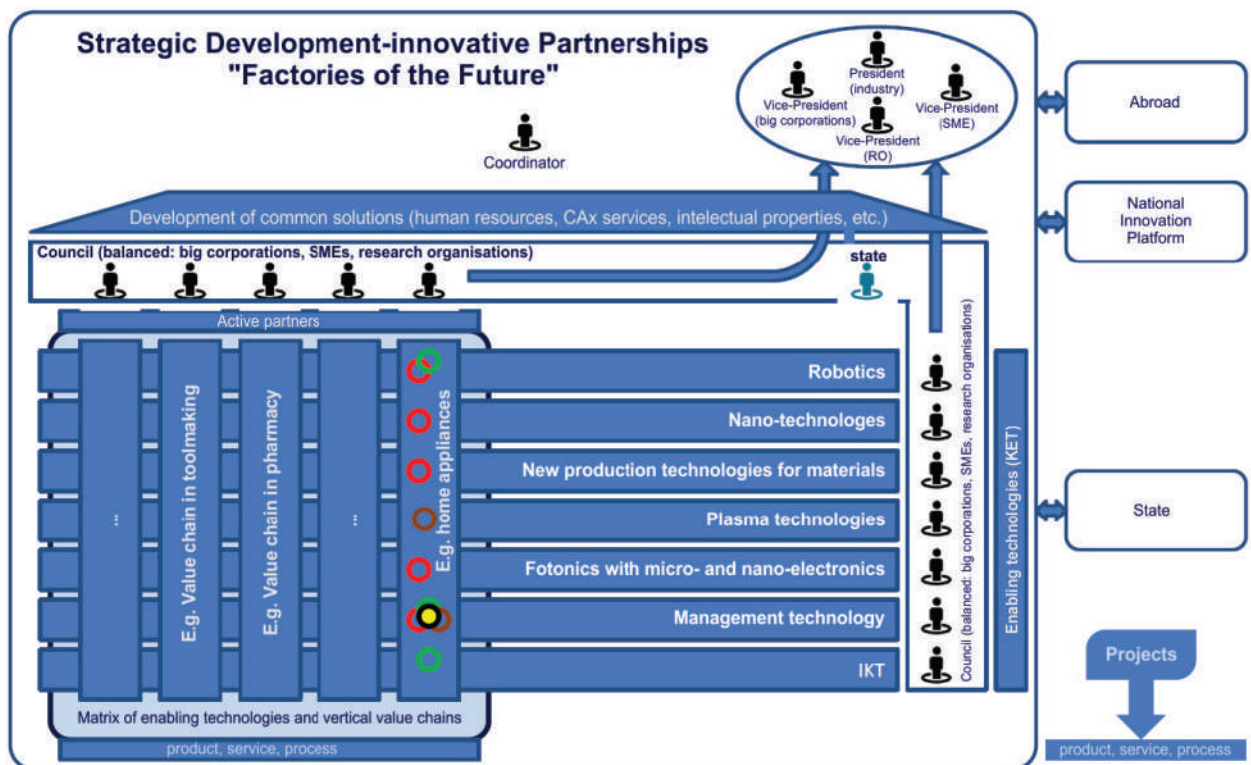


Fig.8. Strategic Development Innovation Partnerships – “Factories of the Future” [20]

b) The system encourages information and knowledge sharing among the SME and VDD contributors,
 c) Since the involved parties within the collaborating SME and VDD share information and knowledge throughout the project, a concurrent development of mechanics, embedded systems and control algorithms is the design policy.
 Design and development costs in the presented VDD and SME collaboration are about a third lower compared to the hypothetical costs of development in an SME alone. Marketing data compiled by the SMEs marketing department shows that the developed products are competitive. The long-term existence of the VDD is assured, as there are enough SMEs in need of its services.

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COST ESTIMATION IN THE EARLY STAGE OF PRODUCT DEVELOPMENT

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Aco ANTIĆ

Abstract: Manufacturing costs are one of the main measures of success of the developed product design, both in terms of cost effectiveness and the aspect of market success. The inability of enterprises to quickly and effectively determine the costs can significantly endanger their survival on the global market. Depending on the stage of product development, there are different ways of determining costs, in the early stages cost estimation is usually performed, while the later stages are more focused on detailed cost calculation.

The main subject of this research is related to the importance of cost estimation in the early stage of product development. The paper shows the classification of methods for cost estimation with the detailed representation of the analogical method for cost estimation. Its verification is performed for the case of cost estimation of proposed product – housing, for different variants of manufacturing technologies, through the application of the developed software for cost estimation.

Key words: Product development, Conceptual design, Conceptual process planning, Cost estimation

1. INTRODUCTION

The main tasks of modern manufacturing are determined by the level of development of manufacturing technologies on one side, and the characteristics of modern market demands on the other side. Customers in the modern market no longer ask the question “Is it possible to manufacture the product?”. They ask instead “What is the optimal manufacturing process?”. Optimal manufacturing process is understood as the process which result is a product with minimum manufacturing costs and time and who meets market requirements in terms of

functionality, reliability, quality, aesthetics, environmental protection, etc. [1, 2].

Cost estimation is very significant in terms of project sustainability and cost reduction in product development and product manufacturing. Based on the cost estimation, more successful decision-making can be achieved about the selection of optimal product alternative from the aspect of its cost-effective manufacturing [3, 4]. One of the important tasks of the cost estimation is to ensure sufficient amount of reliable information about the cost elements in the early stage of product development. For that purpose, appropriate methods for cost estimation are developed, such as intuitive, comparative, analogic and parametric [5, 6].

This paper is focused on the significance of the cost estimation in the early stage of product development, or in the conceptual process planning stage. The classification of methods for cost estimation is shown and one analogical method for cost estimation is shown in detail. The development of the software for cost estimation is represented, as well as its verification for the proposed example part – housing, for different variants of manufacturing technologies and production volumes.

2. SIGNIFICANCE OF THE COST ESTIMATION IN THE EARLY STAGE OF PRODUCT DEVELOPMENT

The influence of the stages of product development on the total product cost, time and quality depends on the product type, production type, environment and many other techno-economic factors. Generally, the influence of the conceptual process planning is around 5-10% in the total manufacturing cost, and wrong decisions in this stage of process planning can affect the increase in manufacturing cost in more than 60%, Figure 1 [7, 8]. It is therefore necessary to consider the production problem as early as possible, in the stage of product design, i.e. the development of product concept, because the costs are even higher due to product changes if they are made in the later stages of product development [9, 10].

However, despite the significant progress in manufacturing and information technologies, better decision-making in the early stage of product design is very complex due to the reason it assumes a number of unpredictable factors in manufacturability, quality, reliability, maintenance, etc.

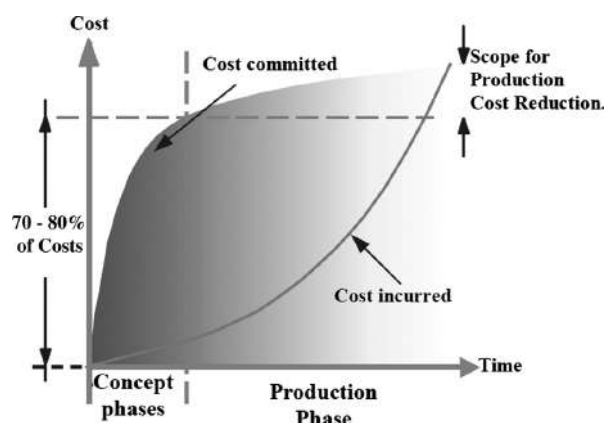


Fig.1. Manufacturing costs in product development [8]

Two basic functions of product development are product design and process planning which should be integrated as much as possible [11]. The process planning is a set of activities used to decide how and with what resources a product can be manufactured in cost-effective and concurrent way [12]. The process planning is a complex function which is divided into several hierarchical levels. First and the highest level in this hierarchy is the conceptual process planning, as an activity of preliminary assessment of possibilities and effects of manufacturing a conceptually planned product. The output results of the conceptual process planning are used in the stage of the conceptual product design, as well as in the stage of the detailed process planning [13]. One of the main tasks of the conceptual process planning is the estimation and reduction of manufacturing costs, which is given in more detail in the following chapter.

3. MANUFACTURING COST ESTIMATION

Determination of manufacturing costs is the basis for making good decisions in manufacturing practice because production systems that are not able to quickly and successfully determine costs have the problem of development and survival in the global market. Depending on the stage of product development, there are different ways to determine these costs; in the initial stages the estimation is commonly performed, while the cost calculation takes place in the later stages.

Many researchers have studied the problem of cost estimation, such as those in [3-6, 8-11, 14] and many others. A number of methods and techniques for the calculation and estimation of manufacturing costs are developed. Some authors have attempted to include as many manufacturing processes as possible, while the others have oriented solely to certain groups of similar processes. In the papers [5, 6], the classification of methods and techniques for the determination of costs is given, according to the way of realization, stage of application in the process of product development and precision of the methodology, Table 1.

The given table shows that the analogical and parametric methods are recommended for the conceptual process planning. The cost estimation within these methods is performed on the basis of known costs of similar parts, i.e. product and process parameters taken for comparison.

Appropriate information about conceptual model of a product, selected material and conceptual process is necessary for the cost estimation. Specific costs in this stage of product development are orientation (level of error is 14-30%), which can aggravate the quality assessment of product design. However, these costs can be very important and sufficiently reliable for this level of process planning, because the main goal at this stage is not to determine actual or final costs, but to estimate costs and select appropriate and most favourable variants of manufacturing technologies.

For the detailed or final process planning, analytical methods for cost calculation are recommended, such as ABC (*Activity-Based Costing*) method, or TD ABC (*Time-Driven Activity-Based Costing*), and others. In metalworking industry modern methods are based on features (feature based methods) [14].

By the content of activities the conceptual process planning substantially matches with the DfM/DfMA-Design for manufacturing and assembly, where manufacturing cost estimation is also included. The most famous DfMA methodologies are: *Boothroyd-Dewhurst (DFMA)* [9], *Lucas (DfA)* [15] and *Hitachi-AEM (Assembly Evolution Method)* [16], which are represented in more detail in [17]. On the basis of these three methodologies the corresponding softwares are developed, including the most famous among them, *DFMA software, Boothroyd Dewhurst Inc. USA* [18], within which the *DFM Concurrent Costing* module is covered. This module is focused on the selection of manufacturing processes and quick estimation of manufacturing costs.

In recent years, major software companies in the field of CAx expanded their products/softwares towards the development of PLM systems. So, for example, PTC which developed PTC/Creo, is also responsible for Windchill (www.ptc.com/product/windchill/cost), Siemens which developed UniGraphics (NX) has its PLM system (www.plm.automation.siemens.com/en_us), Dassault Systems which developed CATIA and commercialized SolidWorks has its own PLM Solutions (www.3ds.com). All these systems offer cost-estimating capabilities during product life cycle, and also including the manufacturing cost estimation [19].

Table 1. The classification of methods for cost calculation and estimation [5, 6]

Method	Description	Best application	Accuracy (level of error)
Intuitive	Evaluation of cost regarding personal knowledge and intuition	Global production planning	30-50 %
Comparative	Evaluation using similar parts	Global production planning	30-50 %
Analogical	Case-based evaluation, definition of main parameters for comparison with previous cases	Conceptual design	14-30 %
Parametric	One or several parameters are chosen to be critical. They are used along with coefficients to evaluate the cost.	Conceptual design	14-30 %
Analytical	Direct and indirect costs are considered. Each cost is calculated and then they are all summed to get the product cost	Detailed design	5-15 %

In addition, a significant number of CAD-based software is developed for the cost estimation and/or manufacturability analysis of product design. These softwares can be integrated and embedded as specific modules in the corresponding CAD systems [8, 19]. Some of these softwares are the following:

- **DFMPro** (dfmpro.geometricglobal.com) which is successfully integrated within various CAD platforms. It is designed for the manufacturability analysis in CAD environment (DFMPro-Rule Manager). Based on the rules and guidelines, corrections of design can be made within this environment, as well as the cost estimation in machining, sheet metal processing, plastic injection moulding, casting and assembly.
- **SolidWorks Costing** (www.solidworks.com) which is fully integrated into SolidWorks, is designed for the cost estimation for sheet metal and machined parts.
- **Autodesk Simulation DFM** (www.autodesk.com) which can be integrated into various CAD platforms, is designed for the manufacturability analysis for plastic parts, manufacturing cost estimation and environmental impact assessment.
- **SEER-DFM** (www.galorath.com), designed for users in the area of management, finance, engineering, industrial design, as well as manufacturing. It is used for the evaluation of process parameters and trade-offs impacting various factors (such as, ease of manufacturing and assembly, material and process selection, failure and repair rates, cost estimation).
- **MicroEstimating** (www.microest.com) developed by *Micro Estimating System Inc.*, offers computer-aided process planning and computer-aided cost estimation for the machining and fabrication industries. MicroEstimating employs Machine Tool Emulation, Knowledge-Based Machining, and Automatic Feature Recognition.
- **Costimator®** (www.mtisystems.com) is designed for easy configuration of products assuming the capabilities of a manufacturing facility. The system contains hundreds of process and cost models which assumes a wide spectrum of manufacturing processes and features that are implemented for prebuilt, ready-to-use manufacturing cost models.
- **aPriori Production Cost Management** (www.apriori.com) is a tool for manufacturing cost estimation for machining, casting and plastic injection moulding.
- **CustomPartNet** (www.custompartnet.com), is an online tool for manufacturing cost estimation. It allows users to perform quick and efficient calculations on the basis of which the product design and cost estimation are facilitated.

The next chapter presents the software for cost estimation which is based on analogical costing methodology and suitable for cost estimation in the early stage of product development, i.e. at the conceptual process planning stage.

4. COST ESTIMATION SOFTWARE

4.1. Basis for development of cost estimation software

In developing the observed software, a method that belongs to the group of analogical methods is applied, according to Swift and Booker [10]. This method is based on the cost estimation for similar parts, whereby basic parameters for comparison with previous solutions are defined. The parameters match the data from manufacturing practice of Great Britain industry that are systematized and analysed statistically, and then embedded in appropriate tables and dependency graphs. The algorithm structure of cost estimation software as well as the module of developed conceptual CAPP system is shown in [20].

Manufacturing cost model (T) of specific component is composed of: material cost (T_m), processing cost (T_i) and surface coating and/or heat treatment cost (T_t):

$$T = T_m + \sum_{i=1}^n T_i + T_t \quad (1)$$

Material cost (T_m) are defined as the volume of raw material V_P multiplied by the cost of material per unit volume C_{mt} :

$$T_M = V_P \cdot C_{mp} \quad (2)$$

The total volume of raw material is easily determined when it comes to simple shapes such as rod, sheet and others. However, when the total volume of raw material is complex or not known, then it can be calculated as the finished volume of component V_f and waste coefficient W_C , which takes into account the amount of material to be removed and is dependent on particular manufacturing process and the shape complexity of component. Thus, the formulation is the following:

$$V_P = V_f \cdot W_C \quad (3)$$

As a basis for determining the processing cost (T_i), the cost of producing an "ideal product" are taken into consideration, with correction of the relative cost coefficient that shows the difference between the observed and the ideal product, depending on the type of material, shape complexity, specific dimensions, tolerances and surface finish:

$$T_i = P_C \cdot R_C \quad (4)$$

where:

- P_C – The cost of producing an ideal product,
- R_C – Relative cost coefficient

The relative cost coefficient is calculated based on the following equation:

$$R_C = C_C \cdot C_{mp} \cdot C_S \cdot C_{ft} (\max C_i \text{ ili } C_f) \quad (5)$$

where:

- C_c – Relative cost associated with shape complexity,
- C_{mp} – Relative cost associated with specific material-process compatibility,
- C_s – Relative cost associated with a section thickness of the product
- C_t – Relative cost associated with tolerances,
- C_f – Relative cost associated with surface finish.

For an ideal part/product all the represented coefficients are equal to 1. After determining the relative cost associated with tolerances C_t and relative cost associated with surface finish C_f , the coefficient that has greater value is taken into account and is designated with C_{ft} . For determining some coefficients that exist in the formulations for calculation, it is first necessary to perform the classification of products according to geometrical shape. All products/parts are divided into three classes (A - rotationally-symmetrical and rotationally-asymmetrical parts, B - prismatic and full boxed parts, C - prismatic and thin-walled boxed parts), which are further divided into five different subclasses according to classification system given in [10].

4.2. Application of the developed software

Representation of the given method for the cost estimation and developed software is performed for the sample part – housing ER1.002, made of aluminium alloy EN AW-2024, Figure 2.

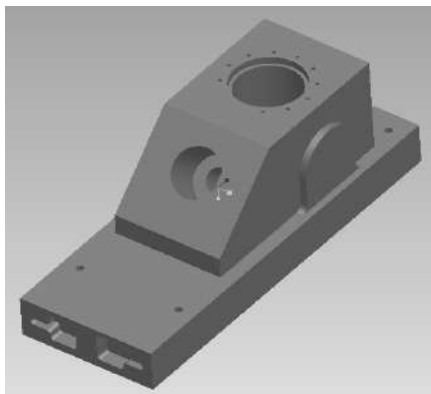


Fig.2. 3D model of housing

This type of aluminium alloy belongs to the alloy series 2000, so called “hard alloys” which are commonly used in aerospace and defence industries. They are characterized by high firmness but require annealing or long-term aging in order to achieve maximal characteristics. Good machinability is a positive characteristic of this series but it lacks good corrosion resistance.

Based on structural and technological characteristics of the parts, such as basic shape and complexity, material, quantity and others, the following variants of manufacturing processes are taken into consideration by using the software – module for manufacturing process selection [20], table 2.

According to the classification system, the observed housing belongs to the B3 class of parts – prismatic and full boxes, Figure 3. In order to analyse the cost, from the aspect of planned production volume in an appropriate

production system, the quantities that belong to various methods of production are taken into consideration (10, 50, 100, 400, 1.000, 10.000 and 100.000 (parts/year). These production methods vary from job and small-scale production to large-scale production.

Table 2. Variants of manufacturing processes

Variant No.	Manufacturing process
Variant 1	CNC machining from full blank/raw material (CNC)
Variant 2	Gravity Die Casting (GDC) + CNC machining (CNC)
Variant 3	Pressure Die Casting (PDC) + CNC machining (CNC)
Variant 4	Closed Die Forging (CDF) + CNC machining (CNC)

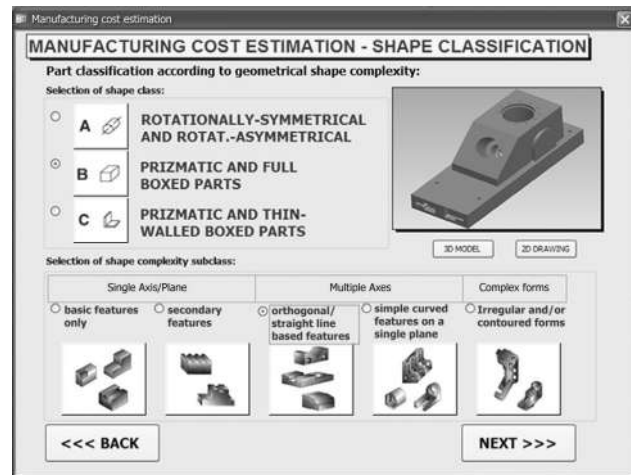


Fig.3. Classification of the housing in the software

In the following, Figure 4 shows the input data for the variant 1 which is consisted of one basic manufacturing process – CNC machining and heat treatment, with production quantity of 10 parts/year. Based on the input results, the output results assume the obtained values of costs for the housing and observed variant of manufacturing process. These costs are divided into material cost, processing cost and heat treatment cost, with the graphical representation of the share of each of these costs in the total manufacturing cost, Figure 5.



Fig.4. Representation of the applied software for the estimation of housing manufacturing cost for variant 1

Variation of the data for all four given variants of manufacturing processes for housing is performed by changing the input values with the variation of production volumes as well. The estimated output values of manufacturing cost are obtained as the output result. These results are given in Table 3.

Material cost:	In pence:	In eurocent:
$T_m = V_p \cdot \text{Cost} = 1400000 \cdot 0.00083 = 1168.64$	1168.64	934.91
Processing cost:		
$T_i = 2(P_d + R_d) = 990$	990	792
Heat treat./surf. coat. cost:	200	160
Total manuf. cost:	$T = T_m + T_i + T_t = 2358.64$	1886.91

Fig.5. Output results of the estimation of housing manufacturing cost for variant 1

Figure 6 shows the dependency graph representing dependency of the estimated manufacturing cost on the input quantities for the selected variants of manufacturing processes for the observed part – housing, based on the results from Table 3. According to the obtained results of cost estimation, it can be concluded that the cheapest variant for job and small-scale production is the CNC machining, while the other variants are more favourable for middle-scale and large-scale production.

Table 3. Cost estimation values (given in €/parts)

Q (parts/year)	Variants of manufacturing processes			
	Var. 1	Var. 2	Var. 3	Var. 4
10	18.87	487.93	567.13	710.16
50	12.53	174.21	195.22	210.75
100	11.42	54.54	62.45	77.19
400	10.72	19.52	20.31	24.40
1000	10.47	10.98	11.85	13.71
10000	10.24	6.49	6.62	7.23
100000	10.00	5.90	5.87	6.37

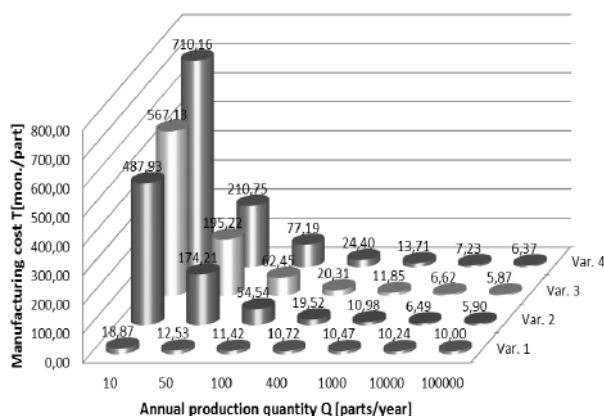


Fig.6. Graphical representation of the dependency of cost estimation on production volume for the adopted variants of manufacturing process for housing part

5. CONCLUSION

Product development takes a central place in product life-cycle in which product quality is generated, as well as large number of errors which should be eliminated at the earliest possible stage of development.

The main activities in early stage of product development are: conceptual design and conceptual process planning whose output result is also the cost estimation. An enterprise which is unable to quickly and successfully estimate costs and on their basis to define manufacturing costs i.e. product price, can significantly jeopardize its survival on increasingly demanding global market.

This paper shows the significance of cost estimation in the early stage of product development, the classification of methods for determining costs, as well as the appropriate softwares for cost estimation. Brief representation of the analogical methodology for cost estimation is given. On the basis of this methodology the software with identical name is developed. This software is applied for estimation cost of the observed part-housing for various variants of manufacturing technologies and production volumes with the results given in this paper.

The cost estimation methodology described in this paper enables quick estimation of manufacturing cost which significantly facilitates the work of designers. Although analogical methods are not as reliable as analytical, for this level of design they can be very significant and sufficiently reliable due to the fact that the main goal in this stage is not to determine real costs but to estimate manufacturing cost and select the most favourable variant of manufacturing process. For the purpose of sustainable competitiveness almost every production organisation will be forced to adopt a philosophy of integrating product design and process planning and, according to that, to apply tools for cost estimation in early stages of product development and tools for determination and control of costs throughout the entire product life-cycle.

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IMPROVEMENT OF E-LEARNING PROCESS OF PACKAGING RAPID PROTOTYPING COMPUTER NUMERICAL CONTROL MACHINE SYSTEMS

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Abstract: Research is focused on improvement of packaging prototyping through distance learning in the field of graphic engineering and design. The improvement includes the visualization of packaging prototyping CNC (Computer Numerical Control) systems, with elements of theoretical knowledge as well. 3D visualization represents the real graphic systems, with interactivity between virtual model and user. Virtual 3D visualization allows examination of all system elements with the insight into the knowledge base containing information about the theoretical and practical functioning of each element. packaging, e-learning, rapid prototyping, virtual space, contemporary graphics systems.

Key words: packaging, e-learning, rapid prototyping, virtual space, contemporary graphics systems.

1. PACKAGING AS AN IMPELLER OF PRINTING INDUSTRY IN THE FUTURE

By analyzing the market trend, it is estimated that the market for packaging will, in the next five years, reach 303.60 billion, with a projected CAGR (compound annual growth rate) of 7.17% from 2015 to 2020. Figure 1 shows the analysis of the market value of packaging industry in 2015 and the expected market share in 2020, expressed in billions of dollars for specific types of packaging materials. As it can be seen on the graph, usage of paper and cardboard materials increases and exceeds the use of plastic, metal, glass and other packaging materials. Paper and cardboard segment is projected to grow at the highest CAGR of 8.08% from 2015 to 2020. It is expected that the market of paper and cardboard packaging will reach a

value of 213.4 billion dollars, by 2020 [1]. Growth in the market will affect the development of small and medium-sized enterprises for the production of paper and cardboard packaging. Innovative shapes, the process of assembly, storage and distribution of packaging products will be the key to good positioning in the market. Thus, the need is created for devices that will allow the quick prototyping of packaging, and accelerating the development of new products. The aim of the paper is to present development of applications for e-learning which will in an interactive manner, using a virtual environment, simulate packaging prototyping on CNC systems for die cutting. Use of CAM systems for cutting blanks is an essential process, whether production is large or small.

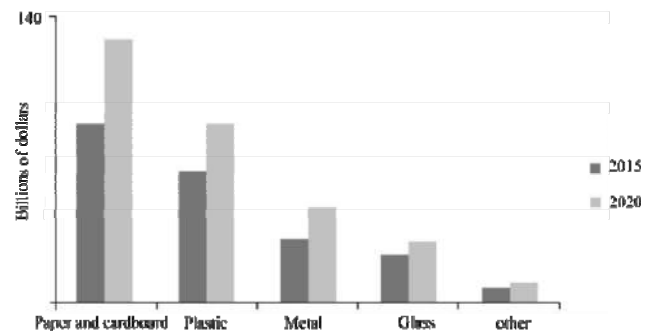


Fig.1. Analysis of the market value of packaging from 2015 to 2020.

In order to make the learning process more efficient, all advantages and benefits of e-learning were incorporated into knowledge transfer application, along with a virtual model of prototyping CNC device.

2. E-LEARNING

The beginning of XXI century is unequivocally marked by the exponential growth of computer applications in all aspects of engineering. According to definition of the American Association ASTD (American Society for Trainers and Development) from 2001, e-learning is an instructional content or learning experiences delivered or enabled by electronic technology.

E-learning is used for a decade as learning aided by information and communication technologies. With the introduction of new "instructional" media, such as images, slides, film, animation, begins a new era in the field of education. The use of multimedia system allows simultaneous presentation of several different media sources – text, image, video, sound (speech, music, various sound effects), graphics, animations, as well as the storage, search, and data processing. An animation represents the main aspect of multimedia educational presentations. The two-dimensional animation is the most commonly used type of animation, but the 3D animation is becoming more and more popular [2].

3. CAD SYSTEMS

It is certain that the computer-aided design (CAD) has revolutionized modern engineering. In engineering, CAD is widely used for the design and development of various products. CAD increases the productivity of engineers /

designers, improves the quality of the design and creates a database for production. Figure 2 shows a computer-generated technical drawing of selected slotted cardboard box, developed for production. The technical drawing is the foundation for quality of production and development of products using CAD systems.

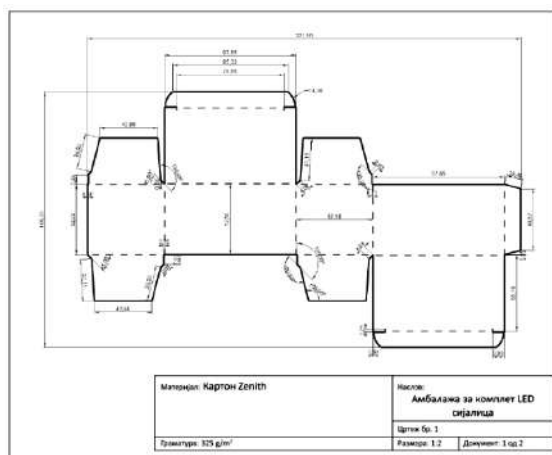


Fig.2. Technical drawing of packaging

4. PACKAGING

Packaging is shaped or unshaped material designed for enclosing or protecting products for sale, storage, distribution and use. Packaging is an inseparable companion of goods through its various phases from the final stages of production to its end user [3].

4.1. Packaging box

Packaging box represents the most common form of cardboard packaging, thanks to its properties. Cardboard boxes are usually constructed as a slotted box, specially designed to be delivered to buyers in flat 2d form, and then easily assembled [4].

4.2. Prototyping

The main objective of prototyping is the opportunity to test the physical characteristics and behavior of packaging, therefore, it is of great importance that the model is as close as possible, by the characteristics, to production product. Today, prototyping mainly uses CAD / CAM systems, where cutting machine is guided by computer CAD drawing (Figure 3) [3].

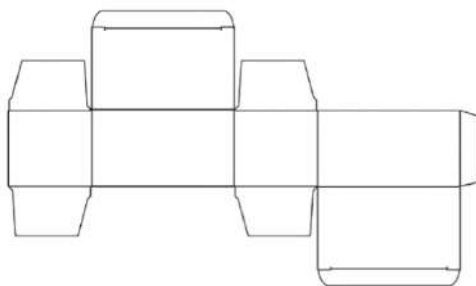


Fig.3. Cutting and creasing lines created for package box in EngView

Prototyping machine Aristomat SL 1317 was chosen for modeling in a virtual environment. Aristomat SL 1317 is

an excellent representative of the prototyping device for packaging, because, with proper selection of tools, it offers the possibility to process (mainly cut and crease) plastic, laminated and textile materials, as well as corrugated cardboard for the packaging industry.

4.3. EngView Package Designer

EngView Package Designer is specialized CAD / CAM software for packaging production. Supported materials are paper, paperboard, cardboard, corrugated cardboard, and inverted corrugated board. It is designed as a set of functional modules covering the entire process - from conceptual design to tooling for die cutting.

5. PROCEDURE OF CREATING INTERACTIVE ANIMATION

5.1. Modeling

It has been shown that 3D can affect the activation of the spatial memory, which may improve long-term memory [6]. In purpose of accurate modeling, it is necessary to perform a detailed analysis of the cutting device. 3D animation is the most common type of visualization. It allows the representation the largest as well as the smallest parts of the machine with great realism.

Figure 4 shows a wireframe model of cutting head. In addition to the basic geometric form and in their transformations, in most cases, modeling is done using modifiers. Modifiers change the shape and structure of the object. In the process of modeling various modifiers were used along with modifiers such as extrusion, chamfering, Boolean operations, smoothing, etc.

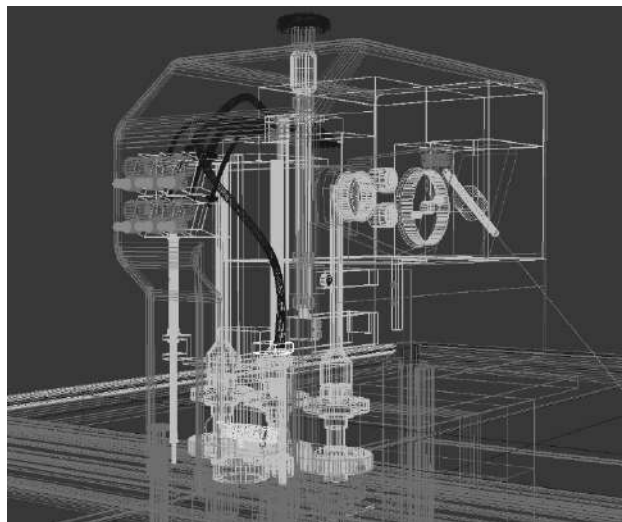


Fig.4. The wireframe model of cutting head

5.2. Materials

Given that realistic view, contributes to the reduction of abstraction of theory and therefore better adoption of knowledge in subjects [7], special attention was given to the choice of materials and lighting in the scene. The materials choice determines the way in which objects reflect light and pass it on creating indirect illumination. The material is used in a scene for color, brightness, and transparency. The materials are, in addition to light, what makes the scene more realistic.

Mapping is a method of projecting the image information (materials) to the surface. The image is not bonded to the surface but is projected mathematically, using modifiers [5]. Many materials are also used bitmaps, to simulate the appearance of complex surfaces such as e.g. marble, wood, brick, etc. In most cases, when assigning material to objects predefined materials were used, with exception of surfaces that had specific information in form of writings (Figure 5.)



Fig.5. Render with assigned materials.

5.3. Lighting

In three-dimensional graphics lighting is one of the most important factor for obtaining high quality and photorealistic presentation. The light source, in 3ds Max, generates shadows and contributes to the atmosphere and realism of the image.

The usual manner of lighting the scene consists in the use of the main light source, for the basic lighting scene, and one or more additional light sources which emphasize the shadows.

5.4. Camera

The camera determines which part of the scene or 3d model user will see. Camera position essentially determines the composition of the final image. Target Cameras consist of a camera and target. Each of these elements can be moved independently but the camera is always pointed at the target. Target is then linked with 3d model ensuring its movement along with the object. Target cameras are very easy to manage because of these characteristics.

5.5. Animation

In order to create an animation of machine Aristomat SL 1317, it was necessary to study how it operates and determine the moving parts trajectory. The animation exploits the persistence of human vision, which refers to the optical illusion whereby multiple related still images in quick succession is perceived as a continuous movement. Each individual image is in the process of animation is called frame. The animation in 3ds Max is made by using keyframes. Keyframes are made on the timeline by simply moving an object in the 3d scene (Figure 6).

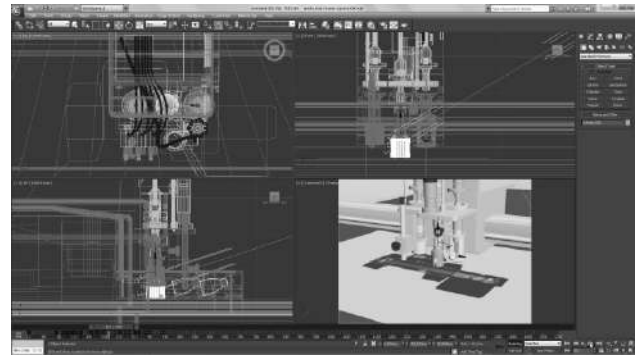


Fig.6. 3D software interface with timeline

5.6. Rendering

Rendering is the final process of 3D graphics creation, where 2D images or animations are rendered from the 3D scene. Resulting image is a derivate of the 3d models conjoint with above-mentioned factors such as lighting and material, mathematically calculated (Figure 7).



Fig.7. Final render of the 3D model

5.7. Development of interactive applications in the Adobe Flash software

After the animation is rendered in 3ds Max an interactive version of the same was made in Adobe Flash, with buttons and information on important parts of the machine.

The main feature that had to be achieved through the application is interactivity. Interactivity has a very important role in how the knowledge is adopted and the speed of adoption for users. The interactivity of applications for e-learning is reflected in the ability of users to manipulate the contents.

Using the programming language Flash ActionScript 3.0 each button is activating the specific animation. This detailed instruction is written in AS3 code. Figure 8 shows a segment of an interactive application which invites scene shown in Figure 9.

Upon completion of the animation, the new buttons appear on certain parts of the machine. By clicking on the red button learner can open a new page containing images and theoretical description (knowledge base) of a given part (Figure 9).



Fig.8. After the animation is finished learner has access to the knowledge base for any part of the device.

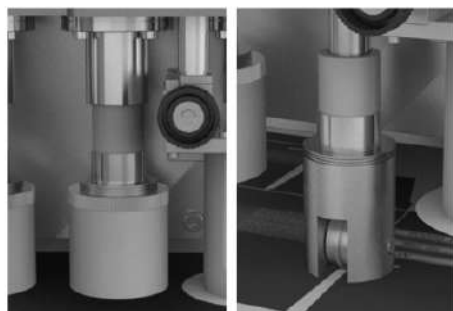


Fig.9. The page of application with the knowledge base about tools for stamping

6. CONCLUSION

Through interactive 3D animations, the learner can analyze the process of packaging prototyping for the particular case of graphic cutter Aristo Mat 1317 SL. In this way, without the real world machine for prototyping, the learner is provided with unambiguous information about operation carried out by the device and its basic parts, which significantly facilitates the acquisition of knowledge.

Further development of applications will gravitate towards the increase of the interactivity and upgrade of simulation, where the new products will be automatically animated. Automated animation could be guided by cutter lines in the same way computer guides real world CNC cutter. This will add more personalization and interactivity in learning the process.

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DEVELOPMENT OF IMPROVED WHEEL HUB PROTOTYPE THROUGH IDEALAB PLATFORM FOR STUDENTS'S CONTEST

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Abstract: The paper presents results of improving wheel hub design, of a bicycle trailer, organized through the student's contest. The whole process is based on crowd sourcing idea performed in several steps, through open iDEALab innovation platform, specially oriented towards co-working between small and medium enterprises (SME-s). The paper is also presenting the further development of the three best student's ideas which are finally prototyped on 3D printing machine. The final results are discussed in the paper.

Key words: open innovation, crowd sourcing, idea platform, product development, 3D printing

1. INTRODUCTION

Open innovation in the university-industry co-working is considered as one of the most important elements in fostering innovation performance of one region. Therefore there are many initiatives that support these activities in different ways. The Tempus project "Fostering students' entrepreneurship and open innovation in university-industry collaboration - iDEA lab" (Ref. No. 544373-2013) aims to develop a physical and virtual environment for generating, developing and commercializing innovative students' ideas through relevant trainings, mentoring and technology put at their disposal. Following the entrepreneurial or open innovation route, it will foster the co-working possibilities between universities and enterprises, especially between students and SME-s, advance employment potential of graduates and enhance the companies' innovativeness. This paper focuses on the virtual segment of the platform. It presents the idea, structure and the first results in the application of this innovation platform specially oriented towards co-working of students in solving a specific problem defined by one SME. Section 2 describes the

iDEA lab platform concept, relating its elements to the existing literature and practice on open innovation and innovation contests. The structure of the iDEA lab platform is represented in Section 3, followed by Section 4, focusing on the case study of the co-working course of study on the iDEA lab platform. Section 5 offers conclusions and opens some questions for the future research and improvement of this platform.

2. IDEA LAB PLATFORM CONCEPT

The core idea behind iDEA lab project is to motivate students and young researchers to actively use their intellectual potential to generate innovative ideas. The iDEA lab platform will be a knowledge repository and open innovation platform that will integrate all physical iDEA labs established through the project and will provide live events, lessons on specific subjects, online workshops, seminars, different multimedia contents, alumni and mentors sections. It will be the corner stone of the regional marketplace for ideas, as one of the iDEA lab project's results **Error! Reference source not found.**

Considering that companies, especially SME-s, have limited resources to support their product development processes, their management has started to look for fresh ideas and competent individuals outside their borders. It is claimed that it is more effective to encourage a diverse group of people outside the company, or even the discipline, to seek innovative solutions [1]. Companies have started to distribute problem solving tasks to larger groups of people, in order to mine the collective intelligence, assess quality and process work in parallel [3]. These strong tendencies towards crowdsourcing follow the view that large groups of people are smarter and wiser than an elite few, no matter how brilliant they are. They are better at solving problems, fostering innovation, coming up with wise decisions, even predicting the future [4]. This open approach to product development offers a great opportunity for companies to access valuable knowledge from multiple sources (including small companies, universities and consumers) for the creation of innovations. This is the outside-in process whereby new ideas and technologies are acquired from partners and brought into the innovation pipeline [3].

As the world is getting more and more networked, companies are recognizing the power of the Internet as a powerful platform for collaborative innovation [5]. The Internet helps companies gain more ideas for innovation, opening their innovation funnel through crowdsourcing and widening the scope for screening ideas [6]. Through the creation of virtual communities, it allows companies to tap into the social dimension of knowledge shared among groups of people with shared interests [5]. These communities consist of people, who interact socially, a shared purpose, such as an interest, need, information exchange or service that provides, policies that guide social interactions, and computer systems, to support and mediate social interaction and facilitate a sense of togetherness. [7]

In the search for knowledge within these communities that can deliver innovative products and services, innovation contests are being used for the acquisition of

ideas, as a mechanism chosen by companies to perform open innovation [3]. They are a way to engage with external sources of knowledge (third parties, the "crowd"), such as individual entrepreneurs, students, experts and small firms, who are asked to submit interesting solutions for a particular contest challenge, that satisfy certain criteria within a defined timeframe [3]. Members of the crowd do not see, nor have rights to use the proposed solutions: the outputs are closed and owned by the sponsor [8], who offers the prize to contributors of the best solutions in return for the right to use and exploit them.

An innovation contest is defined as a/an (IT-based) competition of innovators who use their skills, experience, and creativity to provide a solution for a particular contest challenge defined by an organizer [9]. It is the invitation of a private or public organizer to a general public or a targeted group to submit contributions to a certain topic within a timeline, which are reviewed, selected and rewarded by established committees [10]. According to Boudreau and Lakhani (2013), innovation contests are the right choice when it is not obvious what combination of skills or even which technical approach will lead to the best solution for a problem. They are most effective when the problem is complex or novel and when it comes to design problems, where creativity is crucial [11].

2.1. Structure of the Idea Lab Platform

Existing open innovation platforms comprise crowdsourcing contests on corporate websites, initiated by companies themselves, platforms intermediaries, run by organizations that connect companies and solvers through innovation contests, as well as companies' (open) innovation management softwares. The iDEA lab platform represents an intermediary platform, in the form of the third party web-based innovation marketplace, which acts as a knowledge broker [12], allowing companies to access unbiased knowledge, and to gain insights into opportunities that lie beyond the companies' immediate field of view [13].

Following the analyses of twelve intermediary platforms (Innocentive, Idea Connection, Jovoto, Hypios, Ideaken, Innovation Exchange, Idea Bounty, Eyeka, Top Coder, Edison Nation, Atizo, Kaggle) and their basic aspects (type, interface, challenges, disciplines, sectors, research library, community and news), as well as the literature review on this topic, the structure of the iDEA lab platform has been developed (Table 1). The contest organizers on this platform are made up by an industry partner and an academic institution, targeting students as primary contributors.

Table 1. Basic aspects of the intermediary platforms

Innovation contest platforms	Type		Inter-faces		Challenges			Disciplines				Sectors		Research library			Community			News		
	Contest organizers	Contest agents	Companies	Solvers	Idea / theoretical proposal	Concept solutions	Sourcing technologies	Research	Any	Data science	Digital products & services	Design	Corporations	Government & Non-profits	Articles & White-papers	Case studies & Best practices	Webinars	Open access to profiles	Open access to solutions	Forum	Blog	News/letter
Innocentive	x		x	x	x	x			x				x	x	x	x	x	x		x	x	x
Idea Connection		x	x	x	x	x	x	x					x	x	x	x		x		x	x	x
Jovoto	x		x	x		x						x	x	x				x	x	x	x	x
Hypios	x		x	x	x	x		x	x				x		x						x	
Ideaken	x		x	x	x	x			x				x					x		x	x	
Innovation Exchange	x		x	x	x				x				x	x				x		x		
Idea Bounty	x		x	x	x				x				x	x							x	x
Eyeka	x		x	x	x	x			x			x	x	x	x	x	x	x	x	x	x	x
Top Coder	x		x	x		x					x		x	x	x	x	x	x		x	x	x
Edison Nation	x		x	x	x	x			x				x	x	x	x		x		x	x	x
Atizo	x		x	x	x				x				x	x				x	x	x	x	x
Kaggle	x		x	x		x		x		x			x	x	x			x		x	x	x

Having in mind the classification for innovation platforms according to two key attributes – platform operator and platform purpose, iDEA lab platform can be described as a problem-oriented third party operator, which is focused

on finding solutions for specific challenges and awarding the best proposal [14]. Additionally, contests that take place on this platform can be described according to the list of ten key design elements for innovation contests [9].

Concerning the media choice, these innovation contests are run online, with the possibility to become mixed in later stages when the valuable solutions are being improved through personal contacts among participants (students) and organizers (university professors and teaching assistants, with the support of partner company's representatives). The tasks are highly specific and call for elaborated concept solutions, within the time frame of six months (during summer or winter semesters). Motivation to contribute one's competences to an innovation contest is fostered by a reward system that is adapted, as much as possible, to the needs of the target group. The rewards are realized by monetary prize and social motivation, like positive feedback, reputation among relevant peers and self-realization. Community functionality is provided through the elements which foster interaction, like information exchange and topic related discussion on the platform or face-to-face during lessons or consultations. Evaluation of the submitted solutions is realized by the jury, consisted of the companies' representatives, as well as university professors, teaching assistants and researchers in the field.

In order to increase the efficiency of innovation contests, iDEA lab platform supports multi-round contests, in which the first round is played with a large pool of contestants who make relatively little investment in the idea generation phase [15]. In the second round, after the submitted ideas are evaluated and the promising ones are selected, the limited pool of solvers work on the idea development and refinement in closer contacts with the contest organizers, focusing on the concept development. In each innovation contest the participants, i.e. companies or university laboratories, research centers, institutes as solution seekers, university as an intermediary, and students as problem solvers, all have their defined roles. Therefore, the iDEA lab platform comprises these three crucial segments (Figure 1) within the university environment that allow solution seekers and problem solvers to fulfil their own assignments.

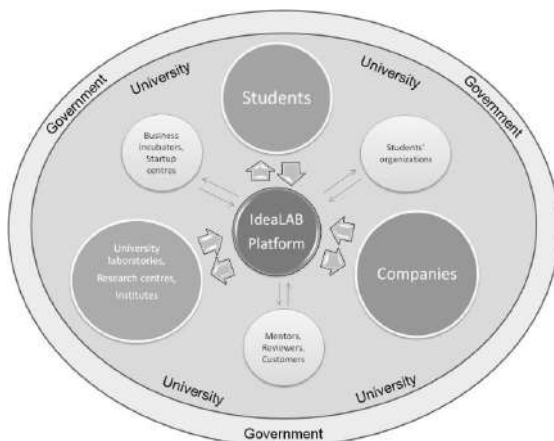


Fig.1. iDEA lab platform structure

As solution seekers, companies should propose a specific task to be solved and lend their support to an interesting idea/product/project, and university laboratories, research centers, institutes, are supposed to invite students to join for a specific research project and to give an infrastructural & expert support for an interesting

idea/product/project. Solution seekers should also provide a financial compensation for solvers in return for solutions in order to retain the IP rights to them [1]. The role of students is to solve tasks in a specific challenge/contest, to give their opinion/vote on a specific challenge, to propose a product/service idea, to propose a business model or some activity/idea.

The incentive structure for participants needs to be simultaneously attractive for the participants and appropriate for the company [6]. Since students invest considerable time and effort, a mixture of monetary (cash, scholarship, etc.) and non-monetary prizes (student job, internship, professional certificates, etc.) should be a successful stimulation for both intrinsically and extrinsically motivated students. Concerning the scope of innovation contests, there are two major strategic areas of application - greater good and solutions for corporate challenges [17]. Apart from finding the solutions for a company's problems, the development of students' skills and competences is another main objective of these contests in student innovation contests and also represents a strong incentive for students. On the other hand, HEIs are also stimulated to tap into a talent pool of their student groups, and offer them research internships to foster ongoing research on these institutions.

3. STUDENTS INNOVATIVE SOLUTIONS IN COWORKING WITH SMEs

The proposed platform structure is tested in real environment to gain necessary insight into functionalities, usability, etc. Several projects were launched simultaneously. All projects had a defined task with supporting documentation. The desired outcome for the projects was a functional prototype of a given subassembly on an existing product.

The product used for the contest was the "Croozier Kid for 1 Bicycle Trailer", which is a product of the CROOZER GmbH (Figure 2).

The first defined project dealt with modifications on the »Baggy wheel« connection to the trailer (Figure 2-1), while the second project dealt with modifications on the »Handle bar« connection to the trailer (Figure 2-2). The third defined project dealt with modifications on the »Wheel hub« (Figure 2-3), while the fourth project dealt with modifications on the »Hitch arm« design (Figure 2-4).



Fig.2. Croozier Kid for 1 Bicycle Trailer

The functional requirements were defined in detail for all of the projects by the manufacturer. The projects were

accessible for third-year students of mechanical engineering at Subotica Tech – College of Applied Sciences in Subotica, Serbia and at Faculty of Technical Sciences in Novi Sad, Serbia.

The remainder of this chapter gives a more detailed insight to the »Wheel hub« project. Some of the most important requirements were as follows:

- Possible minimum hub width under consideration of the loading force per wheel of 1.500 N
- Integration of only two bearings within the hub.
- Attaching the brake disc to the hub, not to the spokes
- Designing of the axle system with current OD, able to carry the load of 1.500 N
- Keeping the push button release system.
- Easy to assemble: Hub, bearings, brake disk, axle, spokes
- Rubber cap can't ripped off
- Reduce weight by 25%

3.1 Analyses and the process of ideation

During the testing, students were able to choose the task they were interested in solving and had the access to all information needed, such as (functional requirements, CAD drawings, other available analyses and calculations). Of course, they were able to communicate via the system messenger with the task administrator if they had any additional questions.



Fig.3. Student's group work on Wheel Hub task

3.2 Student's contest through the platform

The projects were set to be performed in three independent phases. In the first phase all registered students who wanted to participate in the project uploaded their ideas as 3D PDF models with comments. These were initial ideas that have not been designed in detail. All uploads were analyzed and graded independently by a group of reviewers and the representative of the manufacturer.

The analysis was performed based on the agreed criteria between the reviewers and the representatives of the manufacturer. It was important to meet the main requirement/function of the task in the students' proposals. So, reviewers had to check whether the "basic criteria" were met and provide comment if there was some possible problem or whether the basic criteria were not provided.

The first phase was not eliminatory and all students had a chance to upload the improved solutions for the second phase, according to the received suggestions.

The second-phase uploads were made also as 3D PDF models in the assembled and/or disassembled structure with comments. These improved uploads were made in more detail and with more specific suggestions regarding the functionality.

The example of a students' ideas uploaded in the Phase Two are presented in Figure 5 and Figure 6.

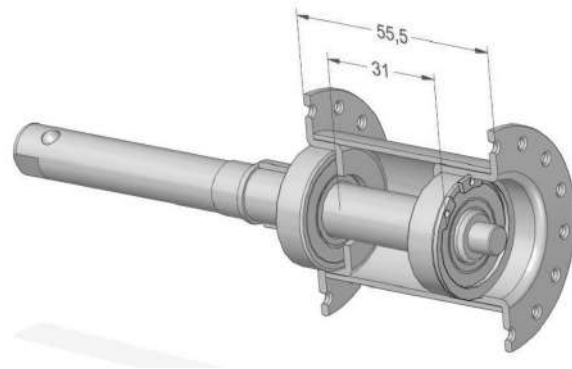


Fig.4. Example of an idea uploaded in Phase Two

All uploads in the Phase Two were also analyzed and graded independently by a group of reviewers and the representative of the manufacturer. The analysis started with checking for the fulfillment of all basic criteria from the first phase. They all had to be met, otherwise the proposal was eliminated. The criteria for the second phase were the so-called "quality criteria" for different functional requirements and the uploaded solutions could be graded on a scale from 1 to 5 (5 being the highest number of points).

Figure 5 is an explanatory presentation of the parameters and the user interface for the "Reviewer" for the analysis procedure in Phase Two for the project "Wheel Hub" connection to the trailer.



Fig.5. Part of the user interface for the "Reviewer" for Phase Two

After Phase Two, all participants were invited to improve the best ideas from the Phase Two. The best ideas were opened up and made accessible to the whole group of participants, so that everybody could contribute to the best chosen solutions. The third phase of the project included further improvements of ideas, with 3D sketches

in greater detail, taking into account the real measurements of the product and possibly, the used materials and technologies. Moreover, all functional requirements had to be taken into consideration. The idea was to provide refinement of the solutions by all community members. It meant that somebody could upload a solution with a much better design and more optimized than the initial proposal that was selected in the second phase.

4. 3D PRINTING THE BEST MODELS

After finishing the student's contest, three best solutions were printed by students in iDEALab laboratory and compared with the original wheel hub (Figure 6).

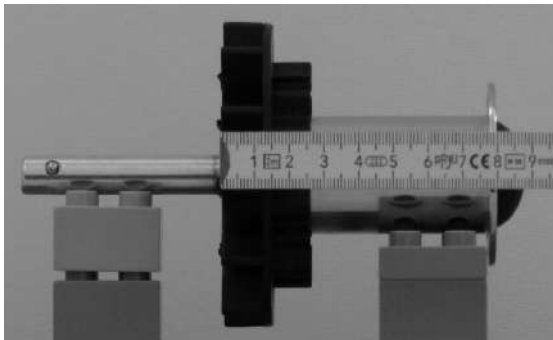


Fig.6. The original wheel hub

Functional prototype #1

Total length of the Prototype #1 (Figure 7) with integrated brake: ~62mm. Distance between bearings: unchanged. Spokes' geometry: changed slightly due to distance reduction by 18 mm. Static and dynamic stress: changed slightly due to changed position of spokes. All new parts except the brake are supposed to be metal parts.

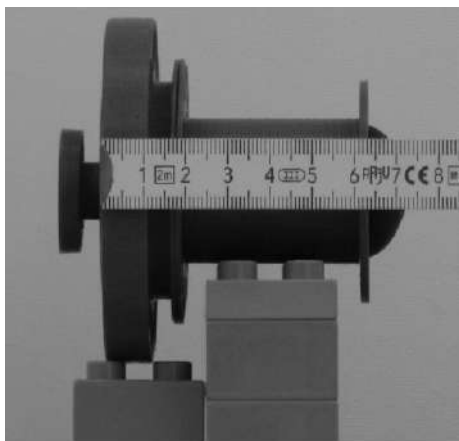


Fig.7. Wheel hub prototype #1

Functional prototype #2

Total length of the Prototype #2 (Figure 8) with integrated brake: ~62 mm. Distance between bearings: unchanged. Spokes' geometry: changed slightly due to height change on the left side. Static and dynamic stress: practically unchanged. All new parts are supposed to be metal parts.

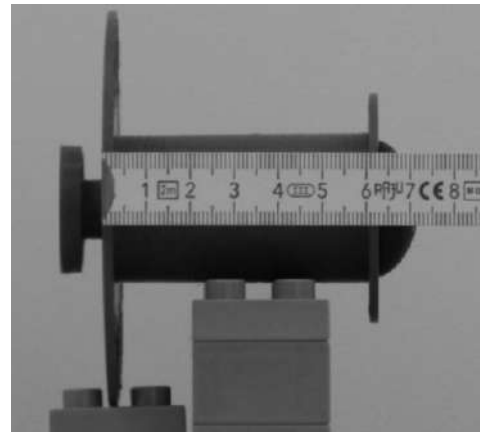


Fig.8. Wheel hub prototype #2

Functional prototype #3

Total length of the Prototype #3 (Figure 9) with integrated brake: ~62 mm. Distance between bearings: unchanged. Spokes' geometry: changed slightly due to height change. Static and dynamic stress: practically unchanged. All new parts except the central part are supposed to be metal parts. Central part is supposed to be produced from plastic material with metal implants to support the bearings (this is not presented).

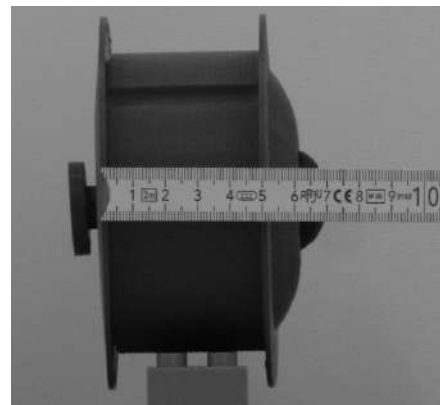


Fig.9. Wheel hub prototype #3

At the end, all three 3D printed models were compared with the original wheel hub (Figure 10). Total length of wheel hub with integrated brake is shortened by ~18 mm in all suggested solutions. This is due the fact that the distance between the bearings is unchanged, and because the shaft mounting is also unchanged. Spokes' geometry is changed only slightly. Static and dynamic stress practically remains unchanged.

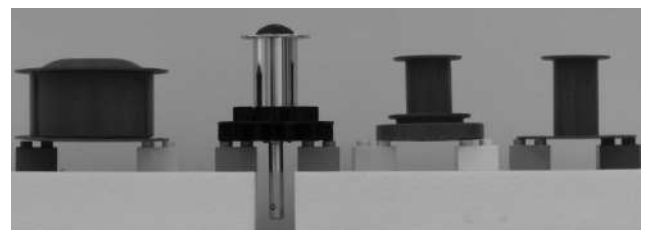


Fig.10. The comparison of three improved wheel hubs with the original wheel hub

5. CONCLUSION

In comparison with internal product development, co-working provides companies with the opportunity to increase the number of sources of new solutions, by embracing the competences and intelligence that are not present inside their borders. Companies should not necessarily engage their own resources to find the right employees, to motivate and monitor their effort. They can benefit from involving a large number of innovators in their product development processes. Through a student innovation co-working activity, the company Croozer GmbH created a pool of potential solutions.

Since students are identified as a powerful source of innovative solutions and it is shown that there is a great potential for the cooperation between universities and industry in this sector, there should be a specialized platform to support this kind of collaboration, such as the iDEA lab platform. There are some crucial issues that have to be taken into account for the further development of this platform. This platform has to be simple and intuitive for all parties involved in innovation contests. The motivation mechanisms (incentives) have to be well-developed for each challenge to attract students, who should be educated to become open innovation ambassadors through their experience gained in these innovation contests. There already are group problem solving and entrepreneurial idea development tasks that are included in existing courses at universities, as well as different contests organized for student population. In order to develop the iDEA lab further, these contests should be connected to this platform. However, there is lack of the experience for a specific challenge deadlines, reviewing criteria and respond/reaction of the "crowd" (students). These are some open questions that still need answering, in addition to the issue of intellectual property, concerning the rights among different partners, as well as the sustainability issue of the iDEA lab platform.

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TRANSFORMING PRODUCT- CONSUMER COMMUNICATION THROUGH AUGMENTED REALITY TECHNOLOGY

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Abstract: *For communication on the product-consumer relation to be successful, it has to use an appropriate channels. This applies to all types of communication, including advertising, promotional material, information on a website in any of the formats electronic, print, braille, etc. In order to cater to the needs and preferences of all customers' integration of all the formats can be beneficial. Augmented reality is not new technology, but just recently it has become able to provide integration of mentioned formats of communication. Wide availability of the internet access through mobile devices has made possible to use this technology in the place of sale, consumer homes or almost anywhere. This paper aims to present possibilities for integration of advertising, packaging, user guides or any other form of product-consumer communication by usage of augmented reality technology. This can lead to far better user experience and provide advantage for the early adaptor organizations implementing this approach to the communication with the consumers.*

Key words: *packaging, communication, augmented reality*

1. INTRODUCTION

Today every successful business is becoming a digital business, in sense that design, production, distribution, marketing are penetrated by the digital technologies. Having in mind that majority of the consumers in modern countries are becoming digital consumers communication channels are changing. Social networking and mobility are used to transform purchasing experience and product usage. In response to the rise of the digital era and grooving number of digital consumers organizations

providing products must look to digital technologies as leverage to ensure consumer engagement and competitive advantage in the market. Appropriate channels must be used for product – consumer communication in order to convey the certain information about the product. All types of communication, including advertising, promotional material, user manuals, maintenance instructions, information on a website, etc. in any of the formats electronic or print are conceived in the way that insure comprehensible and easily absorbable information flow. Often these formats are presented at the different time and place. In order to cater to the needs and preferences of all customers' integration of all the formats is needed. With so many different possible consumer points to contact it's more important than ever to create a unified experience across all channels. In the modern age of mobile technology domination it is possible to make unified Omni-channel information dissemination strategy from a customer's first information about the product (website, TV or magazine advertisement, etc.), place of sale where the purchase decision is made, to the user manuals, maintenance and disposal instructions.

According to a Kitewheel report, for United States, brands are failing to meet consumer expectations to deliver an experience that is seamless across both digital and traditional media. 53% of brands report that their main problem is an inability to offer a seamless consumer experience and achieve cross channel synergy. Brands must bridge this channel gap, as 54% of consumers expect seamless experience across digital and physical media [1]. As mobile technologies and widely accessible internet connection continue to reshape product purchasing and usage processes the organizations must find creative ways to seamlessly integrate physical and digital media. This is critical for offering pleasant user experience for today's consumers who alternate frequently or simultaneously use digital and physical media. Seamless interactions is crucial for utilization of digital channels in order to enhance consumer experience. The Point-of-Purchase Advertising Institute's research shows that 70% of buying decisions are made in the store [2]. According to Deloitte, about half of all in-store retail sales were influenced by digital media in 2014, and digital media is projected to influence 64 percent of in-store retail sales by the end of 2015 [3]. This is not prediction of packaging obsolescence in the point of purchase communication and its impotence in the purchasing decision making processes but rather prediction of packaging evolution to the next level.

Using augmented reality technologies can be utilized to achieve the goals of ensuring maximum consumer emersion by combining material and digital reality. Augmented reality can serve as an extension of the conventional product-consumer communication. This is achieved by including real-time modification of perceived context on the basis of interactions among the user, the environment and focal objects using augmented reality technologies. Augmented reality differs from virtual reality in that there is no attempt to completely replace the real world, the users are still able to interact with the real world, and at the same time perceive enhanced views with augmentations imposed on top of the real world image [4].

2. AUGMENTED REALITY TECNOLOGY

Augmented reality is novel human–computer interaction technology that overlays computer-generated virtual elements onto real-world environment images. The information display and image overlay are context-sensitive, which means that they depend on the observed objects thereby enhancing a user’s perception of reality by combining physical and digital content. This technology is widely available through mobile devices. [4, 5, 6]. Basic classification of technologies utilized can be done according to tracking technique, display technology and interaction techniques, table 1.

Table 1. Classification of AR technologies

Tracking techniques	Sensor-based techniques
	Vision-based techniques
	Hybrid techniques
Display technology	See-through displays
	Projection-based displays
	Handheld displays
Interaction techniques	Tangible AR interface
	Collaborative AR interface

Especially interesting for application is Vision-based tracking technique which can be further divided into Marker tracking and Marker-less (Non-marker) technique. Marker tracking technique where the corresponding image descriptors are provided beforehand and stored into the database. Most of the applications are using marker-based technique. Marker-less (Non-marker) tracking technique where application recognizes images that were not provided to the application beforehand. This is much more difficult to implement because the recognition algorithm running in your AR application should identify patterns, colors or some other "features" that may exist in camera frames [7]. There are also Sensor-based tracking and hybrid techniques.

2.1. Augmented reality hardware

Availability of the hardware is one of the driving forces behind explosion of augmented reality applications and move from the laboratories to the everyday life of the people. Minimal requirements are image input device, processing machine and display, figure1. Simplest setups consist of web camera PC computer and monitor. In order to ensure mobility and full usability of the technology modern mobile device such as phone or tablet with back facing camera and appropriate application installed. Usability can be expanded even further with wearable technology such as Google glass or similar products. Such wearable technology frees users’ hands and enables him to perform manual tasks while he is given additional information or instructions.

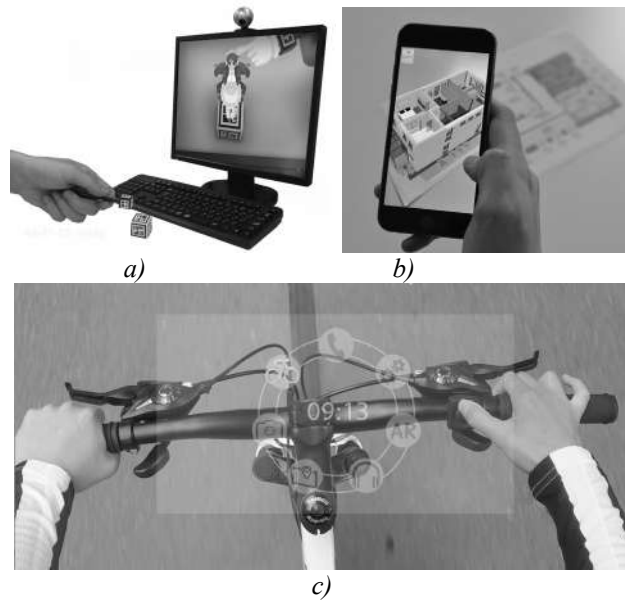


Fig.1. AR hardware a) Personal computer, b) Phone, c) Wearable hardware

2.2. Augmentation process

The augmentation is produced after a series of transformations, as shown in the process for the creation of AR using marker-based tracking, figure 2. First, the real video image is transformed. Then, this image is processed in order to determine position of the markers (containing an image pattern that is compared to patterns stored in a database). Next, the algorithm determines pattern orientation as the base for the coordinates frame and calculates the real position of the digital camera in relation to the physical marker. After that, the 3D objects are placed over the markers, and the final image is rendered and sent to the display [8].

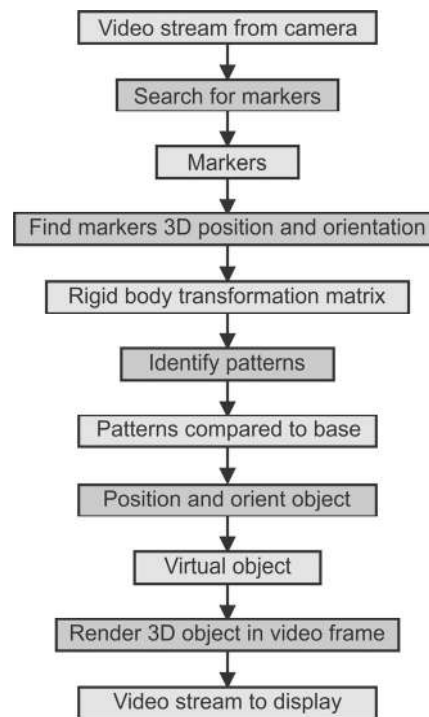


Fig.2. The process for the creation of AR using marker-based tracking

3. AUGMENTED REALITY APPLICATIONS

Although augmented reality is still in its infancy it must be taken in to consideration as a useful tool. Accenture Augmented Reality Survey finds 59 percent of participants have never had an opportunity to see augmented reality in a live situation, 30 percent is not sure how augmented reality works and 23 percent does not consider augmented reality particularly useful [9]. The same survey rivals that augmented reality would increase your likelihood of purchasing the product in case of use for:

- Virtual furniture app: 86%
- Virtual dressing room: 88%
- Color changing app: 73%
- Catalog app: 65%
- Shopping app: 75%
- Shoe sampler app: 87%
- Virtual vehicle manual: 75%
- Gamification app: 61%

Two years later, in 2016, we are witnesses of rise in use of such augmented reality capabilities, shown in figure 3. IKEA Catalog enables users to place selected furniture in their own room, gamification of content used by McDonalds, shopping assistance by Fitnect application and promotional usage by adding 3D, sound, video and other content to the packaging (or rather triggered by packaging image) are some of the applications of augmented reality technology for product-consumer communication enhancement.

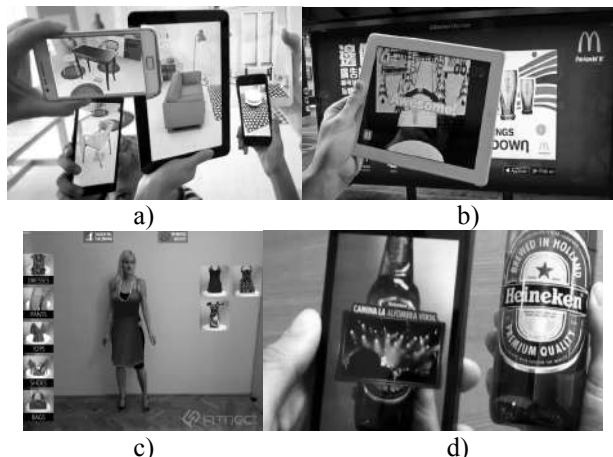


Fig.3. Examples of augmented reality applications:

- a) IKEA – virtual furniture,
- b) McDonalds – gamification of promotional panel,
- c) Fitnect – shopping assistance,
- d) Heineken – addition of video content to packaging.

3.1. Product promotion

As shown in figure 3 augmented reality can be powerful tool for product promotion. Adding additional content to packaging thus occupying consumer attention and developing relation between product and consumer can be beneficial at the point of purchase.

Additional content can be in the form of useful information presented as text, sound, images, video, 3D object or 3D animation, it can be even interactive introducing gamification elements.

3.2. Consumer education and training

Virtual reality has a long history in education of military, medical, industrial training, etc. in developed counties. Augmented reality is offering to overcome limitations of virtual reality opening up a whole new field of applications and enhance the effectiveness and attractiveness of training and learning, figure 4. Proper demonstration is key for success in hands-on courses. Now instructions can be given in real time while performing the task. Using augmented reality technology can reduce the risks when using expensive equipment which is often inaccessible for training and thus reduce or avoid training costs if mistakes are made during skills training. Advanced applications can provide authentic learning experience and 'active' training through immersive simulations and engage consumers in ways that were never possible before.

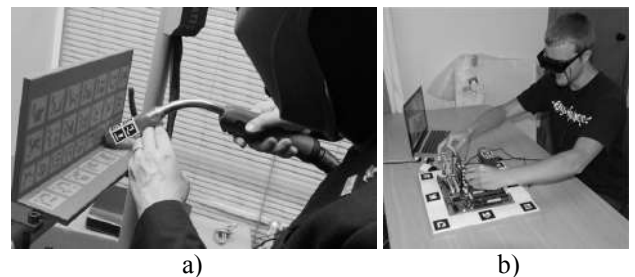


Fig.4. Examples of augmented reality applications for training, a) Welding training, b) Motherboard assembly training

3.3. Product maintenance

Development of applications especially in the area of product maintenance, especially for industrial systems, is beneficial and can ensure following of proper procedures in disassembling and assembling operations, thus avoiding additional. Providing information in real-time while operation is done is crucial. Placing markers on parts of equipment can help identification and ensure correct order of operations. By using wearable electronics hands of operator are free as mentioned before. For instance, BMW have developed a concept for augmented reality glasses, which assist mechanics in performing maintenance on the cars. The glasses point out the part that needs replacing, the screws that need turning, while an audio track talks the mechanic through the steps of the repair, as shown in figure 5.



Fig.5. BMW augmented reality maintenance concept

Recognition of equipment in factory setting and matching it to the digital map makes paper layout plans obsolete through a now technological feasibility to mix augmented reality with geolocation, recognition and tracking technologies.

4. TECHNICAL CHALLENGES IN AUGMENTED REALITY

Unlike applications for product promotion applications intended for training and maintenance require a higher level of accuracy in tracking. Main problems are high frequency motion as well as rapid camera movement. Further development of computer vision, inertial and hybrid tracking techniques, alongside using laser, RFID and other types of sensing devices will be required for augmented reality to achieve full functionality in all segments of product-consumer communication. Registration or placing of virtual objects in the correct position is a difficult as different tracking methodologies have different error sources, due to latency problems between data streams, synchronization and computational delays, misalignments between sensors, and/or incorrect registration algorithms [7].

5. CONCLUSION

The novelty of the augmented reality, instant information and enhanced perception will change the way communication between products and consumers. Adoption of the technology is in the hands of people and companies, but implementation possibilities are vast. Having in mind limitation of the hardware it is necessary to study the best tracking method for a particular application in order to ensure the best results and user experience.

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A NEW CONCEPT OF BICYCLE FRAME DESIGN

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Abstract: For the research purposes two of bicycle frame concepts have been developed. The biggest attention is given to the aesthetic and ergonomic aspects. After developing the concept of bicycle frames, their CAD models are created and FE analysis is performed. The distribution of loads is examined in critical frame areas, as well as deformation occurred under the influence of the real working load. The discussion of the results is performed and recommendations are given how the proposed conceptual solutions can become practically feasible. At the end of the paper are given the directions for further research on this topic.

Key words: bicycle, bicycle frame, design, FE analyses

1. INTRODUCTION

Developing of new products, for widespread usage, is tight related to the industrial design principles [1]. Following of this principles greatly increases the possibilities of the product success on the market. After the first working concept of bicycle, which is designed by Kirkpatrick Macmillan, 1839, there is no a second and third. Ordinal numbers give way to the improved details, new frame designs and improved ergonomics, but original concept stays the same.

The innovative ideas of the manufacturers and construction designers to minimize aerodynamic drag, to improve comfort, minimizing the mass of the frame, maximizing lateral stiffness in the load transfer from the hands and feet to the drive, maximizing the strength capabilities of the frame to allow for a higher load capacity or better load distribution, and adjusting the vertical compliance of the frame to tune the softness of the ride [2, 3] that means comfort and safety ride. These are the reason of contribution in the development occurs in the design of bicycle frame and consequently the need of analysis of bicycle structure come to rise. The frame of

the bicycle is the main structure to support the external loads. Traditional materials of the bicycle frame are the steel or aluminum alloy. With help of theoretical or numerical calculations, the strength and stiffness of the bicycle structures can be predicted and modified to the optimal design before the manufacture of the prototype and actual commercial products.

A promising solution is provided by a tool of structural engineering; the Finite Element analysis. Performing Finite Element Analysis (FE analysis) on bicycle frames has become a common activity for bicycle designers and engineers in the hope of improving the performance of frames. This is typically achieved by balancing priorities for key requirements, including minimizing the mass of the frame (possibly using competition rules to constrain this), maximizing lateral stiffness in the load transfer from the hands and feet to the drive, maximizing the strength capabilities of the frame to allow for a higher load capacity or better load distribution, and adjusting the vertical compliance of the frame to tune the softness of the ride. Depending on the type of physical problem being analyzed, the field variables may include physical displacement, temperature, heat flux, and fluid velocity to name only a few [4]. A lot of work is done on the analysis of bicycle frame; some of that work is presented here. FE analysis has been used to analyze composite, aluminum and steel bicycle frames [5–8] with the aim of understanding physical behavior and improving performance relating, however a comprehensive study on the influence of key geometric parameters on the stiffness of frames has not been conducted. Soden and Millar [9] used “beam” elements to evaluate stresses of a steel bicycle frame during common cycling situations and compared the FE results with the experimental ones based on strain-gauges measurements. Hull and Bourlochi [10] proposed a new design method based on a FE analysis evaluation of bicycle frame stresses at several crank angles. Davis and Hull [11] are analyzed the design of aluminum frame bike to fatigue failure. Numerous studies in this field have been carried out in terms of the load and the ergonomic aspects of the analysis on the existing bike frames. There are a smaller number of researches done on the topic of improving the design of the bicycle frame.

Those activities proposed FE analysis as an instrument to improve frame performances in term of strength, high stiffness and low weight by means of a better knowledge of stresses. The designer, using FE analysis, knows which part of the frame is more stressed and could modify the thickness of the tube or its mean diameter.

The authors of the paper used the methodology of industrial design with a concept that includes the analysis of objective circumstances to design two bicycle frames. From the aspect of industrial design, special attention was paid to the aesthetic appearance and ergonomic properties of the bicycle frame. In order to comprehend the real conditions in which the product will be used, creating a 3D models and technical documentation is conducted in the software package Autodesk Inventor Professional 2014®. The aim is to create the shape of the product in accordance with the requirements of technology features, ergonomics and markets, using computer programs which results provide visual and other information necessary for making real product scheduled for industrial production.

To meet the requirement for the design of a bicycle frame is necessary to consider the influence of other components of the bicycle in different exploitation conditions.

2. CONCEPTUAL DESIGN CHOISES

Based on preliminary review of available literature sources and existing solutions bicycle frame design, it is accessed the innovative development of two bicycle frame concepts (Fig. 1).

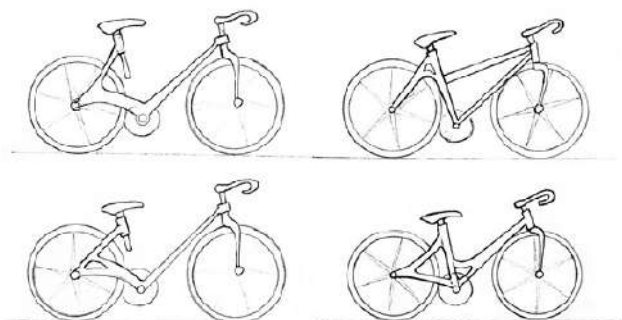


Fig.1. Conceptual design of the bicycle frame

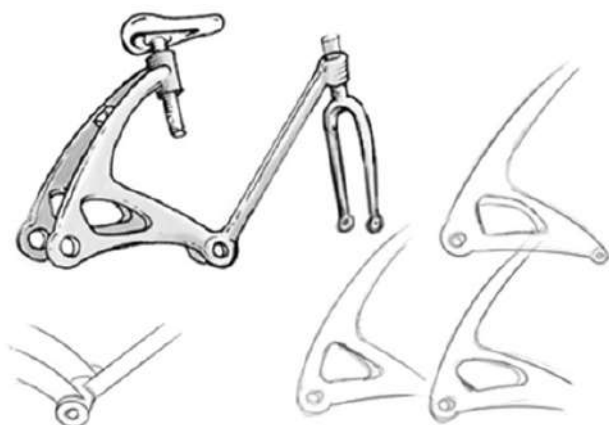


Fig.2. Concretization of the new bicycle frame solution

The final setup of the bicycle frame includes a set of pipes and plates. Frame can be observed as two parts, front and rear. The rear part of the frame is composed of circular pipes $\varnothing 28$ mm, with a wall thickness of 2 mm and two curved plates with thickness of 7 mm. The plates are curved in a form of the letter "A". The front part consists of a circular pipe $\varnothing 34$ mm, curved pipes elliptical shape and rectangular full profile pipe 7 mm and 20 mm. The front and rear sections are connected through a circular pipe $\varnothing 38$ mm, with a wall thickness of 9 mm. The frame is made of aluminum. Piping parts and panels are welded. Overall dimensions of the frame are 926 mm and 572 mm.

3. FE ANALYSIS OF BICYCLE FRAME

Basic requirements to the bicycle frame must fulfill is holding other components of bicycle at certain distances. The frame is a stationary part of the bicycle. Deformation of this part may lead to damage of the whole bicycle, so it is recommended detailed stress analysis.

3.1. Analysis of first bicycle frame concept

Finite element method has been used for testing a new model of bicycle frame (Fig. 3.), and preliminary analysis results are obtained. Analysis of stress and strain state is conducted in Autodesk Inventor 2014[®]. Discretization of bicycle frame is performed by three-dimensional finite element mesh, which is in the form of parabolic tetrahedron.



Fig.3. First bicycle frame CAD model

Constraints are placed in two places: at the rear axle and on the point where the steering pipe relies on a fork. The constraints are shown in Fig. 3. Bicycle frame is loaded with the forces in the value of 2000 N, places the seat and steering wheel (Fig. 4.)



Fig.4. First frame constraints



Fig.5. First frame loads

3.2. Analysis of second bicycle frame concept

In the bicycling world, the weight of the bike is very important characteristic. The only part of the bicycle which could support changes of weight is bicycle frame. The new frame design that has been tested in the previous section did not show satisfactory results, and gain access to redesign solutions. Changes were made at the last, and the front of the shoulder. Plates that are planned as bearing parts of the rear of the facilitation. They are also the reduced thickness. The thickness of the board now is 6 mm. In front of the frame, the reducing volume of pipe 4 mm reduces weight. In Figure 6, you can see the model created in the aforementioned gives software package.



Fig.6. Second bicycle frame CAD model

4. ANALYSIS RESULTS

Displacements, which occur due to the operation of the present loading of the frame design, are shown in Figure 7. The maximum deflection of the pipes in place a seat pipe and it is 2.314 mm. At the rear axle, there is no deformation.

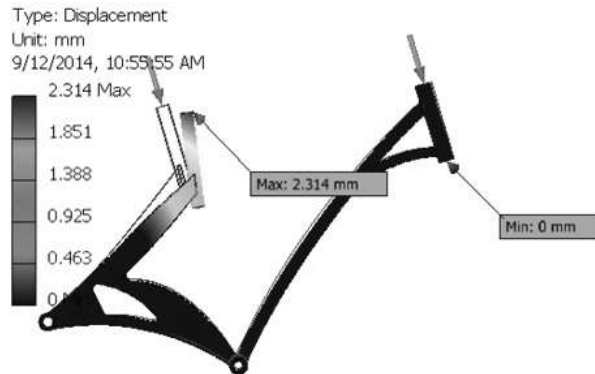


Fig.7. Displacement of first frame

Von Mises stresses are shown in Figure 8. The maximum stress is on the place where is the bicycle fork is 213.2 MPa.

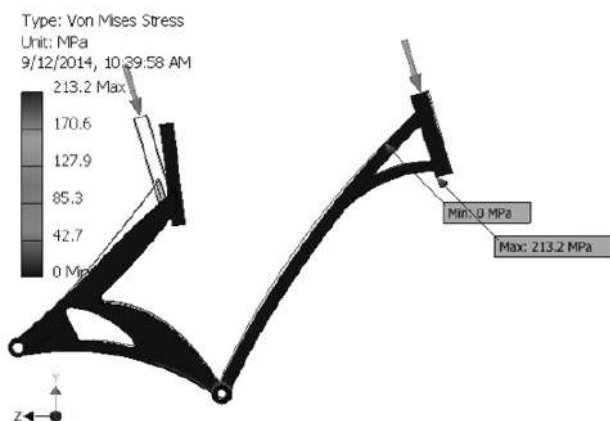


Fig.8. Von Mises stress of the first frame

Deformations on the loaded design of the frame are shown in Figure 9 and show the maximum deformation in place a seat pipe, and it is 3.318 mm. There is no deformation at the rear axle.

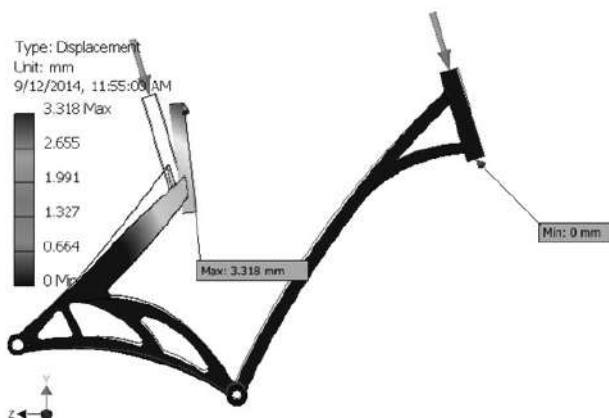


Fig.9. Displacement of the second frame

Von Mises stresses are shown in Figure 10. The maximum stress on the inside, at the place cuttings plate and it is 125.2 MPa

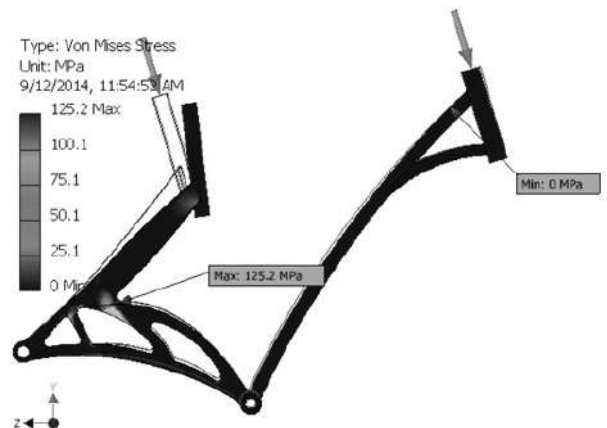


Fig.10. Von Mises stress of the second frame

Because of the stress concentration at radiuses, the mesh elements are smaller at those spots. Results that are more accurate can be obtained if mesh elements is changed to the brick elements. This operation wasn't an option because it hard to perform brick elements on the complex shapes like these bicycle frames.

5. DISCUSSION OF THE RESULTS

The obtained numerical results displacement and Von Mises stress led to the conclusion that a new conceptual solution of first bicycle frame will support the load that has been assigned. The new frame was required, and weight reduction, as this design has not been achieved. The second frame is heavier compared to existing frames by 0.29 kg. In order to reduce weight is recommended to facilitate their cup or plate thickness.

The weight of the first frame is larger than second bicycle frame. Second bicycle frame weighs 1.962 kg. By reducing, the weight decreased and the stress obtained at the initial results of the second solution. Critical stress is now located at the rear of the frame. The disadvantage of this solution is the deformation. Both solutions greatest deformation at the target seat pipe and second bicycle frame displacement 3.318 mm. The value of this displacement is small, less than 4 mm, or viewed in relation to a given initial solution that is eight times larger. To check whether this displacement is negligible recommended conducting additional analysis that would give relevant information. Table 1 presents a comparative overview of the results of first and second bicycle frame with a given displacement and Von Mises stresses.

Table 1. Comparison gained results of the first and second bicycle frame

	First frame	Second frame
Max displacement	2.314 mm	3.318 mm
Max stress	213.2 MPa	125.2 MPa
Mass	2.51 kg	1.962 kg

6. CONCLUSION

Based on the literature sources presented in this paper, we can say that the research topic is extremely relevant and attractive, since the creation of the bike to the present day. Many researchers have been working on the analysis of how the bike, an extremely widespread means of transport do even better.

For the purposes of this study were developed two concept bike frame. Both concepts are developed original solutions. In both concepts, the primary goal was to meet the aesthetic and ergonomic requirements. For both conceptual solutions are developed CAD model, to which reference is made FE analysis. In both models is adopted as the material aluminum. After the analysis of the resulting stresses and displacements. The stresses and displacements for selected material in larger limits than allowed. To make this conceptual solutions bicycle frame made, in practice it is necessary to change the types of materials or to the critical areas amplification device that would prevent displacement and reduce stress concentration.

This is an improvement, which is planned, in the further work using the methods of structural and topological optimization. In future research on this topic in the elaboration of the plan obtained by the methods of structural frames and topological optimization. Such an approach would in the future, combined with FE analysis, and gave excellent results.

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3D MODELLING OF CONSTRUCTION TOWER CRANE

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Abstract: A 3D model of a construction tower crane was developed using SolidWorks software. Selection of main elements, drive units and accompanying mechanisms was performed. Dimensions and parameters of the most critical elements of the design were determined based on previous calculations, partly presented in the paper. In development of the model, the loads influencing stability and carrying capacity of the tower crane were analysed according to standards.

Key words: tower crane, 3D model, SolidWorks

1. INTRODUCTION

Tower cranes are the most commonly used equipment on building construction sites. They play an essential role in material handling, placement, assembly and erection operations.

The trolley of the crane runs along the boom moving the lifted load and reaching a fixed distance from the tower. This distance depends on the counter balance weight and the weight of the load.

There are many design variations of tower cranes, depending upon the manufacturer and the intended use. The tower crane can be erected on a minimum ground area or within a building (for example, within the elevator shaft or other floor opening). To increase their range and versatility, some tower cranes are mounted on under-carriages running on rails, rather than on a fixed base. There also is a truck-mounted type [1].

2. CALCULATION

A tower crane basically consists of: mast, jib, counter-jib, counterweight, ballast and trolley [2].

Basic elements of the tower crane are shown in Fig. 1:

- 1 – base frame;
- 2 – bottom fixed tower;

- 3 – tower section;
- 4 – special climbing section;
- 5 – swing chassis;
- 6 – crane cabin;
- 7 – counter jib;
- 8 – counterweight;
- 9 – hoisting mechanism;
- 10 – counter jib tie;
- 11 – tower head;
- 12 – jib tie;
- 13 – jib;
- 14 – trolley;
- 15 – hook.

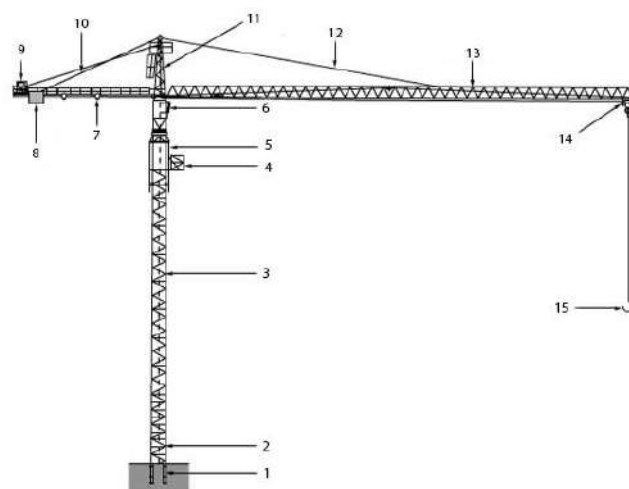


Fig.1. Overview of a tower crane elements

The concept of calculations is based on standards SRPS EN 14439 and SRPS EN 13001. Constant loads acting on a crane during operation (load weight and acceleration, inertial and gravitational forces during vertical load lifting and loads from vibrations of moving tower crane elements) are taken into account [3-5].

By introduction of dynamic factors in calculations, dynamic forces that arise during operation of drive elements of the tower crane are taken into account.

Wind is an important environmental factor for the stability of tower cranes. Strong winds impose extra loads on the tower crane and may affect the crane operation, the crane's rated capacity and the crane's stability.

The avoidance of the following risks was taken into account: risks of collision between the moving crane and fixed objects, risks of over flying forbidden or critical areas and risks of collision between several cranes in motion.

Initial construction (Potain MD 310 C K16) is traditional version of the tower crane with the stationary jib and tower head. There are many different designs of this construction which mostly depend on three possible types of the base frame (concrete, stationary and moving steel base frame). In the paper, a tower crane design with stationary steel base frame with code name ZD 463 and

height of 46 m was analyzed. Basic jib characteristics, such as length (minimal of 30 m, maximal of 70 m) and accompanying load curves are available in technical documentation of the crane. For the analyzed design, a counter-jig is used that has length of 18 m and is formed from the two separate elements. The tower crane load may be either 8 t or 16 t and primarily depends on selected pulley and hook assembly. Designed construction was created based on adopted catalogue values for ZD 463 type with jib length of 45 m.

3. 3D MODEL FORMING

Designed model of the tower crane was created using “SolidWorks 2014” software, Fig. 2. Considering the complexity of the model, presented figures were taken from “SolidWorks Viewport”, in order to more easily delimit the edges of certain component elements compared to classic rendered model [6].



Fig.2. Rendered model of the tower crane

Basic elements of the steel construction of the base frame are three separate H-profiles. In order to achieve greater stability of the base frame steel construction, H-profiles are laterally attached to additional boxed profiles. Design of the base with ballast blocks and legs is shown in Fig. 3. Two different types of concrete blocks were used as the ballast. The first type blocks (cuboid) form the base on which the second type blocks (trapezoidal) were placed. Two blocks of the first type and a total of twenty blocks of the second type were placed (five on each side). The grid structure of the mast consists of four separate vertical boxed profiles, additionally strengthened. The ladder that enable access to the cabin were also set, Fig. 4. The type of ladder access may be determined by available space in the tower. Landings between ladder sections, with changes in direction of the ladder, should be provided if there is available space in the tower. This

system will minimize operators fatigue by allowing for rest breaks and reduce the risk of injury, for example, in the event of workers falling off the ladder.

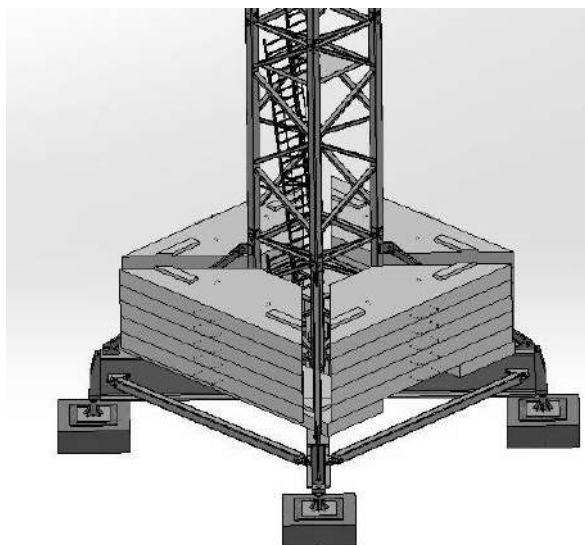


Fig.3. Mast and base

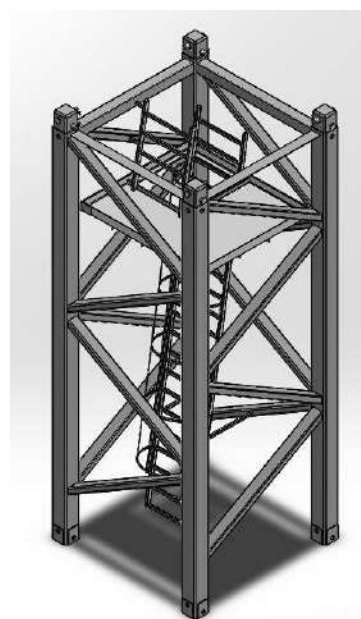


Fig.4. Grid structure of the mast

The risk of serious or fatal injury from crane collapse is very high during tower crane climbing operations. The climbing frame has to cope with significant static and dynamic forces involved in climbing. The cabin was placed on a corresponding platform. In order to achieve better visibility of the space on the front lower part of the cabin, a glass was set instead of the sheet metal. The location of the operator within the cabin of the crane may limit the view of hooking-on or unloading, requiring the presence of a well-trained signaller. Crane controls are designed and located so the operator can manage the crane efficiently. A slewing mechanism for turning the upper crane construction is shown in Fig. 5.

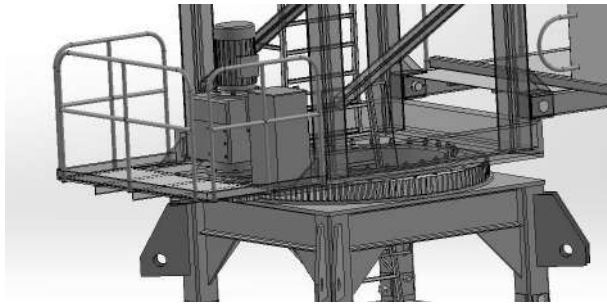


Fig.5. Slewing mechanism

Counter-jib of the tower crane consists from two individual sections. Leading elements are I-profiles strengthened by additional welded profiles, as shown in Fig. 6.



Fig.6. Counter-jig of the tower crane

Hoisting mechanism (engine, clutch, brake, gear unit and drum) are also set on the longer section of the counter-jig, Fig. 7.

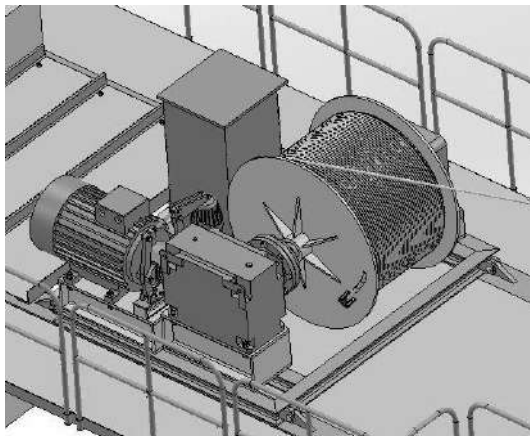


Fig.7. Load lifting mechanism

Tower crane counterweights are critical in ensuring crane stability. Counterweights should be secured to the crane in the manner specified by the crane manufacturer. A counterweight too light for a load and jib configuration may cause the crane to overturn in the direction of the suspended load. A counterweight too heavy for the load and jib configuration may cause the crane to fall over backwards. Access to the ballast blocks and other

elements was achieved by application of the side platform, as shown in Fig. 8. The tower crane jig also consists of a number of separate sections.

Swinging, hoisting, and trolleying motions are powered by electrical drive placed at a convenient location on the tower crane.

The most demanding assembly of the whole construction is the hoisting mechanism. The brake is placed at the output shaft of the electric motor. The gear unit, the clutch, the drive drum and adopted steel rope for load lifting were dimensioned based on the given load lifting speed.

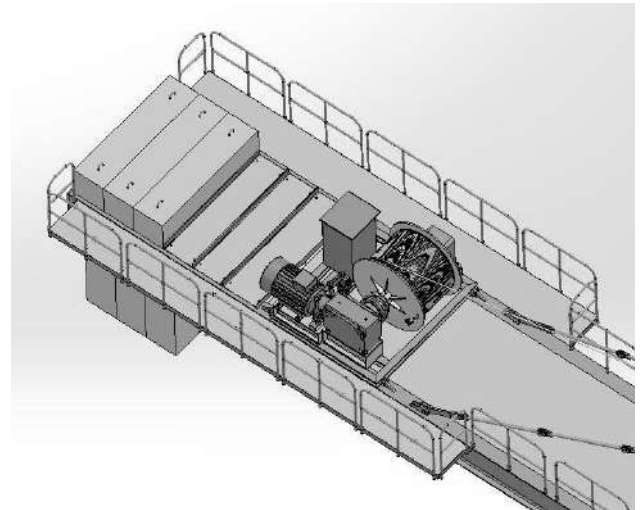


Fig.8. Section of the counter-jig with load lifting mechanism and ballast

The trolley shown in Fig. 9 consists of a chassis or metal base and wheels which move along the rails. The rails that belong to the jig are longer than those of the counter-jig. The trolley moves through a steel cable and a pair of sheaves, one of which is coupled to the drive motor and the other is coupled to the sensor (encoder) for measuring the distance travelled in one direction or another.



Fig.9. Rendered model of the jig and the trolley

The positions of the walkway, handrail and steel wire rope were defined. The trolley, Fig. 10, bears the rope sheaves, Fig. 11, which guide the rope to the pulley and hook assembly, Fig. 12.

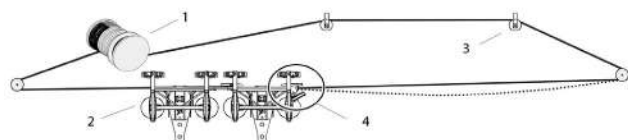


Fig.10. Movement of the trolley: (1) - drum, (2) – twin trolley, (3) – leading pulley, (4) – attachment of one side of the rope with trolley

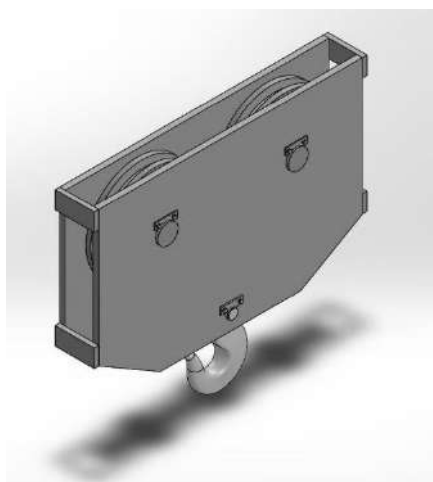


Fig.11. Pulley and hook assembly

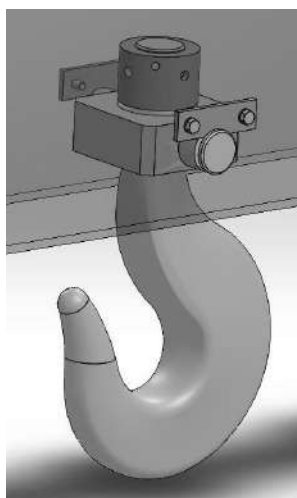


Fig.12. Hook, crossbeam and screw nut assembly

4. CONCLUSION

Creation of a 3D model of the tower crane was described in the paper, based on tower crane Potain MD310 C K16 type ZD 463 construction. The input parameters were taken from the accompanying technical documentation [7].

A real design of fast-mounting tower crane was analyzed. Basic elements were listed, but also some specific details of the analyzed construction were given. Position of the crane was defined according to the building site configuration. Special attention was given to the process of assembly, first of all to achieving the necessary tower height and jib position depending on height of the objects being built.

The most frequent causes for static and dynamic loads of the structure were presented. Influence of the wind, as a special case of dynamical loading was also considered.

The 3D model was created using “SolidWorks” software – the basic modules and “Weldments” module. Created model was exported to “KeyShot” 3D rendering and animation software, to be additionally processed in order to achieve the best photorealistic visualization.

By modelling of the tower crane, increased safety for construction sites, improved operator comfort, increased safety during climbing, and, finally, more flexibility for owners may be achieved. Tower crane users should consult manufacturers, suppliers and owners on operations and safety requirements and address any concerns before using the equipment.

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STRUCTURAL SYNTHESIS OF THE MANIPULATOR OF THE THERMOFORMING MACHINE

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Abstract: Items produced by thermoforming have significant applications in the packaging of food and pharmaceuticals so their production must follow strict hygiene requirements. In order to achieve full control of the thermoforming process, the working assemblies of thermoforming machines are becoming more mechanized. This paper presents the process of the structural synthesis of the fully mechanized thermoforming machine module which is used for finished product manipulation. Subassemblies of the manipulator and their characteristics are defined based on the function-means analysis. By combining subassemblies one can form all possible variants of the manipulator structure and, in this way, significantly ease the design process.

Key words: manipulator, thermoforming machine, design

1. INTRODUCTION

Thermoforming is the process of thermal treatment of plastic. The goal of the process is the manufacture of various three-dimensional plastic forms. Forms are used in two ways: as the final product or as the raw material in some other product manufacturing process [1].

Input forms for the thermoforming can be either plastic foil or plastic sheets. Thickness varies from 0.15 to 1.5 mm for foil and from 1 to 6 mm for sheets. Some often used materials are: polystyrene, polycarbonate, ABS, PET, PVC. These materials can be easily formed when heated and, when cooled, maintain that form. This property gives a products made from these materials a wide range of applications for the packaging, in the food and pharmaceutical industries [2].

Thermoforming is done on special machines through several operations. Each of these operations is realized on a separate module of the machine [3]. The concept development of the manipulator used for handling finished thermoformed products is presented in this paper. The manipulator is designed to perform as a module of

the thermoforming machines developed with previous research.

2. THERMOFORMING PROCESS

The thermoforming process is presented in Fig. 1. It has three main operations: foil manipulation, product forming and finished products manipulation.

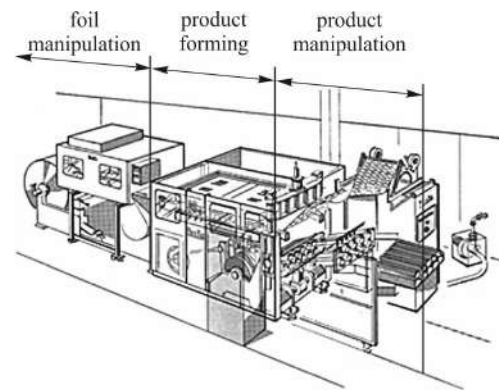


Fig.1. Thermoforming machine and its main operations

In the first operation, the raw material (plastic foil or sheet) is heated to the annealing temperature. In the second operation, products are formed in the working module by a special tool using pressure or vacuum technology. Typically, the tool has two halves –the upper and lower half (Fig. 2).

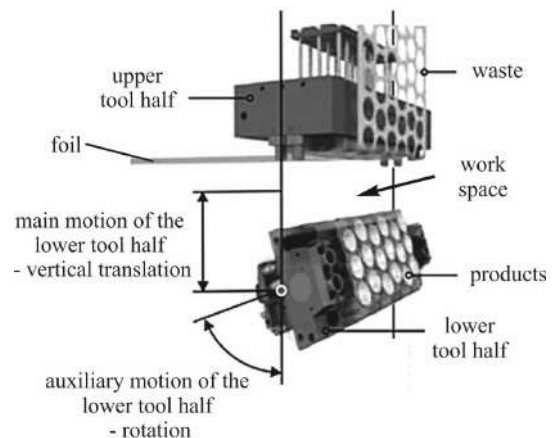


Fig.2. Thermoforming tool

The upper half remains stationary while the lower one moves according to the production process. When the lower half is in the lowest vertical position (it is not moving at that moment), a heated foil is positioned between the halves. Next, the lower half moves upward and presses the foil to the upper half. A period of tool inaction follows during which the products are formed. At the end, the lower half moves downwards and the products are extracted from the tool and moved away.

Two possible ways of product manipulation are possible: first, when the products are removed from the working module while still connected to the foil and separation is done on a new machine module; and the second one, when the products are separated from the foil on the working

module and the waste is transported from the machine. In this case, the tool has to perform an additional operation - separation. The products are left in the tool and their extraction presents a specific challenge.

Finished product manipulation is done in the third operation of the thermoforming process. The first phase of the operation is extraction of the products from the tool while the second phase is product stacking for further counting and packaging. Product extraction can be done in two ways. The first way is when the lower tool moves downward and the products are blown out of the tool through the working space to a net or a container. In such machines it is sufficient for the tool to perform only vertical translation. Products are taken from the container and stacked manually so they became non-sterile and can be deformed in the process thus making them nonfunctional. The second way are machines with complex tool motion. A mechanism extracts the products from the lower tool half through the working space. As the working space is relatively small and cloistered from both above and below (with the tool halves), the products are hard to access. This problem is solved with lower tool half rotation - products are exposed so the extraction mechanism can access them freely.

Thermoformed products have to be sterile. Because of this, completely mechanized manipulation devices are being developed to eliminate direct contact between a human operator and the product. Manipulation devices, beside their main function which is to transport thermoformed products, have to fulfill several other design requirements: high working speeds, simple mechanical structure, reliability and compactness.

3. MANIPULATOR STRUCTURE

Manipulation device consists of three basic assemblies: receiver, intermediary and transporter - Fig. 3.

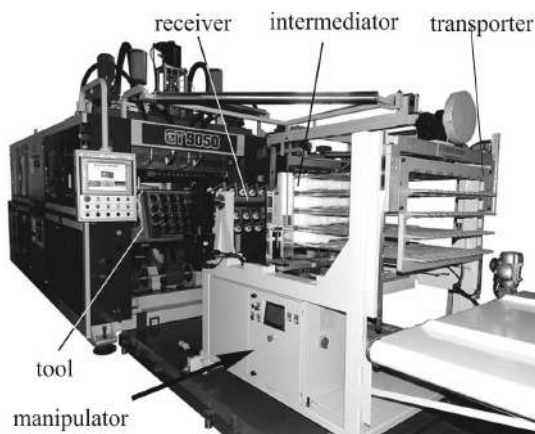


Fig.3. Manipulation device for thermoforming machine

1. Receiver - assembly which extracts products from the tool and transfers them to the next assembly (intermediator or transporter).
2. Intermediator - assembly which takes products from the receiver, stores them and transfers them to the next assembly (transporter).
3. Transporter - takes products from the intermediary (or receiver) and transfers them to the place of the stacking first phase.

3.1. Receiver

The receivers main task is to extract the product form the tool using vacuum pumps and to take them out of the work space. During extraction, the receiver is positioned parallel to the tool. After the taking hold of the products, the receiver moves toward the next assembly while performing both translation and rotation. This complex motion is generated with specific mechanisms - Fig. 4. The rotation angle of the receiver can be either 10° or 80° . First case is presented in Fig. 5-b and Fig. 5-d. A four bar mechanism with two sliders and linear actuation of the input link 2 is used (Fig. 4-a). The receiver is connected to link 3 which performs the needed planar motion. In this case the products are transferred in a horizontal position. Second case is presented in Fig. 5-a and Fig. 5-c. A crank slider mechanism with linear actuation of the input link 2 is used (Fig. 4-b). The receiver is connected to link 3 which performs the needed planar motion. In this case the products are transferred in a vertical position.

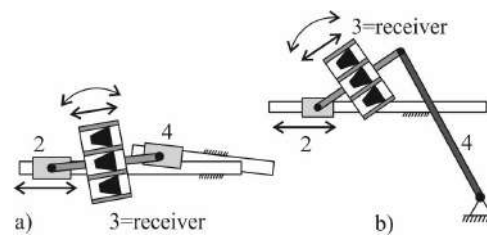


Fig.4. Mechanism for the receiver motion generation: a) receiver rotation angle 10° , b) receiver rotation angle 80°

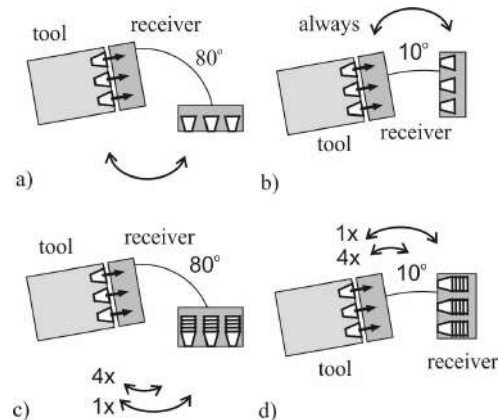


Fig.5. Receiver assembly solutions: a) rotation 80° , one set - P1, b) rotation 10° , one set - P2 c) rotation 80° , several sets - P3 and d) rotation 10° , several sets - P4

The second important characteristic of the receiver is the number of the product sets that it can store:

- 1 set - Fig. 5-a and Fig. 5-b. The receiver takes sets of products one by one and transfers them to the intermediary. It has one long stroke which is repeated. This receiver is simple with small dimensions.
- several sets - Fig. 5-c and Fig. 5-d. The receiver takes one set of products and stores products within itself. It performs several short strokes moving toward, taking products and moving away from the tool. After the last set of product has been taken, the receiver performs one long stroke to go out of the work space and to transfer the products to the next assembly. This receiver has a more complex design and larger dimensions.

3.2. Intermediator

The intermediary takes sets of products, one by one, from the receiver, stacks several sets into groups within itself and then transfers the grouped products to the transporter. The mediator's main task is to store products which were taken from the receiver. This is done because the duration the receiver's working cycle is equal to the duration of the machine cycle (time for one product set manufacture) while the duration of the transporter's working cycle is significantly longer.

The manipulator can be designed in two ways. First, without the mediator - Fig. 6-c. In the case when the receiver stores several sets of the products, the intermediary can be omitted as the transporter has enough time to realize its function. Design without the intermediary simplifies the manipulator structure. Second, with the intermediary – Fig. 6-a and Fig. 6-b. In the case when receiver takes only one set of the products, the intermediators existence is necessary because products are going to be stored in it. The manipulator with the intermediary, although more complex, offers more design possibilities.

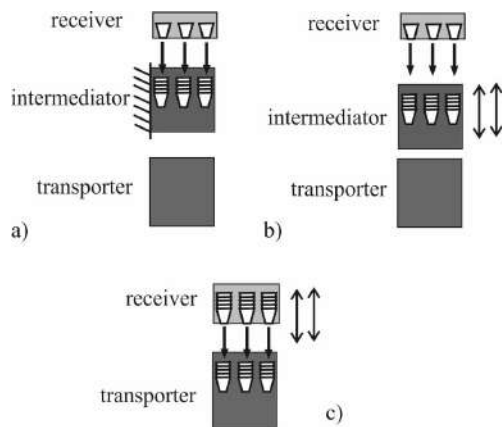


Fig.6. Intermediator assembly solutions: a) intermediary exists, it is stationary – M1, b) intermediary exists, it is not stationary – M2 and c) intermediary does not exist – M3

If an intermediary exists then it can be designed as a:

- Stationary intermediary – Fig. 6-a. The only advantage of this case is freedom in the conception of design solutions for elements for product transfer.
- Non stationary intermediary – Fig. 6-b. Intermediator may have translational motion which eases access and transfer of the products. This motion can be synchronized with the receiver or the transporter.

3.3. Transporter

The transporter transfers products to a space outside of the machine work space and prepares them for stacking. The transporter can perform:

- One motion – Fig. 7-a and Fig. 7-b. The transporter performs only one motion – the main motion, which transfers products into the position where they are going to be ejected from the transporter.
- Two motions – Fig. 7-c and Fig. 7-d. Beside the main motion, the transporter performs an auxiliary motion. During the auxiliary motion, the transporter moves toward the intermediary and assists in the product

transfer. The auxiliary motion is translational and it is necessary for a manipulator designed without the intermediary. Main and auxiliary motions can be generated with a one drive mechanism but, more often, they are separate and demand two independent drives.

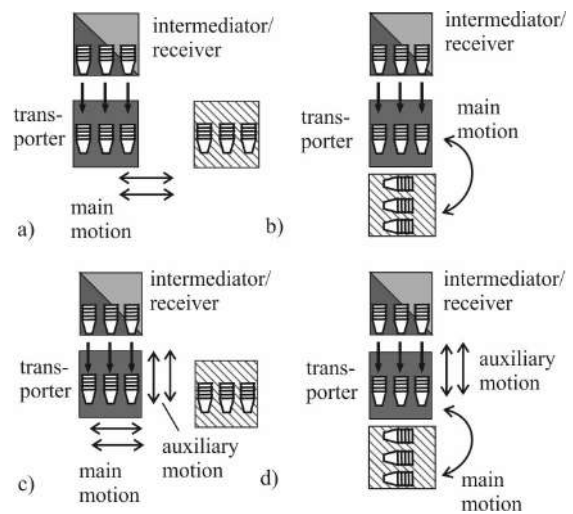


Fig.7. Transporter assembly solutions: a) one motion, translation – T1, b) one motion, rotation – T2 c) two motions, main motion is translation – T3 and d) two motions, main motion is rotation – T4

The main motion can be:

- Translation – Fig. 7-a and Fig. 7-c. Transporter moves translationally to bring products into the stacking position. This motion is perpendicular to the direction of product transfer from the intermediary.
- Rotation – Fig. 7-b and Fig. 7-d. Transporter moves rotationally to bring products into the stacking position. Rotational motion needs more space than translational but this way products can be ejected in a free space far away from the machine.

4. POSSIBLE MANIPULATOR STRUCTURES

Based on the previous considerations, it is possible to form all possible variants of the manipulator structure. The manipulator is formed from three assemblies: receiver P, intermediary M and transporter T. Considering the number of assemblies and their characteristics (P1 - P4, M1 - M3, T1 - T4) there are 48 possible manipulator structures but not all of them have practical sense. All possible combinations of assemblies i.e. manipulator structures are presented in Tab. 1.

Table 1. Possible manipulator structures

P1 M1 T1	P1 M3 T3	P2 M3 T1	P3 M2 T3	P4 M2 T1
P1 M1 T2	P1 M3 T4	P2 M3 T2	P3 M2 T4	P4 M2 T2
P1 M1 T3	P2 M1 T1	P2 M3 T3	P3 M3 T1	P4 M2 T3
P1 M1 T4	P2 M1 T2	P2 M3 T4	P3 M3 T2	P4 M2 T4
P1 M2 T1	P2 M1 T3	P3 M1 T1	P3 M3 T3	P4 M3 T1
P1 M2 T2	P2 M1 T4	P3 M1 T2	P3 M3 T4	P4 M3 T2
P1 M2 T3	P2 M2 T1	P3 M1 T3	P4 M1 T1	P4 M3 T3
P1 M2 T4	P2 M2 T2	P3 M1 T4	P4 M1 T2	P4 M3 T4
P1 M3 T1	P2 M2 T3	P3 M2 T1	P4 M1 T3	
P1 M3 T2	P2 M2 T4	P3 M2 T2	P4 M1 T4	

4.1. Manipulator structure P4 M2 T1

One solution is chosen from the possible solutions set and is presented in Fig. 8. This solution is a combination of the P4, M2 and T1 assemblies.

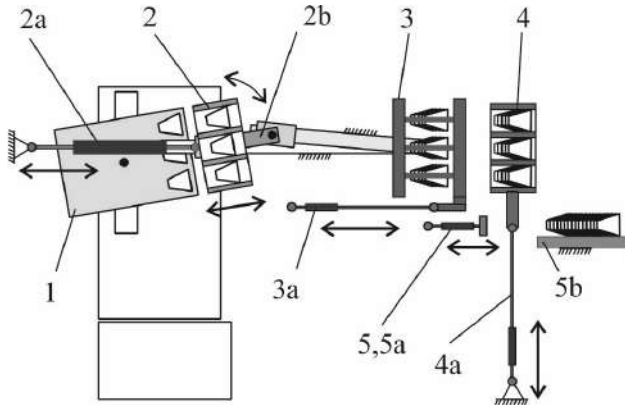


Fig. 8. Manipulator structure P4 M2 T1: 1 - lower tool half, 2 - receiver, 2a - receiver actuator, 2b - receiver mechanism, 3 - intermediary, 3a - intermediary actuator, 4 - transporter, 4a - transporter actuator, 5 - stacking device, 5a - stacking device ejectors and 5b - stacking device platform

After the thermoforming machine has formed the product, the lower tool half (1) assumes position for product extraction – the tool is positioned in its lowest vertical position and is rotated. The receiver (2) stands parallel to the lower tool half, extracts the first set of products and moves it to the right in a short stroke. When the machine finishes next product, the lower tool half assumes extraction position and the receiver positions itself next to the tool, extracts the products, stores them and moves out right. After the receiver has collected a predetermined number of product sets, it, instead of a short, performs a long stroke and comes to the intermediary (3). During the long stroke, the transporter rotates for 10° and assumes a horizontal position, parallel to the intermediary.

The actuator (3a) moves the intermediary (3) translationally to the left and transfer of the products from the receiver to the intermediary is done. The intermediary then moves to the right and transfers the products to the transporter (4).

The transporter is moved by its actuator (4a) translationally to the highest vertical position and it stands next to the intermediary. After it has taken the products it moves downward – step by step.

The ejectors (5a) of the stacking device (5) are activated and eject rows of products, one by one, from the transporter to the platform (5b).

At the end of the transporter motion, several long rows of the product are placed on the platform. They are transported, by rolling and a free fall, to a guide where a final one row of products is formed from which the products are counted and sent to packaging.

5. CONCLUSION

Using these considerations, the designer has insight into the complete set of the possible real design solutions so it makes it an easy task to analyze and evaluate individual solutions. One case of the structure of the manipulator is shown. In the further design process, it is necessary to form several design proposals which all satisfy project requirements and then, after detailed analysis, a final solution can be adopted.

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ANALYSIS OF THE CONCEPTUAL SOLUTIONS OF BIOMASS PELLET MILL

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Abstract: There are two basic principles for the realization of pellet mill. The first is based on the use of the ring die with press rollers and the second on a flat die. Dies can be movable and immovable. This paper presents analysis of the conceptual solutions of biomass pellet mill. Based on the set requirements eight conceptual solutions are proposed. The solutions are based on the use of immovable flat die with press rollers. The solutions are analyzed according to the method VDI 2225. Restrictions (technical and economic) are formed and conditions are set to estimate optimality of conceptual solutions. Based on the performed analysis two solutions stand out. The first is based on the use of motor-reducer and the second has a belt drive. Both solutions have a high and nearly equal values of technical and economic coordinates (score). For adoption of the optimal solution it is necessary to increase the range of estimation or develop both variants on the level of the preliminary CAD model, then repeat evaluation.

Key words: pellet mill, biomass, renewable energy

1. INTRODUCTION

Today a renewable energy sources such as biogas, wind and solar energy, hydropower and others are more and more in use. From renewable energy sources in the future the greatest contribution is expected from biomass [1]. Constantly increasing price of fossil fuels (fuel oil, natural gas, coal) encourage the increasing use of biomass as the cheapest source of energy, environmentally clean and present in large quantities [2]. The most significant advantage of biomass compared to fossil fuel is its reproducibility [3]. In agricultural production there is a large amount of unused biomass - corn stalks, straw, sunflower stalks and remains in pruning fruit trees and vines are readily usable form of energy. Production and use of biomass for energy purposes reduces emissions and contributes to the protection of land and water [4]. Pelletizing process significantly decreases the volume of

biomass and increases the density of the raw material. The density of the dry plant mass with admissible moisture content amounts 80–120 kg/m³ and 650–759 kg/m³ in the pelletizing process. Thereby the required space for accommodation of biomass decreases in relation 7:1 [5]. The basic condition that a fuel must fulfill in order to be found in wide use is to be suitable for transport and storage. The pellets represent most suitable way of storing biomass thereby the use of pellets is possible in a wide specter of combustion plants. The price of pellets is in fourth place, behind the fuel oil, electricity, liquefied petroleum gas, mazut and in front of natural gas, geothermal water, brown coal, hard firewood etc. [6]. Pellets are the most appropriate form of biomass for energy purposes. Due to the large power consumption and the growing needs for energy resources, the pellets have an important place as an alternative energy source. It is therefore necessary to develop technologies that would make the pelletizing process simple, efficient and cheap.

2. THE PELLETING PROCESS

After drying, comminuting and conditioning of biomass, pelletizing process is carried out on the pellet mill (Fig. 1). The main part of the pellet mill is a die with press rollers. There are two types of the die, in the form of a ring (Fig. 2-a) and a flat die (Fig. 3-a). Ring die is used for pressing hard woody raw materials and the flat die for pressing softer material (sawdust, straw etc.).

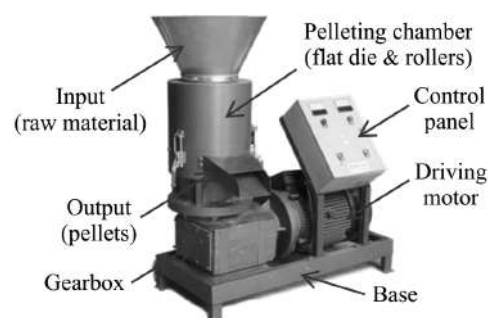


Fig.1. Pellet mill FY-400 [7]

There are two constructions of a pellet mill with a ring die. First, where the die rotates and rollers are not driven, but are rotated due to friction force on the raw material and the second, with immovable ring die and rollers that are rotating and rolling on the die. Mainly is used the first variant. To a ring die rollers suppress raw material radially (laterally) through the holes of the die (Fig. 2-b).

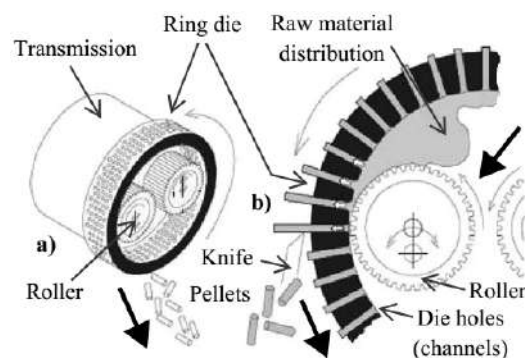


Fig.2. The pelletizing chamber with ring die

There are two variants with a flat die. The first consists of the immovable die at which the rollers rotate and roll, and the second of the rotating die and rollers that are rotating due to friction materials on the die. The first variant is used for lighter and softer materials and the second for stiffer materials. To the flat die raw material is falling from above on the rollers and die, whereby rollers gradually press out the raw material through the holes of the die (Fig.3-b).

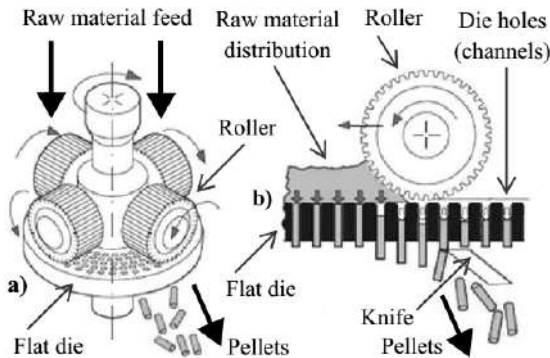


Fig.3. The pelleting chamber with flat die

For the pelleting process of the raw materials the most important tools of the pellet mill are die, press rollers and knife for cutting pellets. During pelleting process, raw material enters in to the pelleting chamber and is equally distributed at the die. On the die a thin layer of raw material is formed by rotating rollers or die. The press rollers create compressed, thick layer of raw materials. Rolling rollers create a strong pressure and clogg holes on the die. From the die exits infinite strand of raw material that is cut with a knife to the desired length.

For the production of pellets important characteristic is ratio between the hole diameter and lenght of hole (channels) on the die. In the pelleting process of woody mass (chip or sawdust) ratio for pressing is 3:1–5:1. Raw materials with small bonding strength require longer channels for pressing and vice versa. The temperature of the raw material in the channel (hole) of the die grows with increasing length of the channel, whereby the hardness of the pellet increases. The pelleting requires a continues supply of the press with homogeneous raw material, comminuted enough, with constant moisture content, which for certain types of pellet mill and concrete woody raw material is 8–12%. When pelleting agricultural biomass the moisture content in the raw material is 12–14%. Small speed of the rollers 2.5 m/s enables good pellets quality. Thin layer of raw material in pelleting chamber enables large traffic capacity. The clearance between the rollers and the die is 0.5–1 mm.

3. ANALYSIS OF CONCEPTUAL SOLUTIONS

Pellet mills with a flat die have several advantages. They have simple construction. The raw material is fed by gravity. Adjusting the clearance between the press rollers and the die enables pelleting of different kind of raw materials. During the pelleting a process of compressing the raw material can be monitored. They are easier to clean and maintain. They have a significantly lesser weight and dimensions comparing to pellet mills of the

same power with ring die. The endurance of flat dies is longer. Pellet mills with ring die are significantly complex and more expensive to produce Basic requirements for pellet mill realization are functionality and reliability, lesser driving force, high efficiency, compact construction, low weight, small size and low cost of production. The force generated by the motor must be significantly smaller than the force of the press rollers which is achieved by the principle of reduction. The flat dies are simpler and less demanding in terms of production and assembly than the ring dies, therefore only the flat dies are analyzed.

3.1. The conceptual solutions

Based on the set requirements eight conceptual solutions are proposed for pelleting of biomass (Fig. 4). The solutions are based on the use of immovable flat die with press rollers. For reduction of torque from the motor to the press rollers geared, chain and belt drive are considered.

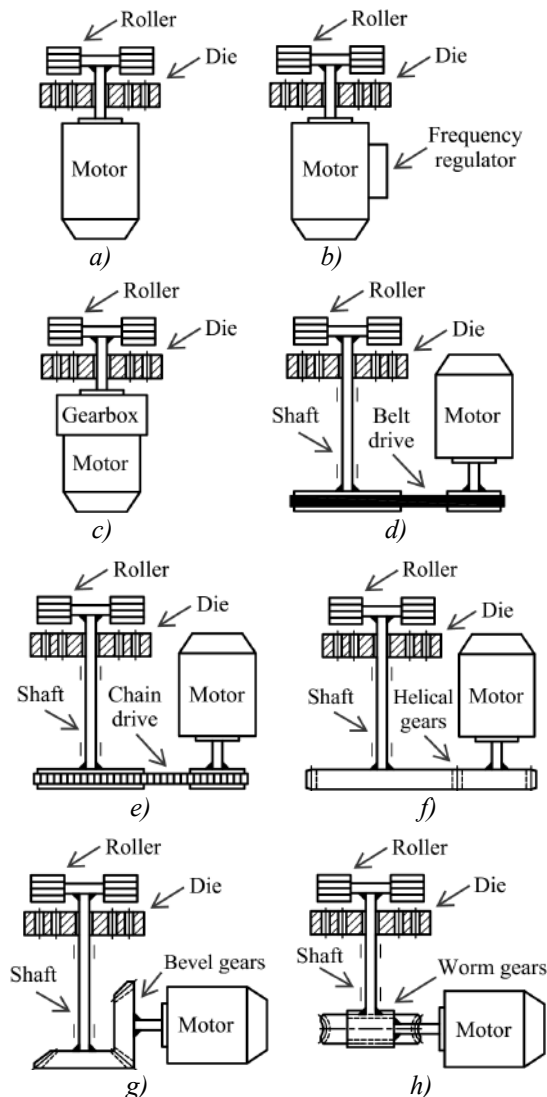


Fig.4. Conceptual solutions

Conceptual solution from Fig. 4-a consists of a die and of press rollers that are attached to the motor shaft. The solution is simple, but requires the adoption of motor for in advance known environment. Therefore it is necessary

to calibrate biomass with the adequate moisture percentage before pelleting. A variant from Fig. 4-b is similar to the previous and contains a frequency regulator for controlling the motor which enables to change the speed of rollers, depending on the structure of the biomass. Lack of the variant is the price of the frequency regulator that exceeds the price of the motor. The solution from Fig. 4-c consists of a die and of press rollers that are attached to the shaft of motor-reducer. The solution is suitable for pelleting bulkier biomass which requires powerful and reliable drive. Conceptual solution from Fig. 4-d consists of a die with press rollers of belt drive and motor. Torque is transmitted from the motor and transforms through a belt drive on the shaft and press rollers. Applying pulleys with different diameters (belt drive reducer) the speed range of the rollers is extending and enables pelleting different kind of raw material. Belt drive allows a short-term overload during work. Variant on Fig. 4-e for reduction uses a chain drive and variant from Fig. 4-g cylindrical gears. Unlike of belt drive the ratio with the chain drive is constant. In conditions of excessive dust, dirt and filthiness a drive with belt pulleys is more suitable. Variant from Fig. 4-g for reduction uses a spiral bevel gears. This type of gear is less noisy, enables the transmission of high power and has a significantly higher carrying capacity compared to cylindrical gears of the same characteristics, but also a higher price because they are more complicated to produce. Proposed variant enables horizontal position of the motor. Variant solution from Fig. 4-i for reduction uses a worm gear. This drive is used for pelleting bulkier biomass and requiring high power and lower number of revolutions of the press rollers.

3.2. The optimal conceptual solution

Within this operation the optimal solution is chosen from a set of proposed variants. Constructional solution must fulfill several boundary conditions and criteria. The variant that in to the necessary extent fulfills all the constraints and criteria is considered optimal. One of the criteria is technical and the second economic. Technical implies a degree of fulfillment of function to which conceptual solutions are intended, and the economical

acceptability of variant solutions for realization. According to the recommendations VDI 2225 [8] the method for selecting the optimum conception variant is defined. It is a two-criterion method to whom should fulfill two function objectives respectively the criteria (technical and economic). For each of them restrictions are introduced that determine the area of acceptable solutions. It belongs to a group of methods for decision-making that is carried out over a set of, in advance known, rationally placed variants. The conception variants are defined only in terms of the principles of work. Materials, technology, shape etc. are not defined. Even if they are, a mathematical optimization model would be difficult to implement. For this reason, this method is based on a subjective evaluation (assessment) of conception variants.

Subjective evaluation is an important characteristic of this method that allows solving complex issues in a simple way. It is based on an estimate in which extent is fulfilled each of the boundary conditions.

The fulfillment degree of boundary conditions is estimated according to the grades, which according to VDI 2225 can be 0-4. Grade 0 means that a solution is not acceptable. Grade 1 means that a solution is not good, but it can be tolerated. Grade 2 means that the solution is acceptable, grade 3 that the solution is good and 4 that the solution is very good respectively ideal.

Based on the described method, technical and economic constraints are formed for evaluation of conceptual solutions. Within the technical constraints the boundary conditions are weight, dimensions, compact, reduction, reliability, speed control, overload, assembling/dismantling, and in the context of the economic constraints are number of parts, complexity of parts, complexity of assembly, motor size, speed control device and maintenance. To certain boundary conditions is increased or decreased influence for evaluation of solutions which is involved by a coefficient k_j (have a value greater or less than 1). In Tab. 1 are shown the results of evaluation of conceptual solution. For each variant $i=1...k$ based on grades p_j ($j=1...n$) is determined a technical $X_i=0...1$ and economic coordinate $Y_i=0...1$, according to equations:

Table 1. Evaluation of conceptual solutions according to technical and economic criteria

	Restrictions (boundary conditions)	k_j	Conceptual solutions								Ideally $p_{j\max}$
			a	b	c	d	e	f	g	h	
Technical	Weight	0.75	4	4	3	3	3	2	3	2	4
	Dimensions	0.75	4	4	4	3	3	3	2	2	4
	Compact	1	4	4	4	3	3	3	2	2	4
	Reduction	1.25	1	1	3	3	3	3	3	4	4
	Reliability	1.5	2	2	4	4	4	4	4	4	4
	Speed control	1.25	1	4	1	3	1	2	1	1	4
	Overload	1.5	1	1	3	4	3	3	3	3	4
	Assembling/dismantling	1	4	4	4	3	3	2	2	2	4
	X-coordinate		0.58	0.68	0.8	0.83	0.72	0.71	0.65	0.66	1
Economic	Number of parts	1	4	4	4	3	3	3	2	2	4
	Complexity of parts	1.25	4	4	4	4	3	3	2	2	4
	Complexity of assembly	1	4	4	4	4	4	3	2	2	4
	Motor size	1.5	1	1	2	3	3	3	3	4	4
	Speed control device	1.5	4	1	4	4	4	4	4	4	4
	Maintenance	1	4	4	4	4	3	3	3	3	4
	Y-coordinate		0.84	0.69	0.9	0.91	0.84	0.80	0.69	0.74	1

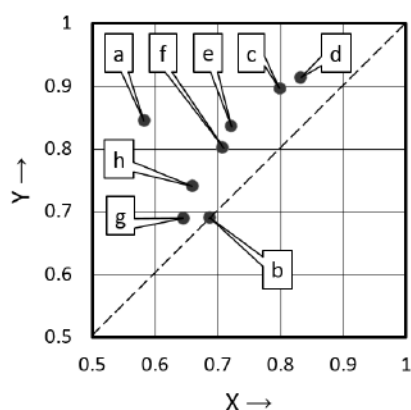


Fig. 5. The acceptability of conceptual solutions:
X-technical and Y-economic score

$$X_i = \frac{k_1 p_1 + k_2 p_2 + \dots + k_j p_j}{k_1 p_{1\max} + k_2 p_{2\max} + \dots + k_j p_{j\max}} = \frac{\sum_{j=1}^n k_j p_j}{\sum_{j=1}^n k_j p_{j\max}} \quad (1)$$

$$Y_i = \frac{k_1 p_1 + k_2 p_2 + \dots + k_j p_j}{k_1 p_{1\max} + k_2 p_{2\max} + \dots + k_j p_{j\max}} = \frac{\sum_{j=1}^n k_j p_j}{\sum_{j=1}^n k_j p_{j\max}} \quad (2)$$

Obtained grades above 0.8 classifies conceptual solutions as very good. Grades above 0.7 in the group of good, and below 0.6 in the group are unsatisfactory. The optimal solution is the one that nearly equal fulfills both criteria (technical and economic), and technical and economic values (score) are high enough. On Fig. 5 that are dots closer to the dashed line at an angle of 45° and farther from the origin. From Fig. 5 two solutions can stand out that have high and nearly equal values of technical and economic coordinate (score), above 0.8. The first solution (c) is based on the use of motor-reducer and the second (d) have a belt drive. Both solutions exclude the use of device for speed control (frequency regulator). Applying pulleys with different diameters (belt drive reducer) the speed range of the rollers is extending and enables pelleting different kind of raw material, which is very significant. Considering the subjectivity of the method in estimation of boundary conditions, it is not recommendable that the decision is made based on very small differences, which is here the case. Therefore, for this to variants is needed to repeat the evaluation but with greater differences in grades. The second possibility is to develop both variants on a level of preliminary CAD model. Only then can more certainly give grades about technical and economic capability of variants and choose the optimal solution that exceeds the scope of this paper.

4. CONCLUSION

In paper is shown analysis of the conceptual solutions of biomass pellet mill. Based on the set requirements eight conceptual solutions are proposed and all are based on the use of immovable flat die with press rollers. Based on the method VDI 2225, restrictions (technical and economic)

are formed and conditions are set to estimate optimality of proposed solutions. Based on the performed analysis two solutions stand out. The first solution is based on the use of motor-reducer and the second have a belt drive. Both solutions exclude the use of device for speed control (frequency regulator). Applying pulleys with different diameters (belt drive reducer) the speed range of the rollers is extending and enables pelleting different kind of raw material. Both solutions have a high and nearly equal values of technical and economic coordinates (score). For adoption of the optimal solution it is necessary to increase the range of estimation or develop both variants on the level of the preliminary CAD model, than repeat evaluation.

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DETERMINATION OF BASIC MECHANICAL PARAMETERS OF THE TRACTOR TYRE BY USING UNIVERSAL APPROACH

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Abstract: Tyre behaviour exerts crucial impact on every aspect of vehicle dynamics response. It is therefore of interest to have access to appropriate tyre model in order to address vehicle behaviour optimization by means of computer simulations. In this paper procedures for determination of some basic parameters of tractor tyre by using simple test facility and universal measuring techniques are described, including dependence between tyre vertical load, static radial deflection and contact length on the hard level ground, linearized radial stiffness and damping coefficients. Impact of tire pressure and vertical load on investigated quantities and relationships is also examined. Set of tyre parameters that can be used in simple tyre model was obtained. Linearized stiffness and damping coefficient values can be used in certain applications of tyre viscoelastic structure modelling. Relationships between tyre load, deflection and contact length can be useful when investigating tyre geometric or filtering properties such as enveloping behaviour.

Key words: : tractor tyre. testing, model parameters

1. INTRODUCTION

Tyre behaviour exerts crucial impact on every aspect of vehicle dynamics response. For the contemporary tractors, due to simultaneous increase of travelling speeds as well as trailing load capacity, issues of handling and stability of tractor itself or in combination with the trailer have become increasingly important for the public road safety [3]. At the same time optimization of tractor vibration behaviour always was and still is one of the most challenging task in tractor research and product development [2]. In order to be able to address such topics by means of computer aided vehicle dynamic simulations, proper model of tyre behaviour is of substantial significance for the validity of simulation results. For development, parameterization and/or

validation of such tyre models, appropriate test facility is needed to investigate real tyre response. Such facilities are as a rule expensive due to high technical level, and this property is even more pronounced in the case of tractor tyres, due to large dimensions and much higher operational loads. For this reasons, tractor tyre test facilities are relatively rare and researchers who might want to explore vehicle dynamics of contemporary agricultural tractors by means of computer simulations have limited access to tyre data required for appropriate model.

The goal of this paper is to propose procedures to obtain at least basic tyre model parameters, by using simple test facility and universal measuring techniques and approaches, more accessible in usual budget-constrained university environment. The following features will be included:

- Dependence between tyre vertical load, static radial deflection and contact length on the hard level ground
- Linearized radial stiffness and damping coefficients
- Impact of tire pressure and load on investigated quantities and relationships.

2. TEST FACILITY

Experimental facility used in these investigations was described in more details in a number of previous papers (e.g. [4,6]) so that only the brief description will be presented herein. Facility is depicted in the Figure 1.

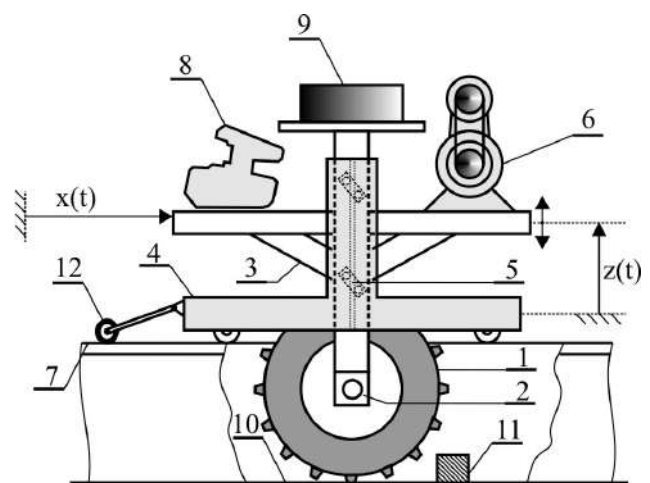


Fig.1. Composition of the test facility: 1-tested wheel, 2-wheel axle bearing, 3-wheel mounting frame, 4-wheel guiding cart, 5-vertical guide, 6-driving motor with reduction gearbox, 7-cart guiding rail, 8 and 9-changeable weights for vertical wheel load adjustment, 10-level ground, 11-road irregularity, 12- longitudinal travel sensor; $x(t)$, $z(t)$ -longitudinal and vertical travel of the wheel rim

Its main features are as follows:

- Tested wheel (1) can roll freely along test ground (10), on which different forms of road profile (11) can be mounted to investigate tyre enveloping behaviour or vertical dynamics
- Longitudinal guidance for the facility is provided by the cart (4) and the rails (7)

- Wheel axle (2) is pivoted at the bottom of the carrying frame (3), which can move freely in vertical direction with regard to the cart, within the guides (5), providing the wheel with additional vertical degree of freedom
- Tyre vertical ground reaction is determined by the weight of the wheel itself, frame (3) with equipment (e.g. driving system (6)), and exchangeable weights (8 and 9)
- Induction displacement transducer is mounted between the frame (3) and the cart (4) to register change of the vertical position of the wheel with respect to the cart

3. DESCRIPTION OF MEASURING PROCEDURES

3.1. General procedure and equipment

For the measurements carried out in the scope of this work, the following equipment has been used, as depicted in Fig. 2: displacement transducer, force transducer, universal measuring amplifier and PC computer for data storage.

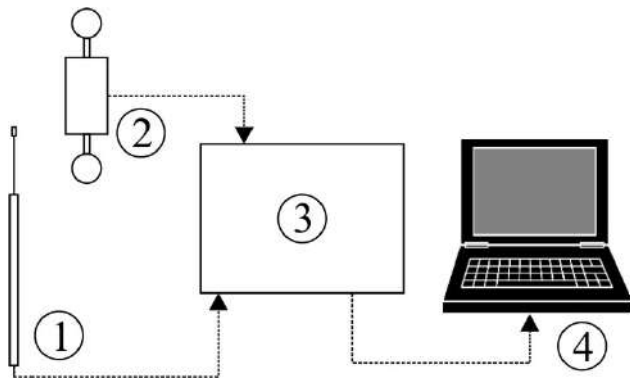


Fig. 2. Measuring equipment: 1-displacement transducer, 2-force transducer, 3-measuring amplifier, 4-PC

Main components, and the basic measurement principles, which will be discussed subsequently, are presented in the Fig. 3. The upper end of the force transducer (2) is connected to the hook of the manually operated hydraulic crane (not presented in the Fig. 3), while the lower end is connected to the wheel carrying frame (4) by the mean of the lifting cable (3). Applying lifting force F_L , tyre ground reaction will change, together with tyre deflection and contact length. Since the force transducer is connected in series between the crane and wheel carrying frame, it is loaded by the lifting force F_L which enables its direct reading from the transducer signal.

Tyre deflection change is determined from the signal of the displacement transducer, which is located between the wheel guiding cart (4, Figure 1) and the wheel carrying frame (3, Figure 1), which moves in vertical direction together with the wheel rim. Finally, contact length is obtained by using two thin (<1mm thickness) rigid plates of rectangular shape, touching opposite edges of the tyre contact area, and measuring their longitudinal distance by using the measuring tape.

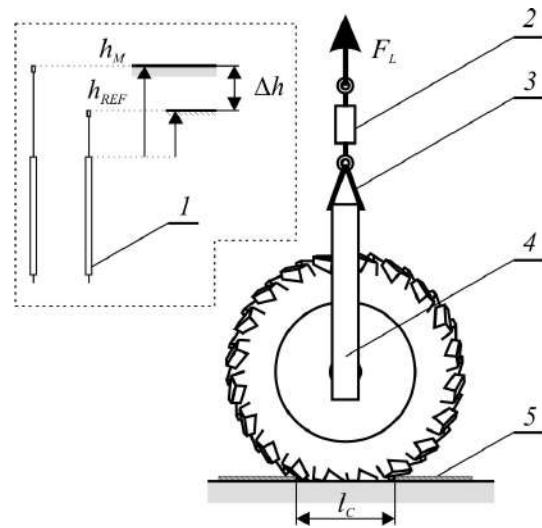


Fig. 3. Measuring principle and main components: 1- displacement transducer, 2-force transducer, 3- lifting cable, 4-wheel carrying frame, 5-rigid plates of small thickness, F_L - lifting force exerted at the top, h - vertical tyre position, l_c - contact length

Values of measuring signals are recorded on a computer, and their instantaneous values can be read from the computer display using measuring software.

All measurements are carried out with 3 levels of system mass (i.e. total weight of the carrying frame together with the wheel and all mounted equipment and loads, determining tyre ground reaction), and 5 levels of tyre pressure, shown in the Table 1. Considerations about the choice of these values are discussed in [5].

Table 1. Operational conditions: system mass and tyre pressure

Mass values [kg]				
Level 1	Level 2		Level 3	
660	960		1440	
Tyre pressure values [bar]				
Level 1	Level 2	Level 3	Level 4	Level 5
0.8	1.1	1.4	1.7	2.0

All measurements were made using the test wheel of the size 12.4R28 ★121 A8 (8 P.R.)

3.2. Procedure for the determination of dependence between tyre vertical load, static radial deflection and contact length

When the lifting force F_L (Fig. 3) is not applied, tyre ground reaction is equal to the weight of the carrying frame together with the wheel and all mounted equipment and loads. In this equilibrium position, tyre deflection and contact length have values that can be adopted as referent. Referent reading of the system height is obtained by the displacement transducer signal. It is not necessary to transform its signal to obtain zero value at this point, because the relative change of its value is compared to initial one, and the absolute signal values are not of

importance. Applying some initial lifting force through the crane, a new static equilibrium position is obtained, for which it applies:

$$F_L + Z_G = W \Rightarrow Z_G = W - F_L \quad (1)$$

where:

- F_L - lifting force
- Z_G - vertical tyre ground reaction
- W - weight of the carrying frame together with the wheel and all mounted equipment and loads

Using known value for W and force transducer signal value for F_L , the actual value of the ground reaction Z_G can be determined directly from the expression (1).

Ground reaction Z_G will change with the change of the force F_L . At the same time tyre deflection and the contact length will change accordingly. The following labels are now introduced (see also Fig. 3.):

- $Z_{G0} = W$ - ground reaction in the absence of the lifting force ($F_L = 0$)
- l_{C0} - tyre contact length when $Z_G = Z_{G0}$
- f_0 - tyre radial deflection when $Z_G = Z_{G0}$
- l_C and f - universal contact length and deflection values (for $F_L > 0$)
- h_{REF} - displacement transducer signal value when the wheel is in the referent position ($F_L = 0$)
- h_M - displacement transducer signal value for universal wheel position ($F_L > 0$)
- h_0 - displacement transducer signal value exactly in the position when Z_G becomes zero ($F_L = W$)
- $\Delta h = h_M - h_{REF}$ - difference between the instantaneous and the referent displacement transducer signal

By gradually increasing F_L , ground reaction will be decreasing according to (1). Exactly observing the instant when Z_G becomes zero (bottom part of the tyre circumference still just touching the ground but without producing any reaction force), value for deflection f_0 is obtained from the instantaneous value of the displacement transducer signal h_0 :

$$f_0 = h_0 \quad (2)$$

Tyre deflection at arbitrary state (for the universal value of F_L) can then be calculated from:

$$f = f_0 - \Delta h \quad (3)$$

By using relationships (1), (2) and (3), by varying F_L , dependence between tyre load Z_G and the deflection f is obtained as a series of numerical values.

Varying of the lifting force F_L has to be done very gradually and in both directions (increase and decrease), in order to eliminate impact of tyre hysteresis. At the same time, after the force/deflection increment has reached a certain value, force variation is temporarily paused to carry out measurement of the tyre contact length, as shown in the Fig 3. and 4.

Measured value of the contact length is then recorded manually, together with the actual value of the lifting force. Series of values is then entered into computer memory for the storage and subsequent analysis and calculations. Establishing dependence between ground force and contact length, relationship between contact length and tyre deflection is also obtained, taking into account correspondence between the ground force and deflection. Described procedure for contact length

measurements is technically simple to carry out, but it is also rather manual-labour intensive and time consuming. The whole described procedure is repeated for different values of the tyre pressure and vertical load.

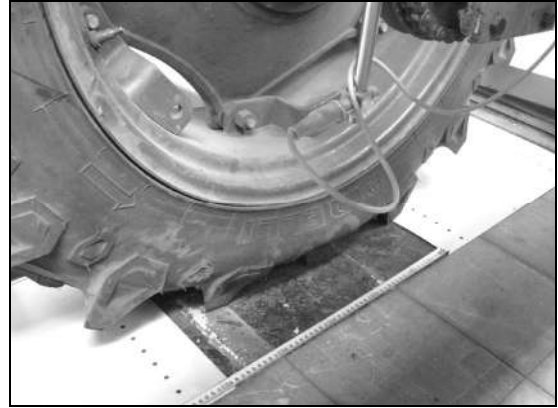


Fig.4. Measuring tyre contact length

3.3. Procedure for the determination of linearized radial stiffness and damping coefficients

Coefficient of stiffness can be determined in the following ways:

- Statically, according to the slope of the curve of tyre deflection versus vertical load, and
- Dynamically, according to the natural frequency of tyre vertical vibration

Damping coefficient can be obtained through the time response of free vertical damped vibration by using logarithmic decrement method. Vibration is excited by the free fall of the tyre from the certain height level ("drop test"), which does not have to be much higher than the static equilibrium height.

Logarithmic decrement δ is used both in calculation of damping coefficient and dynamically determined stiffness coefficient:

$$\delta = \frac{1}{T_D} \cdot \ln \frac{z_i}{z_{i+1}} \quad (4)$$

where:

- T_D - quasi-period of damped vibration
- z_i, z_{i+1} - two adjacent amplitude peaks of the time response curve

Knowing vibrating system mass m , damping coefficient k can now be calculated:

$$k = 2 \cdot \delta \cdot m \quad (5)$$

Free natural frequency of the system ω_0 is:

$$\omega_0 = \sqrt{\left(\frac{2\pi}{T_D}\right)^2 + \delta^2} \quad (6)$$

Dynamically determined stiffness coefficient c is calculated from:

$$c = m \cdot \omega_0^2 \quad (7)$$

Finally, relationship for statically determined stiffness coefficient c is:

$$c = \frac{\Delta Z_G}{\Delta f} \quad (8)$$

where approximation of linear tyre response is adopted in the area of nominal vertical tyre load.

It is well known from the previous investigations (e.g. [1]) that both tyre stiffness and especially damping are dependent on both excitation frequency and the tyre rolling speed. Therefore, results obtained in this work should be viewed as more or less rough approximation. More broad set of test data in view of free damped vibration response with different mass values (meaning different natural frequencies of the system) would though enable empirical modelling of damping coefficient dependence on excitation frequency.

4. MEASUREMENTS RESULTS

4.1. Vertical load, tyre deflection and contact length

By using described procedure for the determination of dependence between tyre deflection and the vertical load, results were obtained that are presented in the Fig. 5.

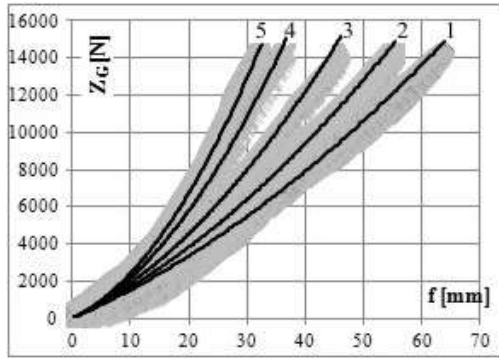


Fig.5. Dependence between tyre deflection f and the vertical load Z_G for different pressure values; 1-0,8 bar, 2-1,1 bar, 3-1,4 bar, 4-1,7 bar, 5-2,0 bar

Fig. 5. shows that, for full range of tyre loads, dependence between tyre load and the deflection is clearly non-linear. However, considering only limited load range in the area of tyre nominal load (i.e. taking very small loads and deflections out of consideration), Fig. 6. shows that linear dependence between load and deflection for this load range represents very acceptable approximation.

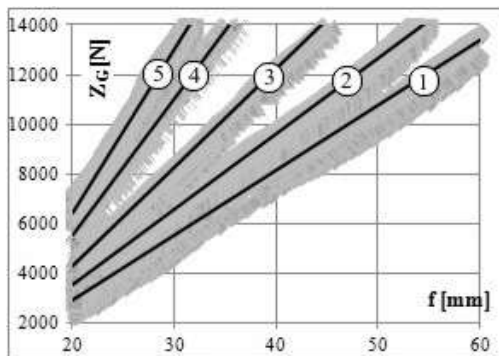


Fig.6. Linear approximation of dependence between tyre deflection f and the vertical load Z_G for limited load range (pressure values as in Fig. 5)

Dependence between contact length and tyre deflection is shown in the Fig. 7. Relationship can be well approximated by degressive power growth trendline. Results clearly show that this dependence is of pure geometrical nature, i.e. it is not affected by the tyre pressure or vertical load.

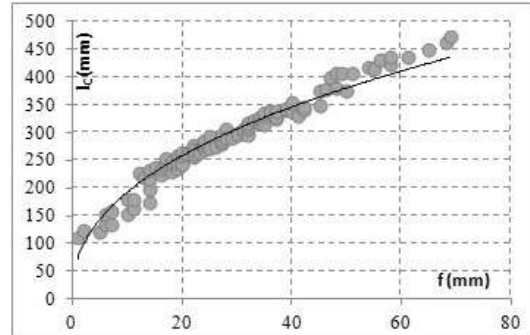


Fig.7. Dependence between contact length l_c and tyre deflection f

4.2. Tyre radial stiffness and damping

Results for the statically determined stiffness coefficients for different tyre pressure values were obtained according to the relation (8), by using curves shown in the Fig. 6. For dynamically determined stiffness, relations (4), (6) and (7) were used. Parameters T_D and z_i (expression 4) were obtained from the oscillogramme, i.e. time history of tyre free response after the drop-test. An example of the oscillogramme is shown in the Fig. 9. Both statically and dynamically determined coefficient values for different pressure levels are presented in the Table 2., and their dependence on pressure is also shown graphically in the Fig. 8.

Table 2. Overall results of the stiffness coefficient calculations

Pressure level	Stiffness [N/m]	
	Static	Dynamic
1 (0,8 bar)	261088	377384
2 (1,1 bar)	304957	394732
3 (1,4 bar)	364064	461916
4 (1,7 bar)	520127	537827
5 (2,0 bar)	573638	585737

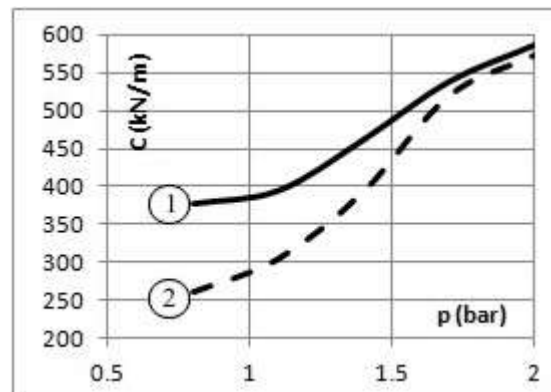


Fig.8. Dependence of statically and dynamically determined stiffness coefficients on pressure

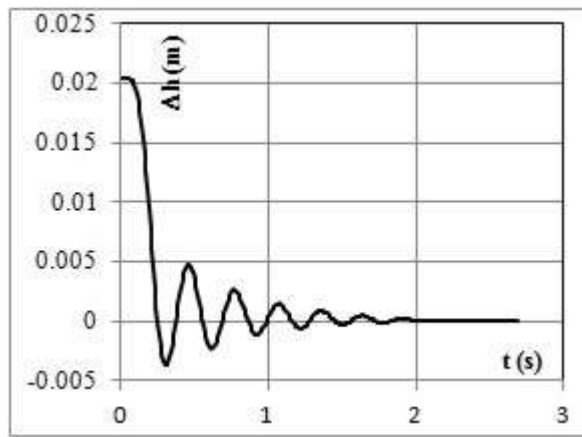


Fig.9. An example of the oscillogramme - time history of tyre free response after the drop-test; t - time, Δh - tyre height with respect to static equilibrium

Results presented in the Table 2 for dynamically determined stiffness coefficient values represent mean values of measurements for three different levels of tyre static load e.g. system mass/weight (according to Table 1). Results for individual values of tyre static load are shown in the Table 3.

From the results in Table 3 a trend toward rise of the stiffness with higher tyre load can be noticed (though with certain deviations). This phenomenon could be attributed to nonlinearity of the dependence between tyre static load and radial deflection (Fig. 5). For higher loads, tyre vibrates about central position corresponding to greater static deflection, exhibiting greater slope of the load-vs.-deflection curve, which corresponds to higher value of linearized stiffness coefficient.

Table 3. Results of dynamically determined stiffness coefficient calculations

Pressure level	Dynamic stiffness [kN/m]		
	m=660kg	m=960kg	m=1440kg
1	339,2	377,2	415,7
2	355,1	398,0	431,0
3	403,6	512,0	470,2
4	462,6	619,4	531,5
5	486,4	683,7	587,0

5. CONCLUSION

This paper describes procedures for determination of some basic tractor tyre parameters by using simple test facility and universal measuring techniques. Dependences between tyre vertical load, static radial deflection and contact length on the hard level ground, as well as stiffness and damping, were investigated.

All described measuring procedures were repeated for different values of tyre pressure and vertical load, which enabled investigation of impact of these two important tyre operational parameters on its behaviour from the observed points of view.

Measurement results shown acceptable level of deviation, so that they were easy for analysis and interpretation. Dependence between deflection and ground force shown non-linearity, although linear behaviour can be used as good approximation in the area of nominal tyre load. It was observed that the dependence between tyre deflection and contact length is of purely geometrical nature, i.e. it is not affected by the tyre pressure and vertical load. Some discrepancy was observed between statically and dynamically obtained results for linearized stiffness coefficient, which is ascribed to non-linear nature of tyre response.

Linearized stiffness and damping coefficient values can be used in certain applications of tyre viscoelastic structure modelling. Caution is though needed as real tyre exhibits dependence of these values on rolling speed and excitation frequencies. Relationships between tyre load, deflection and contact length can be useful when investigating tyre geometric or filtering properties such as enveloping behaviour.

ACKNOWLEDGEMENTS

This paper was done as a part of the researches on the project TR35041 – "Investigation of the safety of the vehicle as part of cybernetic system: Driver-Vehicle-Environment" and the project TR31046 "Improvement of the quality of tractors and mobile systems with the aim of increasing competitiveness and preserving soil and environment", supported by Serbian Ministry of Education, Science and Technological Development.

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STRESS AND STRAIN STATE OF CYCLOID GEAR UNDER DYNAMIC LOADS

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Miloš MATEJIĆ

Abstract Besides the large transmission ratio, extremely compact design and high efficiency ratio, cycloid drives are recognizable according to very good working characteristics in dynamic load conditions. Thanks to the simultaneous coupling of almost one third of the number of cycloid gear teeth, cycloid drives can accept large short-term shock loads without any consequences.

The most important element of cycloid drive in terms of geometric and kinematic complexity (cycloid gear tooth profile represents equidistant of shortened epitrochoid) as well in relation to the nature of the load on which is exposed, certainly the cycloid gear. This paper analyses the stress-strain state of cycloid gear under real working conditions (in terms of dynamic loads). Observed the different load cases and finite element is determined distribution of strains and stresses on the body cycloid gear.

Key words: cycloid drive, cycloid gear, dynamic load, FE analysis

1. INTRODUCTION

Cycloid drives from day to day due to excellent performance characteristics that they have (large transmission ratio, a very compact design, light weight, high efficiency ratio, exceptional reliability ...) are increasingly present in today's global market, Figure 1. The first mathematical model for calculation of basic geometric and kinematic parameters cycloid drives, and loads that occur on its main elements define the Kudrijavcev, [1]. A very detailed analysis of the distribution of load and calculation of efficiency ratio of cycloid drives is presented in the papers [2,3,4]. Sensinger conducted an optimization of basic parameters cycloid drives (cycloid gear tooth profile, efficiency ratio, stress state), [5].



Fig.1. Model of single stage cycloid reducer

The large amount of modern researching's is aimed at developing new concepts of cycloid drives in order to obtain a more favourable performance of this group of power transmission, [6,7,8]. Detailed analysis of the stress-strain state of cycloid gear in static conditions (numerical and experimental) is presented in this paper [9]. However, in real working conditions the dominant are dynamic loads. Kahraman has defined a series of dynamic models of planetary gears that quite accurately describe their dynamic behaviour in real working conditions, [10]. When it comes to the dynamic behaviour cycloid drives, researching in this field are in a quite smaller amount related to classical planetary reducers. A result of that researching's presented in the papers [11,12,13].

In this paper the analyses of stress-strain state of cycloid gear are presented, under the influence of dynamic loads on a specific single stage cycloid reducer, Figure 1.

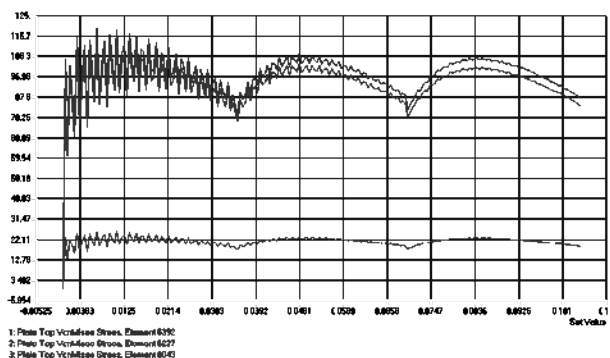
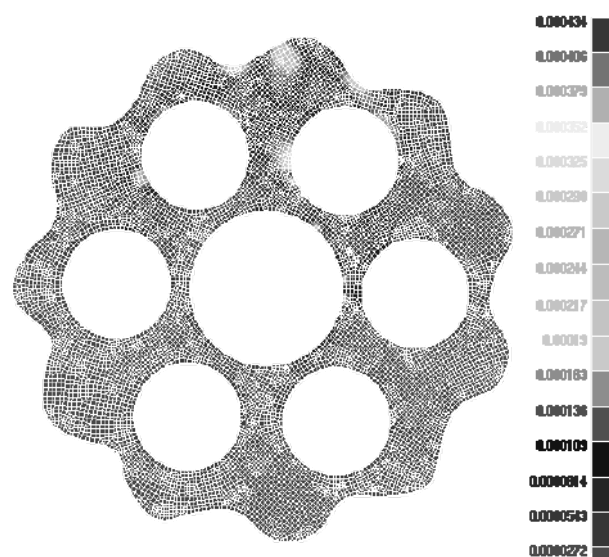
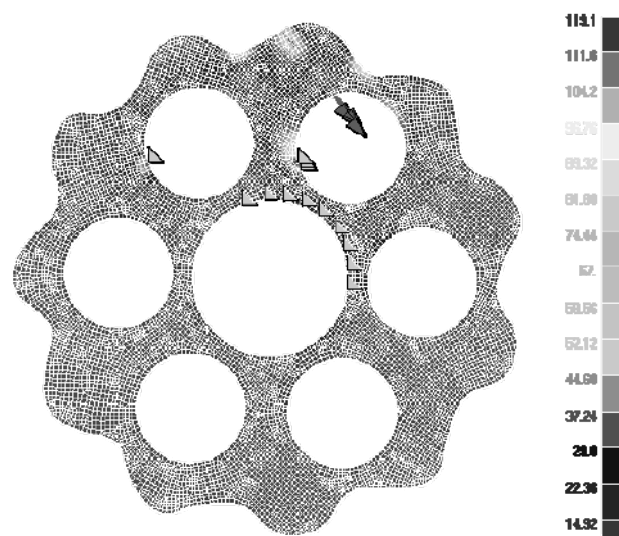
2. NUMERICAL ANALYSIS OF STRESS-STRAIN STATE OF CYCLOID GEAR UNDER DYNAMIC LOADS

Li Lixing defined expressions for the calculation of s and backlash of cycloid gear teeth and rollers of stationary central gear, [14]. Based on these expressions, in the paper [15], a mathematical model is developed that defines the dynamic behaviour of single-stage cycloid drive, Figure 2. The main output parameter of the mathematical model is a normal dynamic force F_{din} which occurs in contact cycloid gear tooth and roller of stationary central gear.

In this paper the single-stage cycloid reducer, with the following working characteristics, has been analysed:

- Input power: $P_{in} = 5,5 \text{ kW}$;
- Input number of revolutions: $n_{in} = 1500 \text{ min}^{-1}$;
- Gear ratio: $u = 11$;
- Number of cycloid gear teeth: $z = 11$;
- Eccentricity: $e = 4 \text{ mm}$.

A series of different numerical models of cycloid gear are made. Modelling, pre-processing and post processing were done in the software package FEMAP, while the calculation itself done in the program MSC NASTRAN.



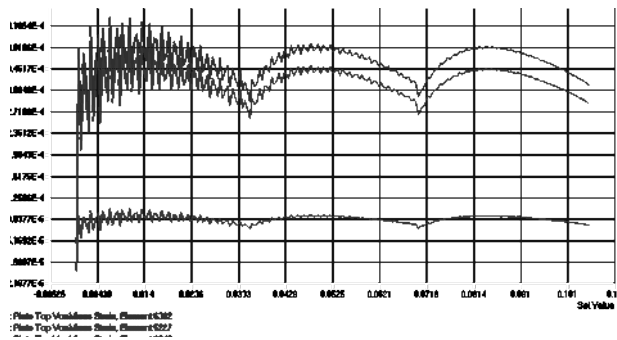


Fig.6. Strain distribution on cycloid gear (model 1) for finite elements 6392, 5227 and 8043

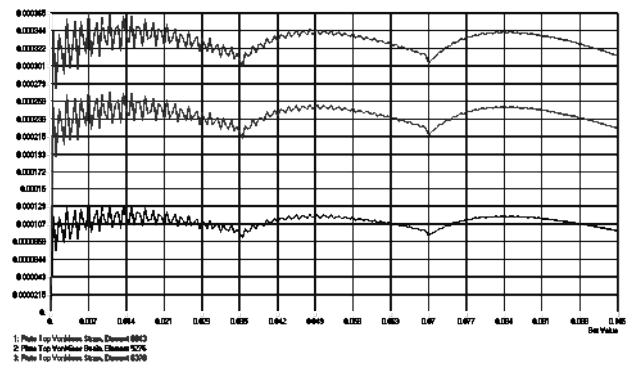


Fig.10. Strain distribution on cycloid gear (model 2) for finite elements 8043, 5276, 6370

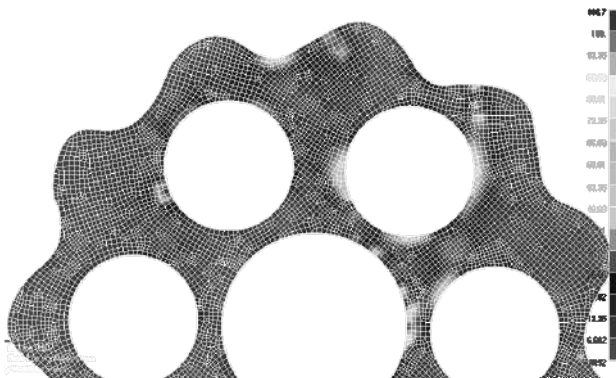


Fig.7. Von-Mises stress distribution on cycloid gear – model 2

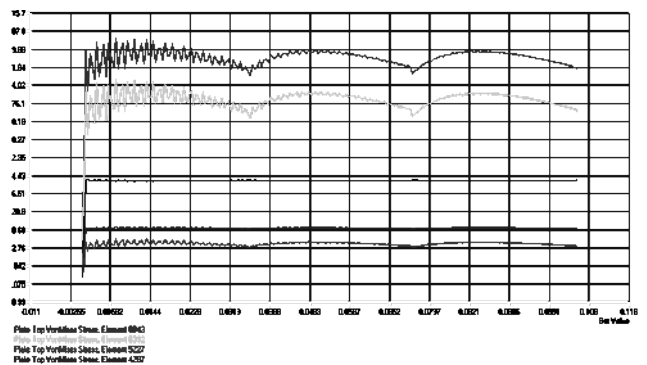


Fig.11. Von-Mises stress distribution on cycloid gear (model 3) for finite elements 8043, 6392, 5227, 4297 and 4527

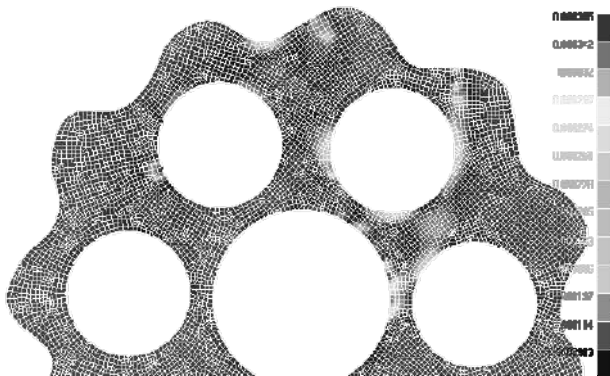


Fig.8. Strain distribution on cycloid gear – model 2

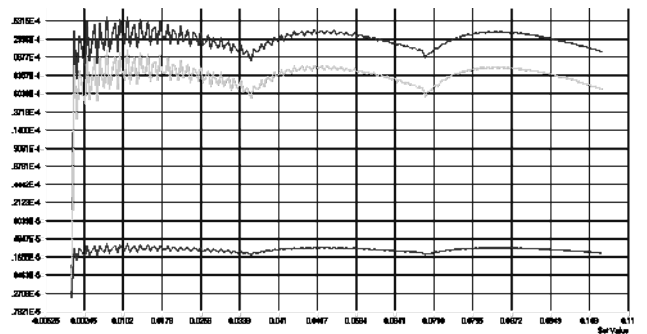


Fig.12. Strain distribution on cycloid gear (model 3) for finite elements 8043, 6392 and 5227

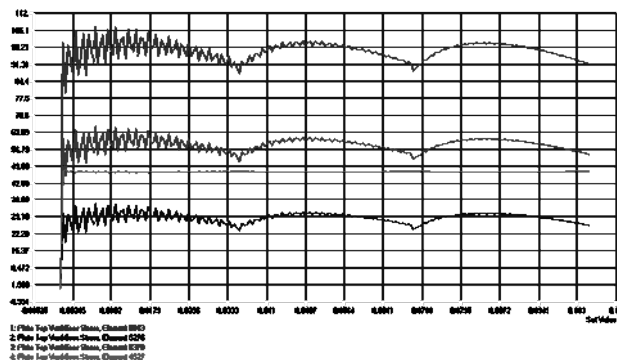


Fig.9. Von-Mises stress distribution on cycloid gear (model 2) for finite elements 8043, 5276, 6370, 4527

The maximum values Von-Mises stress and strain for all three analysed models are given in Table 1.

Table 1. Maximum values of Von-Mises stress and strains

Model	Max. Von-Mises stress, MPa	Max. strain, mm
1	119,1	0,000434
2	106,7	0,000365
3	104,7	0,000358

Shown in Table 1, the maximum value of the Von-Mises stress and strain is when the cycloid gear has the single contact with output rollers, on the place of output roller 2 supports.

3. CONCLUSION

In the framework of this paper the analysis of stress-strain state is single-stage cycloid drive, in terms of dynamic loading using FE analysis, was carried out. A whole series of numerical models have made and a large number of simulations were done. Based on the obtained results the following conclusions can be made:

- The shape of the diagram distribution Von-Misses stress and strain selected nodes and finite elements is almost identical to the form of the corresponding excitation dynamic forces, especially in the zone of the same, [15],
- The maximum value of the stress and strain vary from the corresponding maximum values of stress and strain in case the fact of constant excitation force,
- Moving away from the working zone of dynamic excitation force, stress and strain amplitude is significantly reduced,
- The most critical case is the case of a single contact, which cycloid drives almost never is present,
- The values obtained stress and strain are within the allowed limits, which provide normal and reliable operation within a specified work life of power transmitter.
- The next step, which would confirm the above defined conclusions, would be the experimental analysis of the stress-strain state of cycloid gear in conditions of dynamic loads.

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USE OF SUN-AND-PLANET MECHANISM IN EDUCATIVE

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Abstract: In the process of education, laboratory and experimental work is clearly distinguished. In the sense the author presents original construction, as a model, of the sun-an-planet combinative mechanism with frictional elements which successfully replace a very expensive and original sun-and-planet gearing mechanism.

The proposed construction offers following advantages:

- It is easily worked out
- Low cost in comparison to the original one,
- Possibility of checking of the transfer of revolution ratio in comparison to the theoretical data,
- Possibility of receiving various combination of the transfer,

Easily handed and moved from one place to another because of its little weight (mechanical materials are wood and plastics), only some parts are made of iron and aluminium.

Key words: sun and planet mechanism, model of gear box

1. INTRODUCTION

In the process of education, laboratory and experimental work is clearly distinguished. The investigation of the sun-an-planet has many benefiting:

- Possibility of checking of the transfer of revolution ratio during working period,
- Possibility of realization of very high transfer ratio (1:10000) when don't need to transfer power only reduction of number of revolutions,
- Have high of level of efficiency,
- Calm and balanced work, without noise and vibrations
- Uniform forces in the bearings, because of distribution on higher number of satellites,

- Can be free to rely on the working shaft without specific fundamental pillars and so can be exported as flying constructions
- Has small dimensions due to the very large forces which can be transmitted

Regarding the flaws of these notebooks are: high requirements in terms of technology as well as increased cost price compared to ordinary gear trains.

2. CALCULATION

In structurally planetary gear units can be very different, and in this regard will be discussed here. We will take into consideration the planetary gear train is shown in Figure 1, in which seven combinations may occur, or seven different transmission ratios or numbers of revolutions of the output shaft, depending on which of the elements of the drive, driven and fixed part. Table 1 shows the seven combinations with appropriate gear ratios (i).

$$i_{1,2} = n_1 / n_2 = -z_2 / z_1 \quad (1)$$

Negative sign is due to opposite of rotation directions the drive and driven gears.

Calculation of this relationship (1) an ordinary (classical) pinion gear to the outside pair is:

$$i_{1,2} = n_1 / n_2 = z_2 / z_1 \quad (1a)$$

Positive sign is due to same of rotation directions of the drive and driven gears.

When the planetary gear train for transmission lever (P), both the number of revolutions (n_1 and n_2) are reduced to the speed of the transmission lever (n_p), so that the relative reduction ratio as follows:

$$i_{1,2}^P = \frac{n_1 - n_p}{n_2 - n_p} = -\frac{z_2}{z_1} \quad (2)$$

For the external gear pair, ie:

$$i_{1,2}^P = \frac{n_1 - n_p}{n_2 - n_p} = \frac{z_2}{z_1} \quad (3)$$

For the internal gear pair.

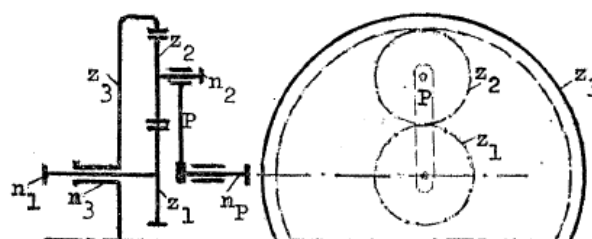


Fig.1. The basic form of the planetary gear carrier which will serve as a basis for the development of a model

The planetary gear train, shown in Figure 1, the gear z_2 (satellite) is inserted between the gear z_1 and z_3 does not affect the transmission ratio only to the direction of rotation of the gear z_1 and z_3 . If the gear z_3 is stagnant (see

the combination 1 in Table 1), then the general formula for the relative transmission ratio is:

$$i_{1,3}^P = \frac{n_1 - n_P}{n_3 - n_P} = -\frac{z_3}{z_1} \quad (4)$$

Insert the pinion gear z_2 between z_1 and z_3 does not affect the transmission ratio, but only to change the direction of rotation (see negative sign $-z_3/z_1$).

Rolled ($n_3 - n_P$) on the right side of the above equation is obtained:

$$n_1 - n_P = (n_3 - n_P)(-z_3 / z_1) = -n_3 \frac{z_3}{z_1} + n_P \frac{z_3}{z_1} \quad (5)$$

By switching right-wing members of the above equation to the left gives:

$$n_1 - n_2 - n_P \frac{z_3}{z_1} + n_3 \frac{z_3}{z_1} = 0 \text{ or:} \quad (6)$$

$$n_1 - n_P(1 + \frac{z_3}{z_1}) + n_3 \frac{z_3}{z_1} = 0 \quad (7)$$

How it is mentioned above that the gear z_3 motionless, in this case, and his speed n_3 be zero, so will now be:

$$n_1 - n_P(1 + \frac{z_3}{z_1}) = 0 \quad (8)$$

here is $i_{1,P} = n_1 / n_P$:

and $i_{1,P} = n_1 / n_P = 1 + z_3 / z_1$ corresponding to the written values in Table 1 for the combination 1.

If the gear z_1 is motionless, then start from the gear z_3 , or the number of his speed n_3 , so we're going to gear z_1 , and his number of revolutions n_1 , and the relative gear ratio. By switching to the right side of the above equation is obtained. By switching both members with the right to the left side of the equation is obtained:

$$i_{3,1}^P = \frac{n_3 - n_P}{n_1 - n_P} = -\frac{z_1}{z_3} \quad (9)$$

By switching to the right side of the above equation is obtained:

$$n_3 - n_P = (n_1 - n_P)(-\frac{z_1}{z_3}) = -n_1 \frac{z_1}{z_3} + n_P \frac{z_1}{z_3} \quad (10)$$

By switching both members with the right to the left side of the equation is obtained:

$$n_3 - n_P - n_P \frac{z_1}{z_3} + n_1 \frac{z_1}{z_3} = 0 \text{ or:} \quad (11)$$

$$n_3 - n_P(1 + \frac{z_1}{z_3}) + n_1 \frac{z_1}{z_3} = 0 \quad (12)$$

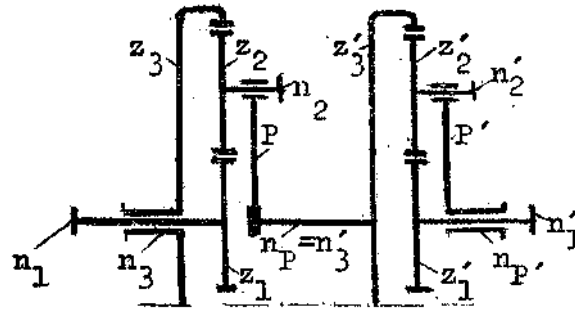
As the gear z_1 motionless to his speed n_1 is equal to zero, we have:

$$n_3 - n_P(1 + \frac{z_1}{z_3}) = 0 \quad (13)$$

here is the $i_{3,P} = n_3 / n_P$:

$i_{3,P} = n_3 / n_P = 1 + z_3 / z_1$ which corresponds to the value written in the Table 1 for the second combination.

From Table 1 it is evident that the planetary gear box can get 7 different combinations, and gear ratios, and thus seven different speeds driven work, which means that there are two coupled planetary gear trains, in Figure 2, this gives $7 \times 7 = 49$ combination.



(9) Fig.2. The combined planetary gear gearbox consists of two planetary gears shown in Figure 1, connected in serie

Table 1.

No of combination	Drive part	Driven part	Fixed part		Gearshift
1	z_1	P	z_3	$i_{1,P} = n_1 / n_P = 1 + z_3 / z_1$	$2 < i < \infty$ (I reduction)
2	z_3	P	z_1	$i_{3,P} = n_3 / n_P = 1 + z_1 / z_3$	$1 < i < 2$ (II reduction)
3	P	z_3	z_1	$i_{P,3} = n_P / n_3 = 1 / (1 + z_1 / z_3)$	$0,5 < i < 1$ (I acceleration)
4	P	z_1	z_3	$i_{P,1} = n_P / n_1 = 1 / (1 + z_3 / z_1)$	$0 < i < 0,5$ (II acceleration)
5	z_1, z_3, P	z_1, z_3, P	$z_1 \text{ i } z_3$ $z_1 \text{ i } P$ $z_3 \text{ i } P$	$i_{1,3} = n_1 / n_3 = 1$	$i = 1$ (direct)
6	z_1	z_3	P	$i_{1,3} = n_1 / n_3 = -z_3 / z_1$	$-\infty < i < -1$ (I speed "back")
7	z_3	z_1	P	$i_{3,1} = n_3 / n_1 = -z_1 / z_3$	$-1 < i < 0$ (II speed "back")

These wide possibilities give a huge advantage over traditional planetary, ordinary gear drives. Now it can be taken that in one particular case, we see the value of the transmission ratio with combination of planetary gear train in Figure 2.

3. RESULTS AND ANALYZE

3.1. Example

Combined planetary gear train, shown schematically in Figure 2, the following information: $z_1 = 25$ (teeth); $z_2 = 14$; $z_3 = 53$; $z_1' = 25$; $z_2' = 14$; $z_3' = 53$; normal module $m_n = 4$ mm; $n_1 = 1000$ r / min, determine the speed of the driven shaft of the transmission lever P (n_p) when the gears z_1 and z_3 are immobile.

3.2. Solution

Since the combined planetary gear train in Figure 2 consists of two identical planetary gear trains, shown in Figure 1, connected in series, we will first determine the

$i_{1,p}$ for the left half of the laptop, and then we'll determine $i_{3,p'}$ for the right half laptop.

$$i_{1,p} = n_1 / n_p = 1 + z_3 / z_1 = 1 + 53/25 = 1 + 2.12 = 3.12$$

here is:

$$n_p = n_1 / i_{1,p} = 1000 / 3.12 = 320.5 \text{ o / min}$$

(see the combination no 1)

From Figure 2 is visible that $n_p = n_3'$ (lever shaft P is also a shaft gear z_3').

Transmission ratio of the right half of the notebook will be: (see combination no 2)

$$i_{3',p'} = n_3' / n_{p'} = 1 + z_1' / z_3' = 1 + 25/53 =$$

$$= 1 + 0.472 = 1.472$$

Here is:

$$n_{p'} = n_3' / i_{3',p'} = 320,5 / 1,472 = 217,73 \text{ o/min.}$$

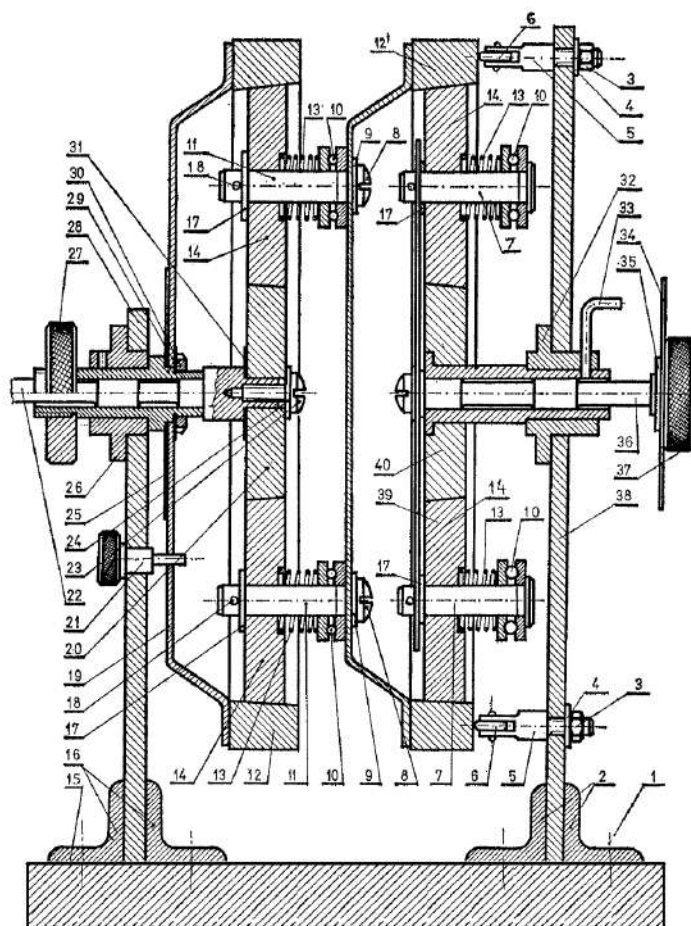


Fig.3. Model of combined planetary gear box

To these theoretical calculations could be clearly presented in the planetary gear train, which is also one of the important didactic and methodological principles, the author of the article is constructed planetary gear in which the gears are replaced by friction car (wheels), thereby enabling very light production, in principle both this notebook will have the same transmission characteristics

as the planetary gear train, because the friction gear does not transfer any forces that would cause some discrepancies transmission ratio with respect to the possibility of mutual sliding friction wheels.

Engineered model combined planetary pinion can be made, or made of wood (plywood), plastic materials, or aluminium or similar alloy (alloy). Production of such

materials does not require any effort even higher power machine tools, especially tools even with greater durability.

It is taken into account the cost of the original planetary gear pinion relative to its model, then it can be seen what a huge savings in the cost, so that the model not only for the much lower cost price was acceptable, but also because of another model has the advantage that compared to the original planetary gear train, and it is the simplicity and transparency of the movement of all its parts, because it is completely open structure.

It should be noted that the production of gear with the internal gear much more difficult and requires special machine tools, while the predicted model replaces the ordinary gear rings that are processed on a conventional lathe quickly and efficiently.

All theoretical calculations can be immediately tested experimentally on the model, so it will be measured for one revolution of the drive part of distance travelled angle of the driven work (on cardboard protractor). Measuring the angle of rotation of some parts of the possibility that they can determine the transmission ratios, as applicable following equation:

$$i_{1,2}=n_1/n_2=w_1/w_2=\varphi_1/\varphi_2=z_2/z_1=d_2/d_1.$$

When was creating the model adopted was the following values:

$$m_n=3 \text{ mm}; z_1=z_1'=21; z_2=z_2'=20; z_3=z_3'=61.$$

The diameters of the friction rounds are determined based on the obtained diameter of gear:

$$d_{o1}=d_{o1}'=m_n z_1=3 \times 21=63 \text{ mm};$$

$$d_{o2}=d_{o2}'=m_n z_2=3 \times 20=60 \text{ mm};$$

$$d_{o3}=d_{o3}'=m_n z_3=3 \times 61=183 \text{ mm};$$

On the model it is possible to jump to the desired parts so that they can get the desired combination, and students can participate in both the theoretical study and the experimental checking of the calculated gear ratios, i.e. speeds or angles.

4. CONCLUSION

If it is taken into account the cost of the original planetary pinion relative to its model, then it can be seen what a huge savings are in the cost, so that the model not only for the much lower cost price was acceptable, but also because of another model has the advantage that compared to the original planetary gear train, and it is the simplicity and transparency of the movement of all its parts, because it is completely open structure.

It should be as well noted that the production of gear with the internal gear much more difficult and requires special machine tools, while the predicted model replaces the ordinary gear rings that are processed on a conventional lathe quickly and efficiently.

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EFFICIENCY AS AN EXPRESSION OF PLANETARY GEAR TRAIN ENERGY LOSSES

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Abstract: During mechanical energy transfer through the transmission and simultaneous conversion its parameters inside the transmission, some mechanical energy losses are appeared. A part of mechanical energy is transformed into the heat energy. In order to characterize quality of energy parameters transformation, the term of efficiency is introduced. Function of efficiency includes a few of important factors that have influence to the energy transformation efficiency in the transmission. Paper deals with the power losses and efficiency of the planetary gear trains. The subject of the paper is basic type of planetary gear transmissions.

Key words: efficiency, energy losses, planetary gear transmissions

1. INTRODUCTION

There is no transmission which works without losses. During mechanical energy transfer through the transmission and simultaneous conversion its parameters inside the transmission, some mechanical energy losses are appeared. A part of mechanical energy is transformed into the heat energy and warms the parts of transmissions. In order to characterize quality of energy parameters transformation, the term of efficiency is introduced. By this parameter the transmission ability for mechanical energy keeping is evaluated. Since that, efficiency is important criterion for the design and evaluation of the construction quality.

There are significant papers dealing with the transmission heating due to power losses presented by efficiency. The review of considerable researches is given in [1]. The subject of this paper is planetary gear transmission. Planetary gear transmission (PGT) is a type of gear trains with many advantages. Since that their application is increasing in mechanical engineering and conveyor

systems as single-stage and multi-stage. Multi-stage planetary gear transmission is obtained from single stage gear trains by linking one or two planetary units' shafts. This paper provides analyzing of power losses and efficiency of planetary gear transmissions

2. PLANETARY GEAR TRANSMISSION POWER LOSSES

Planetary gear transmission power losses, as well as power losses of all gear transmissions, depend on transmission design and work conditions.

The energy losses appear due to friction between the parts in motion. They appear in the gear contact, bearings, oil, sealings etc.

Power losses in the gear contact and power losses in bearings depend on transmission load, but power losses due to oil viscosity and ventilation losses don't depend on transmission load and they are called "idle losses" [2]. Idle losses are experimentally determined for each individual transmission.

[3].

At gear power transmissions, including planetary gear transmissions, it could be accepted that the power losses in the gear contacts is the basic part of power losses.[4]. At planetary gear trains with uniformly distributed satellites, power losses in the central member bearings can be avoided [5].

Unlike the other transmissions, power losses at PGT due to oil bath are considerable. Since that, gears do not soak in oil. Lubrication is carried out by spraying lubricating oil under pressure.

Power losses are heat sources in the planetary gear transmission. The reasons for energy losses in the planetary gear transmission are shown in the Fig 1. Based on experimental researches at the set of two-stage transmissions, can be concluded that share of the energy losses in the gear contacts is 62-90% from the total energy losses. [7]. Lower values refer to the transmission units with the fast rotate gears. The experimental results of other authors [8] indicate similar results.

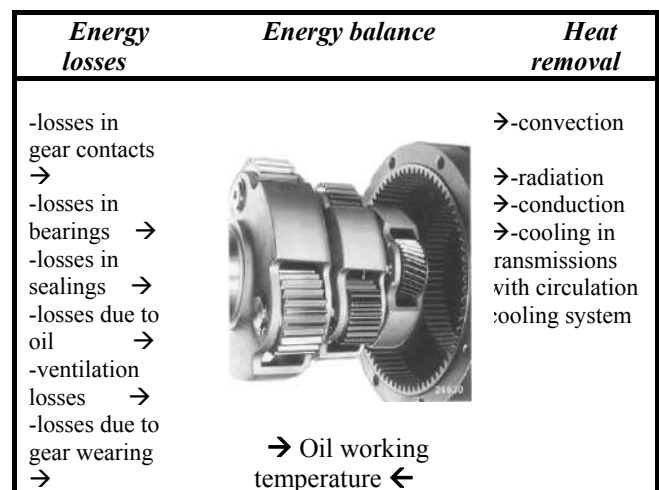


Fig.1. The types of planetary gear transmission energy losses and modes of heat removal

The current results in the study of power losses, heating and efficiency of planetary gear transmissions, were achieved using theoretical and theoretical-experimental methods. All of these methods consider mostly power losses as a result of relative motion of planetary gear transmission elements, especially losses in the gear contacts which are greater than other losses, while there are no results related to research of power losses due to transferred movement. Starting from the aforementioned facts about power losses and current results related to their analytical determination, it can be concluded that always only power losses in the gear contacts is determined because they can calculate for practical application. Bigger errors occur during calculation of power losses in bearings, while calculation of power losses due to oil is almost impossible and because of that experimental data is used. [9, 10]. Based on these researches it is evident that only power losses in the gear contact need to be considered in the mathematical model.

3. EFFICIENCY

Planetary gear transmissions efficiency implies which part of the input power is obtained at the output shaft, i.e. how much power losses during the power transmission process is lost. As a measure of that, efficiency η is used, like a value that multiplied input power.

They could be measured indirectly on made transmission under different working conditions. During the process of transmission design, one of the vital qualitative and quantitative criteria of transmission design is calculated efficiency. Calculated efficiency includes several key factors that affect the energy conversion efficiency.

Planetary gear trains have high efficiency. The reason for this is in the fact that power is transmitted through the PGT partly as a power in the gear contact (power of relative motion) and partly as a joining power (like coupling). Joining power is transmitted without losses. Efficiency of PGT could be slightly higher than efficiency of other types of gear transmission because of joining speed influence.

An especially important role during PGT efficiency determination has losses of PGT with immobile carrier, called basic efficiency, relative efficiency.

Efficiency of PGT in random mode can be expressed by basic efficiency. In [11, 12] can be found expressions for efficiency calculation adapted for different types of PGT and different modes as function of basic efficiency. It is assumed that basic efficiency does not depend on direction of energy flow, $\eta_{13(h)} = \eta_{31(h)} = \eta_0$ [10]. In the case of two degrees of freedom, efficiency changes into the wide interval depending on ratios of speeds of some members.

The procedure for determining expression for basic efficiency is indicated on the basic type of planetary gear transmission.

The basic type of a planetary gear transmission (PGT) is a design which has a central sun gear (external gearing - 1), central ring gear (internal gearing - 3), planets (satellites - 2) and carrier (h), shown in Fig. 2. Planets are in simultaneous contact with the sun gear and the ring gear. This type of a PGT is often used as single-stage

transmission, as a building block for higher compound planetary gear trains.

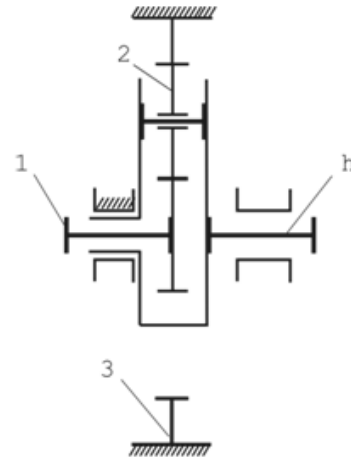


Fig.2. Basic type of planetary gear train (1- sun gear; 2 - planet; 3 - ring gear; h - carrier)

During calculation of basic efficiency different values for partly efficiency can be adopted depending on type of PGT, IT quality and losses in bearings.

The most often, real values is adopted: $\eta_0 = 0.97$ or $\eta_0 = 0.98$ for this type of PGT. These values are obtained by adopting value of efficiency of gear contacts (for both external and internal gear pairs) $\eta_{12} \approx \eta_{23} \approx 0.99$ and efficiency of bearings $\eta_L \approx 0.995$

The analytical expression for basic efficiency can be obtained by using losses coefficient ψ_0 .

$$\eta_0 = 1 - \psi_0 \quad (1)$$

For determining losses coefficient due to gear contact ψ_z , in order to make comparative analysis of variants, it is suitable to use equation (2) [11,12,13] and form (3). In (2) losses coefficient due to gear contact is expressed as a function of the number of teeth of the central sun gear z_1 , the number of teeth of planets z_2 and the number of teeth of the ring gear z_3 . In (3) losses coefficient due to gear contact is expressed as a function of the number of teeth of the central sun gear z_1 and ideal torque ratio t ,

$$t = \frac{|z_3|}{z_1}$$

$$\psi_z = 0.15 \cdot \left(\frac{1}{z_1} + \frac{1}{z_2} \right) + 0.2 \cdot \left(\frac{1}{z_2} - \frac{1}{z_3} \right) \quad (2)$$

$$\psi_z = \frac{t+1}{z_1 \cdot t \cdot (t-1)} (0.15 \cdot t + 0.2) \quad (3)$$

In (3) the number of teeth of the central sun gear z_1 is a discrete variable. Variable t is discrete variable too, since it represent ratio of the number of teeth of the ring gear and the number of teeth of the central sun gear. For graphic view of relation between efficiency η_z and z_1 and t , it is necessary to define the ranges of the variables. In

this paper next ranges are adopted [12]:

$$z_{1\min} = 12 \quad z_{1\max} = 28, \text{ and } t_{\min} = 2 \quad t_{\max} = 12.$$

By calculating according to:

$$\eta_z(z_1, t) = 1 - \psi_z(z_1, t) \quad (4)$$

Discrete set of computing values of basic efficiency considered only losses in gear contact is obtained. Values of variable t change from $t_{\min}$ to $t_{\max}$ with the changes of teeth of the ring gear of one.

The graphic view of this dependance is shown in Fig.3.

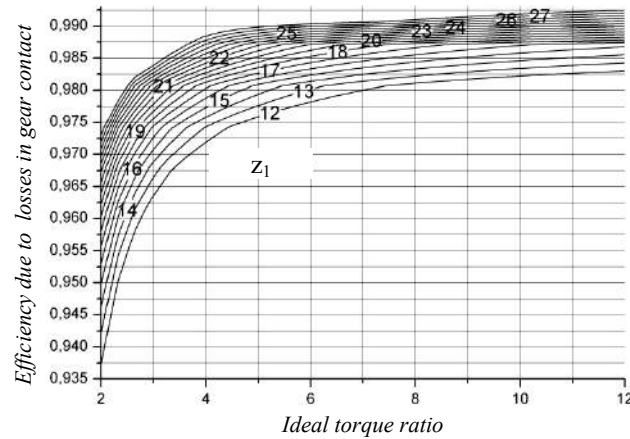


Fig.3. Dependance of computing values of basic efficiency considered only losses in gear contact from variables [12]

It can be noticed in what way losses are increasing and efficiency decreasing by decreasing of teeth of the gears. Also, it can be noticed in what way the change of ideal torque ratio influences basic efficiency. It can be seen that with constant number of teeth of the central sun gear z_1 , efficiency is increasing when the ideal torque ratio increasing, i.e. when the number of teeth of the ring gear z_3 increasing.

By adopting value of basic efficiency according [10], without considering the number of teeth, considerable inaccuracy could be obtained. For example, transmission with next characteristics: $t = 2$ and $z_1 = 12$ according diagram in Fig 3 has basic transmission $\eta_z = 0.9375$. If the value $\eta_0 = 0.97$ or $\eta_0 = 0.98$ is adopted, the significant error is made.

In order to establish mathematic-mechanical model that would considered energy losses in PGT, it is necessary to widen mention model by coefficient considering losses in bearings, losses due to oil and losses in sealings.

For accurately determination of these losses, the experimental results of PGT [7] can be used for widen model establishment.

According to these results [7], the coefficients k_B , k_S and k_C are obtained and the relation between coefficient of losses in gear contact ψ_z and coefficient for determination basic efficiency ψ_0 takes the form:

$$\psi_0 = (1 + k_B + k_S + k_C) \cdot \psi_z \quad (5)$$

The values of these coefficients are:

$k_B = 0,06 \div 0,07$ - coefficient which takes into account losses in satellites bearings;

$k_S = 0,09 \div 0,01$ - coefficient which takes into account losses in satellites sealings;

$k_C = 0,25 \div 0,02$ - coefficient which takes into account losses due to oil.

It is necessary to remark that lower values are related to planetary gear transmissions which gears work with bigger number of revolution, and bigger values are related to planetary gear transmissions that work with lower number of revolution. The widest interval has a coefficient k_C (coefficient due to oil).

On the other hand, starting from the general expression for efficiency determination as a ratio of output power and input power, and considering only losses in gear contact:

$$\eta_z = \frac{P_{out}}{P_{in}} = \frac{P_{in} - P_l}{P_{in}} = 1 - \frac{P_l}{P_{in}} \quad (6)$$

ratio of power losses to input power, i.e. a part of energy that convert to heat is obtained:

$$\frac{P_l}{P_{in}} = 1 - \eta_z \quad (7)$$

The graphic view of this dependance is shown in Fig. 4

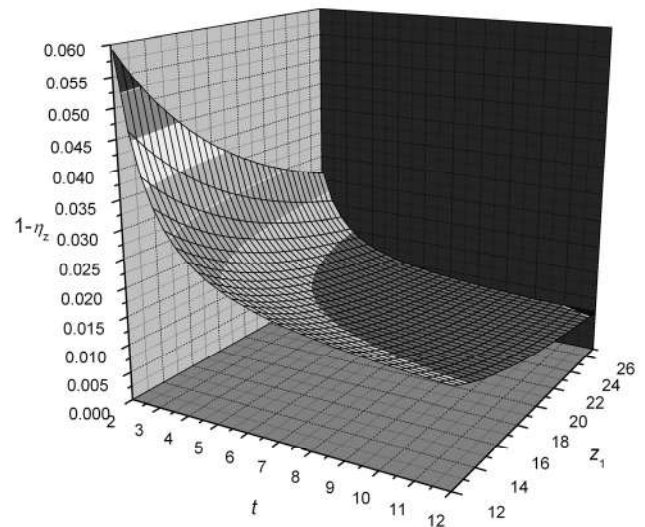


Fig.4. Ratio of power losses to total power

According graphic view can be noticed that when the number of teeth of sun gear z_1 and ideal torque ratio t increasing, efficiency also increasing (as it is shown in Fig.3), difference $(1 - \eta_z)$ is decreasing which means that power losses is decreasing and heating of transmission decreasing too.

By increasing the number of teeth of gears, great dimensions of gears are obtained (diameter and width). This leads to a reduction of surface pressure and the coefficient of friction (because of sliding speed increasing). On the other hand increasing gear dimensions leads to decreasing of gear load. It is obviously that in this case losses due to sliding friction will be decreasing, i.e. efficiency will be increasing.

It can be concluded that optimal design with max efficiency as objective function would be obtained for very great values of variables which define gear dimensions (the highest values that allow constraints) [11,14]. This corresponds with the conclusions that greater values of efficiency require transmission with greater dimension [11].

4. CONCLUSION

Mechanical transmission which works without losses does not exist. During mechanical energy transfer through the transmission, some mechanical power losses have to appear. These power losses heat the transmission.

Transmission efficiency is an expression of these losses. Efficiency is one of the most important criteria for the design and evaluation of the construction quality. By this parameter the transmission ability for mechanical energy keeping is evaluated.

The basic type of PGT is analyzed regarding power losses and efficiency in this paper. First of all, the types of planetary gear transmission energy losses are introduced. Then, efficiency is described as a function of gear teeth and ideal torque ratio. Beside mathematical relation, the graphic view is introduced, too. In the first step only power losses in gear contact is considered. In the following step the model is widened, by introducing the relation that consider coefficients of losses in oil, bearings and sealings.

Starting from the general expression for efficiency, ratio of power losses to input power, i.e. a part of energy that convert to heat is obtained expressed by efficiency. Dependence power losses to total power indicates the similar conclusion as a dependence efficiency from defined variables that optimal design with max efficiency as objective function would be obtained for transmission with greater dimension. Since that, efficiency as an only criterion for optimization is seldom used.

Based on the numerical example it can be concluded that a significant error is occurred when basic efficiency is adopted, without considering the number of teeth.

This approach of efficiency of PGT researching can be successfully used for basic PGT type.

Furthermore, this approach indicates a possibility for application to other planetary gear transmission types and other gear transmission overall.

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LOAD CAPACITY OF CYLINDRICAL WORM GEARS ACCORDING TO DIN 3996-2012

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Abstract: A worm drive is widely used in power and movement transmission, and the proper functioning of the entire machine system depends on it. The advantages of a worm drive, compared to other types of drives, are primarily reflected in its compact design, large transmission ratio, reliability, the possibility of vibration damping, as well as in structural advantages regarding branching and summarising energy, etc. Its capacity is limited by the emergence of a larger number of boundary conditions when in exploitation. This paper considers load capacity calculations of worm gears according to DIN 3996-2012.

Key words: worm gear, load capacity, DIN 3996

1. INTRODUCTION

Worm gears are widely used in tool machines, transport equipment, in vehicles, primarily for power transmission, as well as in fine-tuning and precision devices for movement transmission.

Worm pairs load capacity is nowadays based on the DIN 3996 [1], which was introduced with dimensionless physical parameters, pressure parameter p_m^* , oil film thickness parameter h^* and mean slip path parameter s^* . Current information regarding the load capacity of worm pairs use the median surface pressure on flank σ_{Hlim} of the output torque T_2 as authoritative criteria for the calculation of the flank load capacity. A prerequisite for a correct estimation of worm gear flanks load capacity regards to wear is the accurate knowledge of local flank teeth strain in different coupling conditions.

In the case of worm gear drive the absence of flank teeth wear has almost never been recorded. Research studies [2] show that worm pairs mostly operate under conditions of mixed friction. This indicates that tooth flank wear is

not determined according to the hydrodynamic lubrication theory, but according to Hertz's theory, i.e it depends on the ratio between the radius curvature and the characteristics of the materials in contact.

The research in papers [4], [5] and [6] shows that surface pressure and temperature changes in the contact zone of coupled pairs have a major impact on worm gear wear.

2. BOUNDARY CONDITIONS OF WORM GEARS

In modern construction solutions it is common to use worm gears made of tin bronze and worms made of hardened and whetted steel (Fig. 1). Worm gears are affected by the following boundary conditions:

- Destruction of tooth flanks under pitting,
- A tooth flank wear,
- Tooth breakage,
- Worming, mining and changing thermal stability of gear transmission,
- Worm shaft bending.

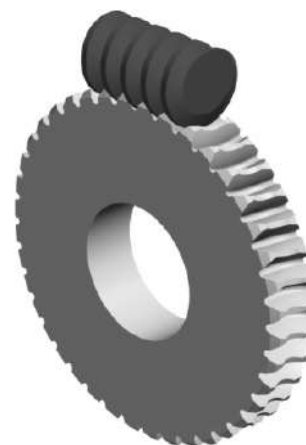


Fig.1. Standard worm gear

2.1 Pitting resistance

Pitting is, in effect, the destruction of a tooth surface (shaped like small holes and dimples) as a result of huge surface pressures and dynamic stress wear.

One can observe the difference between the initial and advanced stages of pitting. The initial stage of pitting is the result of the first phase of gears' transmission. It has a positive effect because it results in a uniform stress distribution and a better contact pattern. When working conditions are unchanged, this type of pitting decreases over time, i.e. the damaging process is digressive. Medium and highly loaded worm gear pairs can be affected by advanced pitting and this constitutes a huge problem. This sort of pitting has a progressive character, so much so that destroyed surfaces are larger while at the same time the contact pattern is smaller. Consequently, the damaged surface keeps expanding continuously and changes in the shape of tooth surface occur. This results in a sudden tooth surface wear, and finally in a tooth breakage.

According to the DIN 3996 [1], the ratio between the limit value of contact stress σ_{Hkr} and mean contact stress σ_{Hm} represents a pitting safety:

$$S_H = \frac{\sigma_{Hkr}}{\sigma_{Hm}} \geq S_{H \min} = 1 \quad (1)$$

2.2. Wear load capacity

During the period of mining wear, the continual tooth material wear width is smaller. In the first phase, the wear has a positive effect because one material rubs against another, thus allowing shapes of tooth surfaces to accommodate and consequently prevent any further wear. However, in the cases of high intensity rubbing, the wear resistance can be a criterion for a working period. The wear basically depends on working criteria. Changing shaft rotation conditions and the working mode, as well as terming the transmission gear on and off, result in increased wear. By using hard and quality worm tooth flanks (case carburising, whetting, polishing) one can reduce wear.

According to the DIN 3996 [1], the wear safety S_W is calculated on the basis of the permissible wear $\delta_{W \lim}$ and abrasive wear in the normal section δ_{Wn} , according to the following equation (2).

$$S_W = \frac{\delta_{W \lim n}}{\delta_{Wn}} \geq S_{W \lim} = 1,1 \quad (2)$$

2.3. Root strength of teeth

Worm gear pairs have dangerous working loads only in worm gear tooth root, considering that they are less resistant and less durable in comparison to worm teeth. In the worm gear tooth root there is a complicated load, and the dominant stresses are shear and bending.

Worm gear tooth breakage is very rare. The most common causes lie in striking overloads, when loads appear larger than the static strength of the material. The wear is also a significant cause of tooth breakage, since wear diminishes its cross-section.

According to the DIN 3996 [1], the safety of the teeth root strength S_F is calculated according to the following equation (3):

$$S_F = \frac{\tau_{Fkr}}{\tau_F} \geq S_{F \lim} = 1,1 \quad (3)$$

where:

τ_{Fkr} - permissible shear stress,

τ_F - shear stress.

2.4. Thermal stability

When designing a gearbox, one should also consider the heat generated inside the gearbox (efficiency of gearing, friction of bearings, and friction in sealing). This parameter is not so important in spur or bevel gearing, but it is in worm gearing. As worm gearing efficiency is considerably lower than that of spur or bevel gearing, far more heat is generated in the gearing process and has to be removed. Therefore, thermal safety has great importance for a correct design, to ensure gearbox function within the permitted temperature range of oil. Thermal design/safety tends to be one of the limiting factors when designing transmissions.

According to the DIN 3996 [1], thermal safety S_T is calculated according to the following equation (4):

$$S_T = \frac{g_{S \lim}}{g_S} \geq S_{T \lim} = 1,1 \quad (4)$$

where

$g_{S \lim}$ - limit value of oil temperature

g_S - oil temperature

2.5. Worm shaft deflection

A worm shaft is loaded by radial, axial and tangential force, which leads to considerable bending of the shaft, considering the relatively large distance between bearings. The shaft deflects due to bending which can lead to contact interference. The tangential force depends on friction against the teeth flanks. Thus, in the event of insufficient lubricating, the friction force increases, which results in a considerable deflection of a worm shaft. When loads and revolutions per minute change, the contact between tooth flanks decreases, as well as the local overload of the aforementioned. This can also result in a deflection of a worm shaft.

Monitoring of the work correctness is based on a control of the contact line between a tooth surface and on a control of a transmission gear thermal stability.

According to the DIN 3996 [1], a worm deflection safety S_δ is calculated according to the following equation (5):

$$S_\delta = \frac{\delta_{\lim}}{\delta_m} \geq S_{\delta \min} = 1 \quad (5)$$

where:

δ_m - a worm shaft deflection,

$\delta_{\lim}$ - a permissible worm shaft deflection.

3. LOAD CAPACITY OF WORM GEARS ACCORDING TO DIN 3996/2012

An analysis of a worm gear load capacity is made for the family of gear drives with the characteristics given in Table 1. The calculation of a worm gear load capacity according to the DIN 3996 [1] has been performed on the basis of 4 criteria:

- Pitting resistance,
- Wear load capacity,
- Root strength of teeth and
- Thermal stability.

For the purposes of a comprehensive observation of worm gears load capacity according to different criteria, the following parameters were varied:

- Centre distance ($a = 63\text{mm}$; $a = 100\text{mm}$; $a = 250\text{mm}$)
- Transmission ratio ($i = 10$; $i = 20$; $i = 30$; $i = 40$)
- Input speed ($n_I = 200 \text{ min}^{-1}$; $n_I = 500 \text{ min}^{-1}$; $n_I = 1000 \text{ min}^{-1}$; $n_I = 1500 \text{ min}^{-1}$; $n_I = 2000 \text{ min}^{-1}$);
- Wheel material (CuSn12-C-GZ; CuAl10Fe5Ni5-C-GZ; EN-GJS-400-15);
- Oil (mineral, synthetic);
- Lubrication (splash, injection).

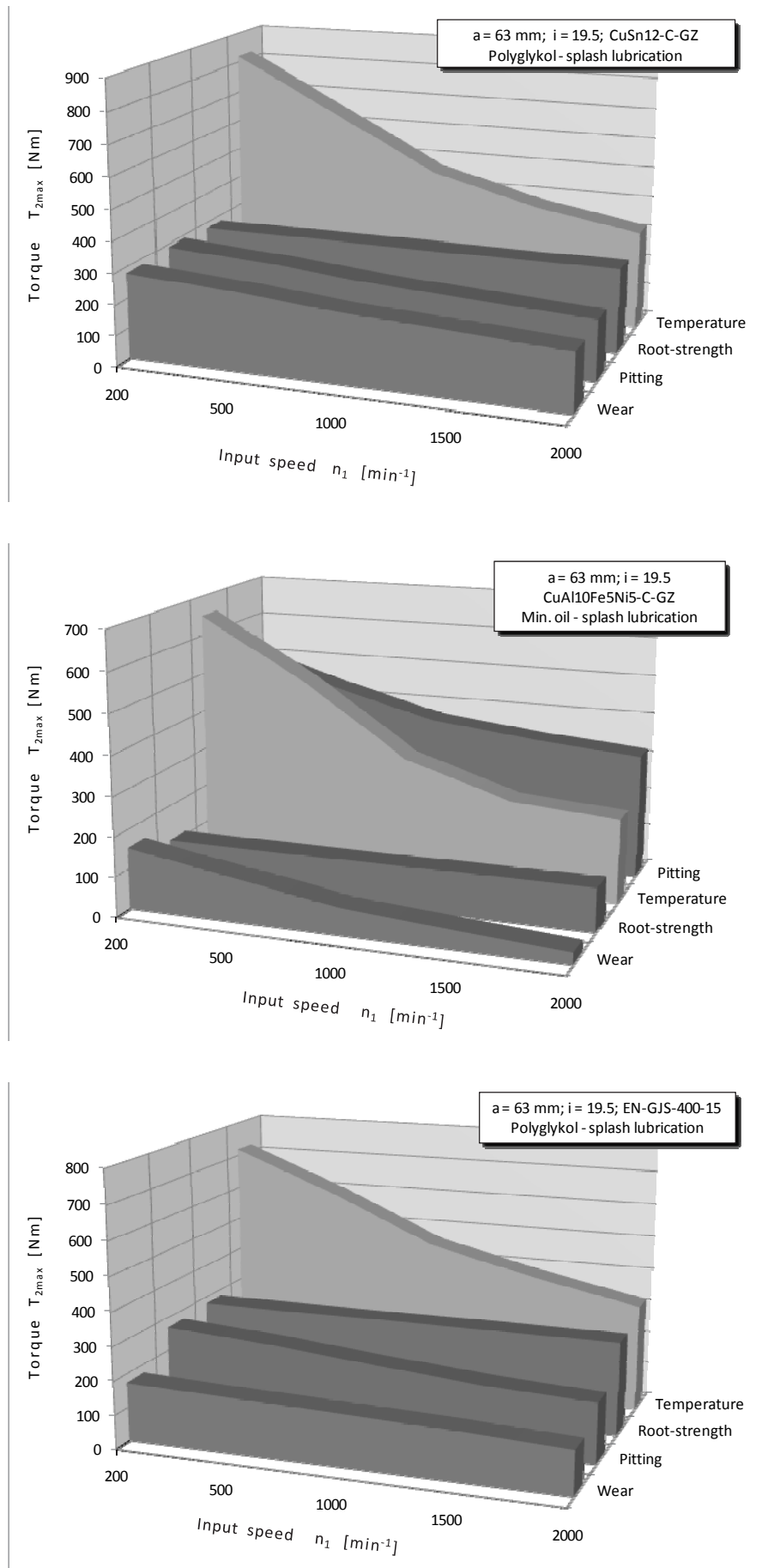


Fig.2. Comparison of worm gear load capacity for different values of input speed n_1 and different materials of the worm gear

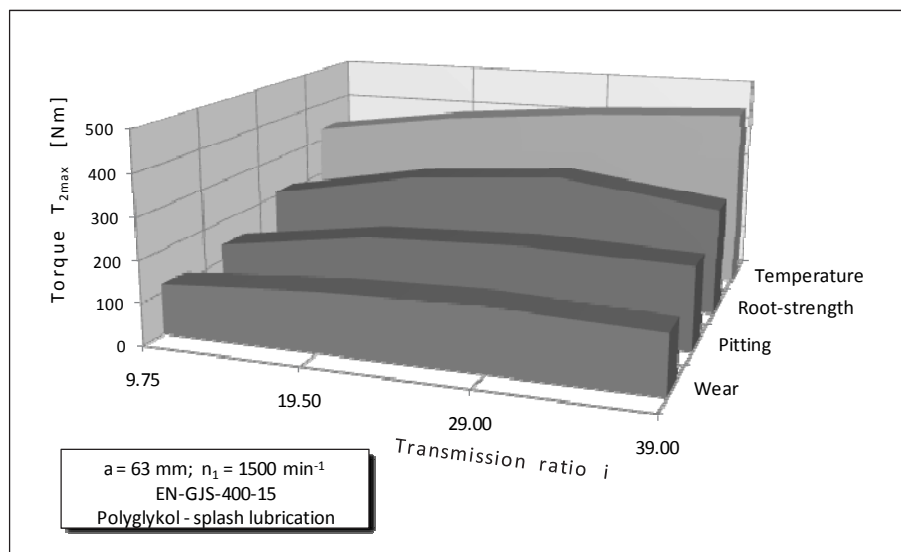
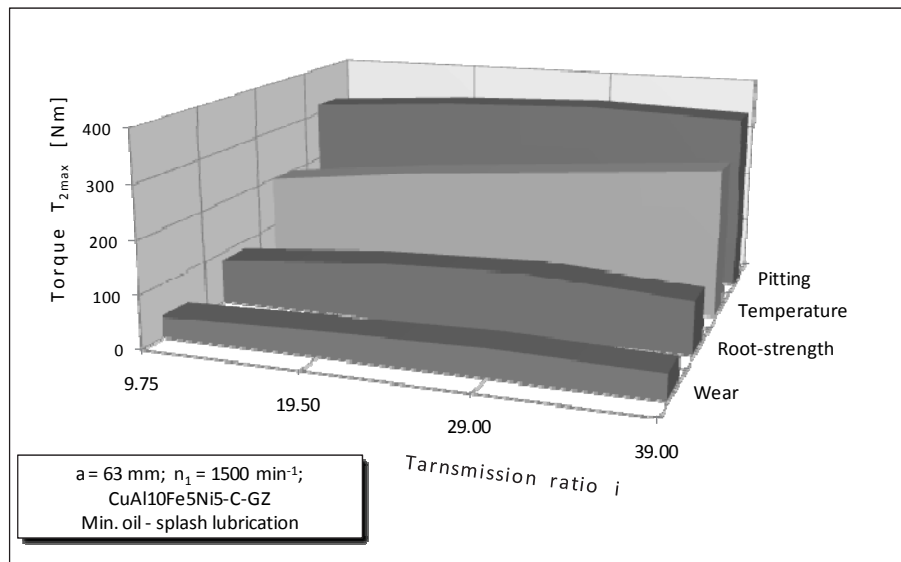
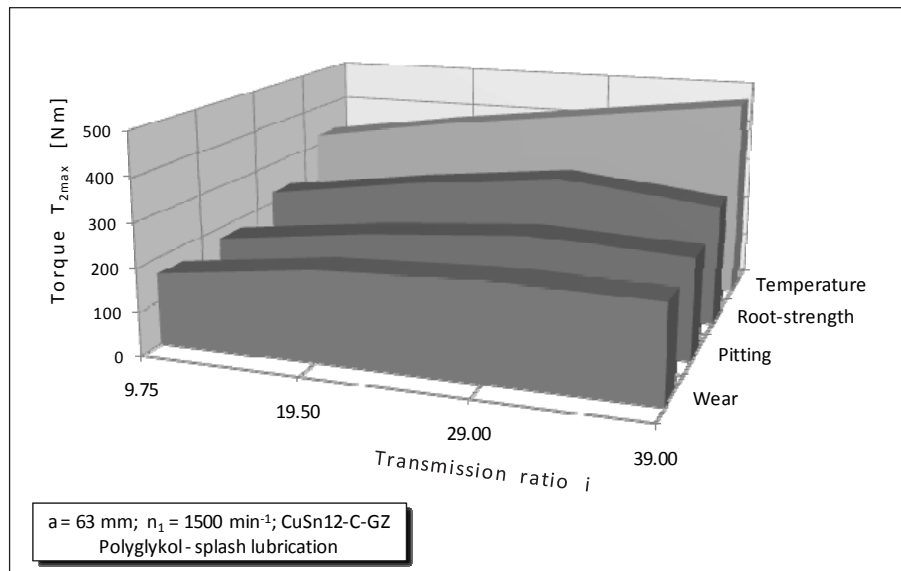


Fig.3. Comparison of worm gear load capacity for different values of transmission ratio i and different materials of the worm gear

Table 1. Transmission gear data

Geometrical size	Values
Centre distance a [mm]	63, 100, 250
Worm gears type	ZI
Transmission ratio i	10, 20, 30, 40
Modul m_x [mm]	2.5, 3.15, 4, 5, 10, 12.5
Lifetime L_h [h]	10000
Number of teeth z_1	1, 2, 3, 4
Number of teeth z_2	20, 30, 40
Wheel material	CuSn12-C-GZ; CuAl10Fe5Ni5-C-GZ; EN-GJS-400-15
Worm material	16MnCr5
Input speed [min^{-1}]	200, 500, 1000, 1500, 2000
Synthetic oil (Polyglykol)	$v_{40} = 220 \text{ mm}^2/\text{s}$; $v_{100} = 41 \text{ mm}^2/\text{s}$
Mineral oil	$v_{40} = 674 \text{ mm}^2/\text{s}$; $v_{100} = 36.9 \text{ mm}^2/\text{s}$

The critical thickness of the weared layer has been determined according to parietal thickness of top of the weared layer.

For each calculation the maximum output torque T_{2max} was obtained for the required safety. The calculation results of the load capacity for centre distance $a = 63$ mm and different materials of the worm gear are shown in Figures 2 and 3. Similar diagrams have also been made for the centre distance $a = 100$ mm and $a = 250$ mm.

Bearing in mind the variation of transmission ratio, speed, worm gears material and centre distance, the analysis required the calculation of 81 gear pairs.

The gear characteristics which are most widely used in practice have been chosen. The boundary condition of a gear is determined by the smallest load capacity. The results of the analysis have been presented in Table 2. Figure 4 presents the results of the analysis by means of a column chart diagram. The diagram shows that the largest worm gear load capacity limitations are related to worm pair flank wear. (45). The second column represents pitting related limitations (36), and the third column represents thermal stability related limitations (18).

One should, however, bear in mind the interconnection between various forms of damage, first and foremost between wear and pitting [7]. Accordingly, the increase in load capacity in comparison to wear significantly affects the load capacity of the entire transmission gear. The same observation applies to thermal stability of transmission gears.

Table 2. Boundary conditions of worm gears

No	Wheel mater.	i	n [min^{-1}]	Centre distance a [mm]		
				63	100	250
1.	Tin bronze	10		W (P) UN	P UN	T, P, NUN
2.		20				
3.		30				P, NUN
4.		40				
5.			200	W, P UN	W UN	P, NUN
6.			500			
7.			1000		P UN	T, NUN
8.			1500			
9.			2000			
10.	Aluminium bronze	10		W NUN	W NUN	T, W, NUN
11.		20				
12.		30				R, NUN
13.		40				
14.			200	R NUN	R NUN	W, NUN
15.			500			
16.			1000	W NUN	W NUN	W, T, NUN
17.			1500			
18.			2000			
19.	Cast iron materials	10		W $\approx$ UN	P, W $\approx$ UN	T, P, NUN
20.		20				
21.		30				P, NUN
22.		40				
23.			200	W UN	P, W $\approx$ UN	P, T, NUN
24.			500			
25.			1000			T, NUN
26.			1500			
27.			2000			

W – wear (45); P –pitting (36); R –root-strength (5);
T – thermal stability (18); UN – uniform load capacity;
NUN – uneven load capacity

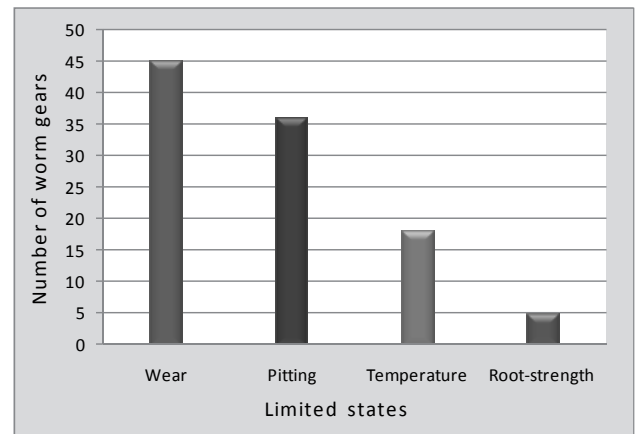


Fig.4. Comparison of boundary conditions of 81 worm gear pairs

From the viewpoint of utilisation of available worm gear transmissions, it is extremely important that worm gear has a similar capacity for all boundary conditions. In the analysis it has been marked as uniform (UN - uniform load capacity), i.e. uneven load capacity (UN – uneven load capacity). Out of 81 gears only 23 gears exhibit uniform load capacity, in 13 gears the load capacity is approximately uniform, while in 45 gears the load capacity is uneven. The best utilisation of available worm

gear resources has been obtained in a gear with a tin bronze pinion.

4. CONCLUSION

Based on this display we can draw the following conclusions:

1. The analysis of worm gear load capacity according to the DIN 3996-2012, along with the variations of the crucial construction parameters (transmission ratio, speed, worm gear material, center distance), shows that their capacity is mainly limited as a result of damage due to wear, pitting and increased operating temperature. Out of 81 analysed gears, 45 gears exhibit critical wear, 36 pitting and 18 increased operating temperature.
2. From the viewpoint of utilisation of worm gear available resources, out of 81 analysed gears, 23 gears exhibit uniform load capacity, in 13 gears the load capacity is approximately uniform, while in 45 gears the load capacity is uneven. The best utilisation of available worm gear resources has been obtained in a gear with a tin bronze pinion.
3. The analysis was performed on worm gears which are most widely used. Thus, the obtained results are of significance for engineering practice.

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ANALYSIS OF SELECTION PROCEDURES OF UNIVERSAL WORM GEAR UNITS

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Abstract: The purpose of this paper is to point out the usual methods of determining the service factor, which is the most influential parameter. In order to provide the proper selection of the size of the worm gear reducer, manufacturers are trying to take detailed look at all the factors which influence the work gear and to provide customers with the best possible solution. The greatest impact on their defining has a safety factor which is used to calculate the gear unit and the effect of an ambient temperature ie. cooling method.

Key words: worm gear units, selection procedures, safety factor

1. INTRODUCTION

It is undisputed that the use of universal worm gears and worm gears in general is reduced to a minimum, because of relatively large losses that occur in them, in addition to a number of advantages they have. They are most commonly used only for short-term operations and in places which require large transmission ratios, within a relatively small body, quiet operation and self-locking (which occurs only at high ratios). In order to reduce this bad feature of the worm gears and the consequences arising from it (substantial warming and relatively short service life, due to the high wear) manufacturers are trying to take a detailed look at all the factors which influence the work gear, to provide customers with the best possible solution [1,2]. The purpose of this paper is to point out the usual methods of determining the service

factor, which is the most influential parameter. The selection of the size of the worm gear is based on this parameter.

2. PROBLEM DESCRIPTION

The selection of universal worm gear units depends on many factors. In most cases, they are only selected in cases of short run, when it is required to reach large transmission ratios within a relatively small space. Also when it is required to ensure that the axis of the input and output shafts are passing each other; when it is required to ensure a peaceful and quiet operation, especially in cases where it is required to provide self-locking, although it should be noted that not all worm gears are self-locking [1,2].

While selecting worm gear unit, apart from the other conditions, (thermal capacity [3,4] and permissible radial and axial load [1,4]), the main condition has to be met. Nominal value of the torque (T_{2N}) should be higher than the value of the actual torque on the output shaft. Actual torque on the output shaft should be increased by all the imbalances which occur during the exploitation and it mostly depends on: driving machine type, working machine type, daily operating period, starts per hour, indicated load during an hour, ambient temperature, type of lubricant, variability of the direction of a rotation, the type of the gear teeth and the desired service life of the gear unit. This condition is usually expressed by the equation:

$$T_{2N} \geq T_2 f_B \quad (1)$$

where is the following stand for:

T_{2N} – Nominal value of the torque which is the Maximum torque at the output. In a quiet load, it can load in a way that working in a single shift it can reach a work span of, typically, 5000 hours of maintenance at an acceptable price [1,4].

T_2 – Output torque at a certain power and at a certain number of rotations per minute

$$T_2 = 9550 \cdot \frac{P_2}{n_2} = 9550 \cdot \frac{P_1 \cdot \eta}{n_2} \quad (2)$$

where it is:

P_2 – output power,

n_2 – rotations per minute of output shaft,

P_1 – input power,

η – efficiency,

f_B – service factor,

$$f_B = \prod_{i=1}^n f_i \quad (3)$$

where is the following stand for:

n – The number of parameters to be taken into account, which differ significantly from manufacturer to manufacturer,

f_1 – factor which takes into account the type of driving machine, electric motor or hydro motor, multi-cylinder combustion engine and the single cylinder internal combustion engine,

f_2 – factor which takes into account the type of working machine,

f_3 – factor which takes into account daily operating period in hours,

f_4 – factor which takes into account number of starts per hour,

f_5 – factor which takes into account real load of gearbox in an hour, ED factor,

f_6 – factor which takes into account ambient temperature, some of the producers consider that the value of normal temperature is 20°C, while others consider it to be 40°C. Some producers allows higher load if the temperature is equal or lower than 10°C,

f_7 – factor which takes into account the type of lubricant and size of the reducer (for synthetic oils, $f_7 = 1$ [3])

f_8 – factor which takes into account variability of the direction of a rotation (for reversible drive some producers calculate that $f_8 = 1,2$ [2]),

f_9 – factor which takes into account the type of gear teeth and thermal capacity,

f_{10} – factor which takes into account the desired service life of the gear unit.

For the proper selection a of gear motor with a specific power and number of revolutions, basic requirement must be fulfilled [1], which is:

$$f_B \leq f_{BD} = \frac{T_{2N}}{T_2} \quad (4)$$

where

f_{BD} – is permissible value of the service factor, which is calculated by the equation (4). Producers give it in their catalogues, for each gear unit, with each input power (power of electric motor) and rotations per minute of the output shaft - n_2 .

The service factor value, or the number of components of the service factor which are taken into consideration (n), varies from manufacturer to manufacturer.

Winsmith company has one of the simplest methods of selection. This company defines the value of the service factor in tables, according to the AGMA standard, depending on the type of driving and working machine and the duration of the drive working machine. It selects the gear unit according to the equation (1).

The ROSSI company has a more complex procedure of selection, because, in addition to these factors, it also takes into account the number of switching over an hour, ambient temperature and effective load during the day. The company takes into account the thermal capacity of the gear unit, ie. requires that the input power (P_1) has to be less than the thermal capacity (P_{tN}) increased by a factor that takes into account the ambient temperature and the effective load gear (f_t).

$$P_1 \leq P_t = P_{tN} \cdot f_t \quad (5)$$

In addition, it takes into consideration the maximum permissible load of gear unit in the short-term operation:

$$T_2 \leq T_{2\max} \quad (6)$$

This method of selecting a gear unit defines in detail the variation factor which occurs during the exploitation Flender Cavex Company has the most complex selection method [3], which requires the fulfilment of several conditions:

1. As far as strength is concerned, the following requirement has to be fulfilled:

$$T_{2N} \geq T_2 \cdot f_1 \cdot f_2 \cdot f_3 \cdot f_4 \cdot f_7 \quad (7)$$

They see the first three factors as one factor and use different symbols to mark them.

2. As far as heating is concerned, the following requirement has to be fulfilled:

$$T_{2N} \geq T_2 \cdot f_5 \cdot f_6 \cdot f_7 \cdot f_9 \quad (8)$$

3. As far as Maximum load is concerned, while starting and stopping gearbox with brake, the following requirement has to be fulfilled:

$$T_{2N\max} \geq T_{2A} \cdot f_4 \cdot f_7 \quad (9)$$

4. As far as maximum permission load of gearing is concerned, the following requirement has to be fulfilled:

$$T_{2N\max} \geq T_{2A} \cdot f_4 \cdot f_8 \quad (10)$$

where :

$T_{2N_{max}}$ – stands for maximum peak permission load, at minimum RPM n_1 , presented in tables for $T_{2N_{max}}$.

A detailed analysis of those parameters and their values show us a different consideration of these factors.

3. PROBLEM ANALYSIS

Driving and working machine have a major impact on the appearance of shock loads as well as the duration of the driving [1], so all manufacturers devote great attention to these factors (table 1).

WINSMITH



Fig.1. Winsmith company worm gear unit [5]

Table 1. Service factor value depending on the type of drive, working machine and the duration of the drive, in case of small number of starts

Electric motor drive			
Drive duration	uniform	moderate	heavy
to 0,5 h	0,80	0,90	1,00
to 2 h	0,90	1,00	1,25
to 10 h	1,00	1,25	1,50
to 24 h	1,25	1,50	1,75
Multi-cylinder combustion engine drive			
Drive duration	uniform	moderate	heavy
to 0,5 h	0,90	1,00	1,25
to 2 h	1,00	1,25	1,50
to 10 h	1,25	1,50	1,75
to 24 h	1,50	1,75	2,00
Single cylinder combustion engine drive			
Drive duration	uniform	moderate	heavy
to 0,5 h	1,00	1,25	1,50
to 2 h	1,25	1,50	1,75
to 10 h	1,50	1,75	2,00
to 24 h	1,75	2,00	2,25

Table 2. In case of larger number of starts (from 10 to 20 starts)

Type of driving machine	Electric motor drive		
Drive duration	uniform	moderate	heavy
to 0,5 h	0,90	1,00	1,25
to 2 h	1,00	1,25	1,50
to 10 h	1,25	1,50	1,75
to 24 h	1,50	1,75	2,00

ROSSI



Fig.2. Rossi company worm gear unit [6]

Table 3. Service factor value

Type of driving machine		Electric motor drive		
Drive duration	Service life (h)	uniform	moderate	heavy
from 2 h	3150	0,67	0,85	1
from 2 to 4 h	6300	0,85	1,06	1,25
from 4 to 8 h	12500	1	1,25	1,5
from 8 to 16 h	25000	1,25	1,6	1,9
from 16 to 24 h	50000	1,6	2	2,36
Number of starts		uniform	moderate	heavy
4		1	1	1
8		1,06	1	1
16		1,12	1,06	1
32		1,18	1,12	1,06
63		1,25	1,18	1,12
125		1,32	1,25	1,18
250		1,4	1,32	1,25
500		1,5	1,4	1,32

FLENDER CAVEX



Fig.3. Flender Cavex company worm gear unit [7]

The influence of working machine and drive duration with an electric motor drive (f_1, f_2, f_3).

Table 4. Service factor value

Type of driving machine	Electric motor drive		
Drive duration	uniform	moderate	heavy
to 0,5 h	0,8	0,9	1
to 2 h	0,9	1	1,2
from 2 to 10 h	1	1,2	1,4
from 10 to 24 h	1,2	1,4	1,6

Table 5. Number of starts (f_4)

Number of starts	to 10	from 10 to 60	from 60 to 240	from 240 to 600
f_4	1	1,1	1,2	1,3

Table 6. The influence of time of effective load, the so-called. ED factor

ED factor	100	80	60	40	20
f_5	1	0,94	0,86	0,74	0,56

Table 7. The influence of ambient temperature and number of revolutions of input shaft (fan speed) - f_6

Temperature (°C)	to 10	20	30	40	50
$n_1 \leq 300 \text{ min}^{-1}$	0,9	1	1,14	1,33	1,60
$300 < n_1 \leq 1500$	0,9	1	1,17	1,42	1,75
$n_1 > 1500$	0,9	1	1,20	1,50	1,90

Table 8. The influence of type of lubricant and the size of the reducer (for synthetic oils, $f_7 = 1$)

The axial height of the gear unit	from 63 to 100	from 120 to 250	from 280 to 450	from 500 to 630
f_6	1,2	1,25	1,3	1,36

Table 9. The influence of variability of the direction of rotation

Revolution in one direction	$f_8 = 1$
Revolution in both directions (reversible drive)	$f_8 = 1,2$

The influence of the type of gear profile and thermal capacity (f_9) is given in a manufacturer's catalogue for each gearbox.

Such a different approach to defining a service factor certainly has a great influence on gear selection. For example, if you are looking for a gear that will be driven by an electric motor, at moderate load, which will run for 8 hours, with 10 starts per hour, the analyzed manufacturers offer the following values of a service factor given in table 2. The differences between the service factors are small, ie. it can be seen that they respect working conditions the most because their influence on the safety factor is the greatest.

Table 10. A summary of the analyzed service factors

Manufacturer	Moderate load, 8 hours, with 10 starts per hour
WINSMITH	$f_B = 1,25$
ROSSI	$f_B = 1,25 \cdot 1,06 = 1,325$
FLENDER CAVEX	$f_B = 1,2 \cdot 1 = 1,2$

If we take into consideration the ambient temperature, for example, 40 °C at operation with a four-pole motor ($n_1 \leq 1500 \text{ min}^{-1}$), then the difference becomes even more significant (Table 3).

Table 11. A summary of the service factors that are included and the ambient temperature of the analyzed manufacturers

Manufacturer	Moderate load, 8 hours, with 10 starts per hour with ambient temperature 40°C
WINSMITH	$f_B = 1,25$ Does not take into account the temperature
ROSSI	$f_B = 1,25 \cdot 1,06 = 1,325$

	The increased temperature limits the input power of the engine, and if this condition is met, the increase in temperature is not taken into account during the selection of the gear unit.
FLENDER CAVEX	$f_B = 1,2 \cdot 1 = 1,2$ In addition to this requirement, the condition $f_B = 1,25 \cdot 0,8 \cdot 1,42 \cdot 0,7 = 0,994$ has to be fulfilled as well; In addition to these two conditions, another two conditions must be met. However, as these conditions are related to the field of study of the first two manufacturers, they will not be mentioned here.

For the torque at the output, for example, $T_{2N} = 300 \text{ Nm}$, revolutions per minute $n = 30 \text{ min}^{-1}$ and effective load of ED = 60 % manufacturers recommended the following gear units:

WINSMITH

This manufacturer recommends the following procedure for gear unit selection for provided working conditions

$$T_{2N} \geq T_2 \cdot f_B = 300 \cdot 1,25 = 375 \text{ Nm}$$

This requirement corresponds to the gear unit marked 7FC, with the center distance of $a = 101,6 \text{ mm}$, input power $P_1 = 1,9 \text{ kW}$ and torque at the output $T_{2N} = 419 \text{ Nm}$. It should be particularly noted that the motor works with $n_{em} = 1200 \text{ min}^{-1}$.

ROSSI

This manufacturer recommends the following procedure for gear unit selection

$$T_{2N} \geq T_2 \cdot f_B = 300 \cdot 1,325 = 397,5 \text{ Nm}$$

This requirement corresponds to the gear unit marked 80, the center distance of $a = 80 \text{ mm}$, input power $P_1 = 1,74 \text{ kW}$, output power $P_{2N} = 1,33 \text{ kW}$, torque at the output $T_{2N} = 407 \text{ Nm}$, maximum torque peak the gear reducer will withstand $T_{2\max} = 710 \text{ Nm}$. It should be

particularly noted that the motor works with $n_{em} = 1200 \text{ min}^{-1}$ and that the number of revolutions of the output shaft is $n_{em} = 31,5 \text{ min}^{-1}$.

The company takes into account the thermal capacity of the gear unit, but at this size and the transmission ratio of the gear unit, permission load is not limited by thermal capacity.

In addition, this company takes into consideration maximum permissible load for short-run drive

$$T_2 \leq T_{2\max}$$

As the driving mode is not specified, it will be counted

$$T_2 = 1,2 \cdot T_2 = 1,2 \cdot 300 \leq T_{2\max} = 710 \text{ Nm}$$

So this size of the gear unit is acceptable.

FLENDER CAVEX

This manufacturer has the most complex selection method and it requires the fulfilment of multiple requirements

1. As far as strength is concerned, the following requirement has to be fulfilled:

$$T_{2N} \geq T_2 \cdot f_1 \cdot f_2 \cdot f_3 \cdot f_4 \cdot f_7 = 300 \cdot 1,2 \cdot 1 \cdot 1,42 = 511,2 \text{ Nm}$$

They see the first three factors as a single factor and use different symbols to mark them.

This requirement corresponds to the gear unit marked 100, with input power $P_1 = 3,3 \text{ kW}$, torque at the output $T_{2N} = 813 \text{ Nm}$, with maximum torque peak the gear reducer will withstand $T_{2\max} = 1060 \text{ Nm}$ and service factor $f_9 = 0,58$. It should be particularly noted that the motor works with $n_{em} = 1500 \text{ min}^{-1}$ and that the number of revolutions of the output shaft is $n_{em} = 30 \text{ min}^{-1}$

2. As far as heating is concerned, the following requirement has to be fulfilled:

$$T_{2N} = 812 \text{ Nm} \geq T_2 \cdot f_5 \cdot f_6 \cdot f_7 \cdot f_9 = 300 \cdot 0,86 \cdot 1,2 \cdot 1,42 \cdot 0,58 = 255 \text{ Nm}$$

Which is satisfactory.

3. As far as maximum load is concerned, while starting and stopping gear with braking, the following requirement has to be fulfilled:

$$T_{2N\max} = 1060 \text{ Nm} \geq T_{2A} \cdot f_4 \cdot f_7 = 300 \cdot 1,2 \cdot 1 \cdot 1,42 = 511 \text{ Nm}$$

Which is satisfactory.

It was calculated that the minimum value of the starting torque or braking torque is higher than $T_{2A} \geq 1,2 \cdot T_2$.

4. As far as maximum permission load of gearing is concerned, the following requirement has to be fulfilled:

$$T_{2N\max} = 1680 \text{ Nm} \geq T_{2A} \cdot f_4 \cdot f_8 = 300 \cdot 1,2 \cdot 1 \cdot 1,2 = 432 \text{ Nm}$$

Where $T_{2N\max}^* = 1680 \text{ Nm}$ - stands for maximum torque peak, at minimum revolutions per minute n_1 , which is given in tables for $T_{2N\max}^*$. In addition, it was calculated that it was the most unfavourable alternating drive. Selected gear unit satisfies.

4. CONCLUSION

Based on the analysis, it is evident that some manufacturers take into account different load mode, which significantly influences the selection of the size and thus the price of a gear unit. Of course, in addition to the standards that strongly define the parameters of the service factor manufacturers have a different experience and approach to their own defining. The greatest impact on their defining has a safety factor which is used to calculate the gear unit and the effect of an ambient temperature ie. cooling method. If the safety factor is lower than the external conditions, it has to be more significantly respected and vice versa. In this particular case it can be noted that one of the manufacturers recommended a slightly smaller gear unit ($a = 80 \text{ mm}$) for the given conditions of the drive, while the other two recommended a gear unit with the same axial distance of $a \approx 100 \text{ mm}$. It is particularly interesting that one of the manufacturers, which has a greater axial distance, has twice as big capacity as the other manufacturers with the same axial distance.

It is undisputed that great attention should be devoted to this extremely important factor.

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NUMERICAL ANALYSIS OF MOTORCYCLE SUSPENSION SYSTEM

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Abstract: A shock absorber is the main part of a vehicle suspension system. This mechanical device has a role to reduce the bumpy road shocks and enable a more comfortable ride. In this study we investigated the behavior of a shock absorber, that is installed on the motorcycle, with three different material of helical springs. 3D model was created using Catia v5 r18. Numerical simulation was done by Ansys workbench 12.0. The results of von Mises stress, deformation and shear stress, in the case of three different materials of coil spring, are analyzed and presented.

Key words: shock absorber, suspension system, helical spring, finite element method

1. INTRODUCTION

A shock absorber presents a mechanical or hydraulic device that is designed to absorb the holes and bumps on the road. Kinetic energy of the shock is transformed into another form of energy, for example, in the heat, and then performs dissipation. A coil spring, as the main part of the shock absorber, is defined as an elastic body. Spring has a role to compress when loaded, and to return to the initial state, when the load disappears. In this study, we used three different materials of helical coil spring: structural steel, spring steel and chromium-vanadium steel. The last two materials are increasingly used in the suspension system. It is very important to determine which steel gives better results in real conditions.

2. LITERATURE REVIEW

A large number of researchers studying the suspension system of the vehicle. The goal is to design an optimal shock absorbers and using quality materials of helical coil, get better vehicle performance.

The authors [2] have presented an analysis of the shock absorber before and after optimization design. They executed the change in diameter of the coil and the use of

two materials showed how different loads affect the operation of the damper.

The author [3] observed the behavior of spring steel as the main material coils. It is a new material, invented by Japanese researchers, that has application in the suspension system. They concluded that spring steel having very high ultimate tensile strength.

Another author [4] has studied the behavior of the shock absorber when using two different materials coils. These materials are structural steel and aluminum alloy. A comparison was made for the same boundary conditions, and the result of research shows that best material for shock absorber is the steel.

As it is known, a composite material made from two or more constituent materials with significantly different physical or chemical properties. The resulting material has a better structure than the individual components. Recent research [5] studying composite materials. It was concluded that composite materials may find use in suspension systems because they have high strength, high stiffness and low weight.

3. MATERIALS AND METHODS

3.1. Model definition

A 3D computer model of a shock absorber was developed using the CATIA v5 r18. The total height is 336 mm. Helical coil spring has the following physical characteristics:

- height 220 mm,
- diameter of wire 10 mm,
- total number of coils 10,
- outer diameter of spring coil 70 mm.

Figure 1 show 3D geometry of a shock absorber.

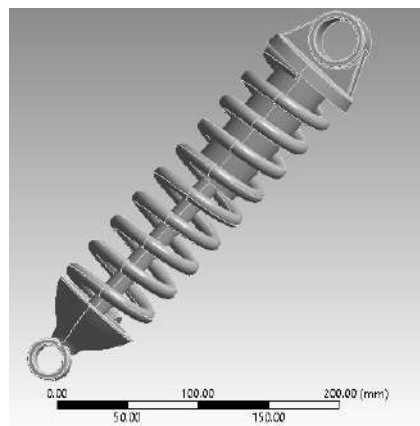


Fig.1. Geometry of shock absorber

Weight of motorcycle is 125 kg. In this study, it is assumed that a vehicle has two passenger of 75 kg. Rear suspension absorbs 60% of the total weight, and the force that acting on the shock absorber has a value of 1618 N. One end of the model is fixed, while at the other end acts mentioned value of the force.

3.2. Material definition

Numerical simulations were conducted with three different materials of helical spring. The following

materials are used: structural steel, spring steel and chromium-vanadium steel. Table 1 shows main properties of used materials.

Table 1. Properties of materials[6], [7]

Properties	Structural steel	Spring steel	Chromium-vanadium steel
Young's Modulus (MPa)	2E+05	2.02E+05	2.1E+05
Density (kg/m ³)	7850	7820	7800
Poisson's ratio	0.3	0.29	0.29
Tensile strength (MPa)	460	570	940
Yield Strength (MPa)	250	360	620

3.3. Finite element formulation

Finite element has become the most popular method for the investigation of the suspension system of the vehicle. This method solves a large number of differential equations to calculate von Mises stress and deformation of the model. The method has been widely used to predict mechanical behavior of shock absorber in various driving situations such as driving over rough road, or sudden braking.

Principle of virtual work is one of the basic principles of continuum mechanics. Applying the boundary conditions in the equilibrium equations [8] virtual work of internal and external forces can be equal, and we have

$$\delta W_{int} = \delta W_{ext} \quad (1)$$

Matrix form of virtual work of the previous equation can be written as

$$\delta W_{int} = \int_V \delta \mathbf{e}^T \boldsymbol{\sigma} dV$$

$$\delta W_{ext} = \int_V \delta \mathbf{u}^T \mathbf{F}^V dV + \int_{S^\sigma} \delta \mathbf{u}^T \mathbf{F}^S dV + \sum_i \delta \mathbf{u}^T \mathbf{F}^{(i)} \quad (2)$$

Applying the principle of virtual work and the constitutive relations for linear elastic material in matrix form, we have

$$\boldsymbol{\sigma} = \mathbf{C} \mathbf{e} \quad (3)$$

Applying the concept of isoparametric interpolation [9] in the finite element, we can write the equation of equilibrium finite elements

$$\mathbf{K} \mathbf{U} = \mathbf{F}_{ext} \quad (4)$$

where K is element stiffness matrix, U displacements at the nodes and F_{ext} - external forces in the element nodes. A basic assumption in the linear analysis of solids is that

the moving solids is infinitesimally small, the material is linearly elastic, and that the nature of the boundary conditions remain unchanged under the action of external loads. About that, last equation is related to the linear analysis of solids because the moving of the nodes is linear function of external forces.

4. RESULTS

Numerical simulation was done using Ansys workbench 12.0. Linear tetrahedral element was used as the final element. Finite element mesh consists of 138962 nodes and 77881 elements.

Numerical solutions of von Mises stress, deformations and shear stress are shown in the following figures. Figures 2, 3 and 4 shown result of simulations for the case when the material of helical spring is structural steel.

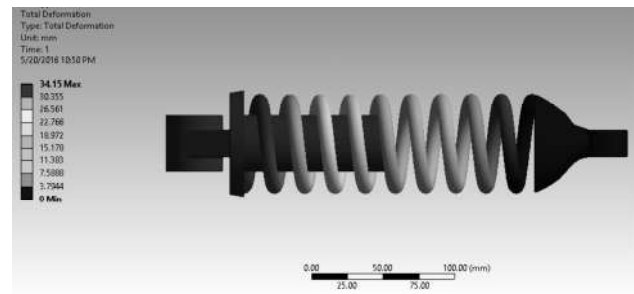


Fig.2. Total deformation of shock absorber for the first case

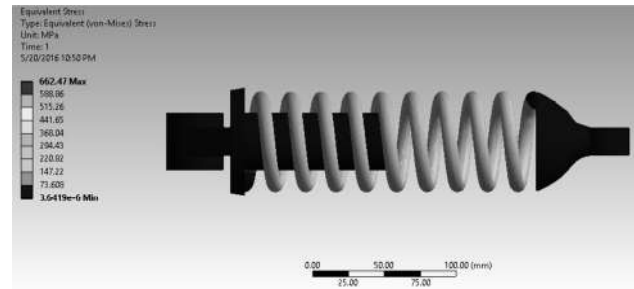


Fig.3. Von Mises stress of shock absorber for the first case

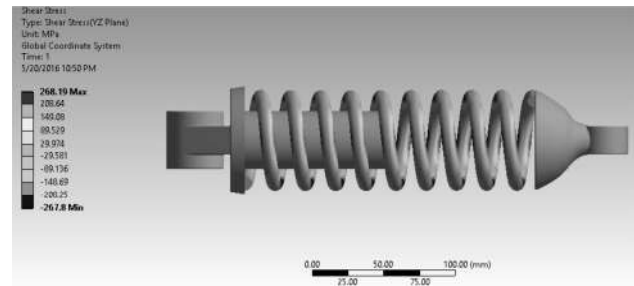


Fig.4. Shear stress of shock absorber for the first case

Figure 2 shows that the largest displacement appears in an area where the force is applied. The highest recorded value is 34.15 mm. Maximum von Mises stress (662.47 MPa) is developed at the inner side of the helical spring (Fig. 3). The maximum value of shear stress, 268.19 MPa, is recorded on a part of the helical spring which comes into contact with other parts of the coils during the compression process.

The simulation results for the second case, ie, when the material of coil is spring steel, are shown in the following three pictures.

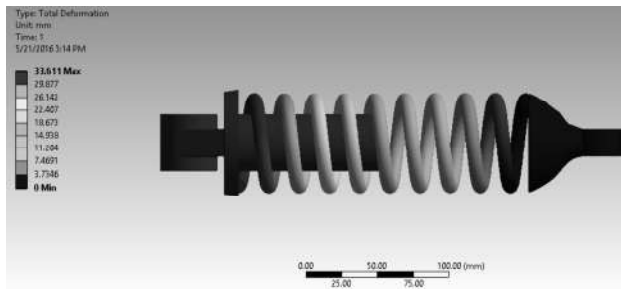


Fig.5. Total deformation of shock absorber for the second case

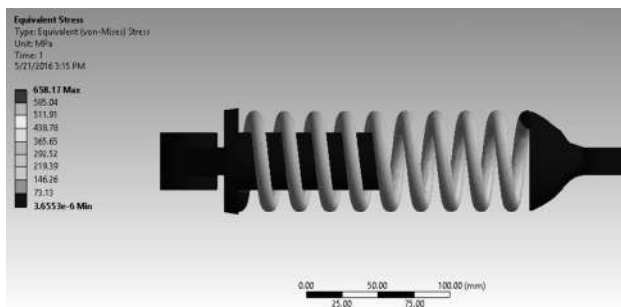


Fig.6. Von Mises stress of shock absorber for the second case

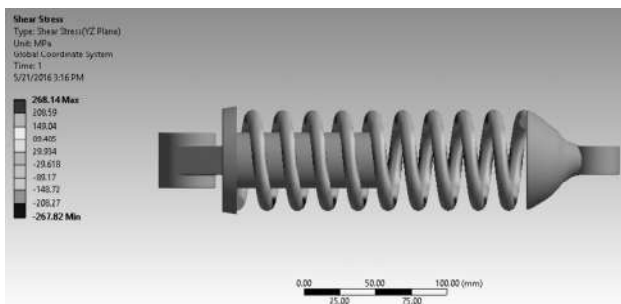


Fig.7. Shear stress of shock absorber for the second case

Results of the second case, show the same von Mises stress distribution, deformations and the shear stress. Maximum deformation is 33.61 mm, while von Mises stress has maximum value 658.17 MPa. Shear stress of shock absorber has maximum value 268.14 MPa. Finally, the third case involves the use of chromium-vanadium steel materials. The simulation results are shown in the following figures.

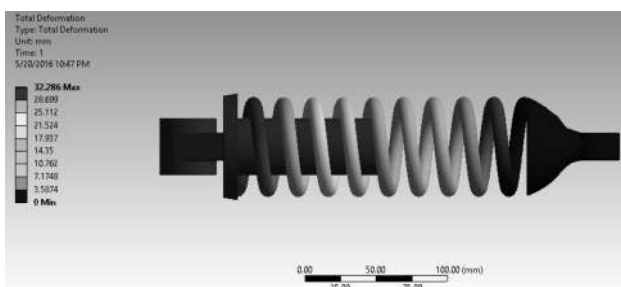


Fig.8. Total deformation of shock absorber for the third case

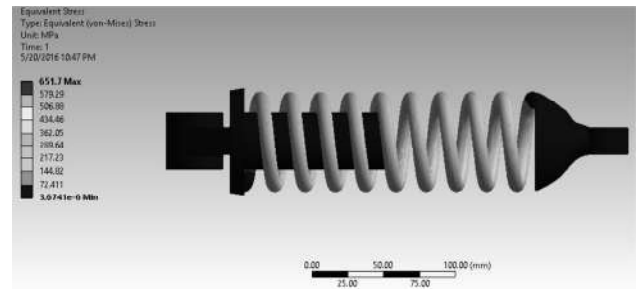


Fig.9. Von Mises stress of shock absorber for the third case

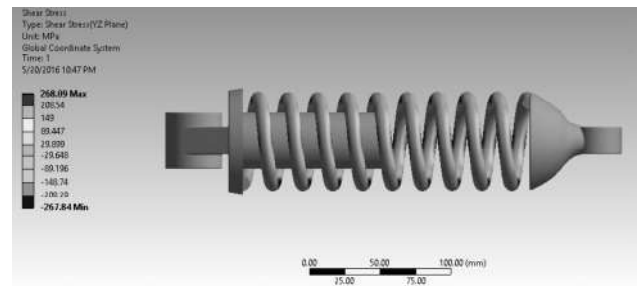


Fig.10. Shear stress of shock absorber for the third case

Fig. 8 shows that the deformations are smallest in this case, and have the maximum value of 32.28 mm. Also, the maximum value of von Mises stress (651.7 MPa) is smaller than in the previous two cases. Shear stress also has a lower maximum value (268.09 MPa). Next figure showed that the best solution, as the material of helical coils, is chromium-vanadium steel, because during the compression springs observed the smallest deformations (Fig. 11).

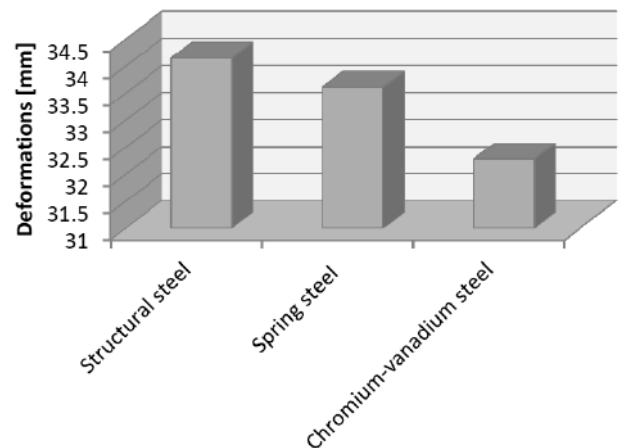


Fig.11. Comparison of the obtained deformations

5. CONCLUSION

In this study, we studied the behavior of the shock absorber in case loads its own weight and the weight of two passengers. In numerical simulations, we used three different materials spring: structural steel, spring steel and chromium-vanadium steel.

Results of the analysis showed that the best solution, as the material of helical springs, is chromium-vanadium steel, because during the compression springs observed the smallest deformations and von Mises stresses.

Our future research will be in the direction of optimization design springs, and testing of dynamic loads. It would be interesting to find a solution by optimizing the design in order to find the application of materials that give a poor results during workloads.

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INFLUENCE OF VANES SHAPE ON FLOW VELOCITY OF VENTILATED DISC IN HEAVY TRUCK BRAKING

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Abstract: Today, the ventilated disc brakes are the most common type of disc brakes used on road vehicles, especially on heavy-duty vehicles. The reason for such wide application mainly lies in the fact that they allow the best dissipation of heat that is generated during brake process. The influence of vanes shape on air flowing velocity through the disc, and pressures that occur as consequence of air flowing, and the influence of disc speed on air flowing velocity is studied in this paper. Air disc brakes for heavy duty commercial vehicle applications with three types of vanes shape are analysed. The analysis how disc speed influencing on air flow velocity in the case of disc with sweep ribs is performed at the end.

Key words: disc brake, heat dissipation, air flowing, cooling, vanes.

1. INTRODUCTION

A braking system is one of the most important safety components of an automobile. Brakes, as mechanisms that provide braking, in fact, consist of a series of elements and special devices, so that together they are mainly used to decelerate vehicles from an initial speed to a given speed.

During braking process, the braking torque is transmitted from the disc on the hub, than to the wheel and at the end on the tire. At the end, the stopping or braking occurs in contact of tire and the road. The rate of heat generation in a friction braking system is a function of the vehicles mass, velocity, and rate of deceleration. If this heat is not dissipated effectively, the temperatures in the brake and surrounding components become too high, high temperatures are responsible for most problems in vehicle braking systems. Such problems include excessive component wear, squeal, judder, brake fading, overheating of brake fluid, seals and other components

and in extreme cases complete failure of the brakes. The fluid temperature may rise to the point where the fluid vaporizes with the subsequent loss of braking. Any improvements to the cooling characteristics of a braking system will reduce the risk of the above problems and provide safer vehicle transport.

It is important to understand that the brake pads absorb only a small quantity of heat during braking. This is very useful during braking, because it provides protection of the brakes and its internal components. A large amount of heat is absorbed and temporarily stored on the disc during stopping (approximately 90 per cent [1]). It is important to point out that the disc capacity is limited. In order to maintain the proper brake function, the disc acts as a heat exchanger. Disc must be able to dissipate the heat that is generated in the braking process into the environment. Solid disc have limited, or lower ability of heat dissipation. As a result this design of disc has a less application on commercial vehicles [2].

Discs that are used on passenger cars and on many heavy-duty commercial vehicles usually have a hollow cross-section, or have holes in them. The main purpose of cross-section or holes is to allow air flowing through the disc and thereby provide better heat dissipation. With such type of construction, the two metal plates are connected by ribs or pillars. Flow through ribs or pillars occur during rotation of the disc, the air passing through the disc takes the heat into the environment and thus cooling of the disc is performed.

In the last decade the requirement on the brake disc cooling is intensified due to significant increase in vehicle speed, weight, and acceleration. Currently, starting from the necessity of higher cooling efficiency, the ventilated disc brake is fast becoming the most commonly used solution in automotive braking system. For years, ventilated brake rotors have been used for their weight savings and additional convective heat transfer from the air channels between outer and inner rotor cheeks. However, the amount of additional cooling due to this internal air flow is not well defined and depends on the individual brake rotor's geometry and the cooling air flow conditions around the brake assembly. Therefore flow analysis and heat dissipation have fascinated many researchers.

Indisputably, to prevent the occurrence of hot judder, excessive heating of the brake disc must be avoided while minimizing temperature variations across the rotor. In order for this to be achieved the brake rotor must be designed to ensure sufficient uniform heat dissipation and thermal capacity [3].

Thus, how to select better geometrical design variables and improve thermal performance of automotive brake rotors is a task that the vehicle designers and researchers are often confronted. Several papers have been done to investigate the various aspects of performance of conventional and ventilated brake discs [4, 5, 6 and 7].

Use of curved vanes requires the left and right side discs to be the mirror images of each other to achieve greater pumping efficiency. Another possible means of increasing the heat dissipation of vented rotors is the use of pillar-vented discs, which separate the two friction surfaces using a series of pillars instead of radial vanes [8]. The pumping efficiency of pillar-vented rotors is less than that

of a rotor with traditional vanes. Wallis et al. [9] have shown that the pillar-vented discs have similar heat transfer rate when compared with discs having curved vanes because of increase in turbulence and hence heat transfer coefficient. Other advantages of these discs [9] include a higher resistance to thermal deformation owing to the more even distribution of material, which results in lower thermal stress build-up within the rotor.

Experimental studies have proved that the cooling efficiency of ventilated brake discs depends on two key characteristics: the mass flow rate through the disc (i.e. the pumping efficiency of the rotor) and the average heat transfer coefficient on the surface of the disc [8, 10].

In this paper, a three-dimensional CFD parametric study of the effect of pin geometry on flow field characteristics within a commercial disc and its variants is performed. Three types of disc brake rotors in heavy truck braking, circular pillar, radial vane and curved vane are analysed.

2. 3D MODEL

Stopping or slowing down of the heavy-duty vehicle requires a large amount of energy. As the result of this energy we have the heat that is in big quantity absorbed by disc. Absorbed heat causes thermal stresses of disc that have a negative influence on disc material.

The rise in temperature of the brake disc in any braking operation will depend on a number of factors including the mass of the vehicle, the rate of retardation, and the braking process duration. In the case of short duration brake applications with low deceleration, the disc and friction material may absorb all the thermal energy generated. As a result very little heat dissipation occurs as the temperature rise in the disc is minimal. In extreme braking operations such as steep descents or repeated high-speed brake applications, sufficient heat dissipation becomes critical to ensure reliable continued braking. As the disc temperature rises it begins to dissipate heat, at steady-state conditions heat generated through braking equals heat dissipation and no further heating occurs. If the heat generation is greater than the dissipation then the temperature will rise, the rate of this rise will depend of the relative quantities of each. If sufficient heat dissipation does not occur the temperature of the disc and friction material can reach critical levels and brake failure may occur [7].

Heat dissipation from the brake disc will occur via conduction through the brake assembly and hub, radiation to nearby components and convection to the atmosphere. Convection to the atmosphere is the primary means of heat dissipation from the brake disc. The use of an internally ventilated rotor will increase both surface area (extra internal area exposed to the atmosphere) and the heat transfer coefficient, due to forced convection created by the internal airflow, with negligible influence on unsprung mass. The material selection and the physical dimensions of the rotor will also have a direct influence on cooling ability.

The aim of this paper is based on determining the best vanes shape, to provide efficient air flowing through the disc, and in that way better heat dissipation and cooling.

An experimental approach, which is naturally the best way in any research, would involve developing a

procedure to study the airflow in and around vented rotors, as well as incorporating a brake test, in order to simulate the airflow around the brakes on a moving vehicle.

Due to an economic aspect and possibility of modern technology, 3D model of air disc brake is made in CATIA and will be used in the analysis. 3D solid model of air disc brake can be seen on Figure 1. The outer diameter of the disc is 410 mm [11].



Fig.1. Air Disc Brakes for Heavy Duty Commercial Vehicle Applications

The actual brake disc involves two rubbing surfaces parted by the blades. One of the rubbing surfaces is attached to the wheel with the help of mounting bolts. The disc brake rotor is rotationally periodic with blades and passages at equal angular spacing. The analysis was performed on 1/18 part of the model which reduced the computational cost considerably. Disc rotor has been modelled using a rotating frame of reference with periodic boundaries. The geometry and the flow pattern repeat themselves at a specified angle about the centreline. Investigation of air flowing will be performed for three different vanes shapes, showed on Figure 2.

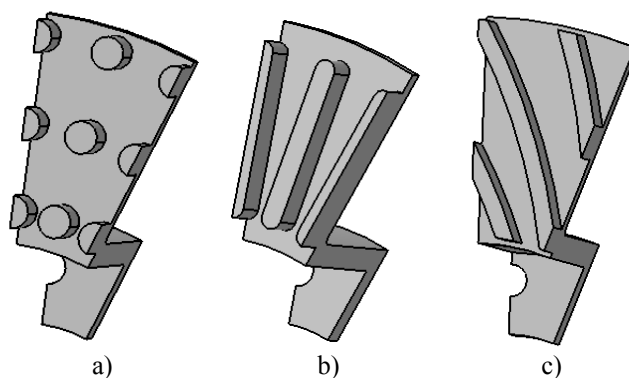


Fig.2. Vanes shapes with a) pillars, b) radial ribs and c) sweep ribs

Taking into account that the disc is cylindrical, it is not necessary to observe the entire disc, it is sufficient to use a slice of disc, in this case 20° [3, 4]. In the analysis, within the boundary conditions, it will be defined that the disc is symmetrical.

2.1. Boundary conditions

Air flow analysis in the heavy-duty commercial vehicle disc is performed using ANSYS, i.e. Fluent Flow module. The space between the inner and outer plate of vented disc, i.e. an area where air flow, is also modelled and is considered in further analysis. The space that is filled with air is shown in Figures 3 – 5.

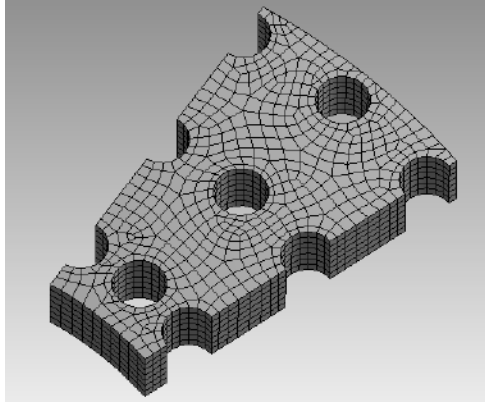


Fig 3. Air space in disc with pillars

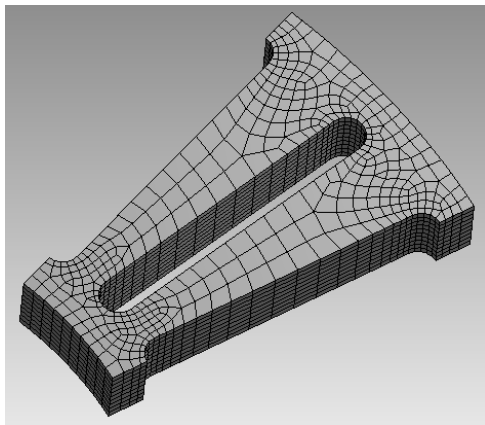


Fig 4. Air space in disc with radial ribs

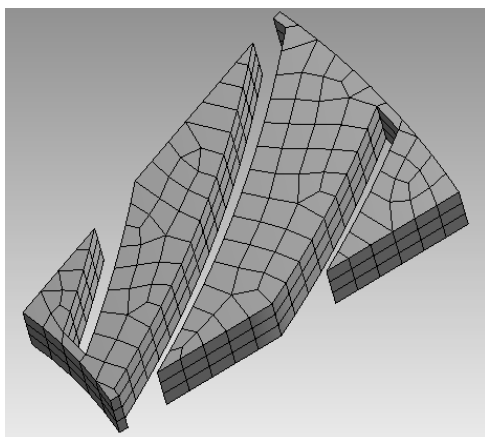


Fig 5. Air space in disc with sweep ribs

The environment temperature and pressure are pre-set and are 25°C and 101325 Pa. It was also necessary to define the air inlet and the outlet, as well as the walls of disc. As stated above, the brake disc is symmetrical and is defined rotational symmetry.

2.2. Analysis of the influence of vanes shape on air flow velocity

Air flow analysis is performed for the wheel speed of 400 rpm, for all three vanes shapes of disc. Results of this analysis can be seen in Figures 6 – 8.

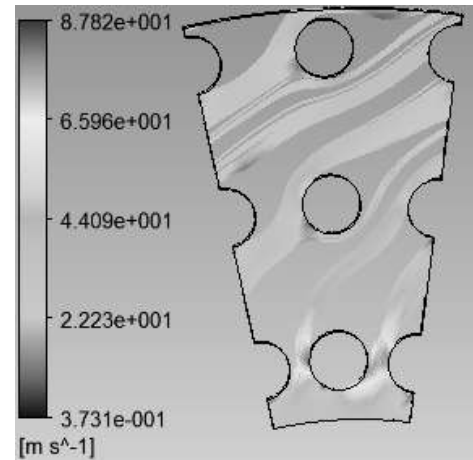


Fig.6. Air flow velocity for disc with pillars

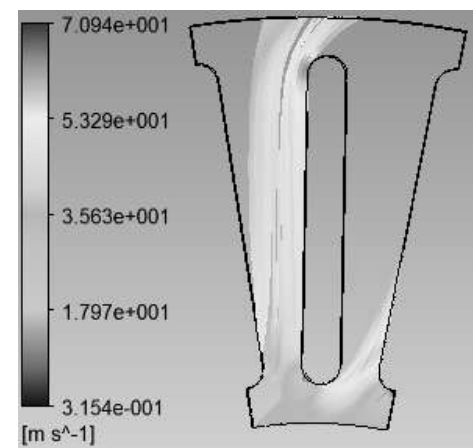


Fig.7. Air flow velocity for disc with radial ribs

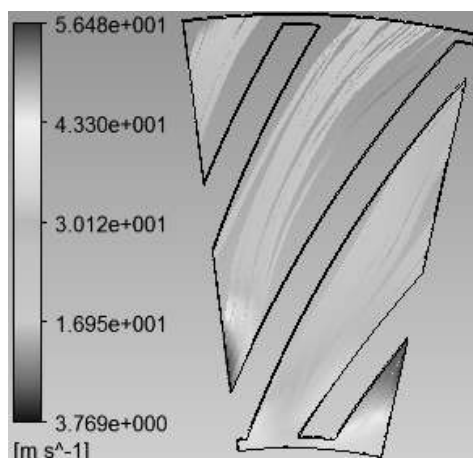


Fig.8. Air flow velocity for disc with sweep ribs

By analysing Figures 6 – 8, it can be concluded that the maximum air flow velocity occurs in case of disc with pillars, and the smallest air flow velocity in case of disc

with sweep ribs. Flow velocity is very important parameter because velocity values have direct influence on the cooling rate of disc. However, this is not the only important parameter, and that is how uniform is an air flow velocity. This parameter is the best for the disc with sweep ribs that can be seen in Figures 6 – 8. The main reason for this is that the rib shape follows the direction of the disc rotation. Based on the foregoing, it can be concluded that the disc with sweep ribs is the best to use in terms of air flow.

2.3. Analysis of the influence of vanes shape on air flow pressure

Analysis of pressure that occurs due to air flow through the disc is performed for the same conditions that have been defined during the analysis of air flow. Results that are obtained during this analysis are shown in Figures 9 – 11.

Based on the performed analysis, one can come to the conclusion that the minimum pressures occur, as consequence of air flow, on the walls of a disc with sweep ribs, and the highest for a disc with pillars. Taking into account the results of previous analysis, it can be concluded that the disc with the sweep ribs is the best choice. As the pressure is relatively small, it can be stated that the pressure doesn't have a big influence on the disc itself.

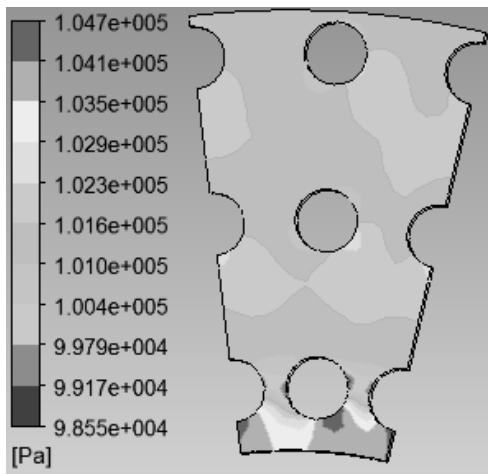


Fig.9. Pressure values for disc with pillars

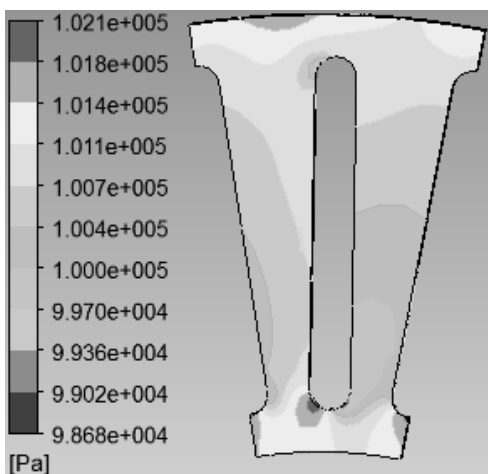


Fig.10. Pressure values for disc with radial ribs

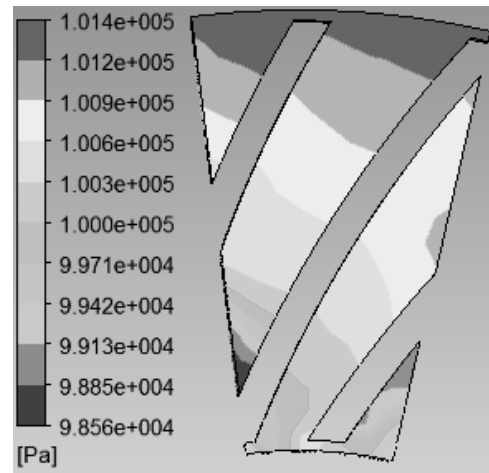


Fig.11. Pressure values for disc with sweep ribs

2.4. Influence of disc rotor speed on air flow velocity of disc with sweep ribs

Based on previous analysis, the most attention is drawn to the disc with sweep ribs. For this reason, an analysis of the influence of disc speed on air flow velocity is performed for this type of vented disc. The main objective of the analysis is to determine the dependence between air flow velocity and disc speed. Results of this analysis are given in Figures 12 – 13.

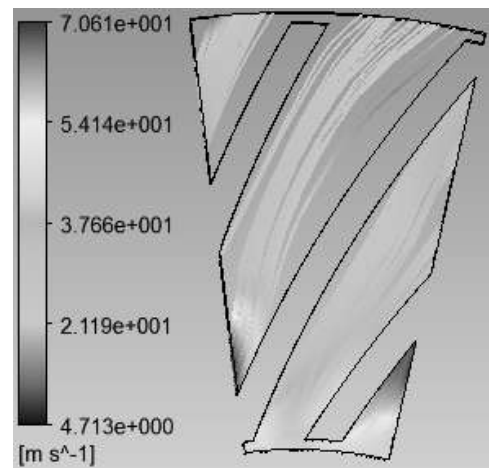


Fig.12. Air flow velocity for disc speed 500 rpm

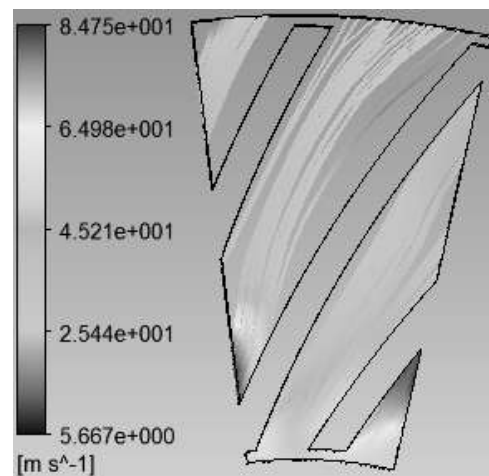


Fig.13. Air flow velocity for disc speed 600 rpm

3. RESULTS

Maximum air flow velocities that are obtained from prior analysis are given in Table 1.

Table 1. Maximum Air Flow Velocity

Disc Speed n [o/min]	Maximum air flow velocity v [m/s]
400	56.48
500	70.61
600	84.75

Based on the data obtained at different disc speeds, can be concluded that with the increase of disc speed, the air flow velocity also increases. Using data from Table 1, can be obtained relationship between air flow velocity and disc speed and it is given by the equation (1).

$$v = 0.1414 \cdot n - 0.0617 \quad (1)$$

4. CONCLUSION

By forming the 3D model of ventilated air disc brake of heavy-duty vehicle and varying shapes of vanes, the following conclusions are drawn:

- The present investigation has clearly revealed that pin profiles affect heat transfer characteristics of a pin-vented disc considerably.
- The highest values of air flow velocity are obtained for disc with pillars;
- The best uniformity of air flow velocity is shown for disc with sweep ribs and
- The smallest values for pressure on disc walls that occurs during air flowing are obtained for disc with sweep ribs.

The best shape of vanes, on the basis of previous analysis is sweep ribs. With increasing the disc speed, the air flowing velocity increase linear. However, use of sweep ribs requires the left and right side discs to be the mirror images of each other to achieve greater pumping efficiency.

ACKNOWLEDGMENT

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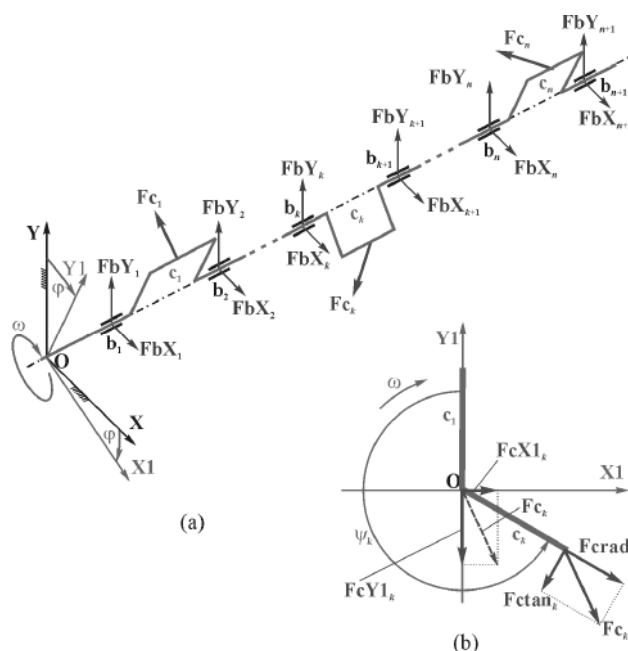
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Key words: computer program, engine crankshaft bearings, polar load diagram

Since high load acting on the main bearings shortens the lifetime of aforementioned engine parts and lowers reliability of the entire engine, it is necessary to reduce the magnitudes of bearing forces as much as possible. In that sense, the best effects are achieved by mounting counterweights on the crank throws. In order to find optimal configuration of counterweights for a crankshaft, it is necessary to construct polar load diagrams of all the main bearings of the crankshaft. The polar load diagram is, in fact, a vector diagram of forces transmitting from a

The paper is structured as follows. Basic principles of crankshaft bearings load determination and final equations used in the development of the "Polar diagrams" computer program are given in Section 2. Section 3 describes the user interface of the program in detail. Some results of the program, obtained by its application on a three-cylinder diesel engine, are presented in Section 4. Section 5 contains conclusions.

Fig. 1 shows the schematic view of an in-line engine crankshaft with $(n+1)$ main bearings with the forces acting on them. Two coordinate systems are also shown in Fig. 1, one of which is stationary (OXY) and the other one (OX1Y1) rotates together with the crankshaft with the angular velocity ω . The position of the coordinate system OX1Y1 with respect to the coordinate system OXY is defined by the crankshaft angle φ . It should be noted that all the forces depicted in Fig. 1 depend on this angle.



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Each crank force F_{c_i} ($i=1,...,n$) that originates from the crank i is distributed to all the bearings, not only to the adjacent ones. The forces F_{c_i} can be calculated in a common way taking into account gas forces and inertia forces that dominate over other forces in IC engines. The gas forces can be measured or modelled by using some of the known methods and the inertia forces can be easily determined by using Newton's second law of motion assuming that the crankshaft rotates with a constant angular velocity. Therefore, the forces F_{c_i} can be considered as known.

In [6] it is shown that each force $Fb_i(\varphi)$ acting on the main bearing b_i , ($j=1,...,n$) can be expressed by its projections on X and Y axes:

$$\begin{aligned} FbX_j(\varphi) &= FbXl_j(\varphi) \cdot \cos \varphi + FbYl_j(\varphi) \cdot \sin \varphi \\ FbY_j(\varphi) &= -FbXl_j(\varphi) \cdot \sin \varphi + FbYl_j(\varphi) \cdot \cos \varphi \end{aligned} \quad (1)$$

where $FbXl_j$ and $FbYl_j$ are the projections of the force Fb_j onto the coordinate system OX₁Y₁:

$$\begin{aligned} FbXl_j(\varphi) &= \sum_{i=1}^n (-\rho_{i,j} \cdot F_{crad_i}(\varphi_i) \cdot \sin \psi_i + \rho_{i,j} \cdot F_{ctan_i}(\varphi_i) \cdot \cos \psi_i) \\ FbYl_j(\varphi) &= \sum_{i=1}^n (\rho_{i,j} \cdot F_{crad_i}(\varphi_i) \cdot \cos \psi_i + \rho_{i,j} \cdot F_{ctan_i}(\varphi_i) \cdot \sin \psi_i). \end{aligned} \quad (2)$$

F_{crad_j} and F_{ctan_j} are radial and tangential components of the crank force, $\rho_{i,j}$ are the influence coefficients [6], ψ_k is the counterclockwise angle between cranks c_1 and c_k (Fig. 1b) and φ_i is the angle that indicates the position of crank c_i during the appropriate engine cycle. Considering that engine cycle in cylinder i advances by angle θ_i relative to the engine cycle in cylinder 1, the following relationship between angles φ_i and φ is valid - $\varphi_i = \varphi + \theta_i$. Based on that calculation, a computer program "Polar diagrams" has been developed.

3. DESCRIPTION OF THE PROGRAM

The program "Polar diagrams" is written by using Mathworks Matlab software. The user interface of the program is shown in Fig. 2. It is divided into five segments that are enumerated accordingly, indicating the order of data input.

The segment 1, with an example basic engine data entered into the form, is shown in Fig. 3. The data involve cylinder bore, piston stroke, length of the connecting rod and engine speed.

Then, the masses necessary to calculate inertial forces of the crank gear should be entered and it is done in segment 2. These are the masses of the piston, connecting rod, crank pin and crank web, as it can be seen for the example data, shown in Fig. 4.

To calculate crank force, cylinder pressure values throughout the whole engine cycle are needed. These values are imported from an excel file to the "Polar diagrams" program. By pressing the button "Calculate crank force" in segment 3 of the program (Fig. 5), a

dialog box is opened so the user can choose the appropriate excel file containing the cylinder pressure values.

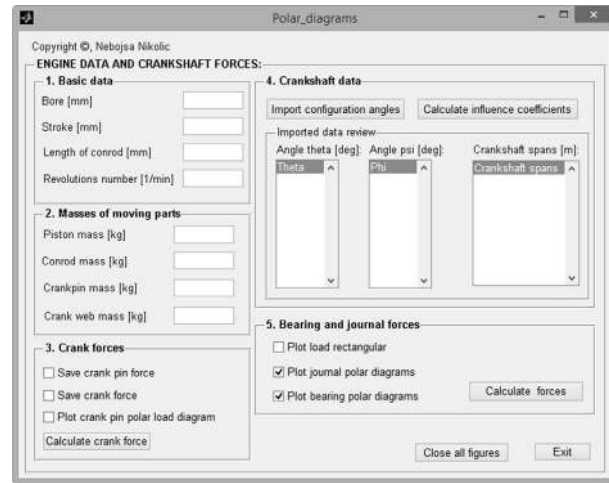


Fig.2. User interface of the program "Polar diagrams"

1. Basic data	
Bore [mm]	101.6
Stroke [mm]	114.2
Length of conrod [mm]	206.4
Revolutions number [1/min]	2500

Fig.3. Input of basic engine data

2. Masses of moving parts	
Piston mass [kg]	1.57
Conrod mass [kg]	2.79
Crankpin mass [kg]	0.865
Crank web mass [kg]	0.458

Fig.4. Input of crank gear elements masses

3. Crank forces	
☐ Save crank pin force	
☐ Save crank force	
☐ Plot crank pin polar load diagram	
Calculate crank force	

Fig.5. User interface segment for calculating crank force

Fig. 5 also shows additional options by which the user can choose whether to save crank force and crank pin force values to excel files and also whether to plot crank pin polar load diagram.

Crankshaft data involve so called configuration angles and crankshaft spans. Engine cycle phase angles (θ - θ) and spatial crank angles (ψ - ψ) are here named configuration angles. Phase angles depend on the engine firing order and spatial crank angles depend on spatial configuration of the crankshaft. All the mentioned crankshaft data are imported from excel files and this is done in segment 4 of the program (Fig. 6).

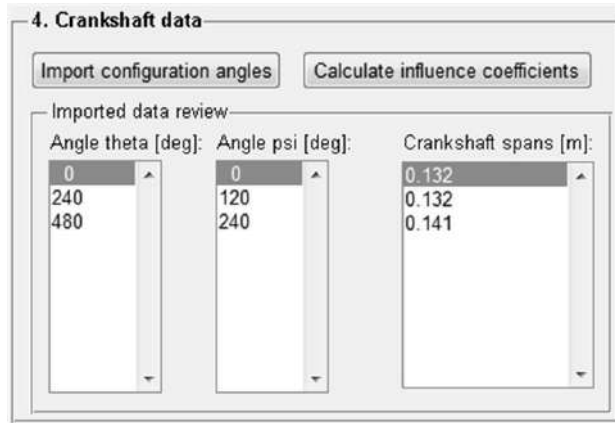


Fig.6. Input of crankshaft data and influence coefficients calculation

By pressing the button "Import configuration angles", a dialog box opens, offering the user to select an excel file containing angles θ and ψ . After the excel file is selected, the imported data appear in appropriate listboxes as shown in Fig. 6 for a three-cylinder in-line engine.

Further, by pressing the button "Calculate influence coefficients", another dialog box opens, where the user can choose an excel file with the crankshaft spans data. The rightmost listbox in Fig. 6 is populated with the data from the chosen excel file and influence coefficients are calculated thereafter.

With the crank force and influence coefficients known, it is possible to calculate journal and bearing forces. This is done by simply pressing the button "Calculate forces" in segment 5, shown in Fig. 7. After the calculation has been completed, appropriate diagrams are plotted for each crankshaft main journal and/or bearing, depending on the checkboxes checked in segment 5. Examples of the diagrams are shown in the next section of the paper.

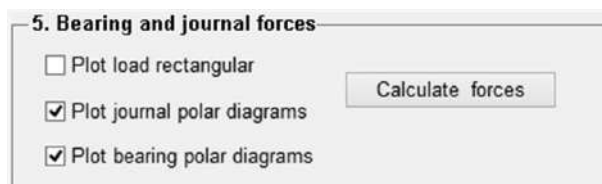


Fig.7. User interface segment for calculating bearing and journal forces

4. ILLUSTRATION OF THE PROGRAM APPLICATION

To illustrate the application of the program developed, a three cylinder in-line diesel engine is taken as an example. Some of the basic data used are already shown in Figs. 3,

4 and 6. Since there are 4 main journals and bearings in the three-cylinder engine, 12 diagrams can be plotted for them and two more for the crank pin and crank pin bearing by using the "Polar diagrams" program. For that reason, not all the generated diagrams can be presented here, but only some of them are shown as an illustration. Polar load diagrams of the crank pin and crank pin bearing are shown in Figs. 8 and 9. Further, Fig. 10 shows the polar load diagram of the main journal 2. Finally, the polar load diagram of the main bearing 2 is shown in Fig. 11.

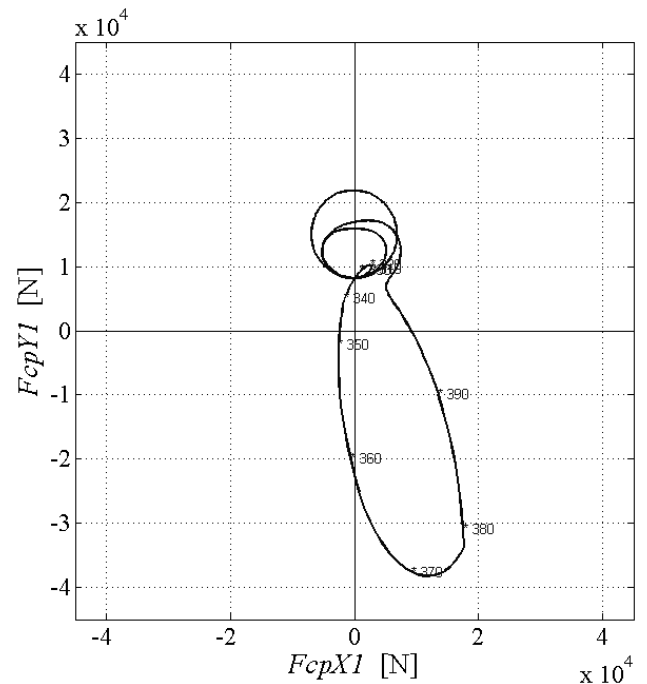


Fig.8. Polar load diagram of the crank pin

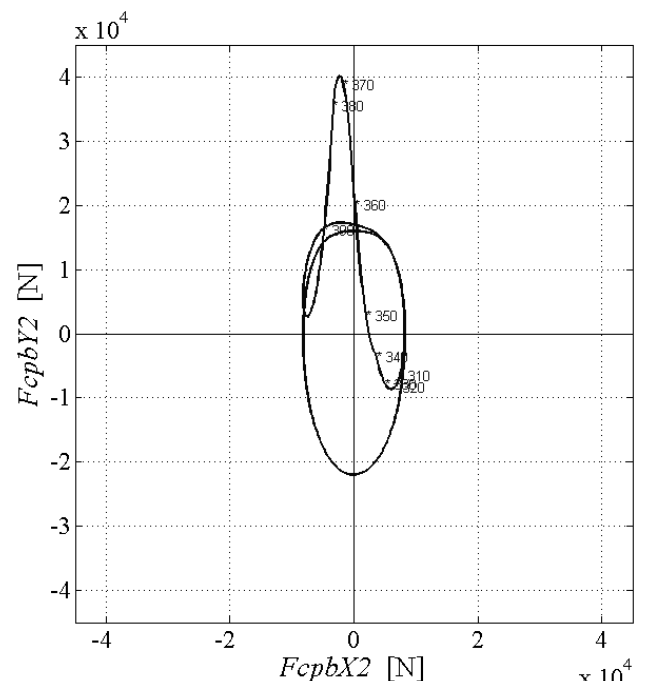


Fig.9. Polar load diagram of the crank pin bearing

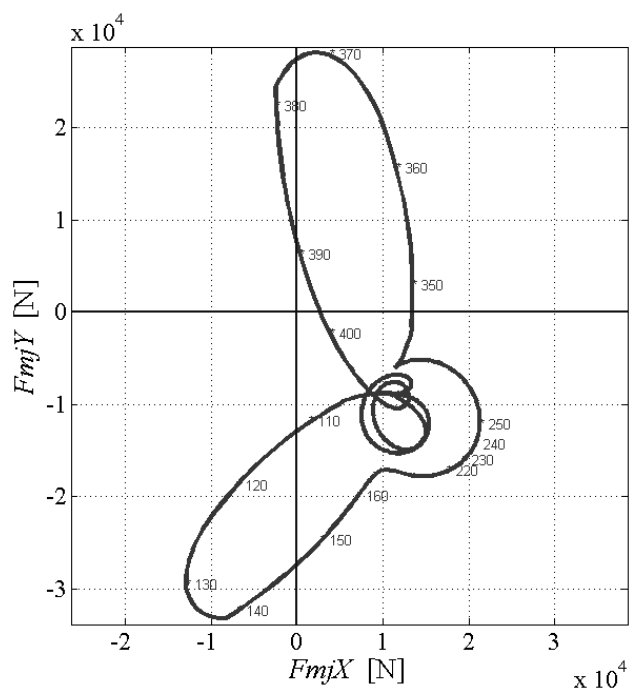


Fig.10. Polar load diagram of main journal 2

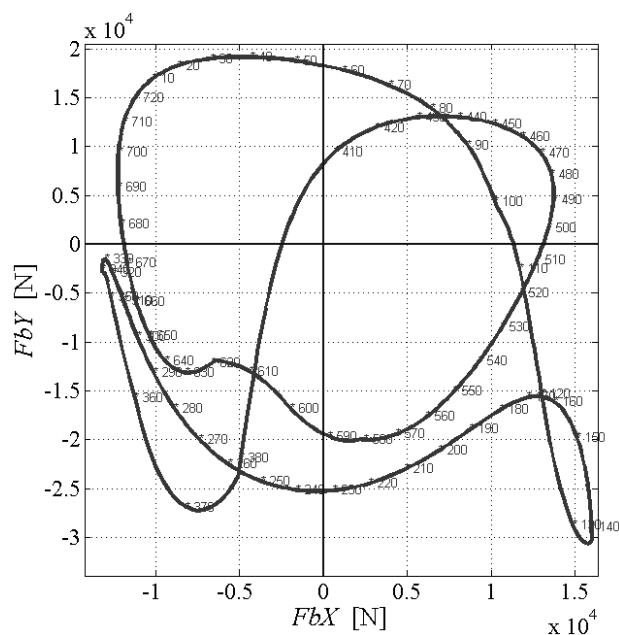


Fig.11. Polar load diagram of main bearing 2

5. CONCLUSION

An interactive computer program for load calculation of IC engine crankshaft bearings/journals has been developed and described in this paper. The program is also capable of plotting journal and bearing load diagrams for various engine input data given. These diagrams provide a visual representation of the load distribution around the inner circumference of main bearings/journals. A part of the program benefits has been illustrated by the results obtained after the program had been applied to a three cylinder in-line engine.

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NONLINEAR KINEMATICS OF ENGINE CRANK-PISTON MECHANISM

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Abstract: The paper deals with nonlinear kinematic behaviour of engine crank-piston mechanism. The basics of crank-piston mechanism's dynamic are presented, followed by analysis of nonlinear kinematic relations connected to kinematic operation of crank-piston mechanism. Simulation of nonlinear kinematics of the crank-piston mechanism of Lombardini 3LDA 450 engine was performed, especially for the piston position, velocity and speed. Corresponding conclusions were drawn.

Key words: crank-piston mechanism, nonlinear, kinematics

1. INTRODUCTION

Since the beginning of the design of internal combustion (IC) engines, engineers had worked on creating new and on improving the existing engines. This means that during designing of a new engine or improving the existing one, it is crucial to determine optimal dimensions of elements of the engine.

A crank-piston mechanism is one of the most important mechanisms in IC engines. It is a functional group that efficiently transforms back-and-forth movement into rotary movement. It is also excellent in converting thermodynamic processes to yield the maximal work and efficiency.

Design of crankshaft assemblies has a common criterion for both gasoline and diesel engines - running smoothness. In gasoline engines, where obtaining high speed has priority, final goal is to make a minimum of moving masses. In diesel engines, where pressures may be twice as high as in gasoline engines, the priority is to control the effects of the gas force [1].

In order to design basic dimensions, assess engine power and consumption and/or evaluate and compare different engines, designers must take into account the following engine characteristics: stroke, bore, piston displacement and compression ratio [1, 2, 3].

Nonlinear geometric problems associated with dynamic effects frequently occur in engines [2]. However, for most analyses, a wide range of simplifications is used, mainly in real geometry, kinematics and boundary conditions. Insufficient knowledge on real kinematics of crank-piston mechanism may result in design of a mechanism with insufficient reliability [4]. Taking into account that simplifications may cause discrepant results, real operating conditions should be used in analysis [5].

The task of kinematic analysis of the crank-piston mechanism can be achieved graphically, analytically or by using computer aided design. In this paper, analytical analysis of nonlinear kinematics of the crank-piston mechanism of Lombardini 3LDA 450 single-cylinder engine was performed.

2. DYNAMICS OF CRANK-PISTON MECHANISM

Kinematically relevant parts of the crankshaft drive - the piston, the connecting rod and the crankshaft perform various movements:

- the piston oscillates back and forth in the cylinder,
- the connecting rod is articulated to the small connecting rod eye by the piston pin that moves back and forth and is also connected with the large connecting rod eye to the crankshaft which rotates, so the connecting rod swings in the plane of the crankshaft circle,
- the crankshaft rotates.

The most important geometric characteristics of the crank-piston mechanism are: connecting rod length, l , crank radius, r , cross sectional area of the piston, A_k , connecting rod sweep angle, β , and crankshaft angular displacement, α , Fig. 1.

During one rotation of the crankshaft, the piston moves from top to bottom and returns to the top dead centre (TDC) and, thus, executes two strokes. It accelerates and decelerates while executing this movement. The crank-piston mechanism movement is described in terms of the crankshaft angular displacement, α , or the angle between the cylinder axis and the crankshaft throw.

From the theory of resistance of materials, machine elements and related fields, it is well known that, it is necessary to have knowledge about forces acting in the system in order to perform mechanical calculations of construction elements.

The forces in the crankshaft drive of an internal combustion engine arise from the gas pressure in the combustion chamber and from inertial forces:

- gas force acting on the piston, F_g ,

$$F_g = (p_g - p_o) \cdot A_k \cdot 10^5 \quad (1)$$

where: p_g is pressure of gases and p_o is ambient pressure,

- inertial force of the piston, F_{io} ,

$$F_{io} = -m \cdot a \quad (2)$$

where: m is piston mass and a is piston acceleration,

- resultant force on the piston, F_R ,

$$F_R = \pm F_g \pm F_{io} \quad (3)$$

■ normal force, N ,

$$N = F_R \cdot \tan \beta \quad (4)$$

■ tangential force, T ,

$$T = \frac{F_R \cdot \sin(\alpha + \beta)}{\cos \beta} \quad (5)$$

■ radial force, R ,

$$R = \frac{F_R \cdot \cos(\alpha + \beta)}{\cos \beta} \quad (6)$$

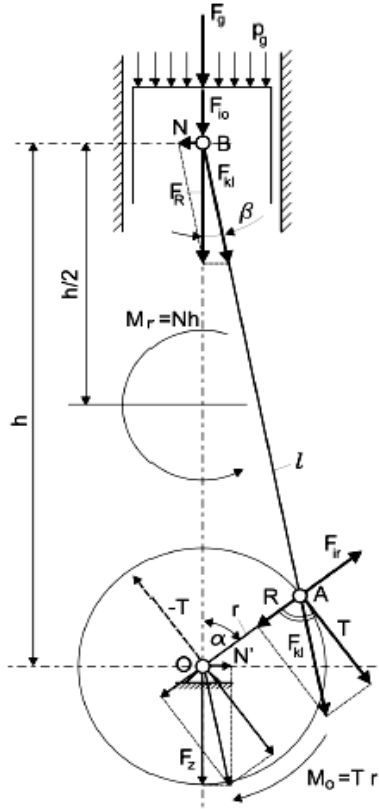


Fig.1. Forces acting on engine crank-piston mechanism

Pressure of gases, p_g , can be measured by indicating a pressure into engine cylinder or by thermodynamical calculations. Ambient pressure, p_0 , piston surface, A_k , and piston mass, m , are constants. The crankshaft angular displacement, α , can be easily measured, while the piston acceleration, a , can be calculated using the basic knowledge of mechanics and mathematics.

3. NONLINEAR KINEMATIC BEHAVIOUR OF CRANK-PISTON MECHANISM

Kinematic analysis of crank-piston mechanism includes the study of its displacement, velocity and acceleration, without consideration of any external forces acting on it. Knowledge regarding these quantities is important for definition of inertial influence of each member of the crank-piston mechanism and determination of the law of variation of volume of the workspace during engine cycle. By transposing the crank-piston mechanism, i.e., shifting the articulation point of the connecting rod to the piston, A , and crank pin, B , out of the cylinder axis, Fig. 2, by

amounts e_o and e_v , respectively, movement of the crank-piston mechanism can be changed as desired. There are shifted crankshaft mechanisms, in which the crankshaft centre is displaced from the axis of the cylinder, and axially offset crankshafts mechanisms, in which the piston pin is moved from the axis of the cylinder. Both offsets enlarge the stroke and move the top and the bottom dead centre from its defined position. [2, 6].

Nonlinear kinematic behaviour of a non-axial crank-piston mechanism may be observed using geometrical and kinematic relations visible from Fig. 2. During derivation of analytical expressions for position, velocity and acceleration of characteristic points, a xTDCy coordinate system was used.

First, the influencing dimensionless factors are defined:

- the connecting rod ratio, λ :

$$\lambda = \frac{r}{l} \quad (7)$$

- relative non-axial characteristic, δ :

$$\delta = \frac{e_v + e_o}{l} \quad (8)$$

By using the definitions (7) and (8) and geometrical relations visible from Fig. 2, the following equations may be written for the connecting rod sweep angle, β :

$$\sin \beta = \frac{r \sin \alpha + e_v + e_o}{l} = \lambda \sin \alpha + \delta \quad (9)$$

Thus, the connecting rod sweep angle, β , can be calculated as follows:

$$\beta = \arcsin(\delta + \lambda \sin \alpha) \quad (10)$$

It is obvious that the connecting rod sweep angle, β , is a nonlinear function of crankshaft angular displacement, α :

$$\beta = f(\alpha) \quad (11)$$

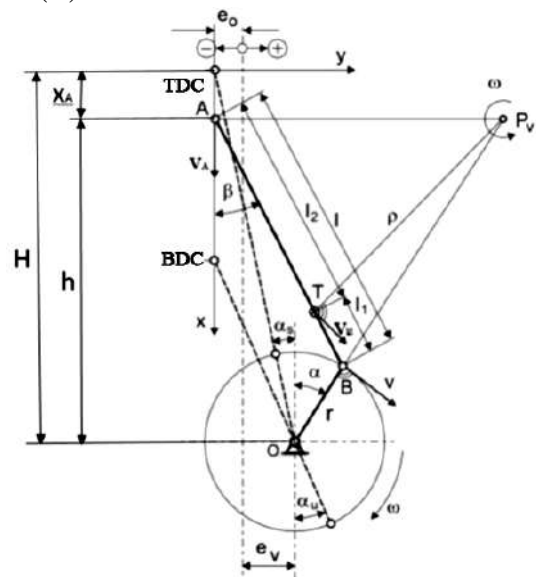


Fig.2. Non-axial crank-piston mechanism

The connecting rod instantaneous angular speed, ω_{kb} , may be calculated as follows:

$$\omega_{kl} = \frac{d\beta}{dt} = \frac{d\beta}{dt} \cdot \frac{d\alpha}{d\alpha} = \frac{d\alpha}{dt} \cdot \frac{d\beta}{d\alpha} = \omega \cdot \frac{d\beta}{d\alpha} \quad (12)$$

where ω is the crankshaft angular speed.

The first derivative of the connecting rod sweep angle, β , with respect to crankshaft angular displacement, α , can be obtained as follows:

$$\begin{aligned} \frac{d\beta}{d\alpha} &= \frac{d}{d\alpha} \arcsin(\delta + \lambda \sin \alpha) = \\ &= \frac{\lambda \cos \alpha}{\sqrt{1 - (\delta + \lambda \sin \alpha)^2}} = \frac{\lambda \cos \alpha}{\sqrt{1 - \sin^2 \beta}} = \frac{\lambda \cos \alpha}{\cos \beta} \end{aligned} \quad (13)$$

Now, the connecting rod angular speed, ω_{kl} , can be obtained:

$$\omega_{kl} = \omega \frac{\lambda \cos \alpha}{\cos \beta} \quad (14)$$

Instantaneous angular acceleration of the connecting rod, ε_{kl} , may be calculated as the second derivative of the connecting rod sweep angle, β , or the first derivative of the connecting rod angular speed, ω_{kl} :

$$\begin{aligned} \varepsilon_{kl} &= \frac{d^2\beta}{dt^2} = \frac{d}{dt} \left(\omega_{kl} \right) = \frac{d\omega_{kl}}{dt} \cdot \frac{d\alpha}{d\alpha} = \\ &= \frac{d\alpha}{dt} \cdot \frac{d\omega_{kl}}{d\alpha} = \omega \cdot \frac{d\omega_{kl}}{d\alpha} = \\ &= \omega \cdot \frac{d}{d\alpha} \left(\omega \cdot \frac{d\beta}{d\alpha} \right) = \omega^2 \cdot \frac{d^2\beta}{d\alpha^2} \end{aligned} \quad (15)$$

The second derivative of the connecting rod sweep angle, β , with respect to crankshaft angular displacement, α , can be calculated, taking into account the equation (13):

$$\begin{aligned} \frac{d^2\beta}{d\alpha^2} &= \frac{d}{d\alpha} \left(\frac{d\beta}{d\alpha} \right) = \frac{d}{d\alpha} \left(\frac{\lambda \cos \alpha}{\cos \beta} \right) = \\ &= \frac{-\lambda \sin \alpha \cos \beta - \lambda \cos \alpha \frac{d}{d\alpha} \cos \beta}{\cos^2 \beta} = \\ &= \frac{-\lambda \sin \alpha \cos \beta + \lambda \frac{d\beta}{d\alpha} \cos \alpha \sin \beta}{\cos^2 \beta} \end{aligned} \quad (16)$$

The final form of the second derivative of the connecting rod sweep angle, β , with respect to crankshaft angular displacement, α , is:

$$\frac{d^2\beta}{d\alpha^2} = \frac{\lambda \left(\frac{d\beta}{d\alpha} \cos \alpha \sin \beta - \sin \alpha \cos \beta \right)}{\cos^2 \beta} \quad (17)$$

Thus the angular acceleration of the connecting rod, ε_{kl} , may be written as follows:

$$\varepsilon_{kl} = \omega^2 \frac{\lambda \left(\frac{d\beta}{d\alpha} \cos \alpha \sin \beta - \sin \alpha \cos \beta \right)}{\cos^2 \beta} \quad (18)$$

When the piston reaches the TDC, the following geometric relation can be applied:

$$H = \sqrt{(r+l)^2 - (e_v + e_o)^2} \quad (19)$$

where H is vertical distance between point O, where the connecting rod is connected to the crankshaft and point A, where the connecting rod is connected to the piston, at the moment when piston reaches TDP, Fig. 2. The instantaneous piston position, x_A , is given by equation (20), depending on the angular displacement of the crankshaft, α :

$$x_A = H - r \cos \alpha - l \cos \beta \quad (20)$$

The instantaneous piston speed, v_A , can be determined as the first derivative of the piston position:

$$v_A = \frac{dx_A}{dt} = r\omega(\sin \alpha + \frac{1}{\lambda} \cdot \frac{d\beta}{d\alpha} \sin \beta) \quad (21)$$

Now, the instantaneous piston acceleration, a_A , has a form:

$$a_A = \frac{dv_A}{dt} = r\omega^2 \left\{ \cos \alpha + \frac{1}{\lambda} \left[\frac{d^2\beta}{d\alpha^2} \sin \beta + \left(\frac{d\beta}{d\alpha} \right)^2 \cdot \cos \beta \right] \right\} \quad (22)$$

The piston position divided by crank radius is calculated as:

$$x_{Ar} = \frac{x_A}{r} = \frac{H}{r} - \cos \alpha - \frac{1}{\lambda} \cos \beta \quad (23)$$

Similarly, the piston velocity divided by peripheral velocity of the crankshaft can be calculated using:

$$v_{Ar} = \frac{v_A}{r\omega} = \sin \alpha + \frac{1}{\lambda} \cdot \frac{d\beta}{d\alpha} \sin \beta \quad (24)$$

Finally, the piston acceleration divided by peripheral acceleration of the crankshaft is:

$$a_{Ar} = \frac{a_A}{r\omega^2} = \cos \alpha + \frac{1}{\lambda} \left[\frac{d^2\beta}{d\alpha^2} \sin \beta + \left(\frac{d\beta}{d\alpha} \right)^2 \cdot \cos \beta \right] \quad (25)$$

In literature, kinematics of the crank-piston mechanism is often observed as axial ($\delta=0$), using simpler mathematical models, where the radicals of power series are developed.

Instantaneous position, velocity and acceleration (x'_A , v'_A , a'_A , respectively) of the piston are then calculated using approximate formulae [2,7]:

$$x'_A = r \left[1 - \cos \alpha - \frac{\lambda}{4} (1 - \cos 2\alpha) \right] \quad (26)$$

$$v'_A = r\omega \left(\sin \alpha + \frac{\lambda}{2} \sin 2\alpha \right) \quad (27)$$

$$a'_A = r\omega^2 (\cos \alpha + \lambda \cos 2\alpha) \quad (28)$$

With increase of the piston speed, inertial forces, friction, wear, flow resistance during intake and engine noise also increase. The maximum permissible inertial forces limit the piston speed and, hence, the rotational speed, ω . On diesel and petrol engines with direct injection, the rotational speed is additionally limited by the time necessary for the mixture formation.

4. RESULTS OF ANALYSIS

The analysis of nonlinear kinematic was performed on the crank-piston mechanism of the Lombardini 3LDA 450 engine situated in the Laboratory for internal combustion engines, fuel and lubricants, at the Faculty of Engineering of the University of Kragujevac, Fig.3.

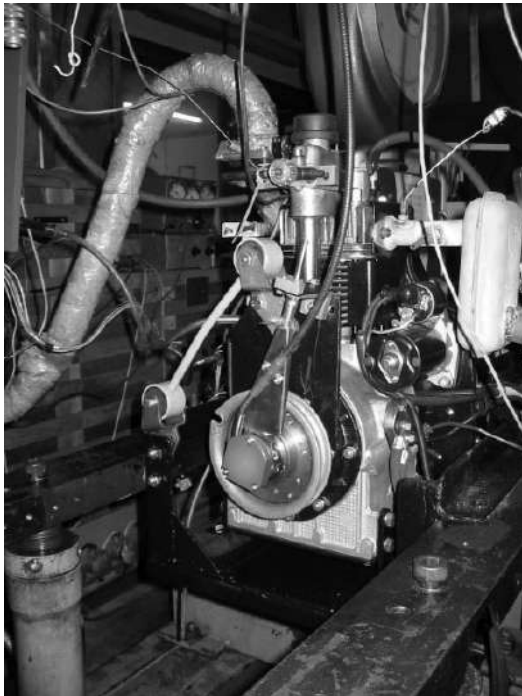


Fig 3. 3LDA 450 engine on the test bench

Lombardini 3LDA 450 is a single-cylinder air-cooled diesel engine with basic specifications given in Table 1.

Table 1. Specifications of 3LDA 450 engine

Capacity	454 cm ³
Number of cylinders	1
Bore x stroke	85 x 80 mm
Compression ratio	17,5:1
Maximum torque	28,5 Nm
Maximum power	7,3 kW
Engine speed	3000 min ⁻¹

Other construction parameter values of the engine that are necessary for kinematic analysis are given in Table 2.

Table 2. 3LDA 450 engine geometric parameters

Connecting rod length, l	0.12 m
Crank radius, r	0.04 m
Offset, e_v	0
Offset, e_θ	0

Simulation of nonlinear kinematics of the engine was performed using a Matlab/Simulink program specially developed for this purpose. In order to execute one working cycle of IC engine, the crankshaft must perform two whole spins or execute rotation of 720°, so kinematic effects that occur in the crank-piston mechanism were observed for the crankshaft angular displacement in the range of $\alpha=0\div720^\circ$. Kinematic analysis was focused on position, velocity and acceleration of the piston.

Fig. 4 shows the piston movement with respect to the crankshaft angular displacement, calculated according to relation (19).

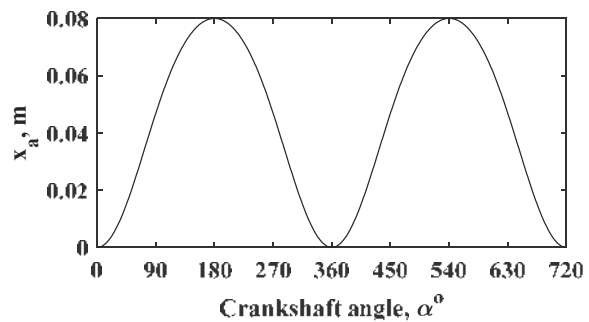


Fig. 4. The piston position, x_A , with respect to the crankshaft angle, α

The piston velocity with respect to the crankshaft angular displacement is shown in Fig. 5. Maximal absolute value of velocity of $v_{Amax}=13,35 \text{ ms}^{-1}$ is obtained for $\alpha=74^\circ$, 288° , 434° , 648° . At TDC ($\alpha=0^\circ$, 360° , 720°) and again at BDC ($\alpha=180^\circ$, 540°), the piston velocity is zero, because the piston reverses direction at those points (in order to change direction, the piston must stop at some point).

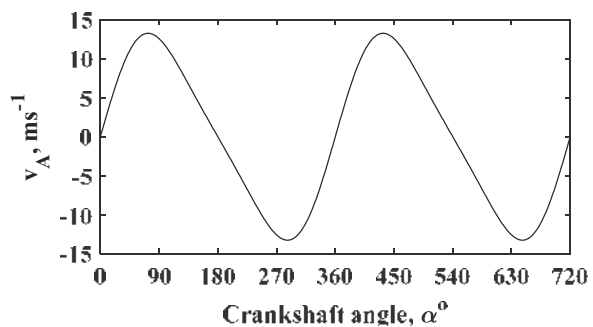


Fig. 5. The piston velocity, v_A , with respect to the crankshaft angle, α

The piston velocity at any rotation point from TDC up to the maximum velocity is greater than at the same number of degrees before BDC. For example, the velocity at 30° after TDC is $v_{A1}=7,66 \text{ ms}^{-1}$, while the velocity at 30° before BDC is $v_{A2}=4,44 \text{ ms}^{-1}$.

The profile of the velocity curve and therefore the location of the maximum velocity, are both influenced by connecting rod ratio, λ . The location of maximum piston velocity, in turn, influences the design of camshaft lobe

profiles in order to optimize the intake event in a particular speed range, and can have an influence on the intake characteristics.

Fig. 6 presents the acceleration of the piston with respect to the crankshaft angular displacement, α .

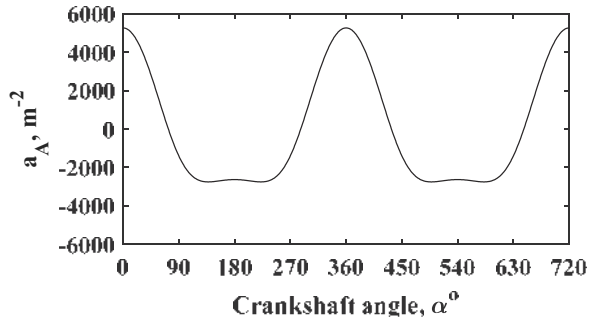


Fig. 6. The piston acceleration, a_A , with respect to the crankshaft angle, α

Maximal piston acceleration of $a_A = 5263.79 \text{ ms}^{-2}$ is obtained when the piston reaches the TDC for $\alpha = 0, 360^\circ, 720^\circ$. Between TDC and maximum piston velocity, acceleration is positive but decreasing toward zero (the piston velocity is still increasing but less rapidly). At maximum piston velocity, the piston stops speeding up and begins to slow down. At that point, the acceleration changes direction. Minimal value of piston acceleration of $a_{A\min} = -2753.68 \text{ ms}^{-2}$ is achieved in four angular positions of the crankshaft, for $\alpha = 138^\circ, 222^\circ, 498^\circ, 582^\circ$. Value of acceleration achieved when the piston reaches the bottom dead point (BDC) is $a_{ABDC} = -2631.89 \text{ ms}^{-2}$ at $\alpha = 180^\circ, 540^\circ$. Thus, the maximum negative acceleration does not occur at BDC, but about 42° on either side of BDC. The diagram of the piston acceleration during one rotation of the crankshaft is symmetrical with regard to crankshaft angular position that corresponds to the position of bottom dead centre. The odd shape at the bottom of the piston acceleration curve is the result of superposition of several orders of acceleration where the first two are the most significant.

The nonlinear kinematic behaviour of the crank-piston mechanism can also be observed from diagram shown in Fig. 7. In this figure, piston position, x_{Ar} , velocity, v_{Ar} , and acceleration, a_{Ar} , normalized according to equations (23) to (25), respectively, are shown. The shapes of the curves and positions of characteristic values confirm previous statements on kinematic behaviour of the crank-piston mechanism.

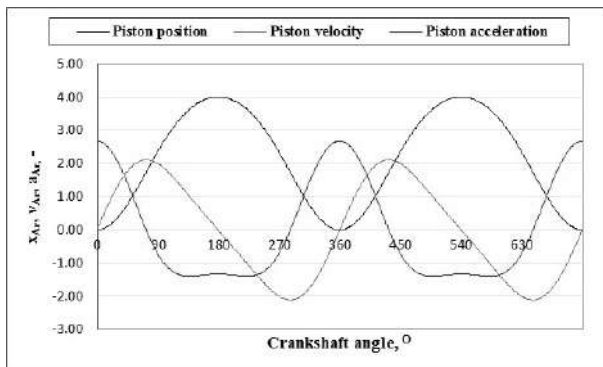


Fig. 7. Normalized piston position, x_{Ar} , velocity, v_{Ar} , and acceleration, a_{Ar}

It is obvious that, as the piston moves from TDC to BDC and back, the velocity is constantly changing, and that the piston velocity is zero at TDC and BDC. The value and location of the maximum velocity with respect to crank rotation are influenced by the connecting rod ratio, λ .

Clearer picture about kinematic behaviour of the observed crank-piston mechanism may be obtained by calculating percentage values of piston movement, velocity and acceleration as functions of crankshaft rotation, Fig. 8. Black line presents piston position as a percentage of piston stroke, blue line presents relative piston velocity as a percentage of maximal velocity during cycle and red line presents relative piston acceleration as a percentage of maximal acceleration during cycle. Positive values of velocity mean that the piston is moving toward the crankshaft; while negative values of velocity mean that the piston is moving away from the crankshaft.

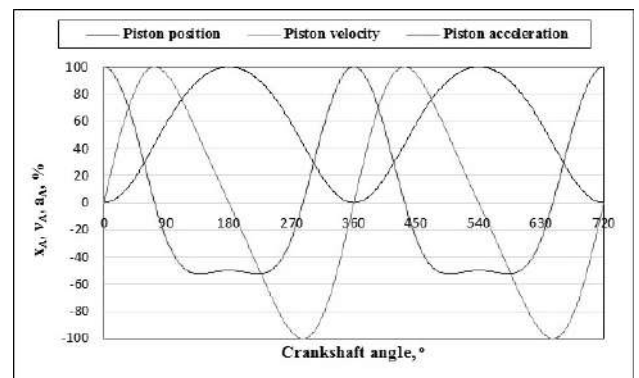


Fig. 8. Percentage values of the piston position, velocity and acceleration

The piston velocity is constantly varying with respect to crankshaft angular position, so, in order to move the piston from the TDC, where velocity equals zero, to the point where velocity is maximal, the piston must be exposed to large acceleration, which also varies with rotation of the crankshaft.

To illustrate the influence of using simplifications like approximation (28) introduced in nonlinear expression for instantaneous acceleration by developing nonlinear functions into power series, a comparative diagram was formed and shown in Fig. 9.

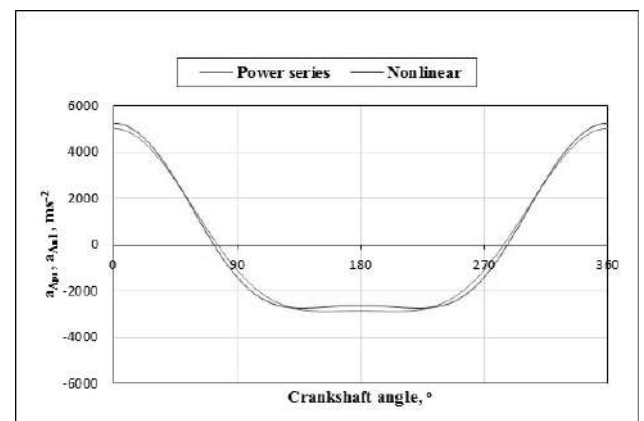


Fig. 9. Comparison between values of piston acceleration obtained by nonlinear relations and by approximation

Introduction of development of nonlinear functions into power series into calculations of the instantaneous piston acceleration has brought maximal relative errors of calculations of 4.3% at TDC and -8.6% at BDC. Thus, the greatest error of approximation of piston acceleration occurs at BDC. It is interesting to observe that there are some apparently high relative errors of calculation of piston accelerations at the crankshaft rotation angle that corresponds to maximal piston velocity. They are the results of mathematical operations (calculated values of acceleration are then very close to zero, but not equal to, which is to be expected) and should not be taken into account. This implies that corresponding improvements in software algorithms and precision of calculations in such situations are necessary.

5. CONCLUSION

Nonlinear geometric problems coupled with dynamic effects in crank-piston mechanisms frequently occur in all types of IC engines. Insufficient knowledge on real kinematics of this mechanism, which kinematically relevant elements (piston, connecting rod and crankshaft) perform various movements, may result in unreliable design of the mechanism.

Knowledge regarding displacement, velocity and acceleration of the crank-piston mechanism is important for determination of inertial influence of each member of the mechanism and of the law of variation of volume of the workspace during engine cycle. Equations describing these quantities consist of nonlinear terms with respect to crankshaft angular displacement. In literature, various forms of simpler mathematical models may be found, where radicals of power series are developed.

The results of analysis of nonlinear kinematic, obtained for the crank-piston mechanism of the Lombardini 3LDA 450 engine show nonlinear behaviour of instantaneous position, velocity and acceleration of the piston. The profile of the velocity curve and the location of the maximum velocity of the piston are both influenced by connecting rod ratio, λ . Maximal positive piston acceleration is obtained when the piston reaches the TDC, while the maximum negative acceleration does not occur at BDC, but symmetrically on either side of BDC.

Another way to observe nonlinear kinematic behaviour of the crank-piston mechanism is to calculate percentage values of piston movement, velocity and acceleration as functions of crankshaft rotation.

Comparison of the results of simulations of kinematic behaviour of the crank-piston mechanism with the results obtained by introducing development of nonlinear terms into power series shows that calculation errors could be made. Maximal calculation errors of piston acceleration occur when piston reaches BDC.

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DYNAMIC ANALYSIS AND PARAMETRIC OPTIMISATION OF THE CONNECTING ROD USING AUTODESK INVENTOR

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Imre KISS

Abstract: This paper presents a case study on optimising a connecting rod, component of a spark-ignition engine, in order to minimise its weight and lower the inertia forces. For this purpose, we use the Autodesk Inventor Professional 2016 (AIP2016) software containing tools that enable either the dynamic analysis to determine the reactions of the connecting rod joints or the finite element analysis for optimising the connecting rod. The result of the optimisation is a lighter and slimmer connecting rod, which implies smaller inertial forces and reduced engine weight, resistant to the stresses subjecting it and ensuring safe operation.

Key words: connecting rod, design, dynamic simulation, optimization, CAD, CAE

1. INTRODUCTION

The connecting rod is the engine mobile part that kinematically connects the crankshaft with the piston.

Due to the working regime and stresses subjecting the connecting rod, the continuous concern for optimising its form, design and operating mode is obvious.

This is highlighted in the papers [1], [2], [3], [4] and [5]. By studying them, we found that the optimisation was performed in order to reduce the connecting rod weight and the inertia forces. In addition, the authors used CAD and CAE software to realise the 3D models and the finite element analysis, these ones enabling the simulation and study of the connecting rod behaviour, in order to be improved.

The purpose of this paper is to conduct a study on optimising a connecting rod, component of a spark-ignition engine with 4 cylinders, for reducing its weight and inertia forces.

As initial data, we considered that the cylinder diameter is 160 mm, the outer diameter of the bolt is 66 mm, and the maximum exhaust gas pressure is 78daN/cm². The

connecting rod is made of steel, with the following mechanical properties: density - 7850 kg/m³, Young's modulus - 210000 MPa, Poisson's ratio - 0.3, elasticity limit - 320 MPa and tensile strength - 610 MPa.

The steps to be taken in achieving the proposed objective are shown in Figure 1.

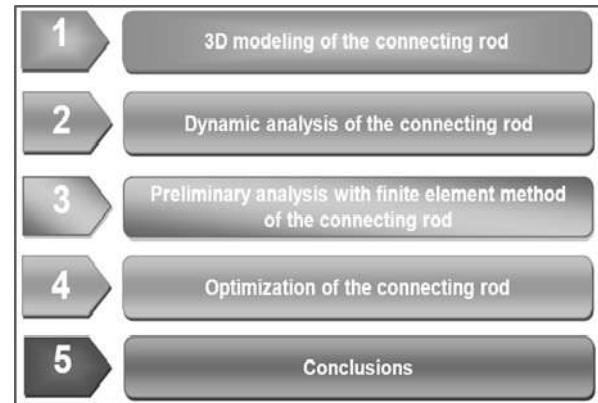


Fig. 1. Steps to take

Based on the initial data and recommendations from literature [6, 7, 8], we established the main dimensions of the component parts of the connecting rod (foot, rod, head and cap), after which we built the 3D model.

For three-dimensionally modelling of the connecting rod, we used the Autodesk Inventor Professional 2016 CAD system, developed by the company Autodesk.

Further, after in the second stage we determined the reactions in the connecting rod joints by using the Dynamic Simulation module from AIP 2016, we conducted a preliminary finite element analysis of the connecting rod using the Stress Analysis module, in order to determine the stresses, total displacement and safety factor.

In the fourth stage, we optimised the connecting rod, establishing the design variables and objective functions; for this purpose, we used again the finite element analysis with the Stress Analysis module, selecting the Parametric Dimensions option.

Finally, the conclusions have been drawn.

2. 3D MODELLING OF THE CONNECTING ROD

The 3D model of the connecting rod, as well as its main dimensions, realised by means of AIP 2016, are shown in Figure 2. It can be seen that we opted for large connection radii between the rod and the head of the connecting rod, for a more uniform and rational distribution of the pressures between the bushings and crankpin, and the stresses in the material. Regarding the connection radius between the foot and rod, we opted for a smaller radius, because the weight increases when the radius is large.

3. DYNAMIC ANALYSIS OF THE CONNECTING ROD

The purpose of this analysis is to determine the reactions acting on the joints of the connecting rod (in its head and foot), considering that the force acting on it is the

maximum force corresponding to the maximum gas pressure. For this, we used the *Dynamic Simulation* module from *AIP2016*.

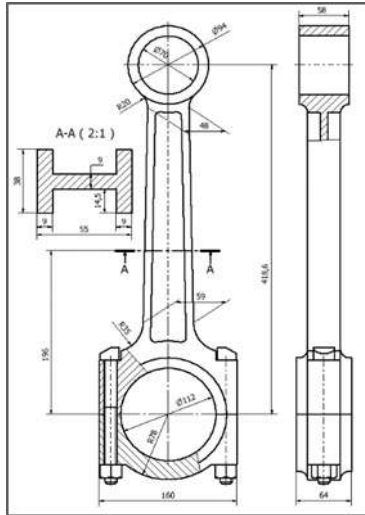
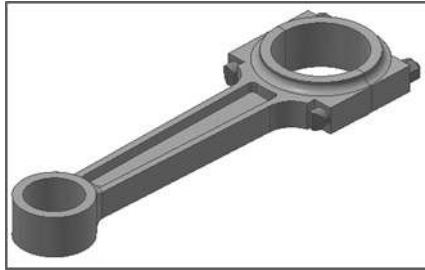


Fig.2. The 3D model of the connecting rod and its dimensions

The dynamic simulation (using the *Dynamic Simulation* module) and the finite element analysis (using the *Stress Analysis* module) are complementary and work well together in *AIP2016*. The dynamic loads applied to a part are determined using the *Dynamic Simulation* module, and then they can be automatically transferred to the *Stress Analysis* module to determine the stresses and displacements. The analysed part is in dynamic equilibrium throughout the dynamic simulation. The finite element analysis is performed at a certain time without taking into account the time variation of the loads [9, 10].

The steps necessary to perform a finite element analysis with loads taken from the *Dynamic Simulation* module are:

1. Calculation of the reactions in joints and inertial forces acting on the part studied with the *Dynamic Simulation* module.
2. Transfer of the loads, determined at the considered time, to the analysed part. The loads consist of reaction and inertia forces, and the part is in dynamic equilibrium.
3. Calculation of displacements and stresses, considering the selected part as being in static equilibrium, by using the *Stress Analysis* module.

3.1. Assembly modelling

To determine the reactions in the joints of the connecting rod, it is necessary to build the assembly whose part it is. For this purpose, the component parts of the *cylinder -*

piston - bolt - connecting rod - crankshaft assembly is three-dimensionally modelled (Fig. 3), and then it is assembled, imposing certain assembly constraints.

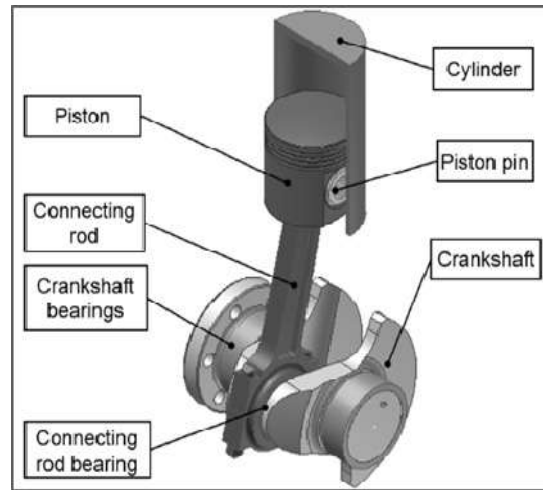


Fig.3. Components of the assembly subjected to dynamic simulation

3.2. Definition of the kinematic couplings

The software is automatically converting the assembly constraints in kinematic couplings, is connecting the degrees of freedom of the immovable parts (*Grounded*) and is grouping the component parts between which there is no relative motion (*Welded*). According to Figure 4, we made five kinematic couplings, whose properties are listed in Table 1.

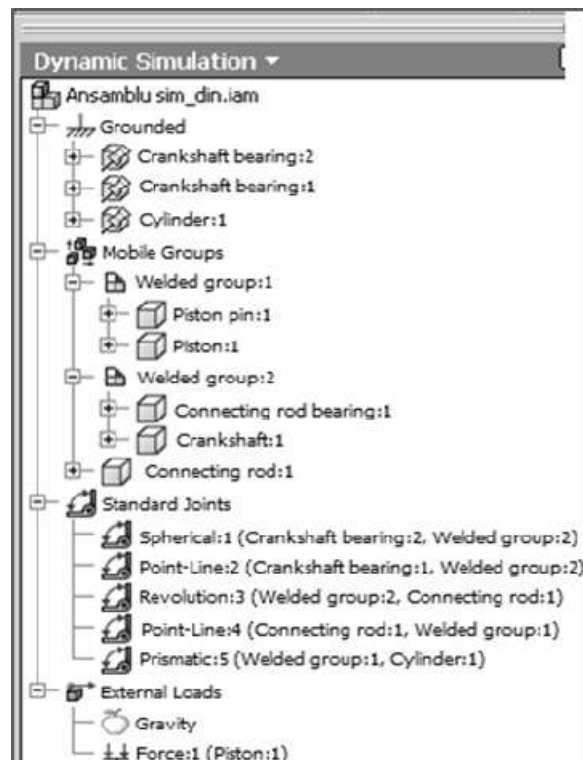


Fig.4. Joints

Table 1. Joints properties

Joint	dof	Damping value	Friction coefficient	Radius (mm)
Spherical 1	dof 3(R)	0,05 Nmm s/deg	0,2	127,56
Point-Line 2	dof 3(R)	0,05 Nmm s/deg	0,2	127,56
Revolution 3	dof 1(R)	0,05 Nmm s/deg	0,2	112
Point-Line 4	dof 3(R)	0,05 Nmm s/deg	0,2	70
Prismatic 5	dof 1(T)	0,05 Ns/mm	0,2	–

3.3. Definition of the environment

At this stage, we add information about the gravitational force and external forces.

It is considered that the maximum force developed by the gas pressure is acting on the piston:

$$F = \frac{\pi D^2}{4} \cdot p_{g \max} = \frac{\pi \cdot 16^2}{4} \cdot 78 = 15682 \text{ (daN)} \quad (1)$$

where $D=16$ cm represents the cylinder diameter, and $p_{g \max}=78 \text{ daN/cm}^2$ –maximum gas pressure.

The force is applied to the piston head, in the direction of the cylinder axis and oriented towards the piston.

As the force developed by the gas pressure must act only at the expansion stroke (downward), we imposed an application restriction to this force depending on the piston speed and crankshaft rotational speed. Thus, the force will act only when the piston speed is less than or equal to 0 mm/s and the crankshaft rotational speed is below 3000 rpm, in this way limiting also the crankshaft rotational speed [10].

3.4. Results

In this stage, it is generated the information about simulation, including positions, speeds, accelerations, reactions, forces and momentums.

After running the simulation, we determine the reactions in the kinematic couplings corresponding to the connecting rod (*Revolution: 3* - head and *Point-Line: 4* - foot), and we identify the times at which they are maximal. In this situation, the two times are 0.07s and 0.075s (Fig. 5), and they are exported for analysis using *Stress Analysis*.

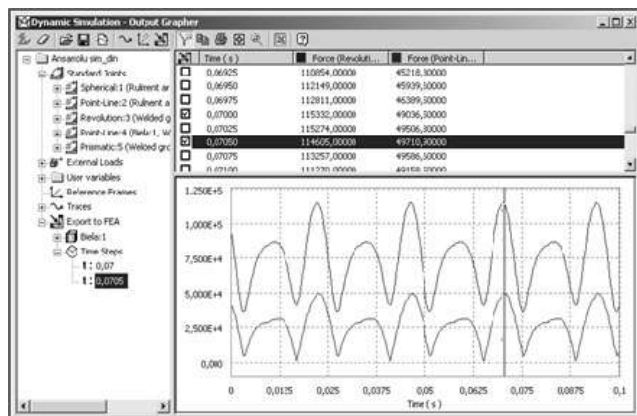


Fig.5. The reactions in the connecting rod joints

4. PRELIMINARY FINITE ELEMENT ANALYSIS OF THE CONNECTING ROD

For the finite element analysis of the connecting rod, we used the *Stress Analysis* module implemented in *AIP2016*. It enables the assessment of the part or assembly in operation, analysis of stresses and displacements based on several resistance theories, determination of the safety factor, integration with the dynamic simulation module, transfer of loads and constraints determined in the dynamic simulation module, animation of the obtained results, etc.

Since the values of the reactions afferent to the two points in time are very close, we are going to perform the analysis only for the time 0.07s, i.e. the reaction in the joint of the connecting rod head is maximal (115274 N) and the reaction in the joint of the connecting rod head is 49036 N). The loads, taken from the dynamic analysis, are: the part weight (*Gravity*), force and moment of inertia (*Body Loads*), reactions in joints (*Remote Force*) and moments in joints (*Moment*) – Fig. 6.

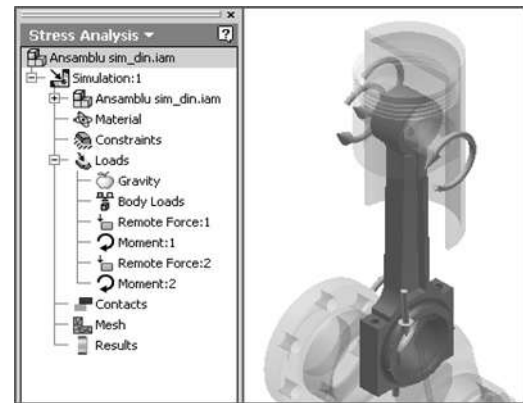


Fig.6. Loads imported from Dynamic Simulation

After running the analysis, the results are shown as von Mises stresses (Fig. 7), total displacement (Fig. 8) and safety factor with respect to the yield strength (Fig. 9).

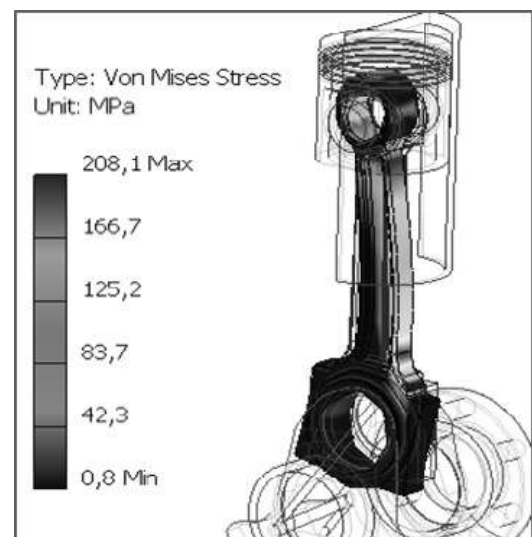


Fig.7. Von Mises Stress

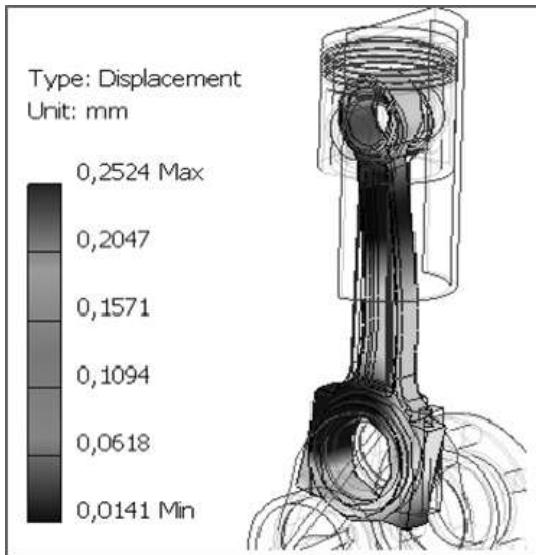


Fig.8. Total deformation

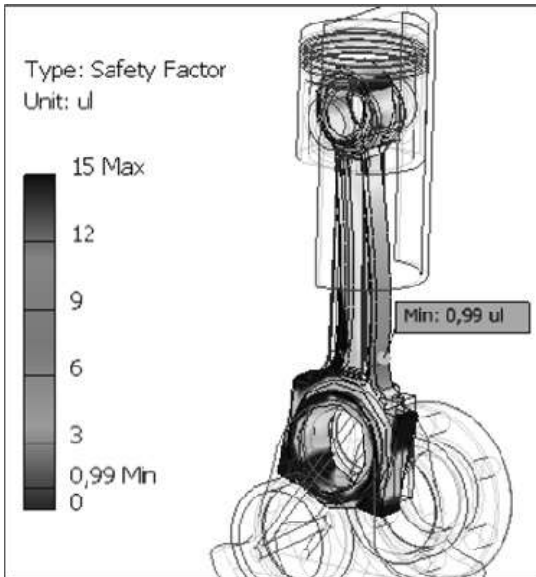


Fig.9. Safety Factor

6. PARAMETRIC OPTIMIZATION OF THE CONNECTING ROD

The optimisation can be performed quickly and efficiently using the *Stress Analysis* module, found in *AIP2016*, and selecting the *Parametric Dimension* option.

The method involves the selection of a number of parameters (design variables), imposition of a variation range and determination of their values for which the design constraints are met (objective functions).

These ones can be:

- Stresses, displacements, deformations lower than the allowable ones or included in a certain range of values;
- Minimum weight;
- Minimum volume;
- Safety factor higher than a setpoint; etc.

6.1. The design variables

The design variables are established in the three-dimensional modelling phase, and are used to generate the geometric shapes that make up the connecting rod.

To optimise the rod, we chose as design variables the values given in Figure 10.

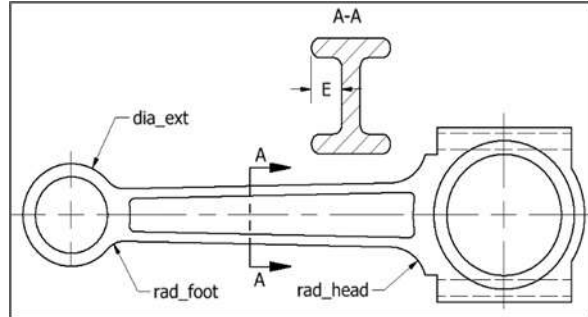


Fig.10. Design variables

The ranges are selected in accordance with the conditions imposed to the connecting rod (mounting, maximum size), etc. In this case, the ranges of the design variables are listed in Table 2.

Table 2. The design variables for connecting rod

Design variables	Parameter	Initial value	Range	No. values
Radius of the connecting rod foot	rad_foot	20 mm	16-24	5
Radius of the connecting rod head	rad_head	35 mm	30-40	5
Outer diameter of the connecting rod foot	dia_ext	94 mm	92-96	3
Free length of the piston rod rack	E	14,5 mm	14-15	3

6.2. The objective functions

The objective functions are established in such way to meet the design requirements. In our case, we opted for:

- Minimisation of the connecting rod weight;
- The safety factor in relation to the minimum yield strength to be less than or equal to 1.2;
- The total displacement to be minimal.

6.3. Results

After conducting the simulation, we obtained the values of the design variables which satisfy the objective functions. In Figure 11, which presents the results obtained for the optimised connecting rod, we see that all the objective functions are fulfilled.

The parameter values, for which the objective functions are fulfilled, are given in Table 3.

Parametric Table					
Parametric Table					
Design Constraints					
Constraint Name	Constraint Type	Limit	Safety Factor	Result Value	Unit
Mass	Minimize			✓ 8,49401	kg
Min Safety Factor	Lower limit	1	1,2	● 1,46738	ul
Max Displacement	Minimize			✓ 0,26497	mm
Parameters					
Component	Feature Name	Parameter Name	Values	Current Val.	Unit
Con rod		rad_foot	16 - 24:5	16	mm
Con rod		rad_head	30 - 40:5	30	mm
Con rod		dia_ext	92 - 96:3	92	mm
Con rod		E	14 - 15:3	15	mm

Fig.11. Results after simulation

Table 3. The optimal values of the design variables

Design variables	Parameter	Initial value	Optimal value
Radius of the connecting rod foot	rad_foot	20 mm	16 mm
Radius of the connecting rod head	rad_head	35 mm	30 mm
Outer diameter of the connecting rod foot	dia_ext	94 mm	92 mm
Free length of the piston rod rack	E	14,5 mm	15 mm

In Figures 12, 13 and 14, we plotted the values afferent to von Mises stress fields, the total displacement and the safety factor for the optimal values of the design variables.

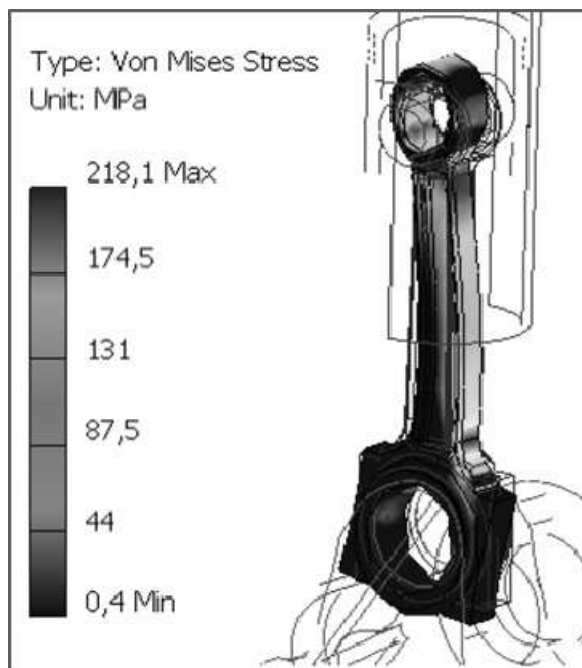


Fig.12. Von Mises Stress of the optimized connection rod

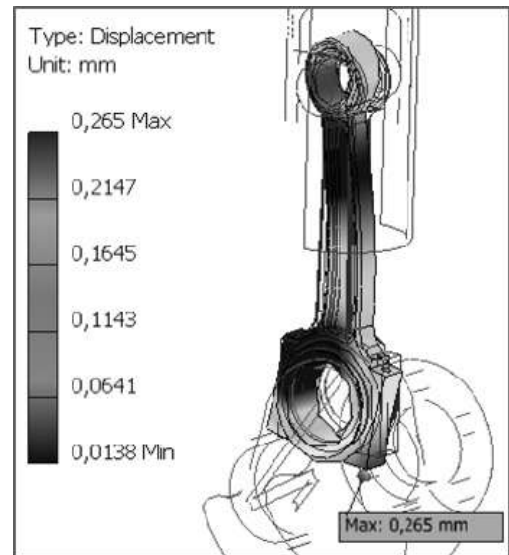


Fig.13. Total displacement of the optimized connecting rod

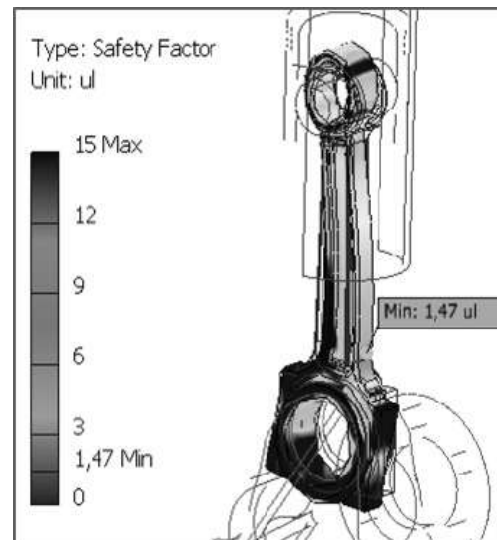


Fig.14. Safety Factor of the optimized connecting rod

7. CONCLUSIONS

By comparing the two versions (preliminary and optimised), we found the following:
The weight of the optimised connecting rod decreased by 0.241 kg compared to the preliminary version, which means that the optimisation goal, from this point of view, has been reached.

Table 4. Comparison between the preliminary and the optimized connecting rod

Objective function	Versions		Difference
	Preliminary	Optimized	
Mass	8,735 kg	8,494 kg	0,241 kg
Safety Factor	0,994	1,467	-0,473
Maximum total displacement	0,252 mm	0,264 mm	-0,012 mm

The minimum safety factor relative to the yield strength for the optimal variant is higher than the imposed one (1.2), so this imposed objective function has been fulfilled.

Compared to the preliminary version, the maximum total displacement afferent to the optimised version is higher, but its value is within the imposed limits found in the literature.

By using the dynamic simulation with the module *Dynamic Simulation*, we waive many simplifying assumptions taken into account in the classical method of calculation. Practically, by dynamic simulation we obtain the reactions in the kinematic couplings at a certain time, without taking into account the variation in time of the loads, which are then transferred to the *Stress Analysis* module to calculate the stresses and displacements.

In this paper, we determined the moments in time when the reactions in the kinematic couplings are maximal, and then we performed the finite element analysis.

In conclusion, the activity of connecting rod optimisation has reached its goal, resulting in a lighter and slimmer part, which implies smaller inertial forces and reduced engine weight, it is resistant to the stresses to which it is subjected and ensures safe operation.

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NEW INTERNAL COMBUSTION ENGINE

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Abstract: In this article was presented one idea of how to improve overall internal combustion engine efficiency. We try to make a brief description of most important and basic parts of a new internal combustion engine. Described engine have several advantages over conventional IC engine.

Key words: IC engine, motor vehicle, CAD

1. INTRODUCTION

Today, there are a very large number of successful constructions of internal combustion engines, which are applied to various fields of science and technology. In some areas IC engines are so dominant without concurrence of other types of engines. This fact suggest that today's internal combustion engines are at a high technical level. However, construction of piston stroke internal combustion engines that are now used is based on inefficient thermodynamic and mechanical concept. It can be said that the main characteristics of today's engine is very small amount of work in relation to used fuel, in other words, today's engines have a very low coefficient of efficiency. Realistically speaking Otto engines today use about 25% of input energy, while diesel construction about 30% (in some cases can be expected a little more). Approximately 35% of the petrol engine and 30% of heat in the diesel engines goes through exhaust and around 33% goes for cooling the engine in both versions, other 7% is attributed to friction and radiation [1]. For illustration can be taken into account combustion of one liter of fuel in the classical combustion engines. Combustion of this amount of fuel frees approximately 39 MJ of power, the engine output shaft is generated only around 13 MJ, while with other 26 MJ engine heated environment (through exhaust and cooling system). Conventional IC engines are based on a relatively simple solution to achieve a thermodynamic cycle while providing mechanical power. While the performance, emissions, and reliability of IC engines have been

improved significantly, the fundamental principle of crank-rod-piston slider mechanism still remains largely unaltered. In theory, the most efficient thermodynamic cycle for IC engines is the Otto cycle [1,2], which consists of isentropic compression and expansion processes and constant volume heat addition and rejection processes [3, 4]. It is generally known that the most important parts of the cycle which determine the efficiency are the constant volume heat addition at high compression ratios [5, 6]. This fact provides the highest thermal potential of the various possible thermodynamic cycles which are suitable for IC engines, and the subsequent expansion process, which converts the thermal potential into work. In reality, neither conventional spark ignition nor compression ignition or even the modern developed homogeneous charge compression ignition or controlled auto ignition combustion processes, can achieve the efficiency level suggested by the ideal thermodynamic cycles [7]. Only the Otto cycle delivers theoretical maximum thermal efficiency. The traditional design of internal combustion engines using a simple slide-crank mechanism gives no time for a constant volume combustion which significantly reduces the cycle efficiency [1].

2. CONVENTIONAL I.C. ENGINE

It is well known that ordinary IC engines are based on a slider-crank mechanism. This way of piston motion provides a relatively simple solution to achieve a thermodynamic cycle to produce mechanical power. As can be seen from theory, the most efficient thermodynamic cycle for IC engines is the Otto cycle [8-12], because the constant volume heat addition is essential for high efficiencies. Common to most reciprocating engines is a linkage known as a crank-slider mechanism [12]. Slider-crank mechanism was shown in figure 1, this mechanism is one of several capable of producing the straight-line, backward-and-forward motion known as reciprocating. Fundamentally, the crank-slider converts rotational motion into linear motion, or vice-versa. With a piston as the slider moving inside a fixed cylinder, the mechanism provides the vital capability of a gas engine: the ability to compress and expand a gas. One of the major disadvantages of the conventional linkage is the fact that this setting produces very limited motion of piston (large changes of volume during combustion) and high mechanical losses due to the friction between piston and cylinder liner. Friction between piston and cylinder is the biggest source of friction in ordinary engine, more than half of all mechanical losses came from contact between parts like piston, rings and cylinder.

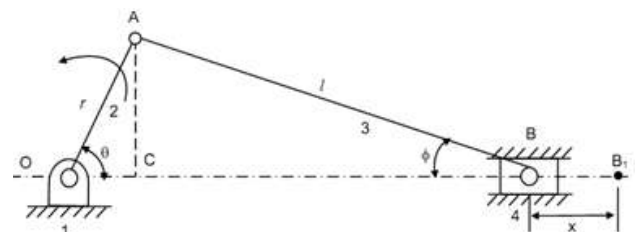


Fig.1. Slider-crank mechanism

3. NEW I.C. ENGINE

Real engine is not able to make heat addition during constant volume. When the engine operating, the piston can only reciprocate continuously between top dead center (TDC) and bottom dead center (BDC) at a frequency proportional to the engine speed. The chemical reaction process associated with combustion events, however, essentially takes a fixed-time to complete, which is relatively independent of the engine speed. In order to maximize the work obtained from the heat energy released by combustion, the air/fuel mixture has to be ignited prior to the piston reaching TDC, and the ignition timing should be adjusted according to the engine speed and the quality of the air/fuel mixture. Clearly, the early stage of the heat release before the piston reaches TDC results in negative work. During the combustion event, the piston movement is defined by the crank rotation, so that truly constant volume heat release is not achievable. The ideal scenario is to initiate and complete the combustion event while the piston remains at the TDC position. A practical method is to reduce the engine crank rotation velocity at the TDC position to provide extra time for completing the combustion. This will then generate a new combustion cycle, quasi-constant volume (QCV), that sit between conventional IC engine combustion cycle and ideal Otto constant volume combustion cycle. Such unconventional SI engine is presented on fig. 2. As can be seen from the described illustration pistons make a movement conditioned by the mechanism consisting of crankshaft and connecting disc. In this article will not be presented detailed description of this concept, since it is not the intention of the authors to propose a kinematic analysis of a new internal combustion engine design but only brief features and advantages over ordinary SI engines. Dwell time or dwell angle is important fact during combustion process. In conventional engine this dwell angle can be changed due to variations of ratio between connecting rod and crank radius. Piston dwell at TDC and at BDC are often mentioned, it should be noted that strictly, there is no dwell period in ordinary mechanism. The piston comes to rest at precisely the crank angle that the crank and rod are in line (TDC and BDC), and is moving at all other crank angles. At crank angles which are very close to the TDC and BDC angles, the piston is moving slowly. It is this slow movement in the vicinity of TDC and BDC that give rise to the term piston dwell. In this described concept there is also no piston dwell in classical sense, there is only very small changes of volume near TDC. If the piston dwells longer near TDC and ignition is initiated properly, there will actually be a longer period of time for the pressure created during combustion to press against the top of the piston. Also, if the dwell period is too long, there is a possibility for unfavorable energy conversion. In real engine ideal Otto cycle have certain disadvantages because there are a lot of heat losses in case when full combustion is obtained in TDC. Optimal heat addition should be found somewhere between pure constant volume combustion and combustion at variable volume-quasiconstant volume combustion.

3.1. Main engine parts

Main engine parts of a new concept are shown in next figures. As can be seen from Fig. 2 main parts are: 1-engine block, 2-engine head (with valves and camshaft), 3-flywheel, 4-crankshaft, 5-double acting piston, 6-piston rings, 7-piston disc and 8-cylinder liner.



Fig.2. Main engine parts

Basic engine design is very similar to well known boxer or flat engine or even rare V-180°. These similarities can be seen from next Fig 3. where is presented flat engine with several parts: 1-engine block, 2-engine head and 3-flywheel.

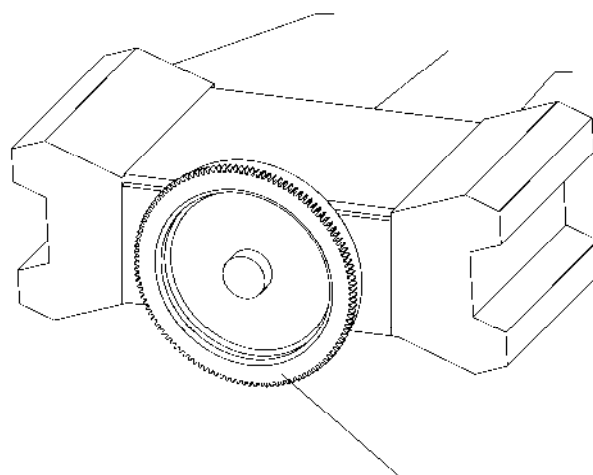


Fig.3. View on new flat engine

Many engine parts are almost the same as we can see in today's internal combustion engine. For example engine block and engine head (as described in Fig.4) are basically the same as in ordinary boxer engine or V-180° engine. In this paper will not be presented parts like camshaft and valves because these parts are completely identical like in ordinary IC engine. Focus will be placed only on parts and assemblies which are significant different than in conventional piston engines.

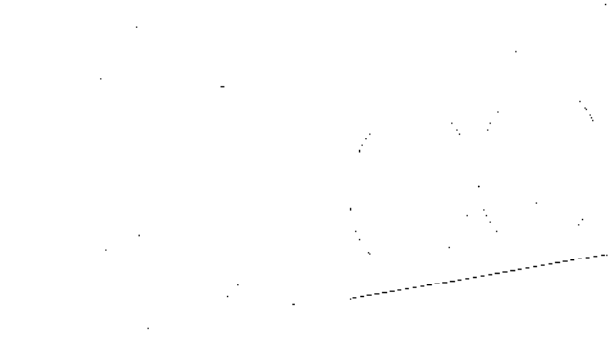


Fig.4. Engine block and engine head

One of the most important parts in every engine is the crankshaft. In this engine design crankshaft is shaped in such way that is able to connect with piston via gear teeth. As an ordinary crankshaft this one also have journal bearing which is necessary because of connection with piston disc. On Fig.5 was presented new crankshaft with gears and flywheel. Gears and crankshaft have rigid connection.

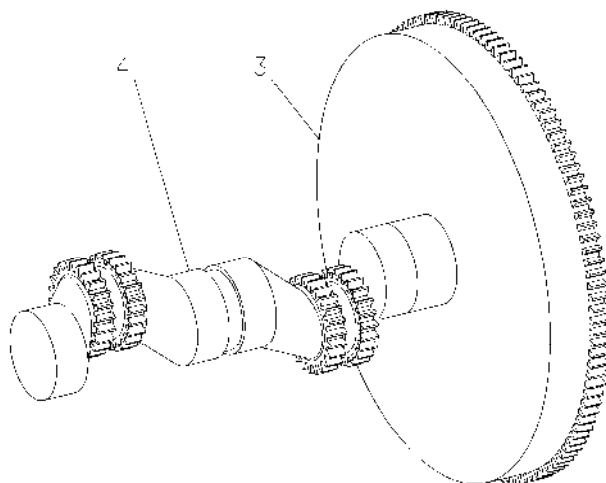


Fig.5. Crankshaft of new IC engine

Double acting piston is one of the most important features of new engine design. One of the possible ways for piston design is described in the following Fig. 6. As can be seen besides two piston head these pistons also have inner gearing. Teeth on the inner gearing are coupled with gearing on crankshaft. Assembly of piston, piston disc and crankshaft is described with Fig. 7. On the top of the both piston side are mounted piston rings (compression and oil). Flywheel is mounted on one side of the crankshaft like in ordinary engine and has exactly the same functions (accumulation of kinetic energy, engine starting, torque, power and motion transfer).

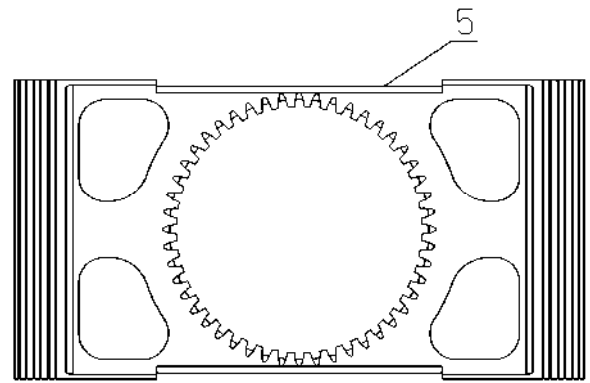


Fig.6. Double acting piston

Coupling of piston and crankshaft through piston disc and gears is presented in following Fig. 7.

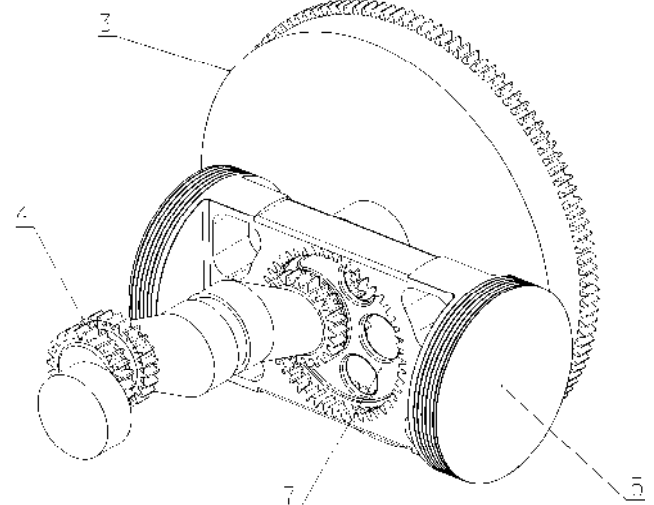


Fig.7. Coupling of piston and crankshaft through piston disc and gears

The idea is to eliminate normal force on cylinder wall. Normal force is product of piston mechanism design and one of the main reasons for high values of mechanical losses in piston engines. Normal force from cylinder surface can be transfer into forces on elements like piston disc (Fig. 8). Piston disc is fully hydrodynamic lubricated and the total friction force is very small in relation to friction between piston and cylinder liner. Piston disc is placed between crankshaft and piston, hydrodynamic lubrication was achieved on both surface, first is in contact with crankshaft and the second is in contact with piston. Contact surfaces can be seen from Fig 8.

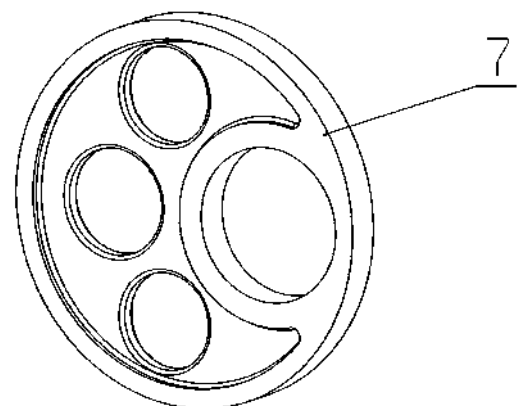


Fig.8. Piston disc

4. ADVANTAGES

A major source of engine wear is the sideways force exerted on the piston through the connecting rod by the crankshaft, which typically wears the cylinder into an oval cross-section rather than circular, making it impossible for piston rings to correctly seal against the cylinder walls. Geometrically, it can be seen that longer connecting rods will reduce the amount of this sideways force, and therefore lead to longer engine life. However, for a given engine block, the sum of the length of the connecting rod plus the piston stroke is a fixed number, determined by the fixed distance between the crankshaft axis and the top of the cylinder block where the cylinder head fastens. Thus, for a given cylinder block longer stroke giving greater engine displacement and power, requires a shorter connecting rod (or a piston with smaller compression height), resulting in accelerated cylinder wear. Idea is to eliminate normal force at all, in this engine design it is possible to reduce mechanical losses with transfer normal force on cylinder wall into the force on gear teeth and force on piston disc which have hydrodynamic lubrication (very low friction). Also such engine design is able to provide such piston motion where heat addition can be done during smaller changes of volume when combustion occurs.

5. CONCLUSIONS

In this article was presented one approach for improvement of spark ignition engine efficiency. Described concept has several advantages over ordinary SI engines. All of these mentioned advantages show that the potential to increase the efficiency of the SI engine conditions is not yet exhausted. The engine used for most contemporary motor vehicles is the four-stroke SI internal combustion engine, in that sense in this paper is presented such engine. A novel Otto cycle engine concept in which intake and compression are carried out through unconventional piston mechanism is presented in this paper. With longer piston dwell near TDC and eliminating normal force on cylinder wall it can be expected that thermal efficiency and mechanical efficiency will be increased. It can be noted that this engine concept have lower number of parts than ordinary boxer engine. It is clear that the efficiency of modern IC engine can not be much increased, which is also one of the reasons for the development of new propulsion systems. However, at the time in which every year world produces a large number of vehicles, where there is still no real alternative, a minimum improvement of any segment of today's engines will certainly be felt on a global level.

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STUDY ON BEHAVIOUR IN SERVICE OF DIESEL ENGINES AND ASPECTS CONCERNING THEIR MAINTENANCE

Olimpia COROIAN

Abstract: Currently, the most used diesel power network Romanian Railways (C.F.R.) uses only 060-DA / DA1 locomotives, it proved to be the most reliable, over the years being brought him numerous modernizations. Maintenance and repairs of them ensure traffic safety and are proved to be important cost elements, which is why we will study and analyze in detail.

Key words: maintenance, diesel, inspection, repairs

1. INTRODUCTION

Locomotive diesel-electric 060-DA is equipped with an engine SULZER having 12 cylinders mounted on two distributions in line and arranged vertically mounted on the locomotive box locomotive, the space between the two management positions, being insulated and soundproofed to these.

2. STUDY ON BEHAVIOR IN SERVICE OF DIESEL ENGINES FOR LOCOMOTIVES

After some time running locomotive engine wear and defects that diminish their performance and can endanger traffic safety unless remedial measures are taken. Maintaining locomotives in running safely is by execution time and at a quality level corresponding revisions and planned repairs, moments in which eliminates the normal wear of the component parts. In Romania the type hauls and repairs planned faced rail vehicles succession of their norms of time (days, months, years) or norms of kilometers from railway vehicles are out of service for revisions and planned repairs are set by ministerial order 1359 / 30.08.2012, namely the railway norm "railway vehicles. Types of planned inspections and repairs. Time and mileage standards for revisions and planned repairs. " According to current legislation, any diesel electric locomotive has a service life of 35 years, during which inspections and repairs runs as follows:

Revision daily, as its name says and runs daily, respectively after a total of 18 diesel engine is running; Technical inspection of repair, RT, running every 60 days, regardless of the number of kilometers traveled; Revision Type 1 R.1, runs every 180 days, regardless of the number of kilometers traveled; Revision type 2 R2, runs every 360 days, regardless of the number of kilometers traveled; Every 720 days, regardless of the number of kilometers traveled is running a new revision of type 2 R2; Revision type 3 GR.3, runs every 1080 days, regardless of the number of kilometers traveled; Lift repair, RR, runs every 4 years or 330,000 kilometers traveled and Overhaul runs every 8 years or 660,000 kilometers traveled (RG - overhaul); Capital repair sheet usually occurs after a lapse of 16 years, to a maximum of 1.32 million kilometers traveled.

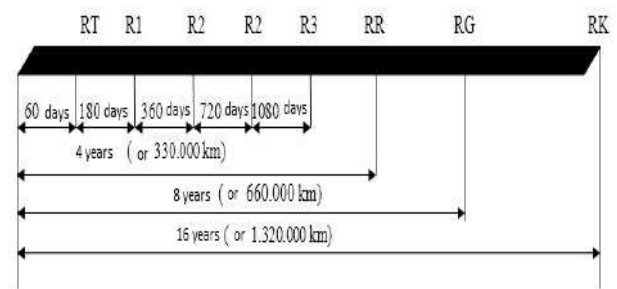


Fig.1. Scheme diesel-electric locomotives revisions

Maintaining operational locomotives have exceeded the normal service is performed only after the vehicles have undergone technical assessments to establish the technical condition of the structure strength (chassis, chassis). Evaluation of technical condition of vehicles is made in relation to a repair with lifting axle / bogies. Given the multitude of parts and complexity of diesel 12 LDA 28, we will focus below on usages problems, overhauls and repairs related to some mobile components, namely the diesel engine crankshaft.

3. CRANKSHAFT

12 LDA 28 diesel engines crankshaft is made from alloy steel with Cr, Mo and Ni, having the following composition: 0.36% C, 0.20% Si, 0.50% Mn, 0.20% Mo, 1 % Ni, 1% Cr, up to 0.035% P, max 0.35% S, max 0.06% P + S, and up to 0.25% Cu [2].

The mechanical characteristics of the shaft are $\sigma_r = 80..100 \text{ kgf/cm}^2$, $\sigma_c = \min 60 \text{ kgf/cm}^2$, $\delta_5 = \min 13\%$, $K = \min 6 \text{ kgf/cm}^2$ and $\sigma_{-1} = 36 \text{ kg/cm}^2$ [2].

In the following, based on data obtained from the SC Remarul February 16 Cluj Napoca, we emphasize the service behavior of the crankshaft.

SC Remarul February 16 SA is one of the companies based in Cluj, the primary activity is the construction, modernization and repair of rolling stock railway, it is a certified and accredited to carry out repair work by international institutions.

Both during and after manufacture revisions and repairs especially crankshaft measurement is essential.

The measurement is made for both the shafts conventional method, taking as reference the line of the geometric axis

of the shaft, and by using the optical apparatus, as shown in fig. 2.

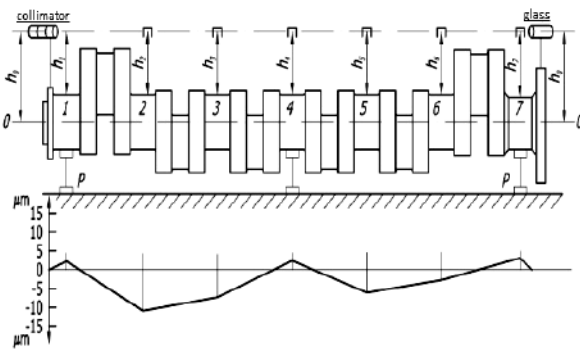


Fig.2. Checking the crankshaft using optical device

With this method are measured irregularities parallel spindles and spindles crankpins levels but the shaft axis. For checking the coaxially, the crankshaft is put on the plate to draw, media hydraulically adjustable, assembled in journals 1, 4 and 7. With the aid of these supports, and the levels of the comparators, the crankshaft is placed with the axis of rotation at the theoretical horizontally, is parallel plate traced. Check axial deviation from the spindles by using comparator. Bring to a comparator foot spindle rotation axis in the plane and is set to 0. Then the spinning shaft and document the indications comparator. The procedure was identical all time.

After checking coaxially spindles move to check sleeves. For this crankshaft rotates until the vertical plane of symmetry passing through the crankpin merges with the plane of symmetry of the tree. In this position comparator goes on top of the generator crankpin and measure some distance connection (measured template) is adjusting to comparator 0. Deviation is measured from the other end of the generating line. Read deviation indicates crankpin generator inclination towards the geometrical axis of the shaft. Also measured and tilt the other side, which brings upright. By rotation of the shaft through all bends are brought into the vertical position and inclination measured sleeves.

Shaft spindles wear mostly due to the engine crankshaft Sulzer Type 12 LDA 28 may present operational following irregularities [1]:

- Wear maximum crankshaft journal or crankpin - 0.05 mm
- Ovality in journals and crankpins - 0,010 mm
- Taper in spindles and crankpins - 0,010 mm
- Bending maximum operating shaft - 0.095 mm
- Deviation from parallelism relative spindles - 0.15%

RR reparation check, with the shaft in assembled, each bearing game. If the game engine crankshaft bearings Type 12 LDA 28 is greater than 0.197 / 0.296 mm, respectively bearing should be replaced. If uneven wear is higher and to all the camps we can say that the shaft is deformed.

A new crankshaft can have a maximum deflection of 0.03 mm, and by mounting to the engine, the bending does not exceed 0.05 mm. Due to the deformation, rotation occurs during opening and closing of the arms crankpins and

journals rotating about an axis inclined to the geometrical axis of rotation of the shaft.

Checking during rotation shaft alignment is done by measuring the variation in the distance between the crank arms. It is considered permissible change of distance less than 0.04 mm. In case of variation between 0.04 ... 0.074 mm, recommended realignment re shaft and over 0,075 mm alignment operation is required.

During general repairs, RG, is usually running cleansing by grinding spindles and crankpins and remediation of deviations from parallelism in their relative. If the required correction (spindles authorized, slotted, etc.) operation is running at the same diameter gear wear all time and crankpins. Correction runs concentric with the geometrical axis of rotation of the crankshaft. If crankshaft brought for examination and repair in the repair plant SC Remarul SA February 16, these are rectified in most cases (over 90%) in figure 3 being presented grinding machine that is running the operation.



Fig.3. Grinding machine SC Remarul February 16 Cluj Napoca

If and crankpin bearing spindles 2100 HP diesel engine, repair rates are:

Table 1. Quotas repair, diesel engine 2100 HP

Repair share	Bearings (mm)		Crankpin (mm)	
	Max	Min	Max	min
A	205,000	204,935	190,000	189,935
B	204,000	203,935	189,000	188,935
C	202,500	202,435	187,500	187,435
D	200,000	199,935	185,000	184,935

Between 2002-2015 DA 001 engine, a 2,100 horsepower engine, was refurbished in SC Remarul February 16 SA in engine reconditioning ward 5 times, each time when conducting major repairs and not during periodic reviews. Operations undertaken during the 2 shaft refurbishment engine DA001 distributions are shown in Table. 2.

Inconsistencies occur in the evolution gear shaft repair study explains that servicing and repair operations on diesel locomotive depots beneficiaries are carried out both in our case C.F.R. and the repair plants. Thus, the first refurbishment carried out in the framework SC Remarul February 16 SA in July 2002 until the second time the engine was brought in factory repair, crankshaft on the distribution I engine was replaced at the beneficiary depot.

Also between July 2011 and April 2013 the crankshaft in the second distribution was replaced at the beneficiary depot.

Table 2. Records of operations on the engine crankshaft DA001

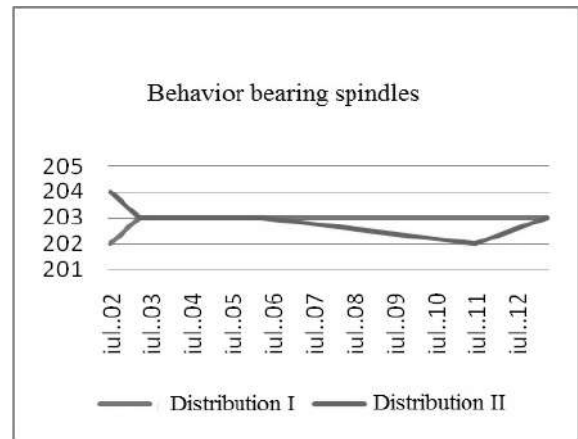
Date	Distribution I		
	Share spindle bearing, mm	Crankpin time share, mm	Operations performed
July 2002	202 (C)	188 (B)	Crankshaft lapping
April 2003	203 (B)	187 (C)	Crankshaft lapping
April 2006	203 (B)	188 (B)	Landing time lapping, Crankpin grinding spindle
July 2011	203 (B)	188 (B)	Lapping line Bearing spindles and crankpins - corrected
April 2013	203 (B)	188 (B)	Time lapped bearing Crankpins rectified
	Distribution II		
	Share spindle bearing, mm	Crankpin time share, mm	Operations performed
July 2002	202 (C)	188 (B)	Crankshaft lapping
April 2003	204 (B)	189 (A)	Crankshaft lapping
April 2006	203 (B)	188 (B)	Crankshaft lapping
July 2011	202 (C)	188 (B)	Landing time lapping, Crankpin grinding spindle
April 2013	202 (C)	188 (B)	Bearing spindles and crankpins - lapped
May 2013	203 (B)	188 (B)	Landing time lapping, Crankpin grinding spindle

Share developments and crankpin bearing spindles D001 locomotive engine, the period studied is shown in Chart1.

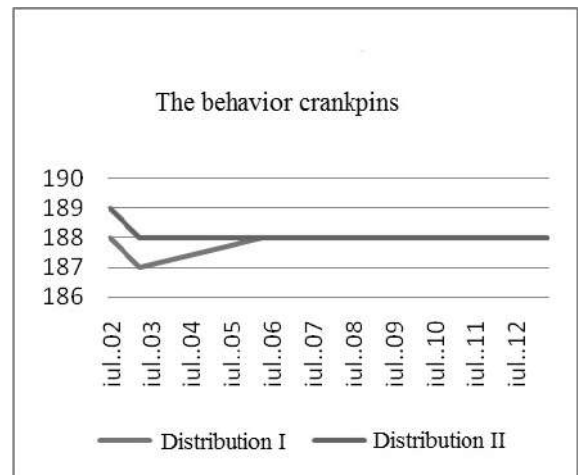
4. CONCLUSION

Studying the behavior in service of diesel engines crankshafts of composition, we can conclude that they have a long operating cycle, as desired in the phase of manufacturing. Friction factors that contribute to the wear of the crankshaft are considered both at the manufacturing stage and during maintenance.

To optimize maintenance and repair propose drawing up a maintenance plan customized for each locomotive in hand, plan the work done both in depots and in factories reparation be linked and have fluency in order to reduce maintenance costs and increase traffic safety.



a)



b)

Fig.4. a) Behavior bearing spindles;
b) The behavior crankpins

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GASODYNAMIC STUDY OF THE INTAKE ROUTE AT A SPARK-IGNITION ENGINE

Sorin RATIU
Vasile ALEXA

Abstract: *This study aims at determining, by experiment, the pressure loss occurring on the intake route of a spark-ignition engine with carburettor. For this purpose, a pilot plant was designed for measuring the pressure at various points on the route, simulating a stationary air flow regime by means of a vacuum pump. Measurements were made for various lifting heights of the intake valve and various opening positions of the throttle body.*

Key words: *intake valve, pressure loss*

1. INTRODUCTION

The components of fresh load entering the engine cylinders during the intake stroke are the fresh air and – in systems with external mixture formation – the fuel (gasoline), in the form of vapours, found in suspension in it. Most of the amount of fresh air enters the cylinders through the channel controlled by the throttle. Additionally, the fresh load can be sucked into the engine through the vapour recirculation system, if it exists. The air mass found in cylinders after closing the intake valve is the decisive factor in relation to the mechanical work produced by cyclically burning of the fresh load, with direct influence on the engine torque. Consequently, the measures to increase the maximum torque and the maximum power almost always involve creating the conditions for obtaining a maximum possible filling ratio [1].

When using carburettors as equipment for achieving the optimal dosage of the fuel mixture, the intake system was characterised by manifolds able to feed one or more cylinders. As a result of its geometry, this system has significant losses of fresh load, leading to lower maximum power and momentum.

On the intake route, two kinds of losses are recorded.

- *pressure losses or gasodynamic losses* caused by the existence of hydraulic resistances on the suction line, which can be quantified using the well-known relation:

$$\Delta p = \xi \cdot \rho \cdot \frac{w^2}{2} \quad (1)$$

where ξ is the pressure loss coefficient characteristic to each component of the intake system, w is the flow rate of the fresh fluid, and ρ is its density.

- *Heat losses* caused by fluid heating from the suction line walls, which determine the final temperature to be higher than the engine inlet temperature, the temperature increase resulting in density decrease and, hence, affecting the filling.

To establish a single criterion for quantifying the filling performance, we must compare the amount of load that actually enters the cylinder with the one that might enter if there were no losses.

The *filling degree or filling ratio*:

$$\eta_v = \frac{C}{C_0} \quad (2)$$

is the ratio between the load *actually retained* in the cylinder (C), after closing the last device for stopping the access of the engine fluid to and from the cylinder, and the amount of load *that might be retained* in the cylinder (C_0), in the condition recorded when entering into the engine, per cylinder capacity unit [2].

This study focuses only on the loss of pressure caused by the existence of hydraulic resistances, for finding them by experimental measurements and highlighting their effects on the process of filling the engine cylinders with fresh fluid.

2. LOSS ON THE ENGINE INTAKE ROUTE

The fuel mixture is flowing through the intake system under the effect of the depression created by the downward movement of the piston from TDC to BDC, in conjunction with the inertial flow effect and the undulating effect of the pressure waves created by the opening and closing valves. At partial throttle openings, while behind it there is a significant depression, the flow is continuous and not pulsating. At large throttle openings, in which case the flow restrictions are minimal due to the effect of acoustic waves propagation, the movement has a pulsating character.

The size of depression in the sucked engine cylinder depends primarily on the gasodynamic resistances that oppose the fresh load passing through the intake system components: air filter, carburettor diffuser, throttle body, intake manifolds, inlet channel of the cylinder head, intake valve port (seat), and intake valve disk. In general, the depression created by the engine depends on many factors, among which the constructive factors are the most important. In practice, it has been found that the depression increases with increasing rotational speed.

The gasodynamic losses are caused, on the one hand, by the abrupt section variations when the fresh mixture is passing through the intake manifolds, intake port and valve and, on the other hand, when changing the direction of mixture passing through the bends of the intake system, as well as by the friction with the intake route walls.

The sudden changes of section and direction, along with the mixture friction with the intake route walls, generate

vortices whose effect is the pressure drop below the atmospheric pressure, at the end of the filling process. The thermal & gasodynamic losses during filling with fresh mixture are assessed using the degree of filling as defined above.

The gasodynamic losses are dependent on the flow velocity of the fresh mixture and, implicitly, on the engine rotational speed. Their sizes increase with increasing rotational speed. So, there is a decrease of fresh charge pressure in the cylinder compared to the one had in the environment and, therefore, a reduction in the mass that fills the available volume of the cylinder.

The pressure drops on the system components, such as: air filter, carburettor diffuser, intake manifolds and the intake channel of the cylinder head, have low values. Of particular importance, however, are the pressure drops on the throttle and the one corresponding to the valve port and the intake valve disk, reaching values of $10 \div 20\%$ of the atmospheric pressure [3].

3. EXPERIMENTAL RESEARCH. EQUIPMENT AND PROCEDURES

This study focuses on theoretical analysis, design and practical realisation of a gasodynamic test stand for providing information about how the fresh gases flow through the manifolds, during the intake process. The purpose of using this device is to find the characteristic parameters that emphasize the perfection of filling.

3.1. How to choose of the intake system model

To materialise the gasodynamic studies, we decided to analyse the intake system of a Dacia engine, type 810.99, equipped with CARFIL 32 IRM carburettor, WEBER licence, centrally positioned on the intake manifold. The cylinder head, cast from aluminium alloy, has two valves per cylinder (one inlet and one outlet). The combustion chamber is wedge-type, with the volume of 35.6 cm^3 . The diameter of the intake valve disk is 33.5 mm [4]. The reason for choosing this intake system is that it has the highest complexity in terms of the gasodynamic resistances found on the route between the air filter and the interior of cylinder.

3.2. Research methodology

To study the intake process in gasodynamic terms, it is necessary to create the same mechanism governing the gas flow through an internal combustion engine. This mechanism is given by the pressure difference between the interior of cylinder and the outer environment.

In this situation, we considered, as methodology for determining the gas flow parameters, the measurement of the pressure differences created between the outer environment and various points, specially chosen, located on the intake route, including the interior of cylinder, for various lifting heights of the intake valve and various positions of the throttle. With the aid of the measured pressure difference values, we can find the gas flows passing through the intake system and cylinders. With the aid of the measured flows and the theoretically determined flows, on the basis of the geometry of the components crossed by gases, we can find a number of

characteristic values of the intake process. Subsequently, we can determine the following parameters: filling degree, flow coefficient of the intake valve, momentary flow coefficient of the hole provided by the intake valve port, resistance coefficient of the intake route, efficiency of flow in the vicinity of the valve, etc.

3.3. Presentation of the experimental stand

For studying the gas flow, it is necessary to create a depression along the intake route. In accordance with this requirement, the gasodynamic research device must be capable of producing negative pressure.

The stand should be able to measure the pressure difference between the external environment and various points located along the intake route, including the interior of cylinder. Considering these aspects, the embodiment includes the following elements:

- equipment for obtaining the vacuum (vacuum pump);
- device for measuring the pressures and pressure differences;
- carburettor, centrally mounted on the intake manifold;
- cylinder head with holder;
- transparent cylinder;
- device for lifting the intake valve;
- device for measuring the lifting height of the intake valve;
- device used to operate the throttle body and to measure its angular position.

To ensure a good accuracy of the experimental measurements, a depression of 150 - 600 mmH₂O is required. The depression change should be continuous, for providing constant values for any geometry of the components to be studied. When studying the air flow through the intake valve, depending on its lifting height from the seat, it is required to change the test pressure for reading the small flows circulated by gases. For this reason, the vacuum pump is equipped with devices able to change the circulated flows and the values of the created depressions.

The measurement of testing pressure values is carried out by using an assembly of manometers with vertical tube filled with liquid. We choose this solution because it satisfies the required measurement accuracy and, in addition, it provides a good sensitivity when reading the values representing small pressure changes. The liquid used in the measuring tube is water.

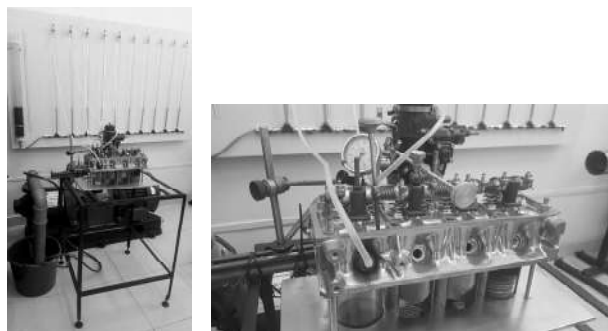


Fig.1. Experimental stand

The route of the pressure ports is shown in Fig. 2. They were positioned so as to measure the pressure drop on each representative section of the intake route.

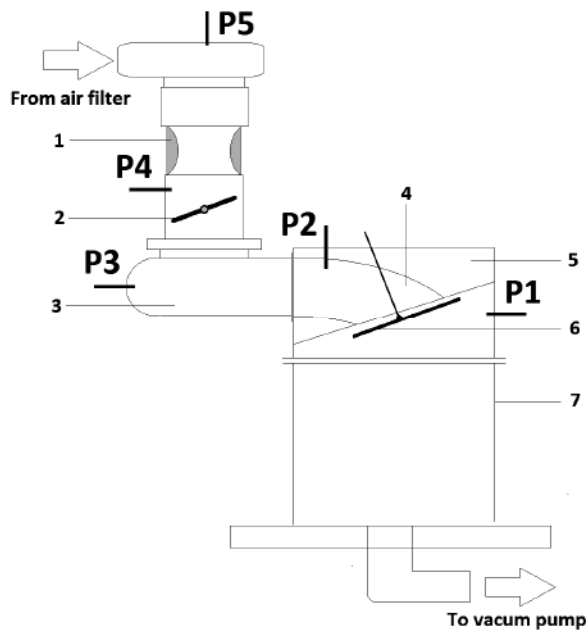


Fig.2. Intake route with the positioning of the pressure ports

The significance of the parts presented in Figure 2 is as follows: 1 – carburettor diffuser, 2 – throttle body, 3 – intake manifolds, 4 – intake channel of the cylinder head, 5 – cylinder head, 6 – intake valve, 7 – cylinder, P1...P5 – pressure ports.

The pressure ports are made in the form of holes in the plate-shaped solid wall (Fig. 3) or cylindrical wall (Fig. 4), over which the air stream flows.

A pressure port has the diameter (ϕ) of 1.5 mm, no burrs, its axis is normal to the cylindrical or plain wall, smooth, placed sideways or sideways-upwards (to avoid possible collection of gaseous inclusions). The flexible pipe is made of transparent plastic material [5].

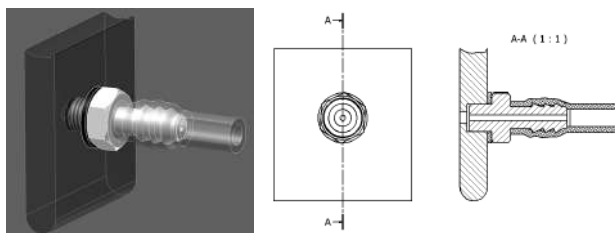


Fig.3. Pressure port in plain wall

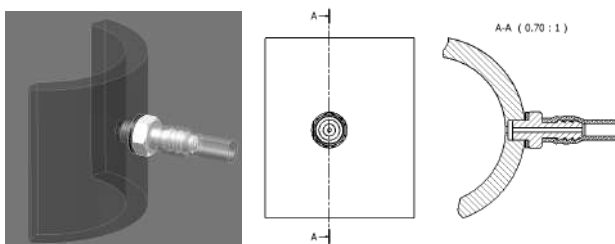


Fig.4. Pressure port in cylindrical wall

We considered 4 positions of the throttle body: 10% open, 25% open, 50% open and 100% open. For each position of the throttle body, we made measurements for 4 lifting

heights of the throttle body, i.e. 0.1 mm, 2 mm, 4 mm, 9 mm (maximum lifting height). All this time, the depression produced by the vacuum pump was kept constant. For each flow regime, as defined by all the above combinations, 3 sets of measurement have been made, the final values representing their arithmetic mean.

3.4. Results and conclusions

Hereinafter, we are going to present the results and conclusions of the experimental measurements, specifying that the measured pressure values are relative to the atmospheric pressure.

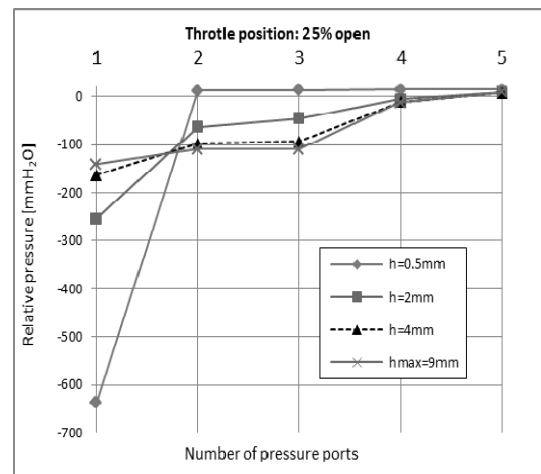


Fig.5. Relative pressure at the 5 ports, for the throttle body position "25% open"

In Figure 5, we can see that the depression values in the measuring points on the intake route, at the same throttle position (25% open), increase with increasing lifting height of the valve (h). On the contrary, the pressure differences between two consecutive measuring points (pressure drop or pressure loss) decrease with increasing lifting height of the valve. This is because, with increasing lifting height of the valve, the flow area in the vicinity of the valve increases and, consequently, the flow rate of the fluid decreases. A low flow speed means lower pressure losses.

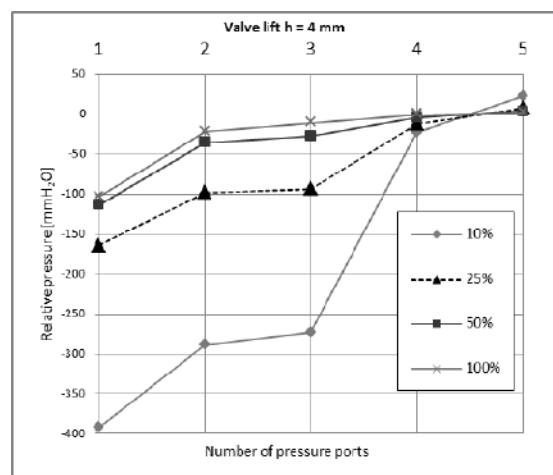


Fig.6. Relative pressure at the 5 ports, for the valve lifting height $h = 4$ mm

Figure 6 shows that the depression values recorded in the measuring points, when the lifting height of the valve is maintained constant ($h = 4 \text{ mm}$), are decreasing as the throttle opens. We can also see that the loss of pressure when passing in the vicinity of the throttle (pressure difference between the ports no. 3 and 4) is lower the more open the throttle is.

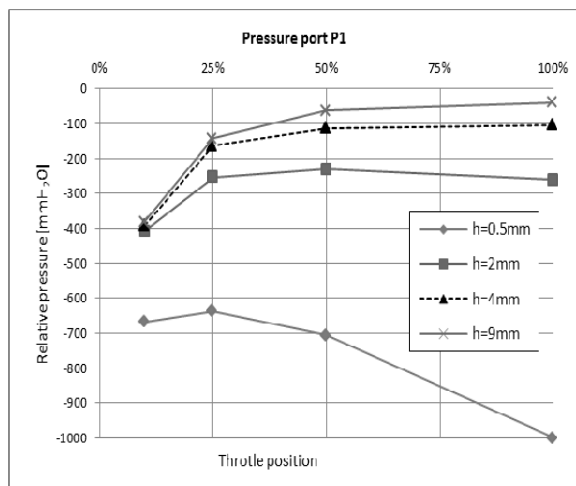


Fig. 7. Relative pressure at the port no. 1 versus the valve lifting height and throttle body position

If we focus on the pressure port P1, located in the combustion chamber (Fig. 7), we can say that, at small lifting heights of the intake valve, due to the extremely high pressure losses, at maximum throttle openings, the depression in the combustion chamber is maximal.

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ON MAGNETORHEOLOGICAL BRAKE FEM MODELING

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Abstract: This paper presents a new approach in magnetorheological (MR) disk brake magnetic design. The possibility of MR fluid's magnetic field routing is considered in this paper. The electromagnetic analysis of the new MR brake (MRB) design is carried out using COMSOL Multiphysics software in its AC/DC module. Necessary material's magnetic properties were previously determined and applied to this simulation. Different number of triangular element are used for different parts of the proposed MRB model, prioritizing parts in vicinity of MR fluid. Post processing was utilized to calculate the magnetic field distribution across the MRB core and magnetic flux density on the central line of the MR fluid gap. Results are compared to those of the classic MRB design model with the same dimensional restrictions. The performances of the proposed MRB design shows improvements over the classical MRB design.

Key words: Magnetorheological brake, Magnetic field routing, Initial magnetization curve, Finite Element Method.

1. INTRODUCTION

Magnetorheological – MR fluids belong to a class of intelligent materials that respond to applied magnetic field with fast, continuous, but reversible change in its rheological behavior, [1]. MR fluids are a type of suspensions. Carrier fluid is usually mineral or synthetic oil, water, kerosene etc., with dispersed micro size ferromagnetic particles. When exposed to external magnetic field, particles form chainlike structures, thus changing the fluid's viscosity.

A wide range of MR fluid devices have been investigated for their applications potential in different systems, systems such as: vibration control, seismic response reduction and torque transfer devices.

A lot of work has been done in the field of MR fluid devices modeling, [2] and [3]. In this paper, emphasis is

on MR brake – MRB as a device to be modeled. There are several different methods to model MR device such as MRB, and one of them is finite element method. To be able to model the MRB, as an electromagnetic system, using FEM, one needs to know its construction restrictions and material's magnetic properties. Typical MRB usually consists of two different material types: nonmagnetic and magnetic material. All nonmagnetic materials practically have the same magnetic properties, which is not the case for magnetic materials. Magnetic properties for MR fluid are easily obtained from its manufacturer. On the other hand, magnetic properties for magnetic materials such as construction steel, usually are not available, so they need to be determined by measurements. The most important magnetic material properties, in MRB magnetic design process, are the initial magnetization and the hysteresis loop, [4].

The objective of this work is to propose a new MRB design, and present its simulation in commercial FEM software. Using some earlier material measurement result, the proposed MRB material's nonlinearity was tackled. Magnetic field and magnetic flux density results, obtained in the simulation, can be used to assess the proposed MRB's braking torque potential. These results are compared to those of a classical MRB design, also FEM simulated. In both cases the same geometrical restrictions are applied. Results are compared and discussed at the end of the paper.

2. THE PROPOSED MAGNETORHEOLOGICAL BRAKE DESIGN

Magnetorheological brake is an electromechanical system that consists of stationary parts: the coil and the stator, and the moving part: the rotor. The coil envelops the stator i.e. the housing. The rotor is immersed in MR fluid and suspended inside the stator. Energized coil generates magnetic field that spreads through MRB's construction in a certain manner. Magnetic field affects the MR fluid's viscosity, hence the MRB's braking torque value.

In some cases, depending on the coil's and MRB's design characteristics, magnetic field effect on the MR fluid will be diminished by the possibility of magnetic field spreading through areas with less magnetic resistance. These areas are usually made of magnetic material, e.g. rotor's shaft and section between disk and coil, through which magnetic field bypasses MR fluid area of the brake. If this is the case, some sections of the MR fluid may not be affected by the magnetic field and the other sections may have only limited magnetic field influence. This situation is illustrated in Figure 1. To counteract undesirable magnetic field spreading, magnetic field routing is necessary.

In this section a brief description of the proposed MRB design, with routed magnetic field, is presented. Figure 2 depicts a cross section of the proposed design. The outer section of the rotor's disk and the housing are composed out of the same magnetic material and the MR fluid surrounds the disk. Conical sleeve is placed around the disk and is a part of the housing.

The use of the nonmagnetic materials in MRB magnetic design affects normal magnetic field spreading path.

Conically shaped sleeve and rotor's shaft are both made out of the same nonmagnetic material. Their presence will route the magnetic field into MR fluid sections of the brake, Figure 3.

The chemical compositions of the steel and nonmagnetic material, proposed for use in the new MRB construction, are given in Table 1.

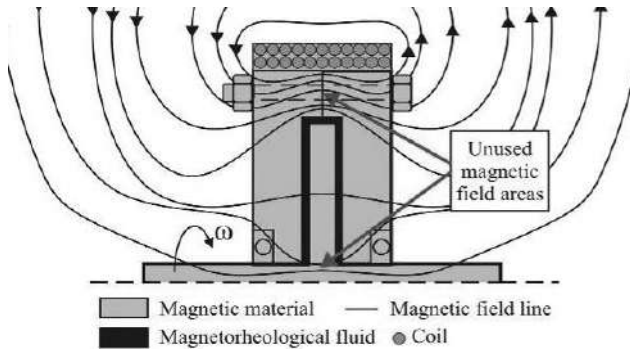


Fig.1. Undesirable magnetic field spreading

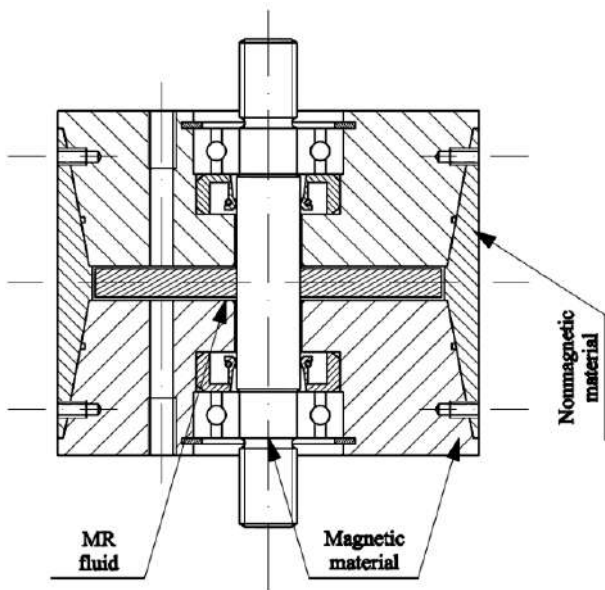


Fig.2. Cross section of the proposed magnetorheological brake design

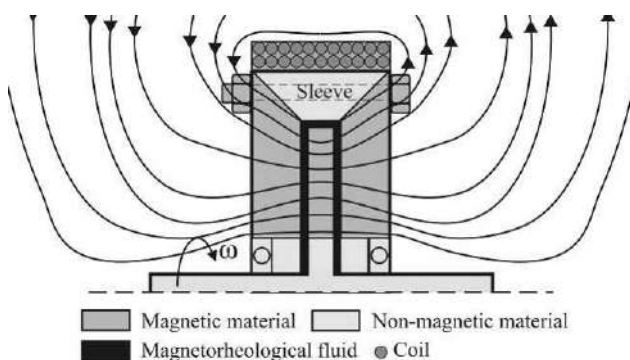


Fig.3. Magnetic field routing

Table 1. Proposed material's chemical composition

Element	C15E, EN 10027	AlMgSi 0.5, EN AW – 6060 T66
C	0.12 – 0.18	-
Mn	0.3 – 0.6	0.1
P	0.035 max	-
S	0.035 max	-
Si	0.4 max	0.3 – 0.6
Cu	-	0.1
Mg	-	0.35 – 0.6
Cr	-	0.05
Zn	-	0.15
Fe	rest	0.1 – 0.3
Al	-	rest

Magnetic properties measurement process, for C15E steel, is presented in the next section. Table 2 holds some basic dimensions of the proposed MRB design.

Table 2. Proposed magnetorheological brake's dimensions

Parameter	Value
MRB's outer diameter (mm)	72
MRB's length (mm)	60
Disk diameter (mm)	60
Disk thickness (mm)	5
Shaft diameter (mm)	10
Total MR gap (mm)	1
MRF volume (mm ³)	≈ 10

3. INITIAL MAGNETIZATION CURVE MEASUREMENT

System such as MR brake is considered to be a magnetic circuit and for the most part is composed out of magnetic material. MR brake's geometry, in regard to its coil, makes it a cylindrically shaped DC electromagnet. For magnetic calculation purposes MR fluid presence can be neglected, because of its small volume share.

One of the important characteristic of magnetic materials is the phenomenon of magnetic hysteresis. This phenomenon is illustrated in Figure 4.

Change in the MRB coil's control current changes the magnetic field and magnetic flux density intensity in magnetic material of the brake. Magnetic field to magnetic flux density holds highly nonlinear relationship. In this process magnetic flux density will increase only to the point of saturation, Figure 4 - points **a** and **b**. Further magnetic field increase will not result in magnetic flux density increase. If this process is reversed, e.g. decrease of the magnetic field will result in the magnetic flux density decrease, but not in the same rate and will lead to the negative saturation point, see Figure 4. Completing the magnetic field intensity variation process will form well known hysteresis loop. Magnetically hard materials have larger hysteresis loop area as opposed to magnetically soft materials. Rule of the thumb is that MR brake should be made of magnetically soft steel.

To be able to analytically determine the braking torque potential, one needs to know magnetic characteristics of the steel in use. As mentioned earlier, magnetic characteristic of ferromagnetic materials usually are not available, as it is the case with C15E. So, appropriate measurements were conducted.

Using well known ballistic measurement method [5], initial magnetization curve for C15E steel was obtained, [4]. Having very precise and accurate measuring equipment, applying movie camera for the recording and analyzing the ballistic galvanometer light beam's position, very accurate results were obtained. Measurement circuit diagram is shown in Figure 5, where primary and secondary windings are denoted as N_1 and N_2 , respectively. The toroidal transformer with the core made of C15E steel sample was constructed, Figure 6. The transformer consisted of 600 turns primary winding and 60 turns secondary winding. There were 6 measurement series, from which the average value was calculated. Measurement results are presented in Section 4 of the paper.

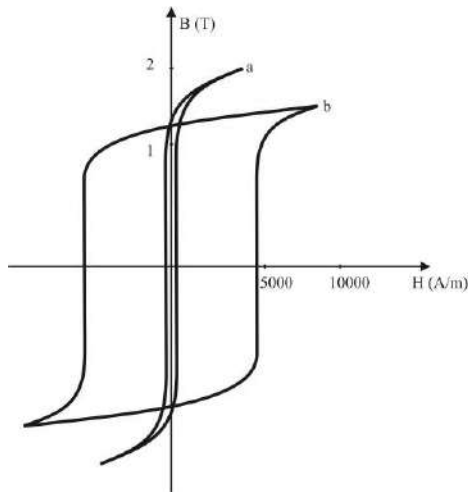


Fig.4. Hysteresis loops,
a) magnetically soft steel and b) magnetically hard steel

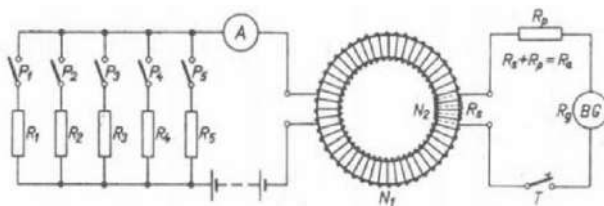


Fig.5. Initial magnetization curve measurement circuit, [5]

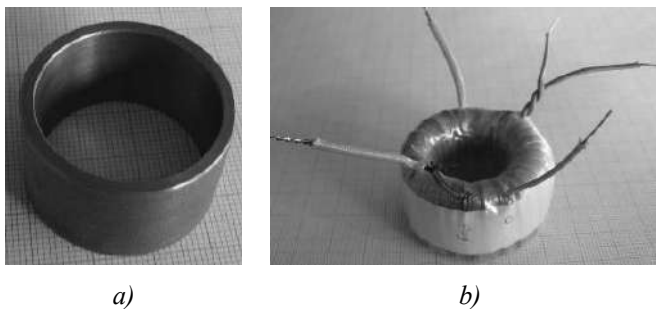


Fig.6. The toroidal transformer
a) C15E steel sample, b) the transformer, [4]

4. NUMERICAL SIMULATIONS

In this section the MRB's numerical simulation and figures of merit are presented. Two MRB designs, proposed and classic, are modeled using commercial FEM software COMSOL Multiphysics – AC/DC module. Due to MRB's axial symmetry, a 2D axisymmetric space dimension option is selected. Magnetic fields subsection is used, because there is no significant electric field influence on the components in an electromechanical system such as MRB. Magnetic field is considered to be static, so stationary domain is used.

4.1. Simulation steps

MRB simulation models are presented in this section. Proposed MRB geometry is presented in Figure 7, a). Every part, of the Model, is numerated including surrounding air boundary, marked as 1. Appropriate material node, containing adequate data, is assigned to every part of the Model. Materials, such as nonmagnetic air and aluminum are selected from COMSOL's material database, but nonlinear magnetic materials, C15E and MRF, are defined using previously measured magnetic properties. These data have been loaded in Comsol as separate files. Note, presence of parts such as ball bearings are neglected because of their steel composition and small volume share in overall construction.

In Magnetic Fields subsection of the Model, additional Ampère's Laws are needed, due to the existence of several different materials. In the same subsection, a Multi-Turn Coil Domain node is added. This node contains inputs for coil.

Figure 8, a) presents meshed Model. Mesh is generated using User-controlled mesh. Element Size is predefined to Extra Fine, with maximum and minimum element size: 8 mm and 0.03 mm, respectively. Approximately 6200 triangular elements are used to solve the following Maxwell's electromagnetic equations:

$$\nabla \times H = J = \sigma(E + v \times B) + J_e \quad (1)$$

$$\nabla \cdot B = 0 \quad (2)$$

To present magnetic field and magnetic flux density intensities across the surfaces of the MRB, a 2D plot options, available in the post processor of AC/DC module, are utilized. The solver was stationary and non-linear in this study.

To compare results a reference model was needed. For this purpose the classic MRB design is modeled and its geometry presented in Figure 7, b). All parts are numerated similarly as with proposed MRB design. Again, parts are assigned with appropriate material nodes. The main difference, in regard to previous design, is that there is no nonmagnetic parts to route magnetic field. Aside this, materials are selected from COMSOL's material database or loaded as separate files.

Magnetic Fields and Mesh node have the same setups as previous.

In Figure 8, b) meshed model of the classical MRB design is presented. Again, approximately 5600 triangular elements are used to solve the same Maxwell's electromagnetic equations, equations: (1) and (2).

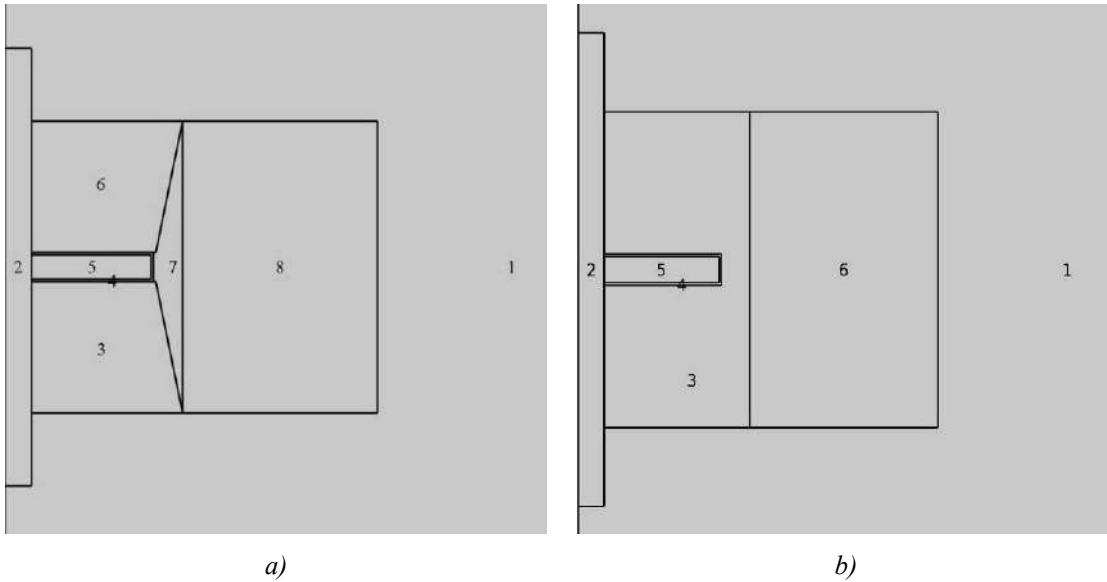


*Fig.7. The Comsol model geometry,
a) proposed magnetorheological brake design, b) classic magnetorheological brake design*

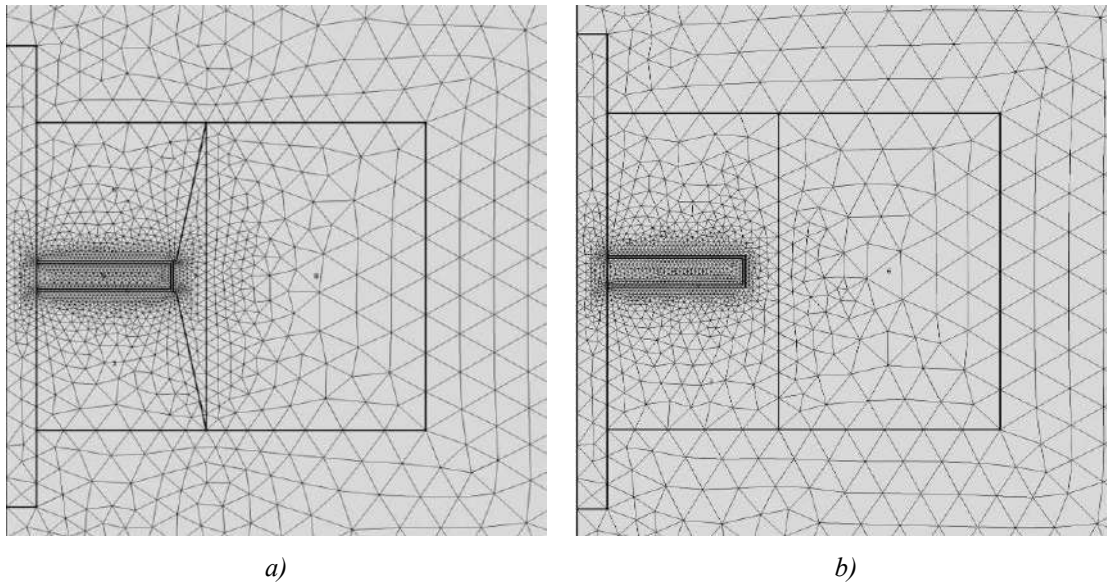


*Fig.8. The Comsol model mesh,
a) proposed magnetorheological brake design, b) classic magnetorheological brake design*

5. RESULTS AND DISCUSSION

In this section, initial magnetization curve and FEM simulations results are presented. Figures 9 a) and b) provide: initial magnetization curves results and their average value, respectively. These measurements were carried out using C15E steel sample and the obtained data are used to define this, magnetically nonlinear, material in FEM simulation process. In Figure 9 a), results of 6 measurements are presented and a good results repeatability is present. Overall magnetic field range, for C15E, is considerably lower than of the AISI 1018 steel, for example. Based on these results, FEM simulations for both MRBs designs are made. Some simulation results are presented in Figures 10 and 11. Figures 10 a) and b) provide 2D surface plots of magnetic flux density of MRBs cross sections. Maximum applied coil control

current is 1 A, in both cases. Magnetic flux density concentration on sharp edges in both models is noted, so rounded edges are highly recommended for future simulations. In Figures 10 c) and d), magnetic flux density intensity change over MR fluid's gap central line is presented. Considerably higher magnetic flux density is notable for design in Figure 10 c) then of the design presented in Figure 10 d). Finally, the magnetic field distribution results, for both designs, are presented in Figures 11 a) and b). Magnetic field concentration in different sections of the brakes is present, resulted from usage of nonmagnetic materials. In Figure 11 b) magnetic field gravitates toward disk's outer rim section. Lower magnetic field concentration, on disk's active surfaces, in this case results in lower MR fluid's viscosity change, thus lower braking torque. On the other hand, in Figure 11 a), due to nonmagnetic material usage and field routing

effect, magnetic field is evenly distributed over the disk's active surfaces resulting in proposed MRB braking torque potential increase.

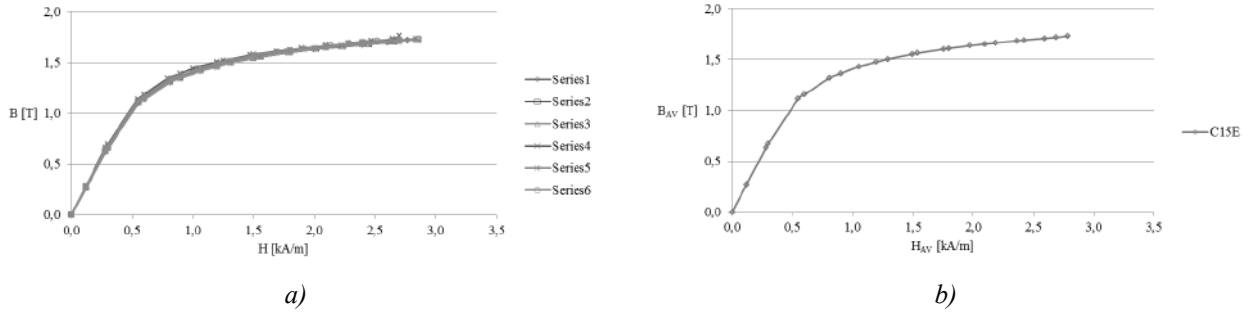


Fig.9. Initial magnetization curve for C15E steel: a) 6 measurement series, b) average magnetization curve

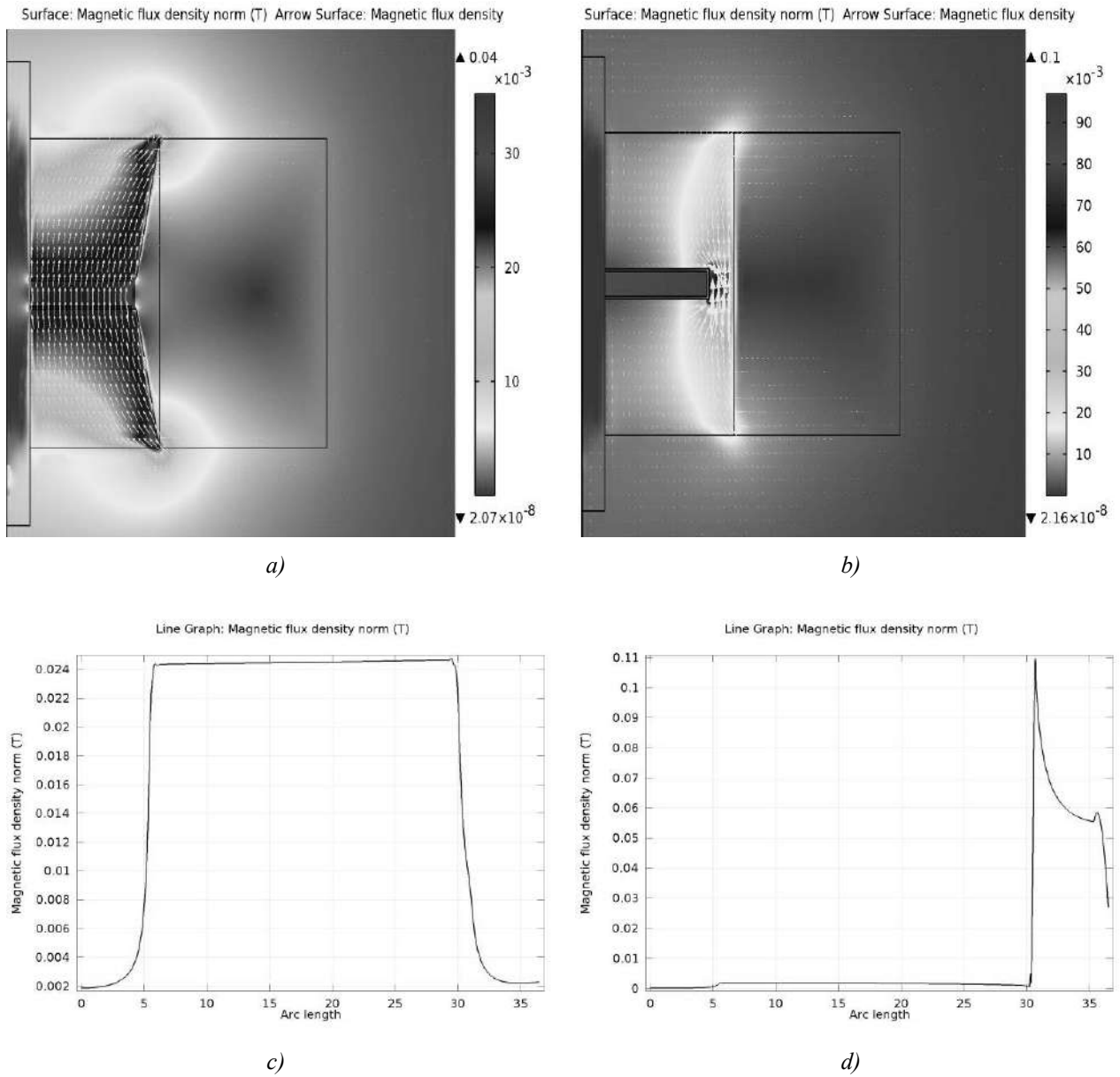


Fig.10. Magnetorheological brakes finite element method modeling
a) proposed MRB magnetic flux density 2D plot, b) classic MRB magnetic flux density 2D plot,
c) proposed MRB magnetic flux density intensity along MR fluid's gap central line,
d) classic MRB magnetic flux density intensity along MR fluid's gap central line

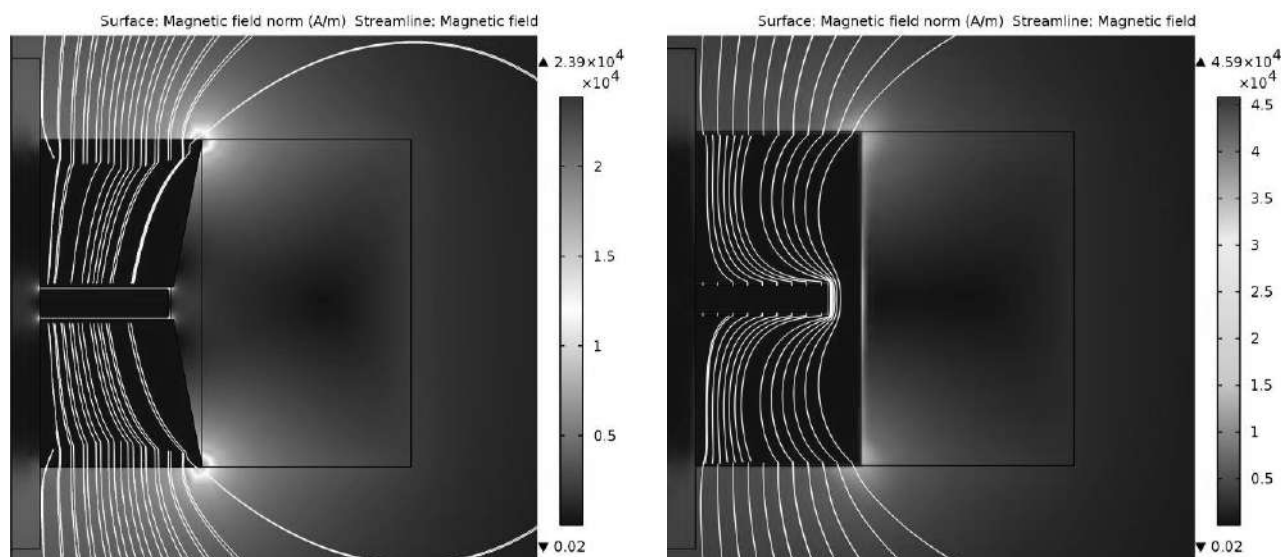


Fig.11. Magnetorheological brakes finite element method modeling,
a) proposed MRB magnetic field 2D plot, b) classic MRB magnetic field 2D plot

6. CONCLUSION

In this paper, a new magnetorheological brake design is proposed. Magnetic flux density intensity and the magnetic field intensities are calculated using finite element based software. Nonlinear relationship between magnetic flux density and magnetic field for commercial MR fluid and steel are applied in simulations. These steel magnetic properties were obtained in previous measurements. The proposed MRB design shows magnetic characteristics improvement compared to the classic MRB design with the same geometric limitations.

ACKNOWLEDGMENT

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THE INFLUENCE OF THE ECCENTRICITY ON SAFETY COEFFICIENT ON A BUTTERFLY VALVE BIPLANE DISC

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Abstract: The paper presents the safety coefficient evolution for different eccentricity locations of a high performance butterfly valve biplane disc. By using structural analysis with finite element method, it was analyzed 13 locations of biplane disc eccentricity. The obtained equivalent stresses for each case are presented and the minimum safety coefficients were plotted depending on the eccentricity. For a certain value of the eccentricity, the safety coefficient has low value, even if neighbouring safety coefficients values are higher.

Key words: biplane disc, equivalent stress, safety coefficient

1. INTRODUCTION

Due to simplicity, safety and maintenance, butterfly valves are the most used types of valves [1] in hydraulic circuits. They ensure the sealing of the fluid toward into downstream equipments. The high performance butterfly valves are used as turbine inlet valves in hydro power plants.

The most important part of the butterfly valve is the biplane disc [2, 3], which can be designed with two eccentricities (Fig. 1): e_1 and e_2 . The eccentricity e_1 is established due to strength of materials conditions, while the eccentricity e_2 takes into consideration the hydraulic shape [4] of the disc and the torque at closing of the butterfly valve.

The biplane disc is a steel welded structure of plates and casted hubs [5]. The mechanical characteristics and physical properties of the used material are: yield point: $\sigma_y=250$ MPa; tensile strength: $\sigma_t=450$ MPa; Young modulus: $E=2 \cdot 10^5$ MPa; Poisson coefficient: $\nu=0.3$.

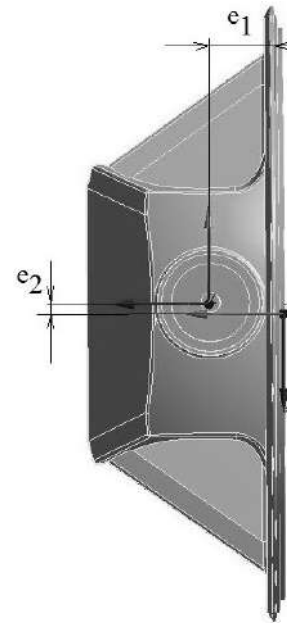


Fig.1. Biplane disc with two eccentricities

2. STATIC STRUCTURAL ANALYSIS

The analyzed biplane disc is of a butterfly valve with 4.9 m inlet diameter and 1.4 MPa upstream water pressure. For a given value of eccentricity $e_1=515$ mm, there were considered 13 cases of the eccentricity e_2 . The eccentricity values were increased with 5 mm, starting from 40 mm till 100 mm (Table 1).

The static structural analysis for each value of the eccentricity was performed with Ansys software. After meshing (Fig. 2) and putting the boundary conditions (Fig. 3), the structural analysis [6] was performed.

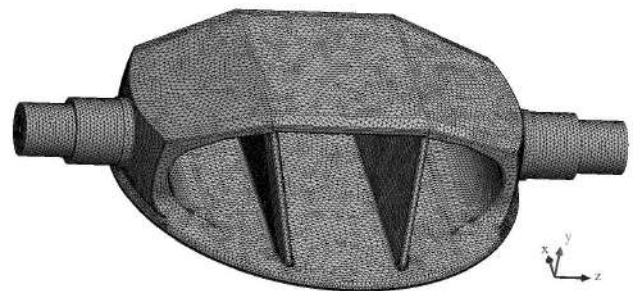


Fig.2. Mesh - biplane disc

The reliability of the biplane disc is characterized by the safety factor. Safety factor is a term that describes the ability of the structure to take external loads higher than those considered in the calculation which may arise in the actual operation of the biplane disc.

Safety factor (c) is defined (1) as the ratio between the yield strength of the material ($\sigma_y = 250$ MPa) and the equivalent stress (σ_{eq}):

$$c = \frac{\sigma_y}{\sigma_{eq}} \quad (1)$$

In this paper, von Mises stress obtained from static analysis is considered as equivalent stress.

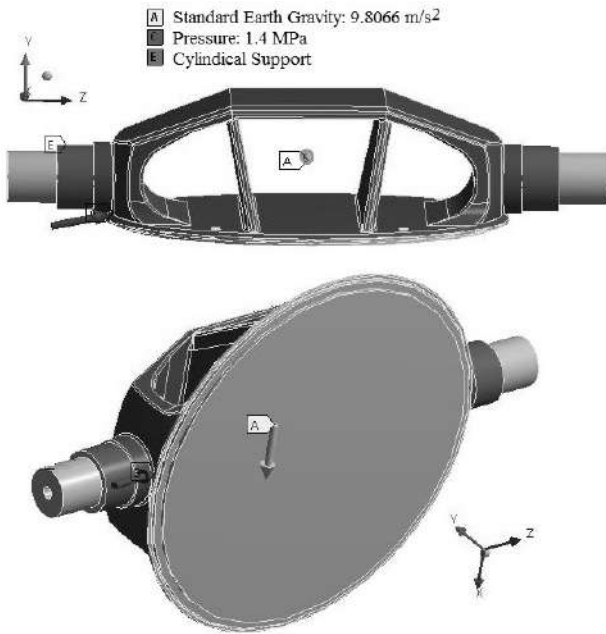


Fig.3. Boundary conditions

The von Mises stress is used to predict yielding of materials under any loading condition from results of simple uniaxial tensile tests.

Figure 4 shows the distribution of von Mises equivalent stresses of the butterfly valve biplane disc obtained by using Finite Element Method (FEM). The higher values of the equivalent stresses are illustrated in shades red colour and the lower values of the equivalent stresses are shown in shades blue colour.

The maximum values of the equivalent stress obtained for the analyzed cases are shown in Table 1.

Table 1. Values of the maximum equivalent stresses

No	Eccentricity e_2 [mm]	Maximum equivalent stress σ_{eq} [MPa]
1.	40	179.61
2.	45	179.57
3.	50	179.43
4.	55	179.16
5.	60	179.28
6.	65	179.11
7.	70	179.14
8.	75	179.23
9.	80	180.84
10.	85	185.78
11.	90	185.18
12.	95	185.13
13.	100	180.20

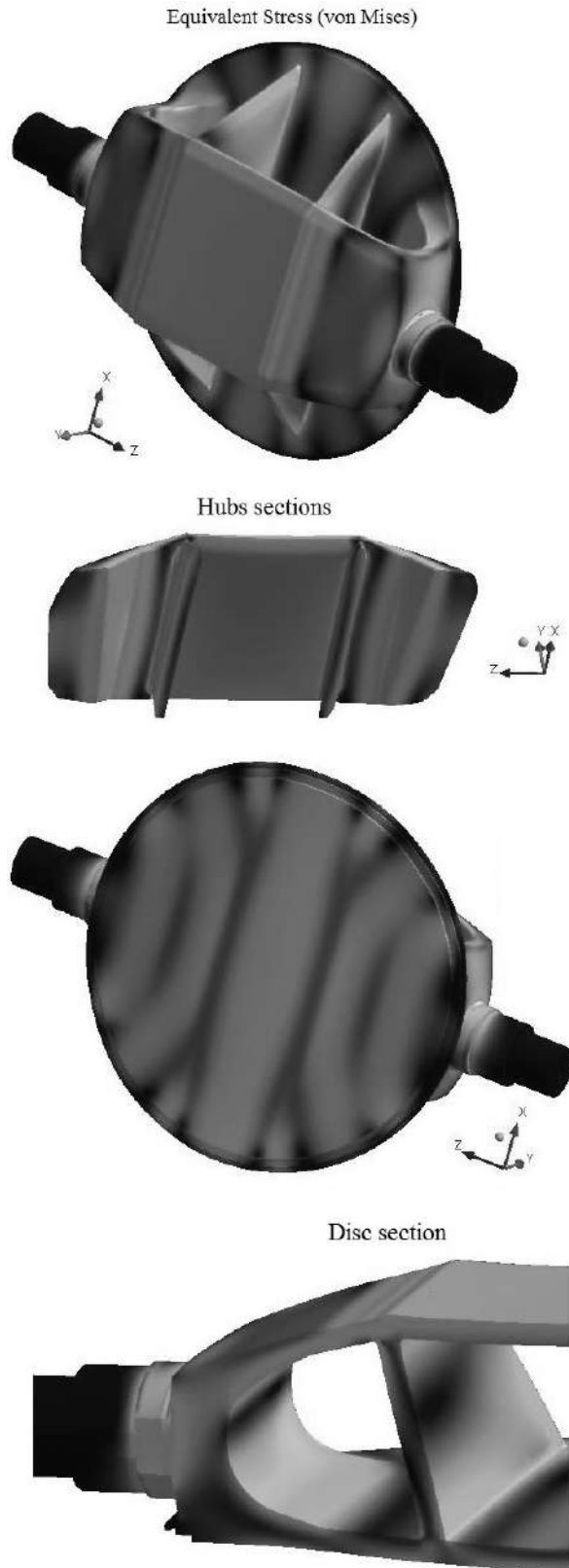


Fig.4. Equivalent stress (von Mises) on the biplane disc

Areas in which von Mises equivalent stresses have maximum values (Fig. 4) are found in the transition region of the biplane disc shafts to the hubs [7]. It can be observed that a three-axial stress states appears in the biplane disc.

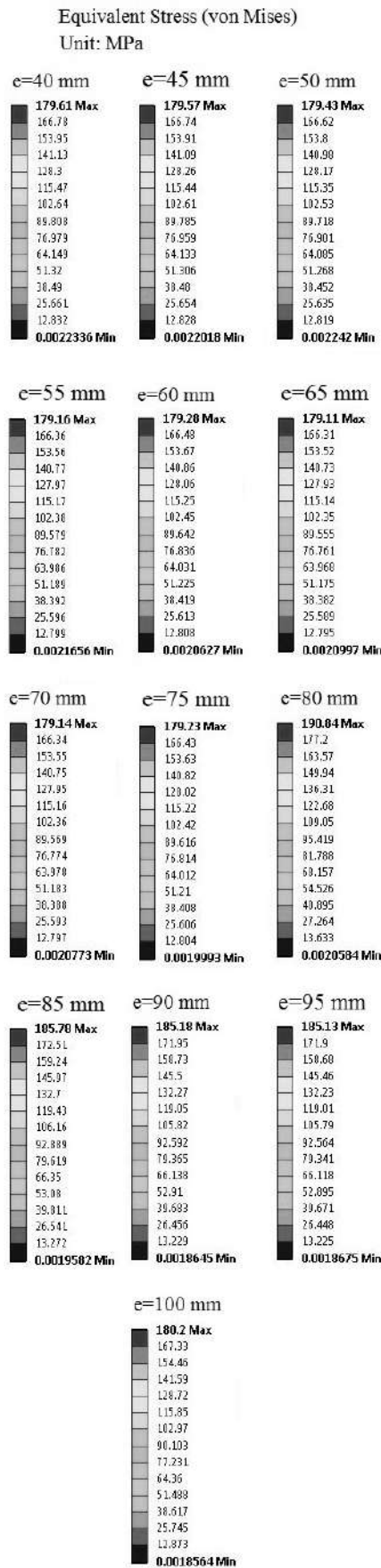


Fig.5. Equivalent stress - FEM results

After moving the eccentricity in all considered cases and structural analysis was performed, the equivalent stress results are presented in figure 5.

3. SAFETY COEFFICIENT EVOLUTON WITH ECCENTRICITY

The safety coefficients distribution on the biplane disc are presented in Figure 6. The lower values of the safety coefficients are illustrated in shades red colour and the higher values of the safety coefficients are shown in shades blue colour.

Areas in which the safety factor has a lower values are found in the transition region of the biplane disc shafts to the hubs, joints between hubs of the biplane plate, joints between biplane plate and ribs, joints between ribs and disc plate and joints between hubs and disc plate.

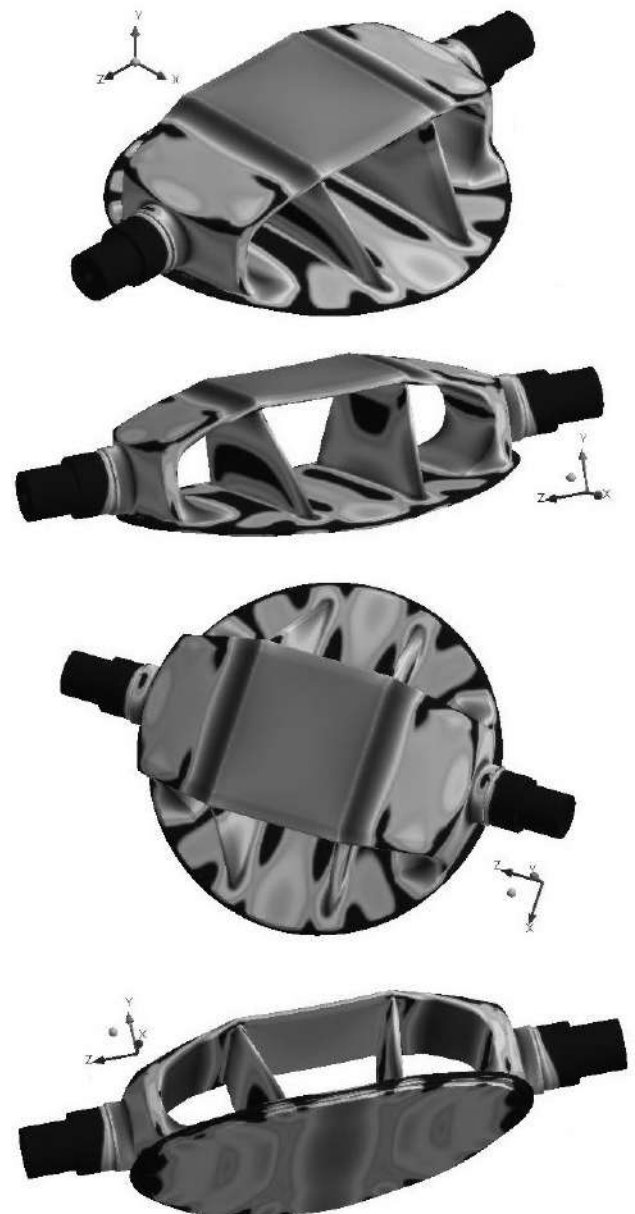


Fig.6. Safety coefficients distribution on the biplane disc

For each analyzed case of e_2 eccentricity location there were obtained minimum values of safety coefficients which are presented in Table 2.

Table 2. Values of the minimum safety coefficient

No	Eccentricity e_2 [mm]	Minimum safety coefficient c_{min}
1.	40	1.3919
2.	45	1.3922
3.	50	1.3954
4.	55	1.3954
5.	60	1.3945
6.	65	1.3958
7.	70	1.3956
8.	75	1.3949
9.	80	1.3100
10.	85	1.3457
11.	90	1.3500
12.	95	1.3504
13.	100	1.3873

Representing all values obtained for the safety coefficients depending the eccentricity e_2 it is obtained the distribution of the minimum safety coefficient (Fig. 7) for all locations of the eccentricity for the considered limits. By analyzing the figure 7 it can be observed that, if $e_2 < 80$ mm, the minimum safety coefficient had a constant value of 1.39-1.40. When $e_2 = 80$ mm, the safety coefficient drops to 1.3. For higher values of eccentricity, the safety coefficient increases to 1.35, and finally, the safety coefficient becomes 1.39.

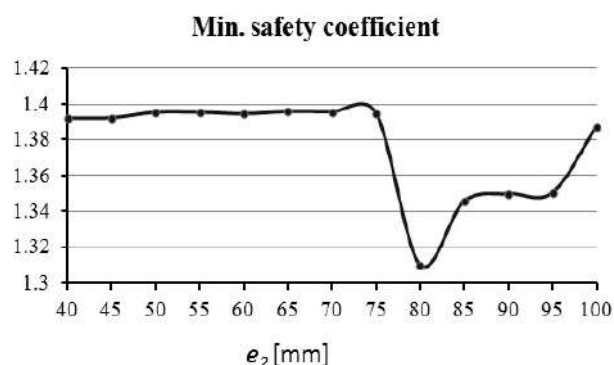


Fig.7. Minimum safety coefficient depending on the eccentricity

4. CONCLUSION

Safety coefficients are calculated using detailed analysis but the structure's ability to carry load must be determined to a reasonable accuracy. After analyzing the obtained results for a biplane disc it can be concluded that due to eccentricity location, the safety coefficient cannot be predicted and must be analyzed for each biplane disc structure.

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DIRECTIONAL DEFORMATION OF THE BIPLANE DISC BY MOVING THE ECCENTRICITY

Cristian Marius MIMIȘ

Abstract: The issue presented in this paper refers at the effect of the double eccentricity on the directional deformation of a butterfly valve biplane disc. By moving of one of the eccentricity for 13 locations and by using the structural analysis with finite element method, the obtained results of the directional deformation for x , y and z axis were plotted, compared and analysed. It was concluded that a linear dependence of the directional deformation depending of the eccentricity location was obtained.

Key words: biplane disc, eccentricity, directional deformation

1. INTRODUCTION

The butterfly valve has the purposes of tight closing water admission into the hydro unit turbine under normal operating conditions and emergency conditions. In fact, it is a safety element for the turbine [1, 2]. The butterfly valve biplane disc is designed with two eccentricities (Fig. 1): e_1 and e_2 . Double eccentric [3, 6] positioning of the disc has the advantages that it permits smooth raising and setting of the disc gasket from the sealing surface. A biplane disc [4] consists mainly of (Fig. 1): disc, ribs, biplane which are manufactured of steel plates and trunnions with hubs which are manufactured of cast steel with the following mechanical characteristics and physical properties: yield point: $\sigma_y=250$ MPa; tensile strength: $\sigma_t=450$ MPa; Young modulus: $E=2 \cdot 10^5$ MPa; Poisson coefficient: $\nu=0.3$.

The butterfly valve biplane disc is shaped to the water flow direction to ensure minimum hydraulic losses [5]. The eccentricities values are limited by the hydraulic shape of biplane disc.

Usually, the e_1 eccentricity is obtained as condition from strength of materials and e_2 eccentricity is chosen by the designer.

2. PREPARING THE ANALYSIS

After 3D modeling of the biplane disc a static structural analysis was performed by using finite element method with Ansys software. The analyzed biplane disc has 4.9 m diameter and 1.4 MPa upstream water pressure. The eccentricity e_1 was chosen at 515 mm and for the e_2 eccentricity were analyzed 13 locations: 40 mm, 45 mm, 50 mm, 55 mm, 60 mm, 65 mm, 70 mm, 75 mm, 80 mm, 85 mm, 90 mm, 95 mm and 100 mm.

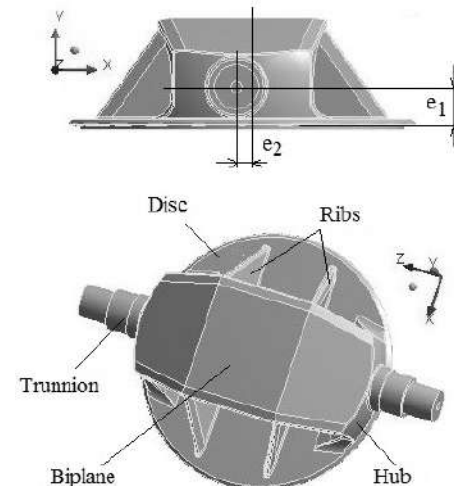


Fig.1. Biplane disc: component and eccentricity definition

For static structural analysis, a tetrahedral [7] finite element was used with the medium size of the mesh of 50 mm (Fig. 2). The boundary conditions [8] are presented in figure 3: cylindrical support on trunnions, water pressure on upstream side of the disc and earth gravity.

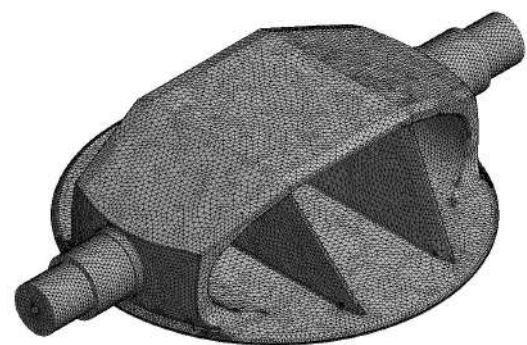


Fig.2. Mesh - biplane disc



Fig.3. Boundary conditions

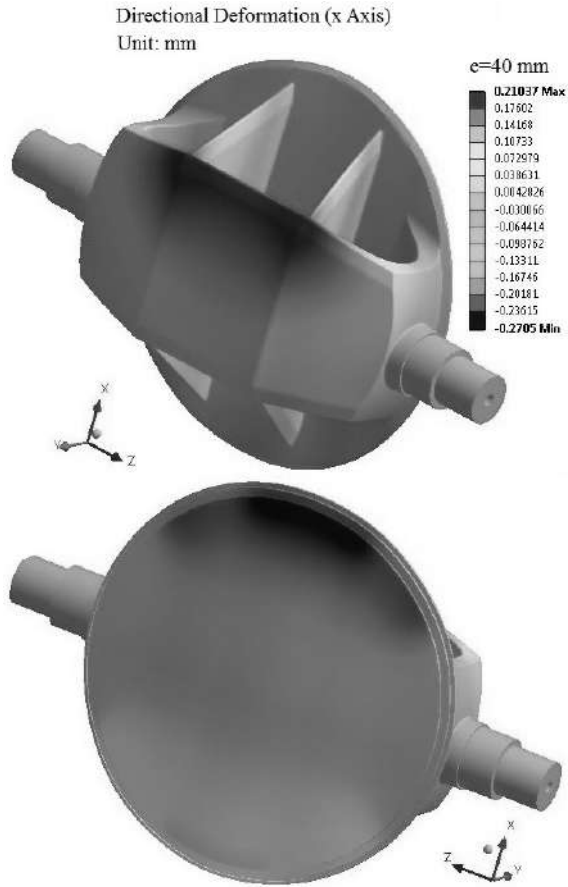


Fig.4. Directional deformation (x axis)

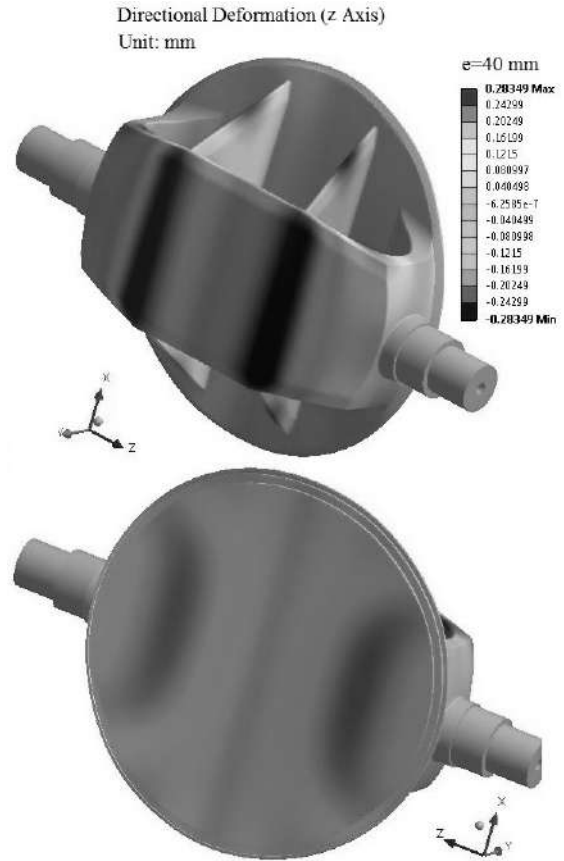


Fig.6. Directional deformation (z axis)

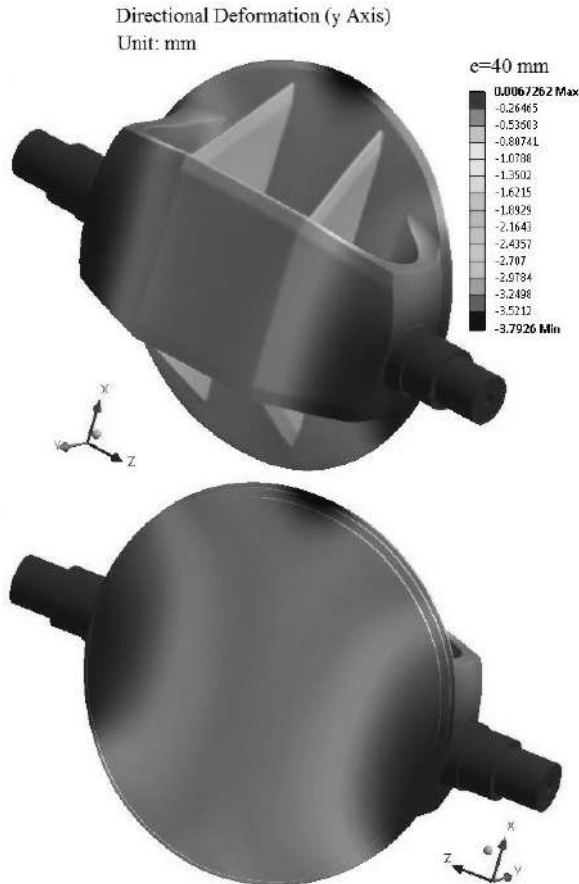


Fig.5. Directional deformation (y axis)

3. DIRECTIONAL DEFORMATION

First, it is analyzed the directional deformation for the eccentricity value of 40 mm. Directional deformation results on x axis, y axis and z axis are presented in figures. 4, 5 and 6.

Values of Directional Deformation (x Axis)

No	Eccentricity e_2 [mm]	$u_{x \min}$ [mm]	$u_{x \max}$ [mm]
1.	40	-0,27050	0,21037
2.	45	-0,27441	0,21978
3.	50	-0,27843	0,22900
4.	55	-0,28242	0,23860
5.	60	-0,28640	0,24800
6.	65	-0,29044	0,25765
7.	70	-0,29449	0,26717
8.	75	-0,29845	0,27651
9.	80	-0,30245	0,28611
10.	85	-0,30639	0,29549
11.	90	-0,31049	0,30511
12.	95	-0,31456	0,31478
13.	100	-0,31864	0,32418

Areas in which the directional deformations on x axis (Fig. 4) have higher values are found on the biplane plate and on the disc plate.

Higher values of directional deformations on z axis (Fig. 6) appear in ribs, joint between biplane plate and ribs, hubs and disc plate.

The maximum values on x and y axis are very small comparing with the biplane disc.

Important directional deformations on y axis (Fig. 5) are found on the disc plate, ribs and biplane plate.

Table 2. Values of Directional Deformation (y Axis)

No	Eccentricity e_2 [mm]	$u_{y \min}$ [mm]	$u_{y \max}$ [mm]
1.	40	-3,7926	$6,7262 \cdot 10^{-3}$
2.	45	-3,8150	$6,7366 \cdot 10^{-3}$
3.	50	-3,8375	$6,7607 \cdot 10^{-3}$
4.	55	-3,8606	$6,7470 \cdot 10^{-3}$
5.	60	-3,8833	$6,7608 \cdot 10^{-3}$
6.	65	-3,9064	$6,7678 \cdot 10^{-3}$
7.	70	-3,9296	$6,7682 \cdot 10^{-3}$
8.	75	-3,9524	$6,7782 \cdot 10^{-3}$
9.	80	-3,9757	$6,7960 \cdot 10^{-3}$
10.	85	-3,9987	$6,7876 \cdot 10^{-3}$
11.	90	-4,0226	$6,7939 \cdot 10^{-3}$
12.	95	-4,0465	$6,8191 \cdot 10^{-3}$
13.	100	-4,0699	$6,8216 \cdot 10^{-3}$

Table 3. Values of Directional Deformation (z Axis)

No	Eccentricity e_2 [mm]	$u_{z \min}$ [mm]	$u_{z \max}$ [mm]
1.	40	-0,28349	0,28349
2.	45	-0,28409	0,28402
3.	50	-0,28457	0,28464
4.	55	-0,28523	0,28514
5.	60	-0,28583	0,28566
6.	65	-0,28639	0,28623
7.	70	-0,28691	0,28681
8.	75	-0,28746	0,28737
9.	80	-0,28806	0,28786
10.	85	-0,28854	0,28849
11.	90	-0,28908	0,28906
12.	95	-0,28966	0,28958
13.	100	-0,29018	0,29015

4. DIRECTIONAL DEFORMATIONS BY MOVING ECCENTRICITY

After applying the structural analysis by using finite element method for each location of the e_2 eccentricity (between 40 mm and 100 mm with steps of 5 mm), were obtained 13 cases of analysis with maximum and minimum values of directional deformation.

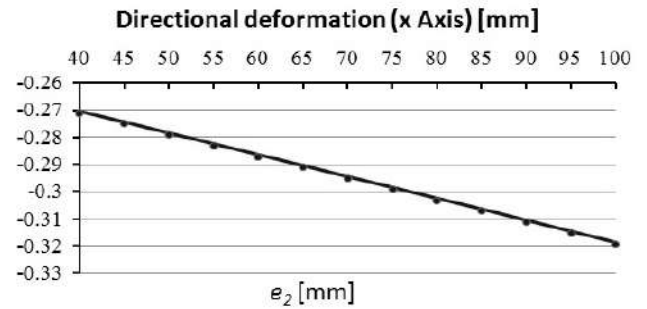


Fig.7. Minimum directional deformation (x axis)

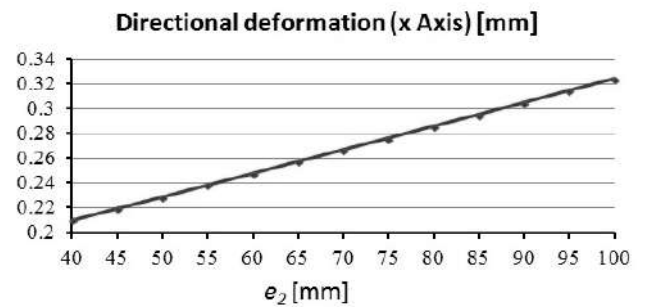


Fig.8. Maximum directional deformation (x axis)

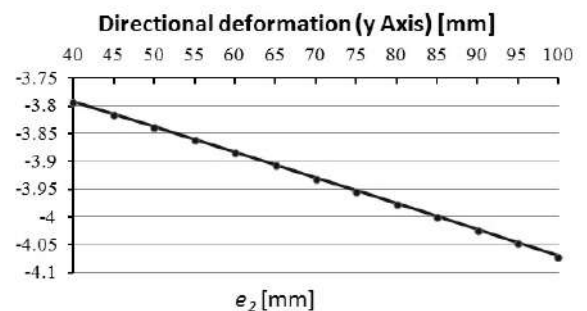


Fig.9. Minimum directional deformation (y axis)

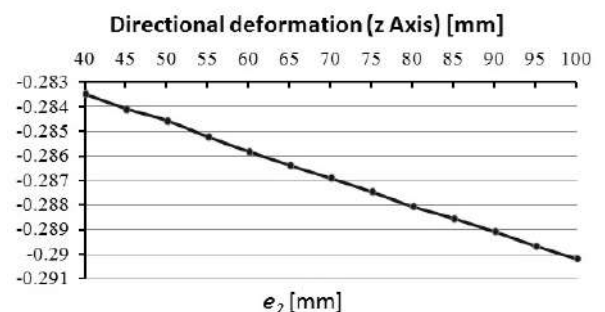


Fig.10. Minimum directional deformation (z axis)

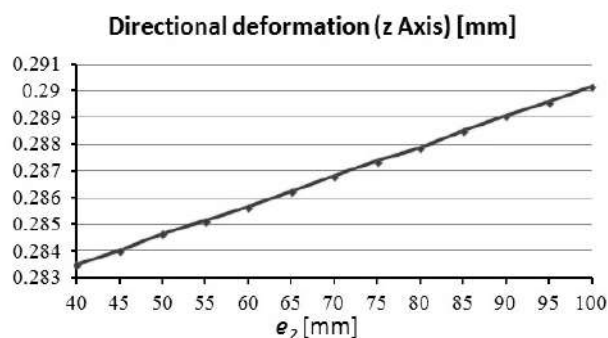


Fig.11. Maximum directional deformation (z axis)

The obtained values for the directional deformation are centralized as follows: in table 1, are written the directional deformation on x axis for minimum and maximum values; table 2 contains the minimum and maximum values for directional deformation on y axis and table 3 is filled with the obtained minimum and maximum values for directional deformation on z axis, respectively.

Taking in consideration that the positive values of directional deformation on y axis are very, very small, they were neglected.

Drawing these results depending the eccentricity, it was obtained the evolution of the directional deformation between the considered limits of the eccentricity (Fig. 7, till Fig. 11).

In figure 7 were shown the directional deformation (x axis) with negative values, figure 8 presents the directional deformations (x axis) with positive values, in figure 9 it can be seen the directional deformation (y axis) with negative values, figure 10 contains the directional deformation (z axis) with negative values and figure 11 we have the directional deformation (z axis) with positive values.

As it can be seen by analyzing the figures 7 to 11, the directional deformation on each axis have a linear tendency. The directional deformations in absolute values increase by increasing the e_2 eccentricity.

Important are the directional deformation on y axis, because these deformation tend to take out the biplane disc from the sealing seat of the butterfly valve body. For $e_2=100$ mm value of eccentricity it was obtained a maximum directional deformation value of ~ 4.1 mm (absolute value) which represent under 0.1% of the biplane disc diameter.

5. CONCLUSION

In absolute values, the maximum directional deformation was obtained on y axis of the biplane disc, in upper and lower side of the disc plate.

By moving the eccentricity between considered limits and analyzing the influence of the eccentricity of the butterfly valve biplane disc on directional deformation results by applying structural analysis with finite element method it was obtained on each axis: x , y and z , a linear evolution of the deformation for a given biplane disc.

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GEOMETRY CHARACTERISTICS OF HUMAN BODY MODEL SUITABLE FOR SIMULATION OF THERMAL COMFORT IN AN AGRICULTURAL VEHICLE

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Abstract: In order to investigate thermal conditions inside a vehicle cabin, some kind of human body model is necessary. Apart from the chosen thermal sensation model, the model of the human body must have an appropriate geometry in order to both influences the cabin environment and have the heat transfer on the similar way like the real human body. The paper deals with geometry characteristics of human body model used in research of thermal comfort. Anthropometric data for thermal manikin from several sources were compared and discussed on aspects relevant for simulation of thermal conditions in the cabin of mobile agricultural machinery.

Key words: human body model, thermal sensation, anthropometry

1. INTRODUCTION

The human thermal sensation is in theory related to the heat exchange between the human body and the environment. One of the factors that influence the heat exchange is the area of the body segment surface exposed to the surroundings. Therefore, the model of the human body that is to be used either in experimental or numerical research of thermal comfort must have the size and geometry that will gave consistent results.

Thermal sensation models are mostly based on specific heat exchange (heat flux expressed in W/m^2) between the body surface and the environment. The convective heat transfer coefficient depends on a body characteristic dimension and air velocity; the radiative heat flux depends on so-called view factor, i.e. ratio between the shape and size of the object in relation to the surrounding surfaces. Therefore, human body shape and size has an influence on convection, radiation and moisture evaporation, as the main modes of heat transfer responsible for human thermal sensation [16].

Anthropometry and biomechanics are usually associated to the geometry of a vehicle interior. Regarding the microclimate and interior aerodynamics, human body in a vehicle cabin presents an obstacle to the air flow and emits heat and moisture [9]. The fact is that direction of the airflow from the air-conditioner vents has significant influence on local thermal sensation [4], [14], [13]. Consequently, the direction of the airflow and relative position of certain body parts could be different for the persons of different sizes and shapes.

There are different approaches regarding either physical or numerical models of the human body, called thermal manikins [2], [10], [15], [19]. Research of Melikov and Zhou confirmed that the manikin surface shape and the surface size are important factors for the assessment of thermal sensation [8].

The aim of this paper is to make comparison of different data about body geometry that are used in physical and virtual thermal manikins. In the first part of the paper, the typical human anthropometry data are given. In the second part, some examples of virtual and physical thermal manikins are described by its shape and size.

2. BASIC ANTHROPOMETRIC CHARACTERISTICS OF THE HUMAN BODY

Engineering anthropometry deals with skeletal dimensions which are measured from certain reference points on the bones, body parts shape, contours, area, volumes etc. [1], [3]. Anthropometric parameters that are closely related to the heat exchange surface are body height, body mass and body surface area. In the automotive engineering, in order to ensure typical and safe driving of the vehicle, the interior geometry must be designed and adjustable in such way to be normally usable by almost whole adult population. Relevant anthropometric dimensions cover a range from 5th percentile female to 95th percentile male sizes. Defining and measuring vehicle seating accommodation in a process of vehicle interior design is done using so-called H-point machine. Its purpose is to insure that drivers of various sizes can comfortably reach and operate vehicle's controls, fulfil the requirements for head space and driver's field of view (standards SAE J826, SAE J1052, SAE J1050 and SAE J941), to mention a few regulations related to the anthropometry. In tractors and machinery for agriculture and forestry there is a different approach. The standards define, among others, seat index point (ISO 5353:1995), dimensions related operator's seating accommodation (ISO 4253:1996) and directive relating operating space (88/41 4/EEC).

However, since the farm tractors or other mobile agricultural machines are mainly driven by male operators, it is presumed that for evaluation of thermal conditions inside the cab, a thermal manikin should have an appropriate size [7]. Therefore, from the population

data, the data for 50th percentile males are taken as a reference values. The data are shown in Table 1. Refer to fig. 1 for body dimension presentation. Body surface area is calculated from body weight m_b [kg] and height h_b [m] using the formula by DuBois [12]:

$$A_{Du} = 0.202 m_b^{0.425} h_b^{0.725} [m^2] \quad (1)$$

Table 1. Anthropometric data for 50th percentile males by Kroemer et al., 1994 [1]

No.	Dimensions	value
1	abdominal extension (stomach) depth, sitting, m	0.299
3	acromial height, sitting, m	0.646
4	bideltoid breadth, m	0.546
14 - 20	forearm length, m	0.290
21	hip breadth, m	0.367
	knee height, m	0.559
27	shoulder – elbow length, m	0.366
26+28	sitting height, m	0.914
29	stature, m	1.756
32	thigh length, m	0.616
	weight, kg	79
	body surface area, m ²	1.946

Important research regarding the anthropometry of the tractor driver was done by Hsiao et al. [7]. The data relevant for this paper obtained from 88 male agriculture workers are shown in Table 2. Dimensions are shown in figure 1. Body surface area is calculated using the formula by DuBois [12].

Table 2. Anthropometry data of the tractor driver by Hsiao et al. [7]

No.	Dimensions	value
1	abdominal extension depth, sitting, m	0.278
3	acromial height, sitting, m	0.607
4	bideltoid breadth, m	0.505
14 - 20	forearm length, m	0.285
21	hip breadth, m	0.406
23	calf length, m	0.427
27	shoulder – elbow length, m	0.367
26+28	sitting height, m	1.372
29	stature, m	1.774
32	thigh length, m	0.423
	weight, kg	87.8
	body surface area, m ²	2.05

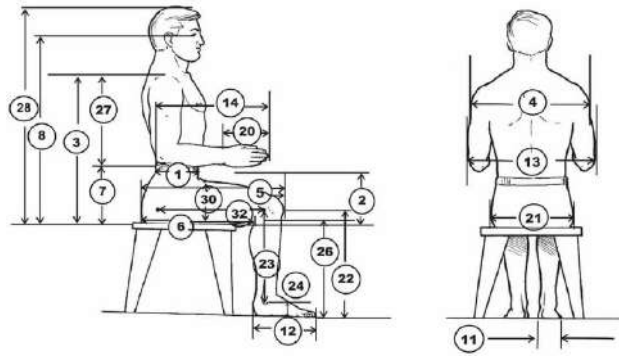


Fig.1. Main dimensions of human body in sitting position [7]

3. EXAMPLES OF ANTHROPOMETRIC DATA OF THE HUMAN BODY MODELS

The main dimensions of body segments that are compared in this research are its characteristic dimensions (diameter, length, etc.) and surface area. Stature and total surface area are taken into account too. The smaller body parts are omitted from the analysis, but they are included in total surface area, mass and overall dimensions.

3.1. 25-node thermal model

Stolwijk and Hardy [12] in their 25-node thermal model have defined a dimensional characteristics shown in table 3. The geometry is simplified and limbs are not divided in smaller segments (Fig. 2). The model presents a thermoregulation system and the surfaces are only an interface for heat exchange between the human body and the surroundings.

Table 3. The dimensions and surface area of simple human body model by Stolwijk and Hardy [12]

Segment	length, m	diameter, m	surface area, m ²
head		0.1976	0.1326
arm	0.56	0.104	0.1268
trunk	0.60	0.294	0.6804
leg	0.80	0.1284	0.2983
body	1.72		1.8877

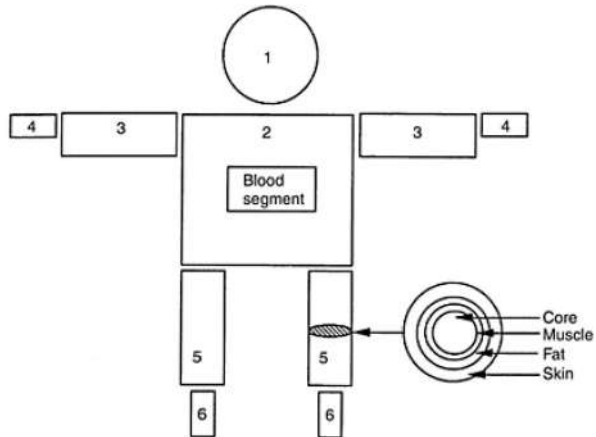


Fig.2. Simple body presentation of 25-node model by Stolwijk and Hardy [12]

3.2. THERMODE 193-node model

Anthropometric data of a Southern Italy standard man were used as a base of a simplified model of the human body [5], fig. 3. This body model was regulated by THERMODE 193-node thermoregulation model, for prediction of local skin temperature. The surface areas shown in the table 4 are calculated assuming a cylindrical shape of the limbs.

Table 4. Anthropometric data for THERMODE 193-node thermoregulation model [5]

Segment	length, m	diameter, (width/depth) m	surface area, m ²
head	0.242	(0.155/0.193)	
upper arm	0.321	0.101	0.102
lower arm	0.239	0.076	0.057
trunk	0.652	(0.360/0.236)	
thigh	0.370	0.183	0.213
calf	0.465	0.094	0.137
body height	1.729		

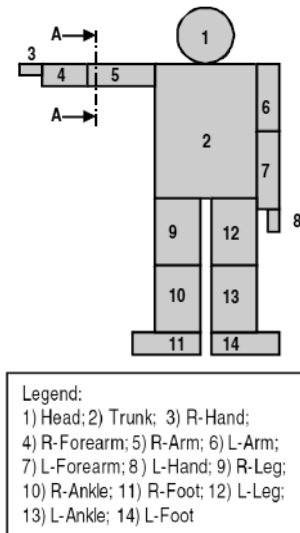


Fig.3. The body segments of THERMODE 193 [5]

3.3. Multi-node thermoregulation model 65MN

Tanabe et al. developed multi-node thermoregulation model (65MN) based on the averaged man [18]. The surface areas of main body segments are given in the Table 5. The surface area of the trunk in this overview is a sum of areas of chest, pelvis and back surfaces. Body height of the model is 1.75 m. Surface mesh of virtual thermal manikin used or radiation modelling is shown in Figure 4.

Table 5. Surface areas of human body model 65MN [18]

Segment	surface area, m ²
head	0.138
upper arm	0.089
lower arm	0.060
trunk	0.523
thigh	0.165
calf	0.112
whole body	1.720

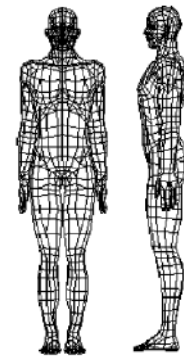


Fig.4. Human body shape for modelling of thermal radiation in CFD [18]

3.4 Virtual thermal manikin by Sorensen and Voigt

Surface area of the individual body segments of virtual manikin used in simulations performed by Sorensen and Voigt [17] are given in the Table 6. They used laser scanned manikin "Maria", made by PT Teknik (<http://pt-teknik.dk>) to create 3D surface for virtual thermal manikin, Fig. 5. The surface area of the trunk given in the table is a sum of areas of chest, pelvis and the back surfaces. The manikin's body height is 1.68 m.

Table 6. Surface areas of virtual thermal manikin by Sorensen and Voigt [17]

Segment	surface area, m ²
head	0.1206
upper arm	0.0762
lower arm	0.0516
trunk	0.4695
thigh	0.1876
calf	0.1082
whole body	1.5940

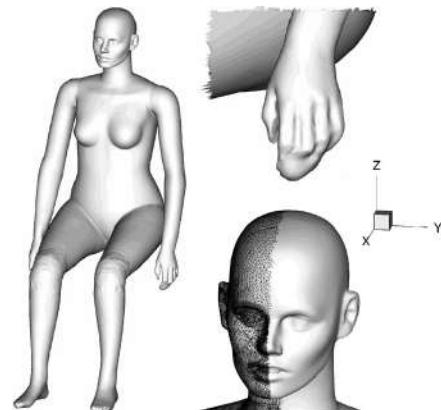


Fig.5. The virtual thermal manikin by Sorensen and Voigt [17]

3.5 Computational thermal manikin by Zhu et al.

Zhu et al. developed 3D computational thermal manikin and compared it with Sakoi's human thermal physiological model of the human body, cited in they paper [19]. There is no data about manikin's height. In the table 7, the surface of the head does not include the surface of the neck and the area of the trunk is a sum of torso and waist areas.

Table 7. Comparison of surfaces area of Zhu's and Sakoi's models [19]

Segment	surface area - Zhu's model, m ²	surface area - Sakoi's model, m ²
head	0.0876	0.1117
upper arm	0.0782	0.0988
lower arm	0.0535	0.0679
trunk	0.5287	0.6518
thigh	0.1171	0.1482
calf	0.0802	0.1024
body	1.4547	1.8370



Fig.6. Body shape used in Sakoi's model [19]

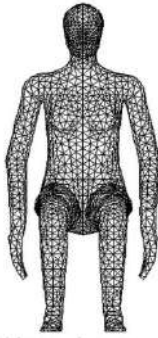


Fig.7. Mesh of the Zhu's computational manikin surface [19]

3.6 MANIKIN2 and MANIKIN3 by Nilsson

The virtual manikin called MANIKIN3 that Nilsson used in his work is of cubical shape and does not have the highest details in body shape reconstruction [10]. However, the manikin is shaped with almost the same size, areas and number of zones as the real MANIKIN2, giving the same constant heat flux to the surrounding air (table 8, figures 8 and 9). The surface area of the head surface in the table is a sum of areas of scalp and face surfaces. The surface area of the trunk consists of areas of upper and lower back and chest surfaces.

Table 8. The surface areas of body segments for MANIKIN2 and MANIKIN3 [10]

Segment	MANIKIN2, m ²	MANIKIN3 CFD, m ²
head	0.1397	0.180
upper arm	0.0808	0.065
lower arm	0.0571	0.070
trunk	0.4802	0.505
thigh	0.1476	0.140
calf	0.1035	0.080
body	1.5919	1.605

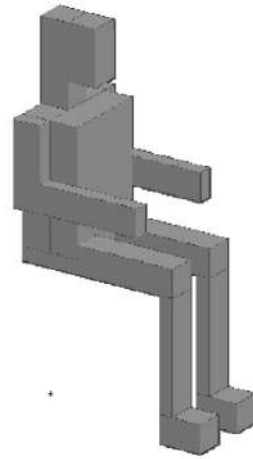


Fig.8. MANIKIN3 [10]



Fig.9. MANIKIN2 [10]

3.7 Areas of body zones for calculation of equivalent temperature

Olesen in Bruel&Kjaer's *Application notes: Evaluation of the Thermal Environment in Vehicles* [11] gives the guide for simplified evaluation of thermal sensation described by local equivalent temperatures. The local values for individual body parts are to be weighted according to the areas of different body parts in relation to the area of the whole body. In the table 9 the surface areas are given, but there is no information on source of those anthropometric data. Figure 10 shows the methods of spatial measurement of equivalent temperatures using special sensors. This human-shaped measuring system and the method present a very simplified physical thermal manikin.

Table 9. Area of different body parts for calculation of weighting factors of local equivalent temperatures [11]

Segment	surface area, m ²
head	0.180
upper arm	0.077
lower arm	0.062
trunk	0.469
thigh	0.160
calf	0.140
whole body	1.751

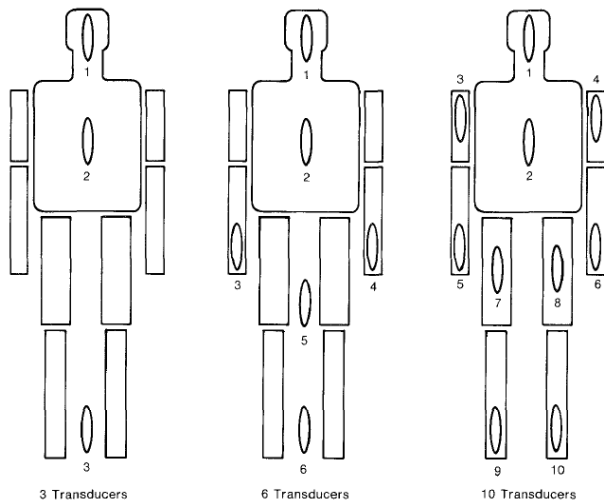


Fig.10. Recommended positions of the equivalent temperature sensors on human-shaped skeleton [11]

3.8 Virtual Thermal Manikin developed by Faculty of Technical Sciences (VTMFTS)

The virtual thermal manikin developed on Faculty of Technical Sciences – University of Novi Sad is shown on Fig. 11 [15]. This model is hereafter referred to as VTMFTS. VTMFTS is experimentally verified and its purpose is comparative evaluation of settings and design of the vehicle cabin air-conditioning and air distribution systems. The main body dimensions are adopted from CATIA database for 50th percentile European male, and the body is divided into 18 segments. The some geometrical characteristics of VTM are given in table 10. The smaller body segments are not listed in the table (neck, hands, and feet). The posture is defined by characteristic points in places where the main body joints are (hips, shoulders, neck, elbows etc). Standing body height is 1.696 m and sitting height is 1.266 m.

Table 10. The main geometrical characteristics of VTMFTS

Segment	length, m	diameter (width/depth), m	surface area, m ²
head	0.200	(0.150/0.195)	0.175
upper arm	0.293	0.090	0.094
lower arm	0.271	0.074*	0.074
trunk	0.539	(0.316/0.223)	0.516
thigh	0.430	0.145*	0.136
calf	0.416	0.110	0.145
whole body			1.795

*average diameter

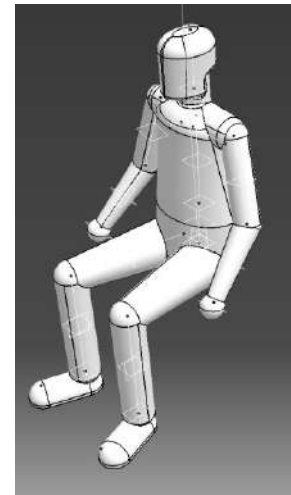


Fig.11. Virtual thermal manikin developed on FTS – Novi Sad

4. DISCUSSION

There is a variety of approaches of modelling of the human body geometry, apparently regardless of the available computational power or software used. Some models realistically present the shape of the human body, like those described in [17], [18] and [19]. Such models can be difficult to define mathematically and consequently it is more demanding to make changes in posture and size. On the other hand, other very geometrically simplified models are presented too [10]. Most of the manikins or human body models are described by segments surface areas, but there is no data about size and/or shape. The segment surfaces of 25-node [12] and THERMODE 193 [5] models are important for thermoregulation and radiation heat exchange, and those models are not directly related to CFD simulations. In order to compare characteristics it would be important to have complete geometrical description of the model. Anthropometric data of a sample of agricultural workers are similar to data of 50th percentile male. Segments surface areas are not available for comparison, but it can be assumed that they could be of similar areas too. Size of VTMFTS and THERMODE 193 are similar to size of the 50th percentile male, making them suitable for application in simulation of thermal conditions in cabins of agricultural machinery. Other models described in this paper are smaller in size, especially those based on female physical manikins.

5. CONCLUSION

The paper deals with geometry characteristics of human body model used in research of thermal comfort. Anthropometric data for thermal manikin from several sources were compared and discussed. The result showed that anthropometric data of a sample of agricultural workers are similar to data of the 50th percentile male. Size of the virtual thermal manikin developed on Faculty of Technical Sciences (VTMFTS) is similar to size of the 50th percentile male, making it suitable for application in simulation of thermal conditions in cabins of agricultural machinery.

Other important relation between the anthropometry and air-conditioning system is position of the HVAC controls regarding the hand reach. Furthermore, the simplicity of the controlling and automatization of the system are the important ergonomic parameters that should be considered during the design of the HVAC system.

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DEVELOPMENT AND MANUFACTURING OF SENSOR CASES FOR MEMS INERTIAL MEASUREMENT UNITS

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Milan RACKOV

Abstract: This paper discusses about the design and simulation of housings for inertial sensors. We used additive manufacturing to develop several models. Deformation analysis for each model was made using the specialized software COMSOL.

Key words: IMU, ABS sensor case, Deformation Analysis

1. INTRODUCTION

Inertial sensors are used in various motion monitoring applications in medicine, entertainment, animation, robotics. Several studies are taking advantage of the small inertial sensors to monitor the movement of the human spine [1].

Additive manufacturing or 3D printing is considered to be the next major technological revolution [5]. Using a 3D printer allowed us to develop and improve the inertial sensor housings. MakerBot printer was chosen because of its accessibility and very intuitive software [6].

The concept of designing cases developed for inhabiting inertial sensors is to protect the sensors and facilitate the handling processes. Also, one major problem that inertial measurement units have is the sensibility to ferrous material which causes them to lose their calibration. In order to reduce the risk of *magnetic disturbances*, a case made of non ferrous material, such as plastic or ABS, was developed. The IMU used in this study is MPU 9150, the world's first 9-axis motion tracking device that offers high-performance at low power consumption and low costs [7].

Another aspect that is addressed in this study is how to create a hard case that would resist to different types of shocks and to have a relative small dimension.

2. MATERIALS AND METHODS

To produce the real cases the MakerBot Replicator 2X was used, with a build volume of 24.6 cm x 16.3 cm x 15.5 cm. Layer height settings: High 100 microns, Medium 200 microns, Low 300 microns. Positioning precision of XY: 11 microns and Z: 2.5 microns. Filament diameter: 1.75 mm and a nozzle diameter: 0.4 mm.

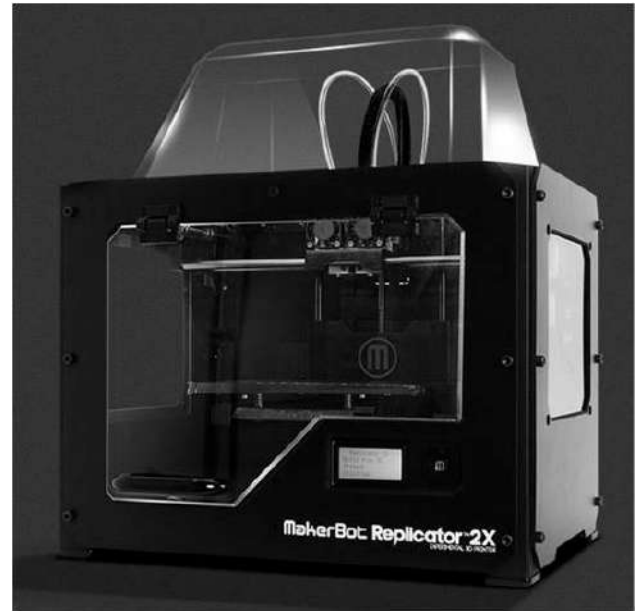
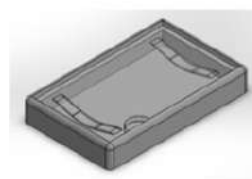


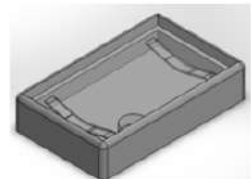
Fig.1. Maker 3D printer
(source: <http://www.makerbot.com/>)

2.1. Design for Encapsulated case

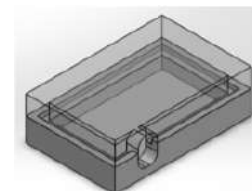
The advantages of the encapsulated model are the following: better protection against dust or even water and the sensor cannot be accessed or removed by patients or other unauthorized people.



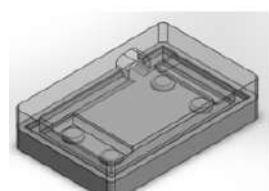
INFO:
1 mm thick
Fig.2 Model 1



INFO:
2 mm thick
Fig.3 Model 2



INFO:
2 mm thick
Fig.4 Model 3



INFO:
2 mm thick
Fig.5 Model 4

First to cases use the same geometry but different size of thickness. To see what is the best solution to make the model small as possible.

For case model 3 we have bad result because the sensor position inside was too big and the sensor can move inside.

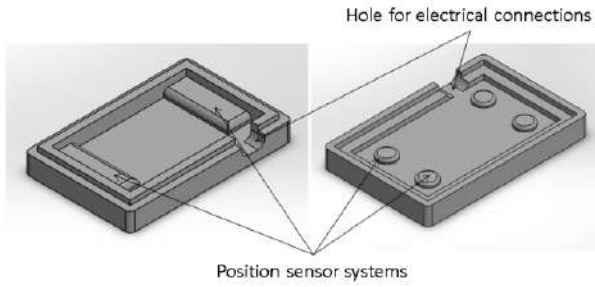
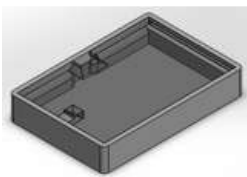


Fig.6. Model 4 – final model

The best case on this class is Model 4, the model can be seen in Fig.6.

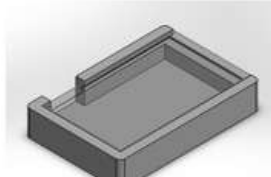
2.2. Design for demountable case

When the fifth model was printed, a problem in the design was observed: the assembly system is small and fragile, thus being easy to deform or even break. In order to avoid this problem, the top part was redesigned and made to slide (sixth model. This model is good in theory, but the 3D printer we used has some limitations and the printed product did not perform as expected. The model 6 was printed to times, one with support material and one without to see the differences.



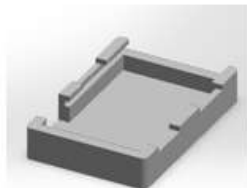
INFO:
1.5 mm thick

Fig.7. Model 5



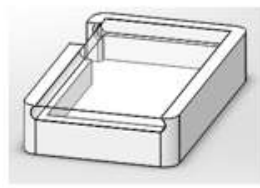
INFO:
2 mm thick

Fig.8. Model 6



INFO:
2 mm thick

Fig.9. Model 7



INFO:
2 mm thick

Fig.10. Model 8

In model 7 we tried to fix the problem by integrating pockets in the design, the manufactured product is presented in Fig.20. Because of the problem we encountered with printing model 7, we decided not to make simulations and improve the design. In the eighth model we changed the contact surface, from a rectangle to semicircle which presented good results in printing and deformation simulation.

2.3. Simulation specifications

The study of stationary forces or external influences are used when field variables do not change over time. This study could give the ability to find the displacement of a surface area and having a constant force in the time domain.

There are many external influences given from the environment that could affect a geometrical design figure such as electromagnetics, heat transfer, fluid flow, chemical reactions and mechanical forces and moments. Mechanical forces and moments are used to compute several solutions, such as a number of load cases, torsional stressed and bending moments, or to track the nonlinear response of a slowly varying load.

The software used to study the casing of the sensor is COMSOL. The general properties of the material used in constructing the casings of the sensor:

Table 1. General properties of the used material

Property	Name	Value	Unit
Density	rho	1020	Kg/m ³
Young's module	E	2300	MPa
Poisson's ratio	nu	0.24	1

The general equation of calculating the deformation at an interval of time with fixed forces could be represented below:

$$e_a \frac{\partial^2 u}{\partial t^2} + d_a \frac{\partial u}{\partial t} - \nabla \cdot (c \nabla u + a u - \gamma) + \beta \cdot \nabla u + a u = f \quad (1)$$

where:

- e_a is the mass coefficient;
- d_a is the damping coefficient or a mass coefficient;
- c is a diffusion coefficient;
- α is the conservative flux convection coefficient ;
- β is the convection coefficient;
- a is the absorption coefficient;
- γ is the conservative flux source term;
- f is the source term;

General formula for total stress:

The total stress in computed via components

$$s^{ij} = s_0^{ij} + C^{ijkl} \left(\varepsilon_{kl} - \alpha_{kl} (T - T_{ref}) - \varepsilon_{0kl} \right) \quad (2)$$

Formula for elasticity tensor component:

$$C^{ijkl} = \lambda g^{ij} g^{kl} + \mu (g^{ik} g^{jl} + g^{il} g^{jk}) \quad (3)$$

For this model we have the following equation:

$$s = S_0 + C : (\varepsilon - \varepsilon_0 - \varepsilon_{inel}) \quad (4)$$

s is total stress

S_0 is initial stress

C is elasticity tensor

The total strain tensor is written in terms of the displacement gradient:

$$\epsilon = \frac{1}{2}(\nabla u + (\nabla u)^T) \quad (5)$$

he gradient of the displacement, which occurs frequently in the following theory, is always computed with respect to material coordinates. In 3D:

$$\nabla u = \begin{bmatrix} \frac{\partial u}{\partial X} & \frac{\partial u}{\partial Y} & \frac{\partial u}{\partial Z} \\ \frac{\partial v}{\partial X} & \frac{\partial v}{\partial Y} & \frac{\partial v}{\partial Z} \\ \frac{\partial w}{\partial X} & \frac{\partial w}{\partial Y} & \frac{\partial w}{\partial Z} \end{bmatrix} \quad (6)$$

$$\sigma \cdot n = F_A \quad (7)$$

$$F_A = \frac{F_{tot}}{A} \quad (8)$$

F_{tot} : is the force applied on the Y axis;

A is surface area;

n is coefficient of deformation;

σ is stress;

3. RESULTS AND DISCUSSIONS

3.1. Simulation result of the encapsulated cases

The simulation process consisted in applying a load on the casing in order to see the deformation and stress zones. The following forces were applied: 9.8 N; 19.6 N; 98N; 980N; 1960N. The software used is Comsol.

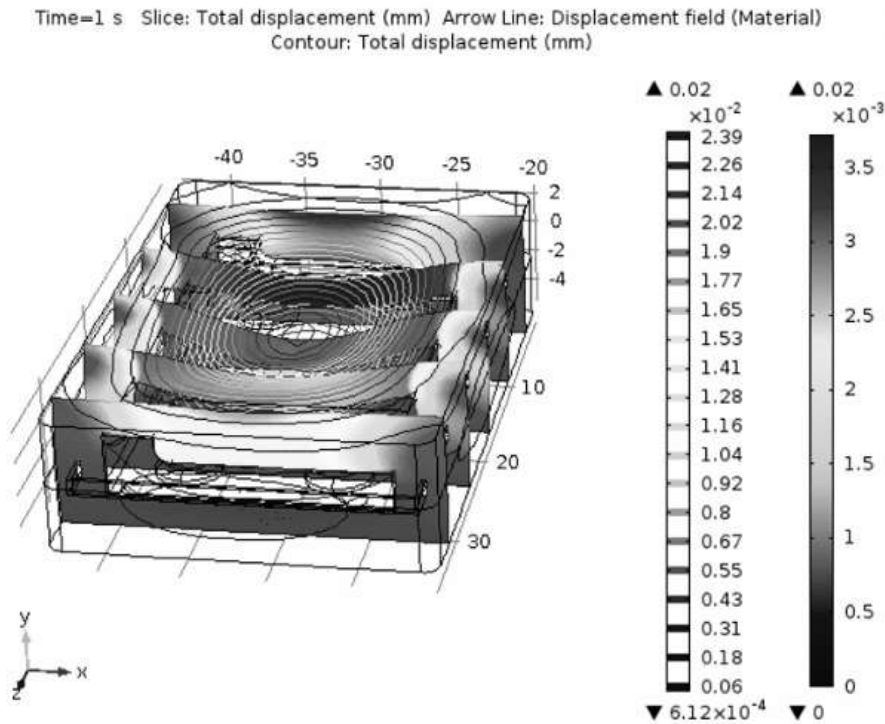


Fig.11 Model 4 – isometric view of the simulation

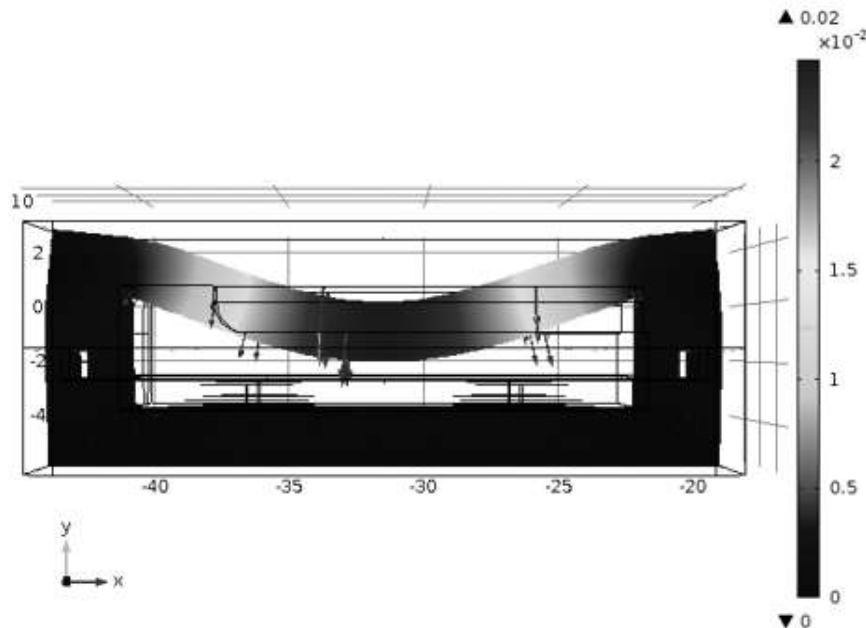


Fig.12 Model 4 – lateral view of the simulation

In Fig.11 we can observe that the biggest displacement/deformation is in the center of the casing, a situation in which the sensor would be damaged. A lateral view of the result of the simulation is presented in Fig.12.

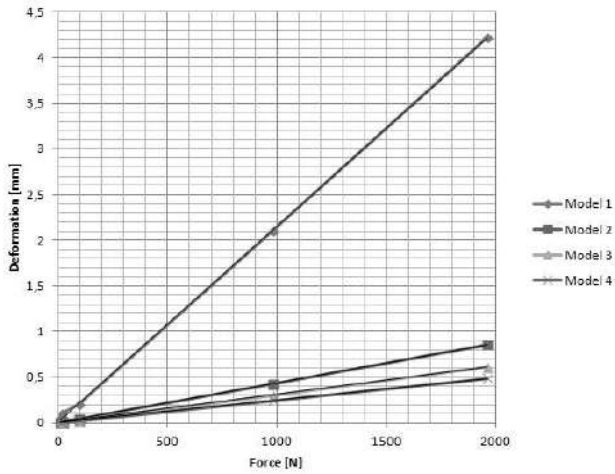


Fig.13. Result of the deformation from the encapsulated model

Several 3D models were designed and simulated. In Fig.13 we can see that the last model has the best behavior under stress.

3.2. Simulation result of the demountablecases

The forces applied on the demountable cases are the same as for the encapsulated cases: 9.8 N; 19.6 N; 98N; 980N; 1960N.

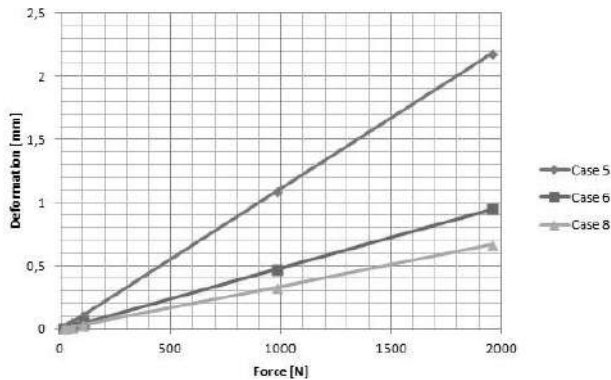


Fig.14. Result of the deformation from the demountable model

The design for model 8 is clearly better than model 5 as it presented a deformation at least two times smaller.

3.3. Manufacturing results of encapsulated cases

In this chapter we present the printed encapsulated cases and the main observations.

In Fig.15 we present the Model 1 of the case and with the red marks is the part where we had problems, due to the limitations of the 3D printer.

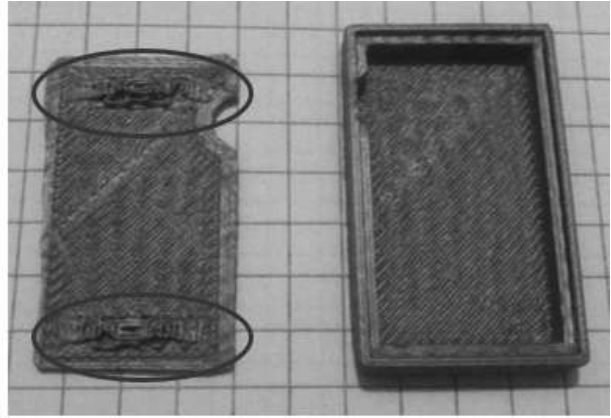


Fig.15. Model 1 – 3D print model with visible problems

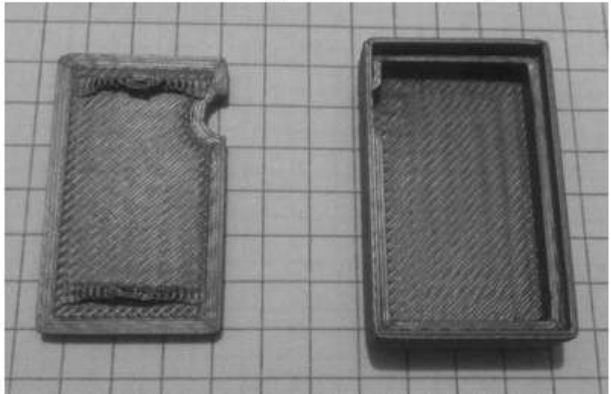


Fig.16. Model 2 – 3D print model with 2 mm thickness

To overcome the problem encountered in model 1, we decided that the minimum thickness that we can use is 2 mm.

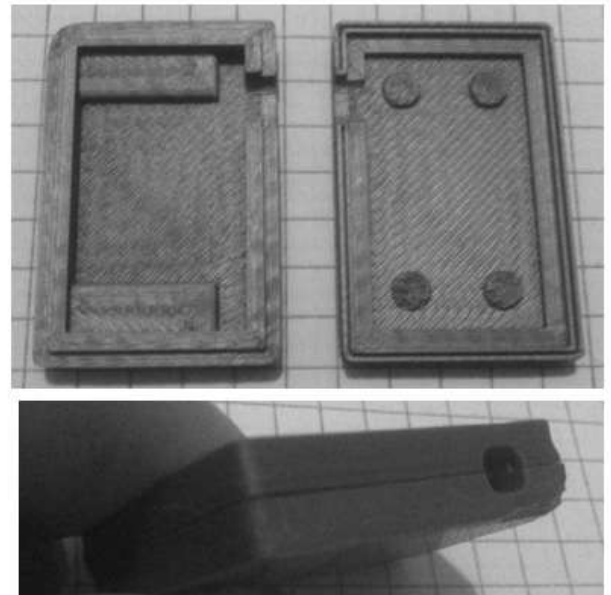


Fig.17. Model 4 – 3D printed model

In Fig. 17 we present the model 4, which is the final model of the encapsulated cases.

3.4. Manufacturing results of demountable cases

In this chapter we present the printed demountable cases and the main observations.

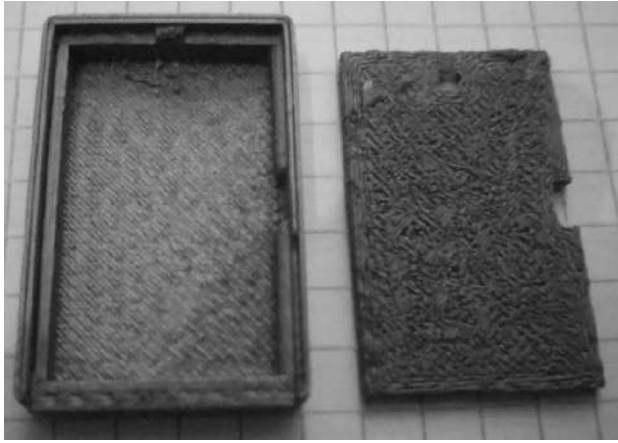


Fig.18. Model 5

The main problem with model 5 was the locking mechanism, which did not work properly.

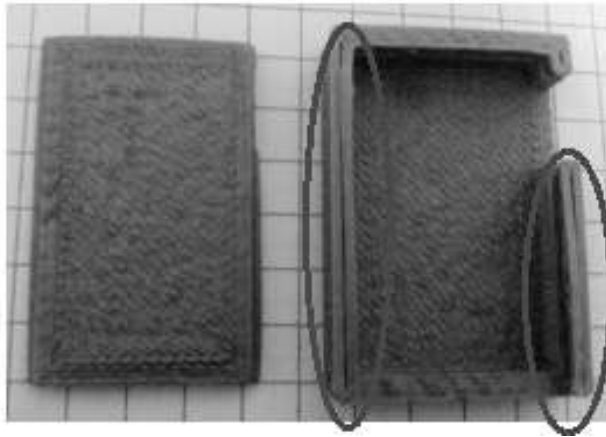


Fig.19. Model 6 – 3D print model with visible problems

In

Fig.19 we can observe that the margins have some gaps which will not allow for the top cover to slide properly.

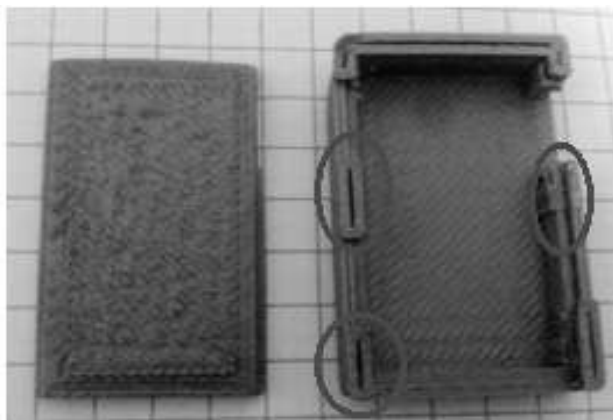


Fig.20. Model 7– 3D print model with visible problems

After printing the model 7, we observed that the problems with model 6 were not resolved.

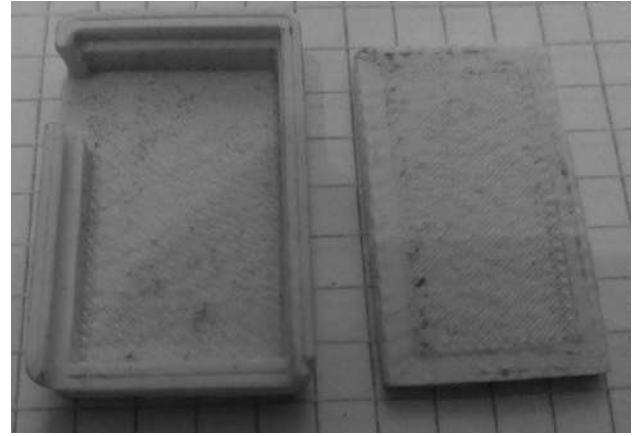


Fig.21. Model 8 Final demountable mode, disassembled



Fig.22. Model 8 – Final demountable mode, assembled

The improved design is presented in Fig.22.

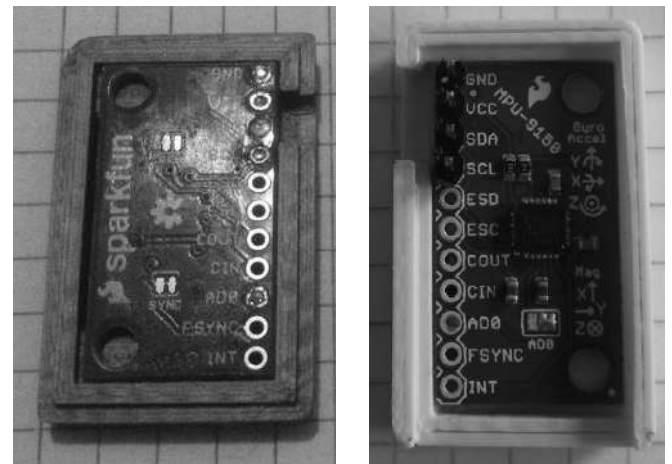


Fig.23. Model 4 and 8 – Final models

The final models for the encapsulated and demountable cases are presented in Fig. 23.

4. CONCLUSION

The deformation for 49 N is only 0.0123 mm, this is the maximum pressure that can apply to this sensor in this specific application. But we also test the case up to 2000N just to see the deformations.

The encapsulated model is ideal for this application because it provides better protection against vibrations, humidity, and sweat.

A disadvantage for the encapsulated case is that the sensor cannot be replaced if it gets damaged or stops working properly.

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AUTOMATIC TECHNOLOGY FOR GLUING CERAMIC HOBS

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Gheorghe ACHIMĂȘ
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Abstract: Accelerated development of engineering sciences in recent years has led to the need for new manufacturing methods moreover, current industrial competition requires new methods of rapid manufacturing at acceptable cost. Assembling the ceramic hobs is carried out by using a synthetic adhesive gluing, which is a relatively new method. Adhesive gluing using synthetic materials has gained in the last decade a wide range of use, gluing sheets or metal parts, non-metallic or combined, especially in light construction and precision mechanics, but also in mechanical engineering.

Key words: gluing, adhesive, ceramic hobs, technological stages

1. INTRODUCTION

Ceramic hobs, but more exactly polycrystalline glass is a material produced by controlled crystallization of glass base. Ceramic glass has the advantage of glass manufacturing and the special properties of ceramics. It is highly resistant to temperature variations including at 800 - 1000°C and very resistant to traction 200-250 MPa. Improvement of cookers led to the design and manufacture of new models. Thus, traditional models that were powered by gas have been replaced by new models supplied with electricity. These new cookers have ceramic hobs under which electric resistors are placed for achieving the desired temperature, thus replacing the conventional gas burners plate.



Fig.1. Classical gas burners plate



Fig.2. Ceramic hob with electric resistors

Ceramic hob is a bonded assembly consisting of: ceramic glass+adhesive+metal frame.

2. GLUING TECHNOLOGY

Gluing is defined by permanent joining of two solid bodies using an adhesive, united parts must not necessarily be of the same material. Gluing is composed of materials to be joined and the adhesive. Strength of bonded joints is primarily based on two types of forces: inner strength, materials cohesion and effects of forces, adhesion (adhesion) occurring at the interface surface of materials to be bonded and adhesive.

Definition of adherence - adherence means the core forces or mechanisms composing each substrate of adhesive, the term refers to all mechanisms of adhesion or forces located in a thin layer (boundary layer) between the substrate and the adhesive itself.

Definition of cohesion—cohesion forces are all forces and mechanisms that hold the adhesive. The cohesion force is the force of attraction that occurs between the atoms of the solid substance and molecules of the liquid (adhesive).

Adhesive gluing technology does not require high temperature that can affect surfaces, concentrations effects are avoided and thermal stress concentrations as well.

There are two ways for assembling the ceramic hobs:

- When ceramic glass is on top of the metal frame, used at built-in hobs;
- when ceramic glass in under the metal frame, used at the oven.

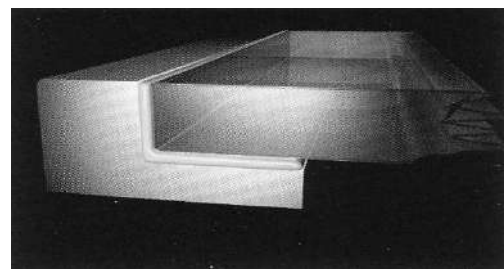


Fig.3. Ceramic plate with ceramic glass on metal frame

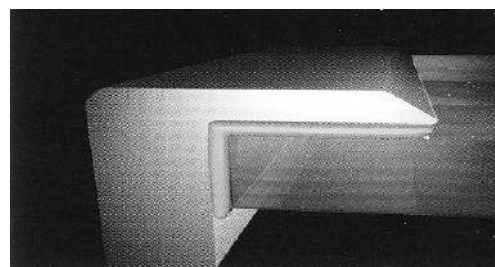


Fig.4. Ceramic plate with ceramic glass under metal frame

2.1. Technological stages

To form homogeneous gluing (there is only a layer of adhesive between materials to be joined) the following order of actions has to be kept:

- Surface treatment process;
- Prepare adhesive;
- -Apply adhesive on surfaces;
- -Join and fix pieces;
- -Hardening adhesive;
- -Finishing operations.

In table 1 are presented the technological stages with related technological suggestions.

Table 1. Technological stages and suggestions

No.	Operation	Technological suggestions
1.	Prepare surfaces.	-Chemical cleaning with carbon tetrachloride, trichlorethylene, methylene chloride, alkaline substances. -Sand blasting. -Application of adhesive substrates, fabrics, cotton or glass, moistened with solvent or otherwise.
2.	Prepare adhesive before use.	-Monocomponent adhesive contain all elements required in hardening. -Bicomponent adhesive are prepared before use. Dosage and means of bonding based on receipt. - Combination time 30 min - 1 day, in conformity with the adhesive.
3.	Apply adhesive on surfaces.	-Manually-by brush, spatula, spray-or mechanically. Possibly by dipping or spreading the adhesive strips. -Thin tapes of 0,03 - 0,1 mm. Application of fabric determines the appropriate thickness.
4.	Prior dry of adhesive layer.	-Keeping pieces outdoor for solvent evaporation before bonding at 15-90 °C, in conformity with the adhesive.
5.	Joining.	-Overlapping parts and maintenance to curing temperature, with or without pressure -Curing duration is from 5 min to 24 hours, meet manufacturer specifications.
6.	Cleaning the joint.	-Clean with solvent, if necessary.
7.	Quality control joint.	-By hammering: high tones - good bonding, low tones - defects. - Ultrasounds. -Breaking Through breaking of samples glued in similar conditions.

		-Through destructive attempts of a number of pieces from the manufactured batch.
8.	Special measures during bonding.	-Free dust rooms, temperature at 18 - 25°C. -After cleaning, pieces should not be touched. Eventually use gloves. -Compliance with precise duration of preliminary drying temperature; in special cases, heating parts. -Uniform distribution of adhesive on bonded surfaces.
9.	Labour protection measures.	-Forced ventilation of the room where bonding is carried out. -Wear safety equipment.

The quality of the soldered joints is determined primarily by the adhesion (adherence), by the effects of forces occurring in the area of contact between the combined material and adhesive.

According to the theory of mechanical joining between the bearing surface and adhesive (or any coating, paint) is achieved by infiltrating the adhesive within pores, asperities (roughness) of the bearing surface.

Based on research carried out in last two decades, the high resistance of the links obtained by increasing the roughness (asperities) of the surface, it can be explained in this way, which derives from the theory of mechanical bonding. If the infiltration of adhesive among the surface inequalities can really be demonstrated, the strength of gluing resulting from the mechanical connection can be significant.

3. INDUSTRIAL ROBOTS

Robots are used in different industries to fix pieces very rapidly and accurately. Some robots, without being limited to orthogonal movements have assemblies with many joints enabling simulation of the movements of human arms. These joint sections must have a high swing accuracy and high rigidity ensuring that sudden stops do not prevent them from operating.

The robot can be defined as a facility for automating operations which used to be carried out by man under the supervision of the eye, eye-hand coordination being carried out by the brain. Besides robots, manipulators perform handling operations.

Robots have a mechanical structure more complex with higher degrees of mobility and is run by a precise program. Manipulators have a mechanical structure, simpler with fewer degrees of mobility and is led by a rigid schedule.

3.1. Main mechanisms of robots

Industrial robots consist of three main groups of mechanisms:

-The driven mechanism of the trajectory (DMT), consisting of those kinematic couplers enabling movement on imposed trajectory. To generate trajectory are necessary three degrees of flexibility: rotation around

the axis Oz; vertical movement along the axis Oz and a radial movement along the axis x.

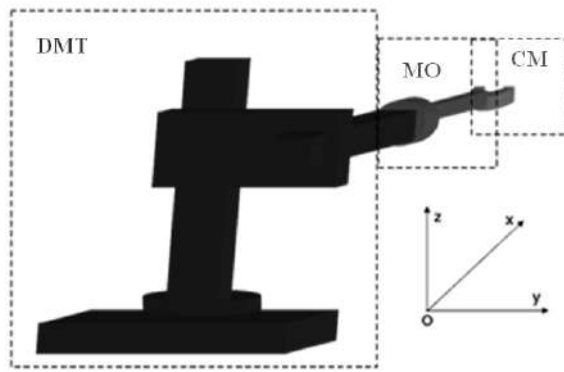


Fig.5. Robot mechanisms

-The mechanism of orientation (MO), consisting of kinematic couplings which ensures the spatial orientation of the object and mechanism performing the rotation by the x, y and z axes, imitating the wrist-forearm of the human hand.

-Clamping mechanism (CM), which provides the fixing of the handled object and has no degree of flexibility. Using the advantage of the industrial robot with high degree of mobility we can use in the soldering process.

3.2. Process automation

The robot chosen to carry out the process is the IRB 4600 from ABB which is a very productive robot, optimized for short time cycles.

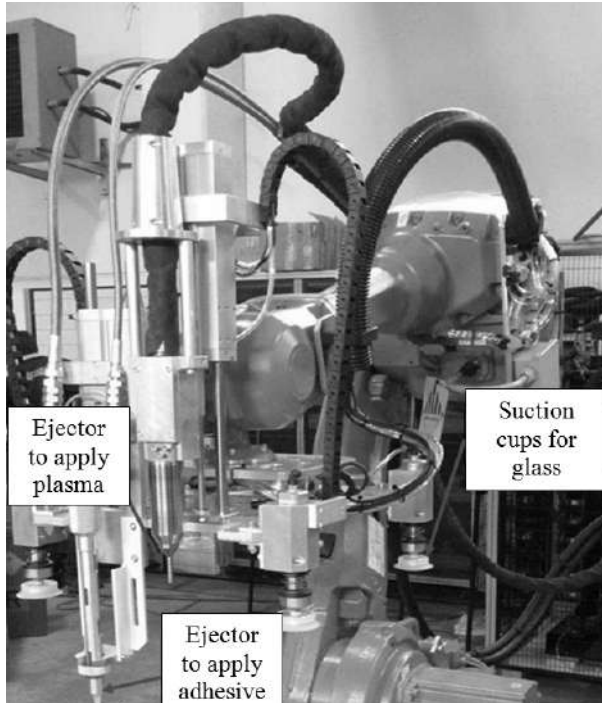


Fig.6. Robot IRB 460 equipped

In order to perform the gluing operation to the end of the clamping mechanism are mounted some devices to supply the technological stages, as follows:

- Device for glass clamping;
- Ejector to apply plasma jet to prepare surface;
- Ejector to apply adhesive.

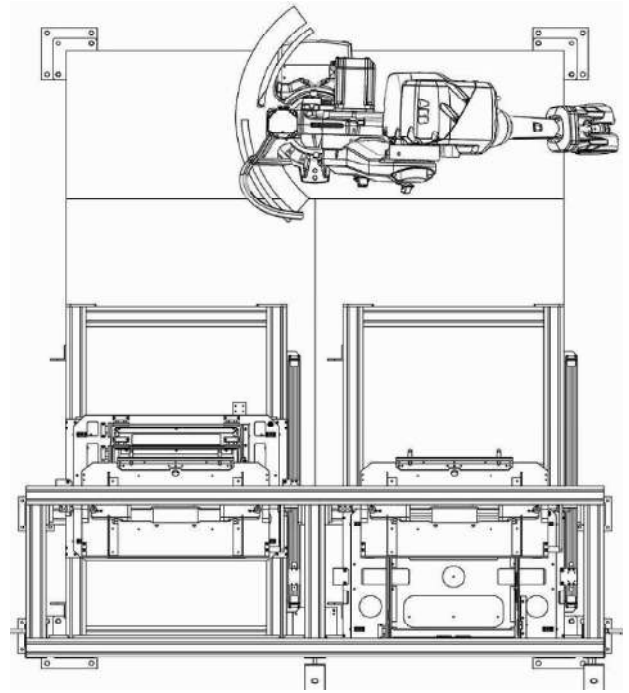


Fig.7. Work stand

Work stand is designed so as two workstations to be successively served. The posts are provided with two devices where the metal frames and ceramic glass are placed and loaded by an operator. While in the first post is carried out the gluing operation, the second is supplied with components. On completion the gluing operation at the first post, the robot move to position two and meanwhile the operator removes the glued assembly and recharge devices then the cycle is repeated.



Fig.8. Successive gluing cycle

Automated gluing operation consists of simultaneous technological stages. First phase is carried out by the operator (1), the second by the robot (2). As work stand is provided with two workstations, the studied gluing operation is a successive-simultaneous operation. Combined successive-simultaneous operation consists of the following stages:

- Supply device with ceramic glass-1;
- Remove bonded assembly-1;
- Supply device with metal frame-1;
- Surface treatment with plasma jet-2;
- Apply adhesive-2;
- Fix ceramic glass-2.

Then, robot moves to the first work stand and repeats the cycle.

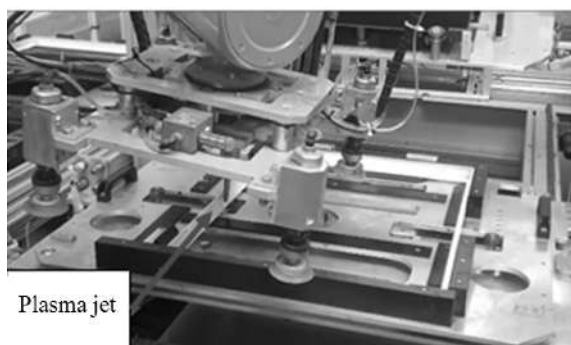


Fig.9. Surface treatment using plasma jet

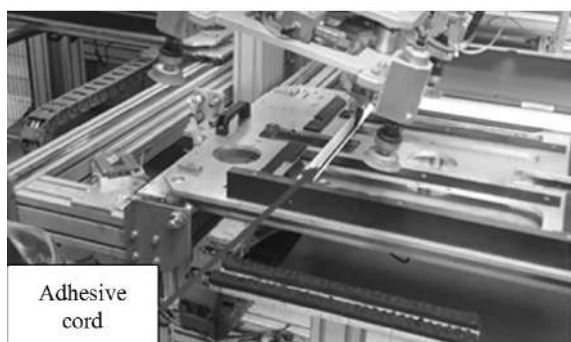


Fig.10. Adhesive application



Fig.10. Fixing ceramic glass

To ensure a good adhesive impregnation into the inequalities of the bearing surface arising from plasma processing, the glued assembly is placed in a pressure device.

4. CONCLUSION

Due to its high degree of mobility and „flexibility” industrial robots have rapidly extended in engineering manufacturing. If the clamping mechanism is multi-functional and associated with a non-conventional technology (as presented), then we obtain a high and improved productivity. Work phases from soldering operations are more complex than those listed in Table 1 but in terms of cycle duration they are shorter, more precisely reduced to minutes or even tenths of seconds.

Use of nonconventional technology with plasma jet leads to an adherent superficial layer, clean without impurities, providing a better grip.

Surface roughness consists of micro-irregularities assembly, these are relatively small related to its depth, but have an important role in obtaining strong gluing.

The higher the surface roughness or asperities the greater cohesive force it provides penetrating the adhesive deeper into fabric to be glued.

Moreover, applying the adhesive using the robot provides a uniform cord of adhesive, improving mechanical resistance and sharing workload evenly.

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COMPARISON OF DIFFERENT FLUIDIC MUSCLES

József SÁROSI

Abstract: *Electrics, hydraulics and pneumatics can be the main motion power of industrial robots. Pneumatic cylinders, pneumatic motors, pneumatic stepper motors and pneumatic bellows are widely used in industrial environment due to their power/weight ratio, power/volume ratio, strength, compactness, simplicity, reliability and cost. Disadvantages of pneumatic actuators can be summarized as follows: difficult to control accurately, air compressibility, compliance and noisiness. Relatively new type of the pneumatic actuators is the pneumatic artificial muscle (PAM) or pneumatic muscle actuator (PMA). Fluidic Muscle made by Festo Company is one of the most investigated commercially available PMA. These muscles can be used in industrial environment as well as in prosthesis or rehabilitation devices. In this paper Fluidic Muscles with different geometric parameters (length, diameter) are compared experimentally.*

Key words: *pneumatic muscle actuator, Fluidic Muscle, experimental setup, force-contraction characteristics*

1. INTRODUCTION

Sensors and actuators are key elements of mechatronic systems. Sensors are required to detect and respond to some type of input from the physical environment (e.g. temperature, position, pressure, force) and actuators are responsible for converting energy into motion [1], [2]. In this study pneumatic muscle actuator as one of the least investigated type of actuators but an important driver element is described and investigated. History of PMAs dates back to 1930s. Unfortunately, due to lacks in technology the production was limited. In 1950s, Joseph L. McKibben was the first who designed an artificial muscle for practical use in medicine. McKibben is often mentioned as the pioneer in PMA. In 1980s, engineers in Birdgestone Company in Japan produced the so called Rubbertuator PMA. Recently, the most often applied is the Fluidic Muscle and also the Shadow Air Muscle (SAM) produced by the Shadow Robot Company.

The structure of the most PMAs can be divided into two main parts: a flexible membrane (e.g. latex, silicone rubber or chloroprene) and a load carrying element (e.g. nylon, fiberglass or aramid). On the basis of their connection, Daerden in [3] discriminates braided muscles, netted muscles and embedded muscles (Fig. 1).

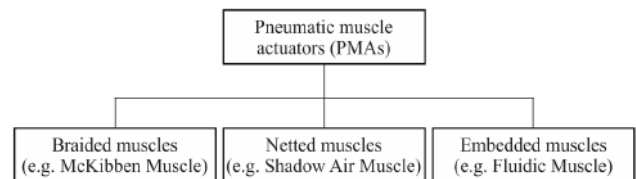


Fig.1. Classification of PMAs

The difference between netted and braided muscles is in the density of the threads in the braided mesh shell (network of load carrying element) surrounding the membrane: it is higher for the braided muscles. In embedded muscles the loaded threads are settled into the elastic tube. In the paper this type of muscles is considered (Fig. 2 and Fig. 3).

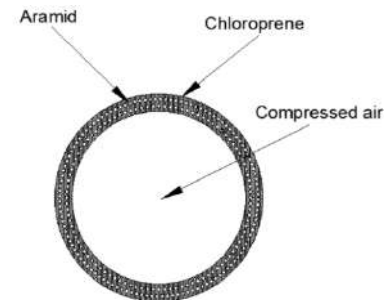


Fig.2. Model of an embedded muscle



Fig.3. Fluidic Muscles with different nominal length and diameter [4]

Working principle of PMAs is simple: when a PMA is pressurized, the flexible membrane tends to increase its volume against the braided mesh shell which is non-extensible, therefore the actuator will be shortened and a pulling force will be produced if the muscle is connected to a load.

This flexible actuator shows similarity to human muscle, because the force and motion generated by PMA are linear and unidirectional. For two-direction motion an antagonistic pair of PMAs or a spring returned PMA has to be used. Typically, one muscle moves the load, while the other serves as a brake. During the motion in opposite direction the mechanisms commute their action. These

serially connected muscles are named antagonistic pair: the muscle for motion is a flexor while the braking muscle is named an extensor or antagonist [5].

PMA's differ from general pneumatic cylinders as they have no inner moved parts and there is no sliding on the surfaces. The main disadvantages are nonlinear and time variable behaviour, existence of hysteresis and step-jump pressure. This is why number of control schemes and static and dynamic models can be found in the literatures. The applied control strategies are collected in [6]: adaptive control, direct continuous-time adaptive control, neurofuzzy PID control, hierarchical control, slave-side control, intelligent controls, neural networks, just-in-time control, Bayesian probability, equilibrium-point control, fuzzy logic control, machine learning, evolutionary computation and genetic algorithms, nonlinear optimal predictive control, optimal control, stochastic control, variable structure control, chattering-free robust variable structure control, energy shaping control, gain scheduling model-based controller, sliding mode control, proxy sliding mode control, neuro-fuzzy/genetic control. Static and dynamic modelling of PMA's is well described in [7], [8], [9], [10] and [11].

Pneumatic muscle actuator can be used for totally different fields: lifting aid, tab punching, robotic prosthesis, wearable exoskeleton robot or jumping robot (Fig. 4).

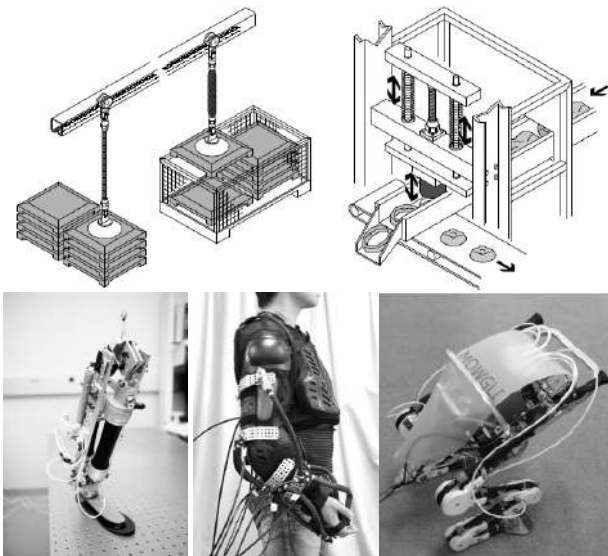


Fig.4. Applications of PMA's [4], [12], [13], [14]

This paper is organized in 4 sections. After Introduction, in Section 2 the applied experimental rig and the experimental measurements are described. The experimental results and the comparisons of Fluidic Muscles are presented in Section 3. The paper ends with Conclusions (Section 4).

2. EXPERIMENTAL RIG FOR MEASURING PRESSURE, FORCE AND POSITION

For investigation of key features of PMA's, force, pressure and position (displacement) must be measured by an experimental rig (Fig. 5) that is well described in [15]. It illustrates the Fluidic Muscle is built horizontally into the

test bed. One of ends of PMA is fixed while the other is movable. The free end of PMA is connected with a spindle which is connected with an accurate position measuring instrument. The rig consists of the following sensors and regulator:

- proportional pressure regulator (PPR) (VPPM-6L-L-1-G1/8-0L6H-V1N-S1C1),
- incremental encoder (LINIMIK MSA 320),
- load cell (Kaliber 7923),
- pressure sensor (Motorola MPX5999D).

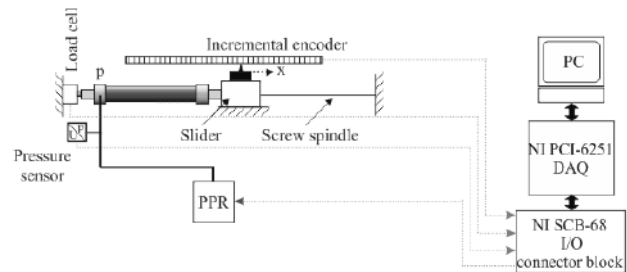
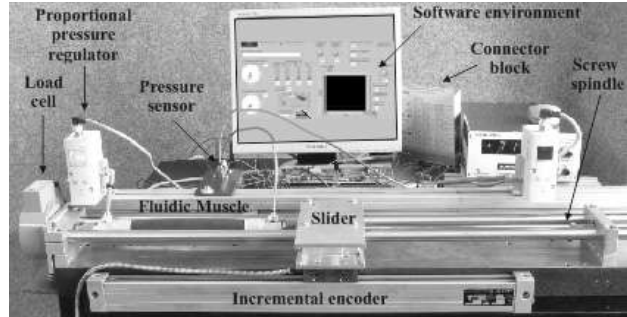


Fig.5. Experimental rig for measuring pressure, force and position

With the help of it, four Fluidic Muscles are: DMSP-10-250N-RM-RM, DMSP-20-200N-RM-RM, DMSP-20-400N-RM-RM and DMSP-40-200N-RM-RM, where DMSP means pressed end caps and integrated air connectors, first number defines the inside diameter in mm, second number defines the nominal length in mm and RM means radial pneumatic connection at both ends. Based on values in 30 points of PMA's measured for various compressions the force-contraction curve is plotted. Measuring at every point is repeated for 5 times and statistically averaged. For acquisition and monitoring of measured data (pressure, force, and position of slider) a LabVIEW program is developed. In Fig. 6, the front panel of the LabVIEW program is shown.

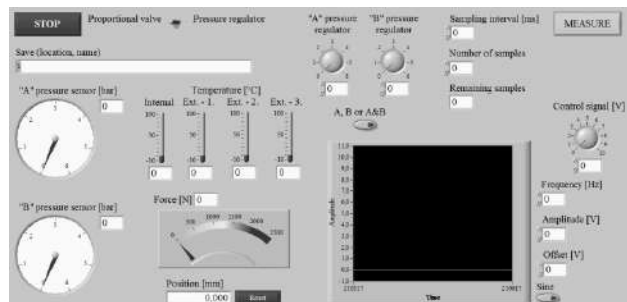


Fig.6. Front panel of the LabVIEW program

3. COMPARISON OF STATIC FORCE DEVELOPED BY FLUIDIC MUSCLES

Firstly, the force-contraction function of Fluidic Muscle type DMSP-10-250N-RM-RM is measured at a temperature of 25 °C. The applied pressure is kept at a constant value (0, 100, 200, 300, 400 and 500 kPa, respectively), while the length of PMA is changed by the screw spindle (see Fig. 5). The pulling force decreases with increasing contraction. At maximal contraction the volume reaches its maximum value and the force drops to zero. It means that contraction has an upper limit at which there is no force developed by PMA. The characteristic is nonlinear. In Fig. 7, the characteristic curves in the interval of 0-500 kPa with increase for 100 kPa are plotted. The maximal force of 996,39 N and the maximal contraction of 20,03% are measured on the pressure of 500 kPa.

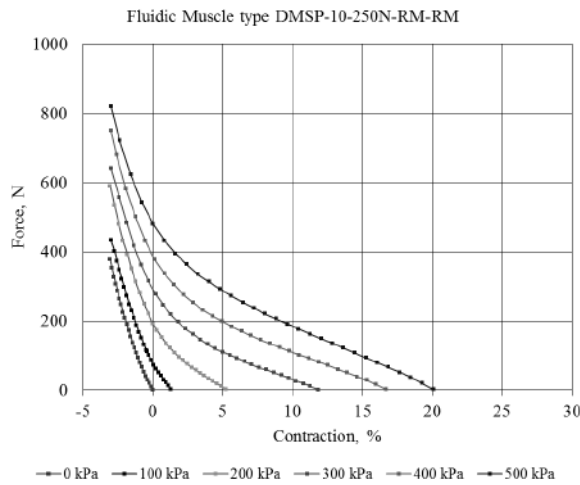


Fig.7. Force as a function of contraction (DMSP-10-250N-RM-RM)

Secondly, the measurement is repeated with Fluidic Muscle types DMSP-20-200N-RM-RM and DMSP-20-400N-RM-RM. Fig. 8 shows the force-contraction function according to [4], while the experimentally determined force-contraction curves are depicted in Fig. 9. In these cases, the maximal force of 2048,32 N and the maximal contraction of 25,88% (200 mm length), furthermore the maximal force of 2060,44 N and the maximal contraction of 27,18% (400 mm length) are measured on the pressure of 500 kPa.

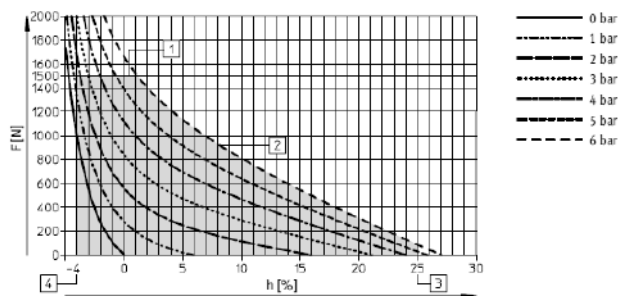


Fig.8. Force-contraction function of DMSP-20 [4]
1-max. force, 2-max. pressure, 3-max. contraction, 4-max. pretensioning

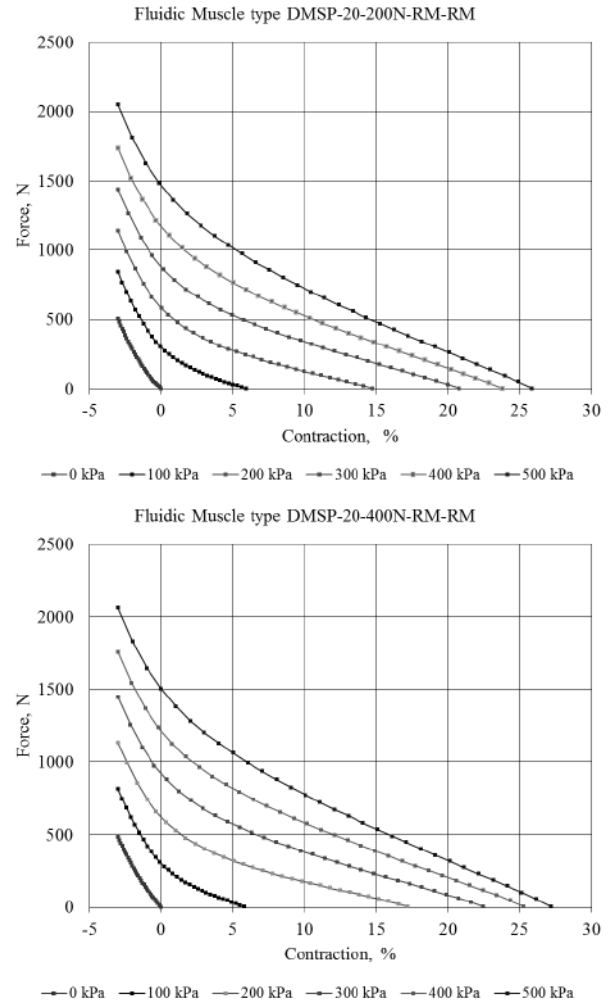


Fig.9. Force as a function of contraction (DMSP-20-200N-RM-RM and DMSP-20-400N-RM-RM)

Comparison of Fluidic Muscles having the same diameter but different length is presented in Fig. 10.

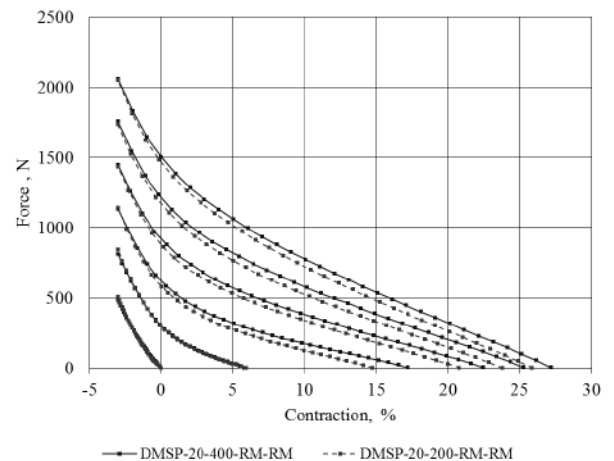


Fig.10. Comparison of Fluidic Muscles having the same diameter

Finally, DMSP-40-200N-RM-RM is investigated: the maximal force of 7273,78 N and the maximal contraction of 28,85% are measured on the pressure of 500 kPa (Fig. 11).

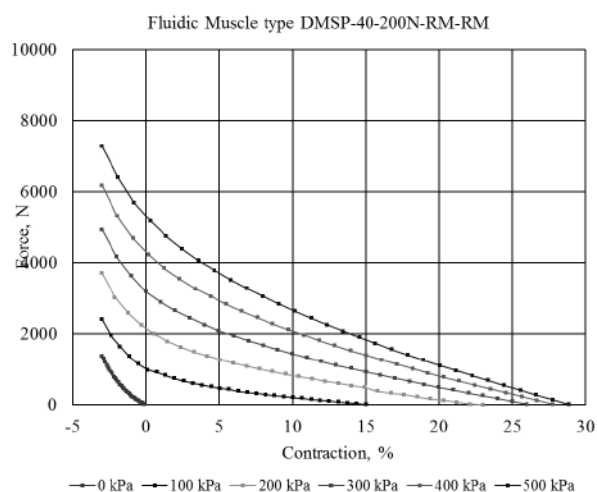


Fig.11. Force as a function of contraction (DMSP-40-200N-RM-RM)

4. CONCLUSION

Based on some investigations the following is concluded: An important difference between pneumatic cylinders and PMAs can be noticed: while the force at cylinders is only a function of pressure and piston size meaning that at constant pressure the force does not change with position in the case of PMAs the force is influenced by displacement. The force produced by PMAs increases with increasing inside diameter and the force depends not only on inside diameter but also length.-

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LUBRICATION REGIME INFLUENCE ON COLD STAMPING PARTS

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Tiberiu Ștefan MĂNESCU

Abstract: *The paper aims to determine the optimal solution for the lubrication elements and guide for stamping in order to obtain consistent parts in a large series regime*

Key words: *stamping, lubrication, guide elements, certified parts, experimental result*

1. INTRODUCTION

During manufacturing, projecting and production in fact of stamped/pressed parts, the primordial role to follow the dimensions, tolerances of the parts is of the active parts correlated with a good guiding of the assembly on presses. In order to obtain parts acc.to heavy loads in the automotive industry is mandatory to keep during fabrication of, the initial active parts dimensions, respective the punch-active plate assembly.

The compliance with initial quota is influenced by more factors. First is the one of execution which refers to compliance with the quality of the prescribed material, in our case C120, then compliance in all respects with the

technological process using 3D machine tools and CNC ones, checkout on 3D measuring devices.

Special attention should be paid to the intermediate heat treatment in order to get a performant assembly having cutting edges worked in quenching + tempering status on 3D machine tools, obtaining so a necessary tolerance range which assures and keeps a hardness of at least 50-55 HRC in the cutting zone.

2. STUDY PURPOSE

During the experiments in normal work conditions as 3 shifts x 8 hours x 5 days a week we found as follows:

1. Lubrication system establishment of columns and guide bushings in use, as:
 - Normal lubrication
 - Reduced lubrication
 - Forced lubrication
 2. Tracking records on shifts of a part „Gusset” in compliance conditions
 3. Choose the lot of the parts to be tracked - 25000 parts
- Establishment of the optimal solution to get conform part

3. MEASURING PROGRAM

For the part „Gusset” it was done on 3D Derby-Etalon machine.

4. EXPERIMENTAL RESULTS

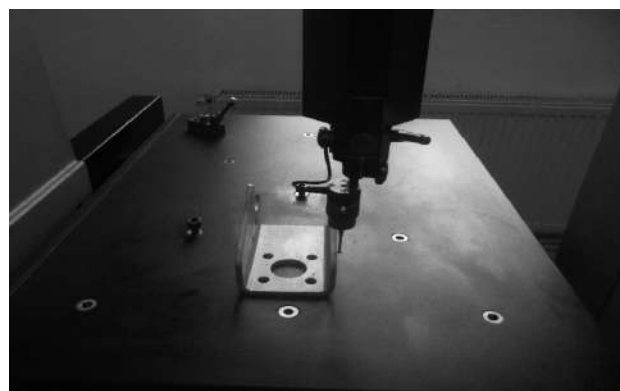


Fig.1 Gusset 922-923

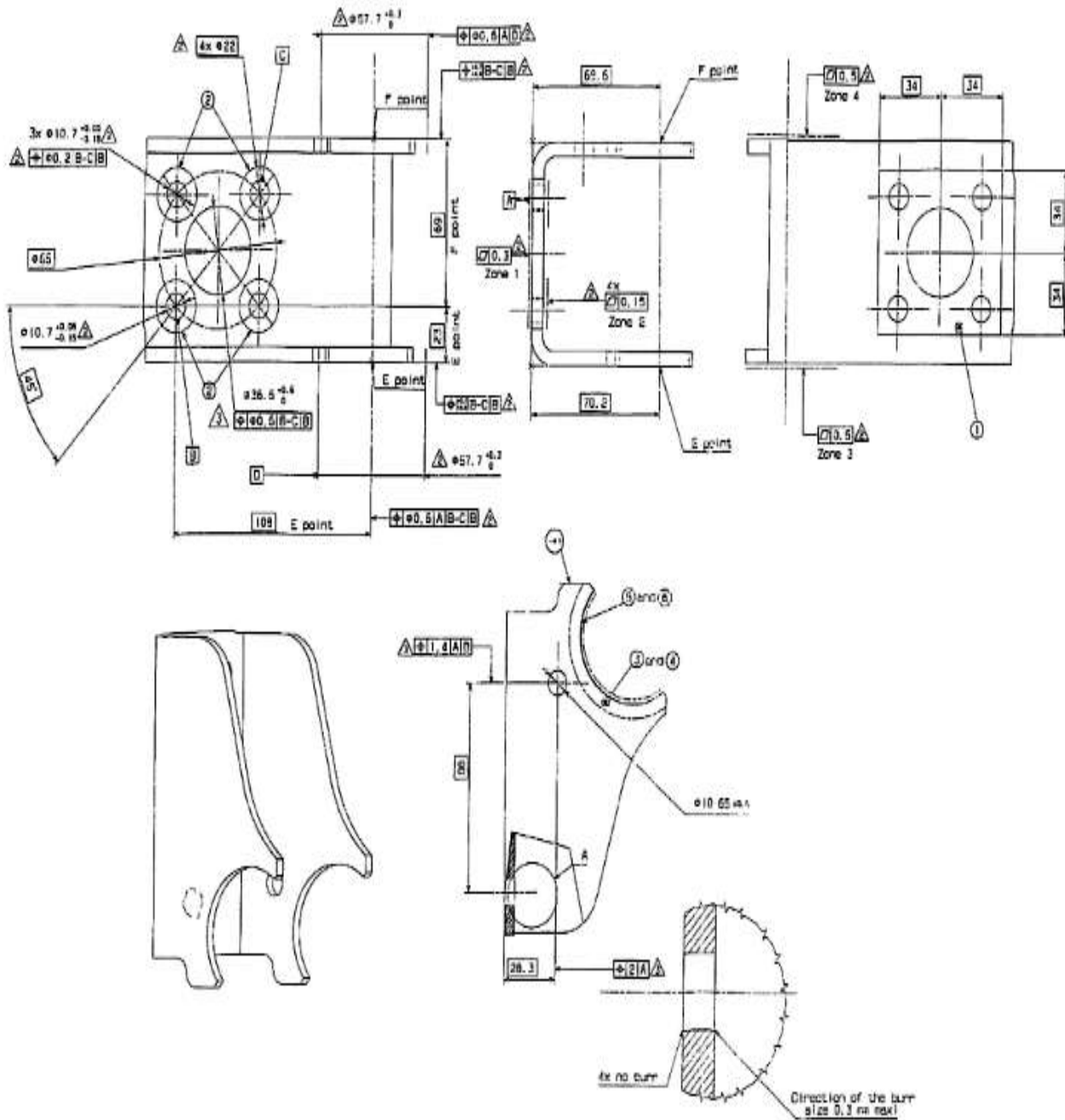


Fig.2 Gusset 922-923

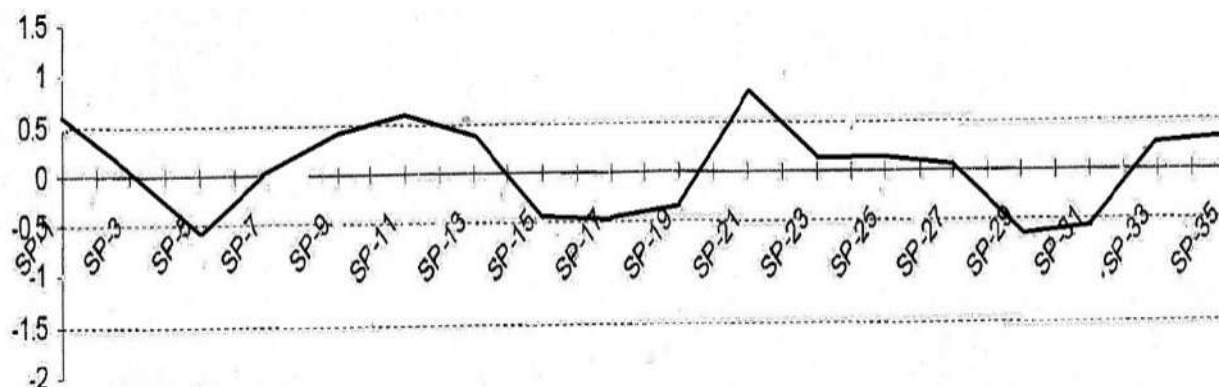
Table 1. Measuring report side openings volume in a normal lubrication

Name	X	Y	Z	dX	dY	dZ	DL
SP-1	2367.589	-657.747	2.205	-0.762	0.005	-0.170	-0.780
SP-2	2364.526	-679.509	15.358	-0.046	0.000	-0.010	-0.047
SP-3	2361.830	-657.033	28.022	-0.816	0.005	-0.182	-0.836
SP-4	2360.318	-678.882	34.226	-0.110	0.001	-0.025	-0.713
SP-5	2357.777	-638.987	49.628	-0.731	0.008	-0.312	-0.765
SP-6	2355.669	-674.900	50.674	-1.418	0.002	-0.093	-1.428
SP-7	2359.206	-611.181	47.024	-0.703	0.012	-0.458	-0.750
SP-8	2350.154	-639.626	86.235	-0.740	0.011	-0.437	-0.789
SP-9	2348.617	-674.868	96.746	-0.692	0.004	-0.154	-0.709
SP-10	2353.411	-654.313	68.661	-0.617	0.007	-0.280	-0.648
SP-11	2344.496	-648.034	113.444	-0.383	0.012	-0.461	-0.434
SP-12	2342.577	-674.144	113.822	-0.823	0.005	-0.184	-0.843
SP-13	2339.786	-653.538	132.467	-0.781	0.012	-0.454	-0.831
SP-14	2338.559	-675.097	131.792	-0.844	0.005	-0.188	-1.864
SP-15	2446.110	-655.682	17.804	0.844	-0.006	0.224	-0.828
SP-16	2443.574	-679.173	29.803	0.699	-0.006	0.223	-0.723
SP-17	2439.652	-657.498	48.833	0.582	-0.006	0.242	-0.608
SP-18	2439.273	-677.194	49.120	0.733	-0.006	0.231	-0.759
SP-19	2436.923	-614.109	63.988	0.599	-0.009	0.335	-0.636
SP-20	2436.591	-648.699	64.119	0.460	-0.007	0.281	-0.491
SP-21	2436.415	-677.717	62.804	0.577	-0.006	0.240	-0.604
SP-22	2432.249	-656.613	83.361	0.821	-0.010	0.362	-0.861
SP-23	2427.859	-641.957	105.203	0.745	-0.011	0.434	-0.793
SP-24	2427.192	-675.776	105.516	0.500	-0.008	0.290	-0.532
SP-25	2420.339	-648.293	142.308	0.388	-0.014	0.533	-0.446
SP-26	2418.314	-675.765	146.182	0.459	-0.009	0.326	-0.495
SP-27	2423.042	-668.296	125.199	0.668	-0.010	0.372	-0.709

Number of points	27
Out of Tolerance	2
Performance	93%
Mean	-0.733
Std.deviation	0.380
Max.Value	-0.047
Min.Value	-1.864

Table 2. Measuring report flat surface volume in a forced lubrication

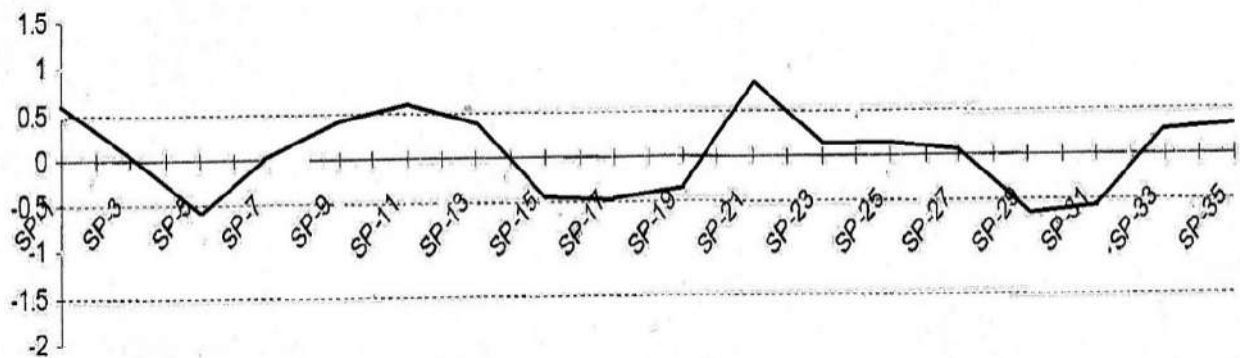
Name	X	Y	Z	dX	dY	dZ	DL
SP-19	2450.687	-650.210	15.170	-0.062	-0.269	0.271	-0.387
SP-20	2450.814	-659.200	14.293	0.063	-0.004	-0.280	0.289
SP-21	2432.114	-693.288	22.949	0.164	-0.012	-0.737	0.755
SP-22	2432.480	-693.222	11.873	0.216	-0.015	-0.967	0.991
SP-23	2365.968	-656.523	-4.682	0.039	-0.003	-0.175	0.189
SP-24	2365.665	-649.250	-3.046	0.074	0.321	-0.322	0.400
SP-25	2364.142	-652.571	3.882	0.017	-0.154	-0.074	0.172
SP-26	2361.061	-655.506	17.859	-0.006	-0.549	0.029	0.550
SP-27	2356.567	-650.408	32.108	-0.113	-0.747	0.506	0.009
SP-28	2353.240	-632.441	43.492	-0.181	-0.169	0.811	0.845
SP-29	2353.996	-612.858	40.630	-0.123	0.309	0.551	-0.645
SP-30	2440.271	-636.587	61.715	-0.148	-0.248	0.664	-0.724
SP-31	2440.919	-614.591	60.992	-0.124	0.262	0.554	-0.625
SP-32	2449.292	-669.078	19.699	-0.001	0.081	0.006	-0.081
SP-33	2447.754	-681.404	26.254	0.074	-0.005	-0.332	0.340
SP-34	2364.290	-668.590	0.854	-0.005	0.797	0.043	-0.798
SP-35	2362.794	-679.756	7.269	0.083	-0.006	-0.370	0.374



Number of points	35
Out of Tolerance	0
Performance	100%
Mean	-0.220
Std.deviation	0.507
Max.Value	0.991
Min.Value	-0.798

Table 3. Measuring report flat surface volume in a normal lubrication

Name	X	Y	Z	dX	dY	dZ	DL
SP-19	2450.687	-650.210	15.170	-0.062	-0.269	0.271	-1.387
SP-20	2450.814	-659.200	14.293	0.063	-0.004	-0.280	0.289
SP-21	2432.114	-693.288	22.949	0.164	-0.012	-0.737	0.755
SP-22	2432.480	-693.222	11.873	0.216	-0.015	-0.967	0.991
SP-23	2365.968	-656.523	-4.682	0.039	-0.003	-0.175	0.189
SP-24	2365.665	-649.250	-3.046	0.074	0.321	-0.322	0.400
SP-25	2364.142	-652.571	3.882	0.017	-0.154	-0.074	-0.172
SP-26	2361.061	-655.506	17.859	-0.006	-0.549	0.029	-0.550
SP-27	2356.567	-650.408	32.108	-0.113	-0.747	0.506	0.009
SP-28	2353.240	-632.441	43.492	-0.181	-0.169	0.811	0.845
SP-29	2353.996	-612.858	40.630	-0.123	0.309	0.551	-1.645
SP-30	2440.271	-636.587	61.715	-0.148	-0.248	0.664	-1.724
SP-31	2440.919	-614.591	60.992	-0.124	0.262	0.554	-1.625
SP-32	2449.292	-669.078	19.699	-0.001	0.081	0.006	0.081
SP-33	2447.754	-681.404	26.254	0.074	-0.005	-0.332	0.340
SP-34	2364.290	-668.590	0.854	-0.005	0.797	0.043	0.798
SP-35	2362.794	-679.756	7.269	0.083	-0.006	-0.370	0.374



Number of points	35
Out of Tolerance	10
Performance	71%
Mean	-0.510
Std.deviation	0.471
Max.Value	0.991
Min.Value	-1.798

5. EXPERIMENT CONCLUSIONS

The test lot of 25 000 parts showed the followings:

a) Guiding elements (bushings + columns) in a normal lubrication, conform parts 93% acc.to table 1.1;

b) Guiding elements (bushings + columns) in a forced lubrication, conform parts 100% acc.to table 1.2;

c) Guiding elements (bushings + columns) in a reduced lubrication , conform parts 71% acc.to table 1.3.

6. RECOMMENDATIONS

After the comparative checkout „part” - control model , cold cutting and perforating die got the working permission for large series production as it was asked for. To obtain a superior conformity percentage it is recommended to use the forced lubrication for serial production up to 50000 pcs/series without the re-sharpening of the cutting elements.

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APPLICATION OF RAPID PROTOTYPING IN MAXILLOFACIAL SURGERY

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Abstract: Nowadays, despite very high precision of medical procedures and equipment, improving of pre-operative and operative processes is always desirable. This multidisciplinary paper deals with the application of 3D modelling and printing in the pre-operational planning stages in maxillofacial surgery. By applying of these methods significant benefits were achieved before and during the operation, the time patient spent in anaesthesia is shortened, and the potential for error is significantly reduced, which is the main goal of all trends in modern medical science.

Key words: rapid prototyping, 3D printing, modeling, maxillofacial surgery

1. INTRODUCTION

Taking into account most recent discoveries, 21st century medicine, with all related fields, is definitely advancing faster than in 20th century. Significant results have been achieved in the field of internal medicine, orthopaedics, oral surgery, prosthodontics, surgery, etc. Maxillofacial surgery is definitely not left behind. One of the most significant developments arises from the possibility of replacing the damaged bone of the maxillofacial region with another bone from different region or some biocompatible implant. In order to achieve high accuracy of these methods it was necessary to develop a sufficiently precise algorithms, apparatus and technology. Improvement of modern hardware and software for human body parts scanning and the compatibility of this software with the modelling and 3D printing software enabled the connection between mechanical engineers and maxillofacial surgeons. Multidisciplinary approach in this case is of great importance because it encourages the

development of new research in each of the mentioned scientific fields. The main contribution of presented research was reflected in the preparation of plastic prototypes of damaged bones in the maxillofacial region as well as healthy bones necessary for the replacement of the foregoing. It was latter shown that the existence of these prototypes significantly contributed to the advancement of the process of implantation and reconstruction of facial bones.

2. IMPLEMENTATION OF THE PROJECT

The idea to initiate this project occurred as a result of long-lasting cooperation between the Faculty of Dental Medicine and Faculty of Mechanical Engineering at University of Belgrade. After a few constructive sessions the plan of the project was developed, and latter implemented in several stages, as shown in following chapters – through several successful cases.

2.1. Obtaining of X-ray scans

The first step in the realization of the project was to obtain an X-ray scan of the patient. After a detailed analysis of the scan and consultations with maxillofacial surgeons (who provided the scan), DICOM file was imported into the standardized software, used to generate appropriate model. A scan set, required for the further preparation, is shown in Figure 1 (red circle indicates the position of the damage hole on the lower jaw due to the effects of aggressive tumor).

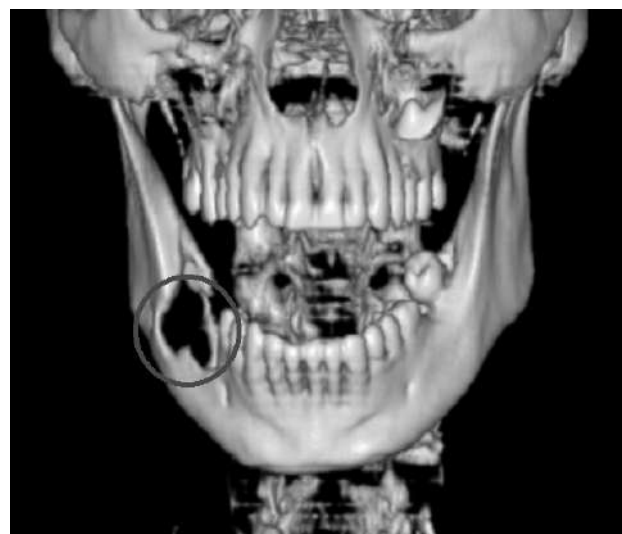


Fig.1. Preview of a head scan with damaged lower jaw bone

2.2. Scan – based prototyping

First step of the project implementation was followed by the drafting of the head model and extracting of an area of particular interest (the area of damage). During the model creation phase, the above-mentioned software provided the ability to isolate bones, skin, tissues, muscles, etc. – and as a result a shell model was obtained. The quality of scanner has a huge impact on the quality of the model. The modern multislice scanners operate on the principle of cross-sections scanning, at a relatively small distance.

When a series of scanned layers, obtained in this manner, are sequenced, the 3D image of the observed object is generated. The basic principle of this kind of scanning method is shown on Figure 2.

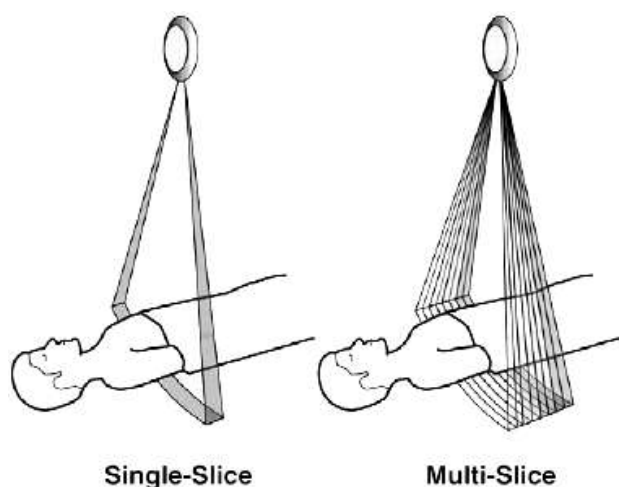


Fig.2. The difference between Single – slice and Multi – slice scanning methods

When the high resolution scanners are used, the distance between two adjacent scanned slices is very small (for example 0,5mm). This can be a problem because, due to the high resolution, complete 3D model is generated from an extremely large number of slices. For processing of such models, a computer with exceptional capacities is needed. The opposite extremes are scanners with very low resolution which, due to the large distance between the two slices, can cause the loss of important information which are located between these slices. The Figure 3 shows a model obtained from a head scan when only bones are isolated. The damage of the lower jaw can be clearly seen.

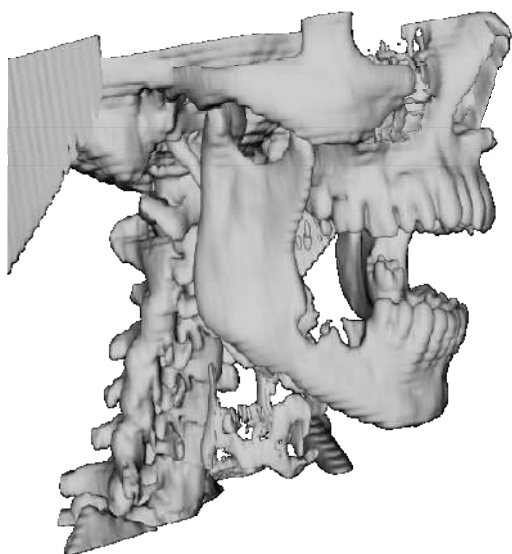


Fig.3. Shell model of the damaged jaw bone

2.3. Making of the printable 3D model

Since the standardized 3D recognition software enables only the extraction of the shell model (surface model with

infinitely thin external walls) in order to obtain printable model further processing is necessary. The main reason for this is the fact that rapid prototyping software does not recognize the surface models and therefore printing phase can't be accessed. At the moment, there are several methods for the conversion of a shell model into a finite volume model. It was decided that this action should be accomplished by adding of a certain amount of material, on the inside surface of the outer wall (shell). Considering that the material is added only to the inside surface of the wall, external parties, especially geometry of damaged bone, remains unchanged. This whole procedure is performed in other standardized software, and a model, after the corrections, has the appearance shown in Figure 4.

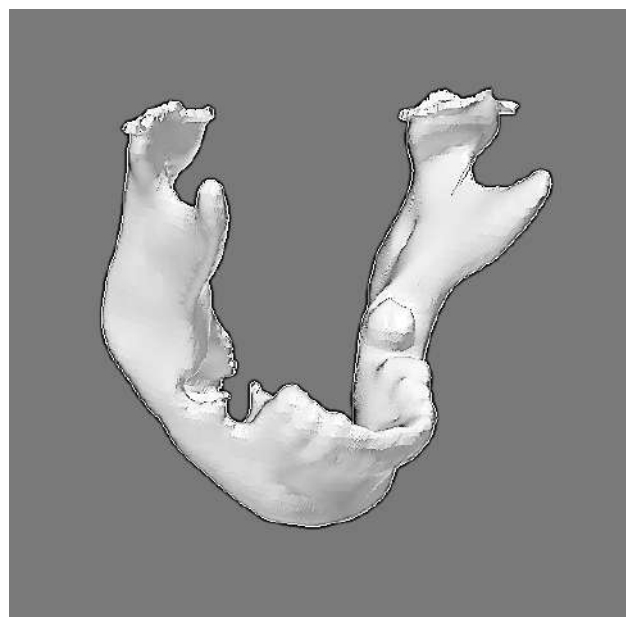


Fig.4. The appearance of volumetric model of damaged jaw bone

2.4. Additional adjusting of the printable 3D model

The final processing of obtained 3D model is performed in the third standardized software, because it has a number of very useful options, used for model simplification – significantly affecting its printing performances. Also, this program serves to remove unnecessary parts of the model and to isolate the desired parts of the scanned human body. In the presented case, the lower jaw bone was successfully extracted from the rest of the head. For that purpose, several commands were used, such as *smoothing* using the Laplacian smooth tool, or *removal of small isolated areas* of the model. These small particles of the model are either randomly generated, or are really part of the model, but they are so small in size that their elimination does not affect the quality of the final model, while on the other hand, it has a significant impact on 3D print facilitating. Work in an environment of mentioned standardized software is shown in Figure 5.

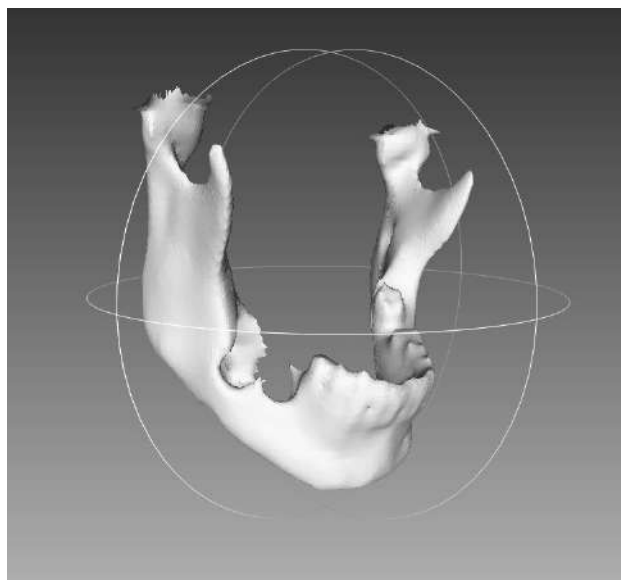


Fig.5. Finishing touches on a printable 3D model

3. 3D PRINTING OF MODEL

After the completion of all previously listed actions, generated printable 3D model was sent to the 3D printer. Printing process was very complicated because it requires the definition of the huge number of parameters whose values can't be found in guide books, so they have the exclusive experimental character. Taking this into account, every model is a story in itself, and as such requires the definition of its own printing parameters. 3D printing works on the principle of bringing the plastic thread to the extruder where it is melted down and delivered to the printing platform. The printed prototype, as well as scan, also consists of layers of melted plastic, where great care is needed as well as the constant presence of the user, because a mistake made in any printing layer entails generating and superimposing errors in all subsequent layers.

3.1. Definition of 3D printing parameters

Some of the main parameters of the 3D printing are:

Extruder temperature – the temperature of the extruder should be optimal, as it is directly related to the speed of adding of the melted plastic to the printing platform.

The speed of material adding – as already mentioned, it strictly depends on the temperature of the extruder, which melts the plastic. The combination of these two parameters must ensure adequate melting and bonding of plastic interlayer. Excessive rate of feed leads to the pouring of plastic in undesirable places. On the other hand, insufficient rate of feed leads to porosity of 3D printed model. In addition to these, another very important parameter is the temperature of plate.

Temperature of plate (printing platform) – it must ensure that there is no contact break between the model and the plate on which the model is set. The plate and printing head gradually depart during printing. This translation allows the formation of a model from a large number of layers. When a printing layer is completed, the printing head is elevated for a specific distance (depending on

desired layer thickness), and then the procedure continues. Regarding to the parameters influencing the structure of the model, the major are:

Layer thickness – this parameter is very important because it directly affects the surface quality of the model. It also indicates whether a particular piece of the model will be completed in one, two or more passes of extruder – thus affecting the overall printing time. On the other hand, a situation when the thickness of the model is smaller than defined thickness of molten plastic layer must be avoided, because, in that case, the degradation of the model geometry occurs.

Infill percent – the structural integrity and overall printing time of the specific 3D model are directly dependent of this factor, taking into account that it defines the percentage of material inside the printed model volume.

The position of the model in the printing area is also of great importance. An unfavorable position of model leads to a situation where software has to generate artificial support of the printed model (so-called supports), thereby unnecessarily increasing the printing time and risk of printing failure. Positioning of the damaged lower jaw bone 3D model in printing area is shown in the figure below (Fig. 6).

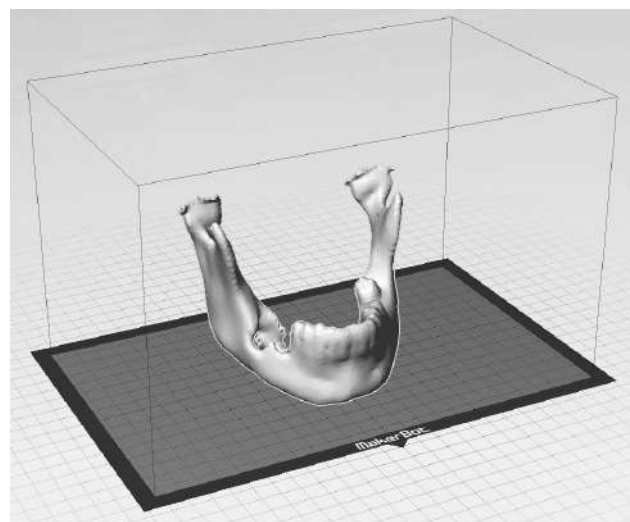


Fig.6. Positioning of the damaged lower jaw bone 3D printing model on printing platform

3.2. Printing material

As a material for 3D printing, the industrial-strength Acrylonitrile Butadiene Styrene (ABS) plastic was used. Its detailed mechanical and physical characteristics are given in the Table 1. ABS is derived from acrylonitrile, butadiene, and styrene. Acrylonitrile is a synthetic monomer produced from propylene and ammonia; butadiene is a petroleum hydrocarbon obtained from the C4 fraction of steam cracking; styrene monomer is made by dehydrogenation of ethyl – a hydrocarbon obtained in the reaction of ethylene and benzene.

ABS plastic combines the strength and rigidity of acrylonitrile and styrene polymers with the toughness of polybutadiene rubber. While the cost of ABS producing is roughly twice the cost of producing polystyrene, it is considered superior for its hardness, gloss, toughness, and

electrical insulation properties. Also, ABS plastic is easily machined (with common machining techniques, including turning, drilling, milling, sawing, die-cutting and shearing) – making it the most common plastic for industrial usage worldwide. ABS plastic can be also cut with standard shop tools and line bent with standard heat strips.

Table 1. Physical and mechanical properties of ABS plastic

ABS plastic		
Physical properties	Density	0,9-1,53 [g/cm ³]
	Thermal conductivity	0,1 [W/mK]
	Linear thermal expansion coefficient	73,8·10 ⁻⁶ [m/mK]
	Temperature range of usefulness	-20-80 [°C]
Mechanical properties	Tensile strength	22 [MPa]
	Tensile modulus	1360 [Mpa]
	Tensile elongation	6 [%]

Regarding to the displayed physical and mechanical properties of ABS plastic, it is clear why this material is most often chosen for 3D printing and subsequent mechanical processing.

4. USAGE OF 3D PRINTED PROTOTYPE IN A PRE-OPERATIONAL ACTIVITIES

When a prototype of damaged jaw bone was finally manufactured, it was mechanically trimmed, getting final shape. The model was then sent to the maxillofacial surgeons. In this particular case, damaged part of the lower jaw bone was replaced by the upper part of the hip bone. The upper part of the hip bone was chosen in this application due to the high similarity with geometry of the lower jaw (so-called U-shape), as shown on the Figure 7. The procedure of hip bone prototyping is identical as for jaw bone and it consists of the same steps – described in previous chapters. This procedure is graphically represented on Figures 8-11.



Fig.7. Matching of the geometries of the 3D printed upper part of the hip and lower jaw



Fig.8. Preview of the upper part of hip bone scan which should be used as a implant



Fig.9. Shell model of hip bone

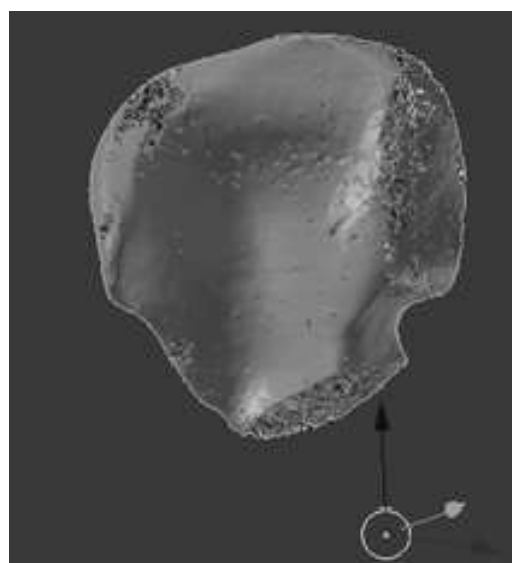


Fig.10. The appearance of volumetric model of hip bone

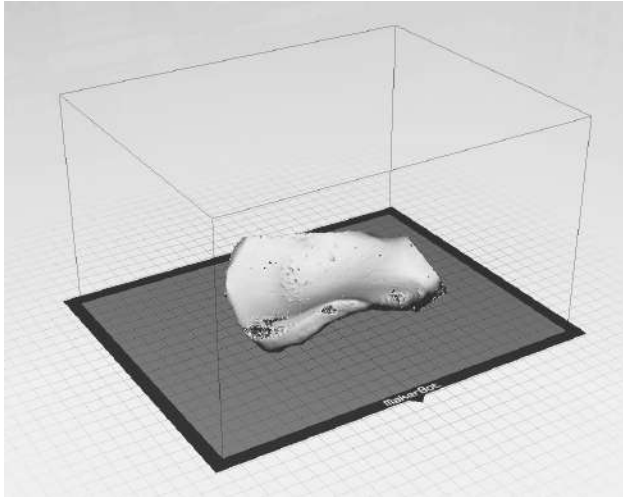


Fig.11. Positioning of the hip bone printing model in the printing area

After rapid prototyping of both presented models, surgeons had several benefits. They were able to determine exactly how foreseen healthy bone can replace the damaged one, use the model to rehearse interventions necessary to remove the damaged bone as well as those for separation of the healthy bone (implant), or determine the exact amount of material for the implementation of the operational process, etc.

For example, a small mismatch between the two shapes (red circle) can be seen on the picture shown below (Fig. 12) – without prototypes, this mismatch could be seen only after cutting of the patient. As it was observed much earlier (before the actual operation), appropriate measures could be taken pre-operatively.

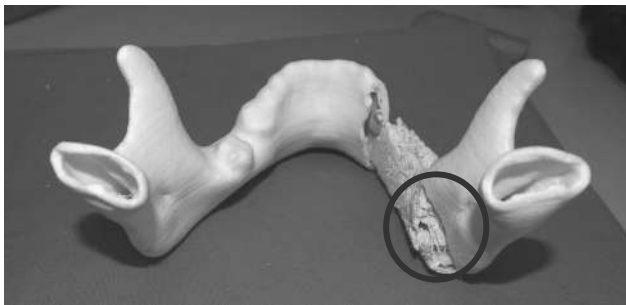


Fig.12. Spotting of the mismatch between part of hip bone (used as an implant) and damaged lower jaw bone

After the removal of the identified defects, the subsequent positioning of implant was made, as well as the shaping of the surgical guides, necessary for the fixation of the implant during and after surgery. The result of these procedures is shown on Figure 13.

The successful completion of described activities (the pre-operative process), caused the surgeons to be fully prepared for performing of successful actual operation.



Fig.13. Adaptation of the surgical guides and a matching of the printed models

5. CONCLUSION

After the completion of described multidisciplinary project, very positive feedback was obtained by the surgeons of the maxillofacial surgery. They were fully satisfied with the obtained prototypes and their fit with the real objects (prototyped human body parts). An application of this, relatively simple, technology has achieved significant effects:

- The time patient spent in the state of general anesthesia is significantly reduced because the number of operations which would have been carried out is done pre-operative.
- The precision during the execution of the operational process was significantly increased.
- Surgeons were familiarized with geometry of the damage long before the opening of the patient.
- Possible mismatches between the implant (obtained from hip bone) and the bone that is receiving the implant (lower jaw bone) can be detected and thus, the occurrence of complications during surgery was prevented pre-operative.

When all of the above is taken into consideration, it could be concluded that, by the usage of described methodology, multiple influences on the improvement of surgical procedures in the field of maxillofacial surgery could be achieved, among them – the most important: humanistic contributions and benefits of the patients.

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CAVITATION EROSION BEHAVIOR OF THE STEEL 17CrNiMo6

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Abstract: In hydraulic drive systems, cavitation erosion affects especially the driving and distribution elements. In most cases the working fluid is oil for which the cavitation erosion potential is smaller than for water. On the other hand, the erosion affects exactly the areas destined to stop the leakage and as a consequence the reduction of erosion is an important condition. The majority of the mobile elements being manufactured from weakly alloyed steels it is important to use suitable heat treatments such as carburizing or nitriding for increasing the hardness of the areas exposed to erosion. The article presents the cavitation erosion resistance researches realized with the vibratory device of the Timisoara Polytechnic University Cavitation Laboratory upon specimens of 17CrNiMo6 steel in two conditions annealed and carburized in a gaseous environment followed by quenching as well as tempering at low temperature. The cavitation erosion resistance of the two states for the analyzed steel was compared with the steel 41Cr4 considered with good cavitation resistance. The eroded area was also examined with performing microscopes. The comparisons show that the carburizing and the applied heat treatments enhance significantly the steel behavior to erosion

Key words: alloyed steel, carburizing, microstructure, mean depth cavitation erosion, mean depth erosion rate

1. INTRODUCTION

In hydraulic drive systems (especially in directional control elements, valves, etc.) the cavitation appear in the passing slots between the mobile parts and the devices body, see Figure 1 and 2. The cavitation intensity is determined by the oil velocity during the passage through the slots, when the pressure can decrease significantly

under the vapor pressure. In Figure 3 is presented the oil flow through a directional spool valve [5]. It can be seen that in the passing slot the pressure became negative and this fact determines the apparition of the cavitation phenomenon, with all their negative effects. The principal motive is the very small dimensions of the slots (the magnitude order is tenth of millimeters) which determine the increase of velocity and the subsequent reduction of the pressure.

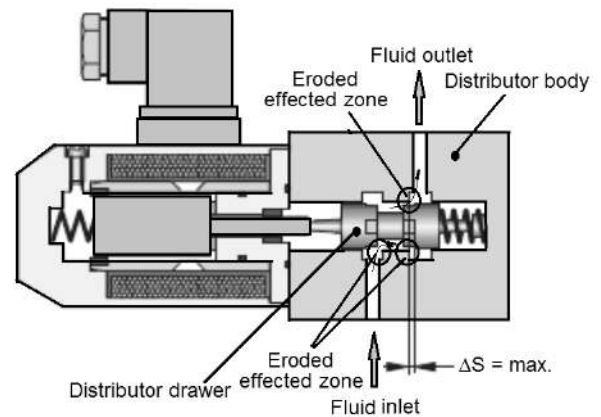


Fig.1. Cavitation eroded areas at a distributor apparatus

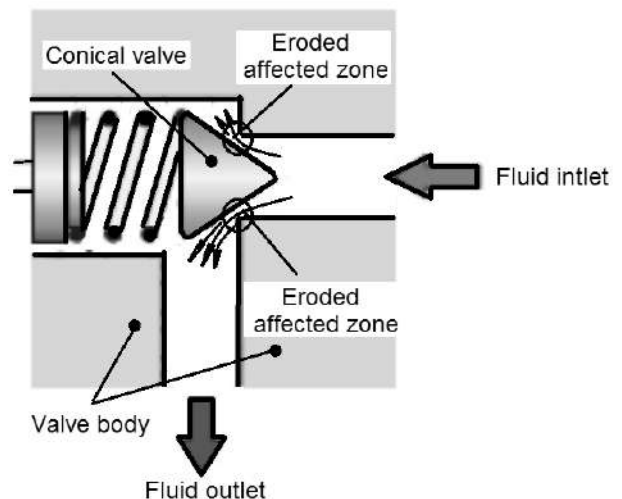


Fig.2. Cavitation eroded areas for conical valves

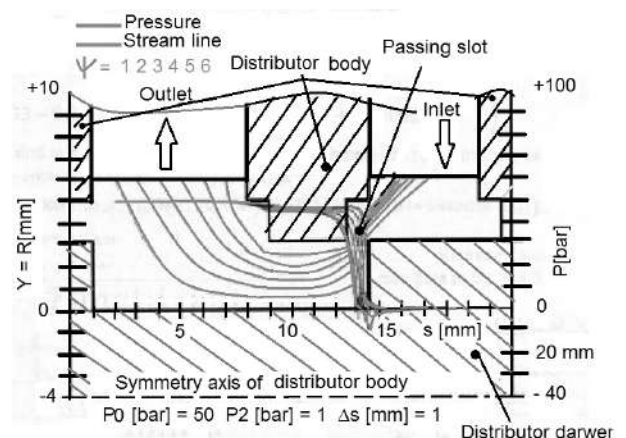


Fig.3. Stream lines and pressure evolution along them, for $x = 1$ mm, (directional control valve)[5]

Because the most noxious effect of cavitation is the erosion of the solid boundaries limiting the flow [2], [3], [8], [9] and the used materials are the special cast irons (for the body of the devices [3]) and the weak alloyed steels (for the mobile elements [4], [6], [7]) it appear the necessity to use special manufacturing procedures able to increase the cavitation erosion resistance of the affected zones and as a consequence to extend their life. Such a procedure which can be applied to the mobile elements is the thermo-chemical treatments which increase the hardness of surface subjected to cavitation. In the present paper there are presented investigations upon cavitation erosion resistance of the low alloyed steel 17CrNiMo6. The experimental device used is a vibratory one existent in Cavitation Laboratory of Timisoara Polytechnic University. Both the device and the applied procedures respect the ASTM G32-2010 Standard.

2. RESEARCHED MATERIAL. LABORATORY DEVICE

The researched material is the low alloyed steel 17CrNiMo6. The chemical composition and the mechanical properties are given in Tab. 1 and 2, both those guaranteed by the manufacturer (given in EN 10084 and symbolized 17CrNiMo6^S) and those measured in our Laboratory (symbolized 17CrNiMo6^M). The samples destined for the chemical and mechanical measurements as well as the cavitation erosion specimens were taken from round rolled bars. From Tables 1 and 2 it can be seen that the measured values are between the guaranteed ones. In the same tables are given also the characteristics of the steel 41Cr4 taken as standard material for evaluating the cavitation erosion behavior.

Table 1. Chemical composition

Chemical Elements [%]	17CrNiMo6 ^S	17CrNiMo6 ^M	41Cr4
C	0.15-0.21	0.18	0.445
Si	≤0.40	0.20	0.28
Mn	≤0.025	0.70	0.72
P	0.50-0.90	0.012	0.025
S	≤0.015	0.011	0.025
Cr	1.50-1.80	1.65	0.94
Mo	0.25-0.35	0.3	-
Ni	1.40-1.70	1.55	-
Al	≤0.05	0.032	-
Cu	≤0.30	0.21	0.17
Fe	rest	rest	rest

S-represent the values given by EN 10084-1998; *M*-represent the measured values

Table 2. Mechanical characteristics

Properties	17CrNiMo6 ^S	17CrNiMo6 ^M	41Cr4
R _{p0.2} [MPa]	590-830	805	790
R _m [MPa]	830-1400	985	808
HB [daN/mm ²]	229-233	228	238
A5 [%]	9-10	9.5	11

S-represent the values given by EN 10084-1998; *M*-represent the measured values

For the tested material there were used two types of samples: in annealed state and after thermo-chemical treatments. The thermo-chemical treatment was applied with the purpose to increase the hardness and simultaneously the cavitation erosion resistance. This treatment was applied directly to the cavitation erosion specimens. The shape and dimension of the used specimens are given in Figure 4 which present such a specimen winded in resin and axially sectioned for electronic microscopy. The thermo-chemical treatment consists in carburizing and quenching followed by low tempering. The diagram of cycles is given Figure 5. The thickness of the carburized layer is of approximate 1.2 mm.

In Figure 6 there are given the structures of the steel both in annealed and after thermo-chemical treatment. In Figure 6a it can be seen the structure formed from pearlite (major component) and ferrite. Figure 6b put into evidence the structural transformation as the consequence of the thermo-chemical treatment, with the formation of globular carbides of the alloying elements (Cr, Mn, Mo) which as a result of the high mechanical properties (hardness, tensile and compression strength) enhance the cavitation erosion resistance.

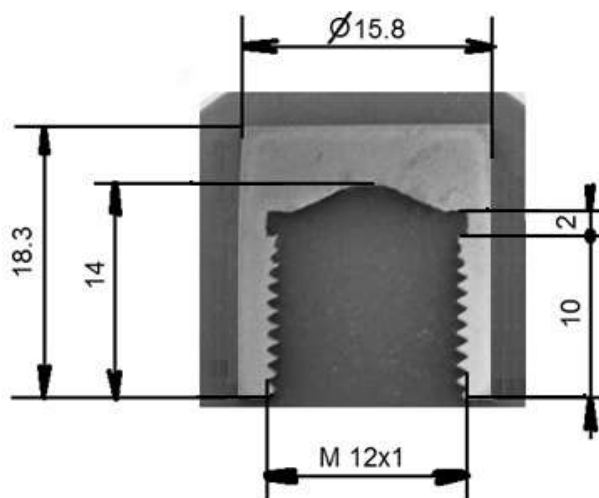


Fig. 4. Shape and dimensions of the specimen used for cavitation erosion tests (dimensions given in mm)

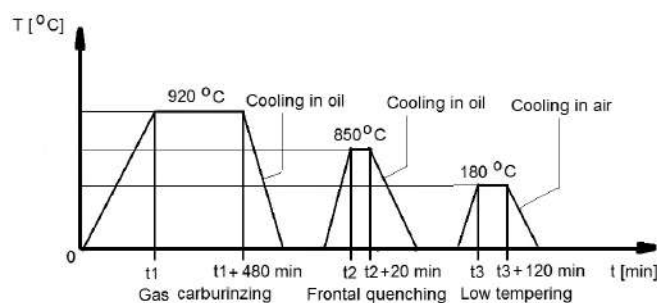
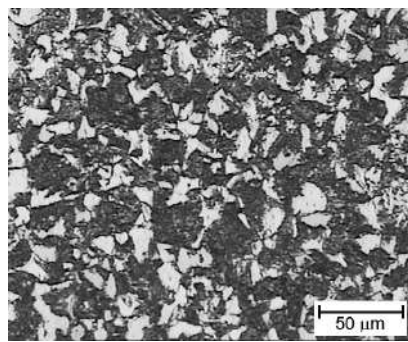
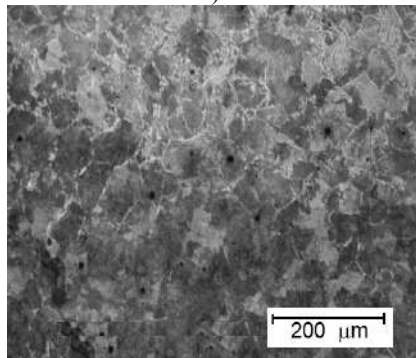


Fig. 5. The diagram of the thermo-chemical treatment



a)



b)

Fig. 6. Steel microstructure (x 200)

a) annealed (pearlite + ferrite)

b) after carburizing, quenching and tempering (cubic martensite formation after tempering)

In Figure 7 is given the variation of micro-hardness in the cross section of the thermo-chemical treated specimens in conformity with the measurement scheme presented in Figure 8. It can be seen that the chromium presence, as principal alloying element, together with the carbon enriching of this layer determine a substantial hardness increase, till approximate 59 HRC. The reduction with 0.8 mm of the carburized layer as the result of the obligatory grinding does not exceed 2 HRC and remain greater than the core hardness (approximate 48 HRC). This fact is beneficial for erosion resistance because all the previous researches ([3], [8], [9], [10]) show that hardness is extremely important parameter for increasing the behavior of the material subjected to cavitation. Each point in the Figure 7 is the mean of three hardness measured values at the given depth point. For testing the hardness distribution of the exposed area the measurements were made on three diameters taking four points on each one. The final conclusion was that the hardness has negligible variations on these points.

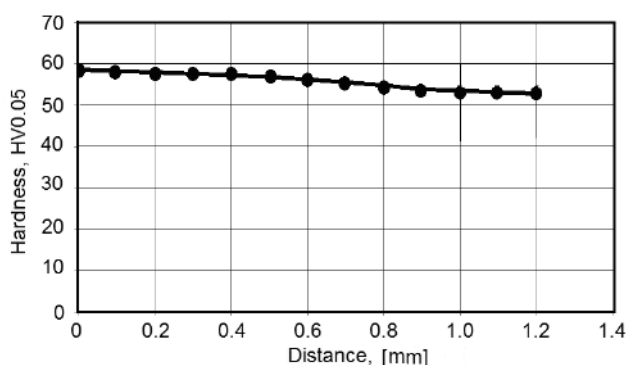


Fig. 7. Variation of hardness in the cross section of the specimen

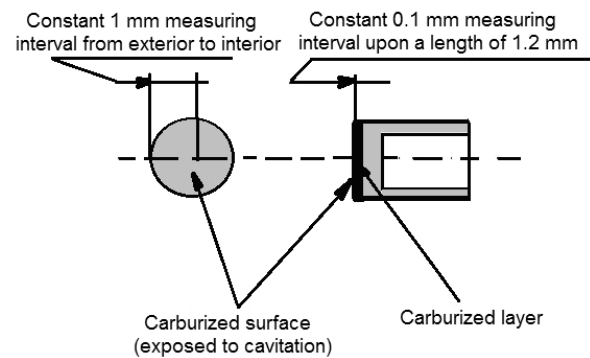


Fig. 8. Details regarding the hardness measurements of the carburized layer

The cavitation erosion research program was realized in Timisoara Polytechnic University Laboratory for Cavitation on a device with piezoelectric crystals [6], [7], [9]. Because the ASTM G32 Standard [11] required that all the cavitation erosions researches to be realized in distilled water, this requirement was fulfilled even if the great majority of the hydraulic installation have oil as working agent. As a consequence all tests were carried out in distilled water at $22 \pm 1^\circ\text{C}$ and so the results can be compared with all other materials tested for cavitation erosion, inclusively with the steel 41Cr4. The running parameters of the device (frequency and amplitude of vibrations, the power of the electronic generator of ultrasounds) have been maintained constants during the research [2], [8]. The specimen preparation, the data registration and processing are those required by ASTM G32-2010 Standard.

3. EXPERIMENTAL RESULTS

3.1. Behavior to cavitation erosion

To put into evidence the evolution of cavitation erosion, after each intermediary exposure period [7], [8], [9] the eroded surface was examined and photographed. The mass losses were also weighted in order to realize the characteristic curves MDE (t) and MDER(t). In Table 3 are given images for one of the three tested specimens at three distinct times with relevance for the erosion evolution.

Table 3. The evolution of the surface erosion after cavitation exposure

Materials	0 min	90 min	165 min
17CrNiMo6 Carburized			
17CrNiMo6			
41Cr4			

images show clearly that the mass loss is increasing constantly till the minute 90 of the exposure time and after that the erosion remain constant. Figures 9 and 10 present the cumulative mean depth erosion (MDE) and

the mean depth erosion rate (MDER) during the 165 minutes of exposure.

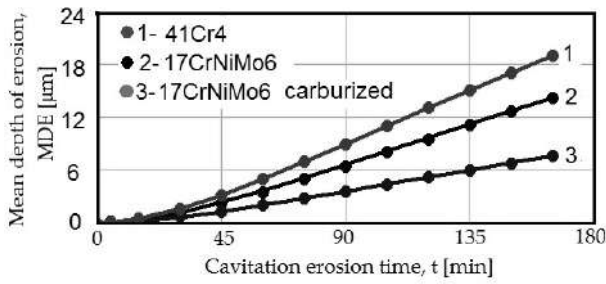


Fig.9 Variation of cumulative mean depth erosion during the exposure

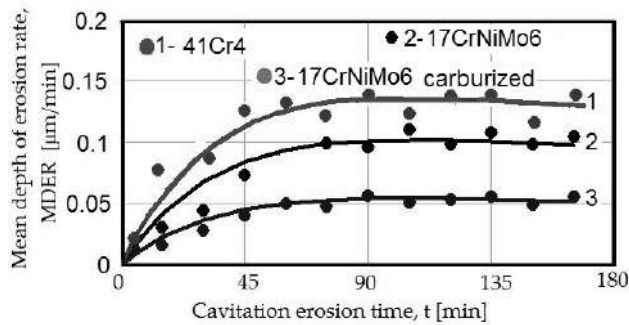


Fig.10. Variation of mean depth erosion rate during the exposure

The reduced scatter of the experimental points, both in annealed state (curve 2) as well as thermo-chemical treated (curve 3) show that the exposed surface of the steel 17CrNiMo6 is homogeneous from the cavitation erosion point of view. From the two tested states, the carburized one (noted with 3) is net superior with regard to cavitation erosion resistance.

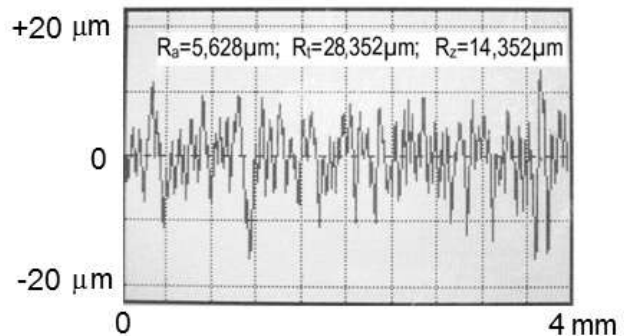
Regardless of the considered parameter, the maximum penetration depth after 165 minutes (MDE Figure 9), or the value at which the erosion rate curve (MDER-Figure 10) has the tendency to become stable, the resistance of the annealed steel is greater with 25% and those of the thermo-chemical treated steel with approximate 60% in comparison with the steel 41Cr4.

Both the curves evolutions and the values of MDE and MDER at the maximum exposure time (165 minutes) show that the thermo-chemical treatment through the proprieties generated in the superficial layer (especially the high hardness see Figure 7) enhances the cavitation erosion resistance with 47%. Evidently this increase is the effect of the mechanical properties of the Cr, Mo and Mn carbides (high tensile and compression resistance, increased hardness [1]) but also by the formation of the tempered martensite, structural component with great cavitation erosion resistance [2], [8], [9].

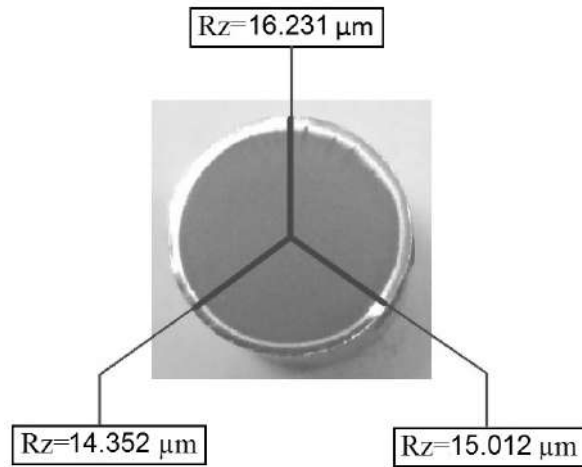
3.2 Erosion morphology

Figure 11 present some images which allow evaluating the cavitation erosion resistance of the specimen manufactured from an annealed sample. All the images were obtained after 165 minutes of cavitation exposure.

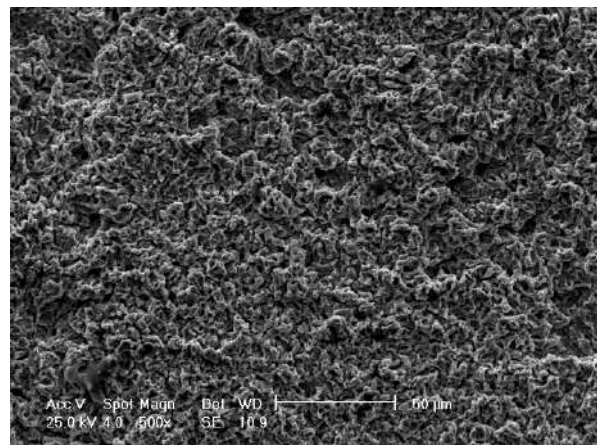
Figures 11a and 11b present images of the area eroded by cavitation. The profile of roughness in Figure 11a and the close values of the Rz roughness in Figure 11b show the uniformity of the erosions upon the whole surface. The explanation consists not only in the high values of the mechanical properties but also in the uniform properties of the entire exposed area. The erosion profile diagram in Figure 11a as well as the close values of the mean roughness Rz from Figure 11b show that the cavitation erosions are uniformly disposed on the surface. The surface micrograph presented in Figure 11c and 11d show that the surface presents secondary inter-granular cracks placed between cavities with different dimensions, especially in the ferrite matrices, the fractures having a fragile character and being situated near the boundaries pearlite/ferrite.



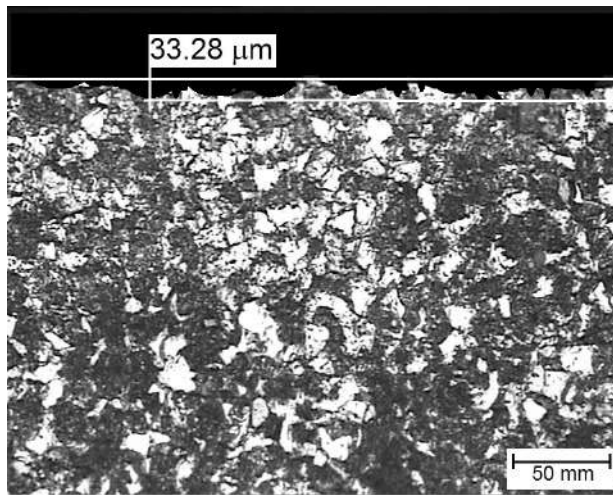
a) Roughness variation upon a measuring direction



b) Specimen mean roughness upon the three measuring directions



c) Eroded surface micrograph (frontal view x500)



d) surface micrograph (axial section x 200)

Fig.11. Specimens (2), annealed, after 165 minutes of cavitation exposure

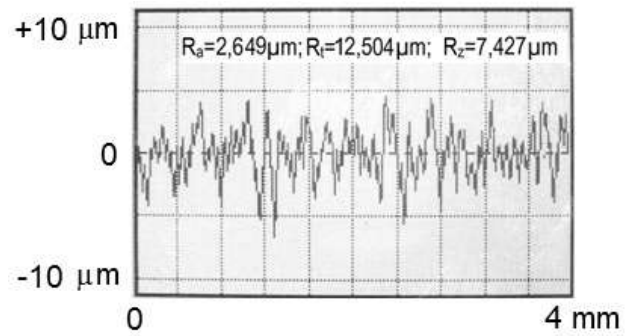
In Figure 12 are given images which allow to evaluate the cavitation erosion of the thermo-chemical treated (carburizing followed by quenching and low temperature tempering) specimens. The Figures 12a and 12b present the cavitation erosion of the carburized surface. The profile diagram in Fig. 12a and the close values of the Rz in Figure 12b show the uniformity of the erosion as a result of the high values of hardness and mechanical resistance of the formed carbides (of chromium, manganese and molybdenum) against the jet impacts or the shock waves created by the implosions of the cavitation bubbles. The reduced level of the profile variation show that the heat treatment after the gas carburizing procedure conducted to a fine structure and to homogenization of the properties in the carburized layer. From the micrographs of Figures 12c and 12d we can see that the eroded surface present fine, but also rude cavitations generated by inter-granular fractures on the lines where the different carbides structure meets.

In Figure 13 are given comparisons between the parameters of cavitation erosion resistance and the roughness of the eroded surface. It is to be observed that for the same type of specimen the differences between the mean roughness Rz and the mean depth erosion is very small. In the case of the annealed specimens Rz is greater than MDE but for the thermo-chemical treated specimens the situation is reversed.

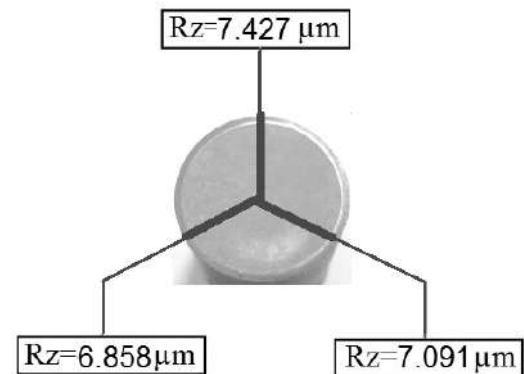
This aspect shows the complexity of the interaction of cavitation influence and the material response which in the annealed state can accept the carbides to be expelled as a result of fragile fracture.

The differences between the parameter RT and the maximum depth DE_{max} which appear in Figures 13 are easy to be explained by the measuring procedure. The profile measurements Rz and Rt are realized on a line of 4 mm length chosen arbitrary on the eroded surface while DE_{max} is the maximum depth measured in the final cross sections. So, the measurements were made on two different lines. The comparisons between the two types of specimens presented in Figure 13, show that carburizing followed with quenching and tempering increases

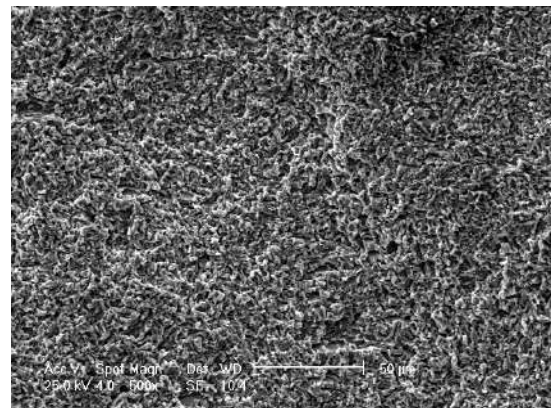
substantially the cavitation erosion resistance against the annealing heat treatment.



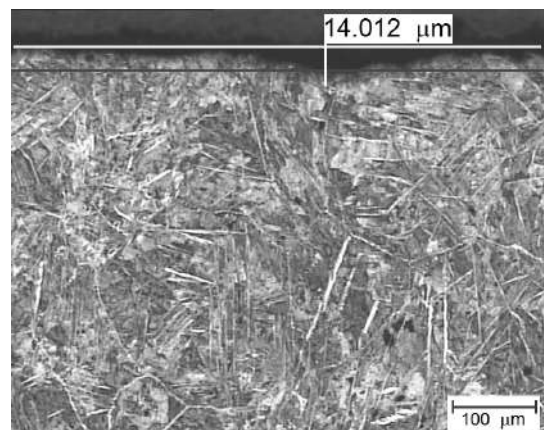
a) Roughness variation upon a measuring direction with a length of 4 mm



b) Mean Rz roughness upon the three measured directions



c) Micrograph of the eroded surface (frontal view x500)



d) Micrograph of the eroded area (axial section x200)

Fig.12. Specimens (3), annealed, after 165 minutes of cavitation exposure

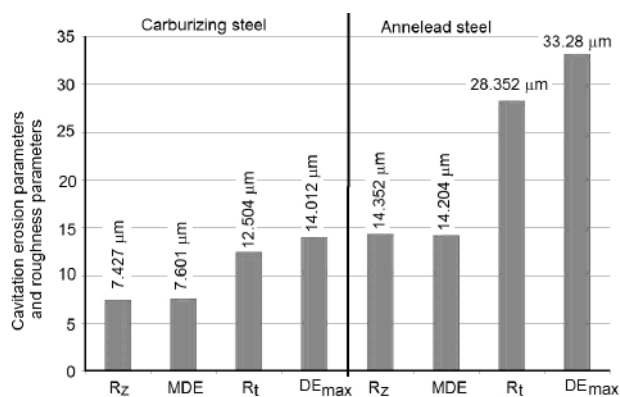


Fig.13. Comparisons between the characteristic parameters

4. CONCLUSION

The researched steel has good cavitation erosion resistance even in the genuine state.

The applied thermo-chemical treatment (carburizing followed by quenching and tempering at low temperature) increase the cavitation erosion resistance with approximate 47%, by modifying the structure and simultaneously increasing the mechanical properties.

The use of these steel together with the thermo-chemical treatment is recommended for the life increase of the details used in hydraulic drive systems.

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LINEAR ELECTRIC MOTORS – NEW POSSIBILITIES FOR SMART LINEAR MOTION

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Igor FÜRSTNER

Abstract: Modern solutions in electric actuators allow direct linear motion without the need of transforming the rotating movement into a linear one. This allows machine design with lower levels of complexity if linear movement is required. On the other hand, the integration of electric actuators and control units transforms them into complex mechatronic devices. Having in mind the possibilities of modern electric actuators and the complexity due to integration of executing and control units, and also the communication with external control units and sensors, as well as other actuators, it can be concluded that more attention than before is required when the mechatronic devices are formed in this way. This integration of actuators, control units and communication technologies is nowadays often referred to as Industry 4.0, which incorporates a number of contemporary automation, data exchange and manufacturing technologies and is a facilitation of the vision and execution of a “Smart Factory”. This research deals with the problem of programming two linear electric actuators, which work simultaneously using an external control unit, and communicate through EtherCAT communication protocol, and present a possible constituent of a Smart Factory. The problem is discussed by presenting a case study of controlling two linear motors.

Key words: linear motor; mechatronic device; smart factory

1. INTRODUCTION

Nowadays, we are witnessing a change in industry conception and organization. While the third industrial revolution introduced advanced electronics and information technology, which allowed for further automation of production processes, today development and establishment of global networks incorporate the machinery, warehousing systems and production facilities in the shape of Cyber Physical Systems. In the

manufacturing environment, these Cyber-Physical Systems comprise smart machines, storage systems and production facilities capable of autonomously exchanging information, triggering actions and controlling each other independently. This facilitates fundamental improvements to the industrial processes involved in manufacturing, engineering, material usage and supply chain and life cycle management. The Smart Factories that are already beginning to appear employ a completely new approach to production. Smart products are uniquely identifiable, may be located at all times and know their own history, current status and alternative routes to achieving their target state [1]. This business approach is named Industry 4.0, which is a term publicly known from 2011, and which includes six design principles that are presented in Table. [2].

Table 1. Design principles of each Industry 4.0 component

	Cyber-Physical Systems	Internet of Things	Internet of Services	Smart Factory
Interoperability	x	x	x	x
Virtualization	x			x
Decentralization	x			x
Real-Time Capability				x
Service Orientation			x	
Modularity			x	

Interoperability means that all Cyber Physical Systems are able to communicate with each other, and are connected over to Internet of Things and Internet of Services in the context of SmartFactory [3]. Virtualization means that a virtual copy of the physical world allows the monitoring of the processes [4]. Decentralization as a demand, which arises due to increasing difficulties to control systems centrally, makes it necessary to keep track of the whole system at any time [5]. To be able to keep track of the whole system at any time, real-time capability is required [5]. To be able to answer to specific customer requirements, service orientation is needed as well [5]. Finally, modularity as an approach enables flexibility regarding to the adaptation of changing requirements by providing functional independency of individual modules [5][6].

Modern mechatronic devices are gaining more ground in multiple areas. Large number of devices contain more and more mechatronic elements [7]. These devices have multiple sensors and actuators. The most widely used actuators are rotating motors, which are used for achieving both linear and rotational movements. They are well developed and studied in detail. Usually rotational motion is converted to linear motion, or the well-known pneumatic/hydraulic systems are used for linear motion. These solutions usually offer a simple and cheap way to achieve the linear motion with good precision and performance. However, there are multiple different options to achieve linear movement. Among these, linear motors are starting to appear as a potential option for solving the problem of linear motion. The linear motor's characteristics are similar to the conventional systems, but

this system has additional positive attributes, which makes it a desirable solution.

The basic principle of the linear motor was known in 1895, but the first functioning model was developed in 1947. It can be imagined as an unfolded rotating motor. There are different types of linear motors, including step by step, DC, AC induction, iron cored and ironless brushless DC linear motors, and tubular linear motors [8]. The linear motor used in this research is a tubular linear motor. This type of linear motor is created when the regular BLDC motor is unfolded, and then folded back along the line, which is perpendicular to the magnetic field's spreading direction [9] [10]. It can be made with short and long coils, but since the long coils are expensive to make the shorter coils are more common. The moving part is usually a long tube, which has neodymium magnets in it with alternating polarities.

If the motor has to work in conditions, which require high precision positioning, an external encoder has to be included. Typically the working force is around 600N. It can achieve 20G acceleration up to 5 m/s velocity [11] [12]. The repetition precision is 0.05 μ m, and the gap is less than 0.5 mm.

The motors which are presented in the case study are the Dunkermotoren Servo Tube STA1104-1116 linear motors [9].



Fig.1. Example of a Servo Tube STA1104-1116 linear motor

The presented research deals with the problem of programing a group of Servo Tube STA1104-1116 linear motors, which work simultaneously using an external control unit, and communicate through EtherCAT communication protocol. This structure presents a possible constituent of a Smart Factory.

The motors are used and the programing is performed as a further development of an ongoing project started as a student contest at Subotica Tech – College of Applied Studies, sponsored by Dunkermotoren GmbH [7].

The reminder of the paper is structured as follows: In the 2nd paragraph the DSP 402 protocol, which enables the communication via EtherCAT, between the motors and the control unit is presented in brief. Also a short overview of SysmacStudio software, which allows the programing of the system is given as well, together with the introduction of the external control unit, i.e. the used programmable logical controller (PLC). The 3rd paragraph presents structure of the system of electric actuators, control unit, and auxiliary components which form the mechatronic system that is presented and the case study. Conclusions are given in the 4th paragraph.

2. THE MOTION CONTROL

2.1. CiA DS402 Protocol

The CiA DS402 protocol [13] is responsible for the implementation of the motion control, which is based on the Elmo DSP working principle. The DSP 402 protocol refers only to operation under load, together with the operation's position, velocity and limits. The protocol is used to enable the motion of the actuators in a CAN network. The protocol is based on the CAN communication, which is called CANopen. It determines the basic communication mechanisms for each actuator in CAN network. In this work the CAN over EtherCAT (CoE) type of CAN implementation is used.

2.2. SysmacStudio Software

The machine control is integrated into only one software, which integrates motion control, safety features, image processing, and information management, logic control, imaging and networking. The software is called Sysmac Studio [14][15][16]. The software provides an integrated development environment, which includes a unique 3D simulation tool.

SysmacStudio integrates configuration, programming and monitoring. Due to the graphic configuration, the control of local devices and networks can be assembled quickly, thus reducing programming time. The software also features a smart editor, which provides online debugging functions, enabling faster and error-free programming.

2.3. Used PLC

The control of the presented mechatronic system is performed by a PLC developed by Omron, which uses a developed Sysmac system. This PLC series enables the functionality, the high speed and secure control, and the reliability and serviceability. It is compliant with IEC 61131-3 standard and the PLCopen function blocks, which belong to the motion controller. It allows fast and precise control over EtherCAT devices.

3. SYSTEM STRUCTURE AND CASE STUDY

3.1. System structure

The structure of the system is presented in Fig 2 and Fig. 3. It consists of two Dunkermotoren Servo Tube STA1104-1116 linear motors and an external control unit, the Omron NJ301-1200 PLC.

3.2. Case study

The case study is structured as follows. First, the kinematics of the motors is presented. After that, some screenshots of the programming and of the program are presented.

Both motors have pre-set constant accelerations (a) with different values for different time ranges which is presented in Table. Acceleration, as well as the velocity (v) of the motors, and the motors' position (p) are given in increments (inc), where one increment equals 1/1024mm.

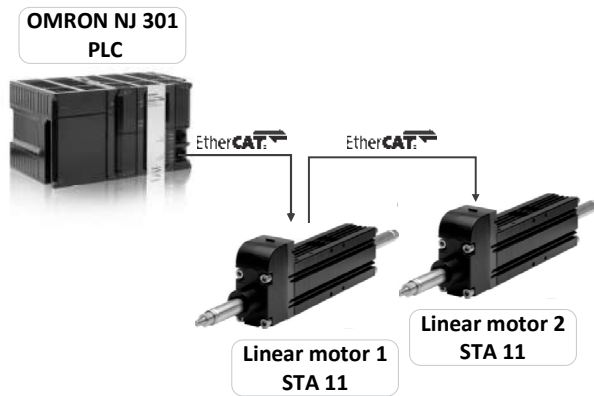


Fig.2. System structure with PLC and linear motors

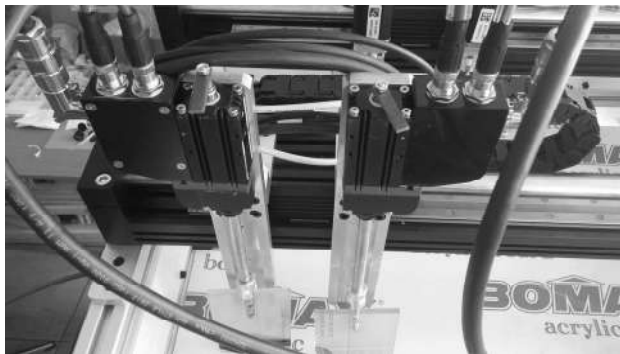


Fig.3. Two Dunkermotoren Servo Tube STA1104-1116 linear motors installed on the system

The motors' velocity is calculated from acceleration (1).

$$a = \frac{dv}{dt} \rightarrow dv = a \cdot dt \rightarrow v = \int a \cdot dt = a \cdot t + \text{const} \quad (1)$$

For different time ranges const has to be determined separately, which is presented in Table 3.. The graphic presentation of the velocities is given in Fig.4.

Table 2.. Acceleration values

acc. motor A [inc/s ²]	acc. motor B [inc/s ²]	time ranges	abs. start time [s]	abs. end time [s]	relative start time [s]	relative end time [s]
1000	0	A	0	0.2	0	0.2
0	1000	B	0.2	0.6	0	0.4
-1000	0	C	0.6	1	0	0.4
0	-1000	D	1	1.4	0	0.4
1000	0	E	1.4	1.6	0	0.2

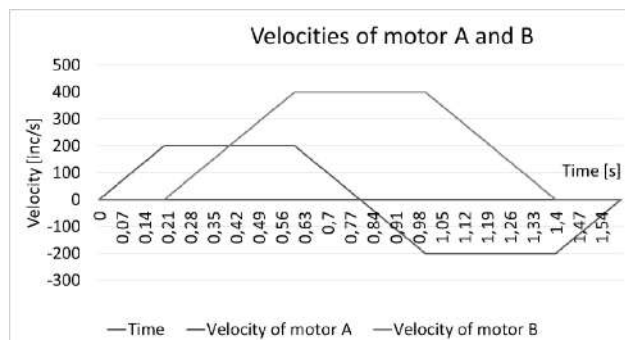


Fig.4. Velocities

For the given time ranges position (p) has to be determined separately as well. Position is calculated from velocity (2).

$$v = \frac{dp}{dt} \rightarrow dp = v \cdot dt \rightarrow p = \int (a \cdot t + \text{const}) \cdot dt = \int a \cdot t \cdot dt + \int \text{const} \cdot dt = a \cdot \int t \cdot dt + \text{const} \cdot \int dt = a \cdot \frac{t^2}{2} + \text{const} \cdot t + \text{const}2 \quad (2)$$

The values for the const2 are presented in Table 4., while the graphic presentation of the positions is given in Fig.5.

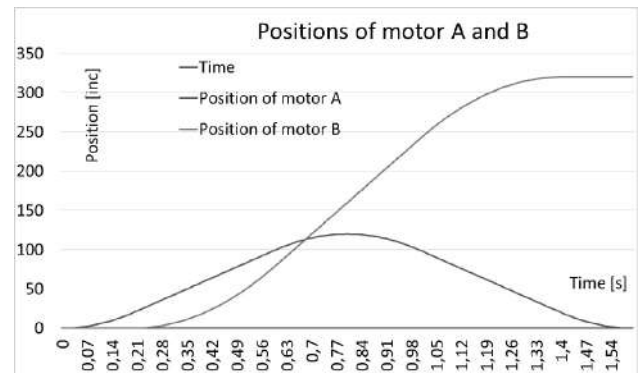


Fig.5. Positions

4. CONCLUSION

New developments of electric actuators allow direct linear motion without the need of transforming the rotating movement into a linear one. This allows machine design with lower levels of complexity if linear movement is required. Also, the ongoing change in industry conception and organization through the establishment of global networks results in the creation of Cyber Physical Systems, which in the manufacturing environment, comprise among others smart machines, which are capable of autonomously exchanging information, triggering actions and controlling each other independently.

The presented research deals with the problem of programming two Servo Tube STA1104-1116 linear motors, which work simultaneously using an external control unit (Omron NJ301-1200 PLC), and communicate through EtherCAT communication protocol, which presents a possible constituting structure of a Smart Factory. The communication via EtherCAT between the motors and the control unit is enabled by using the DS402 communication protocol, while the programming of the system is done in SysmacStudio software.

Table 3. Values of const for motors A and B

Time range	Value of const (based relative start time) $\left[\frac{inc}{s}\right] = v_0 - a \cdot t_0$	Motor
A	0	A
B	200	A
C	200	A
D	-200	A
E	-200	A
A	0	B
B	0	B
C	400	B
D	400	B
E	0	B

Table 4. Values of const2 form motors A and B

Time range	Value of const2 $[inc/s] = p_0 - a \cdot (t_0^2) / 2 - const \cdot t$ (based on relative start time)	Motor
A	0	A
B	20	A
C	100	A
D	100	A
E	20	A
A	0	B
B	0	B
C	80	B
D	240	B
E	320	B

The electric motors are controlled by defined pre-set constant accelerations with different values for different time ranges within the total time range, based on which the velocities and the positions of the motors are obtained. The result of the case study shows that the presented mechatronic system can work properly with the given structure, which is a good starting point towards designing a more complex mechatronic system. Therefore, future research will focus on integrating more actuators in the system, which will work simultaneously as well. Also the control of the system will integrate an on-line control influenced by the end user.

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ANALYSIS OF NEW TECHNICAL SOLUTION IN PROCESS OF DETOXIFICATION ELV FROM ENVIRONMENTAL ASPECT

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Abstract: Adequate detoxification is component of successful recycling end of life vehicle (ELV) processes. During the process of detoxification all materials which are dangerous for the environment are discharged from the vehicle. Chemical composition of the brake fluid makes that this matter take a special place in the process of detoxification because their characteristic which are hazardous for the environment. Consequently, there is a need for development of appliance which will have a possibility of discharging fluid without contact with the environment, which will with the further treatment dispose on the place designated for that and all with the aim to protect the environment.

The aim of the paper is analysis of the new technical solution, appliance for discharging brake fluid from vehicle. There are shown a technical characteristic of the appliance, it is simulated the effect of the appliance on the brake hose and also are shown a possibilities of using appliance on the existing plant model for discharging liquids from vehicles in the process of ELV

Key words: Environment protection, ELV detoxification, brake fluid, technical solution, technical characteristics

1. INTRODUCTION

Motor vehicles with all their components, metal part and toxic fluids which come to the ELV presents important contaminants of environment.

Hazardous materials with complex chemical composition comes with development of science and technology. Some of that materials are not biodegradable on the natural way and their degradation last infinitely long. All this leads to the conclusion that inadequate treatment, begin from the detoxification to the dumping and placing waste are very risky and ruinous job on the environment. One of the most dangerous is the brake fluid.

Brake fluid is the mixture of the polyglycol ethers and glycol with additives which inhibit process of oxidation and corrosion of the brake system [1,2]. Chemical composition like this can cause contamination of land and surface water.

Shown chemical structures point out that the brake fluid represents hazardous waste, unless it is recycled. Mixing with brake oil doesn't allow further recycling.

Brake fluid must not be treated as used oil, as it has characteristics of toxicity. Mix with chlorinated solvents from spray, which is used for cleaning the brakes, makes the brake fluid extremely hazardous [3]. All this reasons create a need for secure treatment of this fluid, in order to avoid contact between environment and operators of the process of detoxification.

2. LEGISLATIVE ASPECT

Legislative aspect related with waste oil is covered with Directive of council 75/439/ESS for dumping waste oils, amended with directives 1987/101/ESS, 91/692/ESS and 2000/76/ES promotes gathering and dumping of mineral lubricants or industrious waste oil that are no longer usable for original usage [4].

Republic of Serbia legislature features rule book about conditions, method and process of managing waste oils [5], which establishes conditions, method and process of managing waste oils which are unusable for purpose they were designed in the first place.

3. THE AMOUNT OF THE WASTE BRAKE FLUID IN THE REPUBLIC OF SERBIA WHICH NEED SAFE DISPENSING AND ADEQUATE TREATMENT

Figure 1. shows number of registered vehicles with average number of ELV in the last 14 years in the Republic of Serbia.

Table 1. The number of registered and waste vehicles in the Republic of Serbia in the last 14 years (2001-2014) [6]

Year	Number of registered passengers cars	Average number of ELV (5,35 % of the number of registered vehicles)
2001.	1382396	73959
2002.	1343658	71886
2003.	1388109	74294
2004.	1449843	77567
2005.	1481498	79260
2006.	1511837	80883
2007.	1476642	79001
2008.	1486608	79534
2009.	1637002	87580
2010.	1565550	83757

2011.	1677510	89747
2012.	1726190	92352
2013.	1770162	94704
2014.	1797252	96153
Σ	21694257	1160677

The weight of the brake fluid per vehicle is 0,4 kg which makes 0,04 % of total vehicle weight [7], with a degree of recyclability 0 %. Multiplying the sum of the average number of registered vehicles in the Republic of Serbia in the last 14 years which comes to the ELV with waste brake fluid per vehicle give results of 464270,8 kg, which is nearly half of tone of these hazardous substances for the period of 14 years. On the annual basis that number is 33162,2 kg [8], which representing not to small amount. Presented data suggest on the need for quality treatment for this toxic substances so do not come to the contact with environment.

4. EXISTING EQUIPMENT FOR DETOXIFICATION OF ELV

Figure 1. shows existing technical solution for discharging fluids from ELV and represent stationary station for combined removal of all vehicle fluids with minimum influence on the environment.

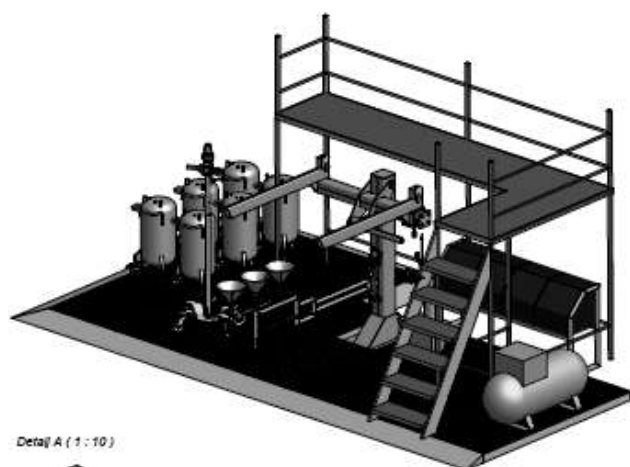


Fig.1. Plant model for discharging liquids from vehicles in the process of ELV [9]

A part of dispensing, which applies on brake fluid, is performed using a funnel that is positioned under the cylinder on the wheels of a vehicle. By unscrewing the screw on brake cylinders, the brake fluid outflows in the funnel by free fall, from where, under the effect of sub pressure, it is transported into the suitable tank for brake fluid acceptance. A connection is installed, on the tank for brake fluid in the motor area, which provides the transfer of air from the compressor into the mentioned tank. The created overpressure in the brake system of the car improves the dispensing of the brake fluid and prevents the fluids from reservation in the brake system of the vehicle [9].

In order to avoid free fall of brake fluid, when unscrewing the cylinder, and with that a possibility of eventually leaking of this hazardous substances outside of the funnel, it is proposed new technical solution, appliance, which will process of dispensing make safer.

5. APPLIANCE WHICH IMPROVES DISPENSING PROCESS OF BRAKE FLUID FROM ELV

Developed device is consisted of undersystems which altogether function is discharging brake fluid out of the brake hose, which are integral part of vehicle.

Fig. 2. shows three-dimensional appliance model which do dispensing of break fluid, while fig.3. shows three-dimensional appliance model separated on belonging elements.



Fig.2. Three-dimensional appliance display for dispense improvement of vehicle brake fluid in the process of ELV [10]

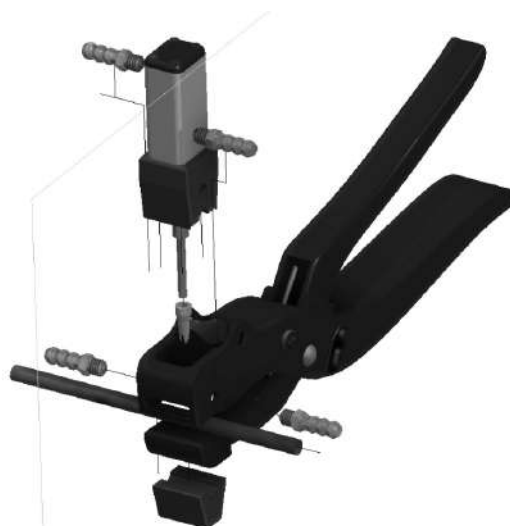


Fig.3. Three-dimensional appliance look for dispense improvement of vehicle break fluid in the process of ELV [10]

Suggested device gives possibility that instead unscrewing screws on brake cylindars and free fall of brake fluid, with direct brake tube perforation with perforation tool, comes to dispensing of fluids which under effect of under pressure transports into matching provided reservoir goes on further treatment. With that it

has been prevented possibility of brake fluid dispense out of reservoir predicted for that type of fluid, thus contact of this chemical toxic matter with environment and their participants.

5.1. Tehnical characteristic of device

Fig 4. represents drawing set with integral elements for positions on device.

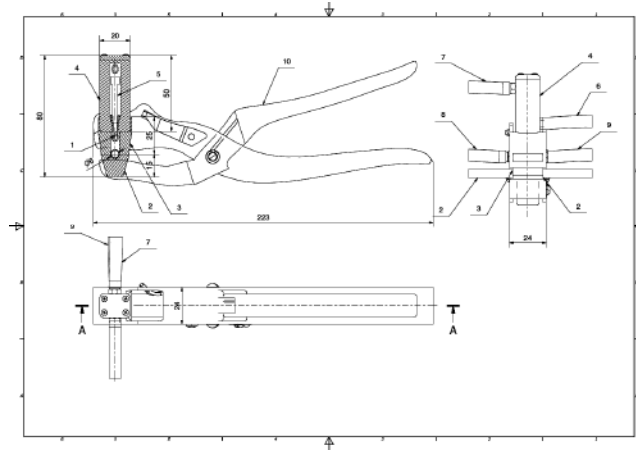


Fig.4. Device drawing set for upgrade of brake fluid dispenser for vehicle on the end of ELV [10]

The device consists of the following parts:

1. Perforation tool; 2. Bottom part of gasket; 3. Upper part of gasket; 4. Mechanism body for perforation; 5. Pricket clip ; 6. Supply of working fluid; 7. Drain of working fluid; 8. Exit to vacuum pump; 9. Vent; 10. Device pliers [10].

5.2. Display simulation of brake hose perforation

Perforation is processing method in which separation of material is done by closed contour. Perforation happens when shearing tension, caused by perforation force is bigger than firmness of consonant material. According to literary sources we take that firmness of material on shearing is $\tau_m \approx 0,8R_m$ where is R_m tightened force of material [10].

On Fig. 2. and 3. brake hoses model is noticable from which fluid is flowed

On newer vehicles, brake pipes have been made out of teflon and strengthened with braided stainless steel which protects teflon, while on older vehicles they are mostly made of from copper and rubber. However, since vehicles which come to ELV are older generation, as referent we take tubes made mostly out of copper and rubber. Concretely for this perforation case is taken copper brake tube, which is 20 mm lenght, 6 mm outside diameter and inside 4,5 mm.

On Fig. 5.,6. and 7. pipe load simulation is shown while force is acting vertically on extended pipe axis.

Fig. 5 showing pipe load simulation with transverse 0,15 kN load, whereby is noticable that deformations are within the limits of elasticity.

Fig. 6. showing pipe load simulation with transverse 0,3 kN load, whereby is noticable that it is coming to plastic deformation, but based on tension shedule we can conclude that it will not come to perforation, because there is no transgression of shearing stress of 320Mpa.

Fig. 7. showing pipe load simulation with transverse 0,5 kN load on which also comes to plastic deformation, but in that occassion it comes to perforation, because needed tension of 320 Mpa is exceeded.



Fig.5. Transverse tube load by 0,15 kN force [10]

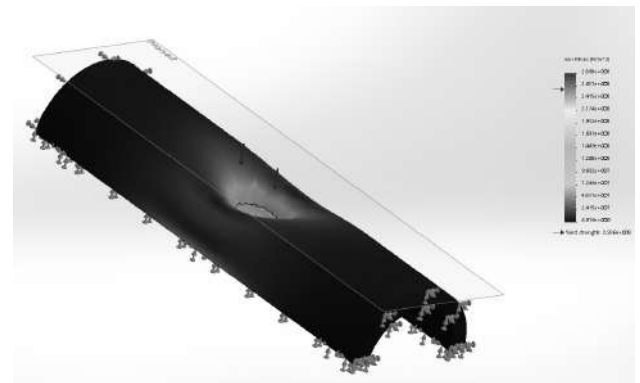


Fig.6. Transverse tube load by 0,3 kN force [10]

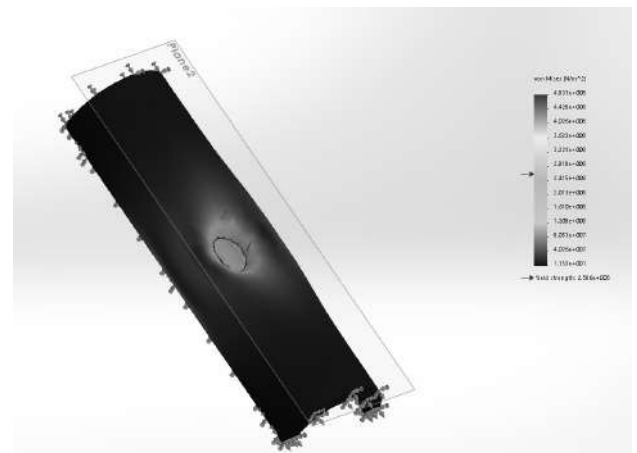


Fig.7. Transverse tube load by 0,5 kN force [10]

6. CONCLUSION

Without successfully completed process of detoxification there is no successfully recycling ELV process. The importance of a successful detoxification is that almost all liquids which are discharging from vehicles are not biodegradable on the natural way and their degradation last infinitely long. One of that liquid is brake fluid which is treated as hazardous waste. Moving to the aim which is adequate recycling vehicles which come to the ELV,

beside raising the awareness of people about importance of this part of recycling process and fulfillment regulations, it is necessary to develop technologies what will create conditions for safe detoxification with environmental aspect.

To this end it is developed technical solution which is presented in this paper and which will improve process discharging of brake fluid from ELV and fulfill needs of recyclers.

Developed solution brings improvement in terms of safer discharging of brake fluid for environment and the protection for the operator of the process of detoxification with which be created conditions for further treatment.

When all activities of detoxification were made on the vehicle then vehicle like that is classified as non hazardous waste and it can be recycled.

ACKNOWLEDGEMENT

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ESTIMATION BY FUZZY LOGIC OF ABRASIVE WEAR PROPERTIES OF COATED VALVES SURFACES BY TIG WELDING

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Uğur ARABACI

Abstract: Abrasion behaviors under different loading conditions in 1.4718 coded steel, the surfaces of which are coated by TIG welding method with the alloys Stellite F, Stellite 1, Stellite 6 and Nickel 60 are modeled by FL method in this study, based on the study previously done experimentally. As experimental study, the surface-coated samples were subjected to abrasion wear under the loads 10, 25 and 40 N through 10, 5 m. As a result of abrasion range, weight losses were measured and abrasion resistance of the samples was examined. The Fuzzy Logic (FL) model was formed according to results of the present experiment. 2 input variables and 1 output variable were used in the fuzzy logic formed. Material and applied load are input variables and weight loss is output variable. The obtained experimental results were transferred to the fuzzy logic system and estimated values were produced for intermediate values, which were not subjected to experiment. The results obtained from the fuzzy logic system were seen to be in harmony with the experimental data.

Key words: Abrasive wearing, valves, coating, TIG Welding, fuzzy logic

1. INTRODUCTION

Abrasion resistance is a property required in many materials in the manufacturing sector. However, testing this property and making experimental conditions

optimum are the studies which take a long time. The established criteria are generally constant and it will provide convenience to computerize these kinds of experimental studies and estimate the results, which will be obtained in the other intermediate values, in the light of certain values since the study of intermediate values causes additional cost and time.

Reflections of developments in computer and information technology on the manufacturing Technologies have introduced Computer Aided Manufacturing (CAM), Flexible Manufacturing (FM) and Computer Integrated Manufacturing (CIM) systems. [1]. This automation and this flexibility in manufacturing not only increase the production amount and improve the quality, but also shorten the cost and production time [2]. Besides, modeling techniques for the control of more complicated systems, accordingly their simulation, monitoring and system analysis have gained importance [3]. Some modern methods have been developed to more reliably model those kinds of systems [4]. The most commonly used of them are the artificial intelligence applications such as fuzzy logic, artificial neural Networks, genetic algorithms. The fuzzy systems are also commonly used in psychology, medicine, science, engineering and many other fields [5]. FL based on fuzzy set theory is a theory of thinking and decision making which enables it to make a consistent and right decision by using fuzzy and uncertain information [6].

In this essay, based on the study previously done experimentally [7], Abrasion behaviors under different loading conditions in 1.4718 coded steel, the surfaces of which are coated by TIG welding method with the alloys Stellite F, Stellite 1, Stellite 6 and Nickel 60 are modeled by FL method. Experimental samples were subjected to static loads with 10, 25 N and 40 N through 10, 5 m. during abrasion experiments. 400-mesh Al₂O₃ type abrasive cloth was chosen as abradant. Coating abrasion values were modeled by fuzzy logic, by making use of the results of experiment. Estimation values obtained from fuzzy logic and abrasion values obtained from experiments were compared.

2. MODELLING AND ESTIMATION BY FL METHOD OF THE ABRASION IN COATINGS

System design has been made to interpret the experimental data with fuzzy logic, as seen in Table 2. 2 input variables and 1 output variable were used in the

fuzzy logic system. A rule block, which includes 24 rules, and 12 membership functions were generated. Certain parts of the results obtained from experimental studies were not entered into the fuzzy logic system. Those results were used to test the fuzzy logic system.

Table 1. Project Statistics

Input Variables	2
Output Variables	1
Rule Blocks	1
Rules	24
Membership Functions	12

Structure of the fuzzy logic system is shown schematically in Figure 1. In this system, the cutting space and punch angle were used as independent variable and the cutting force was obtained as output variable

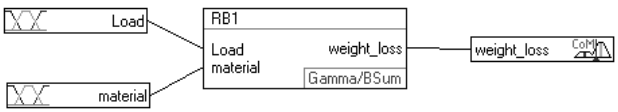


Fig.1. Structure of the Fuzzy Logic System

In the system developed in the fuzzy logic Project, “Compute MBF” method was used as Fuzzification Method and “Center of Maximum (CoM)” method was used as Defuzzification Method. Membership function chart of load, one of the system inputs, is seen in Figure 2

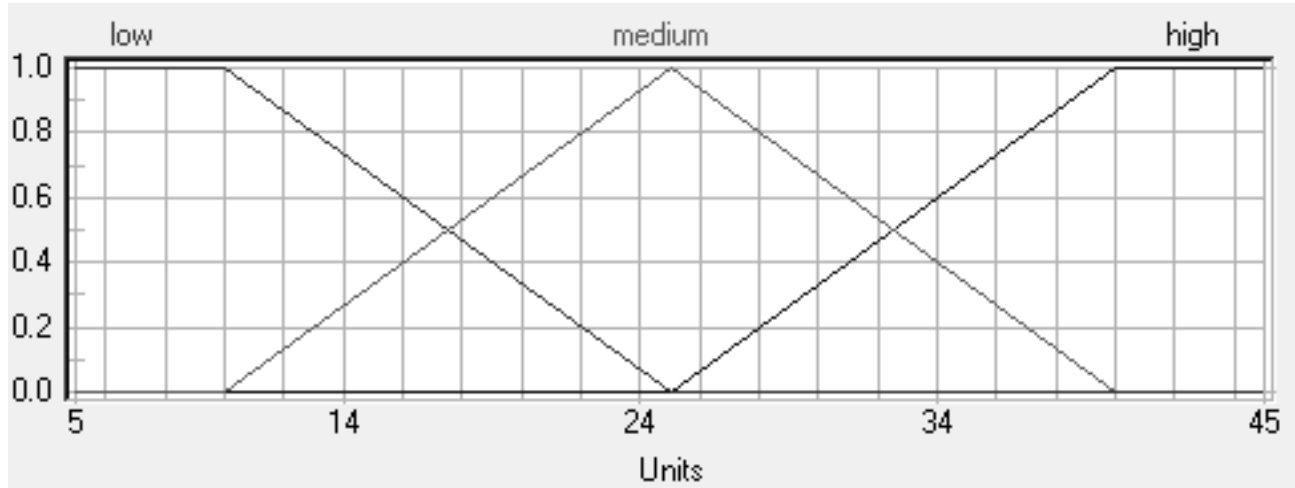


Fig.2. MBF of "load"

Membership function chart of material, another input, is as shown in Figure 3.

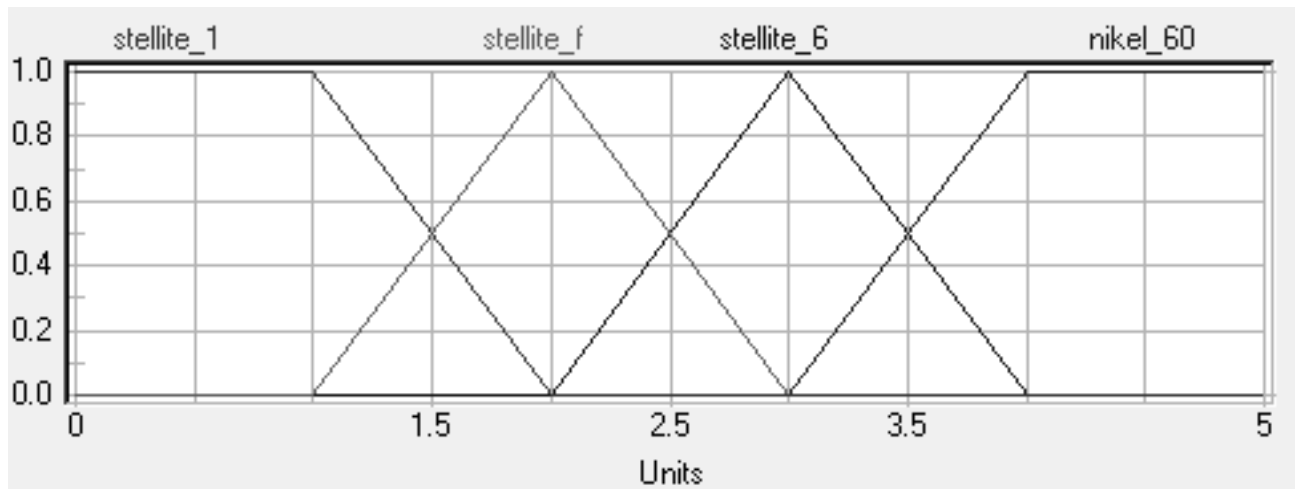


Fig. 3. MBF of "material"

Membership function chart of weight loss, which is single independent variable of the system, is as shown in Figure 4.

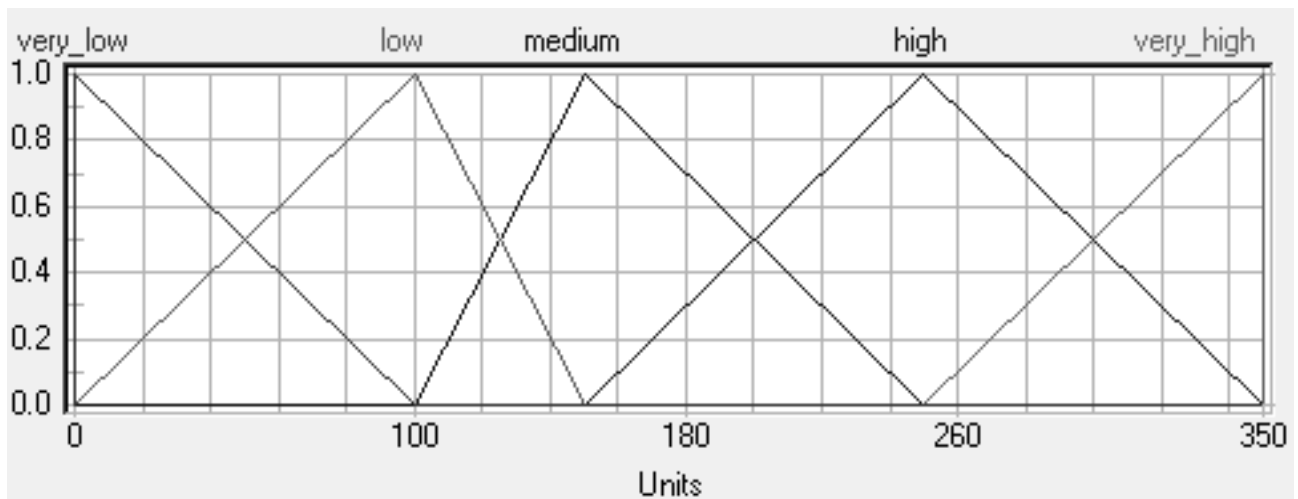


Fig.4. MBF of "weight loss"

The charts of Stallite 1 material, which are formed based on the experimental and fuzzy logic model of the change of material losses according to applied force, are seen in Figure 5. The fuzzy logic model and experimental results are quite approximate to each other. R^2 value was calculated as 0,9959 for two distributions.

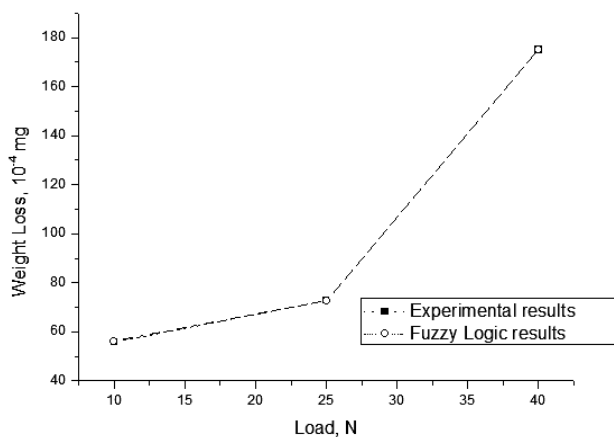


Fig.5. Experimental and Fuzzy Logical weight loss models for Stallite 1

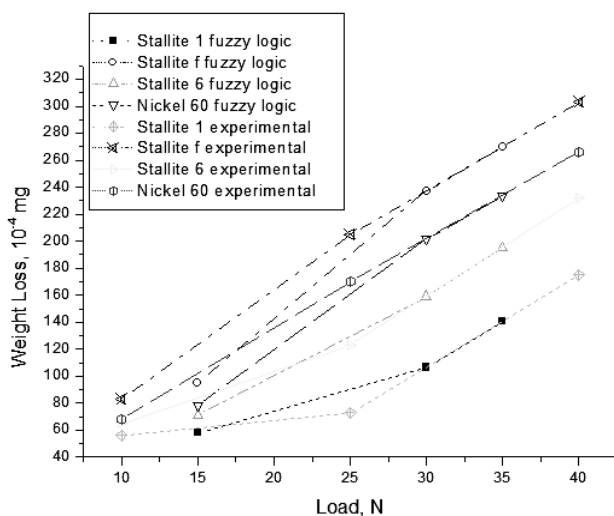


Fig.6. Experimental and Fuzzy Logical weight loss models for different loads

The charts of each material, which are formed based on the experimental and fuzzy logic model of the change of material losses according to applied force, are seen in Figure 6. The experimental results and the fuzzy logic estimations made for intermediate values, which were not subjected to experiment, are given together.

3. CONCLUSION

In this study, a model was set up, in which incomplete and experimentally unstudied abrasion values may be estimated by verbally clustering the experimental data, the abrasion behaviors in 1.4718 coded steel, which is coated by TIG welding method with the alloys Stallite F, Stallite 1, Stallite 6 and Nickel 60, and by establishing fuzzy relationships among those clusters.

Bearing wear characteristic obtained by this model is established as follows:

- The sample stellite 1 has the highest abrasion resistance and the sample stellite f has the lowest abrasion resistance on abrasive wears applied on 400-mesh al2o3 abradant,
- The abrasion losses increase depending on the increase in load amounts applied in abrasive wear experiments,
- The results of the fuzzy logic share similarity with the results of experimental study, and they can be used to estimate the intermediate values.

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END-MILLING FORCE CONTROL SYSTEM WITH SURFACE ROUGHNESS MONITORING

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Abstract: The aim of this paper is to present an adaptive force regulation system in end milling with the newly developed surface roughness prediction model. The relations between measured cutting force, cutting conditions and actual surface roughness of machined surface were determined via adaptive neuro-fuzzy inference modeling (ANFIS). The trained ANFIS model predicts the effect of machining parameters and maximal cutting force on surface roughness during machining. End milling machining process of hardened die steel with carbide end mill was used for modeling. The developed model monitors the actual surface roughness and in case of excessive value provides warnings to the operator or stops the machining process for tool change.

Key words: milling, adaptive control, monitoring, surface roughness, prediction, ANFIS

1. INTRODUCTION

Despite many research, there are no available method for monitoring surface roughness in high-speed end-milling. In milling the relationship between process characteristics and surface roughness is difficult to capture. This is due to the complexity of the relationship between surface roughness and process characteristics. Therefore, an in-process method based on prediction model is required for the real time process monitoring. We attempt to solve this situation by employing the ANFIS to predict the surface roughness based on measured cutting forces and to CNC controller sent cutting conditions. A new surface roughness prediction model is integrated with the existing cutting force control system. The traditional ability of the operator to determine the quality of the machined surface based on his experiences and senses is now the expected role of the predictive model in the upgraded system. Several other models have been proposed to estimate the surface roughness. These include classical statistical approaches as well as fuzzy systems and neural networks. For instance researchers [1,2] developed an approach based on the least-squares regression for estimating surface roughness in machining while [3] have, respectively, used fuzzy pattern recognition for monitoring surface roughness over a limited range of cutting conditions. The capacity of artificial neural networks to capture nonlinear relationships in a relatively efficient manner has motivated a number of researchers to pursue the use of these networks in developing surface roughness prediction models. But in these models, the nonlinear relationship between sensor readings and surface roughness embedded in a neural network remains hidden and inaccessible to the user [4]. In this research we attempt to solve this situation by using the ANFIS system to predict the surface roughness.

This model offers ability to estimate surface roughness as its neural network based counterpart but provides an additional level of transparency that neural networks fails to provide.

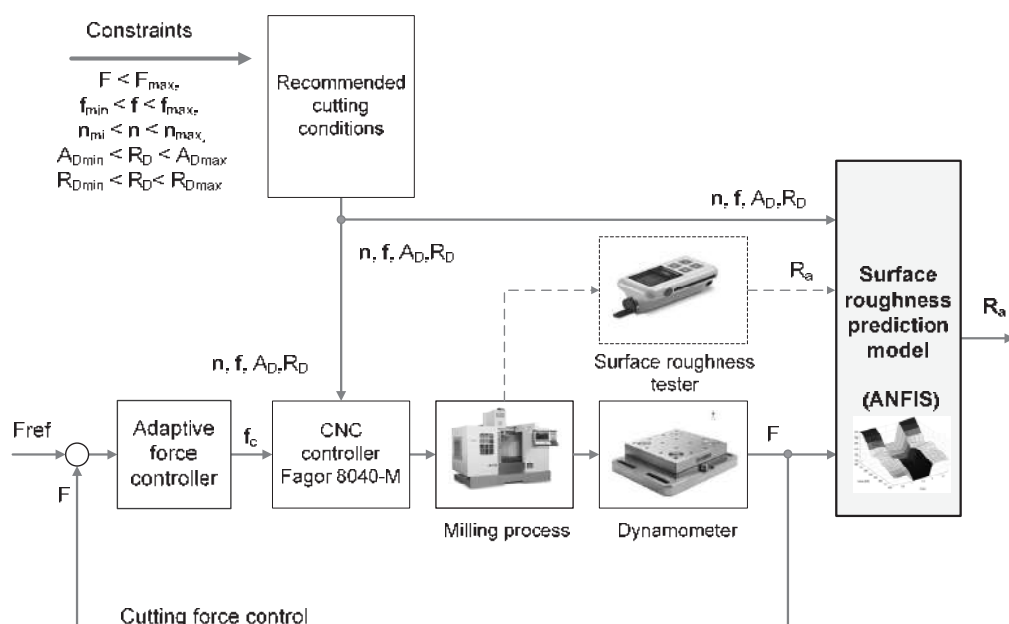


Fig.1. Inclusion of ANFIS predication model in the adaptive force control system

2. ADAPTIVE CUTTING FORCE CONTROL STRUCTURE

The adaptive control system shown in Fig. 1 is a cutting force feedback loop where the feedrate adjusts itself to the actual cutting force F , and varies according to the changes in work conditions during machining. The actual maximal cutting force is sampled every 0.01 seconds and converted to a digital signal. This signal is immediately compared with a predetermined reference cutting force F_{ref} . The maximum measured force is a good starting value for the reference force F_{ref} . The difference between the F and F_{ref} is the control error. The control error is used as the input to the system controller.

The controller adjusts the federate override percentage and sends the determined federate command signal to the federate routine in the CNC controller. A positive error increases the programmed feedrate and consequently increases the produced actual cutting force. A negative

error decreases the programmed feedrate and consequently decreases the actual cutting force. The controller is connected with the CNC controller of a 4-axis machining center, as shown in Fig. 1.

The federate control is achieved through Feed override commands. In order for the controller to regulate peak force, force information must be available to the control algorithm at every 20ms. Data acquisition software (LabVIEW) and the algorithm for processing the cutting forces are used to provide this information. When the adaptive controller is turned on, the feedrates generated by the CNC controller are sent to the feed drive servo-unit and the surface roughness prediction model.

This model predicts surface roughness on-line and displays the value to the operator. If the value of the surface roughness exceeds the predetermined limits, the warning message is sent to the operator.

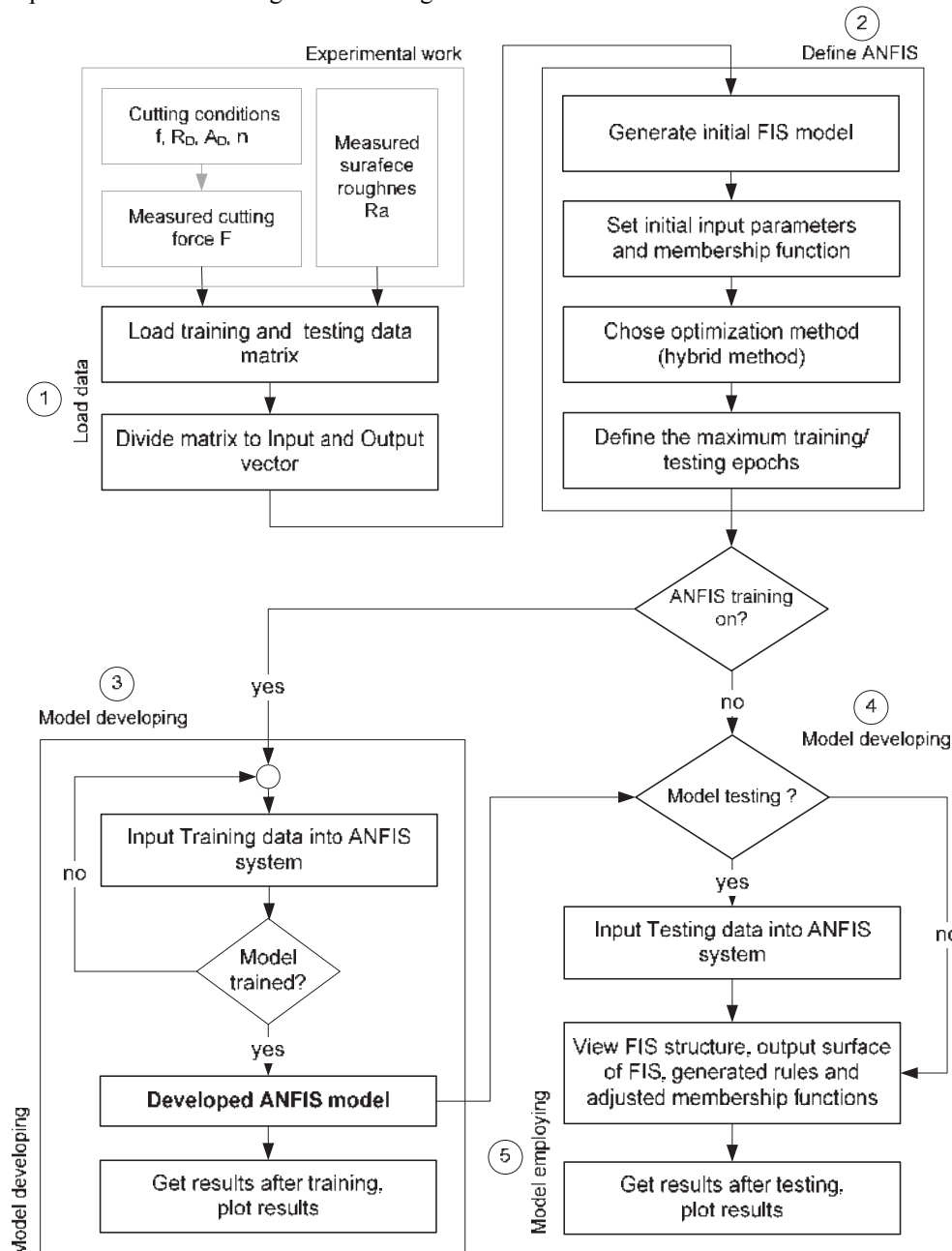


Fig.2. Algorithm for training and employing the surface roughness model

3. ANFIS SURFACE ROUGHNESS MODEL

The aim of this study is to develop an accurate and reliable model for predicting surface roughness during end milling process. The surface roughness prediction model is built according to the ANFIS method. The ANFIS method seeks to provide a linguistic model for the prediction of surface roughness from the knowledge embedded in the trained neural network. By given input/output data set, the ANFIS method constructs a fuzzy inference system (FIS) whose membership function parameters are tuned (adjusted) using a backpropagation algorithm. This allows fuzzy systems to learn from the data they are modeling. FIS Structure is a network-type structure similar to that of a neural network, which maps inputs through input membership functions and associated parameters, and then through output membership functions and associated parameters to outputs. Fig. 2 shows the basic flow chart for predicting the surface roughness via ANFIS.

Five steps are required to develop an ANFIS system. In step 1, the training and testing data are loaded to the system. The process variables are force sensor readings (F), spindle speed (n), feed rate (f), depth of cutting (A_D/R_D), and measured surface roughness. All the data were scaled. The whole data set is divided into the training and the testing set. 350 data points were used in this study. The training data set is used to find the initial premise parameters for the membership functions by equally spacing each of the membership functions. A threshold value for the error between the actual and desired output is determined.

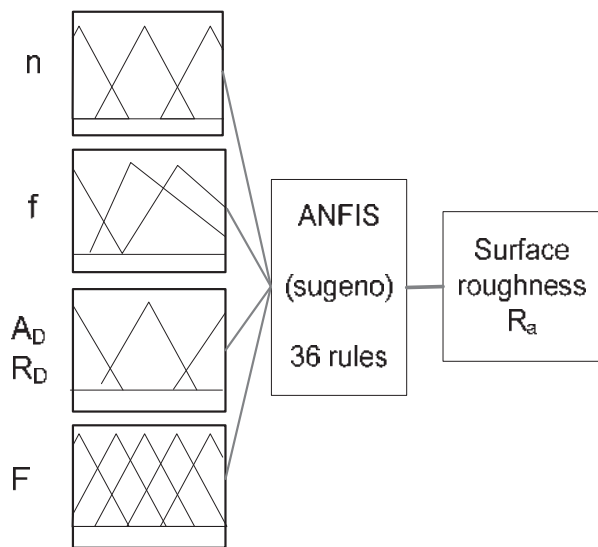


Fig.3. Architecture of ANFIS surface roughness model

The FIS architecture and training parameters were defined in step 2. The optimization method, the tolerance error, the maximal number of epoch, the number of membership functions and the membership functions types are defined. The ANFIS architecture is explained in detail in [5].

The fuzzy inference system under consideration has 5 inputs and one output. The inputs are cutting force sensor signal, and the cutting conditions. The output is surface roughness provided by the surface roughness tester. In

step 3, the training phase is accomplished. With the input–output data, the neuro-fuzzy algorithm is trained, and the unknown parameters are identified.

Fig. 3 shows the inputs, membership functions, and the fuzzy inference system for surface roughness prediction. During the training stage, the ANFIS adjusts its internal structure to give correct output results according to the input features. The process is terminated when the error becomes less than the threshold value.

During training in ANFIS, 350 sets of experimental data are used to conduct 500 cycles of learning. Finally, in the fourth step the trained ANFIS is used to predict surface roughness. After the training, the inference system could estimate surface roughness from cutting force measurement and cutting conditions in real time.

The developed ANFIS model can guide system or operator in tool change decisions making.

4. EXPERIMENTAL SET-UP

Experiments were performed on a CNC machining platform Heller with FAGOR CNC controller. Material Ck 45 and Ck 45 (XM) with improved machining properties were used for tests.

The solid end milling cutter (R216.24-16050 IAK32P) with four cutting edges, of 16 mm diameter and 10° helix angle was selected for machining.

The corner radius of the cutter is 4 mm. The cutter is made of ultra-fine carbide grade coated with TiN/TiCN coating.

The coolant RENU S FFM was used for cooling. The cutting forces were measured with a piezoelectric dynamometer (Kistler 9255) mounted between the workpiece and the machining table.

The data acquisition package used was LabVIEW. The surface roughness was measured by 7061 MarSurf PS1 Surface Roughness Tester. The experimental set up can be seen in Fig. 4.

The experiments were carried out for all combinations of the chosen parameters [6], which are radial/axial depth of cut, feedrate, spindle speed and tool wear. Other parameters such as tool diameter, rake angle, etc. are kept constant. Three values for the radial/axial depth of cut have been selected for use in the experiments: $R_{D1} = 1d$, $R_{D2} = 0.5d$, $R_{D3} = 0.25d$; $A_{D1} = 2\text{mm}$, $A_{D2} = 4\text{mm}$, $A_{D3} = 8\text{mm}$; d - cutting parameter (16 mm). In the experiments the following values for feedrate and spindle speed were varied in the ranges from 0.05-0.6 mm/tooth and 125 –350 min^{-1} , respectively. In this way two sets of data groups were generated, one for training and other for testing.

The cutting force signals were monitored by using a fast data acquisition card (National Instruments NI 9215 A) and software written with the National Instruments CVI programming package..

5. RESULTS AND DISCUSSION

Predicted surface roughness was found significantly sensitive to the measured cutting forces, especially the thrust cutting component. A total of 250 sets of data were

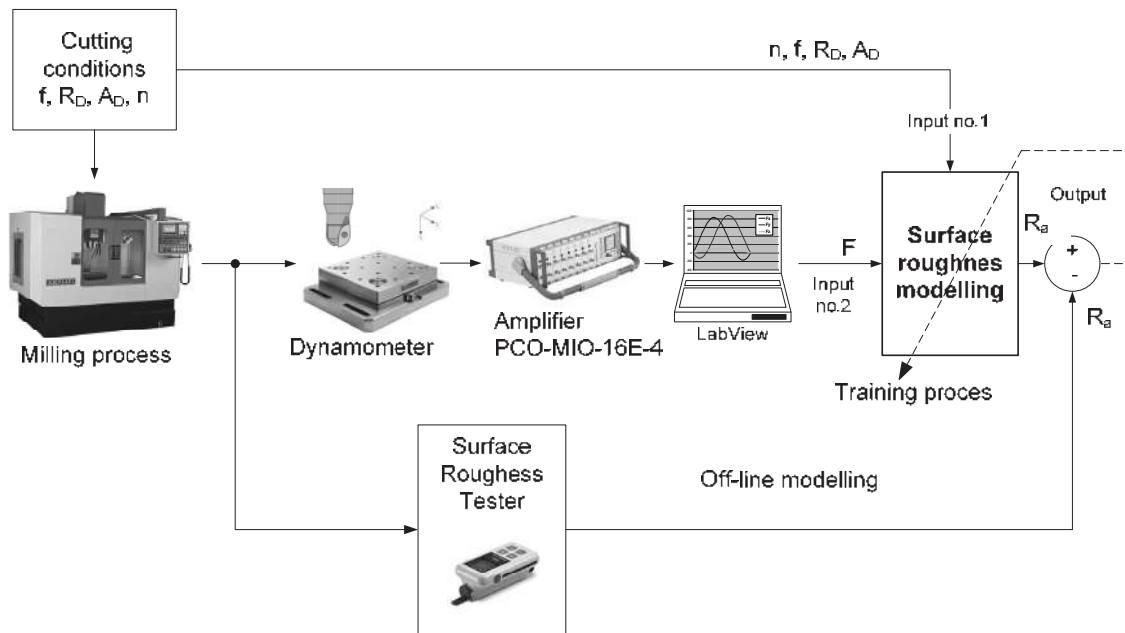


Fig. 4. Experimental set-up for surface roughness modeling

selected from the total of 350 sets obtained in the end milling experiments [5] for the purpose of training in ANFIS. The other 100 sets were then used for testing after the training was completed to verify the accuracy of the predicted values of surface roughness. The best results were obtained when triangular membership functions were chosen for the neuro-fuzzy model.

The average error of the prediction of surface roughness is around 2.9% when triangular membership function is used in ANFIS. The accuracy is as high as 96%. The training was very fast, and the error reached a constant value after about 45 epochs. In this case, there were 36 rules in the fuzzy inference system.

The system with incorporated ANFIS model was capable of predicting the surface roughness in real time.

The accuracy of the training data was 98.1%, and the accuracy of the testing data was 96 %. Wrong predictions accrued when the feed rate and rotational speed were low. This issue can change the cutting forces and affect the tool wear estimation via force signals. The force sensor gives a good estimation of the surface roughness.

5. CONCLUSION

The purpose of this research was to integrate the surface roughness prediction model into the control system aimed at controlling the cutting force by digital adaptation of cutting parameters. The presented ANFIS model predicts surface roughness with 96% accuracy. It is important part of cutting force control system. Surface roughness was measured after each cutting test with a Surface roughness tester. Comparison between the actual surface roughness and the simulated results from the neuro-fuzzy method showed good agreement. Once the ANFIS model was trained the surface roughness can be displayed to the operator on the control panel. In this way it provides additional level of process monitoring.

The efficiency of the control system with integrated ANFIS model is reflected in improved surface quality.

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CLAMPING AND SUSPEND SYSTEMS TO MANIPULATIONS DOCKING RAMPS

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Abstract: This paper presents a comparative analysis of two proposed solutions to achieve the handling and transport platforms docking. The proposed solutions are analyzed by finite element method and validated by laboratory experiments. The solution adopted will be chosen according to the chosen transport equipment.

Key words: finite element, lifting eyes, safety factor

1. INTRODUCTION

Any equipment, any construction element must be equipped with one or more devices suitable for transportation. Only in this way can during each stage of production, assembly, delivery (by train or truck), final assembly at the customer, speed ferry optimal conditions with maximum safety for staff and fixture construction. Attachment points - an absolute necessity that does not give importance to many builders.

In the next table (Table 1) shows the main mounting schemes of different assemblies depending on their weight and shape [1].

Table 1. The main mounting schemes

Scheme	Number of arms	β [°]	Scheme	Number of arms	β [°]
	1	0		2	0-45
	2	0		3-4	45-60
	1	90		3-4	0-45
	2	90		3-4	45-60

Means used to loading - unloading, transshipment goods transported adapters are selected according to the following criteria [2]:

- nature of the goods to be transported (solids or liquids, acids or bases, with or without risk of damage;
- shape and size of elements (in bulk chunks or small, heavy or bulky granular or powdery state, etc.);
- of packing (bags, packages, bottles, containers, etc.);
- how to place the deposit (in piles, stacks, palletized or not, etc.);
- type of means of transport to be achieved;
- the quantity of goods handled per unit time.

Frequently transported objects are equipped with threaded holes for all screws ring DIN. When you want to use very resistant screw attachment points, these holes are usually oversized and require the use of fixings too high [3].

Accessories and devices for catching and suspending pregnancy are built in a variety constructive handled by load characteristics and handling mechanism. Here are some examples (Figure 1) [4]:

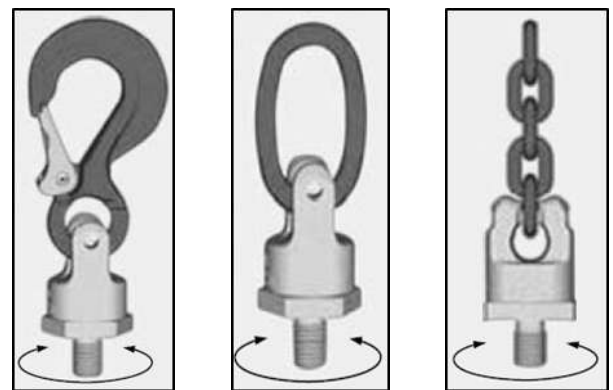


Fig.1. Lifting eyes models

Most often using screws DIN 580 ring of C15 or predict construction sheet which do not allow the possibility of load angle respectively are usually oversized so that smaller hooks opening and flap safety cannot be caught. Docking ramp handling the manufacture, transport, installation and facilities according to of the economic agent can be done using overhead cranes (Figure 2) or a forklift (Figure 3).

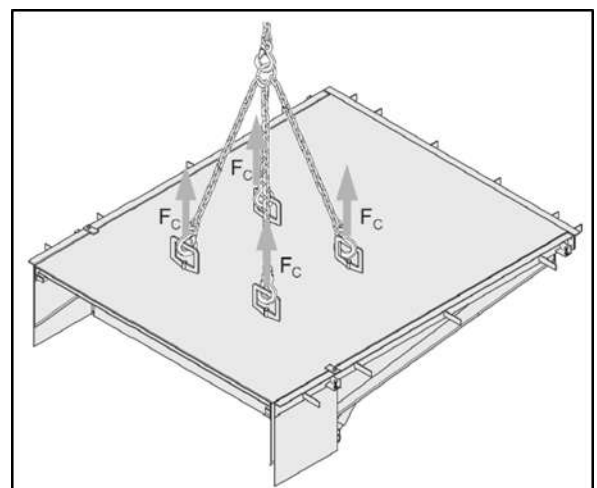


Fig.3. Transport case-using a crane

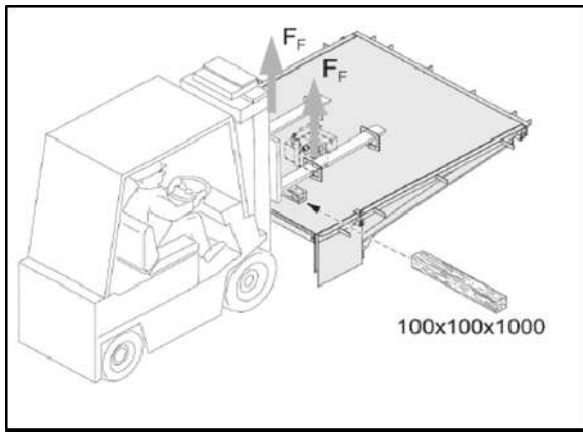


Fig.4. Transport case-using a lifting

In this case, fixing and fastening systems have an important role due primarily to ensure the safety of workers and secondly is affected finished product quality by eliminating non-conformities.

In both cases should be considered gauge assembly to handle its geometrical shape and the binding mode of high load.

Loading gauge assembly docking ramp is different, depending on the type of ramp is designed and respectively its additional facilities [5, 6, 7].

Fixing and fastening system also has a very important role because it influences the production costs of both the manufacturing company and the beneficiary dock ramps.

2. THE PROPOSED CONSTRUCTIVE SOLUTIONS FOR FASTENING SYSTEMS AND HANDLING OF DOCKING RAMPS

Given the above, we propose finding a technical solution to solve the problems faced by the manufacturer of the docking ramps manufacturer.

Like the initial starting date, we will consider two aspects:

- weights and dimensions to the docking ramps transported;
- the transport equipment used.

The main types of docking ramps carried out the following sizes shown in Table 2.

Table 2. The main types of docking ramps

Type	Mass
SD 610 2500x2000x700	780
SD 610 3000x2000x700	870
TD 620 2500x2000x700	950
TD 620 3000x2000x800	1250
LHT 100 kN 3000x2000x800	1750

Special problems are caused by handling and transport using fork, given the typical geometric constructive and its forks, presented suggestive in Figure 5.

For the two types of equipment used to transport, we consider two constructive solutions for fixing and manipulating dock ramps, shown in Figure 6 and Figure 7.

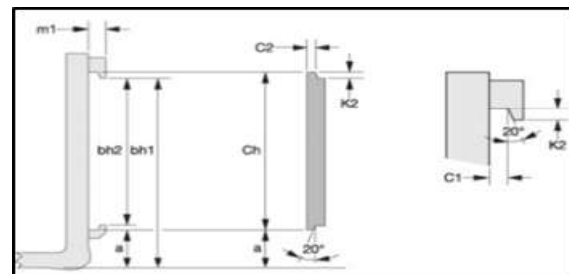
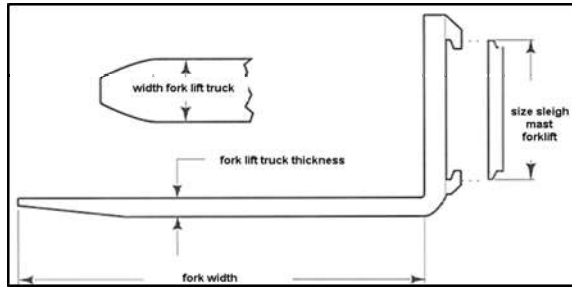


Fig.5. Fork lift truck – geometrical parameters

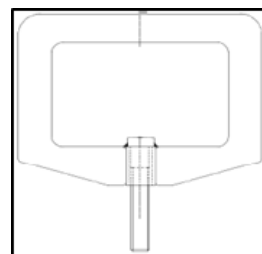


Fig.6. Fork lift truck-variant 1 – geometrical parameters

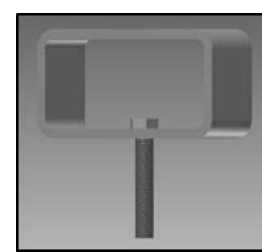
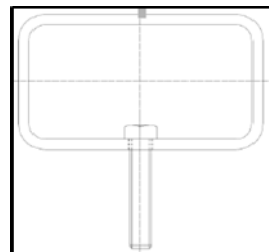


Fig.7. Fork lift truck-variant 2 – geometrical parameters

Axial force an a single lifting eye, according to the charts, when lifting a leveler with load G of charge for two types of transport used data handling relations:

- *using crane*: in this situation, for a single lifting eyes action load $\frac{G}{4}$ and is $\beta = 45^\circ$:

$$F_c = \frac{G}{4} \sin \beta = 0,176G \quad (1)$$

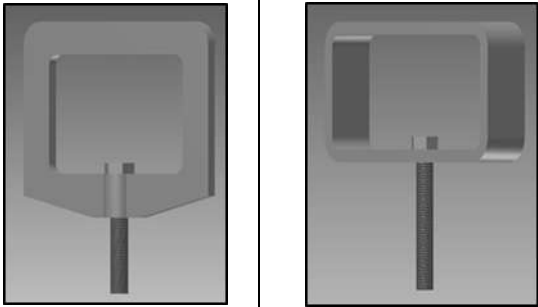
- *using forklift*: in the worst scenario, when lifting only with two lifting eyes, axial force have value:

$$F_F = 0,75G \quad (2)$$

3. ANALYSIS OF THE TWO PROPOSED SOLUTIONS

Based on the two proposed solutions, but also from the two means of handling and transport used make a pertinent analysis on their behavior in exploitation, analysis summarized in Table 3.

Table 3. Maximum leveler weight

					
Using a crane					
Safety factor	S_F	2	Safety factor	S_F	2
Yielding point load	G_Y [kg]	2000	Yielding point load	G_Y [kg]	1200
Maximum axial load	G_A [kg]	1000	Maximum axial load	G_A [kg]	600
Maximum leveler weight	G_{max} [kg]	5800	Maximum leveler weight	G_{max} [kg]	3500
Using a forklift					
Safety factor	S_F	2	Safety factor	S_F	2
Yielding point load	G_Y [kg]	2000	Yielding point load	G_Y [kg]	1200
Maximum axial load	G_A [kg]	1000	Maximum axial load	G_A [kg]	600
Maximum leveler weight	G_{max} [kg]	1333	Maximum leveler weight	G_{max} [kg]	800

Using machines tried to the traction have tried two fasteners and lifting proposed to square axial force-strain diagram, Figure 8 and Figure 9.

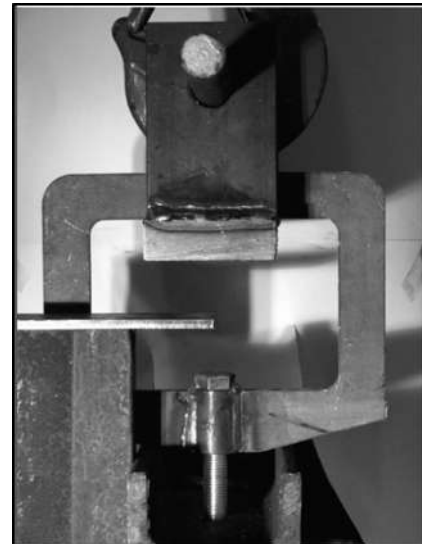


Fig.8. Fork lift truck – geometrical parameters

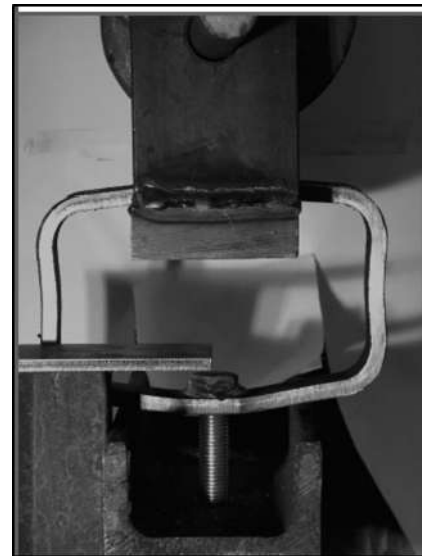


Fig.9. Fork lift truck – geometrical parameters

In order to take into account the test conditions and fastening systems proposed lifting was carried out a simulation using ANSYS Mechanical software package Strength Analysis.

It has been observed that both the elastic and plastic behavior of real fasteners and lifting is approximately equal studied virtual behavior.

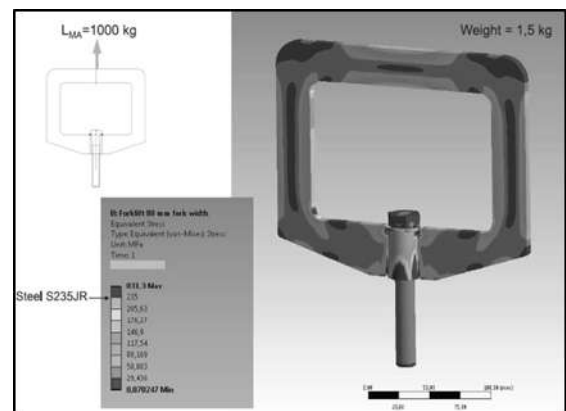


Fig.10. Virtual model 1 – FEA

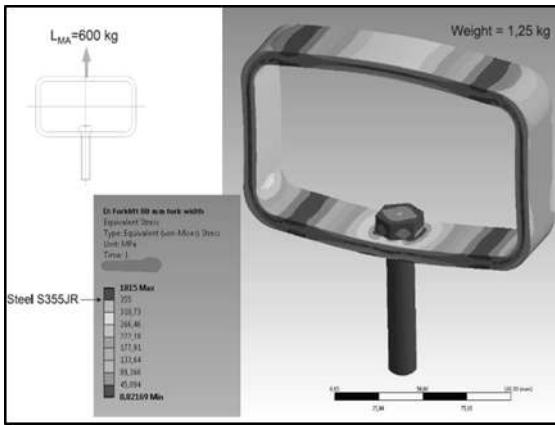
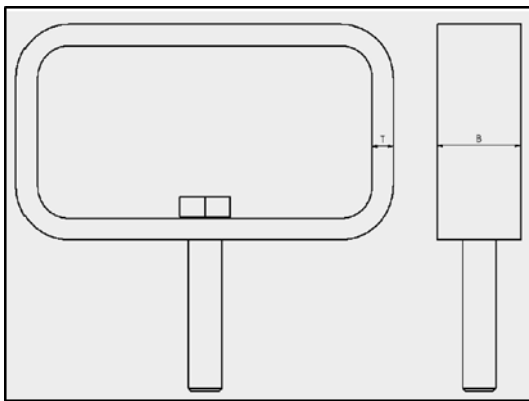


Fig.11. Virtual model 2 - FEA

Among the proposed option 1 (Figure 6) and version 2 (Figure 7) observed that the two provides greater resistance resulting in further dimensional rethinking of this embodiment, Figure 12.



1-BxT-35x8	4-BxT-40x8	7-BxT-50x8
2-BxT-35x10	5-BxT-40x10	8-BxT-50x10
3-BxT-35x12	6-BxT-40x12	9-BxT-50x12

Fig.12. Dimensional variations considered

For the nine proposed dimensional variations were performed tensile testing, resulting graph in the figure below (Figure 13).

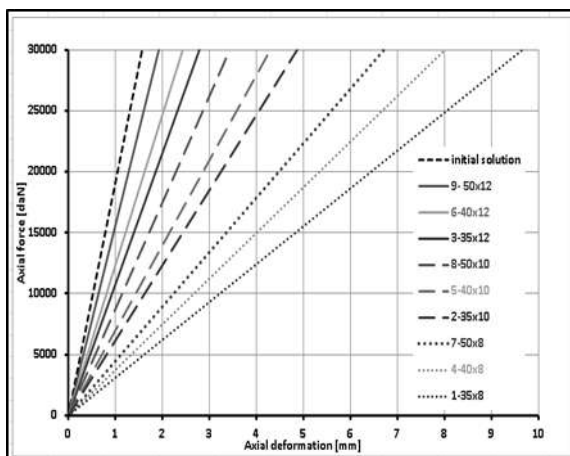


Fig.13. Graph axial deformation-axial force

4. RESULTS AND CONCLUSIONS

While contribute through the implementation of attachment point, which provides safety to:

- minimize the risk of accidents!
- preventing damage and downtime!
- reducing transshipment times!

After analyzing virtual model proposed were identified following:

- increasing width from 35 to 40 mm to increase the resistance factor of 1.1, and an increase in the width of 40 to 50 mm, the same resistance factor from 1.4 increases.
- increasing the thickness of 8 to 10 mm to increase the resistance factor of 2, and an increase in the thickness of 10 to 12 mm, the same resistance factor increased to 3.4.

The optimal variant is the variant 5 (40 x 12), increasing the resistance coefficient of 3.9.

Compared to the version originally proposed, the new version has the following advantages: weight without screw is 1.8 kg and unit cost it is 2.8 euro (initial cost was 4 euro).

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EXTERNAL FACTORS INFLUENCE ON THE METAL COFFERDAM WALLS PROTECTING RIVERS IN CASE OF NATURAL DISASTERS

Silviu Răzvan AVRAM

Abstract: The purpose of this study is the tension and deformation into the metal cofferdam modules used to protect the rivers banks during natural disasters.

This study has a special applicability mainly in the areal Timis & Caras-Severin counties on the Barzava and Timis rivers often affected by natural phenomena.

In this respect, it was considered as opportune to be studied the execution and the technical problems which could happen when using metallic cofferdams as a protective element.

Applications are variable including not only bank protection but also, damming rivers and settlements, lakeside and riverside promenade protection, CCA.

Key words: cofferdam, floods, flow, flow rate, walls

1. INTRODUCTION

The purpose of this study is the tension and deformation into the metal cofferdam modules used to protect the rivers banks during natural disasters.

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Applications are variable including not only bank protection but also, damming rivers and settlements, lakeside and riverside promenade protection, CCA.

2. THE PURPOSE OF THE STUDY

After set by calculating the metallic cofferdam variant (dimensions, checkout, strength), the river water as testing element, changing so the classic method of

riverbanks defense, it was started to study on the realized prototype.

So, after prototype was done in a modular light construction.

It was studied the external factors influence on the metal cofferdam walls in real operating conditions in different working regimes.

For this it was determined the probable maximum flood on Timis river where we intend to implement the program.

We took into consideration the following characteristics:

- To choose the distribution model of „possible maximum rain”
- To calculate the PMP hydrography
- To determine the maximum possible rainfall and to analyze the distribution in time

It is used the statistic method, its algorithm being based on general formula of hydrologic figures frequency analyze proposed by Ghow-Hershfield

$$X_{p\%} = x_n + k s_n$$

$X_{p\%}$ - layer rain corresponding to probability $p\%$

x_n - the average of „n” annual maximal values

s_n - variant of annual maximal values

k - variable statistic coefficient

If instead of $X_{p\%}$ is used x_m = the maximum value observed and instead of k is used k_m , for each of annual maximum string rainfall for a period of : 1 h, 3 h, 6 h, 24 h, we calculate the k_m using:

$$K_m = (x_m - x_n / s_n)$$

In which x_n and s_n consider the maximum values without the biggest registered value.

In fact, we will determine the envelope of k_m , using all the available information for basin of Timis.

It results the calculus formula is :

$$PMP = x_{24h} (1 + c_v k_m)$$

Maximum possible flood calculation VMP is done taking into consideration as basis a system in which inlet is the rain histogram and the outlet flood hydrograph.

The spill formation will be examined in 2 steps:

- Gross rainfall transformation in net rainfall
- Net rainfall transformation in spill hydrograph

We got the following values for the river of Timis



Fig.1. Metal cofferdam



Fig.2. Metal cofferdam



Fig.3. Metal cofferdam



Fig.4. Metal cofferdam

Table 1.

River Segment	F (km ²)	Maxim flows with probability of overtaking (m ³ /s)						Q _{MVP} 12 hrs	Q _{MVP} 24 hrs
		5%	1%	0,8%	0,1%	0,01%	0,001%		
Timis	255	125	250	310	455	670	860	680	880

Table 2.

No.	No of test [pc]	Water speed [m/sec]	River flow [m ³ /sec]	Walls deformations [mm]
1	10	1.0-normal	380	0.133
2	10	1.0	500	0.162
3	10	1.0	680	0.183
4	25	1.25	500	0.304
5	25	1.25	600	0.312
6	25	1.25	680	0.430
7	25	1.50	810	0.95
8	25	1.50	880	0.98
9	25	1.50	1000	1.17
10	10	2.0	1185	1.22

The results are acc. to the maximum flow values obtained through extrapolation of theoretic probability diagram.

Maximum flow calculated for the river Timis - $Q_{\max} = 1185 \text{ m}^3/\text{s}$.

Taking into consideration the above as a calculus base we continue with technical analysis and the study of the effective deformations of the cofferdams walls.

3. CONCLUSION

After tests did in different conditions of water flow and speed we reached to the following conclusion:

- Even the metal cofferdams protecting rivers banks in case of natural disasters are light structures interchangeable and module ones it is recommended a plate wall thickness of minimum 1,25 mm with reinforcements on length.

Another constructive solution in flood conditions, then in extreme conditions, unpredictable has 2 variants :

- A spur of breaking and tranquility waves, respective decreasing water
- Two spur of breaking waves

When can be used the constructive variant of 1 mm thickness with reinforcements of square pipe 25x25x2 mm on cofferdam length.

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INNOVATIVE, SAFE AND COST-EFFICIENT LIGHT-WEIGHTING SOLUTIONS IN THE AUTOMOTIVE WHEEL MANUFACTURING

Imre KISS
Vasile George CIOATA

Abstract: *In automotive wheel design, the balance between strength and weight is critical: too heavy and the car will lose performance, too weak and there is a risk of structural failure. When constructing new light alloy wheels, particular attention is always given to one thing: achieving the perfect balance between weight and stability, which is known as the strength-to-weight ratio. In general, the more stability offered by a tire means the heavier it would be. In the recent days, considerable efforts are being made to reduce the weight of the components, which ultimately reduces the overall weight of the vehicle. In order to sustain all loads and practical conditions wheel materials must be of good quality. In the case of heavy loading condition, steel wheels are preferred and for medium and low load condition, aluminum and magnesium alloy wheels are suggested essential for aesthetic look. Wide varieties of materials are available in the market can be used for wheel rims. Generally used wheel rim materials are aluminum alloy, magnesium alloy and steel. Each material has some advantages over the other. Thanks to the potential weight saving, aluminum castings are good candidates for automotive wheels.*

Key words: *wheel, light alloy, weight saving, low-pressure die-casting*

1. INTRODUCTION

Today, in order to optimize the resources usage and reduce the air pollution, the automobile industry is facing new challenges, with the necessity to improve engines fuel economy, enhance vehicles autonomy and reduce the CO₂ emission. One of the solution, which is being much researched, is the car components weight reduction. There is a range of new materials that have been developed to attend the new weight standards. [7,9,10] Together with lightweight these materials must also deliver acceptable

mechanical properties, easy to manufacture and to assembly capability, good appearance, high durability and good cost-benefit relation. [7,9,10]

The automotive industry continues to look for opportunities to reduce weight and cost while continually increasing performance and safety. Light weighting technologies enhance vehicle performance, (fuel economy, acceleration, braking and emissions). New materials are available to reduce weight; however the incremental cost for the weight reduction can be prohibitive. Vehicle manufacturers recognize removing weight is a key feature to meeting their targets for fuel economy and emission reductions. [7,9-10]

The producers have taken the global lead in the light weighting of passenger cars with aluminum. Europe is the leader for automotive aluminum body applications, in chassis and suspension, bumper systems, wheels, etc., to name just a few. Wheels are now representing about 15% of the average aluminum content in passenger cars and light trucks, and if the main motivation has been styling with mainly cast solutions, weight reduction requirements have led to the development of more technical cast but also forged and fabricated solutions. [1,2,8] Aluminum penetration in wheels was in the year 2000 for European vehicles about 30 to 35%, compared to largely more than 50% in USA and Japan. Now, more than ever, reducing vehicle mass is necessary and light weighting materials, like aluminum and magnesium, can be used to further improve the sustainability and the safety of future generations of cars. [1,2] Aluminum castings, extrusions, forgings and sheets can be found nearly everywhere, including in car bodies, closures, chassis, suspensions and wheels. These light-weighting solutions can be applied with little adaptation across all car models. [3-5,6,9-10]

Innovative, safe and cost-efficient lightweight aluminum bonnets, fenders, doors and bumpers can be found across all car models today. For these parts, switching to aluminum is relatively easy and does not need full re-engineering of the car. In practice, material substitution is generally connected to a model change where there is extensive re-designing in any case. Mixed material design does not present bigger problems, if appropriate design and manufacturing measures are taken. Thus, the weight saving potential could even be significantly greater. [7]

It is important to distinguish between two kinds of weight savings:

- Direct weight saving: the weight reduction due to exchanging a heavier material for a lighter material in one or several components.
- Indirect weight saving: additional weight reduction obtained by downsizing certain components (i.e. brakes, suspension, engine, etc.) to keep vehicle performances at the same level as before. Indirect weight savings come on top of direct weight savings, and can represent up to 50% additional savings on the weight of the complete car.

As indirect weight savings can only be maximized by completely redesigning cars, they are usually lower than 50% and close to 0% when only one single component is made lighter. Weight reduction without any changes to the functionality or safety of the car can be realized not only by replacing heavier materials with lighter materials, but also by introducing new design and manufacturing

concepts or, ideally, by a combination of these measures. It is important to realize that aluminum offers the possibility for light weighting without downsizing and without impairing the safety of the vehicle fleet. Light weighting improves safety for all road users, as it facilitates vehicle handling, shortens braking distances and reduces the severity of accidents. [7,9-10] Therefore, reducing mass of future generations of cars has additional benefits, like:

- keeping constant the car acceleration performance, which is generally reflected by the power-to-weight ratio, saving weight allows downsizing of the power train (engine, transmission, etc.) and thus provides additional weight savings. Keeping the power train unmodified, reduced weight increases the power-to-weight ratio and therefore improves acceleration.
- keeping braking power constant, light weighting shortens braking distance. Keeping the braking performance constant, light weighting
- allows downsizing of the brakes, which offers further weight saving potential.
- saving weight on unsuspended parts like wheels does not only increase a car's comfort, but also reduces the hammering effect on uneven roads, therefore reducing wear of the roads.
- road handling is improved by light-weighting in many different ways:
 - Handling of a lighter car is easier in demanding driving situations.
 - Reducing body weight lowers the center of gravity, improving the car's stability and reducing the risk of rollover.

The relationship between the material properties and the strength, stiffness and weight of a component is very complex and can be strongly influenced by the part geometry, meaning that there is no absolute rule. In practice, it will be necessary to consider the design of each component individually to determine the actual weight reduction potential. [3-5,7,9-10]

2. WHEELS, IN TERMS OF CONSTRUCTION

Automobile wheel is an integral component of each contemporary automobile. The task of an auto wheel is to move a vehicle on a road and to change the direction of the movement. [2,3,10]

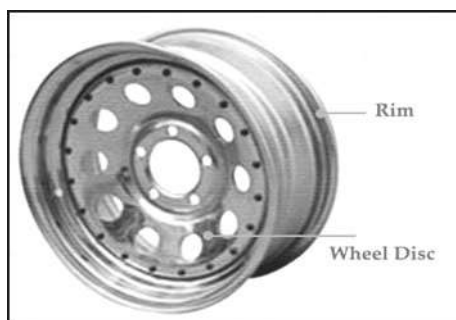


Fig.1. Wheels, in terms of construction

Every automotive wheel consists of a rim and a disc. In terms of construction, a wheel rim is made as a one-,

two- and three-piece auto part', each type having its place in the today's automotive market. Most alloy wheels are made as one-piece product, where a wheel is made in a mold as a single piece. This type of rims is widely used due to its affordability and strength of the structure.



Fig.2. One-piece rim

As for two-pieces rim (also called modulars), the wheel is manufactured of center and outer barrel. Then these separate parts are welded or bolted together and their construction makes for a very strong and lightweight wheel. This type of wheels has more options in sizes. The main disadvantage is the assembling process. They are also cheap to produce so they make up the majority of alloys for regular cars.



Fig.3. Two-pieces rim

As for three-piece rim, the wheel is manufactured with an assembled inside rim half, outside rim half and a centerpiece. Three-piece wheels are offered in a great variety of sizes and offsets.

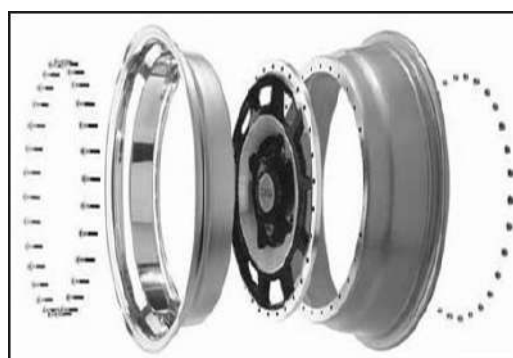


Fig.4. Three-piece rim

3. WHEEL MANUFACTURING MATERIALS

The alloys employed have to meet a range of sometimes–conflicting requirements: [7,9,10]

- good metal–mold casting properties (castability, susceptibility to hot tearing and shrinkage characteristics)
- ability to withstand physical impact (elongation and impact strength)
- corrosion resistance (normal and saline atmospheres)
- fatigue resistance

These requirements have led to the use of proper light–weighting alloys in the automotive industry.

Car rims can be made from several materials. Each material used in a car rim can affect its weight, strength, durability and finish. Quality materials will also drive up the cost of a rim.

Depending on the material, used in wheel rim production, there are several most common rim’s types available in the contemporary automobile market:

- Steel wheel rim. It is a standard type of a rim. Due to its properties, steel is one of the most important engineering and construction material. Steel features great formability, impermeability and durability. Steel is a cost–effective material as it is less exotic material and its manufacturing process is not so complicated. Compared to other materials, one of the main disadvantages of steel is its heavy weight.
- Alloy wheel rim. It is manufactured from an alloy of magnesium or aluminum. One of the main benefits is the lightweight that results in better handling. The alloy wheels reduce the overall weight that in its turn lead to fuel saving. Compared to steel wheels, alloy rims provide better heat conduction. This improves braking performance as the heat is dissipated from the vehicle’s brakes. It also reduce a possibility of overheating. The disadvantage of the alloy rims includes inability to withstand corrosion. It is necessary to apply a paint layer or other coatings to prevent rust. Higher price is another disadvantage of this material. Aluminum alloy wheels are the most popular material used in car rims.
- Carbon fiber wheel rim. Compared to alloy and steel wheels, this type of rims is less popular as its major disadvantage is high price. Carbon fiber composites have high strength–to–weight ratio, toughness, resilience and great fatigue strength. Due to its advantages, carbon–fiber–reinforced polymer is used in manufacturing of supercars. Along with the reduced fuel consumption and longer service life, this type of wheels benefits to performance as well as handling. They are extremely light, less than half that of a steel rim of the same capacity. They are also extremely expensive due to initial cost of raw material and the lot of manual work involved.

Magnesium alloy wheels, considering their high costs, optimum mechanical characteristics and lightness, are produced exclusively for competition. Most common are steel and aluminum alloy, exotic materials like carbon fibers being way out of financial reach of most ofroaders. Most genuine wheels are made of steel, but up to 1/3th lighter than steel and not too expensive, aluminum alloy are good for almost all applications.

The main advantages, when compared alloy wheels with steel wheels are:

- a high styling versatility
- weight (equal or less than steel without styling)
- dimensional accuracy (mass distribution)
- recycling ability
- static and dynamic behavior

Car wheels, also known as rims come in all sizes, designs and materials. The type of metal materials used in car wheels are various, most auto manufacturers now using alloy wheels—a combination of metals—aluminum and magnesium or both.[10]

Alloy wheels are more expensive to produce than standard steel wheels, and thus are often not included as standard equipment, instead being marketed as optional add–ons. However, alloy wheels have become considerably more common since 2000, now being offered on economy and subcompact cars, compared to a decade earlier where alloy wheels were often not factory options on inexpensive vehicles. Alloy wheels have long been included as standard equipment on higher–priced luxury or sports cars, with larger–sized or “exclusive” alloy wheels being options.

Aluminum alloy wheels are less expensive than alloy magnesium wheels but both are now popular choices due to their durability. In addition, whether magnesium or aluminum alloy wheels are choose, both tend to be lighter than the older steel material used in regular steel wheels. Alloy rims make vehicle handling more responsive including braking.

Alloys of aluminum or magnesium are typically lighter for the same strength, provide better heat conduction, and often produce improved cosmetic appearance over steel wheels. Better heat conduction can help dissipate heat from the brakes, which improves braking performance in more demanding driving conditions and reduces the chance of reduced brake performance or even failure due to overheating.

Alloy wheels are usually made of aluminum or magnesium alloys. The advantages of alloy wheel are lightweight, better heat conduction and excellent aesthetic appearance. In the recent technological advancement, considerable attempts are being made to develop Al and Mg alloys to suit to requirement for alloy wheel. Further weight reduction of the wheel hub may be realized by design. Weight reduction is one of the essential criteria of a vesicle as far as the fuel efficiency is concerned. Use of lightweight metals and alloys is now becoming the designer’s choice. [7,10] The existing components, which are made of steel and cast iron, are slowly phased out by replacing with aluminum and magnesium alloys having comparable properties. In fact, metals used in the production of light alloy wheels are aluminum and magnesium alloys. They are non–ferrous metals with physical and mechanical features presenting small quantities of binding elements, such as silicon, copper, manganese, zinc, nickel and titanium.

4. MANUFACTURING PROCESSES

A wheel manufacturing is a complex work that integrates a various production technologies and equipment. In wheel production, in use are two metal processes: casting

and forging. [1,5,6,8,10] Depending on the wheel structure and applied materials, the following types of wheel production are used:

- casting – is a solidification process by which molten metal (aluminum or magnesium alloy) is poured into a mold with the required shape's cavity where the solidification process takes place. Casting has the advantage of allowing implementing various ideas in terms of style. Until recently, gravity casting was widely used by the wheel manufacturers. At present, this technology is replaced by low-pressure die-casting. This technique is aimed at reducing porosity, which is among the main disadvantages of casting.
- forging – is a manufacturing process by which metal is heated up to the required temperature, placed in a certain die and then the compressive force is applied. This manufacturing process eliminates the porosity and avoids that may cause the corrosion and cracks. The limited variation of shapes and high production costs – due to the complex manufacturing technology – lead to the increased price of the wheels.

The major casting processes for wheels are: [1,10]

- low-pressure die casting (mainly)
- gravity permanent mold casting (less used)
- squeeze-casting process (marginally used)

Forged alloy wheels are one-piece wheels formed from a single block of metal by hot forging, hot or cold spinning and machining operations. The forging process permits flexibility in design of the styled disk, similar to cast wheels.

In relation to castings, forged materials exhibit decidedly higher fatigue resistance due to absence of pores and because of a fine, homogeneous microstructure. While cast wheels are performing according to the same load and endurance specifications as forged wheels, the latter are more tolerant to overloads as may be experienced in sports cars.

5. LOW-PRESSURE DIE CASTING OF MODERN ALLOY WHEELS

Magnesium wheels were the first die-cast wheels produced. Magnesium wheels were originally used for racing, but their popularity during the 1960s led to the development of other die-cast wheels, made from any material, from modern aluminum alloy wheels to plastic and composite wheels. [1,6,8]

Casting is a relatively inexpensive way to produce a high quality, strong alloy wheel and many alloy wheels designed for street use are made this way. The most common process of constructing alloy wheels is the one-piece casting, there are a number of methods to do this, and the most basic is the so-called die-casting, this process is used worldwide throughout the casting industry and is not exclusive to alloy wheels.

Among the most interesting processes, low-pressure die casting is certainly worth mentioning, thanks to its peculiarities, allowing, in several cases, an excellent compromise between quality, costs, productivity, geometrical feasibility. [1,9]

Nowadays, it is adopted for casting aluminum- and magnesium-based alloys. The principle of this process is

quite simple: the permanent die and the filling system are placed over the furnace containing the molten alloy.

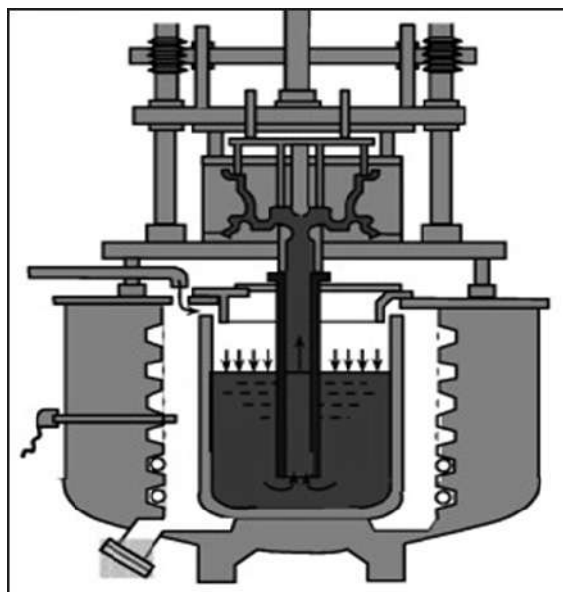


Fig.5. Low pressure die casting of alloy wheels

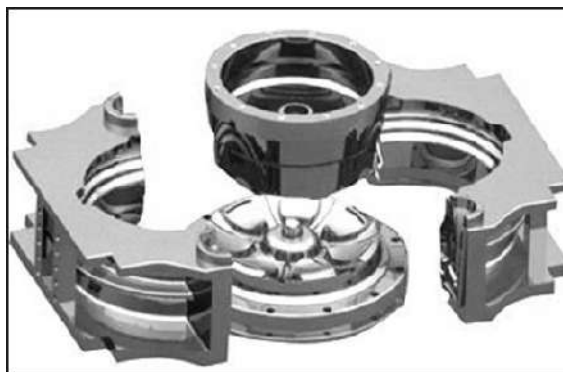


Fig.6. The casting mold

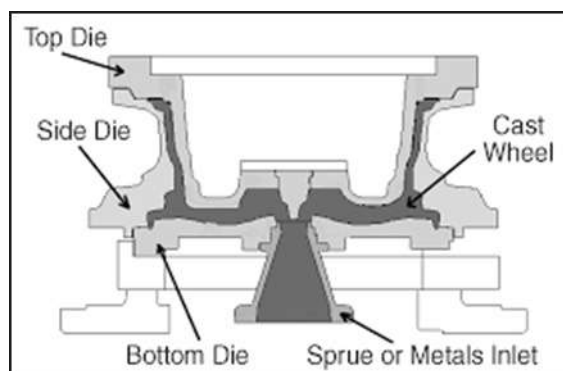


Fig.7. The casting mold and cast wheel

Specific attention has to be done to the design of the die, to control by means of proper cooling circuits, the solidification path of the alloy. The massive region of the casting has to be the last one to solidify and must be placed near to the sprue or metal inlet.

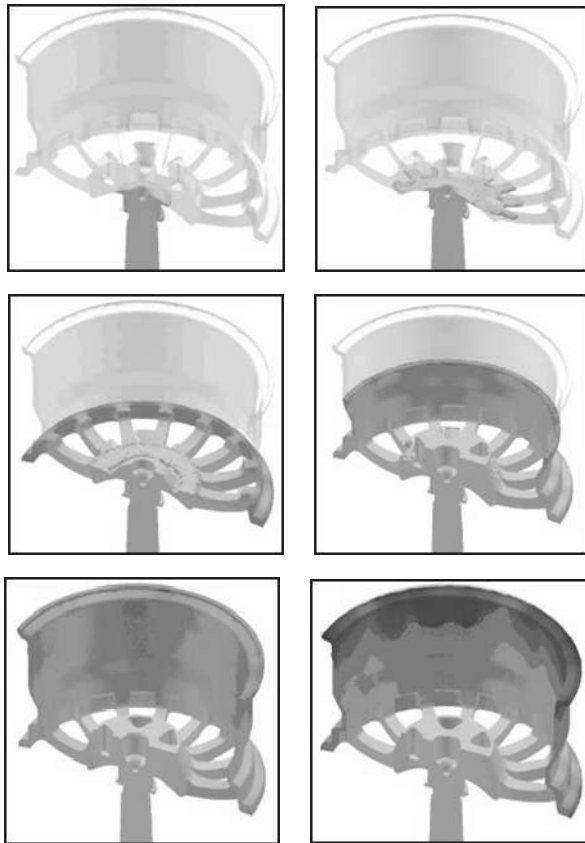


Fig.8. Wheels casting process

The low injection velocity and the relatively high cycle time lead to a good control of the fluid dynamics of the process, avoiding the defects originated by turbulence phenomena.

The die can be design for the production of a single casting or for multiple castings, according to the size required and to the characteristics of the machine.

6. ALUMINUM WHEELS vs. MAGNESIUM WHEELS

Alloys or steels wheel types have advantages and disadvantages for different types of driving; but in general if we want beauty and performance the choice is alloy, and if we want tough, we want steels. While steel wheels are acceptable for use on everyday roads, some drivers prefer the better alternative. In fact, steel wheels are favored because they are tough to damage and are inexpensive, but they are also heavy and clunky looking. Alloy wheels, alternatively, are both better looking and better performing wheels. But, what's best from the alloy's destined for the wheels?

Compare the variety, cost, durability, and handling advantages of aluminum wheels with the weight, heat, and aesthetic advantages of magnesium wheels. Most differences are cosmetically but some are also practical. Unlike steel wheels, aluminum alloy can be cast and worked in many different designs, giving cars a much more individual look, and offering owners the chance to customize even more.

With the many other options to choose from, it is beneficial to have an idea of the benefits and drawbacks

of the two main wheel types: aluminum wheels and magnesium wheels.

Aluminum and magnesium are two metals commonly used in the production of vehicle rims. These metals are abundant in nature, and have similar properties that are used to create efficient and visually appealing rims. Aluminum and magnesium rims are lightweight and increase gas mileage, but they have important differences when it comes to durability and price.

6.1. Focus on aluminum wheels

- a wide range of aluminum wheels are available on the automotive market and they are available from a range of different manufacturers.
- the price of the wheels will vary depending on the size and style, but they are often more costly compared to steel or magnesium wheels. The difference in price usually arises from the difference in the cost to manufacture the wheels.
- compared to magnesium wheels, aluminum wheels are more durable and will last longer than other materials.
- aluminum wheels are stronger than magnesium wheels, with the latter being more delicate and more difficult to manufacture
- they come as standard on some vehicles, making them suitable for regular cars while other types are more suited to specialized cars, such as those used for racing.
- it is possible for the material to be combined with other materials, such as magnesium. This will create alloy wheels that take benefits from each of the materials, which will make them both strong and light.
- using aluminum wheels the handling will often be improved, the car becoming more responsive on operation of the brake and accelerator.

6.2. Focus on magnesium wheels

- magnesium wheels are light wheels, much lighter than steel or aluminum wheels.
- the magnesium alloy might be durable but it does have its limits, and if is don't pay proper attention to the road the investment will losses.
- there's less overheating with magnesium wheels because they spread the heat from the brakes much more efficiently than aluminum wheels.
- the biggest advantage that magnesium wheels offer is in the vehicle's look. Magnesium wheels certainly add a „wow“ factor to a vehicle. That, of course, is the main reason people spend their money on magnesium alloy wheels.
- the main factor weighing against magnesium wheels is the price, they being expensive. Although magnesium wheels do confer some benefits beyond looks, those are relatively minor and not necessarily advantages.

7. CONCLUSION

The automotive industry, in close co-operation with the materials industry, has developed and introduced numerous innovative light-weighting solutions. Key success factors are the product-specific developments, the selection of the proper alloys and the consistent reproduction of the required quality level for the various

product forms (sheets, extruded sections, castings, forgings, etc.). Most important of all is the full exploitation of aluminum-specific design possibilities and the introduction of manufacturing methods highly suited for the forming, machining, assembly and surface finishing of light-weighting components.

Aluminum is the ideal light-weighting material, as it allows a weight saving of up to 50% over competing materials in most applications without compromising safety. Therefore, aluminum light weighting is the enabler of future sustainable cars.

Aluminum is already the dominant material used to make engines and wheels, thanks to the metal's performance, durability and environmental advantages. Automakers are now aggressively applying today's advanced aluminum alloys and designs new light weighting applications as part of an approach to reducing vehicle weight.

Lightweight aluminum designs offer the possibility to improve the safety performance of a vehicle and to reduce simultaneously its mass. The mass-specific energy absorption capacity of aluminum is twice that of mild steel, and compares favorably to the newly developed high strength steel grades. However, car safety is not only a question of the material used. The design applied and the manufacturing concept are even more important.

In order to sustain all loads and practical conditions wheel materials must be of good quality. Wide varieties of materials are available in the market can be used for wheel rims. Generally used wheel rim materials are aluminum alloy, magnesium alloy, and steel. Each material has some advantages over the other.

If original equipment manufacturers require excellent aesthetic shape with very good heat dissipation without compromise with its associated costs then lightweight material such as aluminum and magnesium alloys can be used for wheel rims.

The application of aluminum alloy castings in many mechanical components, especially for cars, has gradually increased in the last years, thanks to the great potential of these materials as replacements for ferrous alloys.

In particular, for those applications in which the necessity of high mechanical properties is combined with the need of a substantial weight saving, aluminum castings seem to be extremely interesting solutions. Moreover, the opportunity of producing cast car components permits a high reduction of the production costs.

The automotive and aluminum industries are working together on reducing the cost of aluminum applications currently used in sport and luxury cars, so that these can also find their place in street going regular cars.

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COMPUTER-AIDED STRIP LAYOUT FOR PIERCING AND CUTTING DIES

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Abstract: Strip layout plan for piercing and cutting dies forms the basis of die design. Designs with classical methods increase waste of time and cause die cost to increase. It is aimed at with this study that strip layout plan for piercing and cutting dies will be formed at computer medium far from user interaction as much as possible. In this way, it is aimed at removing the faults arising from designer and decreasing waste of time lost in the course of design. AutoCAD package program, Visual LISP and VBA (Visual BASIC for Application) programming languages were used in the study made. Strip layout plan at piercing and cutting dies was obtained in a fast, flexible and sensitive way with this study.

Key words: Piercing and cutting dies, computer-aided die design, strip layout plan

1. INTRODUCTION

Calculations, design and production of piercing and cutting dies have an important role in present day's industry. One of the first procedures aimed at by die designer is keeping clearance or waste material percent at strip material to be prepared in accordance with shape and size of the piece to be molded at the least level. Clearance and item number to be molded strip material decreased will increase and production cost will decrease. Item surface area to be produced at one go should be maximum and waste material surface area should be minimum with the object of minimizing strip material clearance. Steel metal forming has gained so much importance in production industry because of the increase in its usage in serial production and led many studies in this area. (1-5). Choi and his friends (1) has made a formulation considering many factors such as availability of standard items, utilisation rate that minimizes material waste for one-way or two-way operations, strip material clearances and design requirements that increase bend resistance.

Parasad and Somasundaram (2) have applied computer-aided die design system for leaving blank in steel metal forming and performed its design. Standard die components data base should be formed with parametric programming that increase the properties of AutoCAD. The biggest advantage of the system implemented is forming the most efficient strip and die layout which contains team configurations information fast. Nye (3) has developed a full and correct logarithm that determines margins for the most efficient material consumption. It is determined at this logarithm that how many pieces will be pressed at one go on the strip, how the piece will be fixed and how the least material rate will be formed. Computer-aided design and production system which was developed for piercing and bending or leaving margin at irregular shaped sheet metal products are introduced in the study made by Chio and his friends (4). Some factors such as information-based system rules, material and thickness of the product, complexity of cutting geometry, cutting profile, availability of press worktable etc. should be considered in system design. Sing ve Sekhon [5] asked three questions in their study to the designer during preparation of strip layout plan. Too many questions increase the possibility of making mistake. Rao has developed a program that designs strip layout at cutting procedures for a sheet metal in the study he made. The program prepared is called "strip layout". Flow diagram of Strip Layout program is shown at Figure 1.

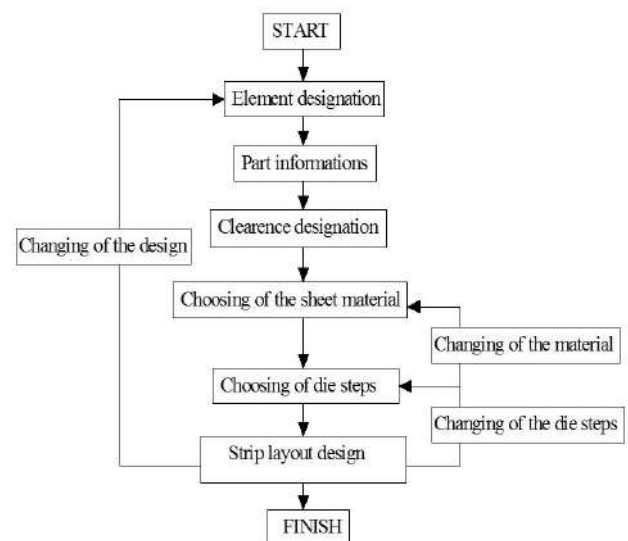


Fig.1. Flow diagram of Strip Layout program

The program prepared perceives the elements on sheet metal item automatically and records to data base that formed in itself. All calculations required for strip layout are determined automatically by the program by using the information recorded. Formulations related to calculations made were used in the program by quoting from the books about sheet metal molding (7-8). Meanwhile, designer is just demanded to select sheet metal material, to determine how many steps strip layout shape and die will consist of. Designer is also asked which procedures

will be performed in which step depending on the number of steps chosen by designer. Designer may not only try different step options and form strip material layout easily, but also may make changes in design of sheet metal product.

2. FORMING STRIP LAYOUT PLAN

It is one of the basic conditions of correct and efficient production to transfer shape and sizes of the item to be molded to strip material. Transferring the most appropriate one of a few strip layouts to be prepared for sheet metal to be molded to strip material is one the most essential steps.

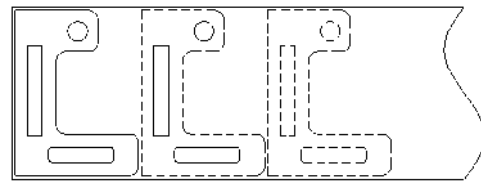
This study aims at producing computer-aided strip layout plan, and the most fundamental property of this study is having no user intervention except compulsory options (material, step number and strip layout ways). Strip material layout plan is designed computer-aided automatically as single filed, double filed, turning inside out, pin-based, one side-fixed and double side fixed way of sheet metal product designed as solid model in CAD atmosphere (Computer-aided design) at the program prepared. The product of which split layout is demanded to be formed should be designed by using solid modeling technique at any drawing or design program considering forming way and position to be made. Designer may enter any figure that may be made in cutting and piercing dies. Since the program prepared works in AutoCAD package program, the item designed should be recorded in any data format (DXF, SAT, DWG etc.) that is accepted by AutoCAD package program.

All elements on the product drawn by using solid modeling technique (holes, notches, main piece etc.) are perceived automatically when they are worked in AutoCAD package program. All lines that encircle sheet metal product were perceived as main piece in the program, all remaining elements in this main piece were set as holes on the product.

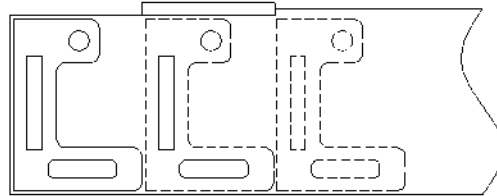
When the program is first operated, it asks designer to select steel metal material. During forming strip material layout plan, 124 different sheet metal material in ERDEMİR product catalog were used (10). The program was designed in a way that it is possible to enter new material species upon request. Designer was demanded to enter split layout way and step number following material selection. Designer was presented five different options herein including pin-based strip layout, single-side fixed strip layout, double-side fixed strip layout, double-lined strip layout and turning inside out strip layout.

Pin-based strip layout plan (Figure 2-a), single-sided fixed strip layout plan (Figure 2-b), Double-sided fixed strip layout plan (Figure 2-c), double-lined strip layout plan (Figure 2-d) and turning inside out strip layout plan which

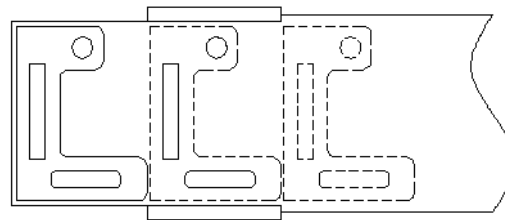
were formed for the same piece in the program prepared have been shown at Figure 2-e.



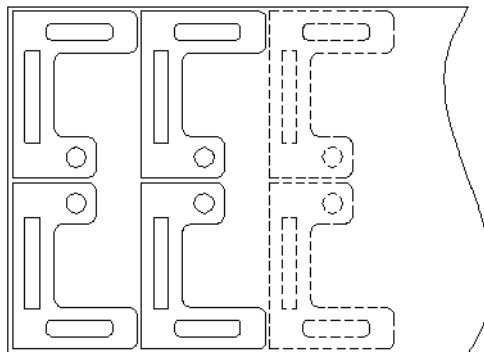
a. Pin-based strip layout plan



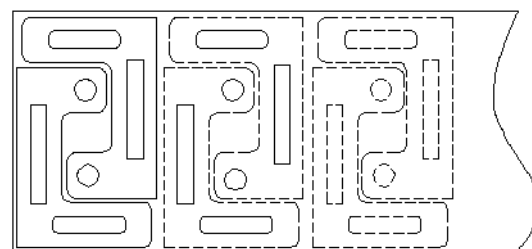
b. Single-side fixed strip layout plan



c. Double-side fixed strip layout plan



d. Double-lined strip layout plan



e. Turning inside out strip layout plan

Fig.2. Strip layout plans formed in the program prepared.

Further steps are required for various procedures on the piece to be molded. Especially piercing procedures which are very similar to each other on the piece should be made

in further steps. It is optional for user in the program in how many steps the piece will be produced. Depending on the number of steps determined by user, user is also asked which procedures will be made in which steps operations' sequence is determined according to user's choice.

Prepared program records formed strip layout plan in the form of DXF data. In this way, design may not only be opened in different CAD/CAP programs (Computer-aided design/Computer-aided production), but also the opportunity for obtaining drawings later and making some changes.

2.1. Sample Implementation

Split material layout plan of a sheet metal piece was formed in this section. It is seen that number of elements are very much and split layout plan of even a very complex sheet metal is formed easily.

Piece sample given is shown at Figure 3, sheet metal product material at Figure 4, step number and determining split proceeding way at Figure 5 and split layout plan formed at Figure 5.

Material with Erdemir Quality Number 6112, Standard Number DIN EN 10130-99 was chosen as split material.

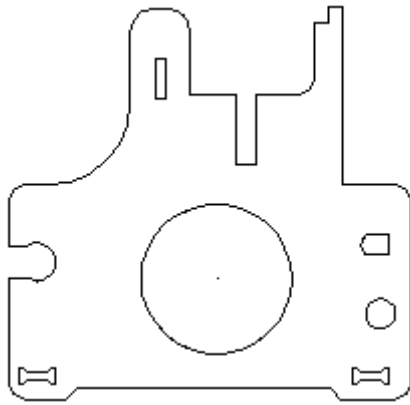


Fig.3. Forming part geometry

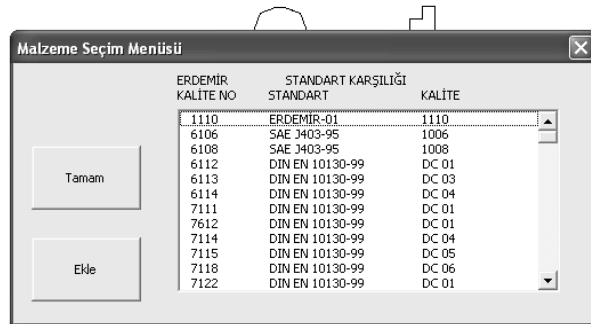


Fig.4. Choosing the material

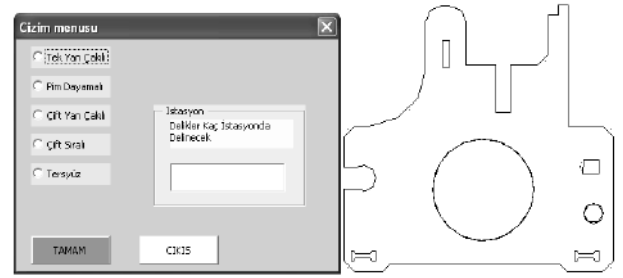


Fig.5. Determining step number and split

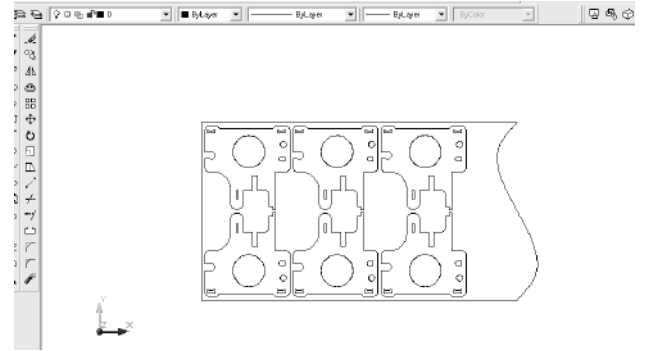


Fig.6. Forming split layout plan proceeding way

As it is seen at Figure5, holes are being pierced at the first stage of sheet metal produced and work piece is produced at the second stage. Furthermore, split layout was chosen as double-lined.

3. CONCLUSION AND DISCUSSION

The opportunity of making the design of piece to be designed as solid model in any CAD/CAP program in the study carried out is among the positive sides of the study. The program was set out for the purpose of minimizing waste of time at split layout plan design, making things easier for designer and minimizing possibility of making mistake.

Steps to be formed during split layout were left to designer's option. It is requested from the user to determine how many steps die will consist of and the procedures to be made in the steps. Prepared program forms die design according to the data entered by the user. Split material is made from material library of which its choice was formed and transferred to the program.

Study carried out facilitates design of sheet metal products with very complex perimeter geometry at a great extent. Since piece geometry in the formed program called "strip layout" is read over the piece recorded by drawing with solid modelling technique, even split layout plan of very complex shaped sheet metal products can be made with ease.

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OPTIMIZATION OF PROCESS PARAMETERS OF SURFACE ROUGHNESS IN AL-7075 DRILLING PROCESS

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Adem ACIR

Abstract: In this paper, the effect of the cutting parameter of surface roughness for AL 7075 material was investigated with Taguchi method. Experimental tests were used Firex[®] coated and uncoated drill tool. The surface roughness was increased with increasing feed rate whereas; the surface roughness decreased with increased cutting speed. The cutting parameter were determined as cutting speed (170m/min, 200m/min, and 230m/min) and feed rates (0.1mm/rev, 0.2 mm/rev, 0.3 mm/rev). Effects of the surface roughness of drilling process were obtained. The most important parameter was found as feed rate with %50.26 followed by cutting speed with %37.36.

Key words: Metal matrix composite; cutting force, optimization; machining; Taguchi method; ANOVA.

1. INTRODUCTION

A surface finish in machining process is a key factor in all areas of tribology and in evaluating the quality of machining operations. Surface quality in the drilling process was effected removed chip, heat treated and cutting parameters. Surface roughness is a measure of the quality of a product and a factor that greatly influences manufacturing cost. Therefore, the selection of suitable cutting parameters for a drill tool and workpiece material is important steps for successful implementation of drilling process. Recently, the effect of the cutting parameters were investigated as cutting speed, feed rate, depth of cut, etc. and tool geometry on surface roughness for various materials. Kosaraju and Chandraker investigated influences of cutting force and surface roughness in turning MDN350 steel with Taguchi analysis [1]. Rajasekaran et al., [2] performed effect of the surface roughness in turning process with Taguchi

method. Günay and Yücel conducted determination optimum parameters surface roughness in turning of high alloy white cast iron [3]. Tsao and Hocheng performed to determine of optimum cutting force and surface roughness in drilling composites materials using neural network and Taguchi analysis [4]. Qasim et al., carried out parametrical optimization of effect of the cutting parameters of machining Al 1045 steel with Taguchi method [5]. Çakıroğlu and Acir conducted Taguchi optimization method of tool chip interface temperature depending on the cutting parameters in drilling operations [6-7]. Haşçalık and Çaydaş investigated optimization of turning parameters for surface roughness and tool life based on the Taguchi method [8].

In this study, the determination of the optimum of the cutting parameters of surface roughness of Al 7075 material was investigated using Taguchi analysis.

2. EXPERIMENTAL SETUP

The drilling TESTS were performed in dry conditions on Johnford CNC vertical milling using uncoated and Firex[®] coated carbide twist drilling tools. Firex[®] product was a multi-layer coating that combines TiN, TiAlN, and TiCN. The point angle and diameter of a twist drill is 140° and 14 mm, respectively. In test, the Al 7075 material was used with 25 mm in diameter and length. The cutting parameters used of experimental test were given in Table 1.

Table 1. Assignment of the levels to the factors

Symbol	Machining Parameter	Levels		
		1	2	3
A	Tool	Uncoated	Coated	----
B	Feed rate, f (mm/rev)	0.1	0.2	0.3
C	Cutting Speed, Vc(m/min)	170	200	230

Surface roughness were measured with Technical characteristics of MahrPerthometer M1 surface roughness measurement device

3. RESULTS AND DISCUSSION

Experimental design method based on the Taguchi and analysis of variance (ANOVA) analysis. The parameter design is contributed to determine the effect of the factor in drilling process of Al 7075. Taguchi design can be provided to determine of optimum drilling parameter of AL 7075 for surface roughness. Experimental design using L₁₈ orthogonal array were presented as shown in Table 2.

Table 2. Experimental design using L18 orthogonal array

Experiment no.	A (Tool)	B (Feedrate)	C (Cutting Speed)
1	1	1	1
2	1	1	2
3	1	1	3
4	1	2	1
5	1	2	2
6	1	2	3
7	1	3	1
8	1	3	2
9	1	3	3
10	2	1	1
11	2	1	2
12	2	1	3
13	2	2	1
14	2	2	2
15	2	2	3
16	2	3	1
17	2	3	2
18	2	3	3

The control factor to determine of optimum surface roughness was selected as tool types, feed rates and cutting speed. The S/N ratio was function to determine the-lower-the-better quality characteristics. The signal-to-noise (S/N) ratio were given as below:

$$\eta = -10 \log \left(\frac{1}{n} \sum_{i=1}^n y_i^2 \right) \quad (1)$$

where η is the S/N ratio for the lower-the-better case (unit: dB), and y_i represents the surface roughness and n the number of repetitions in a trial [15-16]. The experimental values with the S/N ratio obtained from Eq. (1) were given Table 3.

In addition, ANOVA analysis was carried out of surface roughness to determine of effect of cutting parameters. The effects of the cutting parameters were determined with percent contribution ratio (PCR).

$$PCR = \left(\frac{SS_A - (V_e) \cdot (v_A)}{SS_T} \right) \times 100 \quad (2)$$

where SS_A is the sum of squares for parameter A, V_e is the variance of error, v_A is the degrees of freedom of parameter A, and SS_T is the total sum of squares [17].

Table 3. Experimental results and S/N ratio, (η), surface roughness

Experiment no.	A (Tool)	B (Feed rate)	C (Cutting Speed)	Surface roughness	
				Ra (μm)	S/N (dB)
1	Uncoated	0,1	170	2,439	-7,7442
2	Uncoated	0,1	200	1,995	-5,9989
3	Uncoated	0,1	230	1,056	-0,4733
4	Uncoated	0,2	170	2,733	-8,7328
5	Uncoated	0,2	200	2,056	-6,2605
6	Uncoated	0,2	230	1,598	-4,0715
7	Uncoated	0,3	170	3,305	-10,3834
8	Uncoated	0,3	200	2,948	-9,3905
9	Uncoated	0,3	230	2,332	-7,3546
10	Coated	0,1	170	1,780	-5,0084
11	Coated	0,1	200	1,600	-4,0824
12	Coated	0,1	230	1,220	-1,7272
13	Coated	0,2	170	2,522	-8,0349
14	Coated	0,2	200	2,129	-6,5635
15	Coated	0,2	230	1,910	-5,6207
16	Coated	0,3	170	2,841	-9,0694
17	Coated	0,3	200	2,678	-8,5562
18	Coated	0,3	230	2,170	-6,7292

As shown in Fig. 2., the most important parameter of surface roughness were obtained as feed rates with 50.26%. The second important factors were found as cutting speed with 37.36%. However, tool type shown a little effect on the surface roughness as shown in Table 4. Also, the residual error was computed 10.7 % as illustrated in Table 4.

Table 4. ANOVA results for primary surface roughness, μm (for S/N ratios)

Source	DoF	SS	MS	F-test	P-coefficient	PCR (%)
Tool	1	0,1444	0,1443	3,71	0,078	1,701
Feed rate	2	3,1929	1,5964	40,99	0	50,26
Cutting Speed	2	2,3938	1,1968	30,73	0	37,36
Residual error	12	0,4673	0,0389			10,67
Total	17	6,1983				100

Table 5. Results of confirmation tests for surface roughness, (μm)

	Initial cutting parameters	Optimal cutting parameters	
		Prediction	Experimental
Level	A2B2C2	A2B1C3	A2B1C3
Surface roughness, (μm)	2.129	1.12244	1.401
S/N ratio (dB)	-6.563	-1.78961	-2.929
Improvement of S/N ratio	3.634 dB		
Prediction error (dB)	1.139		

The average the S/N ratios determined to influence of each factor at different levels in the experimental tests. As shown in Fig 1. Mean S/N ratio graph for surface roughness were given and S/N ratios were given in Table 3.

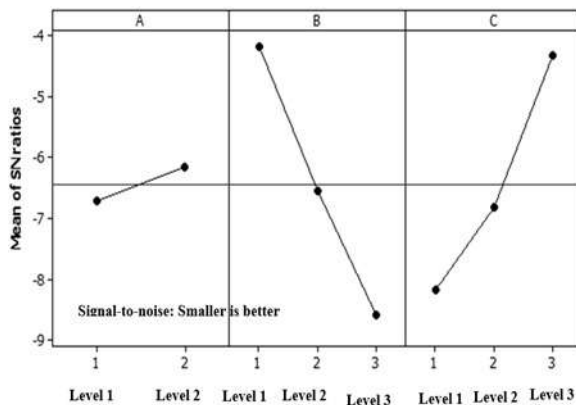
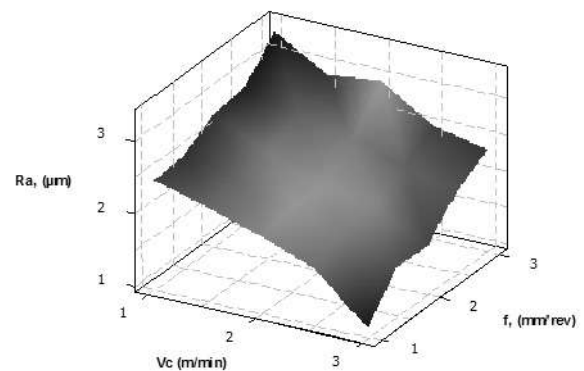
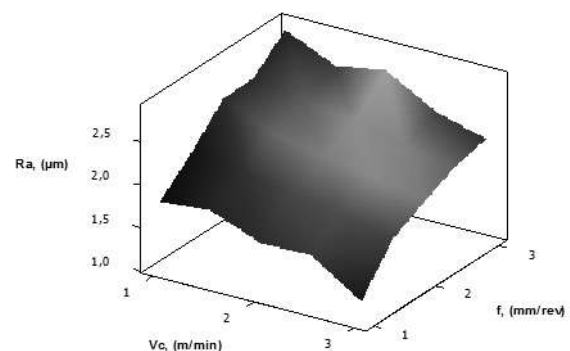


Fig.1. Mean S/N ratio graph for surface roughness

The criteria of the lowest response and lowest S/N ratio for surface roughness obtained were found as A2B1C3 as illustrated in Table 5. According to S/N ratio, drill tool at level 2, the feed rate at level 3 and the cutting speed at level 1 on cutting parameters were determined the best results for optimum surface roughness of drilling process. The surface roughness for coated and uncoated drill tools were given as shown in Figs. 2a and 2b. As illustrated in Figs. 2a and 2b, the surface roughness in drilling process was affected from feed rate and cutting speed. The surface roughness was increased with increasing feed rate whereas; the surface roughness decreased with increased cutting speed. These results were confirmed ANOVA results as shown in Table 4.



a) Uncoated tools



b) Coated tools

Fig.2. Experimental surface roughness results for coated and uncoated drill tools.

4. CONCLUSION

In this paper, the optimization of cutting parameters on surface roughness in drilling process with Taguchi and

ANOVA analysis were carried out. The main results were given as follows:

- The most important parameter of surface roughness were obtained as feed rates with 50.26% followed by cutting speed with 37.36%.
- The surface roughness optimal parameter were obtained as feed rate 0.2 mm/min, cutting speed 230m/min and coated drill tool by using the S/N ratio.
- The surface roughness was increased with increasing feed rate whereas; the surface roughness decreased with increased cutting speed.
- Taguchi method was an efficient and effective method for optimizing cutting parameters for surface roughness in drilling process.

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DETERMINATION OF OPTIMUM PARAMETERS OF CUTTING FORCE IN DRILLING OF B₄C ALUMINUM COMPOSITE WITH TAGUCHI METHOD

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Abstract: In this study, Taguchi method for cutting forces in the drilling operation of B₄C reinforced aluminum metal matrix composite was performed. Composite material was produced using powder metallurgy technique. Drilling operations were conducted using HSS drilling tools having 118° and 130° bit angle at two different revolution speed (800 rev/min, 1200 rev/min, and 1600 rev/min) and feed rates (0.05 mm/rev, 0.10 mm/rev, 0.15 mm/rev). Effect of the cutting parameters of cutting force was obtained with Taguchi Method. The effect of the feed rate was obtained %48 whereas; the revolution effect was found as %34. Also, a correlation was developed and compared with experimental results.

Key words: Metal matrix composite; cutting force, optimization; machining; Taguchi method; ANOVA.

1. INTRODUCTION

To attain net shape and good surface finish is very important machining of ceramic reinforced metal matrix composites (MMCs). The machining operations; i.e., turning, milling, drilling, are applicable to these materials widely. However, machining of the reinforcements in composite materials caused serious cutting force and wear [1-6]. In a previous studies composite materials was performed optimal parameters. Optimization of machining parameters for improving cutting force and

surface roughness in turning of Al6061-TiC in-situ metal matrix composites by using Taguchi method were performed by Ref. [7]. Vankanti and Ganta conducted determination of cutting parameters in drilling process of GFRP composite using Taguchi method [8]. Effect of the parametrical optimization in Drilling Al-Si₃N₄ Metal Matrix Composites with Taguchi Method was evaluated by Ref [9]. Çakiroğlu and Acir investigated tool and chip interface temperature analysis with Taguchi method to determine optimum parameters of drilling process [10]. Parametrical optimization of wear behaviour of Al-Al₂O₃ Composite materials with Taguchi Method was researched by Ref [11].

In this study, the optimization of the cutting parameters of cutting forces of B₄C was investigated with Taguchi method. The main aim was to determine the effect of cutting speed and feed rate type of tool on the cutting force. Also, the statistical correlation was developed and compared with experimental results.

2. EXPERIMENTAL SETUP

In machining used vertical machining center (Johnford VMC-550 Fanuc Series O-M) having a power of 15 kW and capable of a working speed of 3500 rev/min. In drilling considered as revolution, feed rate and tool type as cutting parameters. The cutting parameters were given in Table 1. No liquid coolant was used.

Table 1: Assignment of the levels to the factors

Symbol	Machining Parameter	Levels		
		1	2	3
A	Tool	A1212-118°	A1212-130°	----
B	Feed rate, f (mm/rev)	0.05	0.1	0.15
C	Revolution speed, (rev/min)	800	1200	1600

Cutting forces acting on the B₄C composite samples were measured with a Kistler dynamometer type 9257B. The data of cutting forces were extracted from the Dynoware Software. The main cutting force values were measured with Kistler dynamometer. B₄C reinforced aluminum composites were produced via powder metallurgy technique. The average sizes of the aluminum, copper and B₄C powders were found to be 37 µm, 33 µm and 16 µm, respectively. Produced composite samples were 50 mm x 50 mm x 10 mm in size.

3. RESULTS AND DISCUSSION

The Taguchi method is experimental design method. The parameter design is contributed to determine the effect of the control factor in drilling process. The experimental design can be contributed to evaluate the optimal drilling conditions of cutting force in B4C composite materials. Experimental design using L18 orthogonal array were presented as shown in Table 2.

Table 2: Experimental design using L18 orthogonal array

Experiment no.	A (Tool)	B (Feed rate)	C (Revolution)
1	1	1	1
2	1	1	2
3	1	1	3
4	1	2	1
5	1	2	2
6	1	2	3
7	1	3	1
8	1	3	2
9	1	3	3
10	2	1	1
11	2	1	2
12	2	1	3
13	2	2	1
14	2	2	2
15	2	2	3
16	2	3	1
17	2	3	2
18	2	3	3

The control factor for cutting force was determined as tool types, revolutions and feed rates. The S/N ratio was function to determine the-lower-the-better quality characteristics. The signal-to-noise (S/N) ratio were given as following:

$$\eta = -10 \log \left(\frac{1}{n} \sum_{i=1}^n y_i^2 \right) \quad (1)$$

where η is the S/N ratio for the lower-the-better case (unit: dB), and y_i represents the drill cutting force and n the number of repetitions in a trial [15-16]. The experimental results with the S/N ratio derived from Eq. (1) were given Table 3.

Table 3: Experimental results and S/N ratio, (η), cutting force

Experiment no.	A (Tool)	B (Feed rate)	C (Revolution)	Cutting force	
				Measured (N)	S/N (dB)
1	118	0,05	800	402	-52,08
2	118	0,05	1200	370	-51,36
3	118	0,05	1600	342	-50,68
4	118	0,1	800	438	-52,82
5	118	0,1	1200	410	-52,2557
6	118	0,1	1600	381	-51,61
7	118	0,15	800	459	-53,23
8	118	0,15	1200	442	-52,90
9	118	0,15	1600	423	-52,52
10	130	0,05	800	423	-52,52
11	130	0,05	1200	391	-51,84
12	130	0,05	1600	347	-49,68
13	130	0,1	800	434	-52,75
14	130	0,1	1200	423	-52,5268
15	130	0,1	1600	374	-51,46
16	130	0,15	800	463	-53,31
17	130	0,15	1200	445	-52,97
18	130	0,15	1600	434	-52,75

Also, analysis of variance (ANOVA) was performed cutting force for B₄C composite materials. The effects of the cutting parameters above cutting force with the ANOVA as percent contribution ratio (PCR) was investigated.

$$PCR = \left(\frac{SS_A - (V_e) \cdot (\nu_A)}{SS_T} \right) \times 100 \quad (2)$$

where SS_A is the sum of squares for parameter A, V_e is the variance of error, ν_A is the degrees of freedom of parameter A, and SS_T is the total sum of squares [17]. According to obtained results, feed rates were determined to be the first important factor the feed rate with 48.01% followed by revolution with 34.29%. But, a tool type didn't show effect on the cutting force. Consequently, feed rates were found as the highest effective parameters of the drill cutting force. In addition, the residual error was obtained 17.7 % as illustrated in Table 4.

Table 4. ANOVA results for primary cutting force, F_c (for S/N ratios)

Source	DoF	SS	MS	F-test	P- coefficient	PCR (%)
Tool	1	0.005	0.0055	0.03	0.859	0
Feed rate	2	7.571	3.7856	22.74	0.000	48.01
Revolution	2	5.503	2.7514	16.52	0.000	34.29
Residual error	12	1.99	0,1665			17.7
Total	17	15.08				100

The effect of each factor at different levels can be determined by averaging the S/N ratios in the experiments. The S/N response graphical form was shown in Fig. 1 for main cutting force and S/N ratios were given in Table 3.

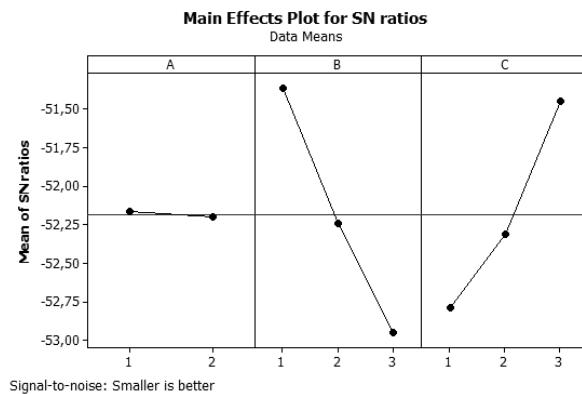


Fig.1. Mean S/N ratio graph for cutting force

The optimal conditions of levels for the criteria of the lowest response and lowest S/N ratio were obtained as A2B3C1 as illustrated in Table 5. It can be seen that the revolution at level 1, the 118° HSS drill tool at level 2 and the feed rate at level 3 of cutting parameters determined

the best results for optimum drilling process for main cutting force.

Table 5. Results of confirmation tests for cutting force, (N)

Initial cutting parameters	Optimal cutting parameters	
	Prediction	Experimental
Level	A2B2C2	A1B1C3
Cutting force, (N)	423	338.44
S/N ratio (dB)	-52.5268	-50.615
Improvement of S/N ratio	1.8463 dB	
Prediction error (dB)	0.0652	

The statistical correlation was developed and compared with experimental results. The main cutting force equation was developed and Eq. (3) was given.

The main cutting force correlation:

The results obtained with main force cutting correlation (Eq. (3)) with experimental results were compared and plotted in Figs. 2. The deviations between correlation results and experimental were within acceptable limit as $\pm 10\%$ for main cutting force components.

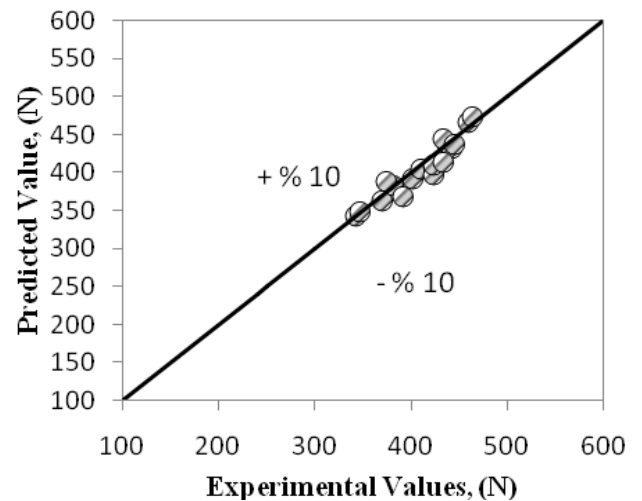


Fig.2. Comparison of experimental values and predicted values

4. CONCLUSION

In this study, the optimum cutting parameter was determined for B₄C composite material drilling process. The main results were as follows:

- The first significant factor of main cutting force with feed rate was obtained as 48.01% followed by revolution (rev/min) with 34.29% according to ANOVA results.
 - Optimal parameter for main cutting force was found as the revolution 1600 m/min, feed rate 0.05 mm/min and 118° drill tool by using the S/N ratio.
 - The statistical correlation for main cutting force was developed and compared. The deviations between correlation results and experimental were within acceptable limit as $\pm 10\%$
- Experimental design method and correlations can be provided the minimum cost and time in drilling process

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EFFECTS OF CUTTING PARAMETERS ON THE CUTTING FORCE AND TORQUE IN DRILLING OF AISI D2 STEEL

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Abstract: Drilling is an important machining procedure. Knowing the best work piece-cutting tool match and the ideal cutting parameters would improve the machining quality especially to overcome challenges faced during the machining of newly developed materials. In the experimental studies, AISI D2 steel is drilled by using 14 mm coated and uncoated solid carbide drills (AICrN) at four cutting speeds (60, 75, 90 and 108 m/min) and three feed rates (0.15, 0.20 and 0.25 mm/rev). The effects of cutting speed, feed rate and coating on the thrust force, cutting force and torque are investigated. According to experimental results, increasing feed rate caused an increase in thrust force and torque in all tests, performed using coated and uncoated drills. Increasing cutting rate decreases torque with uncoated drills while it increases the thrust force. On the other hand, increasing cutting speed with coated drills decreases thrust force and torque. It is observed that feed rate, cutting speed and drill type are the most important parameters in drilling. Best results were obtained at 60 m/min cutting speed and 0.15 mm/rev feed rate for uncoated drills and 108 m/min cutting speed and 0.15 mm/rev feed rate for coated drills.

Key words: AISI D2, Cutting Speed, Feed rate, Coating Application, Cutting Forces

1. INTRODUCTION

Drilling is an important machining application. It is difficult to estimate the results of a drilling process

performed in a closed volume and intervene according to the results. Besides, narrow tolerances expected from hole-shaft moving systems further increases the importance of good drilling applications. That is why many researchers deal with the relationship between the cutting parameters and cutting tool materials for drilling. Kalidas et al. drilled casted Aluminum 356 work piece using uncoated and coated drills (TiAlN / TiN, TiAlN coated HSS and MoS₂ (molybdenum disulfide) coated) and reported that the hole quality drilled with coated drills is better compared to the holed drilled with uncoated drills [1]. Cakıroğlu et al. machined Al2014 material with uncoated and coated (TiN/TiAlN/TiCN) drills, and determined that the cutting force increases by increasing feed rate and it decreases by increasing cutting speed [2]. Ertunç et al. reviewed the studies and determined that the parameters like the material type, feed rate and cutting speed produce different wear types [3]. Bağcı et al. machined AISI 1040 and Al 7075-T651 materials using coated drills (TiN/TiAlN) and determined that cutting forces increase as the feed rate increases [4]. Yağmur et al. machined AISI 1050 material using uncoated and coated carbide drills (TiN, TiAl and TiACN) and determined that thrust forces measured with coated tools are less than those of uncoated tools. They also determined that torque increases by increasing feed rate and cutting speed [5]. Usta, drilled AISI 1050 and 3G40 materials with uncoated and coated (TiAlN) carbide tip tools and observed that thrust force is higher with uncoated tools than coated tools [6]. İşbilir et al. machined a titanium alloy (Ti6Al4V) work piece using a TiAlN coated carbide tool and determined that thrust force and torque increase by the increasing feeding rate [7]. Kıvrak et al. machined a Ti-6Al-4V alloy material using uncoated and TiAlN/TiN coated tool. They measured a 14% less cutting force with coated tool and also determined that feed rate is a more important parameter on cutting forces compared to cutting speed [8]. Meral et al. machined AISI 105 production steel using uncoated and TiAlN coated tools and determined that; coated tools have better performance than uncoated tools, cutting force is affected mostly by feeding rate and tool diameter, and cutting speed has only a slight effect on cutting force [9].

Literature review indicates that the type of cutting tool, work piece material, cutting speed and feed rate are effective on the performance of hole-drilling process among others. In this experimental study, AISI D2 steel is drilled at four cutting speeds (60, 75, 90 and 108 m/min)

and three feed rates (0.15, 0.20 and 0.25 mm/rev) using uncoated and AlCrN coated solid carbide tools with 14 mm diameter. The effects of cutting speed, feed rate and coating application on the thrust force, cutting force and torque are investigated.

2. METIERIAL AND METHOD

Table 1 gives the chemical composition of AISI D2 cold work tool steel that is used in the experimental studies. Mean hardness value of AISI D2 steel is measured as 96 HRB (225 HB), and test samples with 25 mm outer diameter and 35 mm hole length were prepared as through holes were prepared (Figure 2). Drilling experiments are performed in Johnford VMC-550 CNC vertical machining center choosing fixed cutting tool/rotating work piece application. As cutting tool, 14 mm diameter uncoated and AlCrN (Monolayer) coated carbide drills are used following 3D drill depth principle. AlCrN is a coated tool and preserves its hardness especially at elevated temperatures. It also has a high corrosion resistance. Table 2 gives the mechanical and thermal properties of cutting tools used in the experiments.

Drilling experiments are performed in Johnford VMC-550 CNC vertical machining center choosing fixed cutting tool/rotating work piece application. As cutting tool, 14 mm diameter uncoated and AlCrN (Monolayer) coated carbide drills are used following 3D drill depth principle. AlCrN is a coated tool and preserves its hardness especially at elevated temperatures. It also has a high corrosion resistance. Table 2 gives the mechanical and thermal properties of cutting tools used in the experiments. A Kistler 9272 type dynamometer, which works quartz crystal based, and a Kistler Type 5070 amplifier are used to determine three cutting forces (F_x , F_y , F_z) and torque (M_z) at the same time. Dynamometer is installed onto the benchtop and cutting tools are fixed to the dynamometer using a specially designed apparatus. Measured values are converted to numerical values using DynoWare software. Experiments are conducted as the combinations of fou cutting speeds (60, 75, 90 and 108 m/min) and three feed rates (0.15, 0.20 and 0.25 mm/rev). Figure 2 shows the experimental setup prepared for this study.

Table 1. Chemical composition of the AISI D2 steel (% weight)

<i>AISI D2</i>	<i>C</i>	<i>Si</i>	<i>Mn</i>	<i>P</i>	<i>S</i>	<i>Cr</i>	<i>Mo</i>	<i>Ni</i>
	1,58	0,435	0,459	0,028	0,011	11,467	0,88	0,092
<i>Al</i>	<i>Co</i>	<i>Cu</i>	<i>Nb</i>	<i>Ti</i>	<i>V</i>	<i>W</i>	<i>Sn</i>	<i>Fe</i>
0,001	0,011	0,037	0,008	0,003	0,84	0,058	0,03	84,092

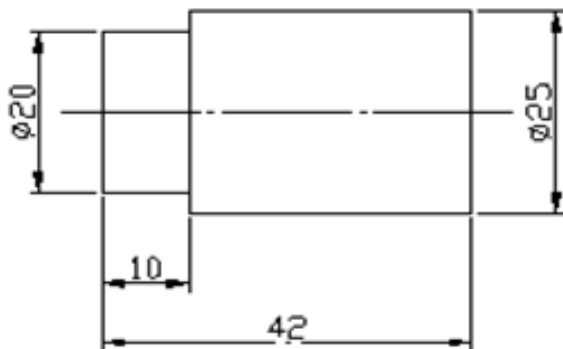


Fig.1. Technical drawing of the test sample.

Table 2. Mechanical and thermal properties of cutting tools used in the experiments.

<i>PROPERTIES</i>	<i>K25 grade (Uncoated)</i>	<i>P25 grade (Coated)</i>
<i>Density (gr/cm³)</i>	14,6	12,6
<i>Compressive Strength (MPa)</i>	5000	4600
<i>Young Modulus (GPa)</i>	590	550
<i>Poisson Ratio</i>	0,22	0,22
<i>Thermal Conductivity (W/mK)</i>	70	45
<i>Thermal Expansion Coefficient (10⁶/K)</i>	5,6	6,7

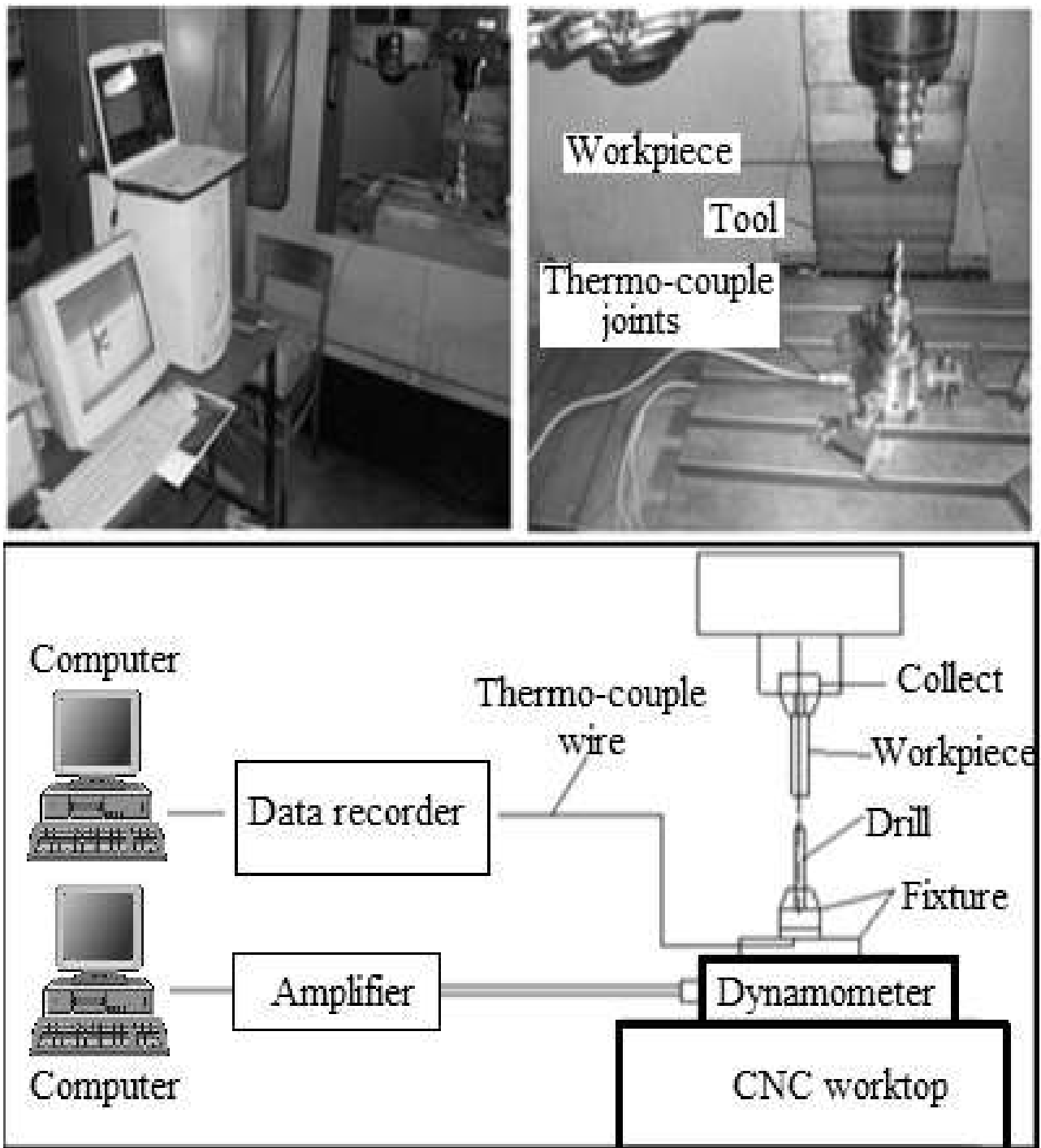


Fig.2. Experimental setup

3. RESULTS AND DISCUSSION

Cutting force, is the combination of the thrust force which drives the drill through hole axis and tangential force that provides rotational motion of the drill to remove chips from the hole. Therefore thrust force and cutting force are related to each other. However in practice, cutting force is accepted to be equal to tangential force (F_t) which is removing the chips. Tangential force produces torque [10].

Figure 3 gives the thrust force, torque and cutting force graphs obtained from drilling AISI D2 steels by using uncoated and coated carbide drills at different cutting-parameter combinations. Graphs indicate that increasing feeding rate increases thrust force, torque and cutting force for all tests performed by uncoated and coated drills. This situation can be attributed to removal of more volume of material through the drilling axis as the feeding rate is increased. Increasing removed material volume cause more cutting force to break off the chips thus a higher torque is produced.

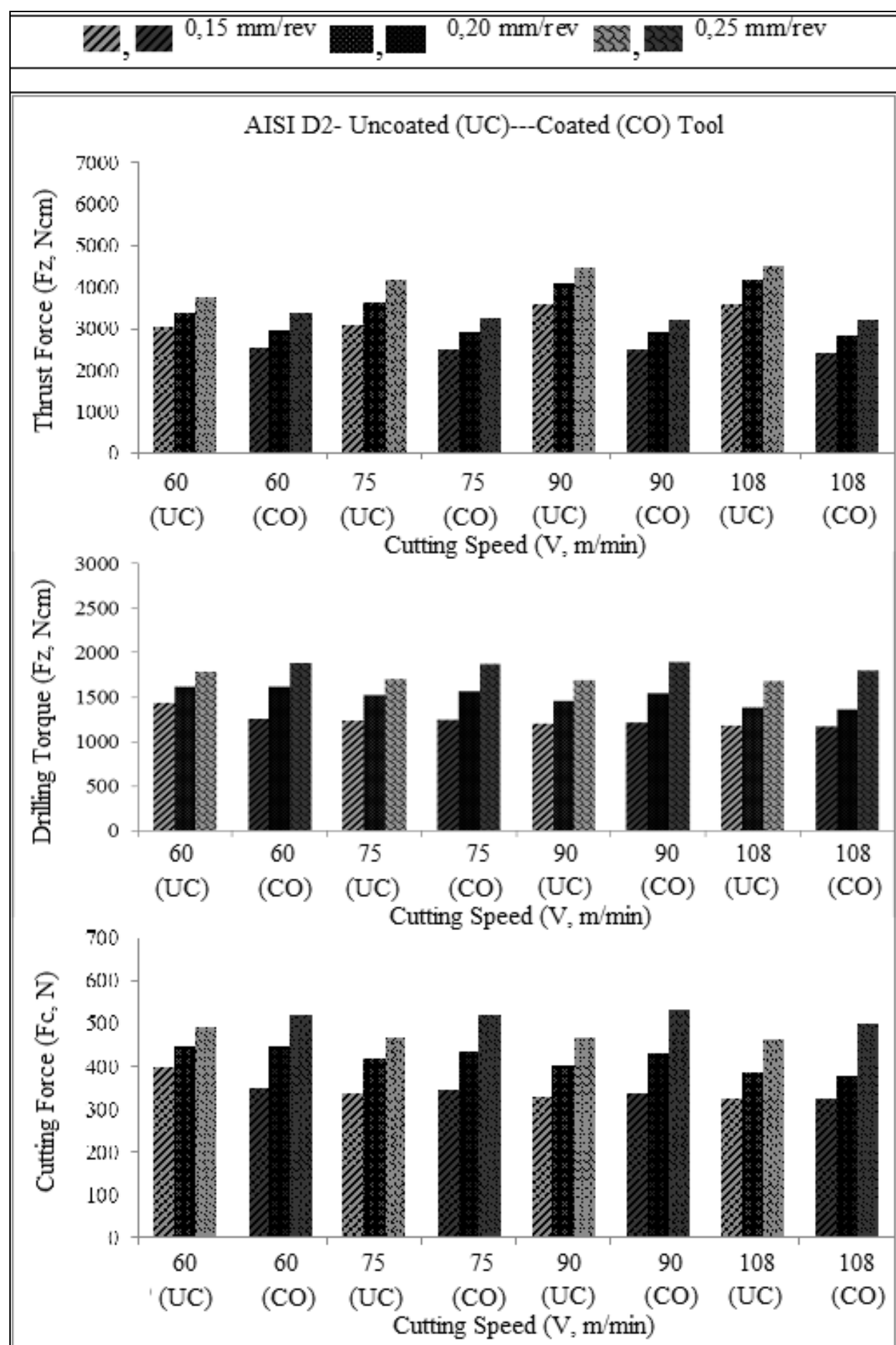


Fig.3. Thrust force, torque and cutting force graphs

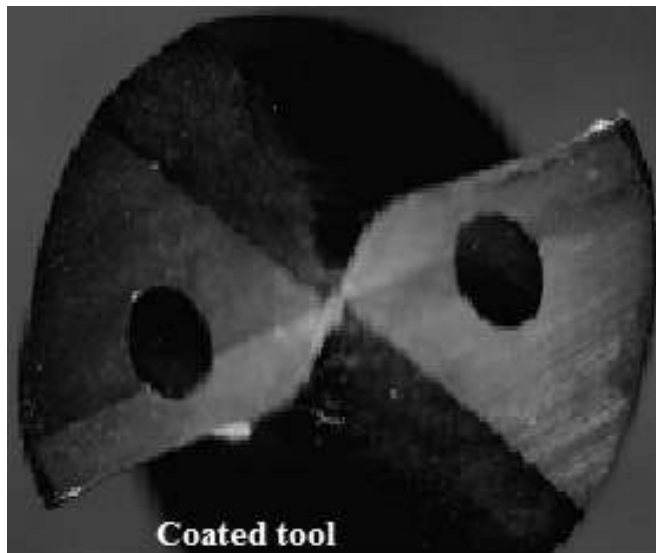


Fig.4. Coated tool

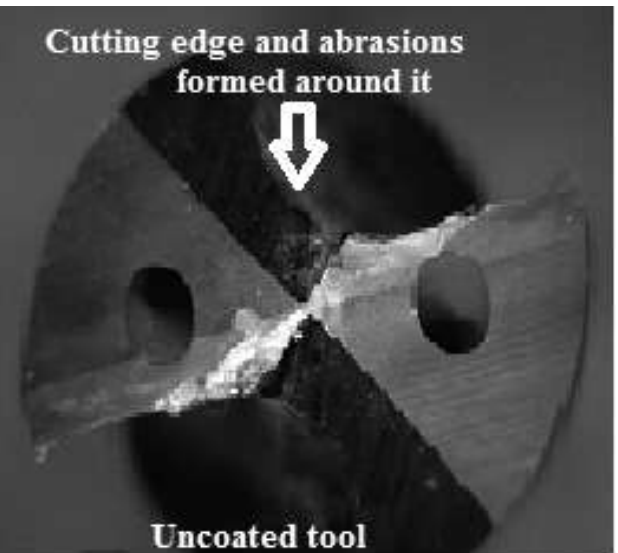


Fig.5. Uncoated tool

As for the cutting tools for the same cutting conditions, the required thrust force for coated drills is found less than uncoated drills. However, torque and cutting force are higher compared to those of uncoated drills. This can be explained by higher wear resistance of coated drill (Figure 4) along with less friction between the drill and removed chips, and on the other hand, tip wear of the uncoated drill (Figure 5). Coated drill continued its process normally during the drilling procedure and removed required volume of chip material within the unit time. Uncoated drill, on the other hand, couldn't manage to go that further because of the tip wear and could remove less amount of material from the work piece than the amount required for the process. Less chip removal requires less cutting force and produces less torque. Cutting force is measured as minimum at 0.15 mm/rev feeding rate for all tests conducted by both coated and uncoated drill bits.

Figure 3 shows that increasing cutting rate decreases cutting force with coated tools which can be attributed to high wear resistance of coated tool and lower friction between the tool and the removed chips. Increasing temperature developed by increasing cutting rate provides deformation of the work piece with less force and facilitates the removal of the chip as well [11]. However, increasing cutting speed with uncoated tools increases the thrust force, and decreases torque and cutting force which can be explained by tip wear of uncoated tool as the cutting speed increases (Figure 5). As the cutting speed increases, tip wear of the tool increases which requires more force to obtain the drilling distance within a certain time. Although the thrust force increases, drilling tool cannot go the distance it has to within a unit of time and thus, chip volume removed per a unit of time decreases which means a decrease in cutting force and torque.

4. CONCLUSION

In this study, AISI D2 cold work tool is machined by using coated and uncoated drilling tools, and the effects of cutting parameters (feeding rate and cutting speed) and coating on the cutting force, torque and thrust force is

investigated. Below are some conclusions that can be made from the experimental results;

- all tests, increasing feeding rate increased cutting forces and it is observed that feed rate is an important parameter for increasing cutting forces.
- Increasing cutting speed increases the thrust force due to drill wear in the uncoated tool while it decreases the cutting force in coated tool.
- For uncoated drills, 0.15 mm/rev feed rate and 60 m/min cutting speed combination gives the lowest thrust force, and 0.25 mm/rev 108 m/min cutting combination results in the highest thrust force.
- For coated drills, 0.15 mm/rev feed rate and 108 m/min cutting speed combination gives the lowest thrust force, and 0.25 mm/rev 60 m/min cutting combination results in the highest thrust force.
- The minimum cutting force for drilling AISI D2 cold work tool steel with uncoated drill is observed with 0.15 mm/rev feed rate and 108 m/min cutting speed combination.
- The minimum cutting force with uncoated and coated drills are observed at 0.15 mm/rev and 108 m/min cutting combination.
- Coated drills are better for drilling AISI D2 steels in terms of wear properties.

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AN EXPERIMENTAL STUDY OF THE EFFECT OF ABRASIVE WATER JET AND LASER BEAM ON THE SURFACE INTEGRITY

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Abstract: *Ti6Al4V alloys are often employed in aeronautic industry because of having excellent properties. Abrasive Water Jet Machining (AWJM) is one of the attractive techniques to machine these materials because AWJM systems do not cause any thermal degradation (distortion) on machined surface. On the contrary, heat affected zone is often seen under machined surface by Laser Beam machining. In this work AWJM and LBM has been compared to the conventional milling using Ti6Al4V workpiece to understand benefits of nonconventional techniques for these materials. After a series of experimental studies has been evaluated for each techniques, surface integrity on machined workpiece has been investigated. The results showed that AWJM is a good alternative method instead of conventional milling*

Key words: *surface integrity, AWJM, LBM, Ti6Al4V*

1. INTRODUCTION

Machining is classified into two categories as conventional and non-conventional machining processes. Non-conventional machining is employed to alternate conventional machining for materials difficult to machine. Ti6Al4V alloys have uncommon mechanical and physical properties such as high strength-to-weight ratio, high temperature strength, high corrosion resistance, good high temperature properties [1-3]. These exceptional material properties restrict the machinability of the Ti6Al4V [4]. Most of conventional machining operations such as turning, grinding, milling, drilling, cutting, etc. are employed in the manufacturing process of aerospace and medical components. The low thermal conductivity of Ti6Al4V causes high temperatures on the tool/workpiece interface. Thus, it affects the lifetime of the tool dramatically. Besides the high temperature strength with a low Young's modulus make the machinability difficult

[5]. As a result of Ti6Al4V's allotropic nature, at low temperatures Ti6Al4V's has hexagonal close-packed structure-alpha phase, or above elevated temperature (about 890 °C) has a body-centered cubic structure-beta phase. These phases affect mechanical strength and micro structure formation [6].

Yang et. al (2010) concluded that the laser parameters such as laser power, laser scan speed, the angle of incidence and the diameter of the laser spot and material and thermal properties strongly affect the depth and width of the heat affected zone [7]. When nitrogen as an inert gas with LBM process was used, subsequently growth of titanium nitride dendrites (crystallization) occurred perpendicularly to the top surface. Lisiecki (2014), implied that high depth of the titanium nitride dendrites are up to 180-250 µm in the surface [8].

In aerospace component parts production processes, AWJM technology which is one of the recent non-traditional method has been employed. AWJM has many supremacy abilities like no thermal distortion, high machining versatility, high flexibility and small cutting forces [9]. Hascalik et. al. (2007) emphasise that the traverse speed of the jet is an important parameter on the surface integrity after machining according to the traverse speed [10].

According the literature review, LBM has some adverse effects on machined surface such as heat affected zone and Titanium Nitride layer. In contrast with the LBM, abrasive water jet machined surfaces are assured better surface integrity. In the scope of this experimental study, the optimum AWJM parameters is investigated and also have concluded LBM is convenient for machining Ti6Al4V alloys. In this work AWJM and LBM has been compared to the conventional milling using Ti6Al4V workpiece to understand benefits of nonconventional techniques for these materials.

2. MATERIALS AND METHODS

2.1. Experimental Setup

Titanium alloy Ti6Al4V having bulk hardness of HV 349 was chosen. Machining was carried out by two different non-conventional techniques LBM, AWJM and one conventional technique milling. Ti6Al4V alloy bar (workpiece) was prepared 10x6x20 (wxhxl) mm due to easily clad on machining system. Chemical composition and material properties are given in the Table 1-2 respectively.

Table. 1. Chemical composition of workpieces (wt%)

Ti	89,464	O	0,18
Al	6,08	C	0,02
V	4,02	N	0,01
Fe	0,22	H	0,0053

The dimensions of workpieces are 10x6x6 mm (wxhxl). AWJM operations are performed by TCI water jet systems and milling operations are performed also Cincinnati universal milling machine under Turkish Aerospace Industry (TAI, Ankara/Turkey) facilities. LBM

cutting processes are studied by FIBERMAK-Fiber Laser-Cutting-Machine under ERMAKSAN (Bursa/Turkey) facilities.

Table. 2. Material properties of Ti6Al4V

Density	4.43 g/cm ³
Modulus of elasticity	113.8 GPa
Tensile strength	620 Mpa
Thermal conductivity	6.7 W/m-K
Melting point	1604 - 1660 °C
Beta transition temperature	980 °C
Hardness	HV 349

In the experiments cutting parameters are given in the Table-3-5. In addition some photos of performing systems can be seen. LBM is given by figure-1 and from AWJM illustrations is given by figure-2. After these machining operations, the photographs of cutting surface were observed by LEICA DMI5000M-optical microscope which is under Gazi University Engineering Faculty Powder Metallurgy Laboratory facility. The workpieces were investigated in an optical microscope at a magnification of 200x. Surface roughness measurements are worked by Taylor Hubson Surtronic 3+ Roughness Measuring Instrument which is under Gazi University Engineering Faculty Powder Metallurgy Laboratory facility. For each sample surface roughness measurement were taken from ten different locations. Surface traverse and longitudinal directions and the results were averaged. To perform micro hardness measurements and recast layer evaluation, selected workpieces were placed and subsequently set in bakelite using a Struers mounting press and then grounded and polished. Micro hardness from traverse section of cutting surface is measured by HIGHWOOD HWMMT-X3 diamond indenter operating with a load of 25 g for 15 s. The Vickers micro hardness measurements [HV_{0,025}] made on the transversal sections were made at 40x magnification at a distance of 10 µm from the top side of the machined cut edge by 30 µm gap and total measurement length is 595 µm.

2.2. Experimental Results and Discussion



Fig.1. Illustration of LBM operation

Table. 3. LBM machining parameters.

LBM	
Laser type	Fiber laser
Power	4000 W
Stand of distance (SOD) height	1 mm
Feed rate	2900 mm/min
Inert gas	N ₂ (%99,5)
Pressure	24 bar
Lens diameter	150 mm
Nozzle diameter	3,5 mm
Focal length	-4,5 mm



Fig.2. Illustration of AWJM operations

Table. 4. AWJM machining parameters.

AWJM		
	P1	P2
Orifice diameter	1 mm	1 mm
Pressure	4000 bar	4000 bar
Feed-rate	30 mm/min	15 mm/min
Abrasive size and type	80 mesh-garnet	80 mesh-garnet
Stand of distance (SOD) height	1,5 mm	1,5 mm
Abrasive-water rate	450 g/min	450 g/min

Table. 5. Milling parameters.

MILLING	
Cutting Speed	256 rpm
Feed-rate	56 mm/min
Depth of Cut	1.5 mm
Tool	D= 125 mm; z=8

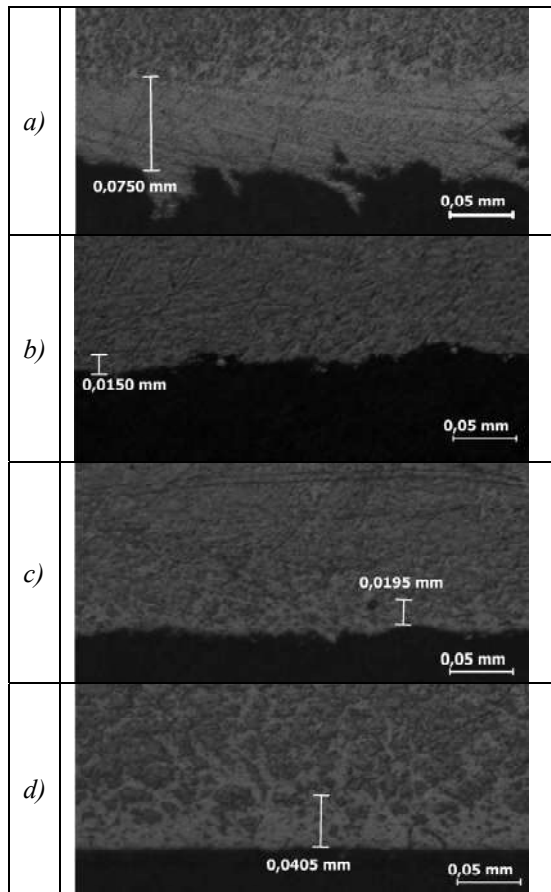


Fig.3. Cross sectional micrographs a) LBM cutting edge, b) AWJM (P1) cutting edge, c) AWJM (P2) cutting edge, d) Milling surface edge

Figure 3-a shows machined surfaces view of workpieces machined by different techniques. Micro structure evaluation of the cutting surfaces of workpieces revealed for different distinct zone for each technique, LBM, AWJM (P1), AWJM (P2) and Milling. On the first stage of the study, nitrogen was used by LBM process as inert gas. From the figure 3-a, it can easily be seen structural changes followings surface cracks, recast layer, micro voids, redissolution of phases. A big thermally influenced heat affected region which is high as 75 μm can be seen on the LBM workpieces.

AWJMed workpieces shows no visible micro voids and also no cracks present on the cutting edges under two cutting conditions (P1 and P2) are identified. Both AWJMed workpieces has initial damage region thickness 15 μm , 19,5 μm respectively. The alteration on surface morphology is generated from the instantaneous penetration of abrasive waterjet. Milled surface shows plastically deformation which is measured 40,5 μm length (Figure 3-d).

The workpieces were also submitted to test surface roughness and micro hardness. Results of surface measurements are shown in the figure 4. LBM surface is the worst surface of all because of used inert gas usage and nature of workpiece materials interactions take place titanium nitride layer. Two different AWJM parameters applied and AWJM (P2) parameter set has given better surface and hence lower feed-rate. Additionally addition extra workpieces were machined using a milling tool for

comparison purpose. Figure also shows allow average roughness among the AWJMed workpieces and milled workpiece.

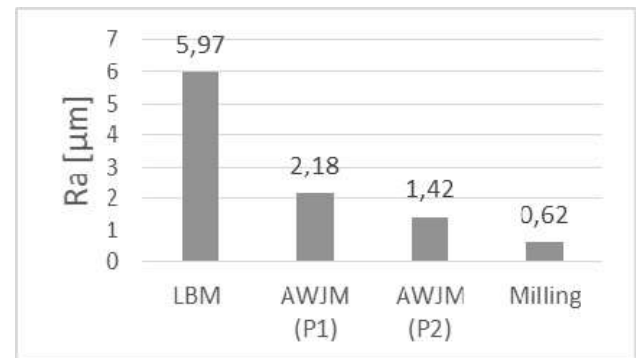


Fig.4. Surface roughness (Ra) results

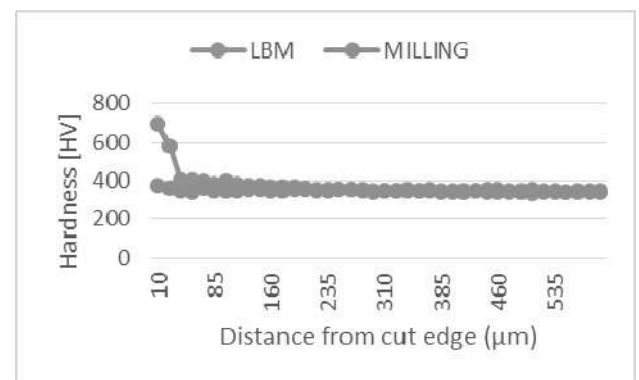


Fig.5. Micro hardness results of LBM and Milling.

Figure 5 shows that, a superior hardness has been investigated on the surface. That hardness is caused by titanium nitride layer and heat affected zone. And also it is observed that, surface hardness is decreasing away from the surface.



Fig.6. Micro hardness results of AWJM (P1) and Milling.

When micro hardness is investigated it has been determined that slopes of AWJMed and milled overlap. After measuring AWJM (P1) and milled workpieces, hardness values were found to be slightly different near the machined surface while AWJM (P2) and milled workpieces has no differences. This is the reason of why AWJM (P1) feed rate is two times faster than AWJM (P2). It is also seen that bulk hardness of Ti6Al4V (HV 349) was observed by measured hardness values.

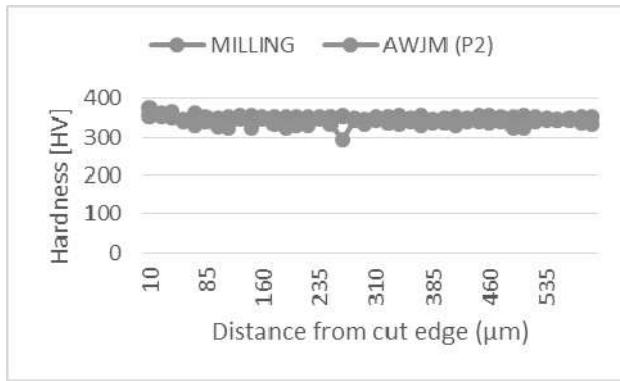


Fig.7. Micro hardness results of AWJM (P2) and Milling.

3. CONCLUSION

In this study Ti6Al4V alloy having superior material properties has been studied by each technology. The conclusions of study as follows;

- LBM with nitrogen as inert gas causes a thick titanium nitride layer formation on the surface. As a result, depending on these results hardness of workpieces and surface roughness has enhanced.
- In terms of surface roughness, better results were achieved using milled operations. And also compared with milled workpiece, hardness of AWJM (P1) were found to be slightly different.
- As the feed rate increases, the hardness increases, plastic deformation layer reduces.

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TIP-JET HELICOPTER PROPULSION SYSTEM TESTING

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Abstract: Tip-jet helicopter relies on pneumatic power transfer system which efficiency is hard to determine and therefore needs to be tested. The paper explains the tests and the test stand used for tip-jet helicopter propulsion characteristics measurement.

Key words: Tip-jet helicopter, thrust measurement, force measurement, acquisition systems

1. INTRODUCTION

Hot cycle system transmits power pneumatically by lightweight ducting and a nozzle that directs high-energy gas from a turbine engine to the rotor blade tips to drive the rotor for helicopter flight.

Reaction driven rotor acts as a power turbine which directly convert the energy of the gases from the engine into rotary power by using light weight ducting (Fig.1.).

With this hot gas powered transfer system, the complexity and excess weight of the gearboxes, shafting, and tail rotor are eliminated. This is due to the fact that the reaction drive concept does not import a torque on the fuselage, and therefore there is no need for a tail rotor [1]. The necessary torque that creates the rotor rotation is now being generated by the moment [Nm] (i.e. rotor radius) and by the force (F_j) created when mass flow is ejected through the tip jets located at the blade tips. The magnitude of this torque – generating force is based on the amount of mass flow ($\dot{m}_j$) ejected, and by the difference of velocity between the tip jet (v_j) and tip speed (v_T).

$$F_j = \dot{m}_j \cdot (v_j - v_T)$$

In order to calculate F_j , the mass flow and velocity of the tip jet, $\dot{m}_j$ and v_j , respectively, must be calculated first. Both of these parameters depend on the results of the engine cycle analysis. Based on these facts, it is obvious that the engine and rotor form a highly coupled system. The variable that couples the engine with the rotor in the sizing procedure of a reaction driven rotor is the tip jet velocity v_j . This parameter is calculated by assuming that the flow expands isentropic from the internal rotor duct pressure to the freestream pressure outside the tip jet nozzles. Equation below shows explicitly how the tip jet velocity is dependent on the temperature and pressure inside the rotor duct:

$$v_j = \sqrt{2RT_{od} \left(\frac{\gamma}{\gamma-1} \right) \left[1 - \left(\frac{P_\infty}{P_{od}} \right) \right]^{\frac{\gamma-1}{\gamma}}}$$

Since it is usually impractical to include a divergent part into the thrusting nozzle, the exit velocity would be close to the sonic one corresponding to temperature of the exhaust gasses. The propulsive efficiency of the thruster in hover can be expressed as

$$\eta_p = \frac{2v_T}{v_T + v_j}$$

2. TESTING SYSTEM

On the tip-jet helicopter, hot compressed air goes out on the nozzles of blade tips, it produces torque for rotation of the rotor system [2].

Power of the tip-jet system is calculated as:

$$P_S = Q_m \cdot \Omega \cdot \eta_p [W]$$

where Q_m is torque of the tip jet system:

$$Q_m = F_j \cdot D [Nm]$$

F_j is thrust force of the tip nozzle, and D is diameter of the rotor.

Ω [rad/s] is angular speed of the rotor.

$$\eta_p = \frac{2v_T}{v_T + v_j}$$

η_p is the efficiency of a jet propulsion system, v_T is tip speed and v_j is the speed of the hot air that is ejected through nozzles on the blade tips (Fig.1.).

From equations above, we can see that the power P_S is directly proportional to F_j (thrust force). That's why it is very important that thrust force is measured properly.

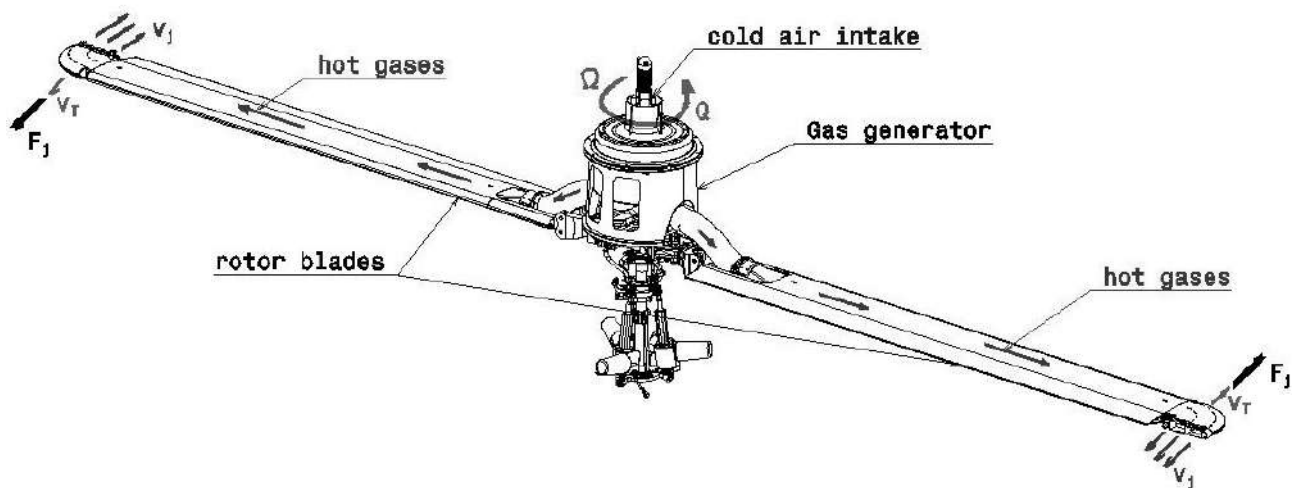


Fig.1. Tip-jet propulsion system

Purpose of the test was to measure thrust force on the blade tips - F_j . In order to realize what is the nature of hot air flow inside the blades and to see where the losses are, the test was performed on actual blades, on cylindrical

tubes of different diameters, with and without flexible pipes. We also wanted to experimentally check, and define nozzle area on the blade tips so it can provide maximum thrust without exceeding critical temperature.

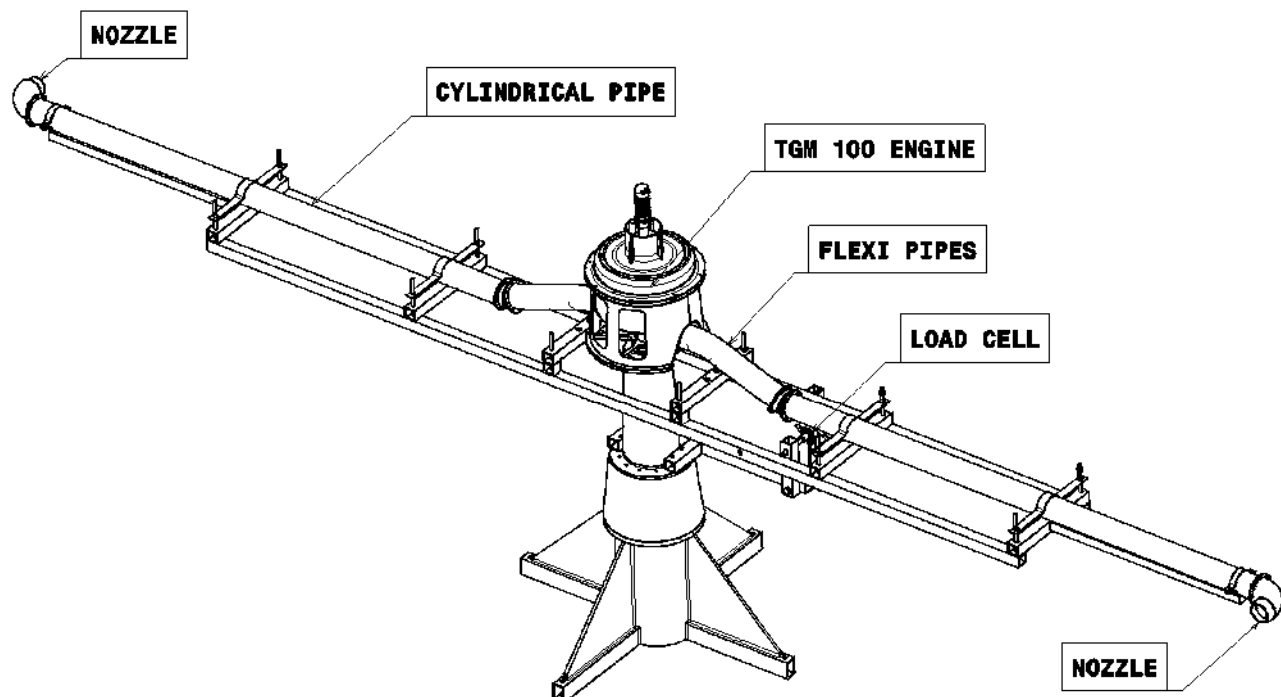


Fig.2. Test stand for measuring of the thrust force on blade tips

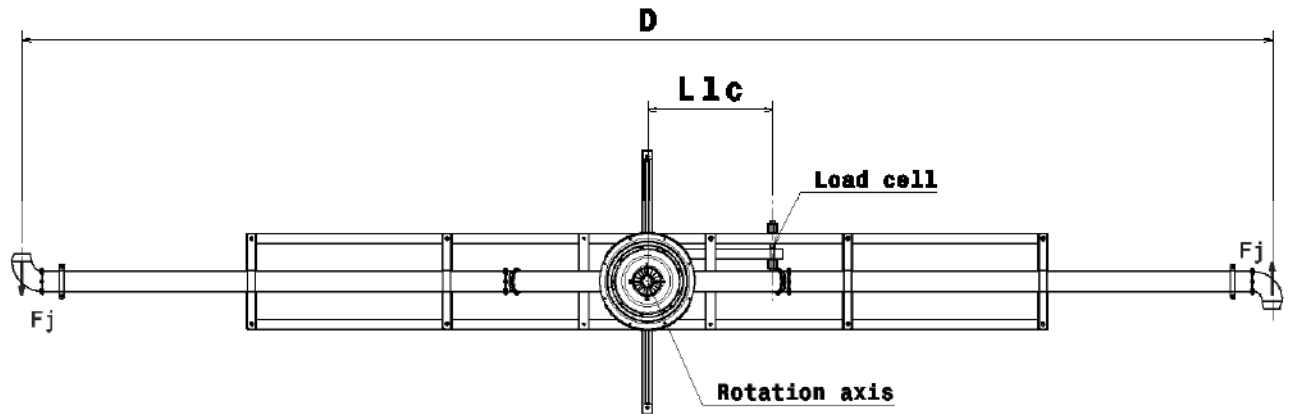


Fig.3. Test stand – load cell position

Test stand is designed to measure thrust force on the blade tips with load cell. Load cell [3] was positioned on L_{lc} distance from rotation axis so it prevents rotation of the system and in the same way indirectly measure the force on the blade tip nozzle (Fig.3.). F_j (thrust force) is calculated:

$$F_j = F_{lc} \cdot \frac{L_{lc}}{D}$$

Where F_{lc} is force measured on load cell, L_{lc} is distance from load cell to rotation axis and D is distance between nozzles on two blades.

We used OMEGA LCHD-2K load cell [4] with capacity of 2Klb (909kg).

For the pressure measurement [5], a pressure transducer PX602-150GV, produced by OMEGA, is used. Since the working temperatures of these sensors are low for measuring the total pressure in the nozzle, where the gas has considerably higher temperature, a special installation was introduced. It consists of a probe for total pressure with a corresponding extension for lowering the gas temperature.

The temperature was measured on one spot inside the nozzle with a K-type thermo couple CH+ AL-, produced by OMEGA.

3. EXPERIMENTAL RESULTS

First test was conducted on actual blades that had nozzle area of 2300mm². Blades with same nozzle area were used on previous hovering test. On that hovering test we managed to lift 210[kg]. Engine rpm, bearing temperature and pump PWM were also measured, with acquisition system that was designed for static test of the engine.

Second test was conducted on cylindrical pipes with inner diameter $D_{in}=76$ mm. Nozzle area was little smaller than on the actual blades – 2200mm². We used same acquisition system as on the first test.

Third test was conducted on cylindrical pipes with inner diameter $D_{in}=70$ mm. Nozzle area was also 2200mm². The purpose of the test was to see how the reduction of inner flow area affect tip-jet ejection performances.

Fourth test was conducted on cylindrical pipes with inner diameter $D_{in}=70$ mm but without flexible pies. The purpose of this test was to verify loses caused by flexible pipes.

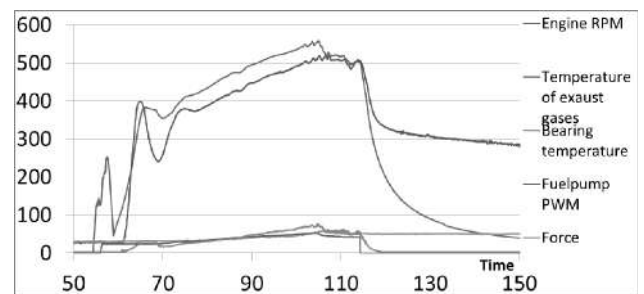


Fig.4. Acquisition system data of one of the tests (nozzle area 2300mm²)

Overpressure on the distributor and on the nozzle was also measured and the results can be seen on Fig.5

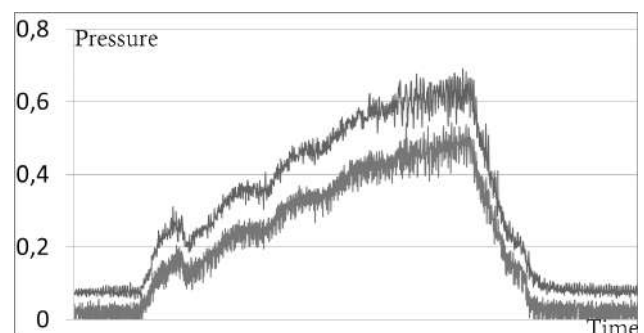


Fig.5. Overpressure on the distributor (red) and on the nozzle (blue)

The last test was with actual blades but with nozzle area of 2000mm². All test data could be seen in table 1.

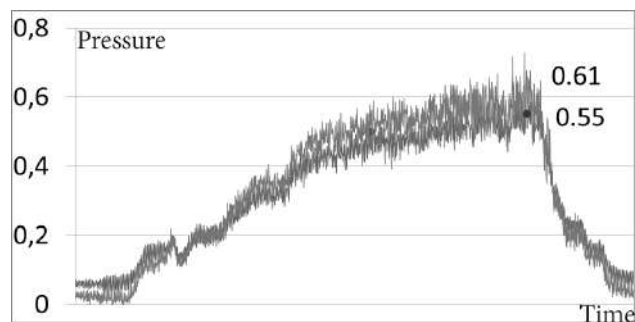


Fig.6. Overpressure on the distributor (red) and on the nozzle (blue) on actual blades

Table 1. Test results data

Test No.	Load cell force F_{lc} [daN]	Load cell position L_{lc} [mm]	Nozzle distance D [mm]	Thrust force F_j [daN]
1.	80	500	4800	8.33
2.	145	500	5000	14.5
3.	155	500	5000	15.5
4.	90	500	2400	18.75
5.	140	500	4800	14.58

4. CONCLUSION

By reduction of nozzle area from 2300mm² to 2000mm² we managed to increase lift force more than 25%. If the losses in flow through the flexible pipes stay the same we can reduce nozzle area little bit more because exhaust gas temperature is 100[°C] smaller than the critical value. By doing that we will increase lift power by additional 5-10%.

Tests with cylindrical pipes with different inner diameter proved that we can reduce area of the channel inside the blades. That allows us to produce extruded aluminum blades with inner Inconel channel for hot air flow. Or, if we continue to use all Inconel blades we can reduce the chord of the blades.

Tests with cylindrical pipes without flexi pipe shown that the losses in the flexi pipes are 17% (thrust force on the nozzle with flexi pipe is 15.5[daN] and without flexi pipe is 18.75 [daN]). It directly reduces the power of the rotor system. Flexi pipes that we use can be replaced with the ones that have better sealing. The system will be a little bit heavier but we expect to cut these loses at less than 10%. In that way we will increase the power of the system by 7%.

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MODULAR CONSTRUCTION OF CIRCULAR MANIPULATOR AS A TEST BED FOR TESTING PNEUMATIC CONTROL

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Abstract: This paper shows a development of pneumatic circular manipulator as a test bed for testing pneumatic control. It is shown three version of manipulator which enable three types of load: compression, straining and bending. Manipulator uses a two types of energy: pneumatic and electrical and has high energy saving potential. In the future, manipulator will be used for investigations in the field of energy efficiency of compressed air systems and electrical systems.

Key words: pneumatic control, circular manipulator, modular construction

1. INTRODUCTION

Identification of energy savings potential within production processes and reducing energy consumption is increasingly important in order to stay competitive [1]. A growing number of researches has recently devoted to this issue [1-3]. Parts of production processes that have high energy savings potential are parts which are used any other type of energy in addition to electricity, such as compressed air. The application of compressed air has a growing trend due to easy and safe generation, manipulation and usage [3]. In addition to other possibilities (for example, detection of air leakage in the system), very important possibilities for energy saving in pneumatic systems are:

- pressure reduction (using a pressure which is enough to match the required power and speed for a given application and using a lower pressure in the working chamber which works without load) and

- re-using of compressed air which is already used.

This paper shows a development of new experimental setup called pneumatic circular manipulator [4] which is used for testing the energy efficiency of pneumatic systems under different conditions. Manipulator uses two types of energy: pneumatic energy and electrical energy and it represents device that has high energy saving potential, like a part of complex robotic cell [1]. The energy consumption of manipulator is dependedent on two factors: firstly, on the type of energy, and secondly, on the type of load.

The work is presented in the following manner. Section 2 gives a basics for development of test bed for testing pneumatic control. In section 3 is shown a detailed description of basic version of circular manipulator and ways for developing a new versions of manipulator. Section 4 gives concluding remarks and topics for future investigations.

2. BASICS FOR DEVELOPMENT A TEST BED FOR TESTING PNEUMATIC CONTROL

For the real testing of above mentioned possibilities, and establishing levels of cost savings that can be obtained, it is necessary to perform tests on a representative setup which enables the expression of these saving possibilities due to its characteristics. Also, it is necessary to identify the most frequent situations in pneumatic control which is applied for the implementation of standard industrial tasks.

The most common used pneumatic actuators are pneumatic cylinders. According to their structure they can be divided into two major groups: cylinders with piston rod and cylinders without piston rod. In order to perform manipulative tasks they are usually equipped with grippers. The most common used types of grippers are mechanical grippers and vacuum cups.

In terms of control, the largest number of applications is so called on-off control where is necessary to implement the final retracted and drawn-out position of the cylinder. However, the significant development in computer control, which enabled more accurate and cheaper position control of pneumatic cylinders, has lead to increased applications of the proportional pneumatic control that allows positioning of the actuator in any point within the stroke of the cylinder. It can also be noticed that different situations occurs if the load is raised with a cylinder positioned on its underside (Fig. 1a), with a cylinder positioned parallel to the raising direction (Fig. 1b) or with a cylinder positioned above the load (Fig. 1c). In the first case the piston rod of the cylinder is loaded on compression (Fig. 2) and there isn't need for special gripping because the load relies on the top of the piston rod. It is only necessary to set a support plate on the top of the piston rod (without additional active elements).

In this case, strain is calculated using formula 1 [5]:

$$\sigma = \frac{F}{A} \quad (1)$$

where F is force and A is cross sectional area.

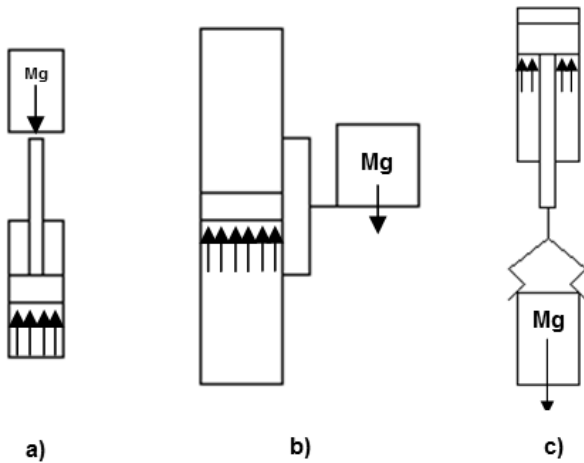


Fig.1. Different ways for raising load

In the second case a cylinder without piston rod is applied. On its outside, this cylinder has a load carrier with a support plate (without additional active elements) where is placed the load. It is obvious that, in addition to mechanical loaded on compression, the cylinder is also loaded with a bending moment (Fig. 3) due to the displaced raising axis relative to the axis of the cylinder. In this case, strain is calculated using formula 2 [5]:

$$\sigma = \frac{M}{W} \quad (2)$$

where M is bending moment and W is axial torque.

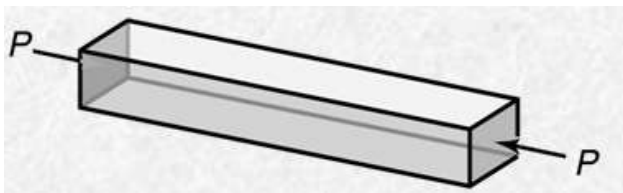


Fig.2. Compression

In the third case, the cylinder is loaded on straining (Fig. 4) and also there is a need for additional active element - gripper. In this case, strain is calculated using formula 1, too.

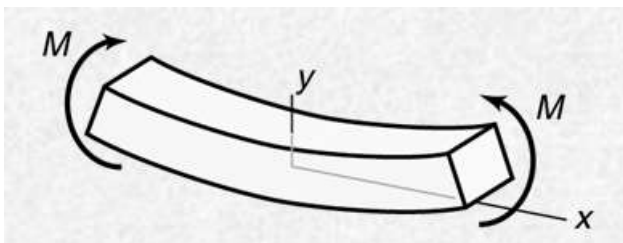


Fig.3. Bending

Also, in the first and the third case the pneumatic force is not the same. In the first case the load is raised with full pneumatic force possible which can be achieved for that diameter of the cylinder, while in the third case the load is raised with less force because it is necessary to subtract the surface area of the piston rod from the surface area of the piston. In the second case, the force is the same for raising and lowering.

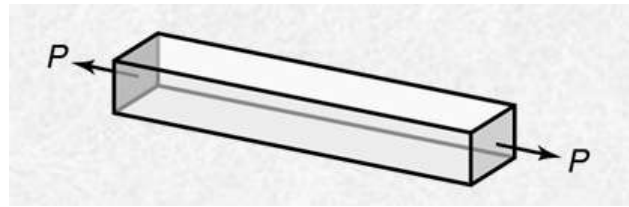


Fig.4. Straining

In order to make a unique test setup able to examine various types of pneumatic control, a modular device called pneumatic circular manipulator is designed. The manipulated workpieces have a closed, circular trajectory. Namely, one cylinder exempts workpieces from the vertical storage unit at the lowest point and brings them to the another cylinder, which raises workpieces to the upper level. At the end of process, the third cylinder (multi-position cylinder) returns them to the storage unit and their circulation ends.

For the raising of the workpieces all the three above mentioned cases can be realized. This analysis has led to the design of such a device.

The starting point was the fact that the storage unit capacity is 6 workpieces that are 100 mm high and 80 mm in diameter. Workpieces are made of steel and their weight is 4 kg. Weight limit is set so that in the case of coupled work of manipulator with a standard industrial robot which has capacity up to 6 kg, robot may exempt them. In order to realise three different positions for the raising cylinder, it must be given enough space (min 700 mm) above and below the vertical storage unit. Also, for the implementation of proportional control it should be given enough space parallel to the body of the cylinder, so a linear encoder can be placed. The board is designed so that all the necessary holes for all the positions of the cylinders and other actuators are drilled.

3. MECHANICAL CONSTRUCTION OF THE CIRCULAR MANIPULATOR

Pneumatic circular manipulator is a new experimental setup, which is developed on Faculty of Technical Sciences in Novi Sad through participation in NeReLa Tempus project [6]. In the first version, the manipulator is consisting of several pneumatic actuators - 3 double acting pneumatic cylinders. One of them is a multi-position cylinder and it is realized by using two double acting pneumatic cylinders, which have identical characteristics. In such way, three different positions are provided. Also a pneumatic gripper is applied which is used for gripping and releasing of the workpieces.

3.1. Description of manipulator operation mode

Workpieces are placed in the vertical storage unit. At the lowest point of storage unit, one horizontally implemented pneumatic cylinder (cylinder A), pushes workpiece forward and places it above the vertically implemented pneumatic cylinder (cylinder B). Cylinder B raises workpiece upwards to the level of the second horizontally implemented pneumatic cylinder (cylinder C), which has pneumatic gripper at its right end. Finally,

cylinder C grasps the workpiece with the pneumatic gripper and places it to the top of the vertical storage unit. Workpiece is sliding down through the storage unit with the help of gravity. When workpiece came down to the initial position, one working cycle is ended.

For easier understanding of manipulator operation mode, in Fig. 5 is shown the travel-step diagram, wherein the left and right pneumatic cylinders of multi-position pneumatic cylinder in the diagram are marked with C and D, respectively.

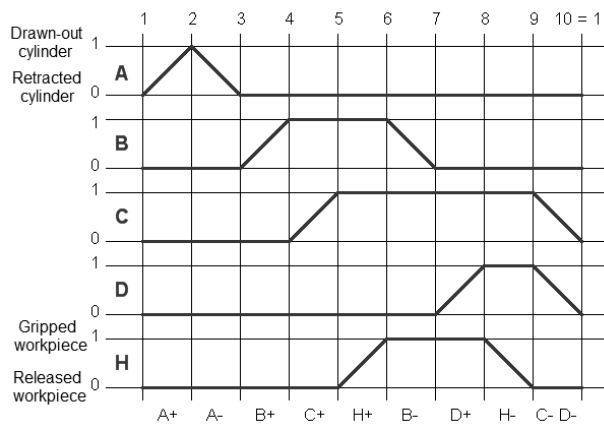


Fig. 5. Travel-step diagram

3.2. CAD model of manipulator

Construction, which includes defining and mutual placement of elements and subassembly of device [5], is made in SolidWorks 2015. The first version of CAD model of circular manipulator is shown in Fig. 6.

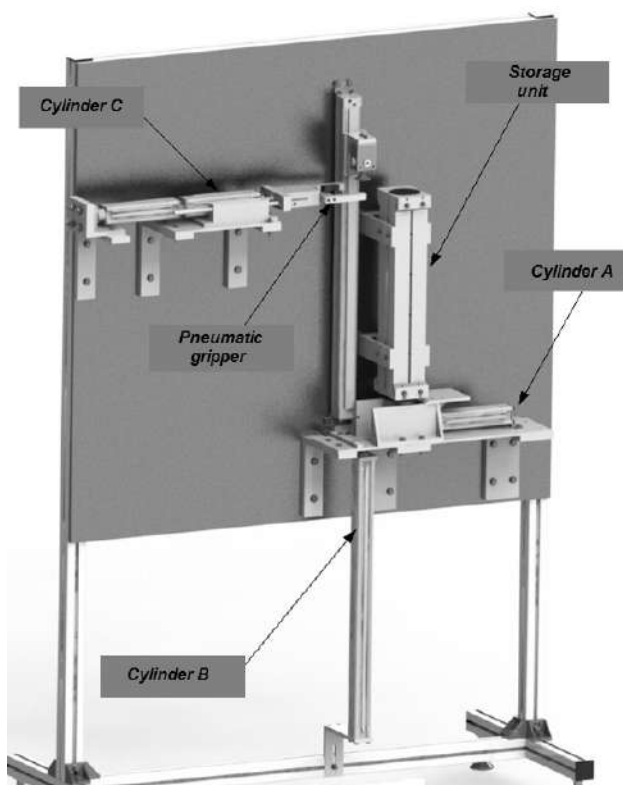


Fig. 6. CAD model of circular manipulator

3.3. Practical realization for the one case

The first version of manipulator has its practical implementation, which is shown in Fig. 7. All pneumatic actuators are from Festo. Cylinders A and C have designation DNC-32-100-PPV-A. Cylinder B has designation DNC-32-600-PPV-A. Pneumatic gripper has designation HGP-25. A supporting structure is a rectangular plate, with dimensions 1300x1200x15, which is made of PVC. Using screw connections, the plate is connected with profiles, with cross section area of 50x50, which is made by the Bosch Rexroth, and thereby manipulator is achieving a stable equilibrium position. Pneumatic actuators and the storage unit are implemented on the supporting plate using aluminum profiles and screw connections, too.

Based on the above, it can be concluded that the manipulator consists of 5 modules:

- supporting structure with Bosh profiles,
- storage unit with workpieces,
- cylinder A with its supporting platform,
- cylinder B with platform for accepting workpiece and cylinder C with pneumatic gripper.

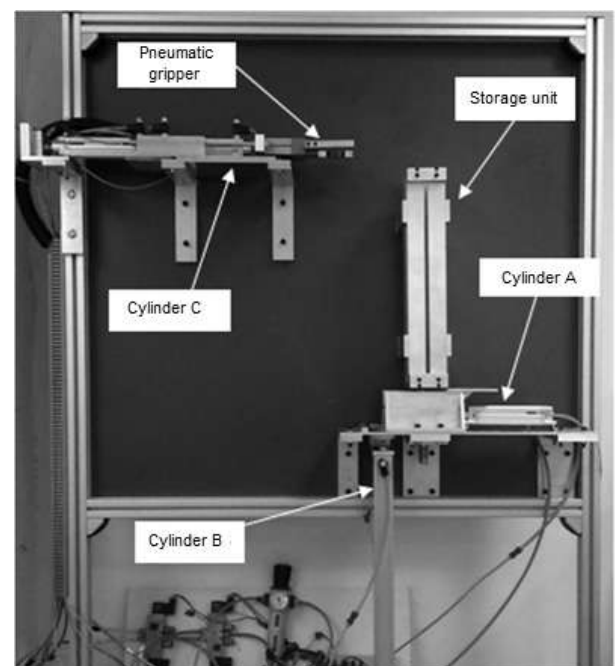


Fig. 7. Practical implementation of the first version of circular manipulator

According to the basic idea, it will be developed two other versions of manipulator in addition to the basic, which is presented above. The main difference will be way of raising workpiece upwards to the level of pneumatic cylinder C and thus we will obtain the different types of load. In this case (when cylinder B raises workpiece), it is compression, as shown in Fig. 2.

The second version of manipulator means using a cylinder without piston rod with platform for accepting workpiece. In such way is obtained a new type of load. It is bending, as shown in Fig. 3.

The third version of manipulator means using a pneumatic suction cup for raising workpiece upwards. The suction

cup will be placed on cylinder which will be connected with the supporting structure using screw connections as can be seen at 3D model of manipulator (Fig. 6). That cylinder will be set up in parallel with storage unit, directly above cilindar B. In that way, it is obtained a straining as type of load (Fig. 4).

4. CONCLUSION

Nowadays, in industrial systems the improving of energy efficiency of the system is crucial for the existence in the market. In this paper, a test bed device for testing of energy efficiency pneumatic control of pneumatic systems is shown. The basic version of test bed device called pneumatic circular manipulator is physically realized. Further researches would be directed to the realization of second and third version of manipulator, testing various types of pneumatic control and development of complex robotic cell which consists of standard industrial robot, pneumatic circular manipulator and biaxial pneumatic or hydraulic manipulator for contact task. CAD model of this robotic cell is shown in Fig. 8. Realization of this cell will enable new tests in the field of energy efficiency and optimization of various parameters.

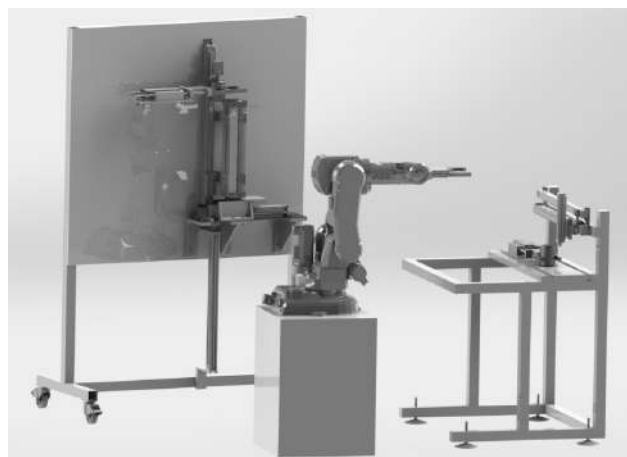


Fig.8. The first CAD model of complex robotic cell

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INVESTIGATION OF MACHINING CARBON FIBER REINFORCED COMPOSITE MATERIALS WITH SOLID CEMENTITE CARBIDE TOOLS

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Ulvi ŞEKER

Abstract: Carbon fiber reinforced composites are being used in aviation and space industry due to their low density, high strength, high corrosion resistance and low thermal expansion coefficient characteristics. In this study, peripheral milling of carbon fiber reinforced composites (CFRP) was experimentally investigated by using coated 4- edge carbide mills. In the study, three different cutting speeds (350, 400 and 450 m/min), four different cutting tools (TiAlN coated with no chip breaker, TiAlN coated with chip breaker, AlCrN coated with chip breaker and diamond coated) and constant feed rate (1270mm/min) were used. This way cutting forces, surface roughness and delamination width were investigated. At the end of the tests it was observed that the increase in the cutting speed decreased the average surface roughness values. In the cutting forces no regular tendency was seen like in the average surface roughness values. Besides it was also seen that the increased cutting speeds decreased the delamination width.

Key words: CFRP, Solid Cemented Carbide, Surface Roughness, Delamination Width, Cutting Forces

1. INTRODUCTION

As a matter of fact, composite materials exist since thousands of years, however their technological development started after becoming insufficient at certain points during the 2nd world war and from then on investigation and development studies on the production and mechanical properties of these materials continued increasingly. The driving force for the developments was to obtain high strength/density and high elastic modulus/density ratios [1]. By the replacement of steel and aluminium with composite materials a weight

recovery of 60-80% and 20-50% is possible respectively. Nowadays, composites seem to be preferable materials for a number of engineering applications. These weight recoveries provide significant energy savings when the increasing energy problems are considered [2].

Machinability behaviors of each material differ from each other because their physical and mechanical characteristics are different. Cutting behaviour of composite materials depend also on the fiber shape, fiber content and fiber harmony of the material. Since composites are not homogeneous, their machinability differs from an homogeneous material like steel. For this reason machinability of composite materials is difficult comparing with other materials. This needs to be very selective for the cutting tools. Composites must be machined by using a sharp cutting tip. In the case of over wearing of cutting edge, fibers will not be cut but ruptured causing the layers to separate. The most common chip removal (machining) processes are drilling and end milling.

By the widely usage of composite material, production and processing of carbon fiber reinforced epoxy composites which are especially used widespread in aviation industry attracted the attention of investigators significantly in the last years. For this reason, in literature there are many studies on the investigation of machinability of carbon fiber. Experimental examination, modelling and analysis of cutting forces occurring in the machining of carbon fiber reinforced composite materials are the principal subjects that the researchers consider [3-7]. Besides, delamination (damage) [8-13] and surface quality [14-19] forming during machining of carbon fiber reinforced composites also take place among the subjects that are investigated. Although the studies related to the machining of fiber reinforced composites increased in the last years, limited number of studies were found in the literature on the milling of these materials (especially on peripheral milling). So in this study, machining of carbon fiber reinforced composite materials by coated carbide end mills of different geometry has been evaluated. In the evaluation, cutting forces occurred during chip removal, average surface roughness of the machined surfaces and the delamination width caused by machining were taken into consideration.

2. MATERIAL AND METHOD

2.1. Material

The carbon fiber reinforced composite materials which will be used in the tests are being widely employed in the aviation sector. These materials were produced by *Odak Kompozit Teknolojileri A.Ş* in accordance with material characteristics used in the industrial applications. Production of materials was carried out in the following way; prepreg was produced by using 245 gr/m² 2x2 3K twill carbon fiber and Huntsman XU3508 resin system. Prepreg material was symmetrically laid on a glass surface with 0° and 45° fiber angles and plates were made by vacuum bagging method. Plates were cured at 120° C for 4 hrs. After producing sufficient amount of prepreg it was cut into desired dimensions. For 20 mm plate 80 layer reinforcement material was used. In Twill texture type, carbon fibers create fabrics in the 0° and 90° directions.

For this reason only 0° and 45° angles were used as the layout angles.

After the completion of layout process, the plate sides were bagged with vacuum equipments and taken into vacuum. The vacuum compressed reinforcements were placed into the furnace. The system was held at 120° C for 4hrs. Under these conditions the glass transition temperature of the cured plate took place at between 116°C - 123° C.

In the tests, solid carbide end mill (having four different geometries) was used. These cutting tools were identified as Geometry 1, Geometry 2, Geometry 3 and Geometry 4 and their codenames were G1, G2, G3 and G4. General characteristics and technical properties of these cutting tools that were used in the tests are given in Table.2 respectively. While G1, G2 and G3 have helical angle of same direction, G4 numbered tool has bidirectional helix. This bidirectional helix tool is advised by the manufacturers to prevent delamination and for this reason was added to the study and was aimed to compare with the tools having helix of same direction.

Table 1. General characteristics of cutting tools used in the tests

Cutting Tool	Coating Type	Friction Coefficient	Number of Edges	Helix Height (mm)	Cutting Tool Height (mm)
G1/ no chip breaker	TiAlN	0,35	4	21	73
G2/ chip breaker	TiAlN	0,35	4	21	73
G3/ chip breaker	AlCrN - based	0,25-0,35	4	21	73
G4/Rigt-Left Helix	Diamond	0,01	4	35	73

Chip removal tests were carried out by using CNC vertical machining center (Johnford VMC-550) of which the technical characteristics are given in Table 2.

Table 2. Technical characteristics of tool machine used in the tests

Power of Machine	5 kW
Maximum revolutions per minute	6000 rev/min
Stroke Length (x, y, z)	600, 500, 600 mm
Accuracy	0,001 mm
Operating system	Fanuc

High cutting speeds are required for the machining of composite materials. The highest revolution speed of the machining center was 6000 rev/min and because of this, speed head was used. (Gisstec Motion) speed head (technical properties are given in Table 3.) was connected to the spindle and increased the revolution of spindle 4,8 times.

Table 3. Technical properties of speed head

Power	Axial Force	Turns Ratio	Weight	Maximum revolutions per minute
1,7 kW	380 N	1:4,8	3,8 kg	30.000 rev/min

For experimentally determination of cutting forces, KISTLER 9272 type dynamometer and Kistler Type 5070 amplifier (working on the basis of quartz crystal) which have the capacity of measuring three cutting force components at a time (Fx, Fy, Fz) were used. Technical characteristics of the dynamometer are given in Table 4.

Table 4. Technical characteristics of KISTLER 9272 dynamometer

Measuring Range Fx, Fy	-5...5 kN
Fz	-5...20 kN
Mz	-200...200 Nm
Accuracy Fx, Fy	-7.8 pC/N
Accuracy Fz	-3.5 pC/N
Accuracy Mz	-160 pC/Nm
Linearity	%≤ ±1 FSO
Capacitance F, Fy, Fz	185 pF
Capacitance Mz	420 pF

For the purpose of determining surface quality to measure the average surface roughness (Ra) values on the machined surfaces, a portable Mahr Perthometer M1 surface roughness device was used. Technical characteristics of this device is given in Table 5.

Table 5. Technical characteristics of Mahr Perthometer M1 surface roughness measurement device

	M1
Scan speed	0,5 m/sn
Scan Force	0,75 mN
Needle nose radius	2 µm
Measuring Range	100-150 µm
Profile resolution	12 mm
Filter	Gaussian
Sampling length (λ)	0,25 – 0,8 – 2,5 (mm)
Ölçme uzunluğu (L)	1,75 – 5,6 – 17,5 (mm)
Parameters that can be measured	Ra, Rz, Rmax

2.2. Method

In this study machining of CFRP materials was investigated and machining tests were carried out by using three different cutting speeds and constant feed rate. When determining the cutting speeds and feed rate, the parameters used in machining CFRP materials at aviation industry were chosen. These parameters are in conformity with the values recommended by the tool producers. The cutting parameters used in the tests are given in Table 6. In this study CFRP material was machined at vertical CNC machining center and the schematic representation of test apparatus is given in Figure 1. The cutting forces creating during the tests were measured by KISTLER 9272 type dynamometer. Measured values were converted into digital values by the aid of DynoWare program. A

screen image showing a measurement taken from the DynoWare is shown in Figure 2.

Table 6. Cutting parameters used in the tests

Cutting Speed (m/min)	Feed Rate (mm/rev)	Side Slip Distance (mm)	Cutting Depth (mm)
350	1270	2	10
400			
450			

Delamination width was calculated by taking the average of the three areas (having the highest value) from the bottom surface where delamination occurred utmost.

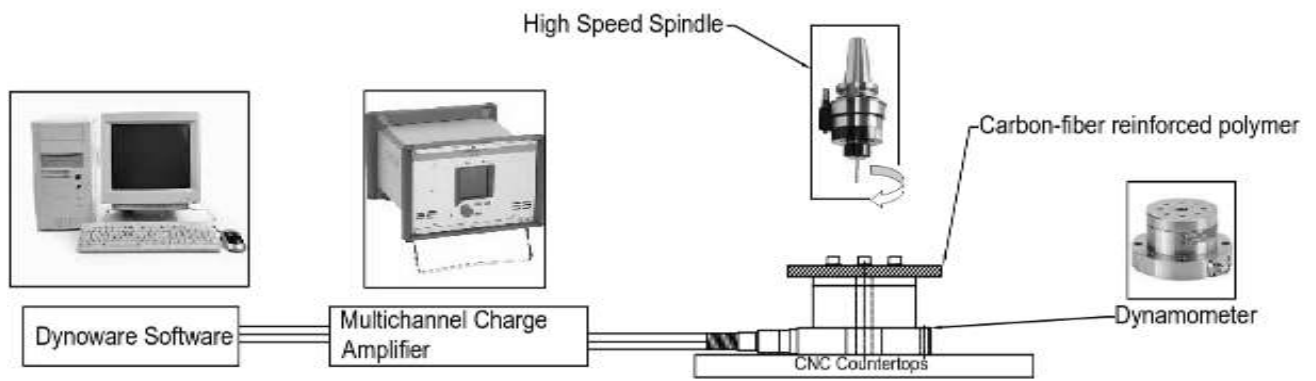


Fig.1. Schematic representation of test apparatus

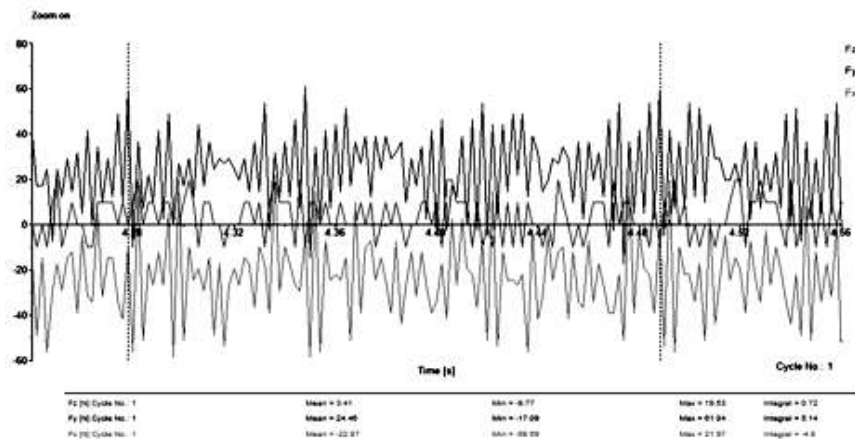


Fig.2. DynoWare program screen image

3. EXPERIMENTAL WORK

3.1. Tables

The obtained values from the tests made with four different cutting tools are given in Table 7. In the graphs prepared by the data from Table 7, the relations between cutting speed and surface roughness, cutting speed and

cutting forces, cutting speed and delamination width are given in Figures 3, 4 and 5 respectively.

When the graphs in Figure 3 were examined, an improvement was observed in the average surface roughness values in all of the tools with the increasing cutting speeds. Improvement of average surface roughness with the increasing cutting speeds is an expected situation and was also verified by the studies in literature [17, 19].

Table 7. Output data obtained depending on the inputs in the experimental work

Experiment Number	Cutting Speed m/min	Feed Rate mm/min	Cutting Tool	Surface Roughness (μm)	Cutting Forces(N)	Delamination Width (mm)
1	350	1270	G1	3,65	58,43	0,29
2	400			3,18	60,28	0,32
3	450			2,67	58,4	0,34
4	350		G2	3,47	60,12	0,33
5	400			3,34	51,92	0,31
6	450			3,15	69,13	0,31
7	350		G3	3,32	72,65	0,29
8	400			2,96	67,73	0,24
9	450			2,65	69,99	0,23
10	350		G4	4,35	92,86	0,18
11	400			3,88	96,28	0,13
12	450			3,62	93,6	0,09

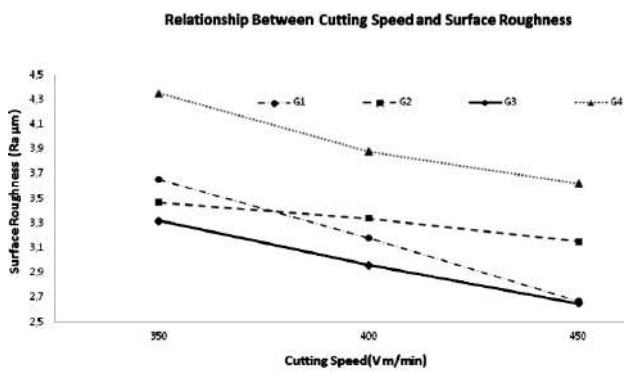


Fig.3. Relation between cutting speed and average surface roughness

As it is seen from Figure 3.1, better surface quality was obtained in the tests made with G3 tool. Bigger helix angle of this tool provided the removal of chip from the work piece and prevented cutting of the chip again affecting the surface quality favourably.

The highest average surface roughness values were obtained in the tests made with G4 no. tool. G4 tool has bidirectional helix angle. For this type of tools it is recommended to bring the helix variation area to the center of thickness which will be machined. This way the tool combines the positive and negative helix design that practices pressure to the top and bottom side of work piece edge. This minimizes any delamination potential (common problem of using greater helix cutting tools) in the machining of CFRP and other engineering composite types. This characteristic which has a positive effect on delamination caused average surface roughness values to increase with the rupture of chip by greater forces during chip removal.

The results obtained from tests made with G1 and G2 tools are similar. The only difference between these two

tools is the different chip crusher geometry. In the machining of CFRP materials chip formation occurs as dust therefore there was no need to chip crusher. It is an expected consequence to obtain average surface roughness values so close to each other with these two cutting tools.

From Figure 4 it is seen that increasing cutting speeds (excluding exceptional cases) decreased the cutting forces. Although the obtained results in the tests made with G1, G2 and G3 numbered tools are close to each other, the cutting forces made with G4 tool came out to be relatively high. It is thought that during cutting, slogging of the tool in both directions depending on the bidirectional helix angle caused the cutting forces to increase.

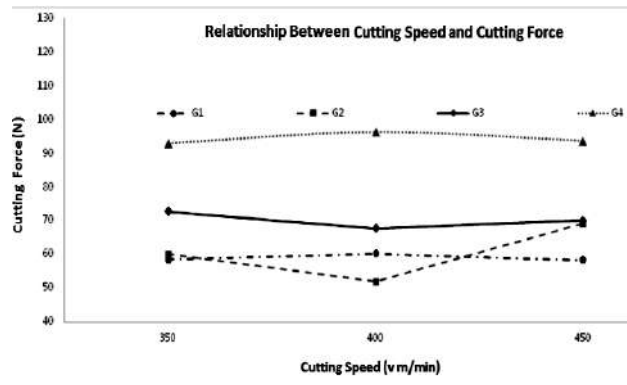


Fig.4. Relationship between cutting speed and cutting forces

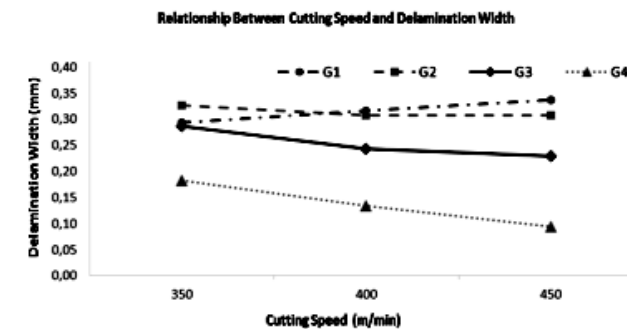


Fig.5. Relationship between cutting speed and delamination width

During chip removal it was observed that this tool was subjected to vibration much more. Exposure of the tool to too much vibration also affected the cutting forces adversely. During chip removal the increasing of forces indicates the slogging of chip when rupturing. During cutting, G4 tool crushed the chip at the same moment by pressing and pulling and the above-mentioned slogging positively affected the delamination width whereas it affected the cutting forces and average surface roughness values adversely. When the graphic in Figure 5 is examined it is seen that the cutting speeds contributed a decrease in delamination width. Unlike the surface quality and cutting forces the best values were obtained with the G4 numbered tool. The G4 tool, during cutting, presses

and pulls the material and forces the chip to crush by jamming rather than separating layers and affect the delamination width positively. Second to this tool another tool is G3 numbered tool having a better delamination width. This can be explained in the way that the tool has a second chip angle and this chip angle decreases the cutting forces and affects the delamination width positively by providing material cutting without slogging. Again no big difference is observed between the G1 and G2 numbered tools as in the average surface roughness values and cutting forces.

4. EVALUATION OF RESULTS

When the data obtained in the tests is evaluated;

- In machining of CFRP materials, increasing of cutting speed affected the average surface roughness values positively. Therefore when CFRP materials are machined with solid cementite carbide tools, the highest cutting speed that the tool permits must be chosen.
- In the machining of CFRP material, cutting forces came out to be low at very parameter. When comparing all of the tools no big differences created among the tools from the point of cutting forces. During machining of CFRP materials due to the necessity of high energy for cutting operation the cutting forces will not be able to be a significant parameter.
- Increasing helix angle affected the average surface roughness values of the machined surfaces positively. Consequently in the applications where the average surface roughness is significant, a higher cutting tool helix angle must be chosen.
- From the point of delamination width, the cutting tool having bidirectional helix seems to be obviously better than the other tools. It will be useful to employ a tool having bidirectional helix to minimize the delamination width. When considering other tools, using the tool having big helix angle will affect the delamination width positively.

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RISK ASSESSMENT EMISSION OF POLLUTION FROM ROAD TRANSPORT IN THE URBAN AREA CITY OF ZRENJANIN IN THE AIM OF ENVIRONMENT PROTECTION – USING THE SOFTWARE ADMS-ROADS

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Abstract: This paper analyzes the impact of traffic as a dominant source of pollution in urban areas using the software tool for spreading pollution. Software ADMS-Roads, used in creation of simulations, is based on the Gaussian, or normal distribution.

Analysis was performed on the transit part of the city of Zrenjanin, which connects the main roads, and which represents the junction between international roads to Romania and the largest centres in Vojvodina and Serbia – Belgrade and Novi Sad. In the immediate vicinity of these roads there are two sensitive community facilities, i.e. a gerontology centre and a primary school, and the aim of the paper is, besides determining the level of pollutants, also to determine the effect of pollution on these two buildings.

Pollutants that are analyzed are the pollutants generated in combustion process in internal combustion engines, which affect air quality: CO, PM10, NO_x, SO₂ and VOC. We analyzed the traffic flows in the streets observed in different time periods and for creating a simulation the worst scenarios were used, i.e. data with the highest traffic flows.

Key words: Environmental protection, urban ecology, ecological risk, road transport, numerical simulation – ADMS-Roads.

1. INTRODUCTION

Next to a variety of industrial plants, power plants, heating plants, etc., traffic is one of the most significant polluters of the environment, whose harmful effects are

seen through pollution of soil, water and earth, and through sound and vibration.

Traffic represents anthropogenic mobile source of air pollution.

According to numerous studies, it is considered that 60% of the world's air pollution comes from fuel combustion in motor vehicles.

Once they get into the atmosphere, gases from combustion of fossil fuels enter into various chemical reactions to form numerous hazardous compounds, among which sulphuric and nitric acid are the most dangerous, which results in occurrence of acid rain, which enter the process of circulation of water in nature, thus destroying the environment. Air pollution creates a variety of pulmonary and skin diseases.

It has particularly damaging effects on children and the elderly. From the above reasons the idea has arisen to choose the intersections near the buildings in which the youngest and the oldest citizens are situated. At the heart of this transit intersection there are two institutions in which reside the mentioned two groups of people; these are a primary school and a gerontology centre.

2. THE IMPACT OF TRANSPORT ON THE ENVIRONMENT

Table 1. shows the most important indicators and impacts of transport on the environment.

Table 1. The most important indicators and impacts of transport on the environment [1]

Impacts	Indicators
Climate changes	Emissions of CO ₂ and CH ₄
Acidity	Emissions of SO ₂ and NO _x
Use of the resources	Power consumption, taking up space
Disappearing biodiversity	Impact on biological world
Air quality	Emission / concentration of pollutants
Water quality	Number of polluted sources, concentration of pollution
Impact on scenery	Important characteristics
Physical limitations	Barriers, the affected
Road traffic accidents	Rates of dead and injured
Noise	Noise levels, range and the affected
Protection of nature and cultural heritage	Impact on such places...

2.1. The impact of pollutants on human health

The dominant pollutants resulting from combustion of fuel in internal combustion engines are PM10, CO, NO_x, SO₂, and one of the most dangerous is VOC. Their harmful effect on human beings is reflected in the following facts.

Suspended particles (PM10) – They cause or reinforce asthma, bronchitis and other lung diseases, and thus reduce the overall body immunity. Although PM10 attack

the entire human population, vulnerable population categories (children, pregnant women and the elderly) are particularly endangered. In addition to health damages, PM10 also reduced visibility during the day because it produces effects which are typical of visibility for haze, often recognized as smog.

Carbon monoxide (CO) – It slows down perception and thinking, can cause death. In conjunction with other harmful substances increases mortality of people with illnesses of respiratory system and bloodstream.

Nitrogen oxides (NOx) – It increases sensitivity to viral infections, especially in asthmatics. It has very serious effects on health with other harmful substances.

Sulphur dioxide (SO₂) – It is very irritating to the respiratory system especially in chronic patients with asthma and bronchitis, also to eyes, it reduces the chlorophyll in plants, has impact on buildings.

Volatile organic compounds (VOC) – Volatile organic compounds from oil are toxic and pollute the air, while some, such as benzene, are extremely toxic, cancerous and cause damage to the DNA structure [1].

2.2. Quantities of pollutants from traffic and legislation in the Republic of Serbia

Table 2. shows the emission of pollutants in the Republic of Serbia, based on data from 2011.

Table 2. Emissions from traffic in the Republic of Serbia for 2011 [2]

Nitrogen oxides (NO _x)	73,211.2 t
Sulphur oxides (SO ₂)	23,105.1 t
Powdery substances (PM _{2,5})	4,379.5 t
Powdery substances (PM ₁₀)	5,170.9 t
Carbon monoxide (CO)	180,708.3 t

Law on Air Protection ('Official Gazette of RS', no. 36/09 and 10/13) comprehensively regulates air quality management.

It specifies measures, organization and supervision of protection and improvement of air quality (air quality, emissions into the air, and other measures for improving the air).

Article 46 of the Law on Air Protection, requires that mobile sources of pollution can be used and marketed if the pollutants in the exhaust gases from these sources do not exceed the emission limit values specified in technical regulations, in accordance with the law.

Emissions from mobile sources of pollution are controlled during regular, emergency and control technical inspection, in accordance with the relevant technical regulations and the law governing traffic safety [3].

3. METHODS

Spreading of a pollutant in the atmosphere and its concentration in the environment depends on meteorological parameters and concentration and type of pollutants. Models of spreading are different in range. For

modelling of regional dispersion (range up to 1,000 km) Euler or Lagrange model is used, while for local dispersion modelling, up to 100 km from the source, Gaussian dispersion model is used. In a Gaussian model, it is assumed that substances which are emitted do not participate in chemical reactions in the atmosphere, that they are carried from their source by the wind in a straight line and mixed with the ambient air both in vertical and in horizontal direction. The said elements characterize a Gaussian, or normal distribution. Software using Gaussian dispersion model or Gaussian distribution model is used in this paper to simulate spreading of pollutants. Commercial name of the software package is ADMS-Roads. ADMS-Roads is a software used for numeric modelling of pollution spread from road traffic and industrial sources, with combining these two sources [4]. ADMS-Roads enables the bringing in of many parameters, including over 7,000 roads and industrial sources through three items and 25 different parameters. The required input variables for ADMS-Roads are wind-speed and wind-direction as well as the Monin-Obukhov length at a specified location in the modelling domain. The traffic-induced turbulence inside the street canyon is considered by ADMS-Roads. The model uses an empirical approach, to define the additional turbulence that is created by the vehicles, which has been determined experimentally [5].

4. DATA SETS FOR STREET MODEL EVALUATION

4.1. Analysis of urban area and traffic data

Figure 1. shows the area of research and numerical simulation of spread of pollutants. The area includes transit part of the city of Zrenjanin that connects main roads, which represents the junction between international roads to Romania and the largest centers in Vojvodina and Serbia – Belgrade and Novi Sad. Size of analyzed area is 1,450 m x 1,300 m, where there are significant institutions for which we want to establish a potential negative impact.



Fig.1. Part of the urban area of the city of Zrenjanin

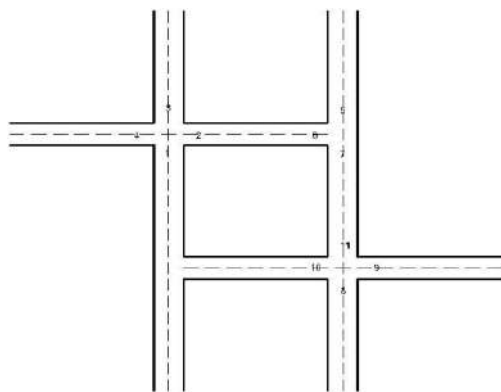


Fig.2. Traffic figure at micro level by accesses

Tables 4. shows data collected for detailed model development and simulation. Software ADMS-Roads generates the results of pollutant spread based on data on geometry of streets and the number of vehicles which pass in an hour period and their average speed in that area. On the basis of data about the size of vehicles, which the software divides into heavy and light vehicles, data on emissions are automatically generated. Data on emissions are generated on the basis of standard reference values in the European Union. The division of vehicles to "Heavy" (H) and "Light" (L) depends on their capacity; and so Heavy vehicles are over 3500 kg, and "Light" vehicles those weighing under 3500 kg (table 3.).

Table 3. Data for streets and vehicles

Street	Width street	Height street	Avg. speed km/h Heavy (H)	Avg. speed km/h Light (L)
1.Miletićeva	35	7	25	30
3.Miletićeva	35	12	30	25
2-6.Principova	11	10	20	15
4.Principova	20	11	20	25
5.Banatska	30	8	30	25
7-11.Banatska	30	8	30	25
8.Birčaninova	12	8	30	25
9.Hercegovačka	12	8	25	20
10.Tomićeva	10	9	25	20

Table 4. Traffic flows structure by moving directions

Street	Vehicle transport 6 ³⁰ –8 ³⁰ h		Vehicle transport 13 ³⁰ –15 ³⁰ h		Vehicle transport 17 ⁰⁰ –18 ⁰⁰ h	
	H	L	H	L	H	L
1.Miletićeva	18	432	35	564	20	258
3.Miletićeva	29	506	54	605	15	216
2-6.Principova	10	317	29	587	6	212
4.Principova	15	239	39	664	5	142
5.Banatska	37	372	94	585	17	254
7-11.Banatska	63	640	129	965	21	373
8.Birčaninova	21	166	30	270	2	106
9.Hercegovačka	8	197	16	295	13	150
10.Tomićeva	2	133	4	97	11	121

Geometry of streets is created using two parameters, width and height. These parameters form a canyon of streets and this parameter affects the way of spreading and directly affects mixing of pollutant with the ambient air, thereby increasing or decreasing the concentration of polluting components in the street. In cases where the canyons are relatively low, pollutants are spread over longer distances, but at concentrations that are significantly lower.

In the table 4. it can be seen that some of the streets are divided into two parts. This was done because different intersections of a street were analysed, which are connecting with the entire whole. Junction of the same street, and its parts had different dynamics of traffic, as branching passenger cars in their parts. This way we obtained a more realistic picture of loading certain parts by road traffic.

4.2. Influence of meteorology on spread of pollutants

Meteorological elements have a great impact on quality of air in the analyzed area, next to concentration and location of pollutants, the most important being air pressure, wind direction and velocity, turbulent flow, humidity, fog, precipitation, air temperature and temperature inversions. The highest concentration of pollutants spreads horizontally in the direction of the wind. It is important to mention that condition in meteorology called calm or "quiet" is a state without the presence of air flow when the polluting components are deposited and concentrated in one place, and therefore the concentration of pollutants exponentially increases. Temperature inversions also greatly affect and can disable normal dispersion of pollutants, which occurs when the earth's surface is rapidly cooled and prevents the air from the lower layers of the atmosphere from moving upwards, because the heat of the air in the upper parts is significantly higher. Low air pressure, absence of wind, high humidity, fog and temperature inversions reduce the spread of pollutants in height and distance, keep them in the ground level and concentrate near sources of pollution. This effect is very pronounced in large urban areas, and most commonly caused mainly from traffic. Eddy flows significantly affect the condition of concentration of pollutants. Turbulence represents irregular movement of air flow where the flow is shifting and uneven. There are two main causes of turbulent flow eddies. Whirling, which is due to air movement next to objects is the result of mechanical turbulence. Particles of hot air that rise from the Earth's surface and large amounts of falling slower molecules of air from the surrounding atmosphere result in thermal turbulence. The most important process in the atmosphere that causes dispersion of air pollutants is called eddy diffusion. Atmospheric whirling causes mixing of contaminated air with relatively clean air during which there is a reduction in concentration of pollutants in a larger volume of air. Pollution in the atmosphere is diluted by air flow in horizontal and vertical direction. The horizontal component of flows is mostly influenced by wind speed and direction, topography, objects on-site and others. The vertical component is affected by air temperature changes depending on the height [6].

4.3. Meteorological conditions and frequency in Banat, in Zrenjanin

Climate in Banat, and consequently Zrenjanin which is located in its central part, belongs to temperate continental type. Continental climate is characterized by very hot and dry summers, while winters are long and harsh. Autumn and spring are moderately warm and short. Summer temperatures can reach up to 44°C, and in winter can sometimes go down to -33°C. Winds in Banat are the strongest in winter. Košava is the most common and the most prevailing wind in this region, mainly from the southeast, and more rarely from the southern direction. Speed of Košava is very variable. It blows at a speed of 5 - 11 m/sec, and in a burst of up to 28 m/sec. Humidity causes condensation of water vapour or the formation of fog and clouds, rainfall, snow. The values of relative humidity are higher in the colder period of the year (76% in autumn, 86% in winter), and lower in the warmer part of the year (spring 72%, summer 68%). Precipitation is unevenly distributed in Banat. Annual precipitation is about 622 mm [7].

4.4. Representation of meteorological data explaining the dominant winds, speed, humidity and the like

Figure 3. shows the wind rose that was used to create simulations of propagation of pollutants resulting from the dominant pollutants, road traffic, in the analyzed area. For the analyzed area meteorological data for the period from 2013. were used. Meteorological data recorded temperature during the year from -6.97 °C to +40.06 °C. Relative humidity ranged from 17.78 % to 90 %, wind speed was up to 6 m/s. Dominant was the southeast wind, which normally is the dominant wind, typical of the area of Zrenjanin and Banat as a whole.

H:\Simulacije ADMS\Zrenjanin- Gerontoloski\Gerontoloski Zrenjanin.out.met

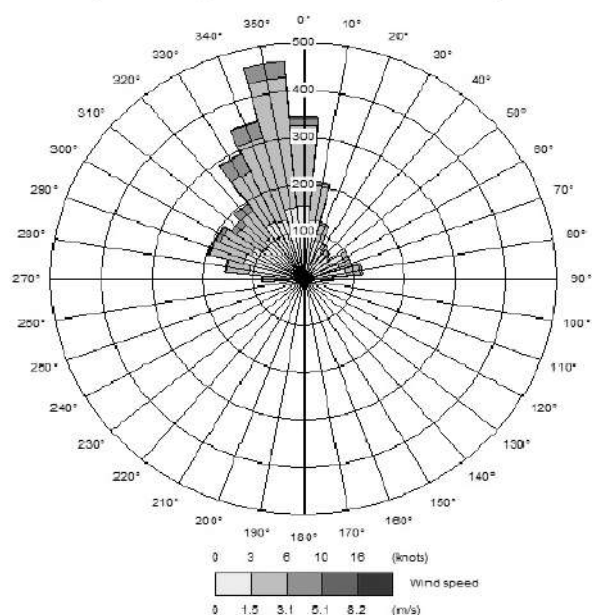


Fig.3. Wind rose city of Zrenjanin

5. RESULTS AND DISCUSSION

To create simulation data, the software enables generating short term and long term emissions. In our case, we generated a long-term simulation using meteorological data throughout 2014.

Figure 4 shows the result of simulation in propagation of CO in the analyzed area. Concentrations of this pollutant dispersion were in the range 1-140 µg/m³. The highest concentrations were identified at the intersection of Principova and Miletićeva, while a large load can be seen on street 4 – Principova. The geometry of this street resulted in a concentration of between 40-70 µg/m³. In the picture it can be seen that gerontology centre is increasingly threatened by these pollutants, due to the fact that it is located at the intersection of the two streets.

h:\Simulacije ADMS\Zrenjanin- Gerontoloski\Gerontoloski LT\Gerontoloski Zrenjanin - LT.glt
Conc ug/m3 CO <All sources> - 1hr

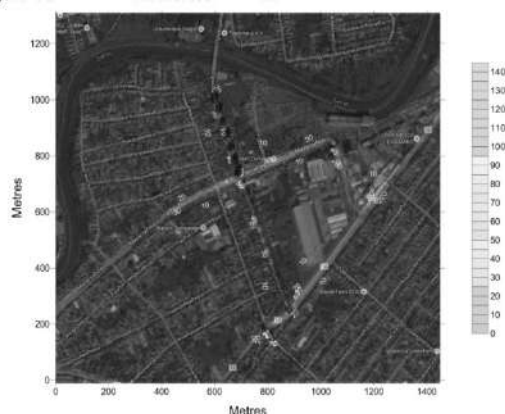


Fig.4 Value of CO

Concentration of pollutants in gerontology centre is at a concentration of 30 µg/m³. Building of primary school "Žarko Zrenjanin" which is located at the corner of Đurđa Smederevca and Tomićeva Street is one of the most important, which is taken into consideration during preparation of this paper. Concentration of CO in the area of primary school is in the range of 10-20 µg/m³, which is much less than concentrations to which gerontology centre is exposed. This is a result of a much lower frequency of traffic than in the streets Principova and Miletićeva which represent transit streets.

Concentrations of nitrogen oxides shown in Figure 5, in the analyzed area, are in the range 10-100 µg/m³. The highest concentrations were recorded in intersections of the study area. NO_x concentration at intersections are 70-100 µg/m³, while in the very streets concentrations are 20-40 µg/m³. It is evident that this pollutant spreads in the broader area and covers a broader area than the analyzed streets, including private homes and a complete industrial complex "DIJAMANT". Concentrations in the broader area to which population is constantly exposed is 15-20 µg/m³.



Fig.5. Value of NOx

Suspended particles represent a significant component in polluting the air and directly affect air quality. The most important division is in their diameter i.e. size, and in literature they are known as PM10 and PM2.5. We have in this case analyzed the suspended particles of 10 μm or PM10.

Figure 6. shows the long-term simulation of dispersion of suspended particles resulting from road traffic. Concentrations of this pollutant are in the range 0.2-5.6 $\mu\text{g}/\text{m}^3$. The highest concentrations can be expected in analyzed sections of streets or at intersections. In the analyzed streets concentrations at ground level of this pollutant will be in the range of 1.2-1.4 $\mu\text{g}/\text{m}^3$ and in figure 5, it can be clearly seen that the gerontology centre is completely surrounded by this pollutant and to the concentration of 0.6-1.2 $\mu\text{g}/\text{m}^3$. There is a variance in relatively constant concentration in Principova Street moving towards the bridge, and where the concentration is caused by the amount of traffic and street dimensions, or, the size of the canyon in the range 1.2-2.4 $\mu\text{g}/\text{m}^3$. In this street eddy concentration of 4.4 $\mu\text{g}/\text{m}^3$ is recorded.



Fig.6. Value of PM10

Figure 7 shows the long-term simulation of volatile organic compounds or VOC. Maximum concentrations of this compound are foreseen of 17 $\mu\text{g}/\text{m}^3$. On the streets, at ground level, the amount of VOC concentrations was foreseen of about 8 $\mu\text{g}/\text{m}^3$ and

specifically in Principova Street. In Miletičeva and Banatska Street pollutant concentrations are in the range of 2-5 $\mu\text{g}/\text{m}^3$. Building of primary school is not exposed to the pollutants to a greater extent, but in immediate proximity the concentration of the pollutants is between 1-2 $\mu\text{g}/\text{m}^3$. Gerontology centre which is located at the intersection of Miletičeva and Principova Street is exposed to the pollutant to a much greater extent, in the range of 6-8 $\mu\text{g}/\text{m}^3$. Concentration of this pollutant in Principova Street at ground level is 8 $\mu\text{g}/\text{m}^3$. In Banatska and Herzegovina Street concentration of pollutants is 4-5 $\mu\text{g}/\text{m}^3$, and in the immediate proximity and residential units the concentration is 2 $\mu\text{g}/\text{m}^3$.



Fig.7. Value of VOC

6. CONCLUSION

In order to protect the air from pollution of road traffic, it is necessary to adopt a series of measures to prevent and/or mitigate the effects of pollutants coming from traffic, among which is the use of "clean" fuels, installation of filters for purifying exhaust gases, greening of affected areas, etc.

Ensuring adequate data on air quality, as well as ensuring their availability are also very important measures in order to reduce environmental pollution. For these reasons this work has been done which aims to present data on air pollution in the urban environment, where, in addition to a large number of residents, there are also "vulnerable" buildings, such as the gerontology centre and the primary school. By gradual and staged development, including consideration of many of the proposed measures and providing information on pollution, we are going up towards the goal, which is general protection of the environment and all the participants in it.

Simulated data show that on the basis of applied meteorological data there is a direct impact of road traffic on air quality in the vicinity of gerontology centre, while the impact on primary school building Žarko Zrenjanin is significantly lower but not negligible. Sensitivity of users of the two institutions, which specifically refers to children and the elderly, directs us to the need to reduce this impact to the least possible measure.

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